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MACHINERY'S REFERENCE SERIES

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No. 31

SCREW THREAD TOOLS AND GAGES

CONTENTS

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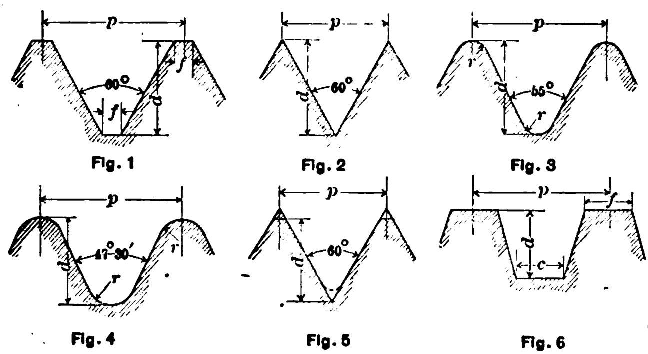
CHAPTER I.

SCREW THREAD SYSTEMS.

Before entering upon the subject of screw thread tools and gages, we will shortly review the more common screw thread systems, giving the most important information regarding each, and will present such tables and formulas as give the basic information which the ordinary mechanic requires. While a great many more systems than are here reviewed have been proposed from time to time, only those which have been officially recognized by mechanical men, or which have gained prestige on account of universal use and adoption, here or abroad, will be treated.

The United States Standard Thread.

The United States standard thread, usually denoted U. S. S., has a cross section as shown in Fig. 1. The sides of the thread form an angle of 60 degrees with one another. The top and bottom of the thread



Figs. 1 to 6. Standard Screw Threads.

are flattened, the width of the flat in both cases being equal to one-eighth of the pitch of the thread. In this connection it may be appropriate to define the expression pitch as well as lead, as these two expressions are very often confused, and the word pitch, in particular, often, though erroneously, used in place of "number of threads per inch." The pitch of a thread is the distance from center to center of two adjacent threads. It is equal to the reciprocal value of the number of threads per inch, or, if expressed in a formula:

$$\text{Pitch} = \frac{1}{\text{Number of threads per inch.}}$$

The lead of a screw thread is the distance the screw will travel forward if turned around one complete revolution. It is evident that for a single threaded screw the pitch and the lead are equal. In a

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double threaded screw, the lead equals two times the pitch, in a triple threaded, three times, etc. The definitions given for pitch and lead should be strictly adhered to, as great confusion is often caused by improper interpretation of the meaning of these terms. Confusion is also caused by indefinite designation of multiple thread screws. The most common way to state the lead and the class of thread is perhaps to say *¼ inch lead, double*, which means a screw with a double thread, which, when cut, has the lathe geared for four threads per inch, but each thread is cut only to a depth corresponding to eight threads per inch. The same condition is also expressed by: *4 threads per inch, double*. These two ways of expressing the number of multiple threads are both correct, but the expression which ought to be used in order to avoid misunderstanding under any circumstances would be: *¼ lead, ⅛ pitch, double thread*.

Returning to the form of the U. S. S. thread, we find that if the thread is flattened one-eighth of the pitch at top and bottom, the depth

UNITED STATES STANDARD THREAD.

Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch
1/16	64	13/16	10	1 3/4	5	3 3/8	3 1/4
3/32	50	7/8	9	1 13/16	5	3 1/2	3 1/4
1/8	40	15/16	9	1 7/8	5	3 5/8	3 1/4
5/32	36	1	8	1 15/16	5	3 3/4	3
3/16	32	1 1/16	7	2	4 1/2	3 7/8	3
7/32	28	1 1/8	7	2 1/8	4 1/2	4	3
1/4	20	1 3/16	7	2 1/4	4 1/2	4 1/4	2 7/8
5/16	18	1 1/4	7	2 3/8	4	4 1/2	2 3/4
3/8	16	1 5/16	6	2 1/2	4	4 3/4	2 5/8
7/16	14	1 3/8	6	2 5/8	4	5	2 1/2
1/2	13	1 7/16	6	2 3/4	4	5 1/4	2 1/2
9/16	12	1 1/2	6	2 7/8	3 1/2	5 1/2	2 3/8
5/8	11	1 9/16	5 1/2	3	3 1/2	5 3/4	2 3/8
11/16	11	1 5/8	5 1/2	3 1/8	3 1/2	6	2 1/4
3/4	10	1 11/16	5 1/2	3 1/4	3 1/2		

of the thread is equal to three-quarters of the depth of a corresponding thread, sharp both at top and bottom. If p equals the pitch of the thread; d, the depth; and f the width of the flat, the following formulas express the relation between these quantities:

$$p = \frac{1}{\text{Number of threads per inch.}}$$

$$d = \tfrac{3}{4} \times p \times \cos 30^\circ = 0.64952\, p.$$

$$f = \frac{p}{8}$$

Formula for the Number of Threads in the United States Standard Thread System.

In order to fix definitely the number of threads per inch corresponding to any given diameter in the U. S. S. system, Mr. William Sellers, its originator, proposed the following approximate formula:

$$p = 0.24\sqrt{D + 0.625} - 0.175,$$

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in which formula p equals the pitch of the thread for any bolt or screw of the diameter D.

This formula is applicable to all screws ¼ inch and larger in diameter. For diameters below ¼ inch the formula is modified so as to read:

$$p = 0.23\sqrt{D + 0.625} - 0.175.$$

This modification, which has met with general acceptance, changing the coefficient 0.24 to 0.23, was proposed by Mr. George M. Bond in 1882. The purpose of the change was to make the formula applicable to screw threads for bolts smaller than one-quarter inch in diameter. Mr. Bond's formula tends to increase the number of threads more rapidly as the diameter decreases, a distinct advantage in the case of small screws.

It will be proper to remark in this connection that screws 11/16, 13/16 and 15/16 inch in diameter, which according to the formula given ought to have 10, 9 and 8 threads per inch, respectively, are in usual manufacturing practice made with 11, 10 and 9 threads per inch.

STANDARD SHARP V-THREAD

Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch
1/16	72	13/16	10	1 3/4	5	3 3/8	3 1/4
3/32	56	7/8	9	1 13/16	5	3 1/2	3 1/4
1/8	40	15/16	9	1 7/8	4 1/2	3 5/8	3 1/4
5/32	32	1	8	1 15/16	4 1/2	3 3/4	3
3/16	24	1 1/16	8	2	4 1/2	3 7/8	3
7/32	24	1 1/8	7	2 1/8	4 1/2	4	3
1/4	20	1 3/16	7	2 1/4	4 1/2	4 1/4	2 7/8
5/16	18	1 1/4	7	2 3/8	4 1/2	4 1/2	2 3/4
3/8	16	1 5/16	7	2 1/2	4	4 3/4	2 5/8
7/16	14	1 3/8	6	2 5/8	4	5	2 1/2
1/2	12	1 7/16	6	2 3/4	4	5 1/4	2 1/2
9/16	12	1 1/2	6	2 7/8	4	5 1/2	2 3/8
5/8	11	1 9/16	6	3	3 1/2	5 3/4	2 3/8
11/16	11	1 5/8	5	3 1/8	3 1/2	6	2 1/4
3/4	10	1 11/16	5	3 1/4	3 1/2		

The Sharp V-Thread.

The sharp V-thread, Fig. 2, is very similar to the U. S. S. thread, except that theoretically it is not provided with any flat, either at the top, nor at the bottom of the thread. In common practice, however, it has proved necessary to provide this thread with a slight flat on top of the thread. There are, unfortunately, some difficulties caused by providing a flat on the top of sharp V-threads, the principal one being that no definite standard for this flat has been settled upon. Some manufacturers have used the same flat as is used for the Briggs standard pipe thread, which, although theoretically rounded at top and bottom, is, in this country at least, made with a small flat on the top of the thread. The width of this flat is selected so as to give exactly the same angle diameter as is obtained when rounding the thread in accordance with Briggs' original proposition. This flat is equal to about one-twenty-fifth of the pitch.

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If p equals the pitch of the thread; d, the depth; and f, the width of the flat on the top of the thread, the following formulas express the relation between the various quantities of the sharp V-thread:

$$p = \frac{1}{\text{Number of threads per inch.}}$$

$$d = p \times \cos 30° = 0.86603\,p.$$

$$f = \frac{p}{25}$$

Attention must be called to the fact that the formula for the width of the flat is selected simply to give an arbitrary value, which is *not* recognized as any *standard element* of the sharp V-thread. In figuring the depth of the thread, this flat is not considered, and the depth is arrived at as if the thread were exactly sharp.

Comparison Between the U. S. S. and the Sharp V-Thread.

The two standards referred to hitherto are the two forms of threads most commonly used in the United States. The objections to the sharp

WHITWORTH STANDARD THREAD.

Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch
$\frac{1}{16}$	60	$\frac{13}{16}$	10	$1\frac{3}{4}$	5	$3\frac{3}{8}$	$3\frac{1}{4}$
$\frac{3}{32}$	48	$\frac{7}{8}$	9	$1\frac{13}{16}$	5	$3\frac{1}{2}$	$3\frac{1}{4}$
$\frac{1}{8}$	40	$\frac{15}{16}$	9	$1\frac{7}{8}$	$4\frac{1}{2}$	$3\frac{5}{8}$	$3\frac{1}{4}$
$\frac{5}{32}$	32	1	8	$1\frac{15}{16}$	$4\frac{1}{2}$	$3\frac{3}{4}$	3
$\frac{3}{16}$	24	$1\frac{1}{16}$	8	2	$4\frac{1}{2}$	$3\frac{7}{8}$	3
$\frac{7}{32}$	24	$1\frac{1}{8}$	7	$2\frac{1}{8}$	$4\frac{1}{2}$	4	3
$\frac{1}{4}$	20	$1\frac{3}{16}$	7	$2\frac{1}{4}$	4	$4\frac{1}{4}$	$2\frac{7}{8}$
$\frac{5}{16}$	18	$1\frac{1}{4}$	7	$2\frac{3}{8}$	4	$4\frac{1}{2}$	$2\frac{7}{8}$
$\frac{3}{8}$	16	$1\frac{5}{16}$	7	$2\frac{1}{2}$	4	$4\frac{3}{4}$	$2\frac{3}{4}$
$\frac{7}{16}$	14	$1\frac{3}{8}$	6	$2\frac{5}{8}$	4	5	$2\frac{3}{4}$
$\frac{1}{2}$	12	$1\frac{7}{16}$	6	$2\frac{3}{4}$	$3\frac{1}{2}$	$5\frac{1}{4}$	$2\frac{5}{8}$
$\frac{9}{16}$	12	$1\frac{1}{2}$	6	$2\frac{7}{8}$	$3\frac{1}{2}$	$5\frac{1}{2}$	$2\frac{5}{8}$
$\frac{5}{8}$	11	$1\frac{9}{16}$	6	3	$3\frac{1}{2}$	$5\frac{3}{4}$	$2\frac{1}{2}$
$\frac{11}{16}$	11	$1\frac{5}{8}$	5	$3\frac{1}{8}$	$3\frac{1}{2}$	6	$2\frac{1}{2}$
$\frac{3}{4}$	10	$1\frac{11}{16}$	5	$3\frac{1}{4}$	$3\frac{1}{4}$		

V-thread, as compared with the U. S. S. thread, are that the comparatively sharp points of the teeth are very frail; that the groove at the bottom of the thread, being sharp, facilitates fracture under strain; that the depth of the thread, being considerably greater than that of the U. S. S. thread, subtracts from the effective area at the root of the thread of the screw, thus impairing the tensile strength of the threaded bolt, and finally, that in case of taps, the sharp V-thread has less endurance and shorter life, and is capable of smaller duty, owing to the frail and easily worn away points of the thread. In spite of all this, however, the sharp V-thread will long continue to be in general use, primarily because it has so thoroughly established itself in the mechanical industries. This form of thread has also another very strong claim, because of being admirably adapted to the making of steam-tight joints. It answers this purpose best of all common forms

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of thread, and all patch bolt taps, boiler taps and staybolt taps are, as a rule, provided with sharp V-threads.

The Whitworth Standard Thread.

The Whitworth standard thread, Fig. 3, is used chiefly in Great Britain, but to a certain extent also in the United States. Its use here, however, has greatly diminished since the U. S. S. thread commenced to gain general approval. The Whitworth standard is the older one of the two, and was the first recognized screw thread system. In the Whitworth standard thread the sides of the thread form an angle of 55 degrees with one another. The top and the bottom of the thread are rounded to a radius determined by the depth of the thread, which is two-thirds of a thread with the same angle which were sharp at top and bottom. The radius at the top is the same as the radius at

BRITISH STANDARD FINE SCREW THREAD.

Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch
$\frac{1}{4}$	25	$1\frac{1}{8}$	9	2	7	$3\frac{3}{4}$	$4\frac{1}{2}$
$\frac{5}{16}$	22	$1\frac{3}{16}$	9	$2\frac{1}{8}$	7	$3\frac{7}{8}$	$4\frac{1}{2}$
$\frac{3}{8}$	20	$1\frac{1}{4}$	9	$2\frac{1}{4}$	6	4	$4\frac{1}{2}$
$\frac{7}{16}$	18	$1\frac{5}{16}$	9	$2\frac{3}{8}$	6	$4\frac{1}{4}$	4
$\frac{1}{2}$	16	$1\frac{3}{8}$	8	$2\frac{1}{2}$	6	$4\frac{1}{2}$	4
$\frac{9}{16}$	16	$1\frac{7}{16}$	8	$2\frac{5}{8}$	6	$4\frac{3}{4}$	4
$\frac{5}{8}$	14	$1\frac{1}{2}$	8	$2\frac{3}{4}$	6	5	4
$\frac{11}{16}$	14	$1\frac{9}{16}$	8	$2\frac{7}{8}$	6	$5\frac{1}{4}$	$3\frac{1}{2}$
$\frac{3}{4}$	12	$1\frac{5}{8}$	8	3	5	$5\frac{1}{2}$	$3\frac{1}{2}$
$\frac{13}{16}$	12	$1\frac{11}{16}$	8	$3\frac{1}{8}$	5	$5\frac{3}{4}$	$3\frac{1}{2}$
$\frac{7}{8}$	11	$1\frac{3}{4}$	7	$3\frac{1}{4}$	5	6	$3\frac{1}{2}$
$\frac{15}{16}$	11	$1\frac{13}{16}$	7	$3\frac{3}{8}$	5		
1	10	$1\frac{7}{8}$	7	$3\frac{1}{2}$	$4\frac{1}{2}$		
$1\frac{1}{16}$	10	$1\frac{15}{16}$	7	$3\frac{5}{8}$	$4\frac{1}{2}$		

the bottom. If p and d equal the pitch and the depth of the thread, respectively, and r the radius at the top and bottom, then

$$d = \frac{2}{3} \times \frac{p}{2} \times \cot 27^\circ\ 30' = 0.64033\,p.$$

$$r = 0.1373\,p.$$

The advantages of the Whitworth thread are that screws with this form of thread have all the strength possessed by screws with U. S. S. threads, and at the same time have no sharp corners from which fractures may start. Screws and nuts with this form of thread will work well together after continued heavy service when other forms of thread would fail. Whitworth threads are used in the United States chiefly on special screws, such, for instance, as screws for gasoline needle valves, where a liquid-tight and yet working fit is desired. It is also often used for locomotive boiler staybolts. The objections to the Whitworth form of thread are that the angle of 55 degrees cannot be measured or simply laid out with ordinary tools, and that the rounded corners at the top and bottom cannot be produced with any degree of accuracy without great difficulty. The Whitworth standard

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screw system is denoted B. S. W. (British Standard Whitworth screw thread) in Great Britain.

British Standard Fine Screw Thread.

The British standard fine screw thread is a system of threads recently adopted in Great Britain. The form of the thread is the same as that for the Whitworth standard, but there is a greater number of threads per inch corresponding to a certain diameter than in the Whitworth system. The fine screw thread system is denoted B. S. F., and applies to screws ¼ inch in diameter and larger.

The pitches for the system of fine screw threads are based, approximately, on the formula:

$P = \frac{\sqrt[3]{d^2}}{10}$, for sizes up to and including one inch; and on the formula:

$P = \frac{\sqrt[8]{d^5}}{10}$, for sizes larger than one inch in diameter.

BRITISH ASSOCIATION STANDARD THREAD.

British Association Number	Diameter		Pitch		British Association Number	Diameter		Pitch	
	Millimeters	Inches	Millimeters	Inches		Millimeters	Inches	Millimeters	Inches
0	6.0	0.2362	1.0	0.0394	13	1.2	0.0472	0.25	0.0098
1	5.3	0.2087	0.90	0.0354	14	1.0	0.0394	0.23	0.0091
2	4.7	0.1850	0.81	0.0319	15	0.90	0.0354	0.21	0.0083
3	4.1	0.1614	0.73	0.0287	16	0.79	0.0311	0.19	0.0075
4	3.6	0.1417	0.66	0.0260	17	0.70	0.0276	0.17	0.0067
5	3.2	0.1260	0.59	0.0232	18	0.62	0.0244	0.15	0.0059
6	2.8	0.1102	0.53	0.0209	19	0.54	0.0213	0.14	0.0055
7	2.5	0.0984	0.48	0.0189	20	0.48	0.0189	0.12	0.0047
8	2.2	0.0866	0.43	0.0169	21	0.42	0.0165	0.11	0.0043
9	1.9	0.0748	0.39	0.0154	22	0.37	0.0146	0.098	0.0039
10	1.7	0.0669	0.35	0.0138	23	0.33	0.0130	0.089	0.0035
11	1.5	0.0591	0.31	0.0122	24	0.29	0.0114	0.080	0.0031
12	1.3	0.0511	0.28	0.0110	25	0.25	0.0098	0.072	0.0028

In the above formulas

P = pitch, or lead of single-threaded screw, and

d = diameter of screw.

This standard is not intended to make the regular Whitworth standard thread superfluous, but is simply supposed to offer a possibility of a standard fine screw thread for such purposes where the regular Whitworth standard would be too coarse.

British Association Standard Thread.

The British Association standard thread is the standard system for screws of small diameter in Great Britain. It is, however, hardly used at all in the United States, except in the manufacture of tools for the English market. The characteristics of the thread form are similar to those of the Whitworth thread, but the angle between the

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sides of the thread is only 47 degrees 30 minutes, and the radius at the top and bottom of the thread (see Fig. 4) is proportionally larger, depending upon that the depth of the thread is smaller in relation to the pitch than in the Whitworth standard thread. If p, d and r signify the pitch, the depth, and the radius at the top and bottom of the thread, respectively, then

$$d = 0.6\,p. \qquad\qquad r = \frac{2\,p}{11}$$

The various sizes of screws in this system are numbered, and a certain number of threads per inch always corresponds to a certain given diameter. The system is founded on metric measurements. It was first originated in Switzerland as a standard for screws used in watch and clock making. This system is therefore also at times referred to as the Swiss small screw thread system.

Briggs Standard Pipe Thread.

The Briggs standard pipe thread is made with an angle of 60 degrees. It is slightly rounded off, both at the top and at the bottom, so that

BRIGGS STANDARD PIPE THREAD.

Nominal Size of Tube	Actual Outside Size of Tube	No. of Threads per Inch	Nominal Size of Tube	Actual Outside Size of Tube	No. of Threads per Inch	Nominal Size of Tube	Actual Outside Size of Tube	No. of Threads per Inch
1/8	0.405	27	1½	1.900	11½	5	5.563	8
1/4	0.540	18	2	2.375	11½	6	6.625	8
3/8	0.675	18	2½	2.875	8	7	7.625	8
1/2	0.840	14	3	3.500	8	8	8.625	8
3/4	1.050	14	3½	4.000	8	9	9.688	8
1	1.315	11½	4	4.500	8	10	10.750	8
1¼	1.660	11½	4½	5.000	8			

the depth of the thread, instead of being equal to the depth of the sharp V-thread (0.866 × pitch), is only four-fifths of the pitch, or equal to $\frac{0.8}{n}$, if n be the number of threads per inch. The difficulty of producing a thread with rounded top and bottom has, however, caused the manufacturers in this country to modify the original standard. Instead of rounding the bottom of the thread, it is made sharp as shown in Fig. 5. The top is slightly flattened instead of rounded, the flat being carried down just far enough to tangent the top circle of the correct thread form. This thread, as indicated by the name, is used for pipe joints and for many purposes in locomotive boiler work. Taps for producing Briggs standard pipe thread are provided with a taper of ¾ inch per foot on the diameter.

Whitworth Standard Thread for Gas and Water Piping.

The form of the Whitworth standard thread for gas and water piping is simply the regular Whitworth thread form, and the only difference from the regular Whitworth standard is the number of

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threads per inch. Most American manufacturers of taps, when making what is called English pipe taps, use the Whitworth form of thread

WHITWORTH STANDARD THREAD FOR GAS AND WATER PIPING, COMMONLY KNOWN AS THE STANDARD GAS THREAD.

Nominal Size of Tube	Actual Outside Size of Tube	No. of Threads per Inch	Nominal Size of Tube	Actual Outside Size of Tube	No. of Threads per Inch	Nominal Size of Tube	Actual Outside Size of Tube	No. of Threads per Inch
1/8	0.385	28	1 3/8	1.745	11	2 5/8	3.124	11
1/4	0.520	19	1 1/2	1.882	11	2 3/4	3.247	11
3/8	0.665	19	1 5/8	2.021	11	2 7/8	3.367	11
1/2	0.822	14	1 3/4	2.160	11	3	3.485	11
5/8	0.902	14	1 7/8	2.245	11	3 1/4	3.698	11
3/4	1.034	14	2	2.347	11	3 1/2	3.912	11
7/8	1.189	14	2 1/8	2.467	11	3 3/4	4.125	11
1	1.302	11	2 1/4	2.587	11	4	4.339	11
1 1/8	1.492	11	2 3/8	2.794	11			
1 1/4	1.650	11	2 1/2	3.001	11			

and the number of threads according to the Whitworth pipe thread system, but make the dimensions for the taps the same as for the Briggs standard in all respects. The taper is also made the same as for the Briggs standard, or ¾ inch per foot.

Square Thread.

The square form of thread is usually made about twice as coarse in pitch as the V or U. S. S. threads, and partly for this reason and partly because of the perpendicular sides of the thread, it is a troublesome thread to cut with taps and dies. There is no standard for the number of threads corresponding to a certain diameter. The depth of the thread is equal to the width of space between the teeth, this space being equal to one-half of the pitch. While theoretically the space between the teeth is equal to the thickness of the tooth, each being one-half of the pitch, it is evident that the thickness of the tooth must be enough smaller than the space to admit at least an easy sliding fit.

The Acme Thread.

The Acme thread, shown in Fig. 6, has of late become widely used, having in most instances taken the place of the square thread on ac-

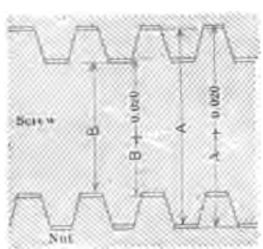

Fig. 7. Acme Standard Screw and Nut.

count of its better wearing qualities, and the comparative ease with which this thread can be produced. Of all standard thread systems the

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Acme thread is the only one which has a standard provision for clearance at the top and bottom of the thread. The screw is made of standard diameter, but the nut is made over-size. The relationship between screw and nut is illustrated in Fig. 7. If the diameter of the screw is A over the top of the thread, and B at the foot of the thread, the corresponding diameters of the nut are $A + 0.020$ and $B + 0.020$ inch. The sides of the thread form an angle of 29 degrees with one another. Considering the screw only, if p is the pitch; d, the depth of the thread; f, the width of the flat at the top of the thread; and c, the width of the flat at the bottom of the thread, then:

$$d = \frac{p}{2} + 0.010 \text{ inch},$$

$$f = 0.3707\, p,$$

$$c = 0.3707\, p - 0.0052 \text{ inch}.$$

The Acme thread has many good points, not the least of which is its strength, and the ease with which it may be cut compared with the square thread. This thread is recommended as a substitute for, and to be used in preference to, the square form of thread.

FRENCH SYSTEM STANDARD THREAD.

Diameter		Pitch		Diameter		Pitch	
Millimeters	Inches	Millimeters	Inches	Millimeters	Inches	Millimeters	Inches
3	0.1181	0.5	0.0197	24	0.9449	3.0	0.1181
4	0.1575	0.75	0.0295	26	1.0236	3.0	0.1181
5	0.1969	0.75	0.0295	28	1.1024	3.0	0.1181
6	0.2362	1.0	0.0394	30	1.1811	3.5	0.1378
7	0.2756	1.0	0.0394	32	1.2598	3.5	0.1378
8	0.3150	1.0	0.0394	34	1.3386	3.5	0.1378
9	0.3543	1.0	0.0394	36	1.4173	4.0	0.1575
10	0.3937	1.5	0.0590	38	1.4961	4.0	0.1575
12	0.4724	1.5	0.0590	40	1.5748	4.0	0.1575
14	0.5512	2.0	0.0787	42	1.6535	4.5	0.1772
16	0.6299	2.0	0.0787	44	1.7323	4.5	0.1772
18	0.7087	2.5	0.0984	46	1.8110	4.5	0.1772
20	0.7874	2.5	0.0984	48	1.8898	5.0	0.1969
22	0.8661	2.5	0.0984	50	1.9685	5.0	0.1969

French and International Standard Threads.

The French and international standard threads are of the same form as the U. S. standard, and the formulas given for the latter form of thread apply to the former. The pitches, however, are stated in the metric measure, and are somewhat finer for corresponding diameters than the U. S. S. thread. This is a distinct advantage, especially in the smaller sizes. The standard thread of the international system is denoted S. I. and was adopted by the International Congress for the unifying of screw threads held in Zürich, 1898. This system conforms with slight variations with the system earlier adopted in France, the French standard thread, denoted S. F. In order to provide for clearance at the bottom of the thread in the nut, the congress referred to above specified that the clearance at the bottom of

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the thread shall not exceed 1/16 of the height of the original triangle. The shape of the bottom of the thread resulting from such clearance is left to the manufacturers. However, the congress recommends rounded profile for said bottom. By this provision choice is given to

INTERNATIONAL SYSTEM STANDARD THREAD.

Diameter		Pitch		Diameter		Pitch	
Milli-meters	Inches	Milli-meters	Inches	Milli-meters	Inches	Milli-meters	Inches
6	0.2362	1.0	0.0394	33	1.2992	3.5	0.1378
7	0.2756	1.0	0.0394	36	1.4173	4.0	0.1575
8	0.3150	1.25	0.0492	39	1.5354	4.0	0.1575
9	0.3543	1.25	0.0492	42	1.6535	4.5	0.1772
10	0.3937	1.5	0.0590	45	1.7716	4.5	0.1772
11	0.4331	1.5	0.0590	48	1.8898	5.0	0.1969
12	0.4724	1.75	0.0689	52	2.0472	5.0	0.1969
14	0.5512	2.0	0.0787	56	2.2047	5.5	0.2165
16	0.6299	2.0	0.0787	60	2.3622	5.5	0.2165
18	0.7087	2.5	0.0984	64	2.5197	6.0	0.2362
20	0.7874	2.5	0.0984	68	2.6772	6.0	0.2362
22	0.8661	2.5	0.0984	72	2.8346	6.5	0.2559
24	0.9449	3.0	0.1181	76	2.9921	6.5	0.2559
27	1.0630	3.0	0.1181	80	3.1497	7.0	0.2756
30	1.1811	3.5	0.1378				

the manufacturers to make the bottoms of their threads flat or rounded as desired, and yet have them conform to an interchangeable standard.

Instrument-maker's System.

The standard screw system of The Royal Microscopical Society of London, England, is employed for microscope objectives and the nose pieces of the microscope into which these objectives screw. The form of thread is the Whitworth form, the diameter of the male gage is 0.7626 inch. The number of threads per inch is 36.

Machine Screw Threads.

The American Society of Mechanical Engineers has adopted a standard system of machine screw threads, which undoubtedly will, before long, become the general standard for small diameter screws. Consid-

MACHINE SCREW THREADS, AMERICAN SOCIETY OF MECHANICAL ENGINEERS STANDARD.

Number	Diameter	Threads per Inch	Number	Diameter	Threads per Inch	Number	Diameter	Threads per Inch
0	0.060	80	7	0.151	36	18	0.294	20
1	0.073	72	8	0.164	36	20	0.320	20
2	0.086	64	9	0.177	32	22	0.346	18
3	0.099	56	10	0.190	30	24	0.372	16
4	0.112	48	12	0.216	28	26	0.398	16
5	0.125	44	14	0.242	24	28	0.424	14
6	0.138	40	16	0.268	22	30	0.450	14

erable care was taken in determining this standard, in order to settle the questions involved so as to meet the requirements of both the makers of taps and screws, and the users. The basic form of the thread is that of the U. S. standard thread.

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Lag Screw Threads.

There is no recognized standard for the sizes and corresponding number of threads for lag screws. The following table, however, gives the number of threads according to common practice. While lag screws are largely made according to these systems, there are, however, a number of different systems in use.

LAG SCREW THREAD SYSTEMS IN COMMON USE.

Diameter	Alternate Systems		Diameter	Alternate Systems		Diameter	Alternate Systems	
	Threads per Inch	Threads per Inch		Threads per Inch	Threads per Inch		Threads per Inch	Threads per Inch
1/4	10	10	1/2	6	6	3/4	4½	5
5/16	9½	9	9/16	5	6	7/8	4½	4
3/8	7	8	5/8	5	5	1	3	4
7/16	7	7	11/16	4½	5			

Gas Fixture Threads.

Thin brass tubing is threaded with 27 threads per inch, irrespective of diameter. The so-called ornament brass sizes have 32 threads per inch. The standard diameters of the thread are 0.196 inch (large ornament brass size) and 0.148 inch (small ornament brass size).

Fine Screw Thread Systems.

We have previously referred to the British fine screw thread system recently adopted. There is a demand for the adoption in this country of a standard system with a U. S. S. form of thread but with a finer pitch than called for by this standard. The Association of Licensed Automobile Manufacturers has adopted such a standard, but it is, of course, not universally recognized. The objection to the adoption of a standard by a single body of manufacturers is obvious. Even if the standard is one which would recommend itself to general use, it would be better if the opinion and the needs of machine builders in general were considered. On the other hand it may be said in defense of the adopted system that automobile construction is so specialized a manufacture that here doubtless may arise requirements which would not present themselves elsewhere.

FINE SCREW THREAD SYSTEM ADOPTED BY THE ASSOCIATION OF LICENSED AUTOMOBILE MANUFACTURERS.

Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch
1/4	28	7/16	20	5/8	18	7/8	14
5/16	24	1/2	20	11/16	16	1	14
3/8	24	9/16	18	3/4	16		

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CHAPTER II.

MAKING THREAD TOOLS.

The chief requisites for cutting a correct thread are a correct threading tool, a correct setting of the tool, and a lathe with a reasonably accurate leadscrew. In making a U. S. thread tool a correct 60-degree angle gage is necessary. To produce such a gage, first plane up a piece of steel in the shape of an equilateral triangle as shown at *a* in Fig. 8. After hardening this triangle, grind and lap the edges until the three corner angles prove to be exactly alike when measured with a protractor. This is now the master gage. To produce the female

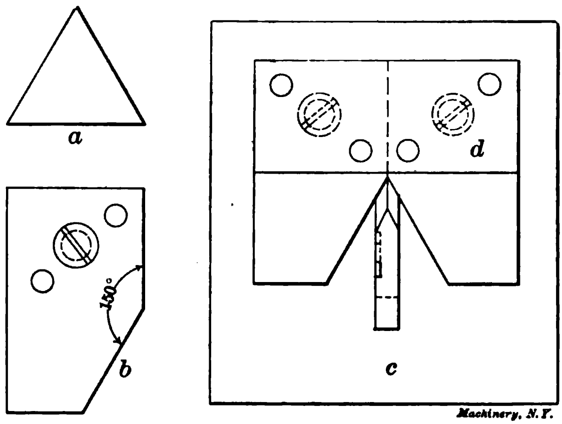

Fig. 8. Gages for Making Threading Tools.

gage, make two pieces, one right- and one left-hand, like that shown at *b* in Fig. 8; harden them and lap the edges that form the 150-degree angle so that they are straight, and square with both sides. When this is done the two pieces should be screwed and doweled to a backing plate *d* as shown, using the master triangle to locate them, thus producing a practically perfect female gage.

In making up the tool, some form of cutter to be used in a holder should be chosen in preference to a forged tool on account of convenience in handling and measuring and the ease with which it may be re-ground without destroying the shape. The tool should be made so that the top will stand level when in the holder, and the clearance should be about 15 degrees, which is ample for a single thread unless the pitch is very coarse. With that amount of clearance the included angle between the sides of the tool in a plane perpendicular to the front edge is approximately 61 degrees 44 minutes. The tool should be planed to that angle as nearly as is possible by measuring with a protractor, then, to test its accuracy, it should be placed

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top down on a flat piece of glass *c* and tried with the 60-degree gage as shown in Fig. 8. After lapping the tool until it shuts out the light when tried in this manner, the angle may be considered as nearly correct as it is possible to obtain with ordinary means. To adapt the V-thread tool thus made to cut the United States standard form of thread, it is only necessary to grind off the sharp edge an amount equal to one-eighth of the depth of a V-thread of the required pitch, or for 20 threads per inch $\frac{0.866}{20} \times \frac{1}{8} = 0.0054$ inch. To test the accuracy of this grinding, a piece of steel should be turned up to the correct outside diameter and a short shoulder turned down at the end to the correct diameter at the bottom of the thread; then the piece is threaded and the tool fed in until the flat of the tool just tangents the shoulder. Then cut a nick in the edge of a piece of sheet steel with the threading

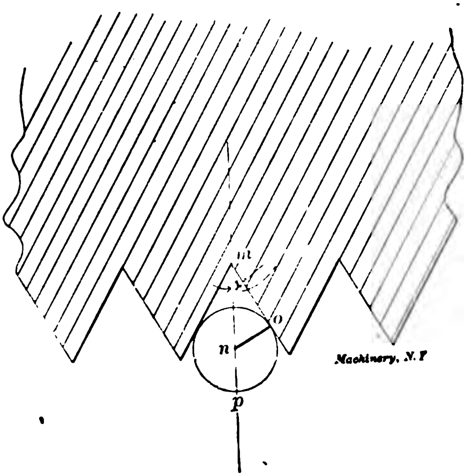

Fig. 9. Measuring the Angle Diameter of a Thread.

tool. This sheet steel piece is now applied like a gage to the threaded cylindrical piece. If the nick in the sheet steel fits the thread so that it shuts out the light the flat of the tool is correct.

In preparing a plug gage for threading it should be made the same as the cylindrical test piece above with a part turned down to the root diameter of the thread except that for V-thread it is customary to leave the shoulder 0.005 inch large on account of the impossibility of producing a perfectly sharp point on the tool. The thread tool should be set level, with the top at the same height as the center line of the spindle of the lathe, otherwise the correct angle will not be reproduced. After a master plug has once been produced, it is not necessary to turn down a portion to the root diameter of the thread, as the work may be compared with the master plug by means of a micrometer fitted with either ball or V points for measuring in the angle of the thread.

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It occasionally happens that a tap is to be threaded, or other external threading is to be done, of an odd size or pitch. Where it is desired to originate a master plug in such cases, three wires may be used for measuring the angle of the thread, placing one wire in the angle of the thread on one side of the piece and the other two on the opposite side, one on each side of the corresponding thread, measuring over the whole with a micrometer. The formula for the micrometer reading is obtained as follows: In Fig. 9, assume that m is the bottom of a V-thread, the circle showing one wire in place. Then angle $a = 30$ degrees; sin 30 deg. $= 0.5$; $\frac{no}{0.5} = mn$ or $2no = mn$. As no and np are radii of the same circle, it follows that $mp = 3no = 1\frac{1}{2} \times$ diameter of wire. Multiplying by 2 to add a length mp for the opposite side gives $2mp = 3 \times$ diameter of wire. Hence for V-thread,

$$\text{Diameter of screw} - \frac{1.732}{\text{No. thds. per in.}} + (3 \times \text{diameter of wire used})$$
$= $ micrometer reading.

For U. S. form we have to take into account the flat at the bottom of the thread, so instead of using the U. S. constant 1.299 we add to it $\frac{1}{8}$ of 1.732, or 0.2165, giving as a constant 1.5155, making the formula:

$$\text{Diameter of screw} - \frac{1.5155}{\text{No. thds. per in.}} + (3 \times \text{diameter of wire used})$$
$= $ micrometer reading.

The subject of measuring threads with the wire system is more completely treated in Chapter V.

Thread Tools for Threads with Rounded Top and Bottom.

While the development of a correct United States or V-thread tool is a thing requiring a great deal of skill and patience, it is easy compared to the task of producing a tool for the round top and bottom thread, of which the Whitworth and British Association standards are the leading examples. In testing for accuracy, threads of this type are not only measured by gages and micrometers, but the curves must match the angle so evenly that when the male gage is tried in the female from either end, no difference can be detected. The difficulty attending this will be the better appreciated when it is known that some of the leading tap and die manufacturers of this country and Europe have failed in producing threads that would pass the British government's inspection.

It may be laid down as a cardinal principle that the best results are obtained by developing the form first with a flat top and bottom as in the U. S. thread, rounding the corners afterward. The first step of all is to produce a correct angle gage; assuming that we are to work out the Whitworth thread, this would be a gage measuring 55 degrees. Make and harden a steel triangle *A*, Fig. 10, with the angle x made as near 55 degrees as is possible by using a bevel protractor; the other two angles are to be equal. Then make an angle iron *B*, making

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sure that *ab* and *cd* are parallel, and that *be* is square with *ab*. Assuming that *C* and *D* are accurate 2-inch and ½-inch plugs, we put in the pins *EE* in such a position that a line drawn through the centers of *C* and *D*, at right angles to their axes, will make an angle of 27½ degrees with *ab*. This can be done by figuring the distance *fg* as follows: In the triangle *lhk*, $hk = 1 - 0.25 = 0.75$ inch.

$$lk = \frac{0.75}{\tan 27\frac{1}{2}\text{ deg.}} = \frac{0.75}{0.5206} = 1.4406 \text{ inch.}$$

$$1.4406 + \tfrac{1}{2}\text{ diameter of } C - \tfrac{1}{2}\text{ diameter of } D = 1.4406 + 1 - 0.25 = 2.1906 \text{ inch} = fg.$$

Set the pin *F* near enough to *D* to keep the corner of the triangle from striking the angle iron *B*. Mount the triangle *A* as shown, and set up the fixture on a surface grinder table using a toe strap in the small hole in *A* to hold it in position, and grind first one edge, and then the other. This gives us the male angle gage. A female gage can be made to this by the method described previously for U. S. thread gages.

The tools to be used in making thread tool (see Fig. 11) include:

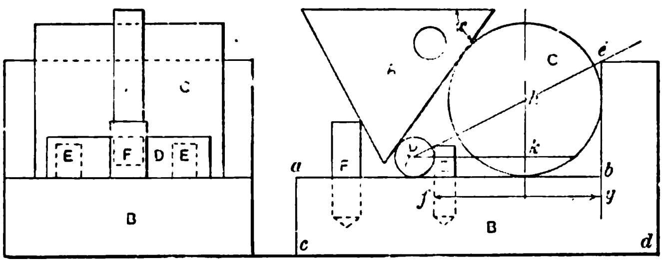

Fig. 10. Method Employed for Obtaining 55 Degree Angular Gage.

an angular tool with a flat point, the width of the point to be such that it reaches to the center of the round in the bottom of the thread, the angle of the tool matching the gage previously made; a female radius tool for forming the point; and a male radius tool for the side radii. For convenience in measuring and getting the exact form required, these tools should be made with the top square with the face at the cutting edge, *i. e.*, without clearance. The sides and back of all should be ground as well as the top. The tool *a* can be ground by means of an angular block made in the same manner as the male angle gage and should be finished by lapping. The tool *b* can be made in two pieces, one a hardened, ground, and lapped wire, and the other a soft piece made up in such shape that the wire can be soldered or otherwise firmly fastened to it in the correct position. The tool *c* should be made up first as at *c′* and hardened. Then lap the hole carefully to size and grind the outside. After measuring the distance from the hole to the back of the tool, the front can be ground off to *ef* and the bevels ground until the depth of the round part is right.

We now require a shaper with an apron made up to hold the toolholder at an angle of 15 degrees, as shown in Fig. 12. The apron

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should fit the clapper-box perfectly. If it does not, it is better to fasten it solid, and let the tools drag back through the cut, sharpening the tools over again before finishing; otherwise, one runs the risk of side shake. With this angular apron we can use the tools made without clearance to produce a tool with correct clearance for the lathe. Two thread tool blanks, one, *a*, of tool steel, and one, *b*, of machine steel, should be set up on the table adapter as shown in the cut with spacing parallels between to avoid interfering with one while planing the other. The blanks should be planed off to exactly the same height, and all measurements for height should be figured from the line *cd*, allowance being made for the difference caused by the 15-degree clearance. Then, after measuring the tools previously made carefully, to determine where the exact center is, we can start forming the blanks, setting the tools sidewise successively by positive measurement from the rib of the adapter. The angular tool comes first and with it we plane down the sides of the tool *a* and the center of *b* so that the point of the tool just reaches the center of the radius. Then using the female radius tool we round the point of *a* and the two points of *b*, coming down until the circle of the tool is just tangent to

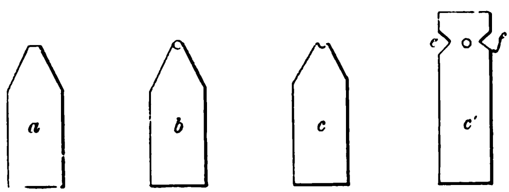

Fig. 11. Tools used for making Whitworth Thread Tools.

the top of the blanks. The male tool will round out the two lower corners of *a* and the center of *b*, being fed down to exact depth.

We now have the thread tool *a*, which can be hardened, and the machine steel blank used as a lap to correct errors in it, reversing the lap occasionally, and using oilstone powder or other fine abrasive as the cutting medium. Great care must be used in putting on the abrasive, as in all lapping operations of this kind points and corners are apt to lap faster than wide surfaces. This operation does not really correct the tool, but equalizes the errors due to imperfect matching of the different cuts, and it can be done so effectively that whatever errors of that kind are left cannot be detected.

To test the tool, turn up a blank plug with a teat equal to the diameter at the bottom of the thread. When this is threaded, the point of the tool should touch the teat just when the outer corners touch the top of the thread. In the angle, the thread should measure by wires, as explained in connection with the U. S. S. thread tool, according to the formula:

$$\text{Diameter of screw} - \frac{1.6008}{\text{No. threads per in.}} + (3.1659 \times \text{diameter of wire used}) = \text{micrometer reading.}$$

For the final test of the fit of the curves with the angle, a tap must

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be threaded with the tool, and a female gage tapped with the tap. The plug just made must screw into this with an equal amount of friction from either end, and show a full contact on the thread. If this last test is not successful it shows that the lapping is not good enough and must be done over. If the plug does not measure right, it is necessary to go back to the planing and plane up another tool, making such

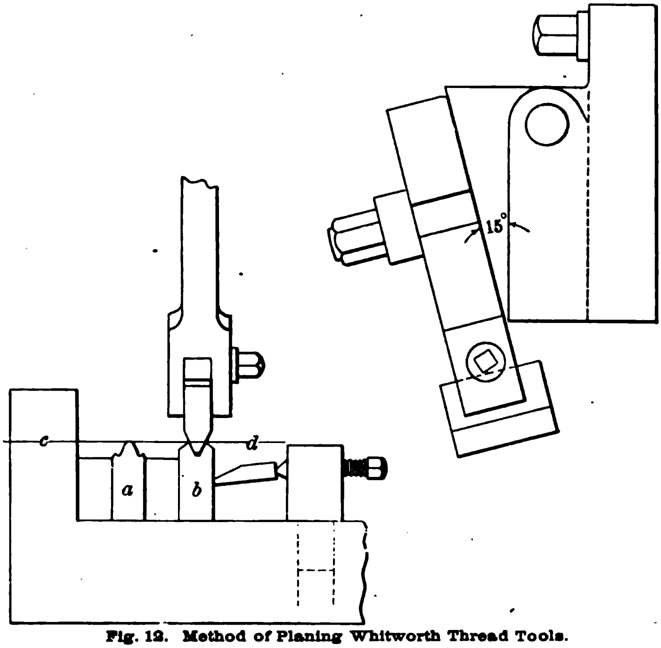

Fig. 12. Method of Planing Whitworth Thread Tools.

allowances as one judges will correct the error. It is sometimes necessary to do this several times before a perfect tool is produced. In the use of the tool in the lathe, great care is necessary to see that it is set at the center of the spindle, and so that the two side curves will scrape the top of the thread at the same time. With the exception of making the angle gage and tool grinding block, this whole procedure has to be carried out for every pitch required.

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CHAPTER III.

TOOLS FOR ACCURATE THREAD CUTTING.

Accurate thread cutting seems to be one of the trade secrets which is not easily mastered or often found described in books. The author has seen and done considerable of this work, and although the methods may not be wholly new, they will, no doubt, be of interest to many. The method of making thread chasers, described in the following, was employed by the Pratt & Whitney Co. several years ago, and the table on page 21 and the hob and fixture for grinding the chasers are original with the author. That old saying "Patience is a virtue," is well recognized when doing this class of work. What can be more exasperating than to have a thread tool tear when on the finishing

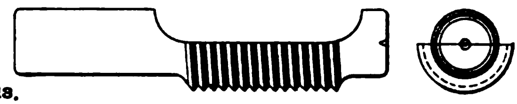


Fig. 13.

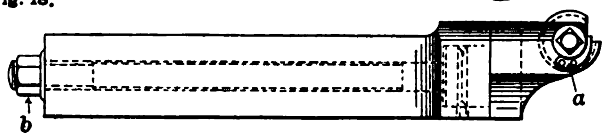


Fig. 14.

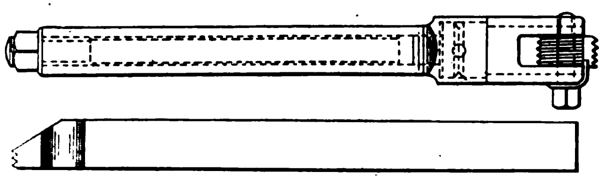
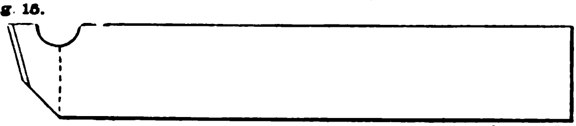


Fig. 15.



Industrial Press, N.Y.

Figs. 13 to 15. Hobs and Chasers for making Accurate Thread Tools.

cut, or, having made a nice plug and ring gage, to have the ring contract and the plug expand in length when they are hardened? There are a great many such difficulties in the path of the thread gage and tap maker. That one little item, of the thread tool tearing the threads is greatly, if not wholly, overcome by adopting a chaser in place of a single thread tool, as the chaser has three to five threads which tend to keep it from tearing into the work. The chaser is also far superior for cutting U. S. standard threads, as the flat top and bottom are sure to be perfect and can always be held to a standard with the aid of a master hob.

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To make a chaser accurately seems at first difficult, but is quite simple when properly understood. The first thing is to make the hob, which is shown in Fig. 13. This requires great care as upon it depend all the tools of that certain pitch. The hobs are all made one-inch

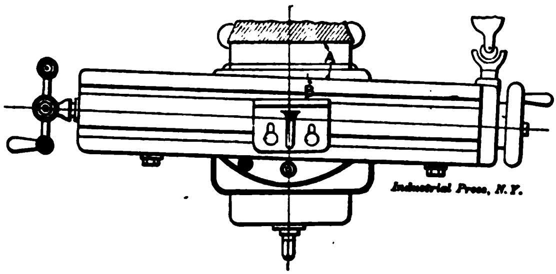

Fig. 16.

Threads per inch	Angle B in Degrees and Minutes.																Angle A	
	Deg.	Min.	Deg.	Min.	Deg.	Min.	Deg.	Min.	Deg	Min.	Deg	Min.	Deg.	Min	Deg.	Min.	Deg	Min.
8	..	....	..	....	3	02	2	17	1	49	1	31	1	18	1	08	2	17
10	..	....	3	38	2	26	1	49	1	27	1	13	1	02	..	55	1	49
12	..	....	3	02	2	01	1	31	1	18	1	00	..	52	..	46	1	31
14	..	....	2	36	1	44	1	18	1	01	..	52	..	44	..	39	1	18
16	..	....	2	17	1	31	1	08	..	55	..	46	..	39	..	34	1	08
18	..	....	2	02	1	21	1	00	..	49	..	40	..	35	..	30	1	00
20	3	38	1	49	1	18	..	55	..	44	..	37	..	31	..	27	..	55
22	3	19	1	39	1	06	..	50	..	40	..	33	..	28	..	25	..	50
24	3	02	1	31	1	00	..	46	..	37	..	30	..	26	..	23	..	46
26	2	48	1	24	..	56	..	42	..	34	..	28	..	24	..	21	..	42
28	2	36	1	18	..	52	..	39	..	31	..	26	..	22	..	19	..	39
30	2	27	1	13	..	49	..	37	..	29	..	24	..	21	..	18	..	37
32	2	17	1	08	..	46	..	34	..	27	..	23	..	19	..	17	..	34
36	2	02	1	00	..	40	..	30	..	24	..	20	..	17	..	15	..	30
40	1	49	..	55	..	37	..	27	..	22	..	18	..	16	..	14	..	27
48	1	31	..	46	..	30	..	23	..	18	..	15	..	13	..	12	..	23
56	1	18	..	39	..	26	..	19	..	16	..	13	..	11	..	10	..	19
64	1	08	..	34	..	23	..	17	..	14	..	12	..	10	..	08	..	17
80	..	55	..	27	..	18	..	14	..	11	..	09	..	08	..	07	..	14
100	..	44	..	22	..	15	..	11	..	09	..	07	..	06	..	05	..	11
	¼"		½"		¾"		1"		1¼"		1½"		1¾"		2"			
	Diameter on which chaser is to be used.																	

NOTE.—All hobs to be 1 inch in diameter right-hand thread. Clearance on chasers, 15 degrees.

in diameter, this size having been adopted so that the angles in the table above could be determined and tabulated. To accurately cut the hob, the tool shown in Fig. 14 is utilized. This consists of a small circular thread chaser held in the body of the tool, the forward part of which is made separate from the shank so that it can be swiveled to suit the angle of the thread on the hob. A small piece of steel, *a*,

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serves as a gage for the cutting face of the circular chaser, so that it can be sharpened and re-set in the holder without disturbing the body of the tool. The nut *b*, on the end of the holder, serves to hold the forward part of the tool securely. After the hob is threaded, it is milled out (as shown in Fig. 13) to its center line and then hardened. The object in milling it in this manner is that it can be easily sharpened by grinding across the face, and this face is also utilized when setting the hob to its proper angle in the milling machine.

The sharpening of the hob is accomplished with the special fixture shown in Fig. 17, which is simple in design, and is made for use on a surface grinder where it is located so that its centers are at right angles with the emery wheel. The most essential point in grinding a hob of this description is to always grind the cutting face radially, in other words, the lower edge of the emery wheel must be in line

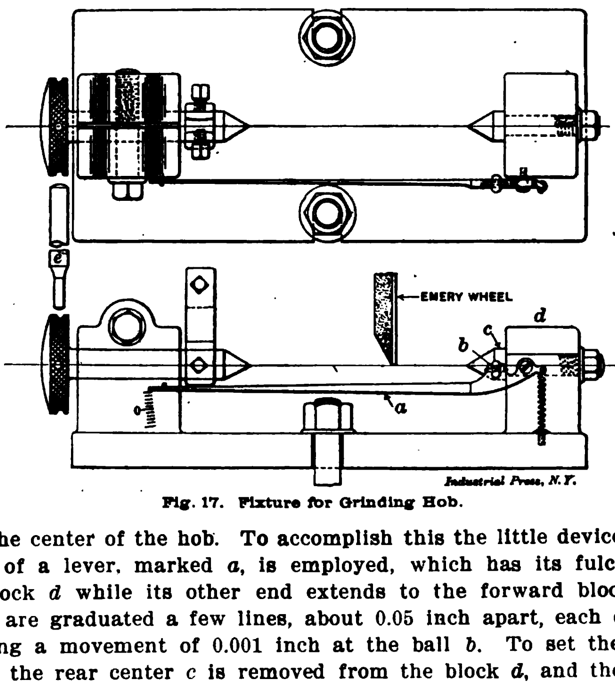

Fig. 17. Fixture for Grinding Hob.

with the center of the hob. To accomplish this the little device in the shape of a lever, marked *a*, is employed, which has its fulcrum on the block *d* while its other end extends to the forward block upon which are graduated a few lines, about 0.05 inch apart, each division equaling a movement of 0.001 inch at the ball *b*. To set the emery wheel, the rear center *c* is removed from the block *d*, and the emery wheel, at rest, is brought down onto the ball *b*. The table of the grinder is run to and fro by hand so that the wheel will pass over the ball; when it forces down the lever so that it registers at zero, this denotes that the lower edge of the wheel is in alignment with the centers of the fixture. The center *c* is then put back in position and the hob, held by a dog, is placed between the centers and sharpened. At no time during grinding is the perpendicular adjustment of the wheel altered.

The hob being completed, the next step is to make, by use of the

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hob, the chaser shown in Fig. 15. This is made of tool steel, hand forged, and planed on all sides. It has a cutting clearance of 15 degrees and is placed against an angle iron which, in turn, is held on a milling machine table. The hob is held between the centers of the machine spindle and the overhanging arm, as shown in Fig. 18, and when the cutting edge of the hob is accurately located, the spindle is locked in position by means of a wooden wedge which is tapped in between the cone and the frame of the machine. The cutting face of the hob and the body of the chaser stand in a straight line and at an angle of 15 degrees with the table of the machine, as shown in Fig. 19. The most essential point in setting up the machine for this job is to

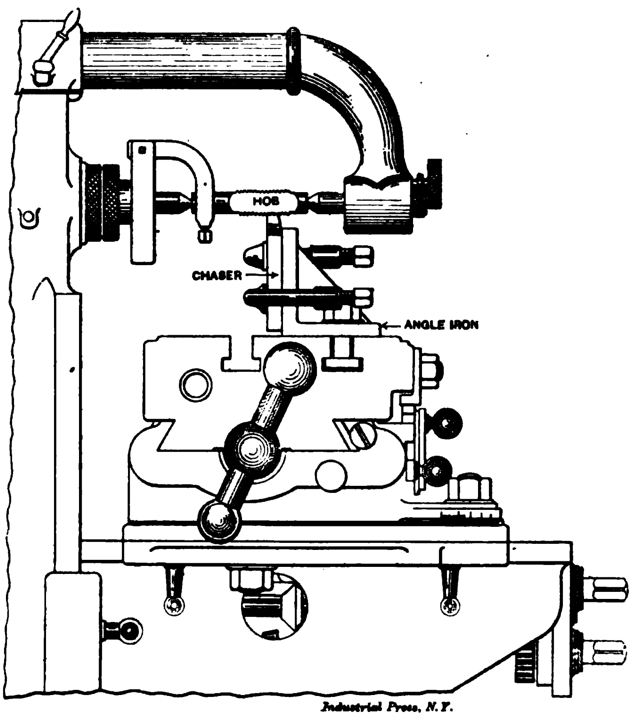

Fig. 18. Milling Machine Arranged for Hobbing Chasers.

get the angle iron located on the table of the machine at the proper angle for the threads to be shaped on the chaser, as a chaser made for use on a ½-inch tap will not work properly on a 2-inch tap of the same pitch, because the angle of the thread is greater on the former than on the latter. The milling machine table must also be swiveled around to the proper angle of the thread on the hob, as the longitudinal movement of the table must correspond to the thread angle. For this purpose we use the table on page 21. The plan view, Fig. 16, shown at the head of this table will serve to illustrate the use of same. This is a plan of the milling machine table, showing it swiveled around,

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and also the angle iron set in the proper position. As will be seen, the table sets at an angle *A*, which is given in degrees and minutes in the right-hand column of the table, while the angle iron is placed on the table of the machine, making with the edge of the table the angle *B*. This is the proper angle for the threads on the chaser. Should it be desired to make a left-hand thread chaser, the angle iron would be placed at the same angle called for by the table, but in the opposite direction.

As an example, we will suppose that it is desired to make a chaser that is to be used in making taps ½ inch in diameter having 26 threads

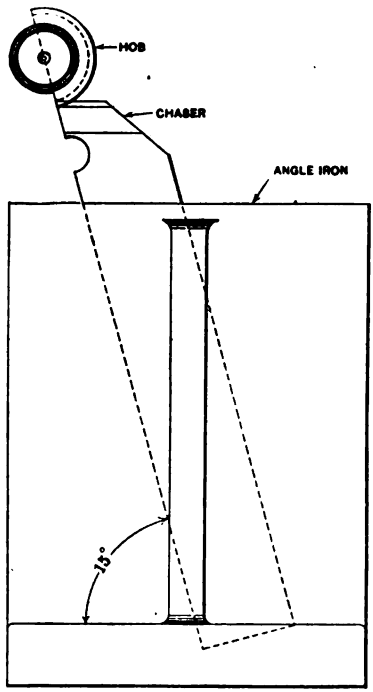

Fig. 19. Hob and Chaser in Position for Hobbing.

per inch. We first look in the "threads per inch" column until we come to 26, then by following along the line we come to the last column, which gives us the angle at which the milling machine table is to be set, or the angle *A*, which equals 0 degree and 42 minutes. On the lower edge of the table are the diameters on which the chasers are to be used, and as in this case this is ½ inch, we follow up that column to the 26 threads per inch line, where we obtain the angle at which the angle iron is to be located on the table of the machine, or angle *B*, which in this case is 1 degree and 24 minutes. The machine being properly set, it is a small matter to shape the thread by moving the table to and fro, gradually feeding it upward until a perfect

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thread is obtained on the chaser. It is advisable to keep the hob well lubricated when cutting to insure a smooth thread on the chaser. A very good lubricant for this purpose is a mixture of one-half turpentine with one-half good lard oil. This will also be found an excellent lubricant for general thread cutting in the lathe.

Another style of chaser which has proved itself very useful may be

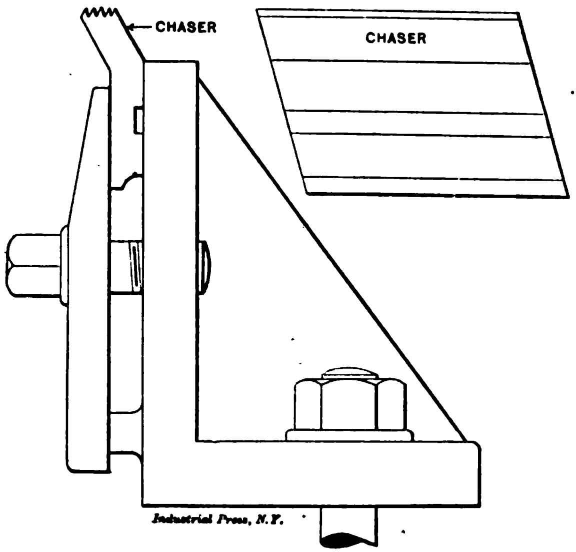

Fig. 20. Fixture for Hobbing Chaser for Pratt & Whitney Thread Tool Holder.

worthy of notice. This is what is known as the Pratt & Whitney chaser, which is shown in Fig. 20. It is made separate from the body of the tool-holder, in the angle iron shown in the same figure. For inside thread cutting, the two tools shown in Fig. 21 can be used very handily. The one marked *a* is for large inside diameters and is composed of a tube through which runs a rod, threaded on each of its ends. Upon the front end is screwed a circular chaser which is held firmly against the tube by a nut on the other end of the rod. This

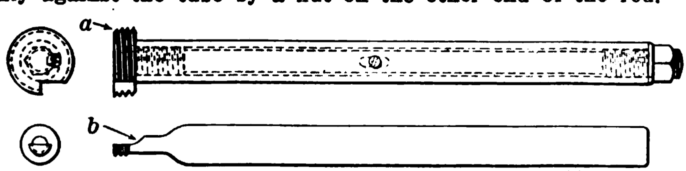

Fig. 21. Chasers for Internal Threads.

makes a very handy device when chasers have to be changed, for by loosening the nut on the end, the chaser can be easily removed with the fingers. The solid chaser, *b*, shown in same figure, is for use in holes of small diameter, one-half of the threaded part being milled off. When sharpening this chaser, care must be taken to always grind the face radially in order to insure accurate results.

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CHAPTER IV.

MAKING THREAD GAGES.

It appears to be the general idea that screw plug gages must be made of tool steel, but it has been found very practical to make them of cold rolled stock, which is very soft and easy to cut, but which, when hardened, gives a surface which is fully as hard as tool steel. This hard surface extends deep enough into the thread gage to permit grinding 0.005 inch deep, enough hard surface still remaining to prevent rapid wear when in use. Another reason for using this soft steel is also that it is not likely to change its shape after having been finished, the same as does even the best tool steel, if it has not been properly seasoned after hardening.

For setting a thread tool for cutting a correct thread, a cylindrical thread gage is made, as shown in Fig. 26. This thread gage has the advantage over the ordinary thread gage on the market, that it can be placed between the centers of the lathe, and consequently one does not depend upon any secondary surface against which to set the thread gage. This is the case with the ordinary thread gages, which have to be lined up either against the side of the face-plate of the lathe, or against the side of the work, and in this way small errors are almost always introduced. The thread gage in Fig. 26 is made of machine steel, hardened and ground all over. The main body, *A*, is provided with three grooves, having an inclusive angle of 29, 55, and 60 degrees, respectively, to correspond with the Acme, Whitworth and United States Standard threads, respectively. When the gage is hardened, the two sides of the grooves are ground with the same setting of the slide-rest, the piece *A* being reversed on the lathe centers while grinding. This insures that both sides of the angle in the gage make the same angle with the axis of the gage.

In one end of the body *A* a hole is drilled; this is ground until the bottom of the hole comes exactly in line with the axis or center line of the body *A*. A hardened and lapped plug *B* is inserted into this hole and held with a set-screw, having a brass shoe at the end. The purpose of this plug *B* is to afford a means for setting the thread tool in the lathe at the correct height, or, which is the same, exactly in line with the axis of the spindle. This is done by merely loosening the clamp which holds the thread tool in its holder; then with the thumb of the left hand on the plug *B*, and the forefinger on the thread tool, it is brought instantly in position, so that the upper face of the thread tool touches the lower side of the plug *B*, as shown in the end view of Fig. 26. When in this position, the clamp of the thread tool-holder is again tightened, and the tool is now placed in the correct position

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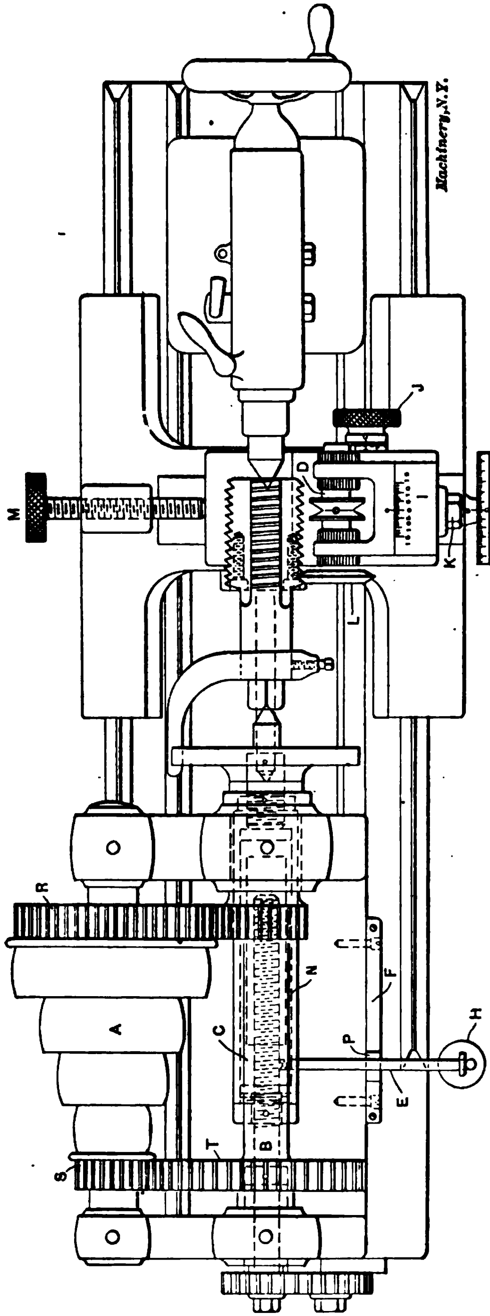

Fig. 22. Lathe for Grinding Taps in the Angle of the Thread.

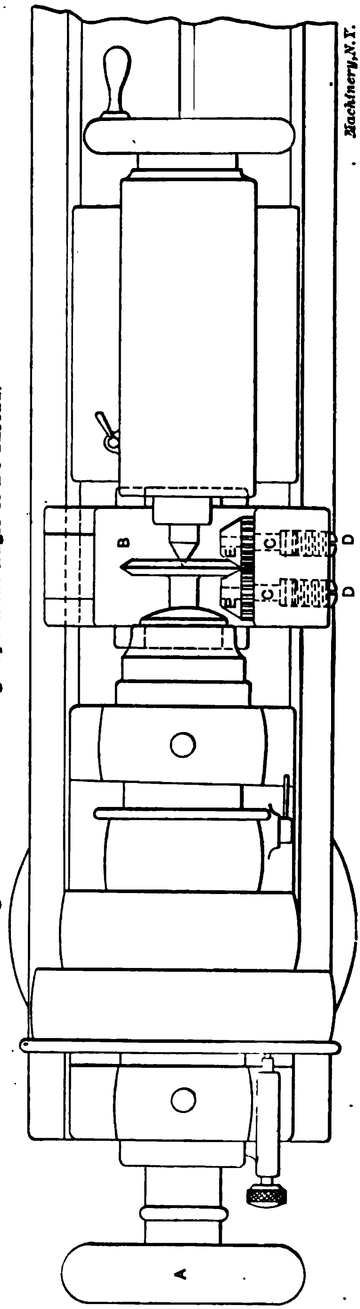

Fig. 23. Bench Lathe with Fixture for Charging Diamond Lap.

as to height. This is a good way of setting the thread tool to the same height as the axis of the lathe centers. This method of setting of the thread tool to height does not necessarily, however, insure that the thread tool in all cases will be set absolutely correct. If the thread tool-holder should be tipped somewhat out of the horizontal position, the top of the thread tool itself would not be horizontal, and consequently, when the gage pin *B* was brought down upon the top of the thread tool, so that the top face would lie perfectly in line with the lower face of gage pin *B*, this pin would not be fully horizontal, and the thread tool would not be set to the exact height of the lathe centers.

With the gage remaining between the lathe centers, the angle of the thread tool is set to a correct central position, sideways. This setting is also a check on the accuracy of the angle of the thread tool. A piece of white paper should be used under the gage and the tool, and a magnifying glass should be employed. First when the tool fits the gage so that all light is shut off, may the setting and the angle be

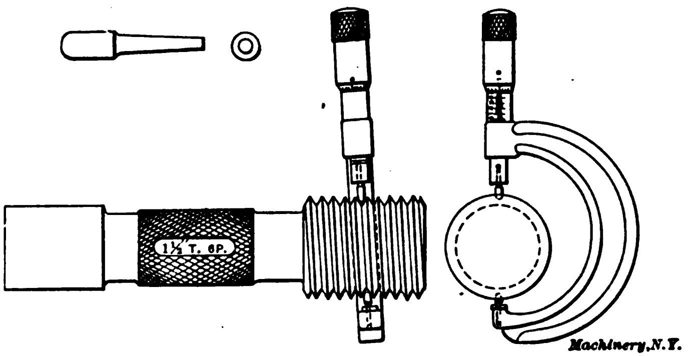

Fig. 24. Comparing Angle Diameters with Ball Point Micrometer.

considered satisfactory. The thread tool being set, we are now ready to proceed to finish thread our screw plug gage, which has previously been roughed out by a chaser having three or four teeth, leaving about 0.005 inch for the finishing single point thread tool. The finishing of the thread is continued until 0.0015 inch is left for lapping. The chaser, as well as the single point tool, should have a clearance of 15 degrees on the front face of the thread tool. This angle has proved to be the most advantageous for all practical purposes.

After having been finish threaded, the screw plug is case-hardened and ready for lapping. A lap made as shown in Fig. 27 is used. It will be seen that this lap is somewhat different from those ordinarily used for this work. The construction shown has been adopted because of the difficulty met with in circular laps which are split on one side for adjustment, but have nothing on the sides to hold the two sections in perfect alignment. Consequently, each of the sides has a tendency to follow the lead of the screw plug when lapping, and difficulty is experienced in getting a thread with perfect lead. The lap here shown,

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therefore, has a dowel pin *A* on each side for the purpose of holding the two sections in perfect alignment, and the adjusting screws *C* are inserted outside of the dowel pins. The two screws *B*, finally, clamp the two halves together. When the lap is assembled and screwed together, it is roughed out in the lathe with a threading tool, or tapped with three or four different sized taps, following one another in proper rotation. The lap is then taken apart, and planed on the inside to permit of adjustment; three grooves are cut in the thread on each side of the lap, for holding reserves of emery and oil. This will permit constant lubrication of the lap, and constant charging when lapping the screw plug to size. The lap is finished with a master tap, which must be made with extreme accuracy. This tap is ground in the angle of the thread, as shown in Fig. 22, and it is finished to a dimension 0.002 inch below the size diameter of the thread plug to be made, in order to permit the lap to wear down to the size when lapping.

The lathe must be revolved very slowly when grinding the master

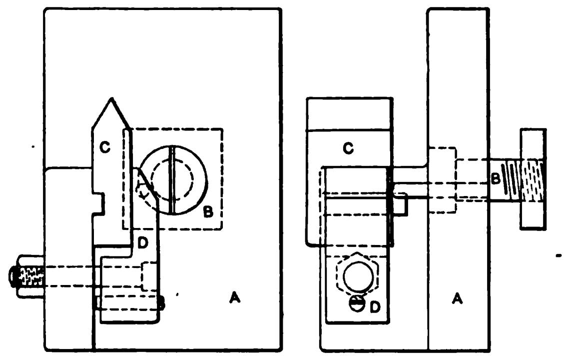

Fig. 25. Gage for Testing the Angle of the Thread.

tap, the revolutions of the spindle being from 20 to 100 per minute, according to the size of the tap. As will be seen in the cut, the cone pulley is placed where the back gears ordinarily are located. Gear *R* is disconnected, and the drive is through gears *S* and *T*. The reason for having the cone pulley in the back, is because it is wanted to use the space directly under the usual location of the cone pulley in the center of the lathe for a mechanism intended to permit a slight adjustment of the lead of the tap when grinding in the angle of the thread.

The feed screw *B* is placed in the center of the lathe bed, directly under the driving spindle, and fits into a solid nut, *C*, from which, through the medium of a casing *N* and a connecting-rod, the carriage is moved. A rod *E* is screwed into the nut *C*, this rod extending over the side of the lathe, and resting upon the edge of plate *F*, which can be so adjusted that it inclines from one end to the other from 0 to 20 degrees. Between this plate and the rod *E*, a shoe *P* is placed. On the extreme end of the rod hangs a weight *H* which holds the rod against the plate *F*. This arrangement serves the purpose of giving a slight change in the lead of the tap being ground, as it is evident

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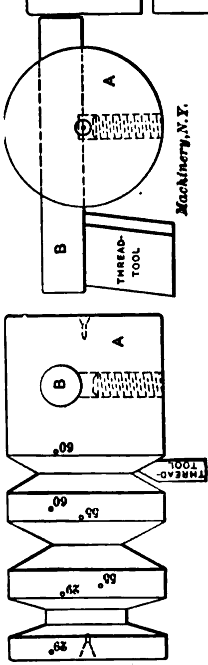

Fig. 26. Gage for Setting Thread Tools.

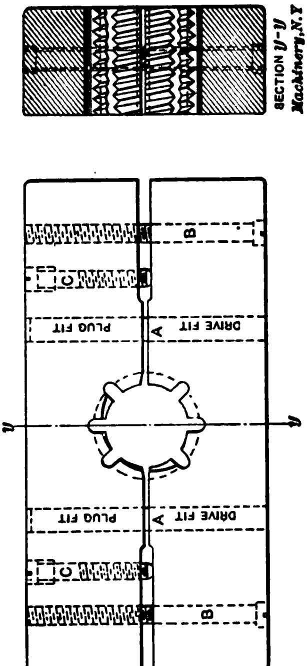

Fig. 27. Lap for Screw Plug Gages.

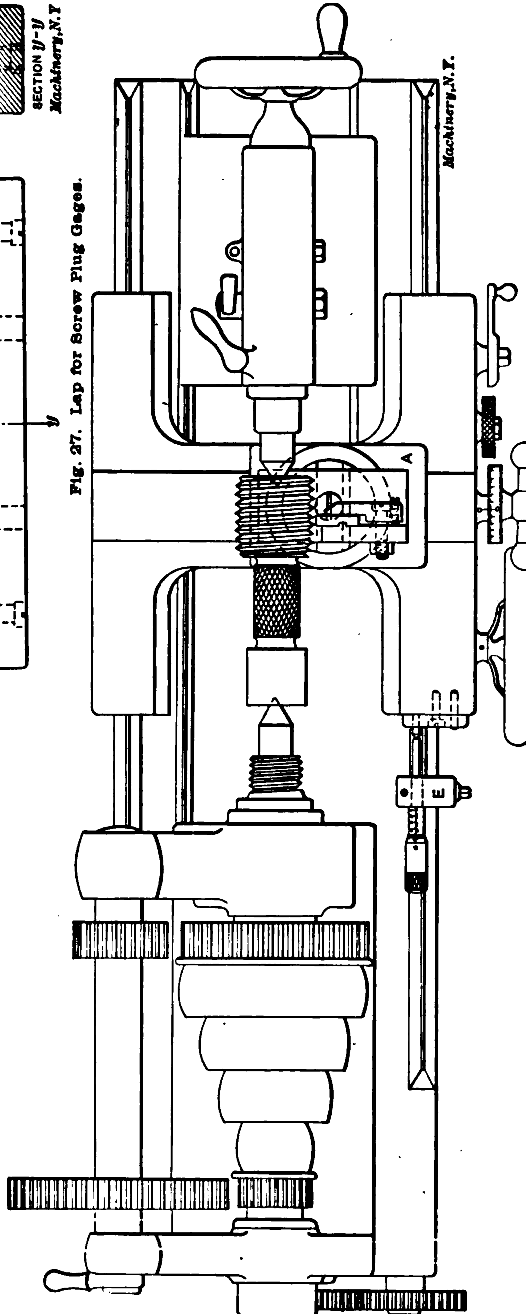

Fig. 28. Final Test of Pitch and Angle of Thread.

that when the rod *E* travels along the plate *F*, on the incline upward, it slightly turns the nut and moves it forward a trifle in excess of the regular forward motion imparted to the nut by the motion of the lead screw. By inclining the plate *F* in the other direction, the motion of the nut may be correspondingly retarded.

A grinding fixture *I* fits the slides on the top of the carriage. On the right-hand side of this fixture is placed a knurled handle *J*, graduated to thousandths of an inch. This handle is for the fine adjustment of the fixture, enabling the grinding wheel to be set correctly to the center of the thread, before starting the grinding operation. The top of the fixture swivels in a vertical plane, so that the wheel *L*, which is made of tool steel and charged with diamond dust, can be set at an angle to the vertical, either to the right or the left, according to the pitch and direction of the thread. This adjustment is made by loosening the nut *K* which binds the head in position when set to the correct angle. The wheel *L* is provided with a shank which fits a tapered hole

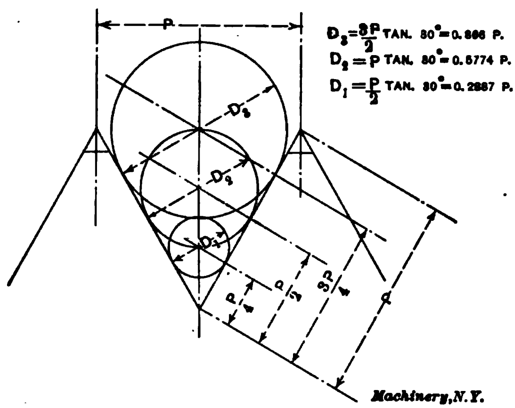

Fig. 29. Formula and Diagram for Determining Ball Points for U. S. Standard and V-Threads.

in the spindle *D*, which latter runs at a speed of 20,000 revolutions per minute. A solid backstop *M* is provided to hold the fixture securely in place while working. The lathe spindle, with the tap, and the grinding spindle run in the same direction, the same as in an ordinary grinder.

A good supply of sperm oil should be used when grinding the tap, and it is necessary to have a cover over the wheel, to prevent the throwing out of oil. This cover, however, is not shown in the cut. Care should be taken not to force the wheel into the work, as if that is done, the shape will soon be destroyed. The wheel should just barely touch the work, and should be fed in a very small amount, say, 0.00025 inch at a time. A sound magnifier or listener should be used, to hear whether the wheel is cutting moderately.

The wheel is charged in the following manner. A chuck, with a tapered hole which fits the shank of the diamond wheel, is placed in the spindle of the bench lathe. as shown in Fig. 23, and the tail-stock

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center is pushed up at the other end to get a good support when charging. Fixture *B* is placed in the bench lathe, and clamped with a bolt and nut from underneath the lathe, about the same as an ordinary slide-rest. The front end of the fixture extends up vertically above the center of the spindle. In this projecting part, two holes are drilled, reamed, and counterbored, at the same height as the center of the lathe spindle. In these holes are fitted two studs *C C* having a T-head inside the counterbored hole. Between the T-heads of these studs and the screws *D D* lie fiber washers, which act as friction stops. On the other end of plugs *C C* are placed hardened and ground rollers *E E* having one end beveled to a 30-degree angle, while the other end has spur gear teeth milled, which mesh into each other. With the slowest speed of the bench lathe, the fixture is fed in by hand, and having two slides at right angles to each other, the same as an ordinary slide-rest, it can be located to the proper position without much trouble. A piece of soft steel wire should be flattened out to make a spade, with

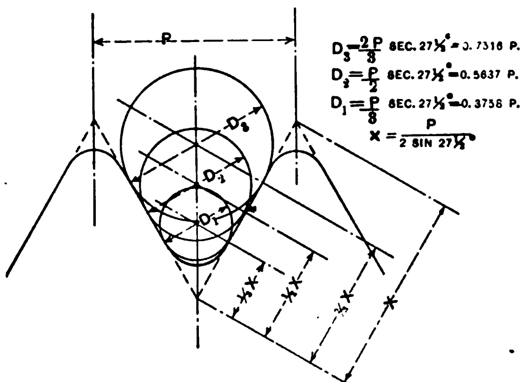

Fig. 30. Formula and Diagram for Determining Ball Points for Whitworth Thread.

which to take up the diamond dust for charging the wheel. One should not try to use a piece of wood, or a brush, as that will only be a waste of diamond. The master tap, which is to be ground, is relieved up to within 1/16 inch from its cutting edge with a file, this being done in order to prevent any more grinding than is absolutely necessary, and to permit the tap to cut freely. The length of the threaded part of the master tap should be about two times its diameter.

The master tap being finished, the lap for the screw plug gages, Fig. 27, is tapped, and ready for use. When charging this thread lap, great care should be taken not to force the lap too much. The spindle of the lathe, where the lapping is done, should be run very slowly, with the back gears in, until the lap is thoroughly charged with emery mixed with sperm oil. Then the lathe may be speeded up to a higher speed, according to the size of the screw plug. It is poor practice to use too much emery on the lap. Reverse the lap often, and use it the same amount on either side. If a large number of screw plugs are to

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be lapped, all of the same size, lap them all, one at a time, with the lap at the same setting. In this way the lap keeps its shape better, and can be used a long while before being retapped. Do not attempt to tap the lap with the master tap when charged with emery, but use a roughing tap first, and also wash out the lap in benzine before tapping. When the screw plug has been lapped to within 0.0005 inch of its size, it is ground on its outer diameter, if it be a U. S. Standard thread plug, and then finished by lapping after being ground. This will permit the top corners to be kept sharp, and better results will be obtained all around.

BALL DIAMETERS TO BE USED IN DETERMINING CORRECT ANGLE OF THREAD FOR V, U. S., AND BRIGGS STANDARD THREADS.

Threads per inch.	D_3	D_2	D_1
32	0.028	0.018	0.010
28	0.030	0.020	0.010
24	0.035	0.024	0.012
22	0.040	0.026	0.014
20	0.045	0.028	0.014
18	0.050	0.030	0.016
16	0.055	0.035	0.018
14	0.060	0.040	0.020
13	0.065	0.045	0.022
12	0.070	0.050	0.024
11	0.080	0.055	0.026
10	0.085	0.060	0.030
9	0.095	0.065	0.030
8	0.100	0.070	0.035
7	0.120	0.080	0.040
6	0.140	0.095	0.050
5½	0.160	0.110	0.050
5	0.170	0.120	0.060
4½	0.190	0.130	0.065
4	0.220	0.140	0.075
3½	0.240	0.170	0.085
3	0.280	0.190	0.095
2⅞	0.300	0.200	0.100
2¾	0.320	0.220	0.100
2⅝	0.320	0.220	0.110
2½	0.340	0.240	0.110
2⅜	0.360	0.240	0.120
2¼	0.380	0.260	0.130

Great care must be exercised during the lapping operation to see that the angle of the thread is correct. The gaging of the angle of the thread is accomplished in the following manner. Three micrometers are used to measure the correct angle. Two ball points of the same size are placed in tapered holes in each micrometer, as shown in Fig. 24. These ball points are ground all over, and made to a shape as shown in the upper left-hand corner in Fig. 24. The body of these ball points is ground parallel, and then the end is turned and ground to a ball shape as shown. Three sets of ball points are used for each pitch, one to measure the thread near its bottom, one at the center,

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and one near the top, as indicated in Figs. 29 and 30. The master screw plug is used for comparison; one micrometer is set to the master screw plug at the bottom of the thread, in the manner indicated in Fig. 24, and is then tried on the thread plug being made. The difference in diameter between the measured diameter on the master gage, and that on the plug being made, is noted. Then the two other micrometers, measuring at the center and near the top of the thread, are used, and the difference between the master gage and the screw plug diameters at the places where these micrometers measure, is also noted. If all three micrometers show the same amount of difference

BALL DIAMETERS TO BE USED IN DETERMINING CORRECT ANGLE OF THREAD FOR WHITWORTH STANDARD THREAD.

Threads per inch.	D_3	D_2	D_1
32	0.024	0.018	0.012
28	0.026	0.020	0.014
24	0.030	0.024	0.016
22	0.035	0.026	0.018
20	0.040	0.028	0.018
18	0.040	0.030	0.020
16	0.045	0.035	0.024
14	0.055	0.040	0.026
13	0.060	0.045	0.028
12	0.065	0.045	0.030
11	0.070	0.050	0.035
10	0.075	0.055	0.035
9	0.085	0.060	0.040
8	0.095	0.070	0.045
7	0.110	0.080	0.055
6	0.130	0.095	0.060
5½	0.140	0.100	0.070
5	0.150	0.110	0.075
4½	0.170	0.120	0.085
4	0.190	0.140	0.095
3½	0.220	0.160	0.110
3	0.260	0.190	0.120
2¾	0.260	0.200	0.130
2⅝	0.280	0.200	0.140
2½	0.280	0.220	0.140
2⅜	0.300	0.220	0.150
2¼	0.320	0.240	0.160
2⅛	0.340	0.260	0.170

in relation to the master plug, then the angle of the thread evidently must be correct. After that, the micrometer measuring at the center of the thread is used to measure the size of the screw plug, comparing it with that of the master gage, until the plug is finished to size.

Figs. 29 and 30 show how formulas are derived for the size of the ball points used in measuring. Fig. 29 applies to a 60-degree thread, either sharp V or U. S. Standard, while Fig. 30 gives the formulas for a Whitworth thread. The diameters D_1, D_2, and D_3, respectively, are the diameters of the cylindrical portions of the ball points used, and are, of course, also the diameters of the half-spheres on the end

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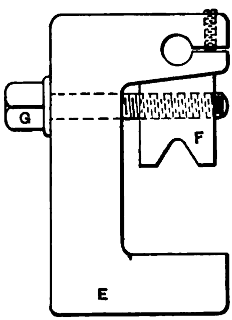
of the ball points. The tables on pages 33 and 34 give these diameters for different pitches, figured approximately from the formulas.

For testing the angle of the screw plug, when finally finished to a limit of 0.0005 inch, it is tried in a testing machine, such as shown in Fig. 28. This machine is simply an ordinary lathe, fitted with a fixture *A*, shown separately in Fig. 25. The tool-post is taken off the lathe, and replaced with this fixture, which is clamped in the T-slot of the tool-post slide, with bolt *B*, Fig. 25. The thread gage *C* is ground all over, and the angle fitted to a master gage. The gage *C* is held by the tongue and groove on the left-hand side of the fixture, and clamped with a strap *D*. To set this gage correctly, in relation to the axis of the spindle of the lathe, as regards height as well as angle, the angle gage, Fig. 26, is used in the same way as has been previously explained in relation to thread cutting. When the fixture has been placed correctly in position, the screw plug is inspected by placing the gage



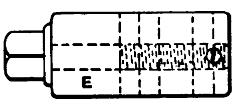


Fig. 31. Holder for Micrometer Stop.

with the hand first to the right and then to the left side of the thread angle. A strong magnifying glass is used with a white paper underneath, and any imperfection of the angle is easily detected, and can be corrected when lapping the last 0.0005 inch to size. If the test gage shows an opening either at the bottom or at the top, the fault is that the lap is worn and must be retapped, or it may be that too much emery has been used. If, for some reason or other, it is impossible to correct the screw plug within 0.0001 inch, when lapping, take a piece of hard wood, or flatten a piece of copper wire, charge it with emery, and hand lap the high points of the angle, while the screw plug is revolving slowly in the lathe. In this way, it is comparatively easy to overcome this trouble, but great care must be taken to follow the thread properly with the hand lap.

To find if a screw thread has a perfect lead, the micrometer stop *E*, Fig. 28, is placed on the left-hand side of the carriage. The holder for

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this micrometer stop is shown separately in Fig. 31. The construction of this stop is very simple. The micrometer head is an ordinary one, as made for the trade by manufacturers of these instruments. The holder *E* is made similar to a C-clamp, with a hole drilled and reamed to fit the micrometer head. A slot is sawed through the upper jaw, with a stop screw on the top, which prevents the micrometer from being clamped too hard in the holder, in which case the thimble would not revolve freely. Underneath this hole the holder is beveled off, and a V-block *F* is held in position by a screw *G*, entering from the side. The micrometer head is placed in the hole provided for it, with its division reading faced upwards, and the screw *G* clamps the micrometer head and the holder *E* at the same time. When the lead of the screw plug is tested, the carriage is moved one inch along the thread. It is understood that the lead-screw of the lathe is not employed in this case, but one depends upon the micrometer for measuring the correct lead of the screw plug.

The master plug may, of course, also be placed between the centers and comparison be made with the master plug. In this case, the micrometer serves as a comparator. A plate is screwed on the left-hand side of the carriage, provided with a hardened stop against which the end of the micrometer screw bears. It is evident that the carriage must not be moved against the micrometer with too much force, but simply brought up to barely touch against the end of the micrometer screw.

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CHAPTER V.

MEASURING SCREW THREAD DIAMETERS.

It is always advisable when measuring screw thread diameters to measure them in the angle, in addition to testing their diameter on the top of the threads and at the bottom of the thread groove, but unless calipers made expressly for the work are at hand, the measurement in the thread angle is apt to be omitted. The tables on pages 38, 39, 40, and 41 were worked out by Mr. Walter Cantelo in 1902 for convenient application in the inspection of screw threads in connection with ordnance inspection for the United States army. The method is known as the three-wire system of screw measurement, because three wires, of the diameter called for in the tables, and applied as shown in the accompanying diagrams, are used in connection with an ordinary flat point micrometer. The dimensions for the standard threads of the systems shown are given in the tables mentioned. For threads of special size or pitch, the values for the various thread parts are easily computed from the formulas given for the kind of thread under consideration. It is especially necessary that the wires used be as nearly round in section as possible, and of uniform diameter.

Two methods of measuring are shown for the 60-degree V, U. S. standard and Whitworth threads, and for each method a formula and table of values are given. The three-wire method is preferred, because the error in the thread groove—if any be present—is taken into consideration twice, while by the single-wire method errors are liable to be introduced by the surface on top of the threads not being exactly concentric with the thread groove. It is evident that for each of the threads the wire to be used in any thread groove is limited in regard to diameters as follows:

The 60-Degree V-Thread.

Maximum diameter of wire $= \dfrac{p}{0.866} = 1.155\,p$, if p equals the pitch of the thread.

Minimum diameter of wire $= 0.577\,p$.

The U. S. Standard Thread.

Maximum width of thread groove $= \frac{7}{8}\,p$.

Maximum diameter of wire $= \frac{7}{8} \times \dfrac{p}{0.866} = 1.010\,p$.

Minimum diameter of wire $= \frac{7}{8} \times 0.577\,p = 0.505\,p$.

The 55-Degree Whitworth Thread.

Let $p_1 =$ distance across thread groove at point where radii are tangent to angle.

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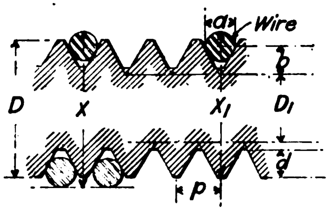
U.S STANDARD



n = number of threads per inch

p = pitch = $\frac{1}{\text{no. of threads per inch}}$

d = depth of thread = 0.6495p = $\frac{0.6495}{n}$

D = diameter on top of threads

$D_1 = D - \frac{1.5155}{n}$

a = diameter of wire { maximum diam. = 1.010p; mininum diam. = 0.505p

b = a

$X = D_1 + 2b + a = D_1 + 3a$

$X_1 = \frac{D}{2} + \frac{D_1}{2} + b + \frac{a}{2} = \frac{D + D_1 + 3a}{2}$

D	n	d	D_1	a and b	X	X_1
1/4"	20"	.0325"	.1742"	0.040"	.2942"	.2721"
5/16	18	.0361	.2283	"	.3483	.3304
3/8	16	.0406	.2803	"	.4003	.3876
7/16	14	.0464	.3292	"	.4492	.4433
1/2	13	.0500	.3834	0.060	.5634	.5317
9/16	12	.0541	.4362	"	.6162	.5893
5/8	11	.0590	.4872	"	.6672	.6460
11/16	11	.0590	.5497	"	.7297	.7086
3/4	10	.0649	.5984	"	.7784	.7643
13/16	10	.0649	.6610	"	.8410	.8267
7/8	9	.0722	.7066	0.100	1.0066	.9408
15/16	9	.0722	.7691	"	1.0690	1.0033
1	8	.0812	.8105	"	1.1105	1.0553
1 1/8	7	.0928	.9085	"	1.2085	1.1667
1 1/4	7	.0928	1.0335	"	1.3335	1.2917
1 3/8	6	.1082	1.1224	"	1.4224	1.3987
1 1/2	6	.1082	1.2474	"	1.5474	1.5237
1 5/8	5 1/2	.1180	1.3494	0.150	1.7994	1.7122
1 3/4	5	.1299	1.4470	"	1.8970	1.8234
1 7/8	5	.1299	1.5720	"	2.0220	1.9484
2	4 1/2	.1443	1.6632	"	2.1132	2.0566
2 1/8	4 1/2	.1443	1.7882	"	2.2382	2.1816
2 1/4	4 1/2	.1443	1.9132	"	2.3632	2.3066
2 3/8	4	.1624	1.9960	"	2.4460	2.4105
2 1/2	4	.1624	2.1210	"	2.5710	2.5355
2 3/4	4	.1624	2.3710	"	2.8210	2.7855
3	3 1/2	.1856	2.5670	0.200	3.1670	3.0835
3 1/4	3 1/2	.1856	2.8170	"	3.4170	3.3335
3 1/2	3 1/4	.2000	3.0337	"	3.6337	3.5668
3 3/4	3	.2165	3.2448	"	3.8448	3.7974
4	3	.2165	3.4948	"	4.0943	4.0474

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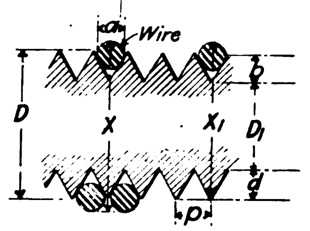
60° V THREAD



n = number of threads per inch

p = pitch = $\frac{1}{\text{no. of threads per inch}}$

d = depth of thread = $0.866p = \frac{0.866}{n}$

D = diameter of top of threads

D_1 = root diameter = $D - 2d$

a = diameter of wire $\begin{cases} \text{maximum diam.} = 1.155p \\ \text{minimum diam.} = 0.577p \end{cases}$

$b = a$

$x = D_1 + 2b + a = D_1 + 3a$

$x_1 = \frac{D}{2} + \frac{D_1}{2} + b + \frac{a}{2} = \frac{D + D_1 + 3a}{2}$

D	n	d	D_1	a and b	x	x_1
1/4"	20"	.0433"	.1634"	0.040"	.2834"	.2667"
5/16	18	.0481	.2163	"	.3363	.3244
3/8	16	.0541	.2667	"	.3867	.3808
7/16	14	.0617	.3138	0.060	.4938	.4656
1/2	12	.0722	.3557	"	.5357	.5178
9/16	12	.0722	.4182	"	.5982	.5803
5/8	11	.0787	.4676	"	.6476	.6363
11/16	11	.0787	.5300	"	.7100	.6987
3/4	10	.0866	.5768	0.100	.8768	.8134
13/16	10	.0866	.6393	"	.9393	.8759
7/8	9	.0962	.6826	"	.9826	.9288
15/16	9	.0962	.7450	"	1.0450	.9912
1	8	.1082	.7835	"	1.0835	1.0417
1 1/8	7	.1237	.8776	"	1.1776	1.1513
1 1/4	7	.1237	1.0026	"	1.3026	1.2763
1 3/8	6	.1443	1.0863	0.150	1.5363	1.4556
1 1/2	6	.1443	1.2113	"	1.6613	1.5806
1 5/8	5	.1732	1.2786	"	1.7286	1.6768
1 3/4	5	.1732	1.4036	"	1.8536	1.8018
1 7/8	4 1/2	.1924	1.4900	"	1.9400	1.9075
2	4 1/2	.1924	1.6150	"	2.0650	2.0325
2 1/8	4 1/2	.1924	1.7400	"	2.1900	2.1575
2 1/4	4 1/2	.1924	1.8650	"	2.3150	2.2825
2 3/8	4 1/2	.1924	1.8900	"	2.4400	2.4075
2 1/2	4	.2165	2.0670	0.200	2.6670	2.5835
2 3/4	4	.2165	2.3170	"	2.9170	2.8335
3	3 1/2	.2474	2.5050	"	3.1050	3.0525
3 1/4	3 1/2	.2474	2.7550	"	3.3550	3.3025
3 1/2	3 1/4	.2664	2.9670	"	3.5670	3.5335
3 3/4	3	.2886	3.1727	"	3.7727	3.7613
4	3	.2886	3.4227	"	4.0227	4.0113

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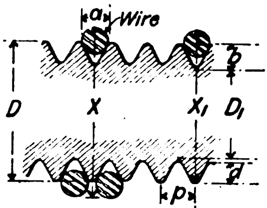
WHITWORTH THREAD



n = number of threads per inch

p = pitch = $\frac{1}{\text{no. of threads per inch}}$

d = depth of thread = $0.6403p = \frac{0.6403}{n}$

D = diameter on top of threads

$D_1 = D - \frac{1.6008}{n}$

a = diameter of wire { maximum diam. = 0.840p; mininum diam. = 0.506p }

b = 1.08205a

$x = D_1 + 2b + a = D_1 + 3.1657a$

$x_1 = \frac{D}{2} + \frac{D_1}{2} + b + \frac{a}{2} = \frac{D + D_1 + 3.1657a}{2}$

D	n	d	D_1	a	b	x	x_1
1/4	20	.0320	.1699	0.040	.0433	.2965	.2733
5/16	18	.0356	.2235	"	"	.3501	.3313
3/8	16	.0400	.2749	"	"	.4015	.3883
7/16	14	.0457	.3231	"	"	.4497	.4436
1/2	12	.0534	.3666	"	"	.4932	.4966
9/16	12	.0534	.4291	0.060	.0649	.6190	.5907
5/8	11	.0582	.4794	"	"	.6693	.6372
11/16	11	.0582	.5420	"	"	.7319	.7097
3/4	10	.0640	.5899	"	"	.7798	.7649
13/16	10	.0640	.6524	"	"	.8423	.8274
7/8	9	.0711	.6971	"	"	.8870	.8810
15/16	9	.0711	.7596	"	"	.9495	.9435
1	8	.0800	.7999	0.100	1084	1.1167	1.0583
1 1/8	7	.0915	.8963	"	"	1.2131	1.1690
1 1/4	7	.0915	1.0213	"	"	1.3381	1.2940
1 3/8	6	.1067	1.1082	"	"	1.4250	1.3999
1 1/2	6	.1067	1.2332	"	"	1.5500	1.5250
1 5/8	5	.1281	1.3048	0.150	.1624	1.7796	1.7023
1 3/4	5	.1281	1.4298	"	"	1.9046	1.8273
1 7/8	4 1/2	.1430	1.5193	"	"	1.9941	1.9345
2	4 1/2	.1430	1.6443	"	"	2.1191	2.0595
2 1/8	4 1/2	.1430	1.7693	"	"	2.2441	2.1845
2 1/4	4	.1601	1.8498	"	"	2.3246	2.2873
2 3/8	4	.1601	1.9750	"	"	2.4498	2.4123
2 1/2	4	.1601	2.1000	"	"	2.5748	2.5373
2 3/4	3 1/2	.1830	2.2926	0.200	.2157	2.9240	2.8370
3	3 1/2	.1830	2.5426	"	"	3.1740	3.0870
3 1/4	3 1/4	.1970	2.7574	"	"	3.3887	3.3194
3 1/2	3 1/4	.1970	3.0074	"	"	3.6387	3.5694
3 3/4	3	.2134	3.2164	"	"	3.8477	3.7990
4	3	.2134	3.4664	"	"	4.0977	4.0490

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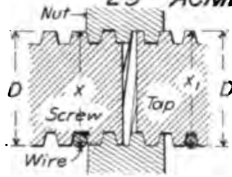
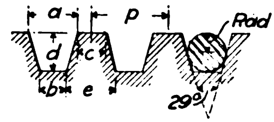
29° ACME SCREW THREAD





SCREW THREAD

p - pitch - $\frac{1}{\text{no. of threads per inch}}$
d - depth of thread - $\frac{p}{2}$ + 0.010"
a - space at top - 0.6293 p
b - space at bottom - 0.3707 p - 0.0052"
c - thickness at top - 0.3707 p
e - thickness at bottom - 0.6293 p + 0.0052"
D - diameter at top of thread
x - D + 0.010"

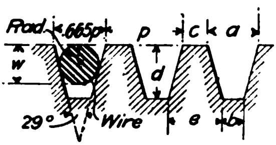
TAP THREAD

p - pitch - $\frac{1}{\text{no. of threads per inch}}$
d - depth of thread - $\frac{p}{2}$ + 0.020"
a - space at top - 0.6293 p + 0.0052"
b - space at bottom - 0.3707 p - 0.0052"
c - thickness at top - 0.3707 p - 0.0052"
e - thickness at bottom - 0.6293 p + 0.0052"
D_1 - diameter at top of thread - D + 0.020"
x_1 - D_1 - D + 0.020"

Threads per inch	p	d	Diameter of wire	Threads per inch	p	d	Diameter of wire
½	2.000"	1.0100"	0.9785"	3	0.3333"	0.1767"	0.1664"
¾	1.500	0.7600	0.7349	4	0.2500	0.1350	0.1278
1	1.000	0.5100	0.4913	5	0.2000	0.1100	0.1014
1⅓	0.750	0.3850	0.3694	6	0.1667	0.0933	0.0852
1½	0.667	0.3433	0.3288	7	0.1429	0.0814	0.0736
1¾	0.571	0.2957	0.2824	8	0.1250	0.0725	0.0649
2	0.500	0.2600	0.2476	9	0.1111	0.0655	0.0581
2½	0.400	0.2100	0.1989	10	0.1000	0.0600	0.0527

The wire used is of such diameter, that when laid in the thread groove of the tap, it will be flush with the top of the threads, and when laid in the thread groove of the screw, it will extend beyond the top of the threads 0.010."

THE BROWN AND SHARPE 29° WORM THREAD



p - pitch - $\frac{1}{\text{no. of threads per inch}}$
d - depth of thread - 0.6866 p
a - space at top - 0.665 p
b - space at bottom - 0.310 p
c - thickness at top - 0.335 p
e - thickness at bottom - 0.690 p
w - diam. of wire - 0.5149 p

Pitch	d	Wire Diam.	Pitch	d	Wire Diam.
2.000	1.3732	1.0298	.3333	.2286	.1716"
1.750	1.2015	.9010	.2500	.1716	.1287
1.500	1.0299	.7723	.2000	.1373	.1030
1.250	.8582	.6436	.1667	.1144	.0858
1.000	.6866	.5149	.1250	.0858	.0643
.750	.5150	.3862	.1111	.0763	.0582
.500	.3433	.2574	.1000	.0687	.0515

The wire used is of such diameter that it will be flush with the top of the thread when laid in the thread groove.

Radius on thread = 0.1373 p, and arc forming top of threads contains 125 degrees.

Then $p_1 = p - 2 \sin 62° 30' \times 0.1373\, p = p - 2 \times 0.887 \times 0.1373\, p = p - 0.243\, p = 0.75\, p$, approx.

$$\text{Maximum diameter of wire} = 0.75 \times \frac{p}{0.887} = 0.84\, p.$$

$$\text{Minimum diameter of wire} = 5/6 \times 0.6068\, p = 0.505\, p.$$

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Principle of Method of Measurements.

The dimension D_1 (see tables, pages 38 to 41) must be considered for both the single-wire and three-wire methods and has values as follows:

For the 60-degree thread: The depth equals $0.866\,p$ and as the apex of the thread angle and root of thread groove are at the same point, it follows that $D_1 = D - 0.886\,p \times 2 = D - 1.732\,p$ or $= D - \frac{1.732}{n}$.

For the U. S. Standard thread: The depth equals 6/8 of the 60-degree V thread, being flattened on top and filled in at the root an amount equaling one-eighth of the V thread depth, and the distance from the top of thread to apex of thread angle at root, therefore, equals ⅞ of the V thread depth, or $\frac{7}{8} \times 0.866\,p$ and $D_1 = D - \frac{7}{8} \times 0.866\,p \times 2 = D - 1.5155\,p$ or $= D - \frac{1.5155}{n}$.

For the Whitworth 55-degree thread: The depth equals 4/6 of the 55-degree V thread depth, being filled in at the root and cut away on top an amount equaling 1/6 of the V thread depth. The depth of the 55-degree V thread would be $0.96045\,p$ and the distance from top of Whitworth thread to apex of thread angle at root equals $5/6 \times 0.96045\,p$ or $0.8004\,p$ and $D_1 = D - 2 \times 0.8004\,p = D - 1.6008\,p$, or $= D - \frac{1.6008}{n}$.

From the foregoing it will be seen how the formulas $x = D_2 + 2b + a$ for the three-wire system, and $x_1 = \frac{D}{2} + \frac{D_2}{2} + b + 2a$ for the single-wire system are produced, and also, it will be readily seen how easily the formulas $x = D - \frac{1.732}{n} + 3\,a$ for the 60-degree V thread and $x = D - \frac{1.5155}{n} + 3\,a$ for the U. S. Standard thread, as given in Chapter II, may be arrived at.

The Acme 29-Degree Screw Thread.

For this thread it is best to use a separate wire for each pitch, of such diameter that when laid in the thread groove of the tap or thread plug gage, it will be flush with the tops of the threads when they are of correct dimensions, and when laid in the thread groove of the screw it will extend 0.010 inch beyond the tops of the threads.

The Brown & Sharpe 29-Degree Worm Thread.

For this thread it is best to use a separate wire for each pitch that will be flush with the tops of the threads when laid in the finished thread groove.

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No. 32

SCREW THREAD CUTTING

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SCREW THREAD CUTTING.

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INTRODUCTION.*

The terms pitch and lead of screw threads are often confused, and particularly in the case of multiple threaded screws does this confusion cause difficulties. Before we therefore enter upon the subject of figuring change gears for the lathe for cutting screw threads, it may be well to make clear the real meaning of the words "pitch" and "lead" and their relation to the number of threads per inch.

The *pitch* of a screw thread is the distance from the top of one thread to the top of the next, as shown in Fig. 1. No matter whether the screw has single, double, triple, or quadruple thread, the pitch is always the distance from the top of one thread to the top of the next. Often, though improperly, the word "pitch" is used in the shop to denote "number of threads per inch." We hear of screws having 12 pitch thread, 16 pitch thread, etc. This is not correct usage of the word pitch, and only tends to cause unnecessary confusion.

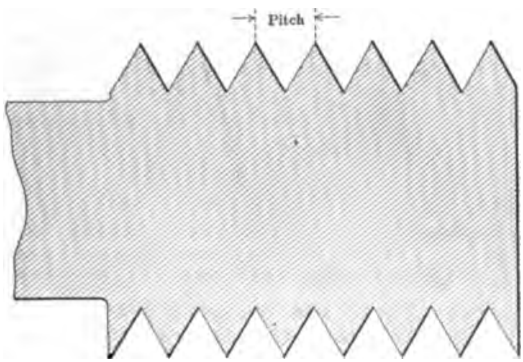

Fig. 1. The Pitch of a Screw Thread.

The *lead* of a screw thread is the distance the screw will move forward in a nut if turned around one full revolution. It is clear that for a single-threaded screw the pitch and the lead are equal, as the screw would then move forward the distance from one thread to the next if turned around once. In a double-threaded screw, however, the screw will move forward two threads, or twice the pitch, so that in a double-threaded screw the lead equals twice the pitch. In a triple-threaded screw the lead equals three times the pitch, and so forth.

The lead may also be expressed as being the distance from center to center of the *same* thread, after this thread has made one turn around the screw. In the single-threaded screw the *same* thread is the next thread to the one first considered. In a double-threaded screw there are two threads running side by side around the screw, so that

* The present introduction, and part of Chaper I, has been reprinted in this book from No. 18 of MACHINERY'S Reference Series, entitled "Shop Arithmetic for the Machinist," because it has been considered that any treatise on screw thread cutting would be incomplete without a thorough treatment on the subject of lead, pitch, and change gearing.

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the *same* thread here is the second one from the one first considered. In a triple-threaded screw it is the third one, in a quadruple-threaded, the fourth, and so forth. However we consider this, we still see that the lead and pitch are alike for a single-threaded screw, that the lead is twice the pitch for a double-threaded, and three times for a triple-threaded, as already stated. The actual relationship is very plainly shown in Fig. 2, where parts of three screws with Acme threads are shown, the first single-threaded, the second double-threaded, and the last triple-threaded.

The main point to remember, however, is that in *any kind of a screw, the lead is the distance which the screw will move forward in a nut if turned around one revolution.*

In this connection it may be appropriate to give the rules and formulas for the relation between the lead and the number of threads per inch. If there are 8 threads, single, in one inch, the lead is

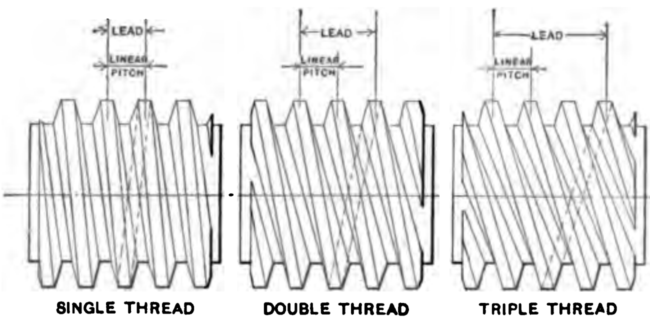

Fig. 2. Comparison between Single, Double and Triple Threads.

evidently 1/8 inch. This we find, mathematically, by dividing *one* by 8, which is the number of threads per inch. The formula, therefore, is

$$\text{lead} = \frac{1}{\text{number of threads per inch}}$$

This formula, expressed in words, says: *The lead of a screw equals one divided by the number of threads per inch.*

Confusion is often caused by indefinite designation of multiple thread screws. The most common way to state the lead and the class of thread is perhaps to say 1/4 *inch lead, double*, which means a screw with a double thread, which, when cut, has the lathe geared for four threads per inch, but each thread is cut only to a depth corresponding to eight threads per inch. The same condition is also expressed by: *4 threads per inch, double.* These two ways of expressing the number of multiple threads are both correct, but the expression which ought to be used in order to avoid misunderstanding under any circumstances would be: 1/4 *lead,* 1/8 *pitch. double thread.*

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CHAPTER I.

CHANGE GEARS FOR THREAD CUTTING.

While the principles and rules governing the calculation of change gears are very simple, they, of course, presuppose some fundamental knowledge of the use of common fractions. If such knowledge is at hand, the subject of figuring change gears, if once thoroughly understood, can hardly ever be forgotten. It should be impressed upon the minds of all who have found difficulties with this subject that the matter is not approached in a logical manner, and is usually grasped by the memory rather than by the intellect. Before attempting to lay down any definite rules for the figuring of change gears, let us therefore analyze the subject. The lead-screw *B* of the lathe (see Fig. 3) must be recognized as our first factor, and the spindle as the second. If the lead-screw has six threads per inch, then, if the lead-screw makes six revolutions, the carriage travels one inch, and the thread-cutting tool travels one inch along the piece to be threaded. If the spindle makes the same number of revolutions in a given time as the lead-screw, it is clear the tool will cut six threads per inch. In such a case the gear *D* on the spindle stud *J*, and gear *E* on the lead-screw, are alike. If the spindle makes twice the number of revolutions of the lead-screw, the spindle revolves twelve times while the tool moves one inch, and consequently twelve threads per inch will be cut. But in order to make the spindle revolve twice as fast as the lead-screw, it is necessary that a gear be put on the spindle stud of only half the number of teeth of the gear on the lead-screw, so that when the lead-screw revolves once the spindle stud gear makes two revolutions.

Simple Gearing.

Suppose we wish to cut nine threads per inch with a lead-screw of six threads per inch, as referred to above. Then the six threads of the lead-screw correspond to nine threads on the piece to be threaded, which is the same as to say that six revolutions of the lead-screw correspond to nine revolutions of the spindle; or in other words, one revolution of the lead-screw corresponds to 1½ of the spindle. From this it is evident that the gear on the lead-screw must make only one revolution while the spindle stud gear makes 1½. Thus, if the lead-screw gear has, for instance, 36 teeth, the gear on the spindle stud should have only 24; the smaller gear, of course, revolving faster than the larger. If we express what has been previously said in a formula we have:

$$\frac{\text{threads per inch of lead-screw}}{\text{threads per inch to be cut}} = \frac{\text{teeth in gear on spindle stud}}{\text{teeth in gear on lead-screw}}$$

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Applying this to the case above, we have:

$$\frac{6}{9} = \frac{24}{36}$$

The values 24 and 36 are obtained by multiplying 6 and 9, respectively, by 4. By multiplying both the numerator and the denominator by the same number, we do not change the proportion. As a general rule we may then say that the change gears necessary to cut a certain number of threads per inch are found by placing the number of threads in the lead-screw in the numerator, the number of threads to be cut in the denominator, and then multiply numerator as well as denominator by the *same* number, by trial, until two gears are obtained, the number of teeth of which are both to be found in the set of gears accompanying the lathe. The gear with the number of teeth designated by the new numerator is to be placed on the spindle stud (at *J*, Fig. 3), and

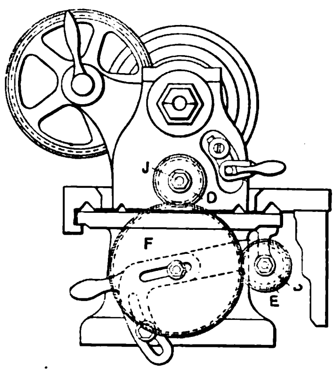

Fig. 3. Simple Gearing.

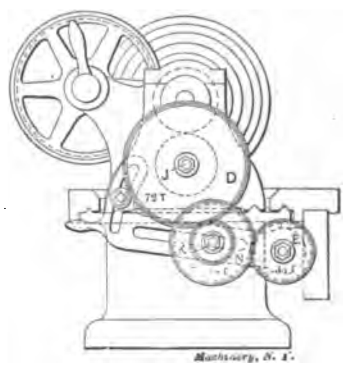

Fig. 4. Compound Gearing.

the gear with the number of teeth corresponding to the denominator on the lead-screw *B*.

A few examples of this will more clearly explain the rule. Suppose the number of teeth of the change gears of a lathe are 24, 28, 32, 36, and so forth, increasing by 4 teeth up to 100. Assume that the lead-screw is provided with 6 threads per inch, and that 10 threads per inch are to be cut. Then,

$$\frac{6}{10} = \frac{6 \times 4}{10 \times 4} = \frac{24}{40}$$

By multiplying both numerator and denominator by 4, we obtain two available gears with 24 and 40 teeth, respectively. The 24-tooth gear goes on the spindle stud, and the 40-tooth gear on the lead-screw. Assuming the same lathe and gears, let us find the gears for cutting 11½ threads per inch, this being the standard number of threads for certain sizes of pipe thread. Then,

$$\frac{6}{11\frac{1}{2}} = \frac{6 \times 8}{11\frac{1}{2} \times 8} = \frac{48}{92}$$

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It will be found that multiplying with any other number than eight would, in this case, not have given us gears with such number of teeth as we have in our set with this lathe. Until getting accustomed to figuring of this kind, we can, of course, only by trial find out the correct number by which to multiply numerator and denominator. The number of teeth in the *intermediate* gear *F*, Fig. 3, which meshes with both the spindle stud gear and the lead-screw gear is of no consequence.

Lathes with Reduction Gearing in Head-Stock.

In some lathes, however, there is a reduction gearing in the head-stock of the lathe, so that if equal gears are placed on the lead-screw and the spindle stud, the spindle does not make the same number of revolutions as the lead-screw, but a greater number. Usually in such lathes the ratio of the gearing in the head-stock is 2 to 1, so that with equal gears the spindle makes two revolutions to one of the lead-screw. This is particularly common in lathes intended for cutting fine pitches or, in general, in small lathes. When figuring the gears this must, of course, be taken into consideration. As the spindle makes twice as many revolutions as the lead-screw with equal gears, if the ratio of the gears be 2 to 1, that means that if the head-stock gearing were eliminated, and the lead-screw instead had twice the number of threads per inch as it has, with equal gears the spindle would still revolve the same as before for each inch of travel along the piece to be threaded. In other words, the gearing in the head stock may be *disregarded, if the number of threads of the lead-screw is multiplied by the ratio of this gearing.* Suppose, for instance, that in a lathe the lead-screw has eight threads per inch, that the lathe is geared in the head-stock with a ratio of 2 to 1, and that 20 threads are to be cut. Then

$$\frac{2 \times 8}{20} = \frac{16}{20} = \frac{16 \times 4}{20 \times 4} = \frac{64}{80}$$

which two last values signify the number of teeth in the gears to use.

Sometimes the ratio of the gearing in the head-stock cannot be determined by counting the teeth in the gears, because the gears are so placed that they cannot be plainly seen. In such a case, equal gears are placed on the lead-screw and the spindle stud, and a thread cut on a piece in the lathe. The number of threads per inch of this piece should be used for the numerator in our calculations instead of the actual number of threads of the lead-screw. The ratio of the gearing in the head-stock is equal to the ratio between the number of threads cut on the piece in the lathe and the actual number of threads per inch of the lead-screw.

Compound Gearing.

The cases with only two gears in a train referred to are termed simple gearing. Sometimes it is not possible to obtain the correct ratio excepting by introducing two more gears in the train, which, as is well known to mechanics, is termed compound gearing. This class

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of gearing is shown in Fig. 4. The rules for figuring compound gearing are exactly the same as for simple gearing excepting that we must divide both our numerator and denominator into two factors, each of which are multiplied with the same number in order to obtain the change gears.

Suppose a lathe has a lead-screw with six threads per inch, and that the number of the teeth in the gears available are 30, 35, 40, and so forth, increasing by 5 up to 100. Assume that it is desired to cut 24 threads per inch. We have then,

$$\frac{6}{24} = \text{ratio.}$$

By dividing up the numerator and denominator in factors, and multiplying *each pair of factors by the same number*, we find the gears:

$$\frac{6}{24} = \frac{2 \times 3}{4 \times 6} = \frac{(2 \times 20) \times (3 \times 10)}{(4 \times 20) \times (6 \times 10)} = \frac{40 \times 30}{80 \times 60}$$

The last four numbers indicate the gears which should be used. The upper two, 40 and 30, are driving gears, the lower two, with 80 and 60 teeth, are driven gears. Driving gears are, of course, the gear *D*, Fig. 4, on the spindle stud, and the gear *P* on the intermediate stud *K*, meshing with the lead-screw gear. Driven gears are the lead-screw gear, *E*, and the gear *N* on the intermediate stud meshing with the spindle stud gear. It makes no difference which of the driving gears is placed on the spindle stud, or which of the driven is placed on the lead-screw.

Suppose, for a final example that we wish to cut 1¾ threads per inch on a lathe with a lead-screw having six threads per inch, and that the gears run from 24 and up to 100 teeth, increasing by 4. Proceeding as before, we have

$$\frac{6}{1\frac{3}{4}} = \frac{2 \times 3}{1 \times 1\frac{3}{4}} = \frac{(2 \times 36) \times (3 \times 16)}{(1 \times 36) \times (1\frac{3}{4} \times 16)} = \frac{72 \times 48}{36 \times 28}$$

This is the case directly illustrated in Fig. 4. The gear with 72 teeth is placed on the spindle stud *J*, the one with 48 on the intermediate stud *K*, meshing with the lead-screw gear. These two gears (72- and 48-teeth) are the ***driving*** gears. The gears with 36 and 28 teeth are placed on the lead-screw, and on the intermediate stud, as shown, and are the ***driven*** gears.

Fractional Threads.

Sometimes the lead of the thread is expressed by a fraction of an inch instead of stating the number of threads per inch. For instance, a thread may be required to be cut having a ⅝-inch lead. In such a case the expression "⅝-inch lead" should first be transformed to "number of threads per inch," after which we can proceed in the same way as has already been explained. To find how many threads per inch there is when the lead is stated, we simply find how many times the lead is contained in one inch, or, in other words, we divide *one* by the

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given lead. Thus *one* divided by ⅜ gives us 2 2/3, which is the number of threads per inch of a thread having ⅜-inch lead. To find change gears to cut such a thread we would proceed as follows:

Assume that the lead-screw has 6 threads per inch, and that the change gears run from 24 up to 100 teeth, increasing by 4. Proceeding to find the gears as before, we have:

$$\frac{6}{2\frac{2}{3}} = \frac{2 \times 3}{1 \times 2\frac{2}{3}} = \frac{(2 \times 36) \times (3 \times 24)}{(1 \times 36) \times (2\frac{2}{3} \times 24)} = \frac{72 \times 72}{36 \times 64}.$$

The rule for finding the number of threads per inch, when the lead is given, may be expressed by the formula:

$$\text{number of threads per inch} = \frac{1}{\text{lead of thread}}$$

which is simply a reversal of the formula given on page 4.

Rules for Selecting Change Gears.

What has been said in the foregoing in regard to the figuring of change gears for the lathe may be summed up in the following rules:

1. To find the number of threads per inch, if the lead of a thread is given, *divide one by the lead.*

2. To find the change gears used in simple gearing, when the number of threads per inch on the lead-screw, and the number of threads per inch to be cut are given, *place the number of threads on the lead-screw as numerator and the number of threads to be cut as denominator in a fraction, and multiply numerator and denominator with the same number until a new fraction results representing suitable number of teeth for the change gears.* In the new fraction, the numerator represents the number of teeth in the gear on the spindle stud, and the denominator, the number of teeth in the gear on the lead-screw.

3. To find the change gears used in compound gearing, *place the number of threads per inch on the lead-screw as numerator, and the number of threads per inch to be cut as denominator in a fraction, divide up both numerator and denominator in two factors each, and multiply each pair of factors* (one factor in the numerator and one in the denominator making "a pair") *by the same number, until new fractions result representing suitable number of teeth for the change gears.* The gears represented by the numbers in the new numerators are driving gears, and those in the denominators are driven gears.

Cutting Metric Threads with an English Lead-Screw.

It very often happens that screws or taps having threads cut according to the metric system are required. The lead of these screws is expressed in millimeters. Thus, instead of saying that a screw has so many threads per inch, it is said that the screw has so many millimeters lead. Suppose, for example, that we have a lathe having a lead-screw with 6 threads per inch, and that a screw with 3 millimeters lead is required to be cut. We can find the change gears to

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be used in the same manner as has been previously explained for screws cut according to the English system, if we only first find out *how many threads per inch we will have if we cut a screw with a certain lead given in millimeters.* Thus, in this case, we must find out how many threads there will be in one inch, if we cut a screw with a 3 millimeters lead. There are 25.4 millimeters to one inch, so that, if we find out how many times 3 is contained in 25.4, we evidently get the number of threads in one inch. To find out how many times 3 is contained in 25.4, we divide 25.4 by 3. Then we get as a result the number of threads per inch. It is not necessary to carry out the division; simply write it as a fraction in the form $\frac{25.4}{3}$, which implies that 25.4 is to be divided by 3. This fraction is the number of threads per inch to be cut. We now proceed exactly as if we had to do only with English threads. We place the number of the threads on the lead-screw in the lathe as the numerator in a fraction, and the number of threads to be cut, which number is expressed by the fraction $\frac{25.4}{3}$, as the denominator. Then we have

$$\frac{6}{\frac{25.4}{3}}$$

This seems very complicated, but as we remember that the line between the numerator and denominator in a fraction really means that we are to divide the numerator by the denominator, then if we carry out this division we get

$$6 \div \frac{25.4}{3} = \frac{6 \times 3}{25.4} = \frac{18}{25.4}$$

If we now proceed as in the case of figuring change gears for any number of threads per inch we multiply numerator and denominator by the *same* number, until we find suitable numbers of teeth for our gears. In the case above we can find by trial that the first number by which we can multiply 25.4 so that we get a whole number as the result is 5. Multiplying 25.4 by 5 gives us 127. This means that we must have one gear with 127 teeth, whenever we cut metric threads by means of an English lead-screw. The gear to mesh with the 127 teeth gear in this case has 90 teeth, because 5 times 18 equals 90.

If we summarize what we have just said in rules, we would express them as follows:

1. To find the number of threads per inch, when the lead is given in millimeters, *divide* 25.4 *by the number of millimeters in the given lead.*

2. To find the change gears for cutting metric threads with an English lead-screw, *place the number of threads per inch in the lead-screw multiplied by the number of millimeters in the lead of the thread to*

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be cut as the numerator of a fraction, and 25.4 as the denominator, and multiply numerator and denominator with 5. The numerator and denominator of the new fraction are the gears to be used. **These same rules expressed in formulas would be**

$$\text{number of threads per inch} = \frac{25.4}{\text{lead in millimeters}}$$

and

$$\frac{\text{number of threads per inch in lead-screw} \times \text{lead in millimeters of screw to be cut} \times 5}{25.4 \times 5} = \frac{\text{gear on spindle stud}}{\text{gear on lead-screw}}$$

Of course, it is sometimes necessary to compound the gears, because the gear on the spindle stud would otherwise get too many teeth, that is, it would be too large. Suppose, for an example, that we wish to cut a screw having 6 millimeters lead on a lathe having a lead-screw with 8 threads per inch. According to our rule and formula the gear

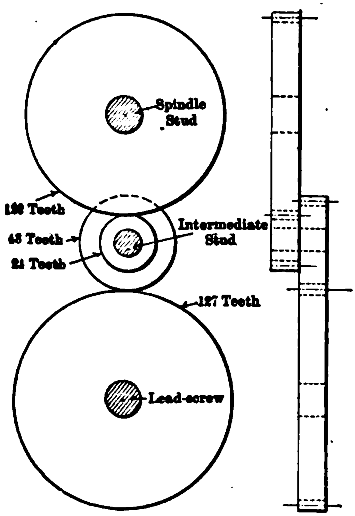

Fig. 5. Example of Gearing for Cutting Metric Thread with English Lead-Screw.

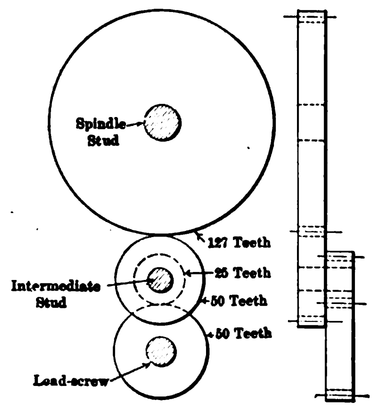

Fig. 6. Example of Gearing for Cutting English Thread with Metric Lead-Screw.

on the spindle stud would then have 8 × 6 × 5, or 240 teeth. As no lathe is provided with a change gear with so many teeth, we must use compound gearing. In this case we would proceed as follows:

$$\frac{8 \times 6 \times 5}{25.4 \times 5} = \frac{48 \times 5}{127 \times 1} = \frac{48 \times 120}{127 \times 24},$$

which is exactly the same method as has already been explained under the head of compound gearing in connection with the figuring of change gears for English screws. The method of mounting these gears is shown in the diagram, Fig. 5.

What should in particular be impressed upon the minds of the reader is that there is *no difference in method* of figuring the gears whether the thread to be cut is given in the English or in the metric system. If given in the latter system, simply transform the "lead in

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millimeters" to "number of threads per inch" and proceed in exactly the same way as if the thread had been given according to the English system.

The 127-teeth gear is always placed on the lead-screw when cutting metric threads with an English lead-screw.

Cutting an English Thread with a Metric Lead-Screw.

The method of figuring the change gears when an English screw is to be cut with a metric lead-screw is simply the reverse of the one already explained. We simply transform the millimeter lead of the metric lead-screw into "number of threads per inch." This we do in the same way as explained before, we divide 25.4 (which is the number of millimeters in one inch) by the number of millimeters in the lead of the metric lead-screw. After having obtained this number of threads per inch, we proceed as usual, putting the number of threads per inch of the lead-screw in the numerator, and the number of threads per inch to be cut in the denominator of a fraction, simplifying the fraction, and multiplying both numerator and denominator by 5 to get the number of teeth in the change gears.

Suppose, for example, that we wish to cut 5 threads per inch with a lead-screw having 4 millimeters lead. The number of threads per inch of the lead-screw is, then, $\frac{25.4}{4}$, and we find our gears by writing our fraction,

$$\frac{\frac{25.4}{4}}{5}$$

This fraction can be simplified by actually dividing $\frac{25.4}{4}$ by 5, in which case we get

$$\frac{25.4}{5 \times 4} \text{ as a result.}$$

Multiplying both numerator and denominator by 5 gives us then,

$$\frac{25.4 \times 5}{5 \times 4 \times 5} = \frac{127}{100}$$

which gives us the number of teeth in our change gears.

The formula expressing this calculation would take this form:

$$\frac{25.4 \times 5}{\text{number of threads per inch to be cut} \times \text{lead in millimeters of lead-screw} \times 5} = \frac{\text{gear on spindle stud}}{\text{gear on lead-screw}}$$

Expressed as a rule this formula would read:

To find the change gears for cutting English threads with a metric lead-screw, *place* 25.4 *as the numerator, and the threads per inch to be cut multiplied by the number of millimeters in the thread of the lead-screw in the denominator of a fraction, and multiply numerator and denominator by* 5. *The numerator and denominator of the new fraction are the change gears to be used.*

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In this case too, of course, it sometimes becomes necessary to compound the gears, in order to get gears which are to be found in the set of gears provided with the lathe. Sometimes, the gears may be available, but they are so large that the capacity of the lathe does not permit them to be placed in a direct train; then, also, it becomes necessary to compound the gears. Take the case which we have already referred to, where we were to cut a screw with 5 threads per inch, using a lead-screw having 4 millimeters lead. We then obtained the gears with 127 and 100 teeth respectively. Now suppose that the lathe does not have a change gear with 100 teeth to be placed in a direct train. The gears to be used in a compound train would then have to be found as has already been described, and, as shown in the following calculation:

$$\frac{25.4 \times 5}{5 \times 4 \times 5} = \frac{127}{100} = \frac{127 \times 1}{50 \times 2} = \frac{127 \times 25}{50 \times 50}$$

The 127-teeth gear is always put on the spindle stud when cutting English screws with a metric lead-screw. A diagram of the arrangement of the gears in the last example is shown in Fig. 6.

If there is any special reduction gearing in the head of the lathe, this must of course be taken in consideration, in a manner as has already been described under the heading "Lathes with Reduction Gearing in Head-stock," on page 7.

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CHAPTER II.

KINKS AND SUGGESTIONS IN THREAD CUTTING.

In the following, a number of kinks and suggestions in thread cutting have been collected and presented. These suggestions have been made from time to time by the readers of MACHINERY, and the methods outlined are in use every day in some shop or other in the country. The names of the persons who originally contributed the descriptions of the suggestions given or the devices shown, to the columns of MACHINERY, have been given in notes at the foot of the pages, together with the month and year when the article appeared.

Indicator for Thread Cutting.

When cutting a thread in a lathe, if the number of threads to the inch being cut is a multiple of the number of threads to the inch on the lead-screw, the split nut may be thrown into mesh with the lead-screw at any time, and the tool will follow the first cut. This is not the case, however, when the number of threads to the inch being cut is not a multiple of the number of threads to the inch on the lead-screw. Because of this, lathes are generally equipped with a backing belt, which is thrown in when the tool has made the desired cut, and the carriage is brought back to the starting point without having been disengaged from the lead-screw, which of course, necessarily brings the tool into the right relation with the work. This is a good arrangement for short threads, say two or three inches in length, but when they are longer, and especially when they are large in diameter (which means slower speed) the backing belt is not a very economical contrivance, because considerable time is wasted while the carriage is being moved by the lead-screw from the end of the cut back to the starting point.

Fig. 7 shows a simple device which may be attached to any lathe, and used to good advantage when cutting threads. It can be fastened to the carriage as shown in the cut, and preferably on the side next to the tail-stock, as very often there is not enough thread on the lead-screw to permit putting it on the opposite side. This indicator is used in the following manner: Start the lathe, and when one of the three points marked *A* of the triangular pointer (see plan view), is opposite the zero mark, throw the split nut into mesh with the lead-screw. After the tool has reached the end of its cut, bring the carriage back by hand to the starting point. Wait until either of the points marked *A* is again opposite the zero mark, then throw the split nut into mesh with the lead-screw as before. If this is done with each successive cut, the tool will always come right with the thread. When the pointer is a triangle as shown, the worm-wheel, which is in mesh with the lead-screw, should be so proportioned that its number of

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teeth is three times the threads per inch of the lead-screw. If, for example, the lead-screw has eight threads per inch, then the worm-wheel should have twenty-four teeth. Then, when either of the points marked *A* is opposite the zero mark, the lead-screw and the lathe spindle would occupy the same relative positions. The device does not work for fractional threads.* This device, it is claimed, was originated in this country thirty or thirty-five years ago by William Gleason, of

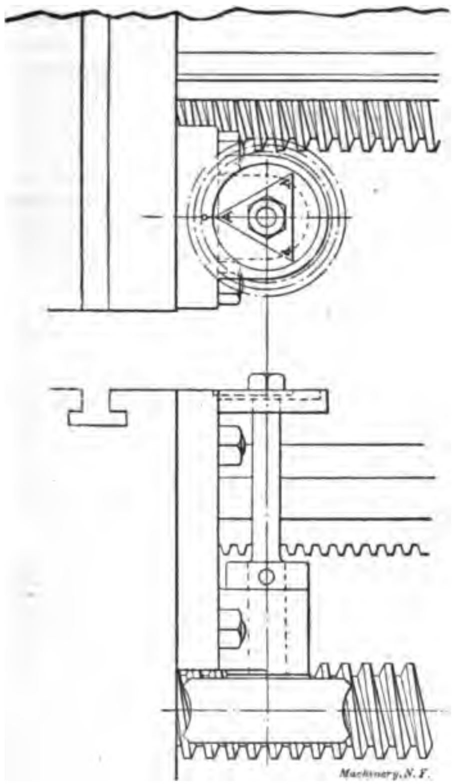

Fig. 7. Device Permitting Opening up the Lead-nut and Running Carriage back by Hand.

Rochester, N. Y. In fact, however, it is much older than that, having originally been invented in England.

Another Thread Catching Device for the Lathe.

The device shown in Fig. 8, which permits the lead-nut to be opened, and the carriage run back when cutting threads, still insuring "catching the thread," was applied to several lathes in the Worcester Polytechnic Institute some years ago by Mr. O. S. Walker of the same city. Mr. Walker states that he first saw it in the shops of the E. W. Bliss Co., Brooklyn, N. Y., nearly twenty-seven years ago.

* Franklin D. Jones, Brooklyn, N. Y., October, 1907.

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The lathes to which this device was applied had the lead-screw at the back, which explains the peculiar engraving. *A* is a casting bolted to the back of the carriage and supporting the split nut indicated by the dotted lines at *C C*. At the left of and supported by *A* is a vertical spindle carrying on its upper end the worm-wheel *E*, engaged with the lead-screw, and at the lower end the disk *D*. The worm-wheel should either have as many teeth as there are threads per inch in the lead-screw *S*, or a number of teeth which is some multiple of the number of threads per inch.

The disk *D* has equidistant slots milled across its periphery, the number of slots being as many as the number of teeth in the worm-wheel is times the number of threads per inch of the lead-screw. In this instance, the lead-screw has six threads per inch, the worm-wheel has thirty-six teeth, and there are six slots milled in the periphery of *D*. Fastened to the lower side of the lower half-nut is the latch *F*, which engages with one of the slots or notches in *D* when the split

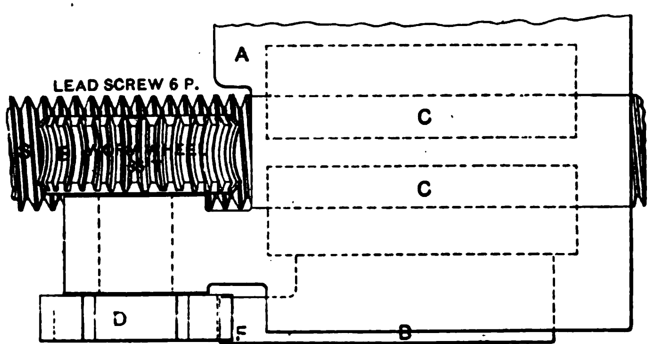

Fig. 8. Another Device of the Same Principle as that in Fig. 7.

nut is closed. It is thus evident that when the split nut has been disengaged from the lead-screw and the carriage run back for a fresh cut, the lead-screw cannot again be engaged until the worm-wheel turns into position for one of the slots to correspond with the latch *F*. The latch being engaged, the worm-wheel ceases to turn, acting then as a sort of half nut on the screw. Therefore the lead-screw can only be engaged at even inches of its length and necessarily the thread-cutting tool must engage with the thread already started when this has a whole number of threads per inch; but on fractional threads, the device fails as constructed. However, by having three slots in the disk *D*, instead of six, fractional threads having one-half for the fraction could be caught and with only one slot in the disk, fractional threads including one-half, one-third, one-sixth, two-thirds and five-sixths could be caught, but under these conditions the time required to bring the notch around so that the latch *F* could be engaged would usually be nearly equal to that required to reverse and run the carriage back, especially on short work.*

* H. P. Fairfield, Worcester, Mass., February, 1901.

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Combination Threading Tool.

Thread-tool holders have always been of considerable interest to toolmakers and others required to produce accurate threads. This interest is largely due to the difficulty of producing a thread-tool holder which fills all the requirements placed on such a tool.

Fig. 9 shows a combination spring and solid threading tool especially useful for working on tool steel. It is made high enough so as to rest on the carriage instead of on the rocker of the tool-post; therefore the tool is always parallel. The cutters are easily and quickly made and may be quickly changed in the holder. They may be easily set, by

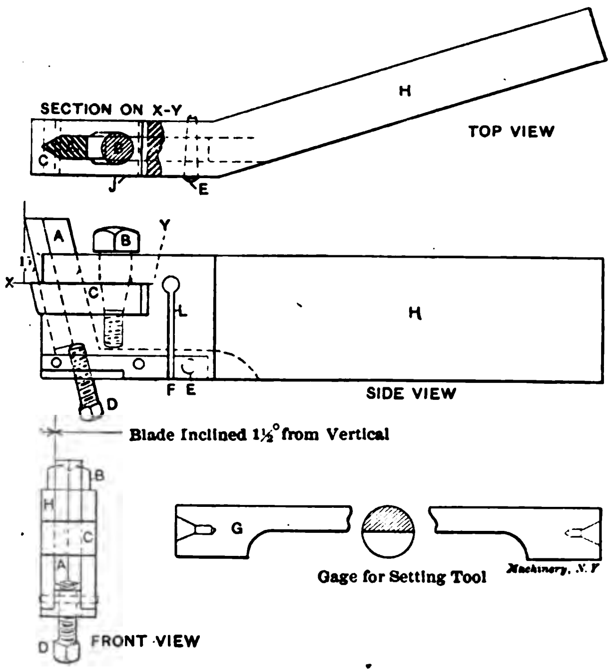

Fig. 9. Special Spring Threading Tool Holder.

bringing the face of the holder at *J* parallel to the face-plate and raising the cutter to the center line with the set-screw *D*. The gage shown at *G*, which consists of a smooth piece of round stock centered at the ends and milled out to the center line for a short distance, furnishes a convenient means for locating the top of the cutter at the proper height.

The construction of the tool will be readily understood. A slot is milled out on the front end of holder *H* at an angle of 15 degrees, to receive the blade *A*. This slot, as shown in the front view, also has an inclination of 1½ degree from the vertical to make the cutter agree with the average inclination of the thread in a U. S. standard screw. In a horizontal slot in the end of the holder is fitted the clamping

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yoke *C*. This has an opening in it through which the blade *A* passes, and is provided with a tapering seat for the tightening screw *B*. As this is screwed down, yoke *C* is drawn in, and the blade *A* is held firmly back against its seat. A saw cut at *F* extends nearly through the holder, thus leaving the upper end flexible to give it the effect of the well-known goose-neck tool. Through a slot in the bottom is passed a tie piece which is pinned fast to the outer division of the holder and may be, if so desired, connected with the shank by the taper pin *E*. This allows the tool to be used either as a solid or as a spring tool-holder.*

Another thread-tool holder is shown in Fig. 10. This was especially designed for the economical use of high-speed steel in thread cutting. One advantage of the holder is that cutters can be broken off from

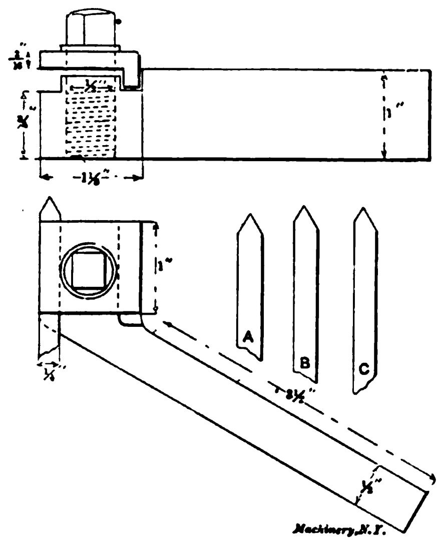

Fig. 10. Thread Tool Holder for High Speed Steel Tools.

the bar and used without further working. By grinding the cutters as indicated at *A*, *B*, and *C*, a variety of pitches can be cut close up to a shoulder.* *

Spring Holder for Threading Tools.

The thread-tool holder shown in Fig. 11 is intended for the blades or single-point cutters made by the Pratt & Whitney Co. The improvement in the design over common holders consists in the provision for permitting the tool to spring away from the work if too heavy a cut is taken. In other respects the principle of the holder is the same as that of the one manufactured by the Pratt & Whitney Co., itself, for these tools. Referring to the engraving, *A* is the body, which is slotted at *B*, proper resistance being given the tool by the set-screw *C* which has a spring at the lower end, acting upon the front part of the holder.

* Everett Kneen, Kearny, N. J., August, 1906.

* * Stephen Courter, Paterson, N. J., August. 1908.

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At *D* may be inserted a blade or key, which will keep the front part of the holder from bending to one side while cutting.

A great many designs of spring tool-holders have been tried, the one shown in Fig. 11 being comparatively common. The difficulty with holders of this kind is that it is almost impossible to adjust the screw for each particular pitch to be threaded so that the spring has the proper tension. It is evident that in cutting a coarse thread there is no need of the tool being as sensitive as when cutting a very fine thread, but there is no means for judging when in each particular case the proper springing action has been attained. Another objection to the design shown below is that it prevents a full and clear view of the thread being cut, the projecting part extending partly above the work. Of all spring thread-tool holders hitherto designed, however, this one is about as good as any. A spring tool-holder for threading

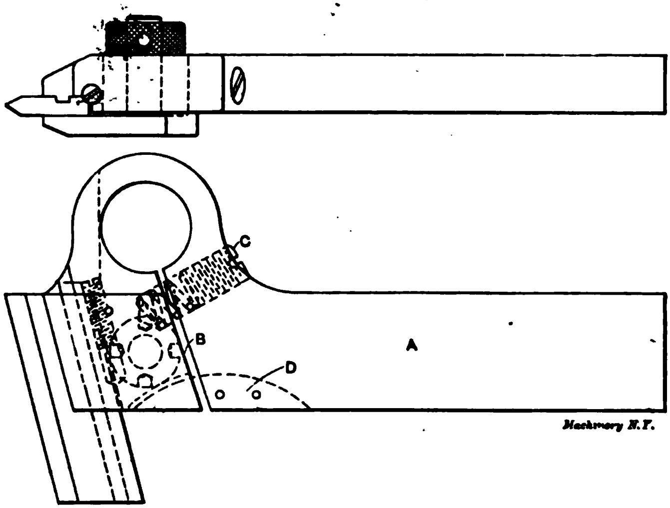

Fig. 11. Spring Holder for Threading Tools.

tools which will overcome the objections mentioned is greatly in demand, and many attempts have been made to solve the problem, but as yet none has been entirely successful.

Tool for Cutting Square Screw Threads.

In Fig. 12 is shown a tool of the chaser type for cutting square screw threads. This tool has been recently patented by Messrs. C. & G. B. Taylor, Bartholomew St., Birmingham, England. Ordinarily, square screw-thread tools, even when they have been used very little, are found to have worn to such an extent that the resulting groove is not as wide as required. It is obvious that it is impossible to regrind these tools after the sides of the cutting teeth have worn down below the required width. With the hope of overcoming this defect, the tool shown in the cut has been designed. As seen, the tool consists of two halves, *A* and *B*, each being provided with teeth which gradu-

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ally cut the groove to the required depth. The required width is obtained by adjusting the relative position of th two tools *A* and *B*, so that the tool *B* widens the grooves already cut by *A*. These two tools or chasers are held in a tool-holder *E*, and the adjustment is

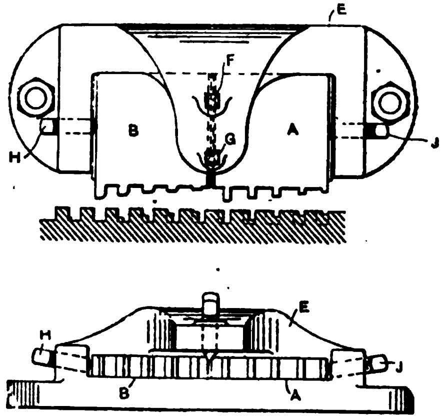

Fig. 12. Adjustable Tool for Cutting Square Threads.

effected by means of two screws *F* and *G* having conical ends, which are forced in between the tools *A* and *B*, these in turn being clamped by the screws *H* and *J*. Whether the tool will prove to possess such practical qualities as will insure for it any extensive application is

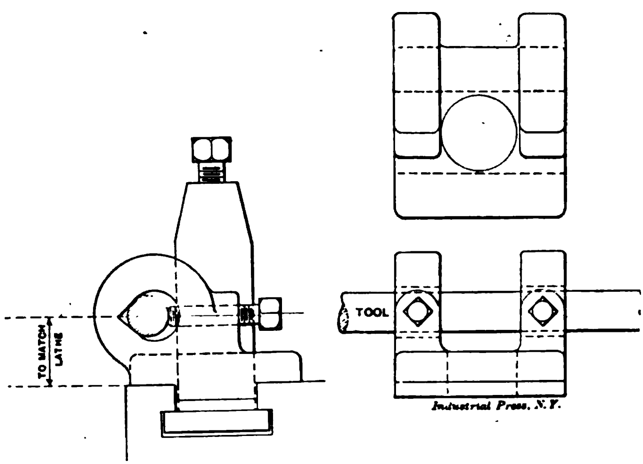

Fig 13. Internal Thread-Tool Holder.

difficult to say, but the idea is ingenious, and may be applied in other cases than that of cutting square screw threads.

Internal Threading Tool Holder.

The lathe boring and threading tool holder shown in Fig. 13 has been in use in a well-known Ohio shop, and has given very good satis-

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faction. It is a plain iron casting, tongued and fitted to the tool-post slide of the lathe in which it is intended to be used, and is clamped in position by inserting a piece of steel in the tool-post and secured as an ordinary tool would be clamped. The threading tool is clamped by two set-screws, and the heart-shaped holes for the tool not only accommodate different sizes of tools, but insure rigidity.

Cutting Triple Threads.

Fig. 14 illustrates an economical device for cutting triple threads. The frame *A* is bolted to the lathe carriage, and it is of such height that the center *B* is in line with the lathe centers. The cutters *C* are held in slides *D*, which slide in grooves planed in the circular part of the frame *A*. These slides are held in place by a circular plate *E*, which also serves to move the slides in and out, through the medium of the cam slots *F* acting on pins in the slides. Plate *E* is held in

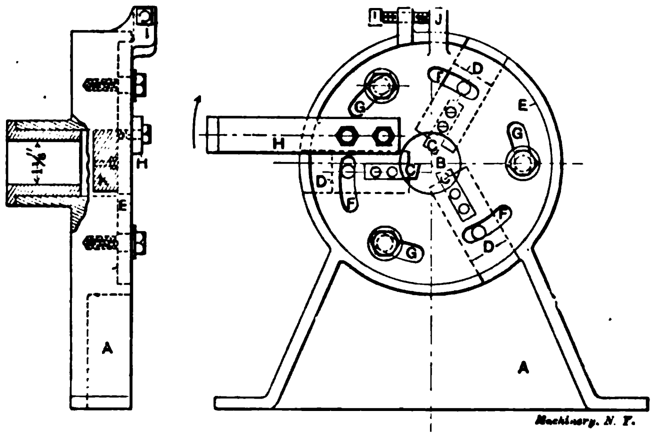

Fig. 14. Device for Cutting Triple Threads.

place by bolts in the slots *G*. A handle *H* serves to rotate plate *E*. A set-screw *I* in a lug cast on frame *A* bears against a lug *J* on plate *E* and acts as a gage to vary the depth of cut.

The tools or cutters *C* are set in the slides *D* so that the top surface is in line with the center *B*. They are fastened to the slides by screws as shown in section at *K*, and they are all set in the same plane so that each one cuts a different thread. At the head-stock end of the lathe a slide is bolted, which engages with arm *H* and throws it up in the direction of the arrow thus drawing the cutters back at the end of the cut. A removable bushing provides adjustment for cutting threads on rods of various sizes. With this device four feet of triple square thread 1⅛ inch diameter and ¾ inch lead can be cut in twenty minutes.*

Face-Plates for Cutting Multiple Threads.

The following method of cutting multiple-threaded screws of two, three, four, five, etc., threads is very simple and mechanically perfect.

* Fred Seaburg, Chicago, Ill., December, 1907.

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A plain circular plate, Fig. 15, to bolt on the face-plate of the lathe, is made, and located and held in exact position with two small dowel-pins, *AA*. Then a number of circles are scored in this plate. The circle nearest the center is divided into three parts, the next one into four parts, which answers for two divisions as well, the next one into five parts, etc. Holes are drilled in the circles large enough in diameter to hold pins for driving the carrier. To use the plate for cutting

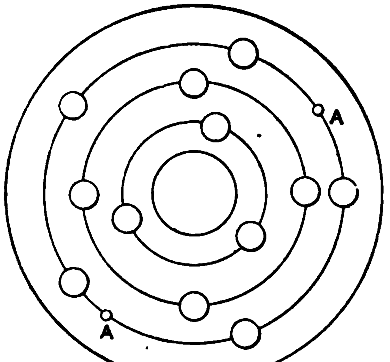

Fig. 15. Device for Facilitating the Cutting of Multiple Threads.

multiple threads, the carrier is moved from one pin to another until each thread is cut. The plate mentioned is kept specially for multiple thread cutting, and can be bolted on by bolts through the back of the face-plate.*

Fig. 16 shows another interesting development of face-plate arrangement for threading lathes, brought out by the firm of Ferdinand Pless, Fechenheim, a. M., Germany, and intended for facilitating the cutting

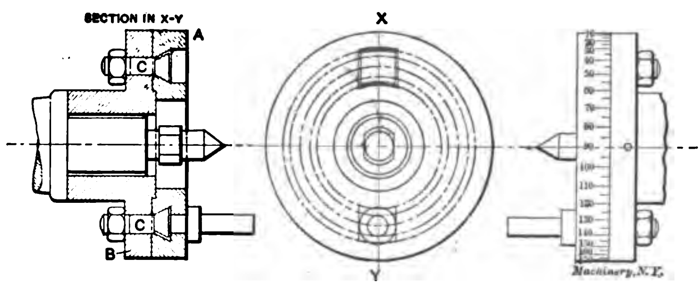

Fig. 16. Another Device for Cutting Multiple Threads

of multiple threads in the lathe. As seen from the illustration, it consists of two parts, *A* and *B*, the part *A* being free to be rotated in relation to the part *B* when the bolts *C* are loosened. The driving pin for the lathe dog is attached to the plate *A*, and in cutting multiple threads, when one thread is finished, the bolts *C* are simply loosened, and the plate *A* turned around in relation to the spindle of the machine an amount corresponding to the type of thread being cut; thus, for

* James H. Gomersall, Germantown, Pa., March, 1900.

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instance, if a double thread is cut, the plate *A* is turned around one-half revolution, or 180 degrees; for triple thread, 120 degrees; for a quadruple thread, 90 degrees, etc. The periphery of plate *A* is graduated in degrees, and a zero line provided on plate *B*, so that the required setting is very easily obtained. On lathes which are con-

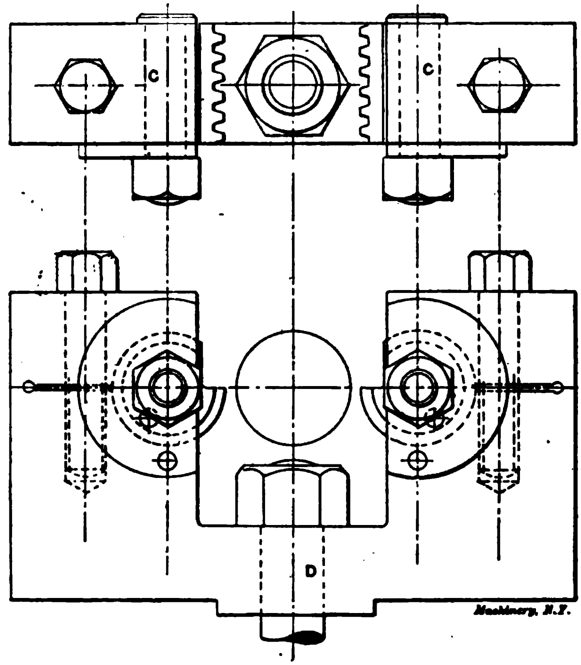

Fig. 17. Double Tool-Post with Circular Formed Thread-Tool.

stantly used for thread cutting the advantage of an arrangement of this type is very evident, as it saves employing any of the more or

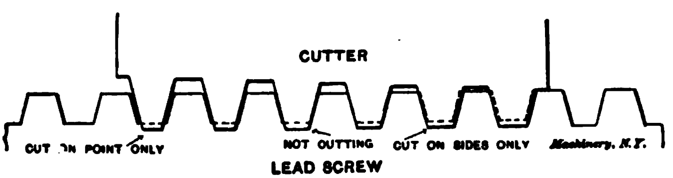

Fig. 18. Cutting Action of the Tools.

less cumbersome methods in vogue for moving the work in relation to the tool when cutting multiple threads.

Method for Cutting Lathe Lead-Screws.

The method shown in Fig. 17 may be used for cutting lathe lead screws. Two cutting tools are used, one in front, right side up, and the other at the back, also right side up, to cut on the reverse trip. The cutting tcols are round, like short sections of the screw to be cut,

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but left-hand to cut a right-hand screw. They are cut with the thread on a taper and the outside turned straight so that the leading cutter tooth will cut to the full depth fed at each traverse, and the succeeding teeth widen the cut, only the last two usually cutting on the full side of the thread, as shown in Fig. 18. The limiting element in using this device is the torsional strength of the screw being cut. The bolts *CC* and their washers and nuts help to hold the cutters in place. Bolt *D* holds the device to the top of the cross-slide, in place of the tool-post.*

Accurate Threading of Taps and Die Hobs.

Experience in tap and die making has taught that it is one thing to make a perfect screw and quite another to make a tap which will perfectly correspond with it. It is well known that a tap shortens in hardening, this shrinkage varying somewhat with different grades of steel, so that a tap and a screw made with the same lead-screw will

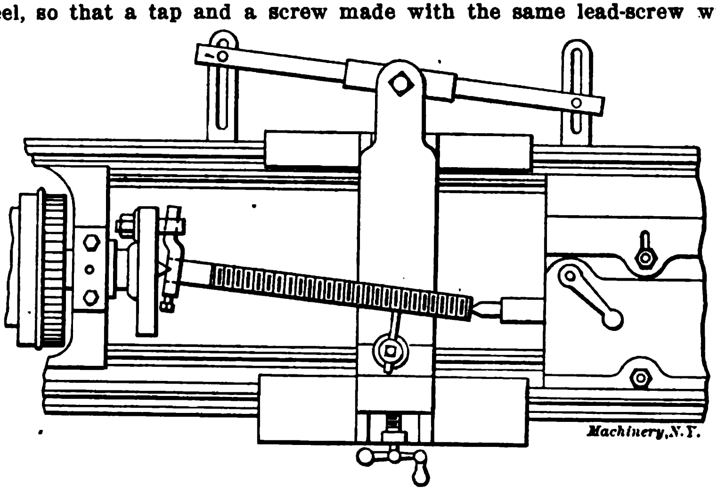

Fig 19. Arranging Lathe for Cutting Taps Long in the Lead.

not correspond in pitch. Therefore, to be accurate, allowance must be made in chasing the thread on the tap for the shrinkage that will take place in the hardening.

To carry out this idea a little further: A hob and a tap may thus be made to correspond, but after a die is hobbed and hardened it will not exactly match the hob or the tap which it is intended to suit. So we see that the hob should be made with an allowance of two shrinkages to counteract the shortening that takes place when it (the hob) is hardened and again when the die is hardened. While it is not the best policy to make taps and dies in the tool-room, it often becomes necessary to do so for sizes varying from the standard, and the problem then presents itself to cut them so as to make the proper allowance for shrinkage. The following method shows how this is accomplished in a very satisfactory manner:

The change gears of the lathe are first arranged as usual for cutting

* E. H. Fish, Worcester, Mass., October. 1906.

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the required number of threads per inch. The tail-stock is then set over either way for a short distance and the taper attachment is set to correspond with the set-over of the tap. The thread is then cut as usual, the tool being set by the face of the tap. The thread cut in this way will be slightly coarser than would have been the case if the centers were in line with the axis of the lathe. The reason for this will be obvious from the cut, Fig. 19, and a little practice will enable the machinist to judge just how much the tailstock should be set over to obtain the required result. For example: If the tap is 10

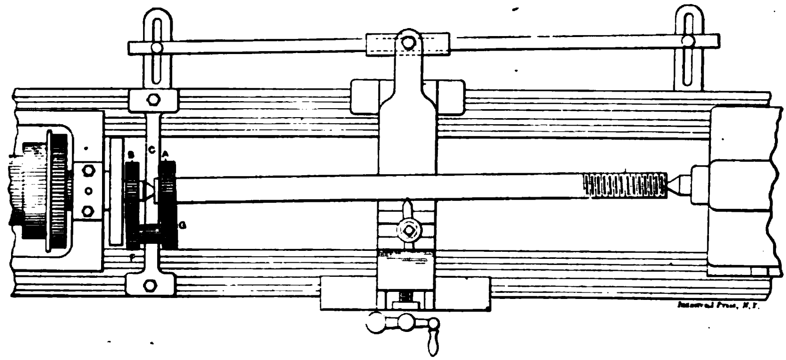

Fig. 20. Another Arrangement for Cutting Taps Long in the Lead.

inches long and is set over ½ inch, while the tool is moving the whole length of the tap, the movement of the carriage parallel to the axis of the lathe would be through a distance of but 9.987 inches, so that the thread is lengthened 0.013 inch; in other words, there are as many threads in 10 inches on the tap as would have been cut in 9.987 inches had the centers been in line with the shears.

If now in cutting this tap we were to use an ordinary dog, driven from the faceplate, it will be apparent that the result would be a

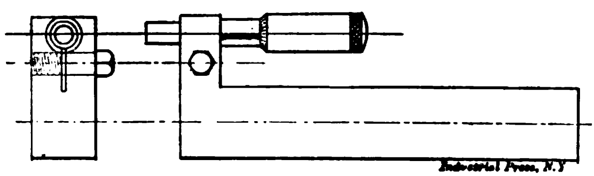

Fig. 21. Micrometer for Testing the Lead of Lead-Screws.

drunken thread, since the velocity of revolution of the tap would not be constant throughout its revolution, but would be variable; therefore a dog as shown in Fig. 19 is required to transmit a uniform motion from the spindle to the tap. One could also employ a small gear *A*, Fig. 20, mounted upon the end of the tap, and a similar gear *B* attached to the live center or to the face-plate. A fixture *C* is used for supporting a short shaft on each end of which is placed a pinion, as shown at *F* and *G*. These pinions are of the same size and engage with the gears *A* and *B*. When they are properly adjusted so as to run freely

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with these gears, they impart a perfectly uniform movement to the tap being chased.

Testing a Lead-Screw.

A method of testing the pitch of a lead-screw, at any position of its length, consists in procuring a micrometer screw and barrel complete, such as can be purchased from any of the manufacturers of accurate measuring instruments, and bore out a holder so that the axis of the micrometer screw will be parallel to its body when the screw is in place, as shown in Fig. 21. With the lathe geared for any selected pitch, the nut engaged with the lead-screw, and all backlash of screw, gears, etc., properly taken up, clamp the micrometer holder to the lathe bed, as shown in Fig. 22, so that the body of the holder is parallel to the carriage. Adjust the micrometer to one inch when the point of the screw bears against the carriage and with a surface gage scribe a line on the outer edge of the face-plate. Now rotate the lathe spindle any number of full revolutions that are required to cause the carriage

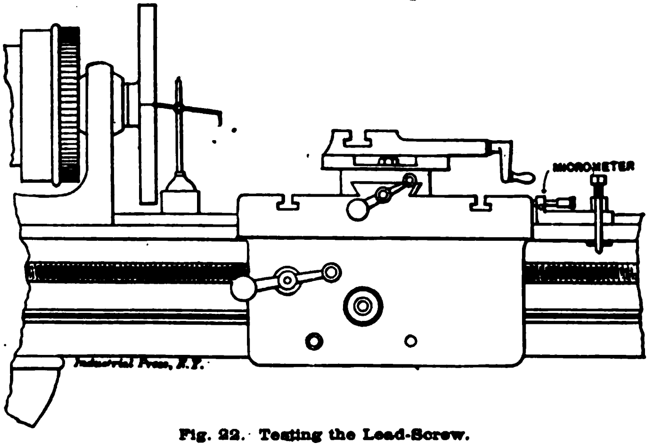

Fig. 22. Testing the Lead-Screw.

to travel over the portion of the lead-screw that is being tested, bringing the line on the face-plate to the surface gage point. If the distance traveled by the carriage is not greater than one inch, the micrometer will indicate the error directly. For lengths of carriage travel greater than one inch an end measuring rod, set to the number of even inches required, can be used between the micrometer point and the lathe carriage. The error in the lead-screw is then easily determined by the adjustment that may be required to make a contact for the measuring points between the carriage and the micrometer screw. The pitch can be tested at as many points as are considered necessary by using end measuring rods, of lengths selected, set to good vernier calipers. The style of holder shown can, with the micrometer screw, be used for numerous other shop tests and as the screw is only held by friction caused by the clamping screw, it can easily be removed and placed in any form of holder that is found necessary.*

* Walter Cantelo, Bridgeport, Conn., July, 1903.

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Cutting a Smooth Thread.

When cutting threads, one often meets with difficulty in obtaining a smooth thread, such as is required for screw gages and taps. One good way to obtain a smooth thread is to turn the tap nearly to size and harden it; then draw the temper to a "light blue." When turning to size, if the tool does not stand up well, draw still lower, the object being to leave just enough temper in the tap to make the steel firm. By taking light chips with a hard thread tool, a glossy, smooth thread will result. Another advantage gained by hardening the tap before finishing is that it will greatly eliminate the chances of the lead changing after the final hardening. A thin lubricant of lard oil and turpentine is an excellent one for thread cutting. When cutting two or more taps it is customary in some shops to rough out both or all the taps, leaving the dogs on them, and for the sizing or finishing cut the taps are chased without moving the thread tool. But if the thread tool dulls a trifle when making the finishing cut on the first tap, the succeeding taps will not be exactly the same size.*

Removing Broken Taps.

To remove a broken tap from cast iron, the hole should first be thoroughly cleaned out by means of a small squirt gun filled with

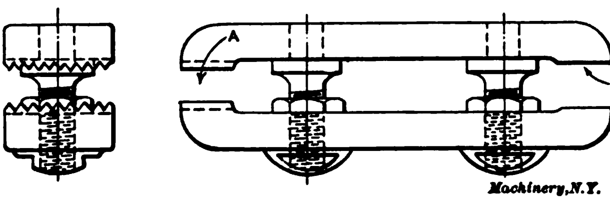

Fig. 23. A Limit Screw Thread Gage

kerosene. All small broken pieces of the tap can be removed with a pair of tweezers. Then the tweezers, which should be as large as possible, should be inserted between the hole and the flutes of the tap and by slowly working back and forth and occasionally blowing out with kerosene, the broken piece is easily released. A through hole, of course, simplifies matters somewhat.**

A Handy Screw Thread Gage.

When cutting threads on screws and bolts, whether by threading dies or in a lathe, much time is wasted by gaging the threads with either a nut or a ring thread gage of the ordinary type. In the case of a piece held between lathe centers, in order to gage the thread with the ring gage, it is necessary to remove the piece from between the centers. The Dresdner Bohrmaschinenfabrik A.-G., Dresden, Germany, is making a gage for measuring the threads of screws, which serves the same purpose as a ring gage, but saves the user considerable time. This gage is shown in Fig. 23. The end marked *A* fits over the threads, and the end marked *C* is supposed not to pass over the threaded screw, when

* F. E. Shailor, Great Barrington, Mass., March, 1907.
** H. J. Bachmann, New York City, January, 1906.

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threaded to the right size. Thus, not only can the size of the threads be tried, but at the same time the gage acts as a limit gage.

Method of Driving Lathe When Cutting Screws of Steep Pitch.

When cutting screws of very quick pitch, or cutting the teeth of spiral gears in a lathe, place a pulley on the lead-screw and lengthen the cone belt so as to drive the lead-screw directly from the countershaft, and drive the spindle back through the change gears. By doing this, the carriage may be driven back and forth much quicker and with less strain on the lathe. When cutting a quick pitch, more power is generally required to operate the carriage than to drive the spindle.*

Case Hardening Ring Thread Gages.

To harden ring thread gages without distortion, anneal the gage after roughing out, and when finished cutting the thread, fill it with powdered cyanide and then heat it in a gas furnace, being very careful to exclude cold drafts as much as possible. When the heat has reached the right temperature, turn the gas almost off, and let the piece remain

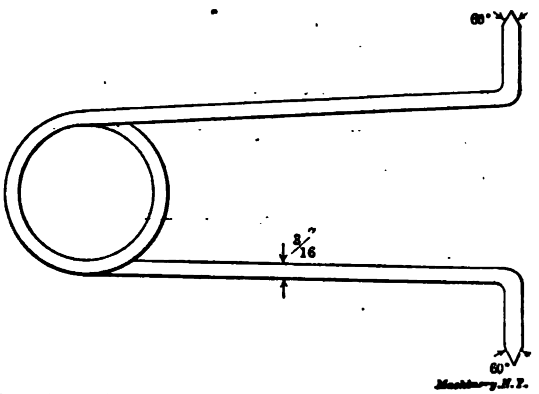

Fig. 24. Tool for Cleaning Threads in Chucks and Face-plates

in the furnace for about ten minutes. Then dip it in oil and keep it moving around in a path shaped like the figure 8. When cool enough, remove it and clean it with kerosene oil.**

Thread Cleaner for Chucks and Face-plates.

The practice of cleaning out the threads on chuck and other face-plates every time they are screwed on the spindle is very necessary to maintain the accuracy of the chuck and should therefore not be neglected, especially by apprentices. The only instrument necessary is a piece of 3/16-inch drill rod bent into the shape of a safety pin and having its two ends bent outward with the points filed to 60 degrees, as shown in Fig. 24. Inserting this little tool between the threads and moving it around by hand insures the removal of all dirt and chips that have accumulated in the threads. In this connection it is also well to remember that after removing a chuck or face-plate from the spindle, it should be laid away face down or with the chuck jaws rest-

* Eugene Wopaletzky, Trenton, N. J., November, 1904.
** Everett Kneen, Kearny, N. J., July, 1906.

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ing on the bench or floor, thus keeping the chips away from the thread as much as possible.*

A Compound Gearing Arrangement.

A certain lathe, which was single geared, was required to have its thread-cutting capacity increased. Compound gearing was therefore arranged for as shown in Fig. 25.

To the left is shown the original gearing on the lathe, with which from 3 to 48 threads per inch can be cut. The key from the end of the lead-screw was removed, and in place of the gear *D*, the extension *A* was screwed on. A keyway was cut in this extension, so that the long key *B* extended from the end of the shoulder at *C* through *A* and into the keyway in the end of the lead-screw. It thus locked the extension onto the lead-screw and also provided a key for the gear *D*. The notch in the key is to facilitate its removal.

In the place of the original intermediate stud, the stud in the center of the engraving was substituted. On this stud the gears were sep-

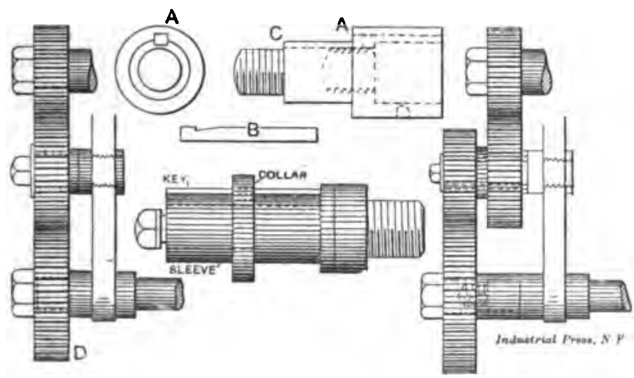

Fig. 25. A Compound Gearing Arrangement.

arated by the collar, while a key, running the entire length of the sleeve, made the gears run together. The complete arrangement is shown to the right in the illustration. By making this alteration it is possible with the same lathe to cut from 1 to 144 threads per inch.**

Thread-Rolling Dies for Small Interchangeable Screws.

Fig. 26 shows a thread-rolling device as applied to a punch press. *A* is a punch holder to fit the punch press. *B* is the bolster, or a piece of cast iron about 1 inch thick, upon which are located two cast iron blocks, one made stationary and the other adjustable by slotting *B*, so that the block can be forced ahead by the set-screw *C*. There is a groove in the stationary block and a tongue in the punch holder *A* to prevent the dies from getting out of line. The screw *D* is for holding a thin piece of steel as a stop so that the thread can be cut to the desired length. The screw *E* holds a wire supporting the piece to be

* H. J. Bachmann, New York City, January, 1906.
** James P. Hayes, Meriden, Conn., September, 1902.

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threaded until the upper die, *F*, comes down and carries it past the lower die *G*. In cutting the die, it may be made in one piece, *H* being the circumference of the thread to be rolled and G_1 the desired length for the lower die. F_1 is the desired length for the upper die, which must be longer than the lower die so that it will roll the wire past the die *G* and permit it to drop out of the way. The part *K* must be cut out when cutting in two parts. The proper angle to which to cut the die depends on the pitch of the thread. The pitch divided by the circumference of the screw to be rolled will give the tangent of the angle. In cutting the die, which must be of good tool steel and hardened after making, the shaper is used. The cut is taken with a tool that can be taken off and put back again without changing its location, such a tool, for instance, as a circular threading tool. In case the point should

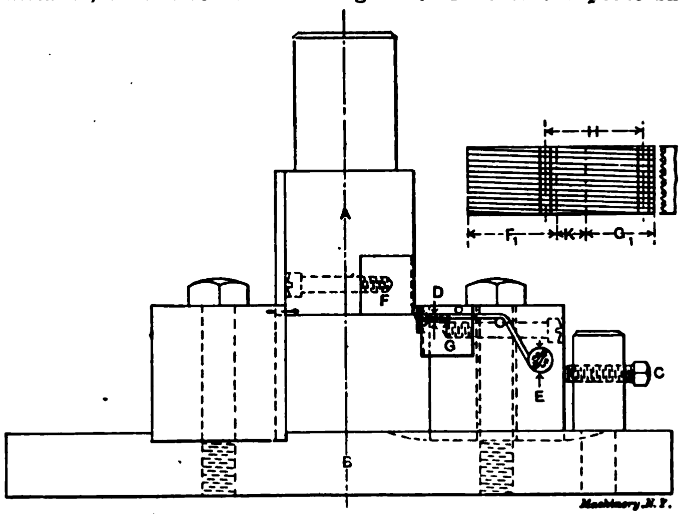

Fig. 26. Thread Rolling Device.

happen to get dull, the tool can then be removed for grinding. If the feed screw has not got the desired graduations on it, a brass index plate can be made very quickly, and used on the machine. The brass plate should be of a good size and cut accurately in a milling machine, and a pointer clamped on the shaper.*

Producing Threads by a Rapidly Revolving Steel Disk.

Fig. 27 illustrates a method used for threading studs, pins, etc., of manganese steel, this material being so hard that it cannot be cut by any kind of tool steel. A plain, hardened tool steel disk, having the edge made according to the angle of thread, is employed. This disk is revolved at a high speed, and at the same time forced into the work, which is revolved slowly. Due to the friction between the edge of the disk and the work, and the softening of the material, owing to the heat generated by the friction, the disk wears away the stock, and, by means

* Stacy Oliver, Great Barrington, Mass., July, 1907.

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of this, creates the thread. The stock is coming off in very small, thin scales like chips, which to some extent remind one of the scales of a fish. An ordinary lathe may be rigged up for the purpose, by removing the tool-post and top-rest, and substituting for them the fixture shown in the cut. The disk must be driven independently by an overhead drum, or some similar arrangement.

The peripheral speed of the disk is usually between 3,000 and 4,000 feet per minute. The operation is unavoidably slow and expensive, and the method is used only when no other way is possible. It is very likely, however, that the efficiency can be increased to some extent by increasing the peripheral speed of the disk, perhaps as high as 24,000 feet per minute, same as used on friction saws.

It is likely that high-speed steel would be preferable to ordinary tool steel as material for the disks, but, as the process described is necessarily slow, and is used only when no other way of threading is pos-

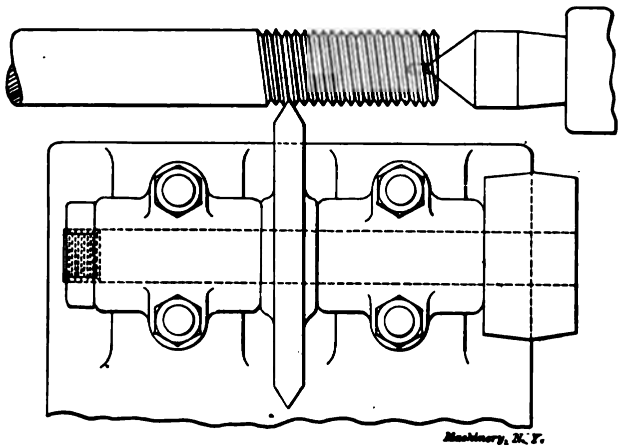

Fig. 27. Cutting Thread by a Rapidly Revolving Hardened Steel Disk.

sible, it has not as yet been developed to the limit of its capacity. There is a certain point in the gradual development of the method above which it becomes economically preferable to employ high-speed steel for the disk, but below this point of development, although high-speed steel may be the best, the ordinary tool steel disk, owing to its smaller first cost, is economically the one to be preferred. A preference for the one or the other kind of steel is influenced by a number of factors, *viz.*, the number of pieces to be threaded per unit of time; the peripheral speed of the disk; the pressure between disk and work; and the efficiency of the system of cooling.

The question of cooling is in itself an interesting one. The reason why the heat does not draw the temper of the tool steel in the disk while the heat is so great that it softens the metal of the work, is that the disk is revolving at a high speed and the work only revolving very slowly, so that a unit of length of the periphery of the disk is in contact with the work but a very short time, while every point on the

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work, at the place where it is cut, is in contact with the disk a comparatively long time. Owing to this the disk has ample time to cool off, while the work accumulates the generated heat. The high speed of the disk also throws the film of air nearest to the disk outward, owing to the centrifugal force, and new cool air comes constantly in at the center, a current of air thus at all times tending to cool the disk.

The cooling thus obtained is found to be satisfactory at the present speed at which the disk is run, but at a higher speed a system of cooling by an air jet, or still better, perhaps, by water, could be employed

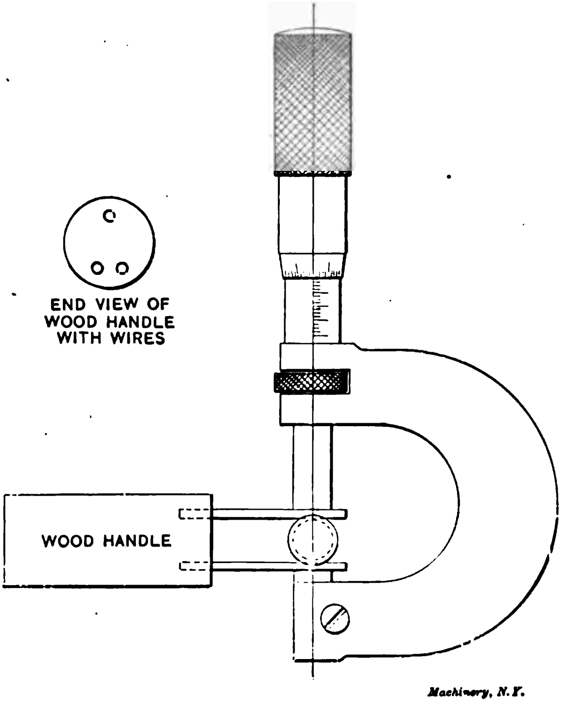

Fig. 28. Holder for Wires when Measuring Taps by the Three-wire System.

to advantage. This would also increase the limits within which an ordinary tool steel disk could be used to advantage. For increasing the peripheral speed of the disk as previously mentioned, undoubtedly the best way would be to increase the diameter of the disk, permitting the number of revolutions to remain the same as before; but at the present stage of the development of this device there are some limitations to the size of the disk, inasmuch as it is used in an ordinary lathe, and the space possible to utilize for the disk is not very great. Another difficulty in increasing the diameter, rather than the number of revolutions, is that for a large diameter disk it is necessary to arrange the disk on an inclined angle in relation to the work in order to

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get a perfect thread, and this necessarily means a more expensive rigging.

The principle involved in this method of cutting threads is the same as that involved in the friction saw. But the principle of the latter machine cannot be carried out to its full extent in the present case, because the steel to be threaded must not be heated more than to a certain degree. Above this limit, increased heating would mean injury to the quality of the steel. The heat also must not be so high that it burns the thread.

If the call for threaded parts of this kind of steel would be great enough so that a special machine would be warranted to be built, then the efficiency of the method could be increased by a careful taking care of all the points previously referred to, but, at the present time, the demand is not large enough to warrant the expenditure of building such a machine.*

To Measure a Thread with Micrometer and Three Wires.

A United States standard or Sellers thread may be accurately measured with a micrometer by the aid of three wires, preferably Stubbs steel. Select a diameter of wire that will lie in the thread nicely and project above the tops of the thread. Use two wires on one side in adjacent V's and one wire on the opposite side, so as to be in the middle plane between them. For convenience these wires may be sharpened and stuck into a block of soft wood, two on one side of the screw and one on the other, as shown in Fig. 28. Then, having gotten the diameter as measured over the top of the wires, the thread diameter may be obtained by the following rule: From the diameter as measured over the wires, subtract three times the diameter of the wires. To the remainder add the quotient obtained by dividing 1.5155 by the number of threads per inch. Suppose that a screw with 14 threads per inch is measured in this manner, using 0.053 inch diameter wire, and the diameter over wires is 0.5507 inch. Subtracting 3 $\times$ 0.053, or 0.159 from 0.5507, leaves 0.3917. To the remainder add $\frac{1.5155}{14} = 0.1082$, giving the diameter as 0.4999, or ½ inch.**

* Oskar Kylin, High Bridge, N. J., January, 1908.
** Ernest Kroff, Cincinnati, Ohio, January, 1906.

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CHAPTER III.

TABLES AND FORMULAS FOR MAKING THREAD TOOLS.

The present chapter contains some information regarding the making of special threading tools, square threading tools, and several tables which will prove useful when making thread tools. It is not a complete treatise on the making of thread tools, but contains such general information as the tool-maker is most likely to require.

Formula for Planing Thread Tools.

Fig. 29 shows a diagram of, and below will be found formulas for, thread tools, with special reference to those used in a Pratt & Whitney thread tool holder, which holder is the one considered the best and mostly used by leading firms. As the planing of thread tools used in this holder is rather particular, and quite confusing to those not familiar with the process, formulas are given by means of which the angles to which the planer or shaper-head should be set, can easily be figured. The formulas will be readily understood from the diagram, but a word may be needed in explanation of "the leading" and "the following" side of the thread tool, the former being that side of the tool first entering the work when a thread is cut.

1. Tool with Side Clearance.

a = depth of thread.
b = width of flat on offset tool.
c = actual width of flat.
d = outside diameter of screw.
v = clearance angle.
w = ½ angle of thread.
y = angle of helix.
x = normal angle (to which to set planer head, when planing tool on side).

$$\tan y = \frac{\text{Lead}}{(d - a)\ 3.1416}.$$

$$\tan x = \frac{\cos y \pm (\cot w \times \sin v \times \sin y)}{\cot w \times \cos v}.$$

Use + for leading side and — for following side.

For Acme (29 deg.) thread and 15-degree clearance angle, the formula can, for all practical purposes, be written:

$$\tan x = \frac{\cos y \pm \sin y}{3.735}$$

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The width of flat on the offset tool is figured from the formula: $b = c \times \cos y$.

2. Tool without Side Clearance.

If the tool has no side clearance, the angle of helix can be considered $= 0$ deg., and above formula reduces itself to: $\tan x = \frac{\tan w}{\cos v}$; for 60 deg. screw thread, United States standard, the formula has this appearance: $\tan x = \frac{\tan 30°}{\cos 15°} = 0.5977$; $x = 30$ deg. 52 min.

In this latter case the width of flat of tool (c) remains unchanged. It will be noticed that formulas are given first for "tools with side clearance" and second for "tools without side clearance"; of course any thread tool ought to be given a side clearance, the amount of which depends on the angle of helix of thread to be cut, but on account

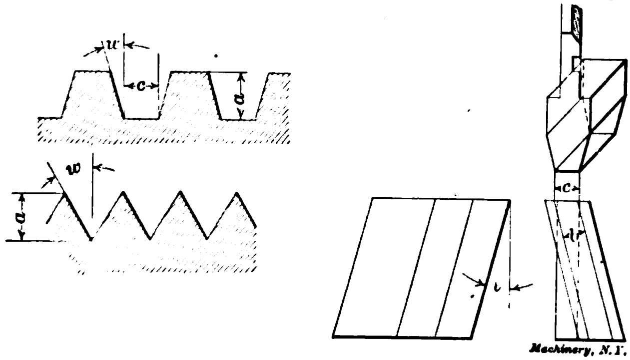

Fig. 29. Diagram Illustrating the Planing of Thread Tools.

of the small angle of helix on fine pitch threads, the necessity of using a tool with side clearance in such cases is reduced to a minimum, and can for practical reasons be dispensed with.

Widths of Tools for Cutting Square Threads.

When cutting square threads it is customary to make the screws exactly according to the theoretical standard of the square thread. The width of the point of the tool for cutting *screws* with square threads is therefore exactly one-half of the pitch, but the width of the point of the tool for cutting taps, which afterwards are used for tapping nuts, is slightly less than one-half the pitch, so that the groove in the tap becomes narrower, and the land or cutting point wider than the theoretical square thread, thereby cutting a groove in the nut which will be slightly wider than the thread in the screw, so as to provide for clearance. An inside threading tool for threading nuts evidently must be of the same width as the land on the tap would be, or in other words, slightly wider than one-half the pitch. This provides, then, the required clearance. The accompanying table gives the width of the point of the tool for all ordinary pitches from one to

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twenty-four threads per inch. The second column gives the width of the point for cutting taps to be used for producing square thread nuts. The third column gives the width of the point of the tool for cutting screws which, as we have said, equals one-half the pitch, and the fourth column gives the width of the point for inside threading tools for nuts. While the table has been carried to as fine pitches as those having twenty-four threads per inch, square threaded screws having so fine a pitch are very seldom used. Some manufacturers of square threading tools, however, make square threading tools for pitches as fine as these, and for this reason they have been included.

Clearance Angles of Square Thread Tools.

In the Chart Fig. 30, reproduced from MACHINERY'S Data Sheet No. 97, directions are given for the use of the diagram presented, by

TOOLS FOR SQUARE THREAD.

No. of Threads per Inch	Width of Point of Tool			No. of Threads per Inch	Width of Point of Tool		
	For Taps	For Screws	For Inside Thread Tools for Nuts		For Taps	For Screws	For Inside Thread Tools for Nuts
1	0.4965	0.5000	0.5035	8	0.0615	0.0625	0.0635
1⅓	0.3715	0.3750	0.3785	9	0.0545	0.0555	0.0565
1½	0.3333	0.3333	0.3363	10	0.0490	0.0500	0.0510
1¾	0.2827	0.2857	0.2887	11	0.0444	0.0454	0.0464
2	0.2475	0.2500	0.2525	12	0.0407	0.0417	0.0427
2½	0.1975	0.2000	0.2025	13	0.0375	0.0385	0.0395
3	0.1641	0.1666	0.1691	14	0.0352	0.0357	0.0362
3½	0.1408	0.1428	0.1448	15	0.0328	0.0333	0.0338
4	0.1235	0.1250	0.1265	16	0.0307	0.0312	0.0317
4½	0.1096	0.1111	0.1126	18	0.0272	0.0277	0.0282
5	0.0985	0.1000	0.1015	20	0.0245	0.0250	0.0255
5½	0.0894	0.0909	0.0924	22	0.0222	0.0227	0.0232
6	0.0818	0.0833	0.0848	24	0.0203	0.0208	0.0213
7	0.0699	0.0714	0.0729				

means of which the clearance angles on the sides of square threading tools may be determined at a glance. The example given in the chart will fully explain its use.

Tables Giving Angles of Threading Tools.

The tables on pages 38 and 39 will be of interest and use to tool-makers. The first table in question deals with the circular threading tool. This kind of tool most generally has its cutting face below its center line, which, of course, changes the angles; it not only changes the angles, but as we lower the cutting edge, the same becomes a convex line. On large diameters this second error is not noticeable to any extent, although it exists from the very moment we lower the cutting face of such a tool. This one item makes it very difficult to accurately give the angle to make such a tool, which should cut an accurate 60-degree thread, when the tool is cut a certain amount under its center line.

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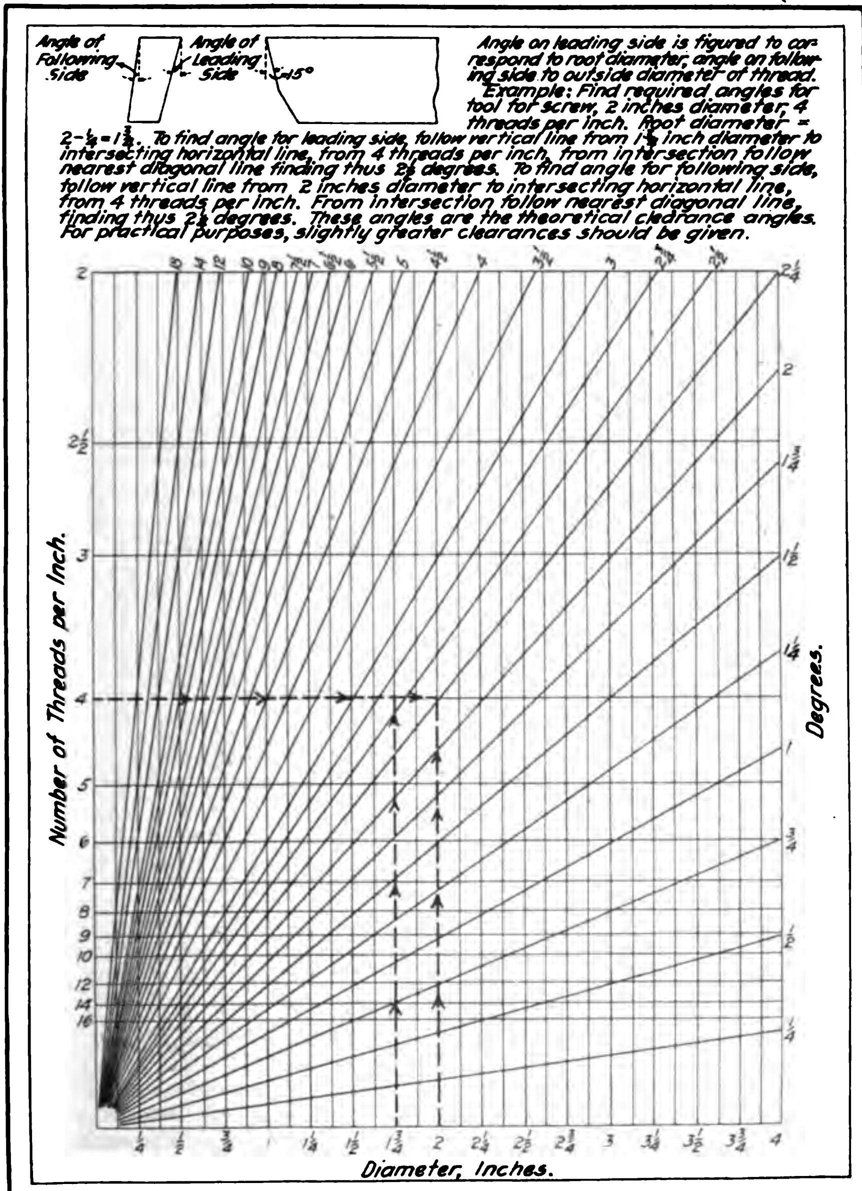
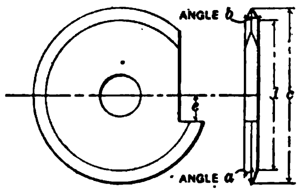


Fig 30. Diagram giving Clearance Angles of Square Thread-Tools

The angles in the table below were computed by taking diameter *d* to be ⅛ inch smaller than *c*; this difference was used throughout the tables. The first column gives the largest diameter of tool, while the second gives the diameter from which the following angles were obtained. The eight columns following give the angle of one side of

CIRCULAR THREADING TOOLS.



a — ANGLE WITH SIDE OF TOOL
b — INCLUDED ANGLE OR 2 × ANGLE a
c — LARGEST DIAM. OF TOOL
$d = c - .125$ (THESE TABLES WERE COMPUTED FROM THIS RATIO)
e — DISTANCE FROM CENTRE LINE TO CUTTING FACE

LARGEST DIAM. OF TOOL — c	DIAM. $d = c - .125$	ANGLES ON CENTRE LINE WHEN $e = .125$		ANGLES ON CENTRE LINE WHEN $e = .1875$		ANGLES ON CENTRE LINE WHEN $e = .250$		ANGLES ON CENTRE LINE WHEN $e = .3125$	
		ANGLE a	ANGLE b	ANGLE a	ANGLE b	ANGLE a	ANGLE b	ANGLE a	ANGLE b
¾	⅝	31° 41′	63° 23′	34° 38′	69° 08′	40° 10′	80° 20′		
⅞	¾	31 15	62 30	33 05	66 10	36 18	72 36	42° 25′	84°.50′
1	⅞	30 56	61 52	32 13	64 26	34 20	68 40	37 50	75 40
1⅛	1	30 43	61 26	31 41	63 22	33 13	66 26	35 34	71 08
1¼	1⅛	30 34	61 08	31 19	62 38	32 29	64 58	34 12	68 24
1⅜	1¼	30 28	60 56	31 04	62 08	31 59	63 58	33 18	66 36
1½	1⅜	30 23	60 46	30 53	61 46	31 37	63 14	32 40	65 20
1⅝	1½	30 19	60 38	30 45	61 30	31 22	62 44	32 13	64 26
1¾	1⅝	30 17	60 34	30 38	61 16	31 09	62 18	31 52	63 44
1⅞	1¾	30 14	60 28	30 32	61 04	31 00	62 00	31 36	63 12
2	1⅞	30 13	60 26	30 29	60 58	30 52	61 44	31 23	62 46
2⅛	2	30 11	60 22	30 25	60 50	30 45	61 30	31 13	62 26
2¼	2⅛	30 10	60 20	30 22	60 44	30 40	61 20	31 04	62 08
2⅜	2¼	30 09	60 18	30 20	60 40	30 35	61 10	30 57	61 54
2½	2⅜	30 08	60 16	30 18	60 36	30 32	61 04	30 51	61 42
2⅝	2½	30 07	60 14	30 16	60 32	30 29	60 58	30 46	61 32
2¾	2⅝	30 06	60 12	30 14	60 28	30 26	60 52	30 42	61 24
2⅞	2¾	30 06	60 12	30 13	60 26	30 24	60 48	30 38	61 16
3	2⅞	30 05	60 10	30 12	60 24	30 22	60 44	30 34	61 08
3⅛	3	30 05	60 10	30 11	60 22	30 20	60 40	30 32	61 04
3¼	3⅛	30 04	60 08	30 10	60 20	30 18	60 36	30 29	60 58
3⅜	3¼	30 04	60 08	30 09	60 18	30 17	60 34	30 27	60 54
3½	3⅜	30 04	60 08	30 09	60 18	30 16	60 32	30 25	60 50
3⅝	3½	30 03	60 06	30 08	60 16	30 15	60 30	30 23	60 46
3¾	3⅝	30 03	60 06	30 08	60 16	30 14	60 28	30 22	60 44
3⅞	3¾	30 03	60 06	30 07	60 14	30 13	60 26	30 20	60 40
4	3⅞	30 03	60 06	30 07	60 14	30 12	60 24	30 19	60 38

Machinery, N.Y.

tool and also the included angle for such tools which have their cutting face 1/8, 3/16, 1/4 and 5/16 inch below the center line. Should one desire to construct such a tool for very coarse threads, say, for instance, for 2 or 3 pitch, it can readily be done with very accurate results by simply manipulating the figures in the table.

Example: A circular tool is to be made, of which the extreme diameter is to be 2 inches and which is to be used for cutting 2-pitch threads; its cutting face is to be 5/16 inch below the center line, and

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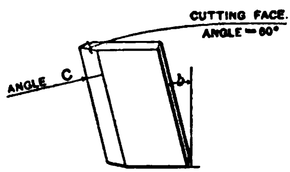
it must cut to the depth of 0.433 inch when the top width of cut equals 0.500 inch. Now the table gives us for a 2-inch diameter 60-degree tool cut 5/16 inch below center, the half angle 31 deg. 23 min., or 62 deg. 46 min. included angle. These angles would be accurate for making a tool that was to be used on threads that have an approximate depth of about 1/16 inch, but for the tool in question we would come nearer right if we reckoned our two diameters, namely, 2 inches for the one and 1⅛ inch for second diameter, since 2 × 0.433 = 0.866, which is nearly ⅞ inch, and 2 inches — ⅞ inch would equal 1⅛ inch for the second diameter. Now if we consider the intermediate diameter 1 9/16 inch—which is found as follows: 2 — 1⅛ = ⅞ inch; ⅞ ÷

STRAIGHT THREADING TOOLS.



ANGLE b = CLEARANCE ANGLE OF TOOL	ANGLE C = ANGLE MEASURED ON FORWARD SIDE OF TOOL. C = ANGLE WITH CENTRE LINE		C = INCLUDED ANGLE	
8	30°	15′	60°	30′
9	30	18	60	36
10	30	23	60	46
11	30	28	60	56
12	30	33	61	06
13	30	39	61	18
14	30	45	61	30
15	30	52	61	44
16	30	59	61	58
17	21	07	62	14
18	31	16	62	32
19	31	25	62	50
20	31	34	63	08

2 = 7/16 inch; 7/16 + 1⅛ = 1 9/16 inch—we find upon referring to the tables that a 1 9/16-inch diameter is not given, so we divide the difference between a 1½-inch and 1⅝-inch diameter; this difference is 27 minutes, half of which would be about 14 minutes. This added to the angle given in the 1⅝-inch line would equal 32 deg. 27 min., which would be the proper angle to make the tool.

We will now turn to the straight threading tool, which is a more accurate tool than the circular, because we have not the convex side to contend with. The cutting edge of a straight tool is always a straight line (provided it is made accurately) regardless of what the clearance angle is, although we have the same problem to contend with in this style of tool as in the circular, namely: when the cutting angle equals 60 degrees, for example, what is the angle on forward side of tool?

The table above gives this angle. As will be seen in this table, the

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first column gives the clearance angles which range from 8 to 20 degrees, inclusive. In the second and third columns are the respective single and included angles, which when measured on the forward side of the tool will coincide with a perfect 60-degree angle on the cutting face.

There is still another item which is of no less importance than any previously mentioned, and which concerns both the straight and circular threading tools, and that is the setting of such tools in the machine so that they may stand in alignment with the angle of the thread that is being cut. Many threads are cut which are smooth on one side and rough on the other; the cause is not having an equal amount of clearance on each side of the threading tool. The old style of lathe tool which was used for threading purposes had a little advantage over the circular and straight tools in this respect, because it had clearance both ways, but with such tools that can be ground without changing their form, we must obtain front clearance only. This makes it more essential to have these tools stand as near in line with the angle of the thread as possible; but when we speak of setting such tools perfectly in alignment with the angle of a thread, we have an impossibility to contend with, because the root of a thread is always smaller in diameter than the apex, and as the lead on both root and apex remains the same, the angle must of course change, when going from one diameter to another. In other words, the angle of the spiral at the root diameter is always greater than at the apex of thread. The most correct diameter to select would be about midway between the root and apex of thread, but as the changes of angles are very slight, and really too slight for practical importance, they are generally computed from the diameter over the apex of the thread.

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The foregoing books, up to and including No. 26, were published and in stock in November, 1908. The remainder will go to press as rapidly as practicable. The complete plan of the series, as stated, is to cover the whole field of mechanical practice, and the editors are preparing the additional titles, which will, from time to time, be announced in MACHINERY.

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No. 33

SYSTEMS AND PRACTICE OF THE DRAFTING-ROOM

CONTENTS

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CHAPTER I.

STANDARD DRAFTING-ROOM METHODS.

The theme of the standardization of methods in the drawing-room is one which is of vital interest to all who are connected more or less directly with this line of work. There are so many leaks possible in the drafting department of any firm, so many ways in which time may be saved by having a way to do things and a place to put them, that a few words upon this topic cannot fail to interest many.

Of course, good light and the least possible amount of noise and confusion in the room during working hours are a foregone conclusion, while the equipment and size of the quarters devoted to this branch of the business must necessarily depend upon the size of the company. Beyond this, however, the way in which the drawing-room ministers to the wants of the factory, and the accuracy and speed with which the drawings are turned out, depend greatly upon the efficiency of the system and the longheadedness of the chief.

About the first step in any good system is the adoption of a number of general rules, governing the production of any new work, which may be easily blue-printed and handed to any new man on his arrival, thus giving him a line on the general way in which the work is desired to be gotten out. These data may conveniently cover such points as sizes of drawings, methods of dimensioning, limits to be used on the work, methods of indicating various finishes, styles of lettering, cross sections, etc. In some drafting rooms they have gone a good deal further than this, and have what may be termed data sheets. These, in addition to the above, comprise a list of the stock of steel in the various sizes, shapes, and qualities, carried by the firm; stock patterns, examples and explanations of the various formulas in use in the shop; and, in general, a collection of data relative to the firm's work, which the draftsman or designer might spend much valuable time in looking up.

On account of the fact that everyone, no matter how careful, will occasionally commit errors, an efficient checking method is essential to do good work. The best method to follow is this: The chief explains to the draftsman, by sketch or verbally, what he desires, and the drawing is made under the supervision of the chief who gives it his approval as regards design. It is then submitted to the superintendent, the chief engineer, or somebody who is in the last instance responsible for results. He either approves it, or orders such changes as he thinks advisable, and the drawing is returned to the draftsman for alteration, if necessary, or, if not, is passed to the checker. The drawing is then thoroughly checked by him for accuracy as to scale, dimensions, and mathematical calculations. If any corrections are found necessary, the drawing is again returned to the draftsman, who

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makes the necessary alterations and returns it to the checker, receiving his approval on same. The drawing then goes to the tracer, who makes the tracing and returns the original and the tracing to the checker. If any corrections are necessary on the tracing, the tracer makes them under direction from the checker, who finally approves the tracing. It is then ready for the blue-print room, and any errors which show up later are held against the checker. This system is very thorough, and the errors that will occur are few and far between.

Another feature of value is the grouping of the various blue-prints covering the manufacture of a certain machine, or a number of machines of similar character, in bound packs or books, located at various points throughout the plant. This obviates the continual replacement of lost prints, which consumes so much time where loose prints are used. Each book is receipted for by the foreman who has the work covered by it in charge, as are also the new prints made necessary by changes in design or dimensions. A record of the location of these books is, of course, kept in the drawing-room, and a man is detailed to keep them up to date. There should be an exception to this rule in the screw machine department, as the prints of the parts should here be mounted on boards and shellaced, the operator having one of these cardboard mounts on his machine where he can refer to it. The mounts not in use are kept in a cabinet for that purpose, where they are easily accessible to the foreman of the department in planning his work. In this connection it might be well to note that all the prints necessary to go into any one department are those referring to operations performed in that particular part of the works, and the drawing-room is generally the only place where a complete set of prints is available.

A good system of handling and recording the changes made on drawings is rather difficult to devise. The following system, however, will prove satisfactory. If the change is a slight one, as for instance the change of a dimension, the tracing is changed, the date of change being noted in the lower right-hand corner, and the various prints are changed by the man in charge of that work. If the design is changed, a new tracing is made, the old one marked obsolete, new prints made and put in the books, and the new tracing filed with the old one. In each case a record of every change is made in a book kept for that purpose, and as the parts are all arranged numerically, it is very easy to refer to this record to find the details of the change in making repairs or filling orders for old parts.

It is remarkable what a number of drawings will accumulate in the course of a few years, when old designs are being constantly brought up to date, and new machines being added to keep abreast of the times. Owing to this fact, it is necessary to have a number of cabinets, with drawers made to fit the different sizes of drawings. In these drawers the tracings are filed, as has been said, according to numerical order. One should also install a card index giving the exact location and size of the drawing of the particular part sought, and in case the number of the part is not known, a cross index to give

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an alphabetical classification. As an auxiliary to this index one should have a smaller one in which are grouped such tools as bits, reamers, special drills, counterbores, etc., which is a great aid and convenience to the designers in making up similar tools, or, as often happens, in adapting the old tool to a new part. In most drafting rooms there are, of course, two distinct divisions, one for the design and production of the drawings covering the machines themselves, and one for the design of the tools necessary for the economic production of these parts. In the former department the data utilized can for the greater part be found in the standard mechanical works or the trade catalogues. In the tool designing department, however, the data is mostly compiled from the book of experience, of which there is no authentic edition.

The following general instructions are intended to be, for the most part, or with minor changes, applicable to the practice of the average drafting room, and, for that reason, are usually confined to principles. Disagreement will often be found with those which are not principles, but they represent excellent practice. They are condensed as much as possible into a few words, as the object has been to state them so concisely that it would require but a few minutes to hurriedly read the whole and to thus make it easier for a beginner to read them every day, or for a checker to glance them through, occasionally, for pointers that might have slipped his mind.

Scale.

Make details to a scale large enough to distinctly show all parts and also to give sufficient room for necessary dimensions. This will sometimes require two or more times actual size. Do not use an unnecessarily large scale when it will also require a larger sheet than is otherwise required.

As to scales, there are more in use than desirable or even necessary, and the use of an odd scale should not be tolerated, notwithstanding the claim made that the use of scales not common prevent scaling the drawing by workmen; but others besides machinists may want to "check up," and then it becomes awkward, to say the least, especially if that particular scale is not at hand.

Occasionally drawings will be noticed bearing the phrases, "Do not scale," "Work to figures," "Report all errors to this office," etc. All tnis seems useless, for general rules and common sense indicate this as clearly as words can do it. It is an insult to the intelligence of the workmen on the part of the draftsmen to try to indicate to him that which is clearly obvious.

Views.

The views required are those necessary to completely and plainly show the piece—no more, no less—except that one view is sufficient if another would show no more than is given by "4 inch dia.," "¼ inch thick," etc., on the one view.

Show pieces in the position they occupy on the construction drawing or on the completed work, when there is no disadvantage in doing so.

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Do not leave wider "open" spaces between the several views of one piece than between those and the views of nearby pieces. Do not crowd views so closely together that there will not be sufficient room for dimensions and notes.

Show long pieces with a portion of the length broken out, when a larger scale is desirable than could be used if the full length of the piece were shown; but do this only when a continuous portion of parts fully shown is thus broken out, or when notes fully explain the cmitted portion.

A part of a view may be shown when the remainder would be only a repetition of what is plainly shown elsewhere.

Always use "third angle" projection; that is, place views nearest the side of adjoining view which they show. Follow the same principle of direction of view in sections, making them on the side where the outside view of the cut-away portion would be. Any deviation from this rule must be very plainly noted on the drawing.

Always draw both right-hand and left-hand pieces when both are to be made, unless the differences are so simple that one or two dimensions can be noted for each without confusion. If the pieces are castings, and but one pattern is required, make notes to that effect, mentioning changes. This also applies to similar pieces cast from one pattern with changes, when a detached view, or portion of one, would often show the change.

Lines.

Make outlines in medium or heavy lines, giving strong contrast with dimensions and center lines, which must always be light but distinct.

Make section lines not less than 1/16 inch apart, except on widths of less than ⅛ inch. Make them lighter than the outline. On ordinary work, narrow sections may be cross-sectioned by free-hand.

Have no "hair" lines in figures, letters or outlines.

Dimensions.

Give all the dimensions necessary to make the piece—no more, no less—and do not repeat them on same nor on different views of the same piece unless for a special reason.

Give dimensions so that the workman will not have to do any important figuring himself.

Give dimensions where and as they will be most useful to the workman.

Do not give dimensions from the center line of a piece when it is not necessary.

Give dimensions from something that the workman can and should measure directly from, if reasonable to do so.

Give dimensions between places having a definite relation rather than otherwise.

Place the shorter dimensions nearest the outline.

Place dimensions so that there need be no doubt as to what they refer.

Do not repeat a dimension many times in a single line where the

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likeness is clear from the drawing. At most, give it two or three times and then include the remainder, or the whole, in a dimension reading like this: 12 spaces at 9″ = 9′ — 0″.

Never crowd dimension lines or figures.

Run the limiting line slightly beyond the end of the dimension line.

On a piece having several diameters in its length, or in similar holes, give the diameters in the side view, or section, rather than in the end view.

Where two things are centrally spaced in relation to a third, and also definitely related to each other, give one dimension from the center to one and another between the two.

Where an inside dimension is given on an outside view and is not much different from it, make distinct by marking "inside," or by giving, in line with it, the remaining dimension.

Do not call for impossible or unnecessary accuracy. If a dimension is calculated 18 1/64 inches and the work is so rough that inaccuracies of 1/16 inch are provided for, leave off the 1/64 inch, but do not disregard the correct figure in further calculations.

Give angles from existing surfaces, or, so that no figuring will be required of the workman in order to measure the angle.

Make the dimension line for an angle an arc with its center at the vertex of the angle. Mark figures "deg." and "min." Do not use their signs.

Mark radii "rad." or "R."

If it is generally understood that all sizes are given in inches and fractions thereof, up to a certain definite number, like 100, the inch mark (″) may well be neglected as time taking and obscuring to the drawing. On 6's and 9's some prefer to use them for the sole purpose of indicating the position they are to read from. Plain Arabic figures, the fractions smaller than the integers, are preferable in making a clear drawing.

Figures and Signs.

Never put letters, figures, signs or arrow-heads on, nor running into, each other, the outline of the piece, nor any but their own dimension lines.

Make figures and lettering read from the bottom or from the right-hand end of the sheet, always parallel with one or the other, except dimensions on lines necessarily on an angle, and on radii, and angles dimensioned in degrees, which may read on an arc of the angle.

Make figures and letters large enough and heavy enough to be distinct—not less than 5/64 inch or 1/12 inch high on ordinary work. That size for fractions, and 1/8 inch to 5/32 inch for whole numbers is good practice.

Make the dash between feet and inches distinct so that 3′ — 4″ shall not be mistaken for 34″.

Make arrowheads plain and neat. Make them blunt if there is danger of doubt as to the line they designate.

Use open and distinct forms of figures and letters.

Make inch and foot marks distinct, but much smaller than the figures.

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Make whole numbers given together with fractions large enough that they will not be mistaken for a part of the fraction.

When the space is not necessarily limited in height, use the horizontal vinculum, or separating line in fractions: $\frac{1}{2}$" rather than 1/2".

Notes and Lettering.

Put notes on the drawing whenever necessary for a distinct understanding of the requirements, as when one part is to have a certain fit with another, except where the same is otherwise provided for, as by gages, micrometer measurements, etc.

Express things in notes whenever they will be plainer that way than by further drawing or dimensions.

On rough works it is often well to note character and use on size of holes thus: "drill 17/32" for ½" bolts," "punch 9/16" for ½" bolts," "core ⅝" for ½" bolt," "ream for ½" shaft," "bore for sleeves No. 65."

When there might be a question about them, mark cast holes whether cored or cut in pattern.

On work having little finishing, mark such surfaces finish, bore, drill, turn, etc.

Run leading lines from notes to holes or other features to which they apply, unless their relation is very plain. Make them free-hand, light, with black ink and an arrowhead at the end indicating unmistakably to what they refer.

Avoid all words and phrases which might be easily misunderstood, or not understood at all.

When a note consists of more than two lines, make them match, vertically, on the left-hand side.

Make spacing between lines of lettering in notes uniform and not wider than the height of the letters.

Except in simple cases, name the sections of a piece and show by a light broken line where they are taken. Letter the ends of the line to read from the side that would be removed to show the section as drawn.

Make all lettering in capitals, vertical, or very slightly inclined to the right, and of uniform height.

Make titles, names of sections, views, and pieces, with letters ⅛ inch high, other lettering 1/12 or 5/64 inch high.

Such statements as "one thus," "one of this"—both of which are actual prevarications—should be abolished as time-taking nonsense. The number of pieces required may be given simply in connection with the material to be used, and the name or number designating the part shown, and nothing more is required. This had better be written, rather than given in numerals, to prevent any possible error.

The general title—as the name of the machine, the parts actually shown, or the word "Details," the party for whom the drawings are made (if other than the makers), and the engineering office making them—should be given. If likely to be sent away, and hence become "foreign" drawings, the address may also be added.

Symbols and consecutive filing numbers should be sufficiently large

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so as to readily catch the eye; in fact, they should be the most prominent thing about the sheet, and it is not a bad plan to have the same repeated on the opposite corner in case the drawing should be turned bottom up.

General Rules.

Exercise your "gray matter" a little before doing things rather than "the boss's" patience afterward.

A rule is sometimes better observed in the breaking rather than in the keeping. Heed the spirit as well as the letter of all instructions.

The men who make the pieces are not draftsman and should not be supposed to know more about a piece than is given on the drawing.

A drawing is plainer to the man who makes it than to the mechanic who first sees it when he just starts on the work. Consider what you would want to know if you were to make the piece correctly under the workman's circumstances.

Strive for clearness, completeness, simplicity and neatness.

While it is admitted that drawings are not in themselves an end, but rather a means to an end, it is equally true that they constitute a sort of universal language in mechanics, and the more perfect and uniform this language can be made the better it must be for all concerned.

Summarizing, and putting our instructions in the most concise form, we may formulate the following exact rules for general drafting-room work:

All drawings should be made on standard size sheets, within standard borders; full size, or to as large a scale as possible.

All views should be placed in the third angle.

Views should be so arranged on the sheet as to leave a clear space for the title at the lower right-hand corner.

Shade lines, *i. e.*, heavy lines drawn on the right-hand and lower side of solids—oppositely on interiors—should be used.

Shading, excepting at the intersection of cylinders, is not desirable.

Use section lines as little as possible.

All figures to be at least 1/8 inch high and clear of view.

All sizes less than 100 inches to be in inches.

No inch marks should be used, except on the 6's and 9's, when alone.

The oblique line between the members of a fraction should not be used whether the fraction is in connection with an integer or stands alone.

Center lines should be made in thin black.

Dimension lines should be made in thin black.

Titles should indicate, (a) the object drawn; (b) for whom or for what purpose; (c) name of engineering office; (d) scale used; (e) date; (f) draftsman; (g) checker; (h) number of drawings in set; (i) individual number of drawing; (j) symbol; (k) numerical order of drawing; and when desirable, (l) designer; (m) size of sheet, represented by symbol; (n) if detail or assembly, and (o) by whom approved.

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Checking a Drawing.

In checking drawings for a machine, the first thing to be done in all cases is to get the general arrangement, and to check the center distances and elevations with the principal dimensions given on the details. After the principal dimensions on the general drawing have been checked, it is used as a lay-out for the details. Each dimension on the detail is checked, and if a similar detail has been used on some other machine, in order to establish a standard, the new one should be made proportional to the old. Finish marks are noted on details where required, and none on surfaces that do not need it, for any finish that can be saved lessens the time in the machine shop. Sizes of all bolt holes are marked and the bolts to be used are called for or noted. Every piece on the drawing, either called for or shown, must be given a letter or pattern number, and the number of pieces required for each drawing ordered. If a certain radius for curve is used in one place, the same radius should be used in each similar place. Care must be taken in checking that standard sizes are used as far as possible, especially in slots for bolt heads. If a slot is made so wide that a standard square head bolt turns in it, a special bolt is required; while if the slot had been made narrower, a standard bolt from stock would have answered the purpose just as well. Whenever practicable, the pieces of a machine should be made so that one can be removed without disturbing the other, this saving a great deal of time when making repairs. As few patterns as possible should be made, and often, with a little foresight, one pattern can be made to answer in two or three places by making some pieces loose so that they can be changed around to suit different conditions. Great care must be exercised that there are no interferences. These can best be detected on the general arrangement and by laying out the adjacent machinery.

In most drawing-rooms, when checking, the tracing is used, and all the changes are noted on the tracing with a pencil—blue, red or black, as the draftsman prefers, while the correct figures are checked with ink. After the man who made the drawing has approved the indicated changes, the tracer makes them without erasing the pencil marks. The drawing is then returned to the checker, who checks off the changes that have been made. As he checks them off in ink, he cleans the tracing with a sponge rubber and anything the tracer may have missed the checker finds during this operation. All figures on the drawing must be clear, so that there can be no doubt in the shop as to the intended dimensions. The pattern and drawing numbers must be looked up to see if the records have been made out properly, and after the detail drawings are all checked the general drawing can be finished. All the pattern numbers are noted and the principal pieces marked with their particular mark and drawing number.

It is, however, a much easier and better way to take a print of the tracing to be checked. All corrections or changes are indicated on

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this print with red pencil, and all figures that are right are checked with yellow pencil. The changed blueprint is given back to the draftsman and he and the checker go over it and decide which is the correct figure or the better way to do the work. After all changes have been approved, the tracer makes the corrections indicated in red pencil. When this is finished the checker compares the tracing and the blueprint to see if all changes have been made and if he finds the dimensions on the tracing correspond to the red ones on the print, he checks those on the print with a blue pencil, marks the tracing "checked," and the checked blueprint is filed. If the work does not go together properly in the shop, the chief draftsman can produce the checked blueprint, see if the dimensions have been overlooked or improperly checked, and so know where to place the blame. When a drawing is checked by this method no check marks show on the tracing, but it is marked in one corner: "Checked by ——," and the date given when checked.

With the first method it is not unusual for a drop of ink to fall on the tracing or for some of the check marks to be blurred. The checker must then spend quite a little time in cleaning off the drawings, a waste of labor which is obviated by the latter method. If there are a large number of drawings for the machine, the tracings in the first method get very badly soiled and crumpled, it being necessary to have them all on the table during the entire process of checking, which often occupies a week. In the latter method, however, the tracings are out only while the tracer is making the necessary changes and while the checker looks them over to see if they are correct.

The following concise rules for checking drawings may prove of value to inexperienced checkers: See that the parts are strong enough to do the work. See that they are of the proper materials and that all the dimensions are drawn to scale and check up with each other. Care should be taken that the proper number of pieces of each part for one machine is called for on the drawing. The different parts should be so designed that the machining may be easily done with the facilities at the disposal of the establishment, and finish marks should be carefully checked to see that some parts are not marked to be finished which should be left rough, and *vice versa.* There should be, also, a correct list of the subsidiary but necessary articles to be drawn from the store house or ordered outside, as, for instance, oil cups, nuts, wrenches, and bolts.

Each part must be so designed that it may be readily removed for inspection or be easily accessible for cleaning and repairs. If the part is heavy, facilities should be provided for handling it, and some parts, such as cylinder heads, should have starting screws by which they can be given the preliminary lift from their seats. All moving parts should be furnished with lubricating devices so located that filling and cleaning may be easily attended to, and with oil passages so disposed that the lubricant will easily reach the surfaces for which it is intended.

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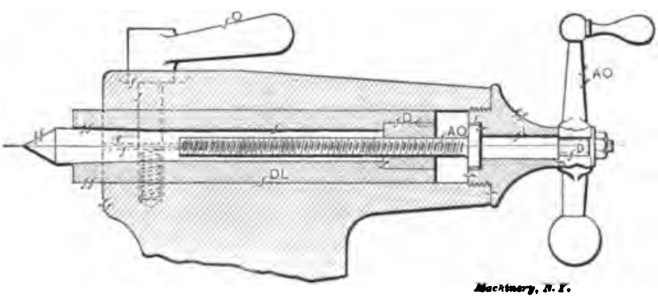
CHAPTER II.

GENERAL SUGGESTIONS IN MAKING DRAWINGS.

Indicating Finished Surfaces.

Fig. 1 shows a simple and convenient system of finishing marks which has been in use in a well systematized drafting room for several years. It will be noticed that the usual *f* is the predominating character, with the addition of another letter at the right, this letter denoting the fit desired in the piece on which it may be placed. This exponent, as it were, has not been chosen so much because it would suggest the character of the fit, but rather for the ease with which it may be made on the drawing, that is, with one stroke of the pen. In the design of special machinery, where the workmen have no past experience to guide them, these marks have saved, to the draftsman, any small and yet important questions as to fit, finish and quality of finish, necessary.

On detail drawings, something to show the fit is essential to make a



Machinery, N.Y.

Fig. 1. Indicating Finished Surfaces.

complete working drawing, and on more or less assembled drawings some marks of this nature are of no less importance, for each man having occasion to use the drawing can tell at a glance what should be a running fit, what a driving fit, what ordinary machine finish, and what polished. The allowance for the fit is preferably made in the holes, the parts fitting them being machined to the exact figure given. This, however, is unimportant, as the allowance could be made on the parts fitting the holes, according to the individual shop practice.

The following table will give a clearer idea of the application and value of the marks. If each man is given a blue-print or card of the finish characters along with the first drawing on which they are used, no further trouble is found in making the men accustomed to their use.

Table of Finishing Marks.

The following marks will be used on drawings to indicate the finish and fits required:

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f, machine finish.

ff, machine finish, (polished).

f^{O}, hand finish only.

f^{S}, forcing fit, — 0.002 for first inch and 0.001 each additional inch.

f^{D}, driving fit, — 0.001 for first inch and 0.0005 each additional inch.

f^{DS}, easy driving fit; exact size.

f^{L}, running fit, + 0.001 for first inch and 0.001 each additional inch.

f^{I}, finish exactly to size.

GD, gear distance.

+ or —, allowance between shoulders.

⟶ key drives this way.

f^{AO}, finish all over.

All allowance for fit to be made in holes. Shafts to given dimensions. All dimensions in inches up to 8 feet.

In Fig. 2 is shown another system of finishing marks which has

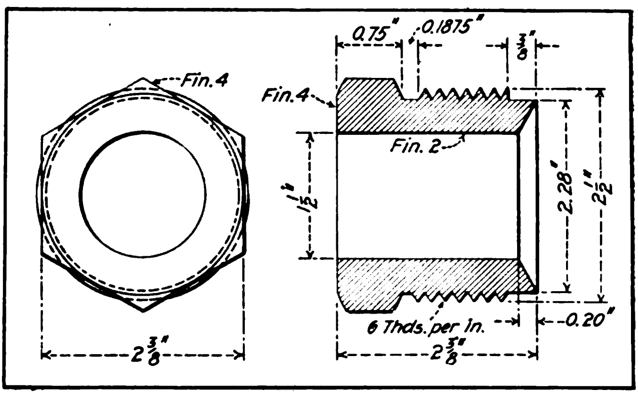

Fig. 2. System for Indicating Finished Surfaces Applied to a Drawing.

also been in practical use for several years in another well-organized drafting room. In this system the various classes of finish are designated by number, as follows:

Finish No. 1 requires surfaces to be extremely smooth and accurate within a tolerance of ± 0.0005 inch.

Finish No. 2 requires surfaces to be very smooth and accurate within a tolerance of ± 0.001 inch.

Finish No. 3 requires surfaces to be smooth and accurate within a tolerance of ± 0.003 inch.

Finish No. 4 requires surfaces to be accurate within a tolerance of ± 0.005 inch.

Finish No. 5 requires surfaces to be rough machined or filed to within a tolerance of ± 0.025 inch.

Finish No. 6 requires that castings or forgings be cleaned of all sand, scale, risers, fins, etc.; and that no thickness of metal when ready to assemble shall differ from drawing dimensions more than ± 5 per cent.

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All allowance for fit is to be made on the female parts. Male parts are to be made to the standard size.

With the above system, used on drawings as indicated in Fig. 2, work can be laid out to any degree of accuracy required and the workman readily knows between what limits of accuracy he must make any dimension. In cases where the piece can be made satisfactorily with one grade of finish all over, a note may be added to the name, etc., of the piece on the drawing to that effect. For example, the specification: Washer—Finish No. 4, attached to the drawing of a washer would indicate that no dimension need be worked closer than ± 0.005 inch to the specified sizes.

Recording Changes on Drawings.

The recording of changes or revisions made on drawings of any kind is of the greatest importance, and Fig. 3 illustrates a very satis-

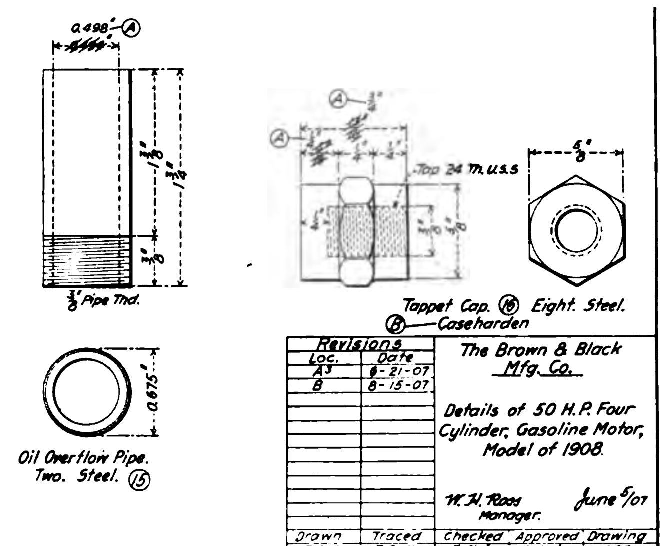

Fig. 3. Method of Recording Changes on a Drawing.

factory method of recording changes on a drawing. The columns headed Revisions show the location on the drawing of dimensions revised and also the date when the revisions were made. The small figure at the right of the letter of revision in the column indicates how many places on the drawing the revision affects. For instance, the revision A^3 affects the drawing in three places, while *B* affects it in only one place, as shown. It is a matter of opinion as to how to remove the old dimension and place the new dimension in its place, but it is preferable that the old dimension be merely crossed out and the new dimension be shown above it, as in Fig. 3.

It is advisable, on account of its simplicity and the possibility for ready reference, that a separate file of revision records for each type

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of machine being manufactured be kept. A card system, of some such form as shown in Fig. 4, is preferred by the writer, because its flexibility allows for sufficient expansion to cover all details concerning any revision that may be made in any drawing. The cards in the revision file are indexed by the drawing numbers, and are arranged in their several indexed spaces in the alphabetical order in which the revisions were made. By the above method a complete record of all changes, on any piece, shown on any drawing and belonging to any machine, can be kept in compact form and in condition for immediate reference at any time.*

Changing Drawings Quickly.

Occasionally a big change in a drawing is necessary, and a print is wanted at once, or a different style of machine is to be made, having the major portion exactly the same as shown on the same previous drawing. The engraving, Fig. 5, illustrates this case. A rapid method for changing drawings is as follows: A tracing shown at *A* is on

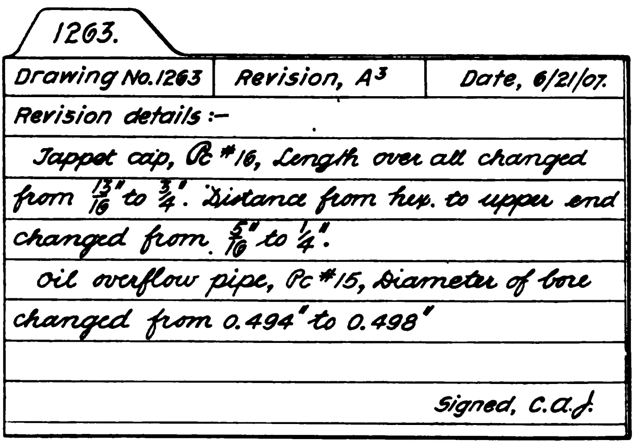

Fig. 4. Card System with Complete Record of all Revisions.

hand, but we wish a print showing this valve with a female pipe thread on both ends. From this tracing we then make a negative, shown at *B*, using a brown process paper. Now with a piece, or pieces, of brown opaque paper cover up all parts of this negative, *B*, that should not show on the required print. From this partially covered negative make a new print, *C*, again using brown process paper. On this new print, *C*, draw with india ink the special part of the new style valve and use the drawing *D*, thus obtained, as an ordinary tracing. The brown lines will print as well as the lines added in India ink.*

Other Time-Saving Methods for the Drafting-Room.

It frequently happens that the drafting room in a large plant gets behind in its work, and the shop requires drawings at once. It is not always advisable to send out pencil drawings, because, if there is to be

* Howard D. Yoder, Wadsworth, Ohio, October, 1907.

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any record kept of the work done in the drawing room, a copy of each drawing should be made before sending it out. Besides that, pencil drawings get mighty greasy and torn knocking around the shop or field. This not only makes it difficult for the workman to understand, but when it is finally returned to the drawing room, tracing it satisfactorily is not possible.

A quick and sure way to solve this problem is to trace the sheet upon tracing paper with a soft pencil. By making the drawing on the tracing paper, not only time but material is saved. This can be done in about one-third the time in which the drawing could be traced upon cloth. Care should be taken, however, to make the lines heavy, so that they will print well; curves and circles can be inked in more quickly and greater satisfaction be given than if they be put in pencil. If

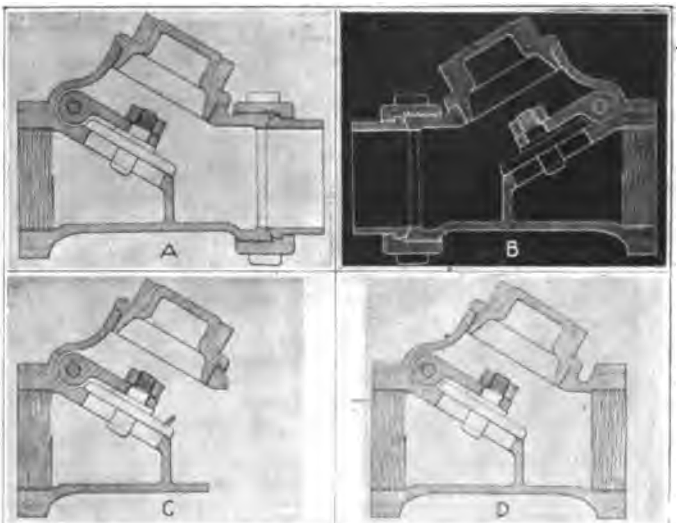

Fig. 5. Method of Changing Drawings Quickly.

strips of drawing paper be pasted on the edges of the paper, it will not tear.

A good method for sending out copies of small or detail work is to have a cross section sketch book about seven and one-half inches by ten and one-half inches. Next to each cross section page should be a number of thin blank pages, preferably three. After the sketch has been drawn in lightly, carbon paper can be inserted between the sheets and the sketch gone over with a glass point or hard pencil. Then these carbon copies can be torn out and sent to the shops, the cross section page being kept for record. These books are handy alike for reference and filing.*

Using Bond Paper Instead of Tracing Cloth.

Thin bond paper is in many respects better than tracing cloth for drawings to be blue-printed. It permits the making of a neat-looking

* F. R. Steuart, Sparrows Point, Md., August, 1906.

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drawing, pencil drawings can easily be made on it, a heavy pencil drawing will blue-print nicely from it, and it is cheaper than cloth. Besides, tracing cloth does not lay in drawers as well as bond paper. When tracings are creased across the lines of the drawing it makes it very bad when the print is taken. There is no trouble on this score with bond paper, and if for no other reason than this many have decided in its favor. Drawings can be inked more accurately by inking the original pencil drawing than by tracing, and it can be done more rapidly, which is another advantage for bond paper.

In many drafting-rooms, with the exception of drawings that must be repeatedly and frequently blue-printed, they make no tracings. Such as they make are made largely for the reason that they blue-print more rapidly. Another reason for tracing is, that if the original drawing were used to blue-print from too often, it would soon become worn out and unfit to make another copy from without much labor. So, for standard erecting plans, etc., it is preferable to make tracings on cloth and keep the original carefully, as in the course of years it is likely to need alterations.

From smooth and semi-transparent drawing paper one can get a first-class blueprint in about two and one-fourth times the number of minutes required for tracing on cloth. The drawing is laid out in pencil, then the useless lines wiped off with a piece of "artgum," which is about half-way between stale bread and velvet rubber in its cleansing properties. It leaves the surface in good shape for inking, and when the drawing is inked it is done. There is no tracing to be made.

Sketching Methods.

For sketching, there are two good methods of duplicating, and which one is used depends generally upon the number of duplicates required. In case but one or two are needed, make the sketch on fairly heavy cross-section paper with "Mephisto" colored copying pencils. The original sketch goes into the shop in this case, after being copied in an ordinary letter book with moistened pads. A second copy can be made on a loose sheet to send off in a letter if needed, but in case a second shop drawing is likely to be needed later, the other method should be used. This consists of a sketch made on thin cross-section paper with a stylographic pen loaded with Higgins' Eternal ink. This ink has sufficient body to yield a fair blue-print, though it is not thick enough to clog the stylo. If a more elaborate sketch is required than a rough free-hand sketch, lay it off in pencil and then go over the straight lines with the stylo, and the circles and arcs with the regular bow-pen or compass, which is much easier than trying to follow a true curved line with the stylo held in the hand.

From these stylo sketches a very good blue-print can be made, and working drawings put in shape for the shop in a very short time. The sketches are filed in indexed envelopes, and the copybook sketches in colored pencil are all indexed in the back of the book, so in either case it is not much of a job to locate an old sketch.

While it is well to make drawings and sketches fairly complete as

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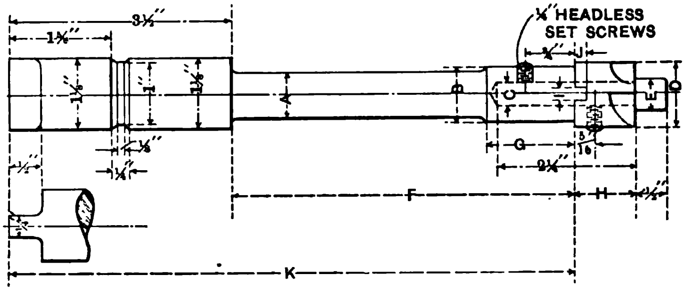
to minor details, at the same time a liberal use of the English language legibly written on the same sheet with the sketch goes a long way to prevent misunderstandings. If draftsmen themselves cannot all agree as to what a certain view of a drawing really represents (and such cases have been spoken of more or less in the technical papers), is it any wonder the man in the shop sometimes has to scratch his head more than twice to see things as the draftsman wants him to? Of course, a written direction can sometimes be read differently by two men, each giving it a meaning of his own, but if the sentences are clear



No.	A	B	C	D	E	F	G	H	I	J	K		
1	3/4"	3/4	5/16	3/4	3/8	4		1	3/16	3/16	7½		
2	3/4	3/4	5/16	5/8	5/16	4		3/4	1/8	1/8	7½		
3	3/4	7/8	3/8	1	3/8	4	1 3/8	1	3/16	3/16	7½		
4	1¼	1 3/8	½	1½	5/8	4	1 3/8	1¼	¼	¼	7½		

Machinery, N.Y.

Fig. 6. Example of Tabulated Tool Dimensioning.

and concise, as all technical writing should be, there is not much chance for trouble here.*

Tabulating Dimensions of Tools.

Fig. 6 shows an idea which will be very acceptable to all tool designing rooms, and also to all regular drawing rooms making tool-room drawings. The foreman of the tool-room wants sketches of a number of sizes of counterbores, having various sizes of pilots, and all fitting the same spindle. To save time in drawing, tracing, and blue-printing, rule up a sheet, as shown in the engraving. All dimensions which are constant, are given in their proper places on the drawing itself, while all those which differ for different sizes are indicated by letters, and tabulated. Vacant columns are left in the table of dimensions, and the tool-maker foreman can now supply the required figures for any

* E. R. Plaisted, Montpelier, Vt., December, 1906.

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new sizes he wishes to make, and turn the print over to his workman without waiting for another sketch. This idea can be used for a great variety of tools and fixtures, and we find it a great saver of time.

A system that in some respects is even better than the one mentioned above is in use in some drafting-rooms. Instead of inserting letters for variable dimensions and tabulating them, the spaces where the variable dimensions are to be given are marked on the tracing with circles filled in with india ink, thus producing on the blue-print a white spot on which the dimensions can be conveniently and plainly filled in with ink. The work of filling in the dimensions on the various blue-prints is hardly more than that of tabulating; but the advantage gained of having one distinct blue-print for each tool aids greatly in preventing mistakes in the shop, due to reading the dimensions wrong in the table.

Method of Enlarging or Reducing Drawings.

Very often it is desired to reduce or enlarge drawings, scroll designs, letters, maps, etc. This can be done to scale or by proportional dividers, but perhaps the simplest and quickest method is as shown in Fig. 7.

The only dimension necessary to lay off is the distance shown at *A B* which of course is the dimension desired for the reduced or enlarged copy. Fig. 7 shows the method for reducing which also applies for enlarging. It will be noticed that large and small rectangles are drawn with a diagonal line through each, and to these lines points are projected, and from there to the space where the copy is desired. The intersection of these lines are points of the copy.

Fig. 8 shows line *C* reversed from that shown in Fig. 7, which causes the copy to become reversed. This is especially convenient where it is desired to obtain a right and left view of any object. In making the third view of an object when detailing it is more convenient to plot it as shown in Fig. 9 than by the usual method of scaling each dimension as it can be done much quicker and with less chance of error. The diagonal line should of course be at 45 degrees in this case. In actual use, dashes cutting the diagonals are sufficient, instead of the construction lines shown in the engravings.

Titles and Border Lines on Drawings.

To be complete, a title for a drawing should include in order of importance:

1. The name of the machine to which it belongs.
2. The shop symbol for the machine, if any.
3. If not an assembly of the whole machine, the name, if there is a commonly used one, of the principal parts shown, as:

16-INCH ENGINE LATHE,
COMPOUND REST.

4. The words Assembly,
Assembly Detail,
Details,

according to which it is.

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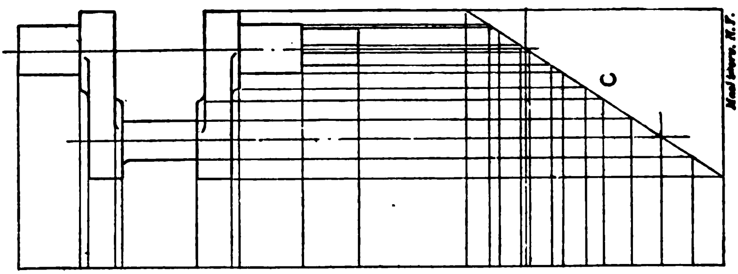

Fig. 8. Obtaining a Left-Hand View from a Right-Hand Drawing.

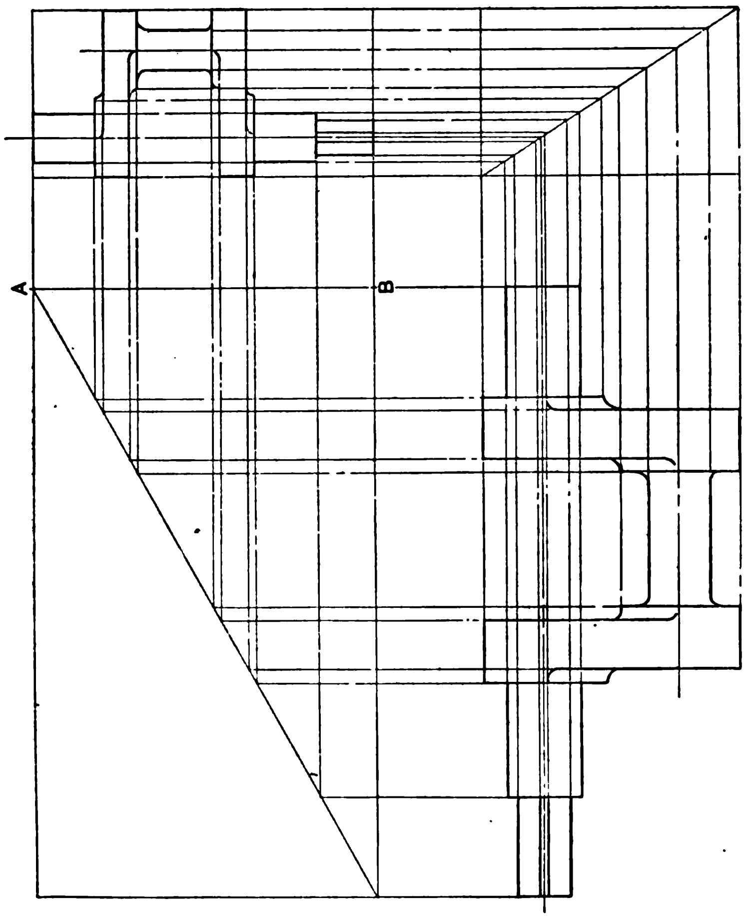

Fig. 7. Method of Enlarging or Reducing Drawings.

5. Name of firm.
6. Date when finished.
7. Name or initials of designer and draftsman and any other person who is responsible for the drawing. The first five items above must be drawn in heavy type, large enough to be instantly read. The last two will meet all requirements if they are simply legible.

Where parts are numbered or lettered, it would be a great convenience if alongside or just over the title, on a detailed drawing, a list of the parts which are shown on the sheet were given.

The length of time to be allowed for making the title and border lines depends, one might say, on about the same condition that a man's dress does. Border lines correspond to collars and cuffs and title to clothing in general. So we would no more leave a drawing without a title than we would go naked; but on the other hand, for shop use we would hardly dress our drawing up as for a reception. So for shop

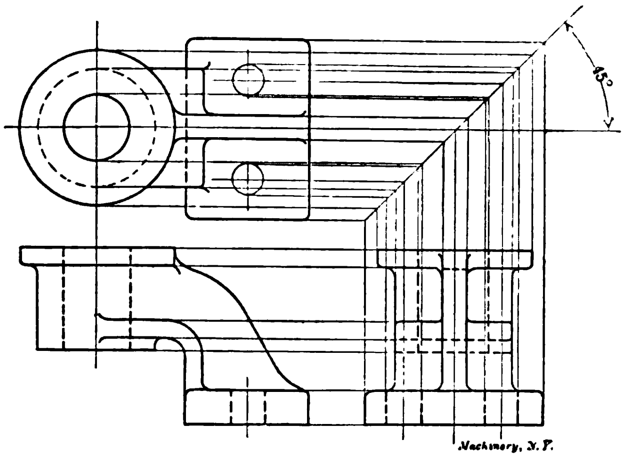

Fig. 9. Obtaining a Third View without Scaling the Drawing.

drawings, leave out border lines altogether and put in the title off hand, being certain that it is clear and legible and always in the same corner of the sheet as the drawings lay in the drawer. But if the drawing is going out to help sell a machine, have a neat, clear title and a plain, medium border.

Pattern-Makers Blue-Prints.

There is a chance for improvement in most drafting-rooms over the method which is in vogue regarding furnishing the machine department with a print containing many dimensions which do not in any way concern it, but which are used by the patternmaker only. When the pattern has been made and castings made from it, and finally, when the machine is finished and no alterations are to be made on the pattern, the pattern dimensions should be omitted from the machine shop print. It is sometimes customary to make two tracings to accomplish this if the piece is complicated, such as machine beds, etc., but

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the following method has the advantage of requiring but the one tracing. A finished tracing is made containing all dimensions both for the patternmaker and machinist. The dimensions for the machinist are inked in as usual, but the pattern dimensions are put in with a soft lead pencil. Several prints are taken from the tracing while in this condition, one furnished the pattern shop and as many filed away as desired. The lead pencil dimensions are then erased and the tracing is ready for making prints for the machine shop. In this way the patternmaker can readily understand and pick out his figures, and the machine shop print is kept free from unimportant dimensions which ofttimes cause considerable trouble.

Making Blue-prints from Typewritten Originals.

It is possible to make good blue-prints from typewritten originals, if one prepares the original properly. What is desired is a sharp, black copy that will make a good, clear blue-print, and the method of making these originals is as follows:

Upon a sheet of copy paper such as used for manifolding, lay a sheet of carbon paper face up; take a piece of tracing paper and lay it on the carbon paper and put them in the typewriter in this order so as to write on the tracing paper. This will give a copy on the front and back of the paper that will be strong enough to take blue-prints from if the following is closely observed. A good black ribbon is required, a new one if possible, but do not use one that has been used for any length of time and that will not strike up a good sharp black letter; a good carbon paper is also required. Use a new sheet every time, so as to get a good uniform letter; the carbon sheet can then be used on regular manifold work so that nothing is lost by it. The paper used for backing is the regular copying paper used for manifolding, but this is immaterial and can be made to suit different machines. The best paper for the original is a good tough grade of thin tracing paper, such Keuffel & Esser's "Series" brand, which takes the ink and carbon well and from which mistakes are easily erased without damaging the paper. As regards carbon paper that will give a good black uniform impression, the "Pilot" brand from the United Carbon Company suits this purpose exactly.

In making erasures, use a soft rubber, as an ink eraser will rough up the surface of the paper, and when the character is struck over, it will smut. To correct mistakes, turn up the paper and place at the back some smooth object (say a piece of glass or a celluloid triangle) and erase on the face; then place it on the face and erase on the back; the desired character can then be struck in.

Two photographic reproductions, slightly reduced, are shown in Fig. 10 of a small typewritten original, front and back, and the fact that it reproduces in this manner clearly is a guarantee that it will blue-print well, provided the writing is done on transparent paper such as any good tracing paper or cloth.

To Make Blue-prints from Heavy Drawing-paper Originals.

Oftentimes a draftsman finds it advisable to make blue-prints of cuts and drawings of which he may have a copy on printing paper or

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heavy drawing paper, etc. This is a long and tedious operation if conducted in the usual manner, but if the sheet to be printed is first given an application, or "coat," of gasoline or benzine, on the face side, and then printed, the result will be better and more quickly obtained than in the ordinary way.

The benzine will evaporate very quickly and leave the original in as good a condition as before the print was taken. Care should be taken that too much benzine is not used as it might spot the original if the blue-print solution were to be reached by any great quantity of benzine. The writer has, however, never experienced this difficulty, although many prints have been made by this method.*

Patching Drawings.

It is the opinion of a great many draftsmen that one can obtain a more accurate drawing in less time with a good grade of bond paper (say "crane" or any other good bond, about No. 18), than on tracing cloth; and if bought in large quantities the maker will tint this paper a soft color that is pleasing to the eye. The drawing is inked in on

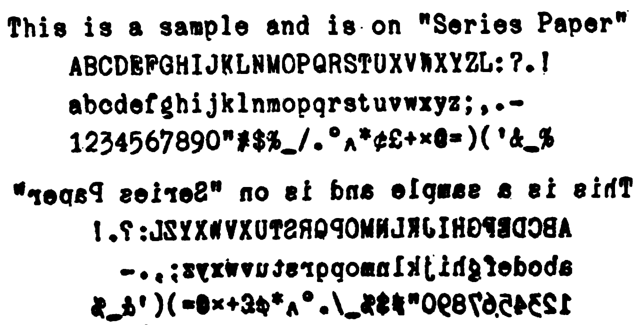
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Fig. 10. Reproduction of a Typewritten Original.

this paper and good blue-prints can be made from it. When making a drawing upon which a great deal of time is to be spent, such as the design of a new machine, the paper is dampened and the edges are glued to the drawing board. When dry it presents a surface which is smooth and which is not affected by any atmospheric changes and moisture that buckle and wrinkle any drawing paper under ordinary conditions. When making changes on these drawings a great saving of time comes in over tracing cloth. The pieces or parts to be changed are cut out and a new piece is pasted in and redrawn. The draftsmen soon get so expert at this that a piece 1½ inches square can be cut out and pasted in in less time than it could be erased on tracing cloth. The piece to be changed is first squared off and cut out with a knife. This is then laid over another piece 3-32 or ⅛ inch larger than the piece that has been cut out and the edges are glued all around with ordinary library paste; it is then pasted on the reverse side of the drawing. Fig. 11 shows an interesting case as far as the number of patches is concerned. Of course the blue-print will be rather light

* R. F. Kiefer, Sharon, Pa., April, 1906.

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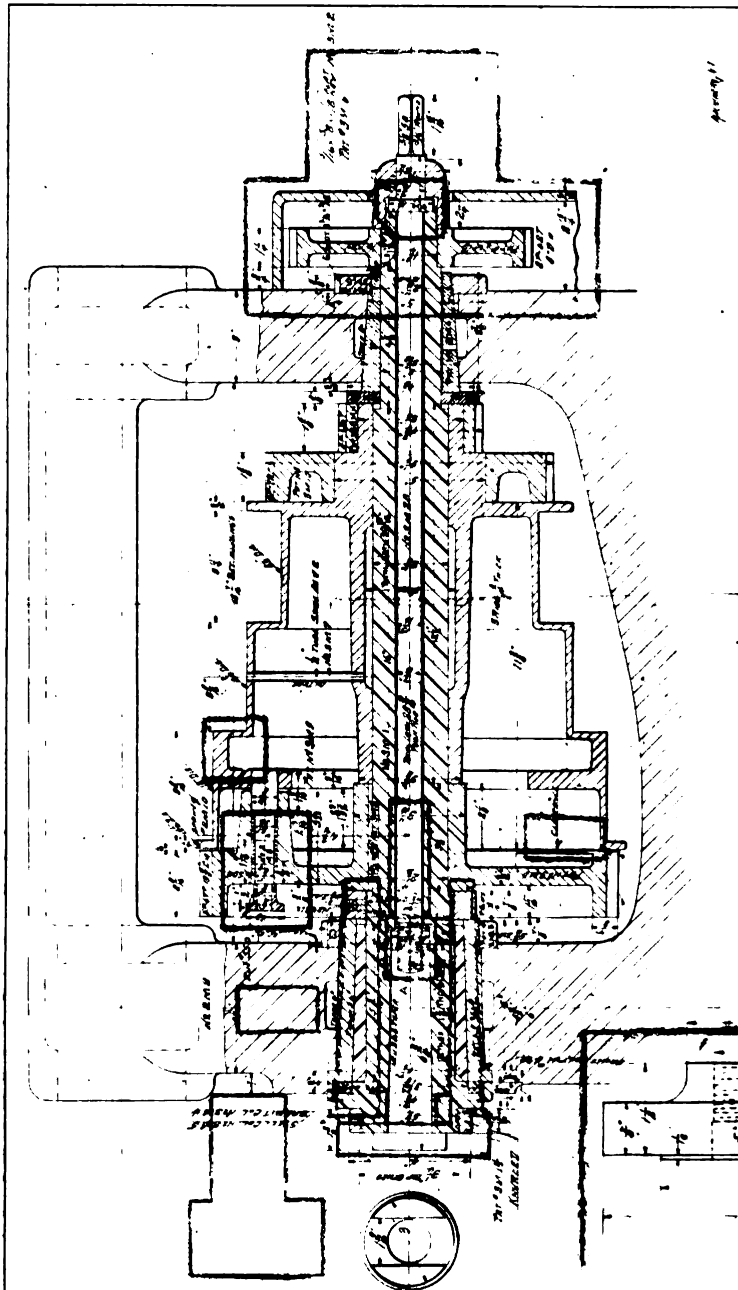

Fig. 11. A Patched Bond Paper Drawing.

around the edges of the patch, but this only indicates to the shop where the changes have been made, and is rather an advantage than otherwise. The engraving, Fig. 11, referred to, is a half-tone reproduction of a discarded drawing on bond paper patched in several places from time to time as changes were made. The work is so neatly done that it is scarcely discernible save when held between the eyes and the light. For this reason the patched effect would not be apparent in the cut, so each patch is penciled around the edge, the penciling being about the same width as the lap, which is only about 1-16 inch in most cases.

Another method used in another large drafting-room engaged in machine tool design, in which, also, bond paper drawings are largely used, deserves attention. Here, when drawings have to be patched, a sheet of clean paper is laid under the affected portion, which is removed by cutting with a sharp knife. The knife passes through both sheets of paper, thus providing a patch to fill the opening at the same time. To fasten this to the main body of the drawing, a piece of transparent paper spread with clear mucilage is used, if the patch is small. If of considerable size, the joint is neatly covered with thin strips of gummed transparent paper about 1/8 inch wide.

The advantage of this method is the smoothness of surface produced. The patch is flush with the main body of the drawing paper and the drawing instruments pass over the joint between the old and new portions without difficulty. It would be especially useful in cases where alterations are made on thick drawing paper. When neatly done with a sharp knife, the joint in such cases is almost invisible. The writer has employed it on tracings, where it worked very well, although it has been found that ordinary library paste is not permanently effective in making the joint. A good clear mucilage should be used.

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CHAPTER III.

DRAFTING-ROOM KINKS.

Ink bottle holders of various designs are constantly appearing in the technical press, and are always of interest to draftsmen. Two typical designs are here shown, which will be found to answer the requirements for serviceable ink bottle holders. The one shown in Fig. 12 is made for a Higgins drawing ink bottle, and is cheap, easy to make, light and neat, and will not tip over easily. This construction does away with toothpick wedges and strips of paper to keep the bottle tight. The round piece fitted in the bottom is held with four No. 4 wood screws, one inch long. If the bottle does not fit tightly,

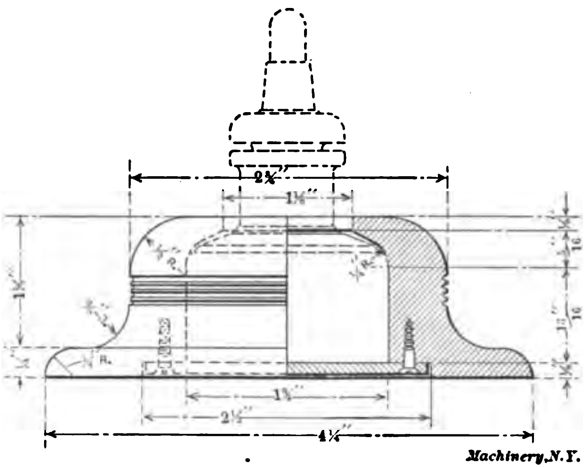

Fig. 12. A Safe Ink Bottle Holder.

put a piece of card underneath. Any pattern-maker can turn out a holder of this description in a short time, and with a coat of shellac it makes a very presentable appearance. The straight grooved sides make it easy and safe to handle. Three or four ¼-inch holes in the bottom, with old lead-pencil rubbers forced in, will make it non-slipping.*

Fig. 13 shows another type of ink bottle holder, used to advantage in some drafting-rooms. This serves a two-fold purpose, *i. e.*, to prevent spilling the ink, and also as a paper weight, which explains the thickness of metal used in the construction. The wood screws are screwed down just tight enough to make the heads a good sliding fit on the flange of the brass case, and when it is desired to remove the bottle, the case is slipped around until the heads of the screws come

* D. C. Turnbull, Mishawaka, Ind., September, 1905.

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into the large diameter holes, when the case can be removed from the oak bottom.*

Re-rolling Tracing Cloth to Prevent Curling.

A problem all, or nearly all, draftsmen are troubled with is the curling up at the edges of tracings, when filed away in drawers. It

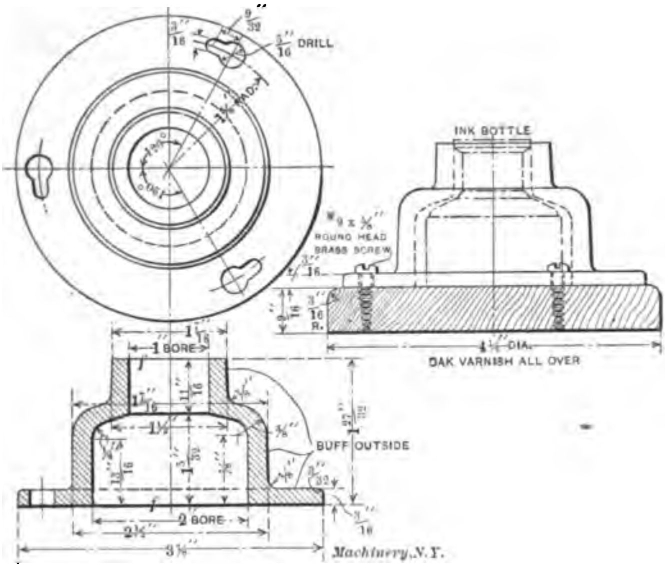

Fig. 13. Another Type of Ink Bottle Holder.

seems almost impossible to make them lie flat, and when put into the printing frame, the edges get folded down and make bad-looking edges on the print. Not very many draftsmen like to use the dull side of the tracing cloth, although the drawings made on this side will keep

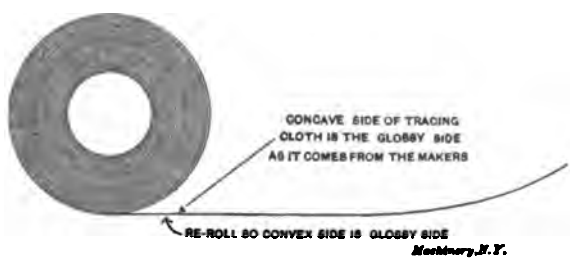

Fig. 14. Directions for Re-rolling Tracing Cloth.

flat much better than when the glazed side is used. The trouble with this curling up of the cloth is due to the fact that when manufactured, the cloth is rolled with the glossy side as the concave side, as shown in Fig. 14, which, of course, makes it curl. To overcome this, re-roll the cloth, putting the dull side in, and leave it lying for a month or

* W. O. Moody, Chicago, Ill., June, 1906.

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so before using. It will be found that there is a great improvement, and that the drawings made on the glossy side are now curled down on the edges rather than up. If one should make some sort of a rewinding device for that purpose, it would be found to be worth the trouble.

Celluloid Templets.

A very handy templet for drawing small curves and circles, where great accuracy is not required, may be made with a piece of thin

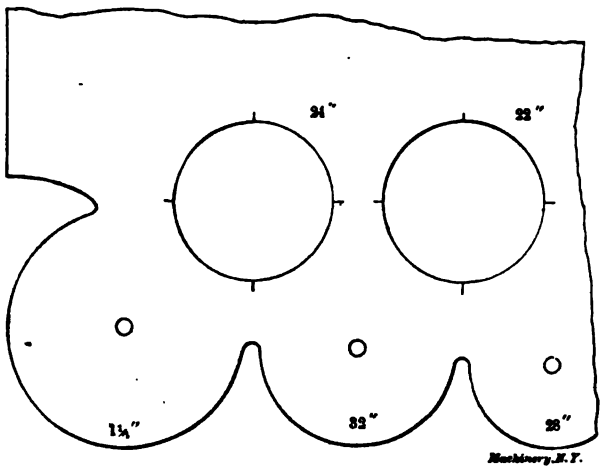

Fig. 15. Celluloid Templet for Small Curves and Circles.

celluloid. The templet, Fig. 15, is about 0.01 inch thick, and is a dark red tint, which is preferable to white. The circles are scratched deeply with bow dividers on one side, the center pushed through, and the same done on the other side until the piece can be broken out. By lightly scratching the center lines just beyond the outside of the circle

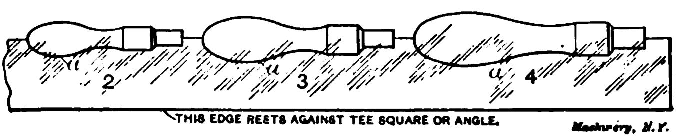

Fig. 16. Templet for Machine Handles.

before the center is removed, the hole can be located on center lines. On the edges, and particularly at the corners, are parts of circles with the center left and enlarged enough for the pencil point to go through to mark it. Sizes may be indicated by numbers giving the thirty-seconds of inch diameter. The holes should be cut slightly large to allow for the pencil point; outside curves, slightly small for the same reason. Similar templets may be made for the irregular shapes which are much used.*

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* E. W. Beardsley, Waterbury, Conn., September, 1905.

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Templet for Drawing Machine Handles.

The drawing of machine handles may be facilitated very much by the aid of a celluloid templet as shown in Fig. 16. If there is a forming tool for each size of handle it is an easy matter to make the templets, and thereby have the drawings the same outline as the formers. If filed out, the outline *a* of the lower half of the handles is scratched on the templets and is filled in with black wax.

Templet for Drawing Nuts and Bolt Heads.

A very handy templet for drawing nuts and bolt heads can be made as shown in Fig. 17. It is made of thin transparent celluloid and facilitates what is perhaps the most commonly repeated work to be done on a drawing. Lines are ruled on the templet, which, when

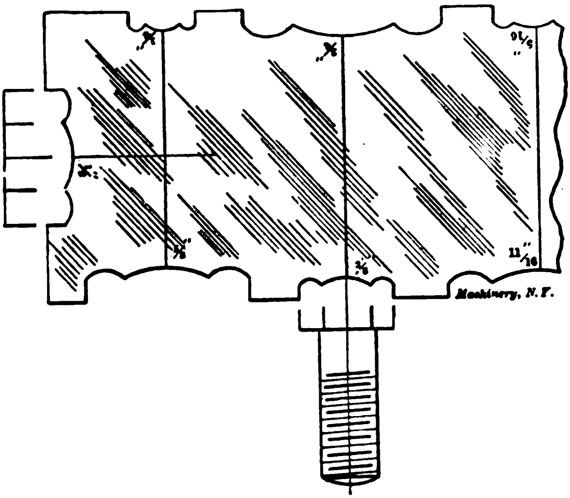

Fig. 17. Templet for Nuts and Screw Heads.

used, are placed over the center line on the drawing, and the curved outlines can then be drawn.

Erasing Shield.

As an erasing shield nothing is equal to a very thin piece of sheet steel with slots cut with a small cold chisel and filed smooth. It should be about 0.003 thick. The slots will not wear perceptibly large during many years of use.*

A Celluloid Protector for Drawings.

It is a well-known fact that in laying out and designing any mechanism of a complex nature several erasures occur which roughen the surface, and the paper then becomes dirty much quicker than otherwise. Assuming that the upper right or left corner of a sheet has a side view which has been worked on for several days, and it is now required to draw a front view of the same object, it is, of course, neces-

* E. W. Beardsley, Waterbury, Conn., September, 1905.

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sary to transfer the center lines, make comparisons, take off measurements, etc., from the side view, and for this reason it is certainly proper to have the side view constantly in sight.

To cover the part already finished with a piece of paper is all right in its way and is better than nothing, but it is annoying to always have to lift the paper covering in order to see the sketch. This trouble may be overcome in a very simple manner; the covering consists of a sheet of celluloid (the thinnest transparent, about 0.010 inch thick) which will cover one-half of the drawing. The price of such a sheet, in this case about 12 × 18 inches, is only 50 cents, and it will save its cost over and over again. It is self-evident that by using this sort of shield the T-square in sliding across the sketch covered cannot injure it, and furthermore, if the celluloid is bent at its lower edge as indi-

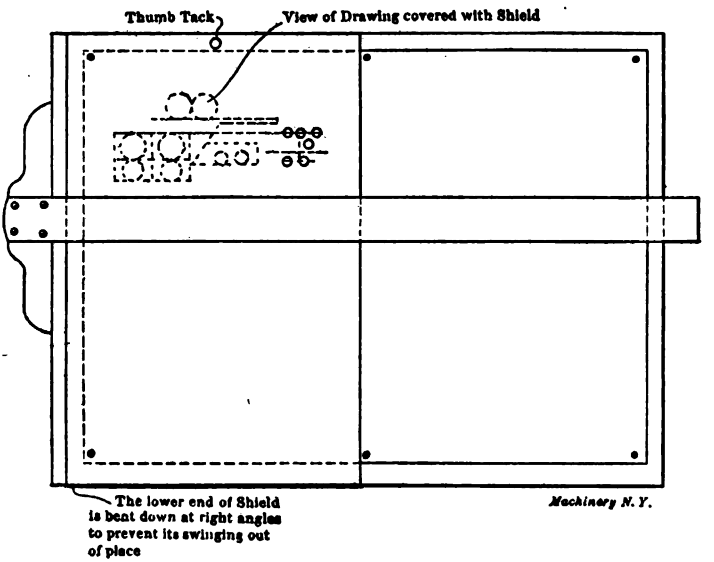

Fig. 18. Celluloid Protector for Drawings.

cated in Fig. 18 and a single thumb tack used at the upper end it is an easy matter to shift the shield from left to right or *vice versa.* If it should be necessary to get at both views occasionally it is not even necessary to remove the tack; simply lift the T-square and swing the celluloid shield so it hangs down from the back of the drawing board.

It is a pleasure to work on a drawing with a shield of this description as the lines can be seen practically as well through the shield as if it were not there; likewise measurements can be very readily taken off.*

Hanger for Reference Drawings.

It is very often that a draftsman is required to use another drawing for reference which he must continually consult, and if it is a large

* Robert A. Lachmann, Chicago, Ill., May, 1906.

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sized drawing it is generally kept on the consulting table beside his drawing board. This compels him to keep forever stretching to see it or constantly getting on and off his stool. The following scheme will prove very useful for avoiding the difficulties referred to.

Fig. 19 will make the device clear without much explanation. *A* is a pine stick ⅜ inch by 1½ inch and about 40 inches long. Onto this stick are screwed three ordinary spring clothes pins. Two strings are fastened to the stick and pass up to two screw-eyes placed in the ceiling the same distance apart as the strings on the stick. Two more screw-eyes are placed in the ceiling near the wall, or any out-of-the-way place, and through these the strings pass down to a weight. The holder is placed parallel to the drawing board, about one inch back of it, and holds the drawings where they can be easily referred to by the drafts-

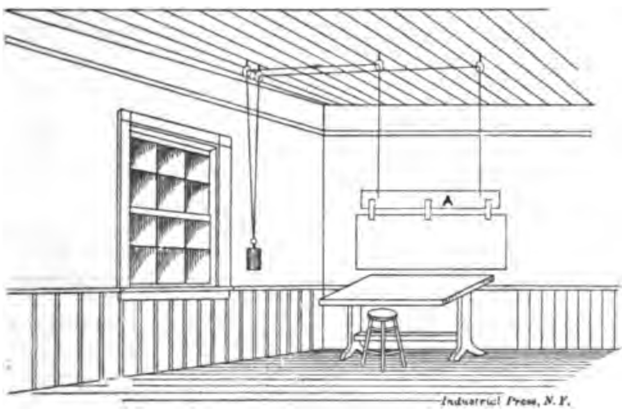

Fig. 19. Holder for Reference Drawings.

man at the same time as they can be raised or lowered to suit his convenience.*

A Time Saving Drafting Kink.

A great saving in time, which even well systematized drafting rooms seem to overlook, can be effected by putting one or two tracers to work cutting up a quantity of standard sizes of tracing cloth, drawing the border lines around them, stamping them with the standard marking, simply leaving out the name of the piece, date, and draftsman's initials, which, of course, are filled in when the tracing is made. This saves every man going and cutting up a piece of tracing cloth each time he is to make a tracing. It makes it very convenient for the tracers, as they can get the size sheet they want, tack it down, and trace in the drawing right away, without bothering with cutting to size, measuring the outline, and putting on the borders and title stamp. This plan also saves a great deal of waste of tracing cloth.**

* Ernest W. Duston, November, 1902.
** F. L. Engel, New Britain, Conn., February, 1908.

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Shading Drawings.

The time required in the drafting room to make simple shade lines to indicate the lower and right-hand edges of solids is time well spent. When a drawing is properly dimensioned, every distance that the workman will need to know being given, there can be no objection to such shade lines other than the time required to make them. The time dwindles into insignificance when the clearness and attractiveness of drawings properly shaded is taken into account. The drawing that possesses snap and life is an inspiration, and it is a real pleasure to work from it. The unshaded drawing has a weakness and dullness that is discouraging. It requires more mental effort or concentration to understand it, and the chances for error of interpretation are greater than with shaded drawings. Shade lines for rounded surfaces, however, are not generally desirable. Unless such shading is *very well* done, the result is decidedly amateurish in appearance. If it is well

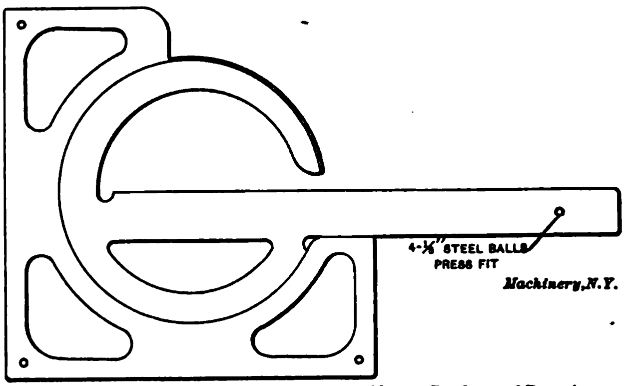

Fig. 20. Method of Raising Protractor Above Surface of Drawing.

done the effect is, of course, pleasing and the subject stands forth clearly, but unless for some very special cases, it should not be favored.

Raising Draftsman's Protractor Above the Surface of the Drawing.

Anyone who uses a Brown & Sharpe draftsman's protractor is familiar with the way it soils the drawing, when put to any extended use, in spite of the utmost care on the draftsman's part. Fig. 20 shows a good scheme for raising the surface of the protractor slightly from the drawing. Four steel balls are made a press fit in the device at about the positions shown, the balls being about ⅛ inch diameter, projecting an equal amount on both sides, thus giving the same results no matter which side of the instrument is up. This addition to the instrument in no way affects its accuracy, and, as the balls bear only on four points, they rub less dirt into the drawing than an ordinary triangle.*

Tool for Spacing Bolt Holes.

In drawing flanges, cylinder heads, etc., where a number of holes are shown on a given pitch circle, it is usually desirable to have them lo-

* M. R. Kavanagh, Detroit, Mich., January, 1908.

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cated correctly on the drawing, for the sake of appearance at least, even if not really important otherwise, and for this purpose the tool shown in Fig. 21 was designed. It is a very convenient instrument, made from transparent sheet celluloid, about 1/64 inch thick. The method of procedure when using this tool is to draw the pitch circle, and then place the instrument over it, with the point *A* on the center. One point of the dividers is placed on the line *A O*, at the point where it intersects the pitch circle, and the other point of the dividers is set at the intersection of the pitch circle and the radial line marked with a number corresponding to the required number of holes in the whole pitch circle. This gives the correct spacing for the number of holes required. If reasonable care is used in the construction of this tool, it will be found to give very close results. The design, of course, can be varied to suit special requirements. The one shown in Fig. 21 was

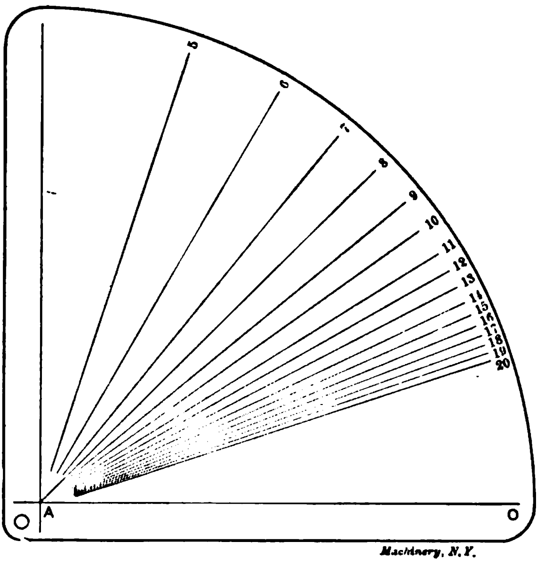

Fig. 21. Tool for Spacing Bolt Holes.

made by laying off the correct angles on the circle having 18 inches radius, and scratching the radial lines with a needle point on the celluloid, after which drawing ink was rubbed into the scratches so as to make them show plainly.

Drafting Tool for Sketching Ratchet Teeth, Etc.

Fig. 22 shows an extremely useful little drafting tool which any mechanic can make in a short time. One can use the slotted blade from a Starrett universal bevel (the blade which has the ends beveled to 30 and 60 degrees), also the clamping screw and nut. The other part is made from a piece of 3/4 × 1/16 inch cold-rolled steel and a piece of 1/4 inch drill rod. It is used for drawing ratchet teeth or any

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similar work. If one has the time, a chuck or clamping device for holding a needle point might be made instead of the solid stem and

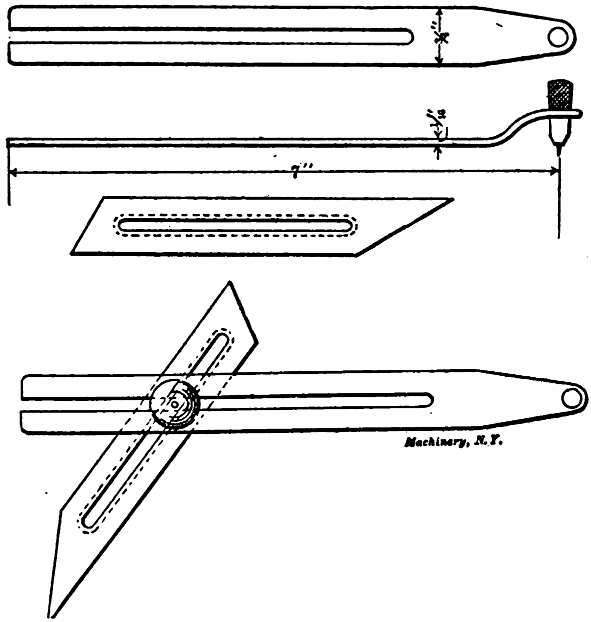

Fig. 22. Drafting Tool for Ratchet Teeth.

point shown. The bevel, without this attachment, is quite handy in drawing as well as machine work.

Simple Device for Drawing Elliptic Curves.

Fig. 23 is a simple device for drawing elliptic curves which is not without interest and value. It has the merit of being quickly adjusted

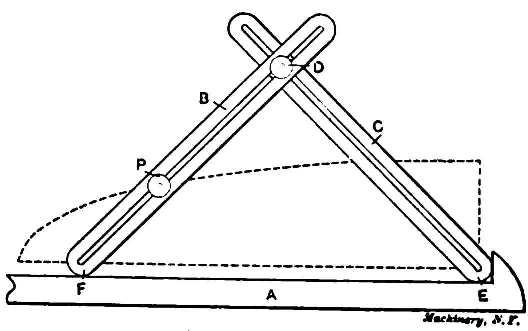

Fig. 23. Device for Drawing Elliptic Curves.

for any required major or minor axes within the limits of the instrument, but has the defect that only one-quarter of the ellipse can be drawn at one setting. *A* is a straightedge with a stop or projection at

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E. *B* and *C* are slotted rods, each having the lower ends truly rounded to perfect semi-circles for the obvious reason that any other shape would distort the shape of the curve. The rods are held together by a joint similar to that used in proportional compasses, that is, the joint may be fixed on the bars in any required proportion, and still allow them to swing on the joint as a pivot. At *P* is the holder for the pencil or scriber, and this also is adjustable in the slot. When drawing an ellipse, the distance *FP* is made equal to one-half the desired minor axis, and *PDE* must be made one-half the length of the major

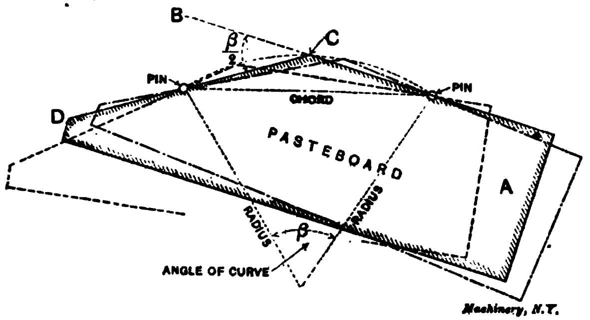

Fig. 24. Plotting a Circular Arc having an Inaccessible Center.

axis. Fig. 23 shows clearly how the curve is drawn, and it will be noted that a short section of the quarter ellipse must be drawn in

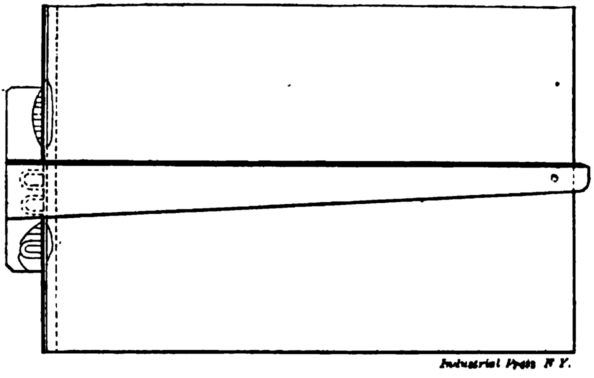

Fig. 25. Magnetic T-Square Attachment.

free-hand, since the pencil at *P* will interfere with the bar *C* when near to the vertical position.

Method of Plotting Curve from Inaccessible Center.

Two pins are stuck into the drawing board at the end of the required curve, and a piece of pasteboard, *A*, Fig. 24, is cut so that the exterior angle *BCD* is one-half that of the required arc. Thus, if the angle subtended at the center is 40 degrees, angle *BCD* would be 20 degrees. To lay off the curve, the pasteboard is pressed against the pins, as shown

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in the sketch and the points of the curve are marked off from vertex *C*. The shade lines show the position of the pasteboard templet, and the dotted lines, other positions that it occupies as the points are laid down.

Drawing Board and T-Square with Magnetic Attachment.

An English novelty in drafting room appliances, shown in Fig. 25, is a T-square having a number of small horseshoe magnets mounted in the head and so arranged that their poles come in contact with an iron strip attached to the left-hand end of the board, the idea being to hold the T-square to its right-angle position, by magnetic attraction alone. This, of course, allows the draftsman to use both hands at his work more advantageously. The magnets are said to oppose little resistance to movement of the blade to any position parallel with its normal position, but considerable force is required to displace the blade from its right-angle position. A drawing board with this attachment may be used in a vertical, sloping or horizontal position with equal facility.

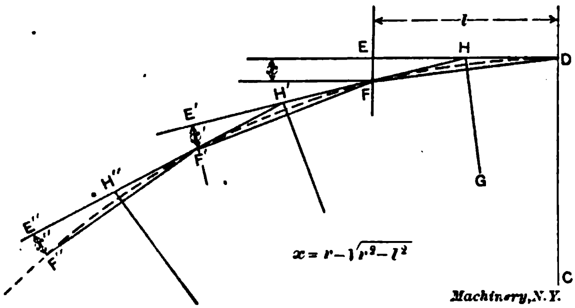

Fig. 26. Laying Out a Circular Arc without a Center.

To Construct a Curve by Points.

Fig. 26 and the accompanying description will be found useful for laying out a circular curve having an inaccessible center or one whose radius is too long to be accommodated on the drawing-board. When the radius r is given, and any cord l is assumed, to construct a curve by points, proceed as follows: First find x by formula in Fig. 26. Draw line *CD*, and at right angles to it the line $ED = l$. Through *E* draw the perpendicular *EF*, and make it equal x (marking the first point in the curve) at *F*. Join *FD* and bisect it in *G*; through *G* draw the perpendicular *GH*, intersecting the line *ED* in *H*, and through *F* and *H* draw the indefinite line *HFH'E'*, making *FE'* equal to l. Erect the perpendicular *E'F'* equal to x, and locate *F'*, which is the second point of the curve; proceed in the same manner for other points of the curve desired.*

Stamping Tracings.

To stamp names or headings on tracing cloth, use a rubber stamp with the ordinary red, blue or green ink found in ink pads. Before

* Geo. H. Waltman, Hokendauqua, Pa., December, 1899.

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the ink has a chance to dry, sprinkle lamp-black over it, using an insect powder sprayer for the purpose. When dry, brush off with a piece of chamois skin. Stamping done in this manner will be found light-proof.

Pencil Sharpener.

Fig. 27 represents a pencil sharpener or pointer. As seen from the illustration, it consists of a piece of thin spring brass bent to the form shown and fastened under the table near the edge by two round-head wood screws. The sandpaper is held between the table and the brass spring. To sharpen the lead of the pencil, slip the point between the leaves of the sandpaper and move it back and forth.*

Making Blue-Prints Without a Frame.

Blue-prints of small size can be made fairly conveniently without a blue-printing frame. An ordinary window can be used if the sun shines through it, and a blue-print can be made in any window. An ordinary thick bath towel is placed behind the print, the tracing being placed against the glass. The towel should be folded into two or three thicknesses and arranged so that no wrinkle or uneven part lies against the print. A small drawing board may then be placed against the towel, but it is better to tack the towel at its corners to the board.

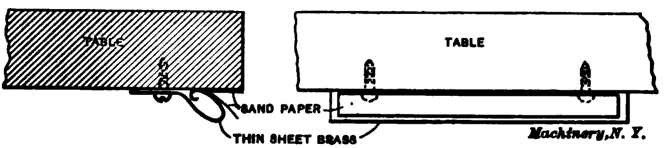

Fig. 27. Pencil Sharpener.

It is also advisable to attach the tracing to the printing paper by small gummed stickers, to keep them in the proper position and to prevent sliding. Ordinary stickers cut into narrow strips will be sufficient, and need engage only a narrow surface on both the paper and the tracing, to serve the purpose. The print can then be frequently looked at without disturbing the relation, and can be easily torn off without injuring it or the tracing. Fragments of the stickers are easily scraped off. The printing paper and tracing can be held by a blank projecting edge against the window while the towel and board are being pressed against them. Any suitable means for holding the board in place may be used.

Another improvised printing outfit used by the writer consists simply in spreading the towel evenly upon the floor where the sun can strike it, placing the printing paper and tracing upon it, and then merely covering these with a thick plate of glass. When first laid down, the glass may be pressed downward with considerable pressure, after which its weight alone will be sufficient to keep the paper smooth. This will be the case if two or three thicknesses of bath towel are used. This plan works perfectly for sheets 10 × 15 inches and below. Larger prints could doubtless be made in this way if weights were put upon the corners of the glass.

* John B. Sperry, Aurora, Ill., October, 1908.

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Another very practical way of making small prints is to use a smooth board with a slightly curved face. A tack placed in each corner of the tracing and print will cause them to snugly lie against the curved surface of the board, making a sharp and clear print. No glass is needed.*

A Blue-Printing Kink.

In the winter months it is hard to judge the time required to make sunlight blue-prints, and it is a very good idea to make a scale of colors with which to compare a test piece of paper. Take a piece of blue-print paper four inches wide and ten inches long. Cover all but one inch with a heavy piece of paper. Expose this one inch for say one minute in a mild light, then move the cover so as to expose one inch more for one minute, which exposes the first inch two minutes, and so on, inch by inch, until all ten inches have been exposed, and the first division has been exposed ten minutes, which in this light, for instance, produces a perfect print. Now wash the paper thoroughly, and when dry you have a scale with which to compare

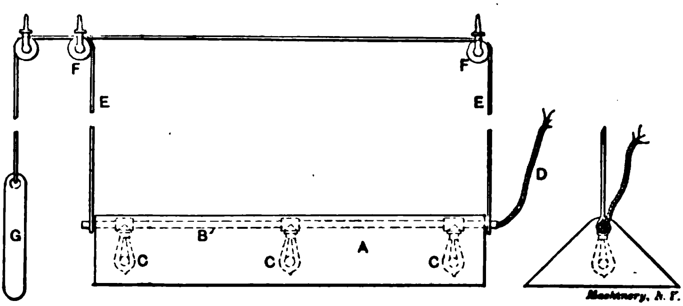

Fig. 28. Adjustable Light Holder for the Drafting-Room.

other prints. Figure out the relative times that would have been required to bring each color to a perfect blue-print, by dividing the whole time by the time exposed, and mark this number on the color, this number to be used as a multiplying factor. Now, in order to get a perfect print, put out the tracing you wish to copy with a small piece of blue-print paper, and leave it, say two minutes, until it would be printed some, but not enough. Then wash, dry with blotters and compare with the scale sheet. Say it was like the fourth color from the bottom. The factor would be 2½. Multiplying this by the two minutes equals five minutes, the time required to make a perfect print in that light. I am pleased with the uniform results obtained.**

The Lighting of Drafting-Rooms.

The lighting of drafting-rooms is a subject often discussed without reaching a satisfactory conclusion for all cases; each one must necessarily be governed by local conditions. The apparatus described in the following is cheap and efficient, and for artificial illumination, when

* C. E. Burnap, Battle Creek, Mich., September, 1908.
**Austin G. Johnson, Two Harbors, Minn., November, 1908.

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electricity is to be used, has proved to be entirely satisfactory in actual practice. Referring to the illustration, Fig. 28, the shade *A* was made of tin, painted dark green outside, and white inside, through which was passed a pipe *B* with T-connections at lamps *C*. The wires *D* were drawn through the pipe and connected with ordinary 16-candle-power incandescent lamps which were fastened at these T-connections. The whole thing was suspended from the ceiling by cords *E* drawn

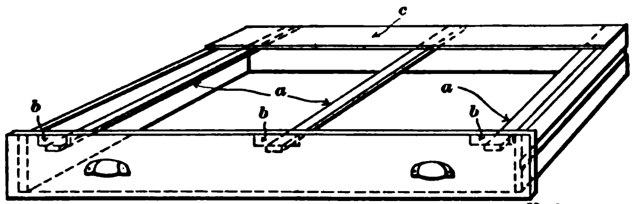

Fig. 29. Practical Drawer for Filing Drawings.

over pulleys *F*. The cords were attached to the counter-balance *G*, and to the shade as shown, allowing the lights to be raised entirely out of the way when not required, or lowered directly over each board sufficiently to shield the eyes of the draftsman and reflect the light over the entire board. The length and number of globes will, of course, depend upon the length of board.

This arrangement may be modified to meet the requirements of the more refined tastes willing to pay for elaborate furnishings. How-

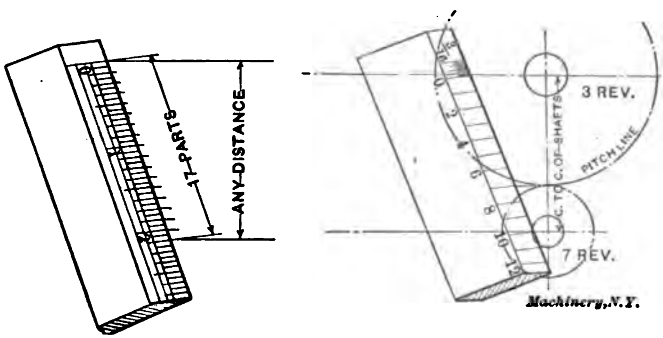

Fig. 30. Convenient Use of Ordinary Scale for Dividing Dimensions in Equal Parts.

ever, the leading features should remain the same: A suspended individual light of wide range, adjustable for height, while permitting the full swing of beam compasses or drafting machine, without encountering supporting brackets. It casts no shadows and protects the eyes of the worker at the board.

Drawer for Drawings.

A simple and cheap method of making drawings lie flat in drawers is shown in Fig. 29. It consists of three blocks *b* fastened to the inside of the front of the drawer, a back top *c*, and three slats *a*, which are slipped in under *c* and *b*. These slats keep the drawings from

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curling up and catching in the drawer case. By using a drawer of this description, the capacity is increased from three to four times.*

The Use of the Ordinary Rule for Dividing.

The operation of dividing a line into any number of equal spaces is a very simple one, and one of the first exercises in geometry, but however familiar men might be with its execution, one rarely uses the rule for this purpose. When it is required, for instance, to draw a certain number of lines or a certain number of threads per inch, in a given space, sometimes the length of the single spaces will be figured, or a diagonally drawn line will be spaced off with the dividers and the spaces transferred with the triangles.

The shortest way is to use the ordinary or draftsman's scale. Use the graduation, nearest suitable to the eye, place the zero mark on one end of the space to be divided, and the number corresponding to the parts required on the line marking the end of the space, or its continuation, mark off points at every graduation, and with triangle or T-square draw parallel lines through these points, as shown in Fig. 30. At the left is shown two lines, the distance between which is to be divided into 17 equal parts. In this case the regular scale of an ordinary rule is used; after the points are marked off, parallel lines are drawn

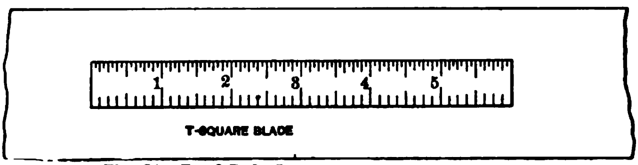

Fig. 31. Steel Rule Recessed into T-Square Blade.

through them. This use of the rule for dividing without going to the trouble of figuring is applicable in a great number of cases. The distance center to center of two shafts is given, and a transmission of spur gearing of the ratio 3 to 7 between them is wanted. The pitch circles can be drawn immediately without figuring their diameters or number of teeth. Draw the distance between centers to scale, hold the zero mark of rule on a line through one center and the graduation 10 ($= 3 + 7$) to a parallel line through the other center; the graduation 7 marks the line tangent to the two pitch circles, as illustrated in the right-hand view.**

Steel Rule Recessed into T-Square.

The following little kink may be of some value to draftsmen. A 6-inch flexible Brown & Sharpe scale is laid into the T-square blade, which has previously been recessed somewhat to fit the scale, as shown in Fig. 31. Shellac is used to hold the scale in place in its recess. The dividers and compasses may be set directly from this scale, and the annoyance of looking for an ordinary loose draftsman's scale is avoided. Scales graduated to 32ds and 64ths of an inch are convenient for fine work.***

* John B. Sperry, Aurora, Ill., July, 1908.
** M. Joachimson, New York City, July, 1908.
*** John Coapman, Rochester, N. Y., June, 1908.

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No. 34

CARE AND REPAIR OF DYNAMOS AND MOTORS

CONTENTS

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CHAPTER I.

DYNAMO AND MOTOR TROUBLES.

A number of small volumes have been written on the care of electrical machinery, particularly dynamos and motors. Most of these books are very useful in assisting the operator in the proper maintenance of the apparatus and the discovery of the causes of faults and breaks which are constantly liable to occur. Almost any given symptom of distress in a dynamo or motor, however, may be due to a number of different causes. This fact, together with the lack of method in the arrangement of some of the books dealing with the subject, often handicaps the beginner in locating the particular fault to which any given trouble is due.

Roughly speaking the various diseases to which dynamos and motors are subject may be placed in six general classes. First, sparking of the brushes; second, heating of the parts; third, noises; fourth, variations in speed; fifth, miscellaneous derangements peculiar to motors as distinguished from dynamos; sixth, miscellaneous derangements peculiar to dynamos and generators as distinguished from motors. It is again possible to divide each of these major symptomatic indications into minor ones. The sparking of the brushes, for instance, may be due, first, to faults of the brushes; second, to faults of the commutator; third, to excessive currents in the armature; fourth, to faults in the armature. Each of these divisions may be again subdivided and an appropriate individual remedy indicated.

To make this clearer, the arrangement in the present chapter has been adopted for stating precisely, and in a limited space, the troubles met with and the remedies to apply. As an illustration we will suppose that the armature of a motor becomes dangerously hot after running for a time. The methodically arranged table or chart which follows is consulted and under the heading of "heating of parts" the sub-head "armature" is found. There are seven different causes given here for heating of the armature. It may be due to overload of the motor, to a short circuit due to carbon dust, etc., on the commutator bars, or it may be caused by a broken circuit, a cross connection, moisture in the coils, eddy-currents in the core, or heat conveyed from a hot box or journals through the shaft. Each of these seven causes may be investigated in turn. For instance, it may be found that the armature core is warmer than the winding which surrounds it. If this is the case, the trouble is due to eddy-currents in the core, or to heat conducted through the shaft from a hot box. If the latter, the shaft will of course be hotter than the armature, and the bearings still hotter than the shaft. If the trouble is due to eddy-currents the armature will be found to be made of solid metal, or to be not sufficiently laminated. In either case the trouble is readily discovered.

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There are two advantages in using a chart of this kind. In the case of trouble with a motor or dynamo, a large text-book is generally too voluminous to be easily used, and, quite likely, is not well enough arranged to permit a quick diagnosis. Then, again, after a person has carefully read over such a work several times, he will still find the chart very acceptable, as a guide which will show him where to look and what to do—something that can be glanced over quickly and can be readily found, which will outline the proper course to pursue. The trained mind will then quickly recall from the larger work the details of the proper method of procedure.

Sparking at the Brushes.

FAULTS OF BRUSHES.

1. *Not set diametrically opposite.*—Should have been set properly at first, by counting bars, or by measurements on the commutator. Can be done if necessary while running; move rocker until brush on on-side sparks least; then adjust other rockers so they do not spark.
2. *Not set at neutral points.*—Move rocker back and forth slowly until sparking stops.
3. *Not properly trimmed.*—Brushes should be properly trimmed before starting by bending back and cutting off loose wires or ragged copper. If there are two or more brushes, one may be removed and retrimmed while running. Clean with benzine, soda or potash (alcohol or ether for carbon); then file or grind to standard jig and reset carefully. For instructions for setting see 1, 4, and 38.
4. *Not in line.*—Adjust each brush until bearing is on line and square on commutator bar, bearing evenly the whole width.
5. *Not in good contact.*—Clean commutator of oil and grit. See that brushes touch. Adjust tension screws and springs to secure light, firm, and even contact.

FAULTS OF COMMUTATOR.

6-7. *Rough; worn in grooves or ridges; out of round.*—Grind with fine sand paper on curved block, and polish with crocus cloth. Never use emery in any form. If too bad to grind down, turn off true in a lathe or preferably on its own bearings, with a light tool and rest and a light cut, running slowly. Armature should have 1/16 inch to ⅛ inch end motion when running, to wear commutator evenly and smoothly. See also that foundation is level. If there is no end motion, file or turn ends of boxes or shoulders on shaft to provide end motion; then line up shaft and belt, so that there is no end thrust on shaft, but so that the armature plays freely endways when running.

8. *High bars.*—Set "high bar" down carefully with mallet or block of wood, then clamp end nuts tightly, or file, grind, or turn true. A high bar may cause singing. If so apply stearic acid (adamantine), candle, vaseline, or cylinder oil to commutator and wipe off; only a trace should be applied. Move brushes in and

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out of holder to get a firm, smooth, gentle pressure, free from hum or buzz.

9. *Low bars.*—Grind or turn commutator true to the surface of the low bars.

10. *Weak magnetic field.*—Broken circuit in field coils, or short circuit in field coils; repair if external, rewind if internal. Machine not properly wound or without proper amount of iron; no remedy but to rebuild it.

Excessive Current in Armature of Generator.

11. *Excessive load.*—Reduce number of lamps and load.
Ground and leak from short circuit on line.—Test out, locate, and repair.
Dead short circuit on line.—Dead short circuit will or should blow safety fuse. Shut down; locate fault and repair before starting again, and put in a new fuse.

Excessive Current in Armature of Motor.

11a. *Excessive voltage.*—Use proper current only, and with proper rheostat and controller, and switch.
Excessive amperes on constant current circuit.—See that controller, etc., are suitable with ample resistance.
Friction.—Reduce load on motor to its rated capacity or less. Clean with benzine. Bearings may be loose or worn out; perhaps new bearings are needed. For bearings out of line, see 30.
Too great load on pulley.—See that there is no undue friction or mechanical resistance anywhere.

Armature Faults.

12. *Short circuited coils.*—(*a*). Remove copper dust, solder, or other metallic contact between commutator bars. (*b*). See that clamping rings are perfectly free, and insulated from commutator bars, and that there is no copper dust, carbonized oil, etc., to cause an electrical leak. (*c*). Test for cross connection or short circuit, and if such is found rewind armature to correct. (*d*). See that brush holders are perfectly insulated, with no copper dust, carbon dust, oil or dust, to cause an electrical leak.

13. *Broken coils.*—(*a*). Bridge the break temporarily by staggering the brushes until machine can be shut down (to save bad sparking) and then repair. (*b*). Shut down machine if possible, and repair loose or broken connection to commutator bar. (*c*.) If coil is broken inside, rewinding is the only sure remedy. May be temporarily repaired by connecting to next coil, across mica. (*d*). Solder commutator lugs together, or put in a "jumper," and cut out, and leave open the broken coil. Be careful not to short circuit a good coil in doing this.

14. *Cross connections.*—Cross connections may have same effect as short circuit; treat as such, see 12. Each coil should test complete, with no cross and no ground.

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Heating of Parts.

ARMATURE.

15. *Overloaded.*—Too many amperes, lights, or too much power being taken from machine. See 11A.
16. *Short circuit.*—Generally dirt, etc., at commutator bars. See 12.
17. *Broken circuit.*—Often caused by a loose or broken band. See 12, 13, and 14.
18. *Cross connection.*—Often caused by a loose coil abrading on another coil or core. See 12 and 14.
19. *Moisture in coils.*—Dry out by gentle heat; may be done by sending a small current through, or causing machine to generate a small current itself, by running slowly.
20. *Eddy currents in core.*—Iron of armature hotter than coils after a run: faulty construction. Core should be made of finely laminated insulated sheets. No remedy but to rebuild.
21. *Friction.*—Hot boxes or journals may effect armature. See 25 and 33.

FIELD COILS.

22. *Excessive current.*—When shunt wound decrease voltage at terminals by reducing speed; increase field resistance by winding on more wire, finer wire, or putting resistance in series with fields. When series wound, decrease current through fields by shunt, removing some of the field winding, or rewind with coarser wire. Excessive current may be caused by a short circuit, or by moisture in coils, producing a leakage. See 24.
23. *Eddy currents.*—Pole pieces hotter than coils after short run, due to faulty construction, or fluctuating current; if latter, regulate, and steady current.
24. *Moisture in coils.*—Coils not dry show less than normal resistance; may cause short circuit or body contact to iron of dynamo. Dry out as in 19.

BEARINGS.

25. *Not sufficient or poor oil.*—See that plenty of good mineral oil, filtered clean, and free from grit, is fed to bearings; be careful that it does not get on commutator or brush holder. (See 12.) Cylinder oil or vaseline may be used if necessary to complete run, mixed with sulphur or white lead, or hydrate of potash. Then clean up and put in good order.
26. *Dirt or grit in bearings.*—(*a*). Wash out grit with oil while running, then clean up and put in order. Be careful about not flooding commutator and brush holder. (*b*). Remove caps and clean and polish journals and bearings perfectly, then replace. See that all parts are free and lubricate well. (*c*) When shut down, if hot, remove bearings and let them cool naturally; then clean, scrape and polish, assemble, seeing that all parts are free, and lubricate well.
27. *Rough journals or bearings.*—Smooth and polish in a lathe, re-

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moving all burrs, scratches, tool marks, etc., and rebabbitt old boxes or fit new ones.

28-29. *Journals too tight in bearings; bent shaft.*—Slacken cap bolts, put in liners and re-tighten till run is over; then scrape, ream, etc., as may be needed, bend or turn true in lathe or grind true. Possibly a new box or shaft will be needed.

30. *Bearings out of line.*—Loosen bearing bolts, line up and block until armature is in center of pole pieces, ream out dowel and bolt holes and secure in new position.

31. *End pressure of pulley hub or shaft collars.*—See that foundation is level and armature has free end motion. If there is no end motion, file or turn ends of boxes or shoulders on shaft to provide end motion. Then line up shaft and belt, so that there is no end thrust on shaft, but so that the armature plays freely endways when running.

32. *Belt too tight.*—(*a*). Reduce load so that belt may be loosened and yet not slip. Avoid vertical belts if possible. (*b*). Choose larger pulleys, wider and longer belts with slack side on top. Vibrating and flapping belts cause winking lamps.

33. *Armature out of center of pole pieces.*—(*a*). Bearings throwing armature out of center may be worn out and need replacing. (*b*). To repair, however, center armature in polar space, and adjust bearings. Loosen bearing bolts, line up and block until armature is in center of pole pieces, ream out dowel and bolt holes and secure in new position. (*c*). File out polar space to give equal space all around. (*d*). Spring pole away from armature and secure in place; this may be difficult or impossible in large machines.

Noises.

34. *Armature or pulley out of balance.*—Faulty construction; armature and pulley should have been balanced when made. May be helped by balancing on knife edges now.

35. *Armature strikes or rubs pole pieces.*—(*a*). Bend or press down any projecting wires, and secure with tie bands. (*b*). File out pole pieces where armature strikes. See also 30 and 33.

36. *Collars or shoulders on shaft strike or rub box.*—Bearings may be loose or worn out. Perhaps new bearings are needed. See also 30 and 31.

37. *Loose bolt connection or screws.*—See that all bolts and screws are tight, and examine daily to keep them so.

38. *Brushes sing or hiss.*—(*a*). Apply stearic acid (adamantine), candle, vaseline, or cylinder oil to commutator and wipe off; only a trace should be applied. (*b*). Move brushes in and out of holder to get a firm, smooth, gentle pressure, free from hum or buzz. See also 3, 8, and 9.

39. *Flapping of belt.*—Use an endless belt if possible; if a laced belt must be used, have square ends neatly laced.

40. *Slipping of belt from overload.*—Tighten belt or reduce load. See also 32.

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41. *Humming of armature lugs or teeth.*—(*a*). Slope end of pole piece so that armature does not pass edges all at once. (*b*). Decrease magnetism of field, or increase magnetic capacity of tooth.

Variations in Speed.

RUNS TOO FAST.

42. *Engine fails to regulate with varying load.*—Adjust governor of engine to regulate properly, from no load to full load.
43. *Series motor; too much current; runs away.*—Series motor on constant current: (*a*). Put in a shunt and regulate to proper current. (*b*). Use regulator or governor to control magnetism of field for varying load. Series motor on constant potential: (*a*). Insert resistance and reduce current. (*b*). Use a proper regulator or controlling switch. (*c*). Change to automatic speed regulating motor.
44. *Shunt motor: regulator not properly set.*—Adjust regulator to control motor.

    Shunt motor: not proper current.—Use current of proper voltage and no other, with a proper rheostat.

    Shunt motor: motor not properly proportioned.—Install better motor, one properly designed for the work.

RUNS TOO SLOW.

45. *Engine fails to regulate.*—Adjust governor of engine to regulate properly, from no load to full load.
46. *Overload.*—Reduce number of lamps and load.
47. *Short circuit in armature.*—(*a*). Remove copper dust, solder or other metallic contact between commutator bars. (*b*). See that clamping rings are perfectly free, and insulated from commutator bars, and that there is no copper dust, carbonized oil, etc., to cause an electrical leak. (*c*). Test for cross connection or short circuit, and if such is found, rewind armature to correct. (*d*). See that brush holders are perfectly insulated, with no copper dust, carbon dust, oil or dust, to cause an electrical leak.
48. *Striking or rubbing of armature.*—(*a*). Bend or press down any projecting wires, and secure with tie bands. (*b*). File out pole pieces where armature strikes. See also 30 and 33.
49. *Friction.*—Clean with benzine. See also 25.
50. *Weak magnetic field.*—Broken circuit in field coils or short circuit in field coils: repair if external, rewind if internal. Machine not properly wound, or without proper amount of iron: no remedy but to rebuild it.

Motor.

STOPS OR FAILS TO START.

51-52. *Great overload; excessive friction.*—Open switch, find and repair trouble. Keep switch open and rheostat "off" to see if everything is in good order. With series motor no great harm will result from failing to start or stopping. With shunt motor on constant potential circuit, fuse may blow or armature burn

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out. Reduce load on motor to its rated capacity or less. See that there is no undue friction or mechanical resistance anywhere. See also 25, 33, and 35.

53. *Circuit open: fuse melted or switch open.*—Find trouble. Put in fuse after opening switch. (If fuse is blown on account of dead short circuit, shut down, and locate and repair fault before starting again.)

Circuit open: broken wire or connection.—Open switch, find and repair trouble as instructed under 13.

Circuit open: brushes not in contact.—Open switch and adjust as stated under 5.

Circuit open: current fails or is shut off.—Open switch; return starting box lever to off position; wait for current.

54-55-56. *Short circuit of field, armature, or switch.*—Test for, and repair if possible. Examine insulation of binding posts and brush holders. Poor insulation, dirt, oil, and copper or carbon dust often result in a short circuit.

RUNS BACKWARDS.

57. *Wrong connections.*—Connect up correctly per diagram; if no diagram is at hand, reverse connections to brushes, or other connections, until direction of rotation is satisfactory.

Dynamo or Generator.

REVERSED RESIDUAL MAGNETISM.

58. *Reversed current through field coils.*—Use current from another machine or a battery through field in proper direction to correct fault. Test polarity with a compass.

Reversed connections.—If connections or windings are not known, try one way and test; if not correct, reverse connections, try again and test.

Earth's Magnetism.—Connect up per diagram for desired rotation; see that connections to shunt and series coils are properly made.

Proximity of another dynamo.—Shift brushes until they operate better. See 1 and 2.

Brushes not in right position.—See 1 and 2.

TOO WEAK RESIDUAL MAGNETISM AND SHORT CIRCUIT.

59. *Too weak residual magnetism.*—Use current from another machine or a battery through field in proper direction to correct fault. Test polarity with a compass.

60. *Short circuit in machine.*—(*a*). Remove copper dust, solder, or other metallic contact between commutator bars. (*b*). See that clamping rings are perfectly free, and insulated from commutator bars, and that there is no copper dust, carbonized oil, etc., to cause an electrical leak. (*c*). Test for cross connection or short circuit, and if such is found rewind armature to correct. (*d*). See that brush holders are perfectly insulated, with no copper dust, carbon dust, oil or dust, to cause an electrical leak. See also 54-56.

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61. *Short circuit in external circuit.*—A lamp socket, etc., may be short circuited or grounded, and prevent building up shunt or compound machines. Find and remedy before closing switch. See also 54-56.

62. *Field coils opposed to each other.*—Reverse connections of one of field coils and test. Find polarity with compass; if necessary, use current from another machine or a battery through field in proper direction to correct fault. Test polarity with a compass. Connect up per diagram for desired rotation, and see that connections to shunt and series coils are properly made. Try shifting brushes until they operate better. If necessary reverse connections and recharge in opposite directions.

OPEN CIRCUIT.

63. *Broken wire.*—Search out and repair as stated in 13.

Faulty Connections.—Search out and repair as stated in 37.

Brushes not in contact.—Search out and repair as stated in 5.

Safety fuses melted or broken.—Search out and repair as stated in 53.

External circuit open.—Search out and repair with dynamo switch open until repairs are completed.

EXCESSIVE LOAD OR RESISTANCE.

64. *Too great load on dynamo.*—(*a*). Reduce load to pilot lamp on shunt and incandescent machines; after voltage is obtained close switches in succession slowly, and regulate voltage. (*b*). Reduce number of lamps and load. (*c*). Bring up to voltage gradually with rheostat, and watch pilot lamp, regulating carefully.

65. *Too great resistance in field rheostat.*—Bring up to voltage gradually with rheostat, and watch pilot lamp; regulate carefully.

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CHAPTER II.

REPAIRS TO THE COMMUTATOR.

The most economical method of repairing electrical machinery in a manufacturing establishment, or electric railway plant, is a subject that should command the attention of the superintendent and electrician. The exorbitant charges of electrical repair concerns and the unnecessary delay in transportation of apparatus make it a practical necessity for companies of any magnitude to do their own repairing. In the present chapter a few suggestions are given for re-filling commutators. As the commutator is the part of a direct-current machine that is subjected to the greatest wear, its re-filling constitutes a large portion of the repairman's work.

It is always advisable, when possible, to purchase hard-drawn copper strips, drawn to gage, and cut them to required lengths. Old commutators are frequently so far out of date that standard sizes of segments

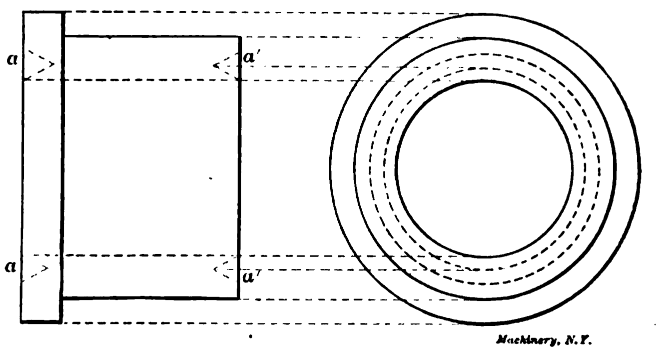

Fig. 1. Copper Casting for Commutator.

will not do. A very good commutator can be made from a copper casting, similar in shape to the assembled commutator, that is, cylindrical in form and enough larger than the original commutator to allow for finishing (Fig. 1). Large castings may be cored out at ends, *a* and *a′*, for collars, thus saving some stock and considerable labor, as it is then only necessary to make a finishing cut after segments are assembled. Bore out the rough casting and drive it on an arbor and place on "centers" of a milling machine. Use a 1/16-inch saw about four or five inches in diameter. Cut as many slots in the casting as there are to be segments, *b*, Fig. 2. By using an indexing head this is a very simple process. Cut the slots to within about 1/8 inch of through, as shown at *c*. The slots *a* for armature leads should be cut in after the commutator is assembled and turned. Now, drive out arbor and catch casting in a vise and finish cutting through the slots with a

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hack-saw. Two blades put in the frame at the same time will make a cut about equal in width to that made by the saw in milling machine. File off any burrs that remain on segments and drill a hole in each one on flanged portion *a*, Fig. 3, in diameter about twice the width of slot cut for lead wires, and a little deeper. This hole aids greatly in soldering in armature leads, as the solder flows at once to the bottom of slot. The insulation between the segments should be micanite about

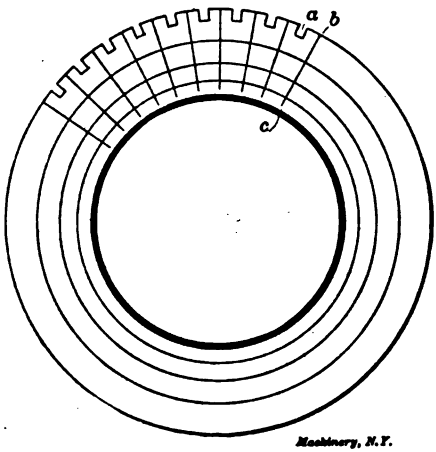

Fig. 2. Slots Cut in Casting for Commutator for Sawing Apart Segments, and for Lead Wires.

1/32 inch in thickness. As the segments are sawed up by a 1/16-inch saw, the rough casting must be made large enough to allow for the difference. For instance, in sawing up a casting into 32 segments, two inches of the circumference would be wasted. Using 1/32 inch micanite would make up for one inch only, so that the rough casting must be one inch greater in circumference—over and above the stock allowed for finishing—than the original size of the old commutator.

The next step is to assemble the segments in a suitable clamp, as

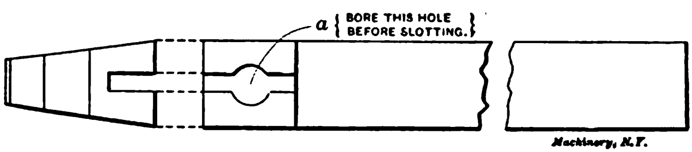

Fig. 3. Segment of Commutator after Sawing Apart.

shown in Fig. 4. This is a cast-iron split ring, the two parts, *c* and *c'*, being held together by bolts *d* and *d'*. A plan of section *c* is shown; *a* and *a'* are dowel pins, and *b* and *b'* are clearance holes for bolts *d* and *d'*. Great care must be taken in assembling the segments to have them all straight, that is, parallel with the axis of the commutator. Now, chuck the clamp, with the segments, in the lathe, and bore out the center to required diameter, then bore out the ends to correspond

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with the old commutator. A templet of tin made to fit the end bore of the old commutator is very convenient for gaging the new one.

It is more economical to make several commutators at one time, so that a temporary shaft, with collars and clamping nuts, should be provided. Such an arrangement is shown in Fig. 5, *d* being a short length of cold-rolled steel threaded at *e* and *e'*; *a* and *a'* collars bored

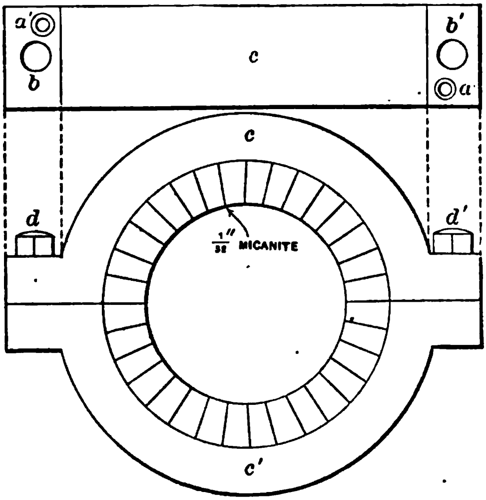

Fig. 4. Segments of Commutator Held in a Clamp for Machining Hole and Ends.

out to slip over shaft, and *b* and *b'* clamping nuts. The temporary shaft should be firmly secured to the newly-bored segments before removing the clamping ring. This done, the ring may be removed and the new commutator will be ready for turning, as shown in section at *c* and *c'*, Fig. 5. Before turning, the commutator should be heated

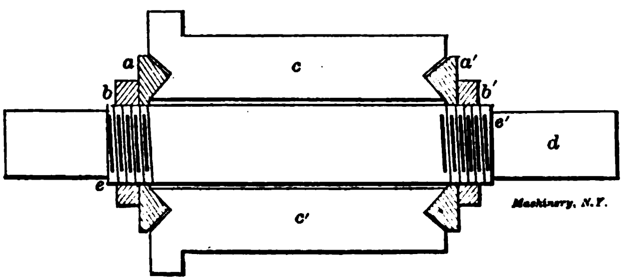

Fig. 5. Arbor for Holding Segments while Turning.

until the shellac oozes from the micanite, then placed on end on a surface-plate with a hole for shaft to extend through. This plate is shown at *d*, Fig. 6.

Place a try-square on the plate and sight along the blade to see that the edge of one of the segments coincides with it, as at *b* or *c*. If not,

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by using a small cold-chisel and hammer, drive the segment one way or the other until plumb. Go all around the commutator in this manner. After straightening all the segments, tighten up the clamping nuts again and allow the shellac to dry. After the finishing cut is taken, the commutator should be returned to the milling machine and the slots cut for lead wires, as shown at *a*, Fig. 2. When all burrs have been removed, we are ready to put on the retaining band, which firmly holds the segments in place until used.

Fig. 8 shows a method of putting on the band. The segments, *a*, are placed between lathe centers, and a heavy piece of manila paper is wrapped around them, as shown at *e*. This is held in place temporarily by a cord, which also serves to hold in place a piece of 1/32-inch brass, *b*. Now cut two fiber friction blocks, *f* and *f′*, to fit in the tool-post, bore a hole and insert a pin in each, *g* and *g′*, to keep the blocks

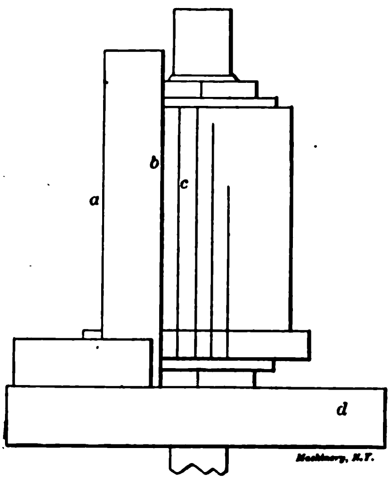

Fig. 6. Testing Alignment of Segments.

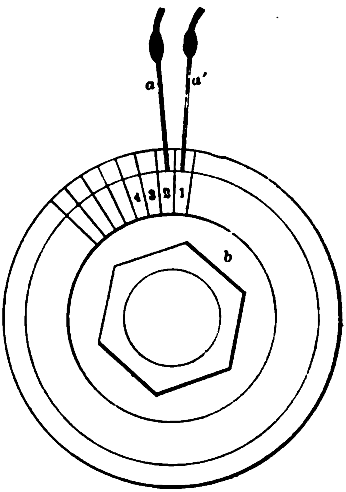

Fig. 7. Testing the Commutator for Short Circuit.

in place. Any amount of tension can be placed on the blocks by the clamping screw *n*.

Take the end of a coil of No. 16 brass wire, and pass it between friction blocks, *f* and *f′*, and catch it in one of the slots, as at *c*. Turn the assembled segments two or three revolutions until the wire is brought over the paper *e*, then cover about one-half the length of the segments closely and very tight. When the desired amount of wire has been wound on, turn the ends *i* and *i′* of the brass strip *b* over on wire, and hammer down, bringing the turn *h* close up to the band. Flow solder over the band with an iron and cut off the ends of wire. The commutator may then be removed from temporary clamping device, when it will have the appearance shown in Fig. 9. The temporary clamping device, the clamping ring, and templets, can be used indefinitely. One clamping ring can be used for several sizes of commutators by using split bushings.

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When removing old segments from a commutator, care should be taken to keep the molded mica insulation on the ends intact. If this is broken it can be replaced by canvas disks, shown in Fig. 11, made up of several pieces shellaced together to obtain a thickness equal to the molded mica. Place the old commutator sleeve, with the rear collar attached, end down on a bench and slip the canvas disk over

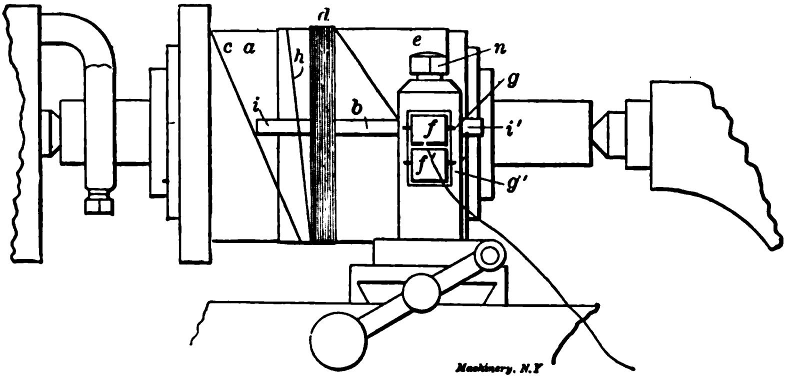

Fig. 8. Method of Putting on Retaining Band.

sleeve to bottom. A hole in the disk should fit tightly over the sleeve, and the outside diameter be about one inch greater than that of the commutator. A thin sheet of flexible micanite must be wrapped around the commutator sleeve, to insulate it from the inside of segments. After placing the assembled segments *d*, over the sleeve, slip on the

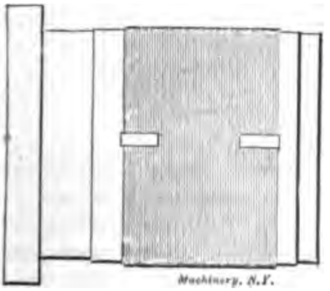

Fig. 9. Commutator with Retaining Band in Place.

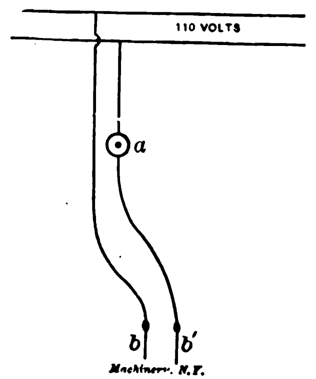

Fig. 10. Arrangement for Testing the Commutator.

upper canvas disk *a*, Fig. 12, then collar *b*, finally tightening up nut *c*. Canvas disks should be put in with shellac, wet. After screwing up the nut firmly, allow all dampness to dry out thoroughly. The canvas disks will then protrude between collars and segments as shown at *a*, Fig. 13. Trim off smoothly, giving a finished appearance like *a* in Fig. 14.

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The completed commutator is now ready for testing. A very convenient and fairly accurate method is shown diagrammatically in Fig. 10. A sixteen candle-power incandescent lamp is connected in series with the mains, and two flexible cords with solid copper tips, *b* and *b'*. Fig. 7 shows the application of the testing arrangement.

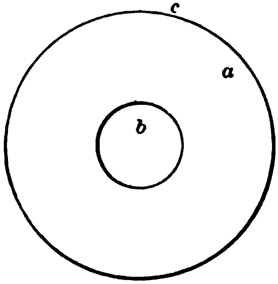

Fig. 11.

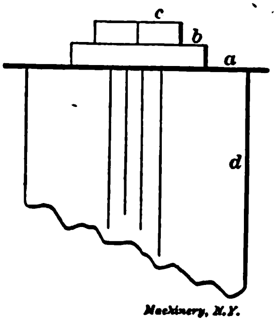

Fig. 12.

The copper tips, *a* and *a'*, are placed on adjoining segments, as at 1 and 2. If there is a short-circuit, the lamp will light. Test each segment in turn in this manner. Then, by placing one of the tips on the end of the collar, as at *b*, and touching the other to each segment in turn, any leakage from segments to core will be found. If no leak is

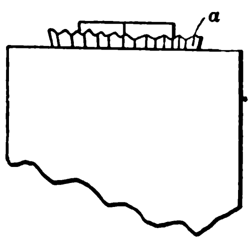

Fig. 13.

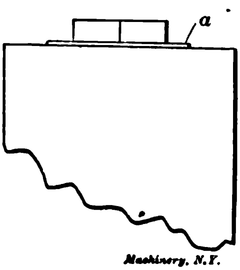

Fig. 14.

found the commutator is ready for use. If a leak or short-circuit appears, the trouble must be located and remedied before using.

Small copper chips wedged in the micanite by the turning-tool often cause a short-circuit between segments. A careful inspection inside and out after turning, will generally disclose any such defect.

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CHAPTER III.

REPAIRS TO THE ARMATURE WINDING.

The repair shop of a manufacturing plant or electric railway plant should have at hand suitable stands or "horses" for holding armatures during the winding process. If the armature is small, short stands may be mounted on a work-bench. When an armature comes in to be repaired, carefully caliper its diameter outside of the bands and the winding. Observe particularly the shape of the ends. As the workman proceeds to tear apart the armature, he should note the size of wire, style of winding, number of coils, convolutions per coil, number of layers, and all other details. All such data should enter a notebook, as this informaiton will be of future value. It is well to head the

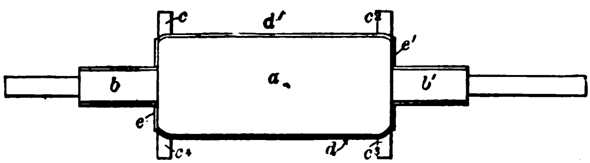
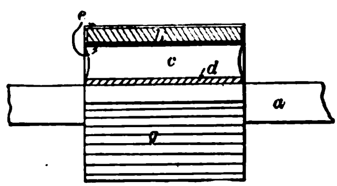


Fig 15.



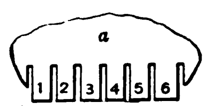


Fig. 16.

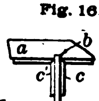


Fig. 17. Machinery, N.Y. Fig. 18.

Figs. 15 to 18.

entry with the name of the machine, horse-power, voltage and current, speed and serial number of the armature.

The first step in unwinding an armature is to unsolder the leads and remove the commutator. Then cut off the bands and remove the wire. If the coils are of the formed type, laid in slots, raise the upper half around the entire circumference, and remove the coils, in the reverse order to that in which they were put in. After the core is entirely stripped of winding and insulation, it is ready to re-insulate.

Fig. 15 is a sectional view of a smooth-core drum armature insulated ready for winding. In the figure, *a* is the core; *b* and *b′* the ends of shaft covered as shown; c^1, c^2, c^3, and c^4 are fiber pegs for separating the coils; *d* and *d′* insulation on core, and *e* and *e′* insulating end disks. Fig. 16 shows the manner of notching the core-insulation to fit between the fiber pegs. Flexible micanite 1/32 inch in thickness,

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and held in place temporarily by a few turns of "flax," forms an excellent insulation. A ring armature, partially in section, is shown in Fig. 17. The shaft, *a*, pressed into the hub, *d*, carries the spider, to which is attached the ring *b*. One wing of the spider is shown at *c*. A flexible micanite insulation, *e*, covering the outside, inside and ends of ring, is held in place by a tight wrapping of cotton tape, as shown at *g*. The wings of the spider must be insulated. This can be done conveniently, as shown in Fig. 18, which shows a sectional end view of an armature, *a* being the armature ring, *b* the spider wing, and *c* and *c'* triangular-shaped pieces of micanite extending the length of the

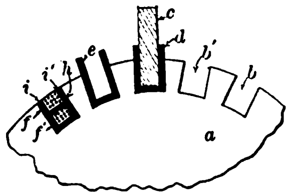

Fig. 19.

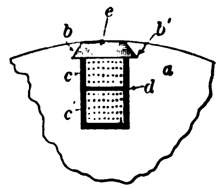

Fig. 20.

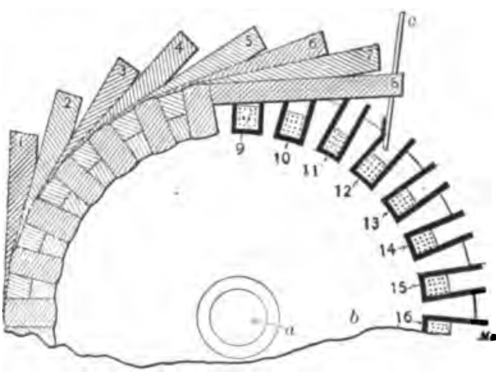

Fig. 21.

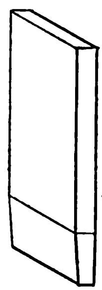

Fig 22.

Machinery, N. Y.

Figs. 19 to 22.

wing. The triangular pieces are retained in position by the wrapping of cotton tape.

A section of a slotted armature is shown in Fig. 19. Here *a* is the end in section; *b* and *b'* are slots not yet insulated; *c* is a hard-wood or fiber block the length of the slot and slightly narrower; *d* is micanite insulation driven in place by the block *c*; *e* is a slot insulated ready for the coil; *f* and *f'* are armature coils; *h* is the insulation between the coils; and *i* and *i'* the slot insulation trimmed flush with core *a*. The micanite is cut into strips of the required size, and folded over the block *c*, to form troughs, as shown. Another form of slot coming generally into use is shown in Fig. 20, in which *a* is the end of armature; *b* and *b'* V-shaped slots cut for receiving the wood retaining-strip *e*; *c* and *c'* coils in the slot, with insulation *d* between them.

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Fig. 21 is an end view of an armature partially in section. This view illustrates the winding process with a form of coil in common use. In the figure, *a* is the shaft; *b* the end of the armature; *c*, a piece of sheet brass the length of the slot and about 4 inches wide; 1, 2, 3, 4, 5, 6, 7, and 8 show the coils with one-half free and one-half in slots; 9, 10, 11, 12, 13, 14, 15, and 16 show one-half of the coil in cross-section. Proceeding to wind an armature of the type shown in Fig. 21, the first process is to insulate the slots in the manner already described, then drive one-half of the coil into place, continuing around the armature, thus filling half of each slot. The armature will then appear as shown. Fig. 22 is a piece of vulcanized fiber, shaped for driving coils into slots. In a four-pole machine each coil will have a pitch of 90 degrees, that is, it will cover one-fourth of the periphery of the armature. The coils slip into the slots easily, with the exception of those on the last quarter. Beginning at a point three-fourths of the distance around the armature from the first coil, the outer half of the coils that have

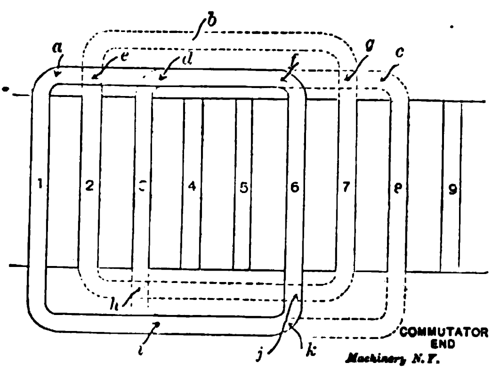

Fig. 23. Development of Surface of Armature.

been laid in, will lap over the empty slots. At this point a little more labor is involved in getting the coils into their respective places. Having completed the first process, start at any point to drive the outer half of the coils into slots. Hold the piece of brass, *c*, before referred to, in the left hand and slip it into the position shown. In this way it guides the coil 8 into the slot. With a mallet and the tool shown in Fig. 22, drive the coil snugly into place. Continue with each coil in like manner until all have been driven into their proper positions.

In Fig. 23, the periphery of the armature is supposed to be laid out flat. In the figure, *a*, *b*, and *c* are coils of the oblong type, which form a chordal winding. It will be seen that the coils are staggered, that is, the projecting ends alternate backward and forward. With this style of winding, the ends of the armature core must be well insulated. The coils, if not pounded into shape with a mallet, will interfere at *d*, *e*, *f*, *g*, *h*, *i*, *j*, and *k*. The coil *a* extends from slots 1 to 6; *b* from 2 to 7, and *c* from 3 to 8. No specific directions can be given here for shaping these coils, as no two makes of armatures are

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alike. By noting carefully the shape of the original coil, it will be an easy matter to form the new one. The winding is executed in a manner similar to the formed-coil winding previously described.

Taking up the subject of smooth-core drum armatures, let us study Figs. 24 to 29 inclusive, which illustrate some common types of winding. Starting with Fig. 24, we will assume each coil, for the sake of simplicity, to be one layer deep and three convolutions in width. Stand facing the commutator end of the armature, and tie the end of wire *a* to the fiber peg. Pass the wire downward over the end of the armature core and between the two pegs, diametrically opposite to the starting point. Turn the armature over and draw the wire tightly along its surface to the back end. Carry the wire around the shaft, on the side opposite to that followed on the front end, and through

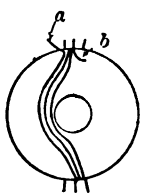

Fig. 24.

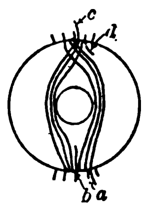

Fig. 25.

Fig. 26.

Fig. 27.

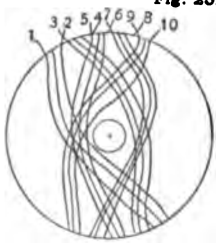

Fig. 28.

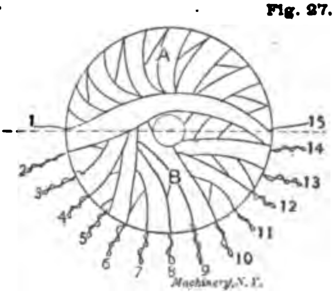

Fig. 29.

Figs. 24 to 29.

the pegs back to the starting point, having in the meantime turned the armature over to its original position. It is convenient to have the reel of wire suspended over the workman's head, so that the wire will pay off freely. Cut the wire from the reel, leaving end *b*. The ends must be long enough to be soldered into the commutator.

Now turn the armature over and begin the second coil to the right of the first one. This will bring the second coil to the left of the first one at the bottom side, as shown in Fig. 25. The ends *c* and *d* are left for connecting to the commutator, as in the first instance. To commence the third coil, turn the armature over again and start at the right of the second coil, Fig. 26, twisting the end *e* around end *b*. Proceed in this manner until all the coils are wound. Fig. 27 gives an idea of the appearance of the armature end after three coils have

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been wound. It will be seen that every second space between the pegs has two ends. The inner end of the first coil is connected to the outer end of the third coil, there being a blank coil between the two, thus forming a closed or re-entrant winding. Any number of layers or convolutions may be used in this winding. Fig. 28 shows a similar winding, that can be used for two layers or a multiple of two. The only difference is that the second coil is commenced at the ending of the third, and so on until every space is filled, as shown in Fig. 29. If the winding when completed is to be four layers deep, the coils from 1 to 15 inclusive will have two layers, and the remaining coils to be wound will also have the same number. The half *A* of the arma-

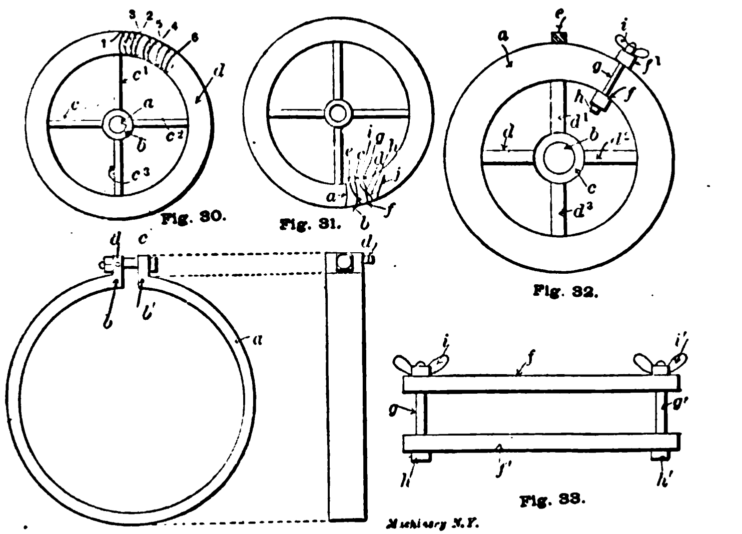

Fig. 30. Fig. 31. Fig. 32. Fig. 33. Fig. 34.

Figs. 30 to 34.

ture will have no ends at this stage of the winding. The second set of coils, whose ends protrude in the half *A*, commences at 15 and extends around to 1. The outer end of one coil joins to the inside of the next. This is clearly shown in Fig. 28, where end 3 of the second coil joins end 2 of the first coil, and so on. It is unnecessary to cut the wire, as it can be left in loops as shown at 1, 2, 3, etc., in Fig. 29.

We will now turn to smooth-core ring armatures. In Fig. 30, *a* is the shaft and *b* the spider hub, to which are attached the wings, c, c^1, c^2, and c^3, bearing the ring *d*. The winding forms a continuous spiral, the end 2 of coil 1 being attached to end 3 of coil 2, etc. As the space or the inner surface of the armature is less than that of the periphery, the winding will have a greater number of layers inside than on the outer surface. An exaggerated view of a method of winding to accomplish such a result is shown in Fig. 31. The wire starts at *a*, then passes around the ring and comes to the front at *b*, passes under at *c*,

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and returns at *d*. It is then carried under at *e*. between *a* and *c*, starting the second layer on the inner surface. From *f*, the wire goes to *g*. and back to *h*, thence to *i*, between *c* and *g*, coming back at *j*. Thus we have five convolutions and one layer on the outer surface, and two layers—one of three and one of two convolutions—on the inner surface.

In Fig. 32 the application of the clamp shown in Fig. 33 is given. This clamp consists of two wood pieces, *f* and *f'*, and two bolts, *g* and *g'*, with heads, *h* and *h'*, and thumb nuts, *i* and *i'*. This clamp serves

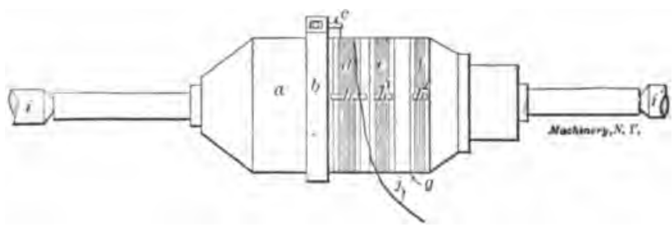

Fig. 35. Application of Clamp in Fig. 34.

to hold the wire of each coil in position while winding, and is moved around as fast as a coil is completed. Referring again to Fig. 32, the wood piece, *e*, is used for filling the gap in the outer surface of the winding caused by the spider wings, *d*, d^1, d^2, d^3. It is made equal in width to the wing, and of the same depth as the winding. In balancing an armature one or more of these strips may be removed and lead strips wound in tape substituted. All armatures must be carefully balanced, which can be accomplished by several methods, one of which has just been mentioned. If the air-gap of the machine has clearance enough, solder may be flowed onto the bands. With slotted armatures,

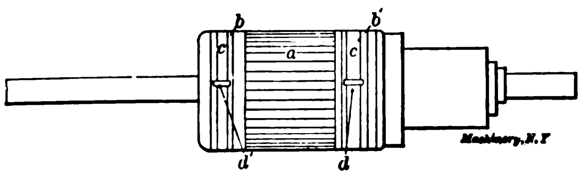

Fig. 36. A Completed Armature.

some makers bore holes in the core on the heavier side, thus equalizing the weight. Another method is to bind a piece of sheet lead on the front end of the armature over the lead wires, by a tape and cord.

As no definite rule can be given for soldering lead wires into a commutator, only a few suggestions will be offered. Tin the slots in the segments and the armature leads before soldering. See that the slot is flowed full of solder. Do not use acid for flux on small wire, as corrosion will take place. Be extremely careful that no drops of solder lodge between or back of the segments. Each coil should be tested for grounds as it is wound, and to make sure that the right ends are connected. A convenient testing arrangement was explained in the previous chapter on Repairs to the Commutator, and can be applied in

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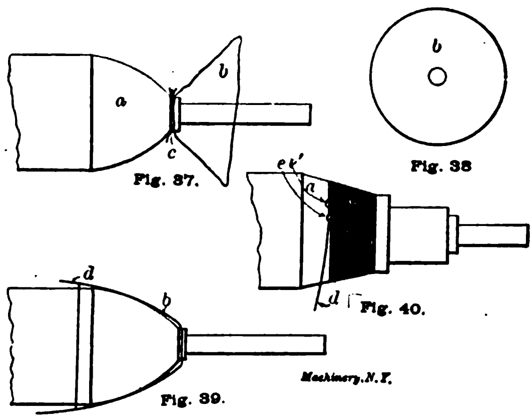
this case also. A magneto bell, a galvanometer or a Wheatstone bridge may be used for testing purposes.

Fig. 34 gives two views of a clamp that is used in winding armature bands, *a* being a hoop of band-iron, with its inside diameter equal to the outside diameter of the armature. The ends *b* and *b'* are bent up as shown, and bored to receive a clamping bolt, *c*. A pin, *d*, is attached to the end, *b*, for fastening the binding wire. Fig. 35 shows clearly the practical application of the clamp. In this sketch, *a* is the armature; *b*, the clamp; *c*, the pin; *d*, the band being wound; *e* and *f*, finished bands; *g*, mica insulation under bands; *h*, h^1, and h^2, brass clips for holding the bands together. The lathe centers are shown at *i* and *i'*. The end of the brass wire *j* passes through the fiber friction blocks in the toolpost.

The manner of winding an armature is essentially the same as that



Figs. 37 to 40.

of a commutator, which has already been fully described in the previous chapter.

A completed armature, with slots of the type shown in Fig. 20, is illustrated in Fig. 36. The core, with wood retaining strips driven into slots, is shown at *a*. Mica strips *b* and *b'*, under bands *c c*, are for the usual insulating purpose. Brass chips *d* and *d'* are attached in the regular manner. Armatures with wood retaining-strips in slots require but two bands, which are wound on the coil ends that project beyond the core.

A method of protecting the ends of a surface-wound drum armature is illustrated in Figs. 37, 38, 39, and 40. A canvas disk, *b*, Fig. 38, is tied by a cord, *c*, to the end of armature *a*, Fig. 37. The disk is then drawn over the end and tied temporarily with a cord *d*, as in Fig. 39. The armature band serves to hold this hood in place permanently. An effective manner of finishing the commutator end of an armature is illustrated in Fig. 40. The end *a* is wound tightly with cord about ¼ inch in diameter, as shown at *b*. Two loops of string, *e* and *e'*, are

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caught under the last two or three turns of cord and the end *d* is passed through them, after which the loops are drawn up, the ends trimmed off and the cord cut close to the last loop. After the armature is completed, it should be given a thorough coating of shellac and placed in an oven. When fully dried out, put on two even coats of P. & B. compound, which gives it a good black and waterproof finish.

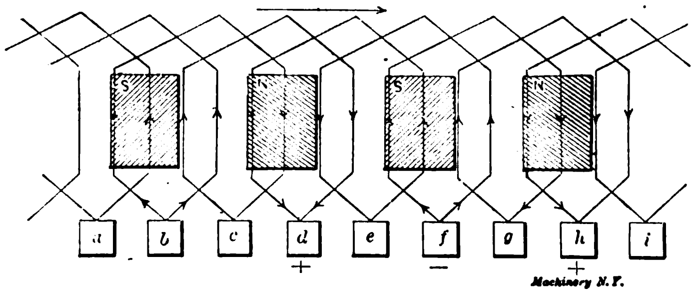
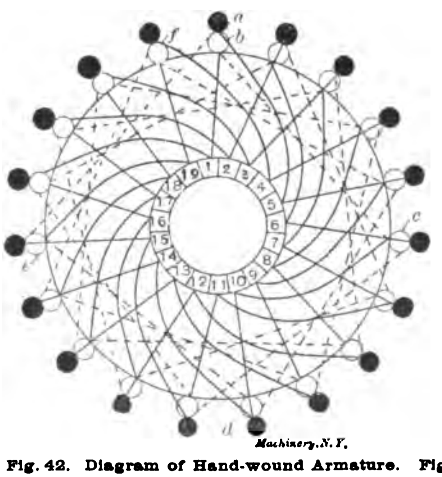


Fig. 41. Graphical Illustration of Lap Winding.

As a complete description of the numerous styles of connecting up armatures is beyond the scope of this book, only a few of the common types will be taken up. For further information on this subject the reader is referred to one of the many text-books on dynamo-electric machinery.

Fig. 41 illustrates, graphically, a lap-winding. The poles of the



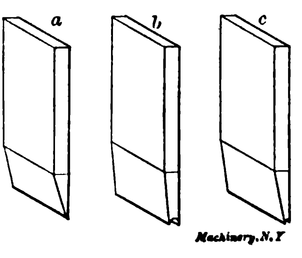


Fig. 42. Diagram of Hand-wound Armature. Fig. 43. Tools for Winding Armatures.

machine are represented at *S*, *N*, *S*, *N*, and the commutator segments are indicated by the squares *a*, *b*, *c*, etc. We will take, for example, the coil starting at *b*. The conductor passes over the left face of the *S* pole, and then returns over the middle of the *N* pole, to the adjoining segment, *c*. This series of loops is continued around the armature, forming a complete circuit. The large arrow indicates the direction

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in which the conductors are moving, and the small arrow-heads on the wires show the disposition of the current.

A diagram of a hand-wound armature, with a wave winding, is shown in Fig. 42. To avoid complications, nineteen coils only are shown. This type of armature is extensively used by the Shaw Electric Crane Company. The small white circles around the circumference of the armature, as at *b*, represent the first layer of wire; the black circles, at *a*, show the outer layer. The dotted lines indicate the conductors passing over the back end of the armature. Starting at the commutator segment 1, the conductor goes to *b*, then to *c*, and connects to segment 10. Here we start a new coil, going to *d*, then to *e*, and connecting to segment 19, one space from the starting point. The third coil starts here and leads to *f*, and so on until one layer is completed, when one-quarter of the circumference of the armature on each side will have ends protruding, that is, the first quarter will have leads, the second will be blank, the third will have leads again, and the fourth will be blank. Start the second layer at the end of the first one. When the winding is completed, each slot will have two ends, as shown in the figure. An armature with formed coils can be connected in a like manner. A few handy tools can be made from fiber; they are shown at *a*, *b*, and *c*, in Fig. 43. These tools are used for driving the wires into place, etc.

Rewinding an armature requires great care and neatness. The dimensions and shape of the original winding should always be closely followed, as an armature which is but a fraction of an inch too large is useless.

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CHAPTER IV.

REPAIRS TO ARMATURE AND FIELD COILS.

An illustration of an armature coil of the most common type now in use, is given in Fig. 44. This coil requires somewhat more labor to prepare than does the rectangular form, but is much easier to use when rewinding armatures, as no portion of it passes over the end; and it has the further advantage of allowing better ventilation. Another type in use is the plain, rectangular form, shown in Fig. 49. Such a coil is easily made, but requires considerable manipulation after it is placed in the armature. The forms over which these two styles of coils are wound are similar in design but different in shape. Fig. 45 shows a form, in which *a* is a standard fastened to bench or

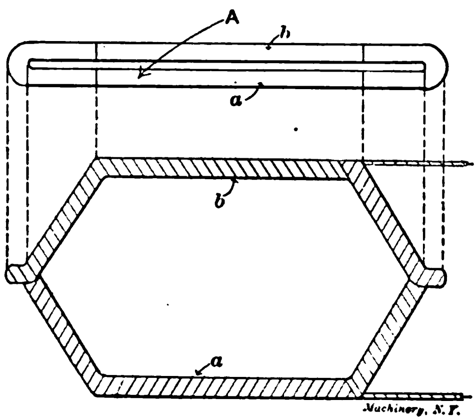

Fig. 44. Armature Coil of Common Type.

floor, and a slot is cut down through the center of this standard to a point two or three inches below the crank. A small bolt, *l*, with thumb-nut, is provided, to place a tension on the crank, by drawing up the slot. This is plainly shown in the figure. For the form proper, we require a piece of hard wood, *b*, rounded at the ends, *d* and *d'*, the size and shape of this piece conforming to the interior of the coil to be wound; and two side pieces of wood, *c* and *c'*, slightly larger in dimensions than the center piece, so that when placed on either side concentrically with the center, a spool is formed. The side *c* is fastened permanently to *b*, while *c'* is held in place by two thumb-nuts, *e* and *e'*, which allows the coil to be readily detached from the form. The flange *f* is attached to the short shaft with crank and handle *j*. The illustration shows elevation and side view. The notches, *g* and *g'*, *h* and *h'*, *i* and *i'*, are for fastening the ends of the wire and for the retaining strips, which will be explained later.

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Fig. 46 gives plan and end views of the shaper for shaping coils of the style shown in Fig. 44, in which *g* is the wood base, *a* is the coil that has been shaped, *b* and *b'* the clamps; *c*, *d*, *e* and *f*, wooden strips, of the shape indicated. The pieces *d* and *e* are fastened to the base *g* by screws. The operation of this shaper will be taken up in its turn. Fig. 47 represents a reel of magnet wire, *a*, swung on a suitable support, *b*, and provided with a tension. The tension consists of a piece of hard brass wire, about No. 10 or 12, fastened to base *b* and passing around a groove in one side of the reel to a weight, *d*. With the wire *e* paying off in the direction indicated, the desired amount of tension may be placed on the reel by varying the size of the weight.

Fig. 48 represents a handle for guiding the wire onto the form. This

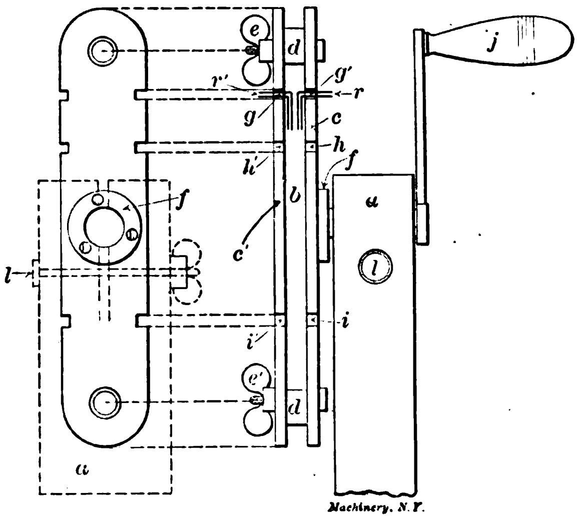

Fig. 45. Form for Winding Armature Coils.

handle should be made of fiber or hard wood. The end *c* is beveled off as shown. A hole or slot, *b*, extends through the handle, through which the wires pass. The handle is held in the left hand, while the form is rotated with the right hand by means of the crank already mentioned. Some styles of coils are wound with two or more wires laid on in parallel at one time. The reason for this is that in machines of considerable size, a single conductor of sufficient carrying capacity would be too stiff for handling. Under such conditions the handle should have a slot at *b* instead of a hole, and a separate reel for each conductor should be provided, one placed back of the other.

The materials used in armature coils are: Double, cotton-covered magnet wire, ½- or ¾-inch cotton tape, insulating tape, a good quality of orange shellac, and some strips of 1/32-inch sheet brass. The cotton tape can be procured at any department store in rolls of various sizes.

A drying oven is an essential part of an electrical repair shop, and

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when there is steam or gas in the building it is an easy matter to provide one. Its capacity must be determined by the size and quantity of the work to be done. An angle-iron frame, with 1/16-inch iron or steel lagging for siding and doors, makes an excellent oven. The doors should be on the front and extend from top to bottom. The interior of the oven must be provided at the lower portion with a suitable stand for holding armatures, and above the stand, metal racks, for drying coils and commutators. Steam coils or gas burners may be used for heating the oven. A good temperature for drying is about 200 degrees F. Too much care cannot be taken in following the dimensions of old coils, as a little variation will cause trouble when placing the new coils in an armature.

In general, do not use metal for driving wires into place; use hard

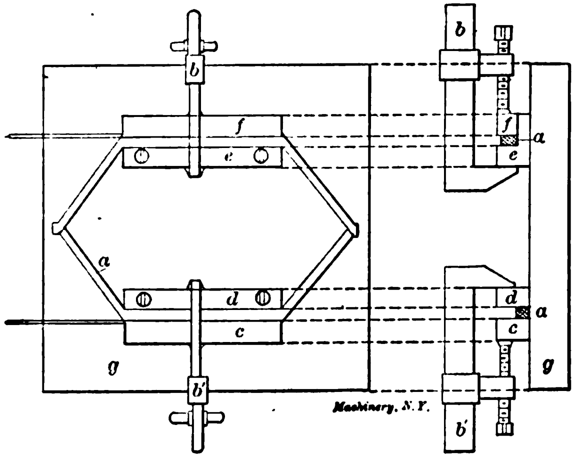

Fig. 46. Shaper for Coils Shown in Fig. 44.

wood or fiber. If any bare spots appear in course of winding, insulate them at once. Do not cut the ends of the wire too short. It is easier to cut off a little more than to splice on. Tape the coils tightly and let the convolutions lap well, so that they will not pull apart when shaped. The lead wires, or ends of coils, should be carefully covered with insulating tape. Soldering in of lead wires is greatly facilitated by tinning the ends before driving them into the commutator slots. The beginner, undoubtedly, will have to make several attempts before completing a perfect coil. Neatness is of great importance, and every portion should be well done before proceeding further.

We will now wind and insulate a coil, and follow out each detail. Assume that a set of coils, similar to that shown in Fig. 44, is to be made. It is supposed that the winder has one of the old coils for inspection. With a wire-gage determine the size of wire, and note whether one or more wires are run in parallel for one conductor. This done, carefully unwind one turn and measure its length; this will be the circumference of the form to wind the new coil on. A coil of this

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style should be wound on a long, narrow form, similar to that shown in Fig. 45. Some armatures have two coils per slot, that is, two coils made up as one, with their respective ends brought out separately. Consider that we are about to wind a double coil, with two wires in parallel for one conductor. This will necessitate winding on four wires at one time.

Arrange four reels of magnet wire, one back of the other, and push the four ends through the slot in one guide. Suppose the desired length of the armature leads, or coil ends, to be six inches. Now bend at right angles the two right-hand wires, about six inches from the ends passed through the guide, then bend the two left-hand wires in the opposite direction. This being done, slip the two pairs of bent ends into the slots *g* and *g′* in the manner shown at *r* and *r′*, Fig. 45. Say that the coil is four convolutions wide and three layers deep. Having secured the ends as described, turn the crank two and one-half times, bringing the outer ends of the coil through the slots on the opposite side to *g* and *g′*, using care to retain the ends in their original order. During the winding process the guide should be held firmly in

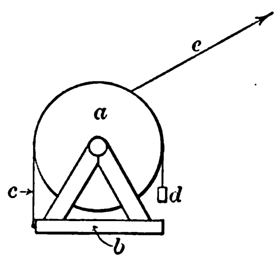
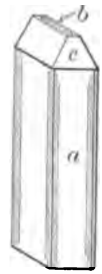


Fig. 47.



Machinery, N.Y.

Fig. 48.

the left hand, and the wires pulled down tightly into the form. Now cut off the wires from the reels, leaving six-inch ends on the coil, as at the beginning.

Cut four narrow strips of 1/32-inch brass, about 1 inch long, and slip one under the coil through the slots *h* and *h′*, *i* and *i′*, and also through the four corresponding slots on the opposite side of the form. Bend the strips up over the coil, and tap down lightly with a mallet, taking care not to break the insulation. Loosen the thumb-nuts, *e* and *e′*, slip off the side *c′*, and remove the coil. Make up the desired number of coils in this manner before insulating them. Having completed the winding, cover with good insulating tape the four ends—that is, the eight wires. These leads should be covered from the point where they bend through slots in the form, to about one inch from the end. The next operation is to carefully wrap the whole coil with cotton tape, giving it the appearance shown in Fig. 44. The leads must be left protruding the distance covered with insulating tape.

We will now shape the coil, as in Fig. 46. The two halves of the coil, after being taped, should be pressed closer together, giving the

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coil the appearance shown at *A*, Fig. 44. Insert one of the halves between the strips *c* and *d*, and clamp it tightly. With the protruding half grasped with both hands, pull it over the strip *e* and clamp as shown. With a small fiber mallet, pound the ends into symmetrical shape. Remove from shaper, and immerse in a suitable receptacle filled with thin shellac. When thoroughly impregnated, allow superfluous shellac to drip off, and place in oven to bake. The rectangular

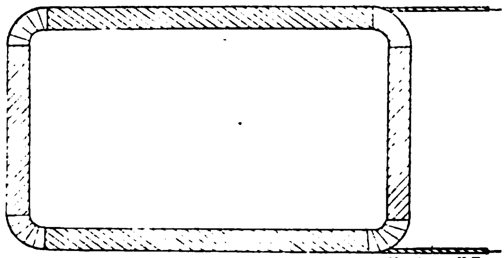

Fig. 49. Plain Rectangular Form of Armature Coil.

coil shown in Fig. 49 can be made on a form similar to the one described, having its dimensions correspond to the shape of the coil. A little practice will render the workman proficient, and enable him

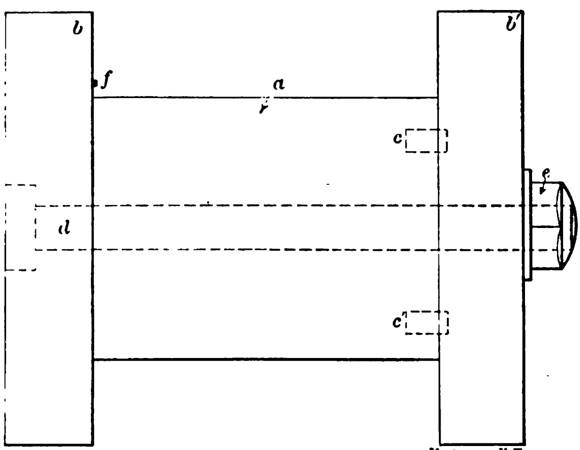

Fig. 50. Form for Winding Field Coils without Spools.

to turn out as good work as can be purchased from electric manufacturing companies.

Repairing Field Coils.

Fig. 50 gives a view of a form for winding coils that have no spools. The shape of this form depends on the style of the coil to be wound. The form is clamped to a lathe face-plate either by bolts extending through the form or through the side *b*. Dowel pins *c* and *c'* serve to hold the side *b'* from twisting, and the bolt *d* and the nut *e* hold the

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whole form together. Fig. 51 represents a guide that is held in the tool-post of a lathe. Attached to the piece *a* is a grooved wheel, *b*, over which the wire from the reel runs. The same arrangement of reels can be employed in winding field coils that is used with armature coils. Fig. 52 shows front and side views of a connector, which consists of a piece of sheet copper, *b*, rolled up at the end, *c*, and sweated into a sleeve, *a*, at the opposite end. The sleeve has a set-screw, *d*, for outside connections on the machine.

A very convenient way of securing the last turn of wire is shown

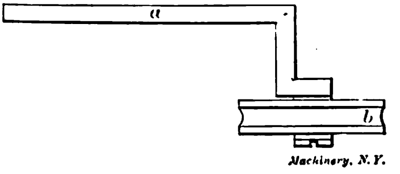

Fig. 51. Guide used in Winding.

in Fig. 53. Here *a*, *b*, *c*, *d*, and *e* represent the convolutions of wire; *f* is a loop of cotton tape with its ends protruding at *g*. The loop is laid on the coil before the turns *c* and *d* are made, then the end of the wire, *h*, is pushed through, and the loop *f* is drawn tight by pulling on the ends at *g*.

We will now start to wind a coil. First, bolt the form to the face-plate of the lathe, then arrange the reel conveniently for paying off

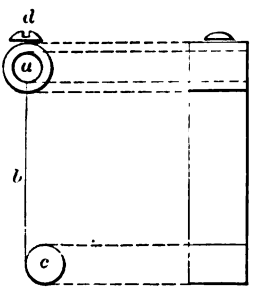

Fig. 52.

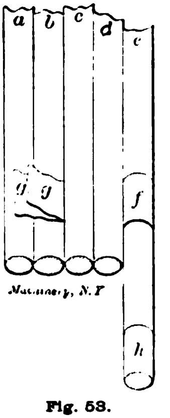

Fig. 53.

the wire. Solder a connector like that shown in Fig. 52 to the end of the wire, and tape thoroughly. Catch the connector behind the pin *f*, provided for that purpose (Fig. 50), and proceed to wind. The wire running over a guide wheel in the tool-post enables the operator to use the tool carriage for guiding the wire as it is wound. With a little practice the operator will be able to run the tool carriage backward and forward with enough skill to permit considerable speed in winding.

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The connectors can be made in different lengths and widths for varying styles of coils. The connector for the outer end of coil will, of course, be short, allowing the sleeve *a* to come on outside of insulation. Having wound the desired number of turns onto form, finish the end with a loop of tape, as shown in Fig. 53, and solder on outside connector. Coils wound of wire fine enough to be flexible do not require connectors, as the wire itself may be left protruding through the covering. Before starting the winding, several pieces of cord or cotton tape must be laid across the form, with ends long enough to tie over the completed coil. Having completed the winding of coil, take off the side of form, *b'*, and remove the coil. Different manufacturers have various methods of insulating their coils, and it is always well to

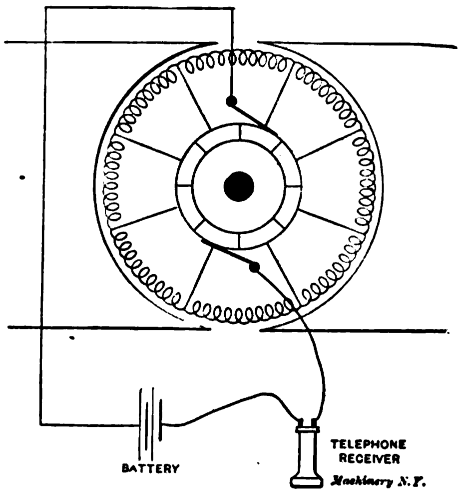

Fig. 54. Testing for Faults in Armature.

treat the new coil in the same manner that the original coil was covered.

Tests for Faults in Armature.

It is very desirable to be able to locate faults in motor or generator armatures around shops with simple apparatus that may be on hand. A method which has proven very reliable and requires only a few cells of battery and a telephone receiver is given below.

Tests for Open Circuit.

Clean the brushes and commutator, and apply current from a few cells of battery having a telephone receiver in circuit as shown in Fig. 54. If the machine has more than two brushes, connect the leads to two adjoining brushes and raise the balance. Now rotate the armature slowly by hand and there will be a distinct click in the receiver as each segment passes under the brushes until one brush bears on the segment at fault, when the clicking will cease. Note that the brushes must not cover more than a single segment.

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If, on rotating the armature completely around, the receiver indicates no break in the leads, connect the battery leads directly to the brushes, as shown in Fig. 55, and touch the connections from the receiver to two adjacent bars, working from bar to bar. The clicking

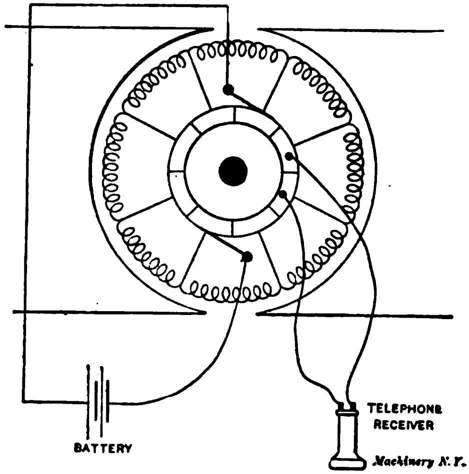

Fig. 55. Next Step in Testing Armature.

should be substantially the same between any two commutator bars; if the clicking suddenly rises in tone between two bars, it is indicative of a high resistance in the coil or a break (open circuit).

Test for Short Circuit.

Where two adjacent commutator bars are in contact, or a coil be-

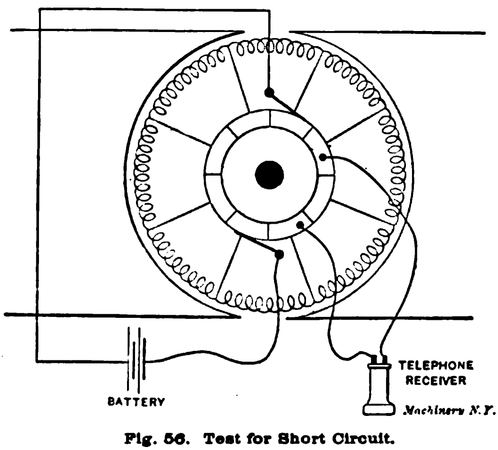

Fig. 56. Test for Short Circuit.

tween two segments becomes short-circuited, the bar to bar test just described will detect the fault by the telephone receiver remaining silent. If a short circuit is found, the leads from the receiver should

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then include or straddle three commutator bars, as shown in Fig. 56. The normal click will then be twice that between two segments until the coils in fault are reached, when the clicking will be less. When this happens, test each coil for trouble and, if individually they are all right, the trouble is between the two.

Test for Grounded Armature.

Place one terminal of the receiver on the shaft or frame of the machine, and the other on the commutator. If there is a click it indicates a ground. Move the terminal about the commutator until the least clicking is heard and at or near that point will be found the contact. Grounds in field coils can be located in the same manner.

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CHAPTER V.

WINDING OF DIRECT-CURRENT ARMATURES.

The following detailed description by Mr. A. C. Jordan, of the various operations performed by an armature winder, accompanied by precise directions and data, originally appeared in the *Electric Journal*, December, 1905, and was published in MACHINERY, March, 1906. The types of armatures to which this description apply are those used in direct-current railway motors, crane and hoisting motors, vehicle motors, bipolar motors and belted generators up to 100-kilowatt capacity.

Tools.

The tools used by an armature winder are as follows:

1 shoe knife,
1 pair seven-inch shears,
1 pair eight-inch pliers,
1 ten-inch screw-driver,

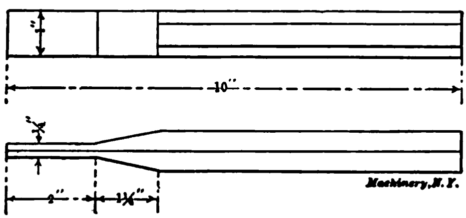

Fig. 57. Wedging Tool.

1 three-pound rawhide mallet,
1 small steel riveting hammer,
1 wedging tool (See Fig. 57),
1 heavy steel drift (See Fig. 58);
Also an assortment of fiber drifts of varying width, length and thickness (See Fig. 59).

The rawhide mallet is used in driving the coils into the slots by means of the fiber drifts, and in bending the coils into shape. The steel hammer is used for straightening laminations or finger plates. It should never be used in bending coils or on any of the drifts. The wedging tool made from a cold chisel is used in driving wedges into the slots as a hammer would injure the insulation of the coils and might bend the laminations.

Core.

An armature core is built up of soft sheet steel laminations. These are stamped of the desired shape and carefully annealed. The stampings are then built up, and keyed to a shaft or spider and held securely

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in place by end plates. Ventilating spaces are left next the shaft or spider and air ducts are distributed at intervals through the punchings by putting in spreaders to hold the laminations apart. The armature in rotating draws in air through the ventilating spaces next the shaft and forces it out through the ducts, thus furnishing a simple and effective means of ventilation. After the core is assembled, the slots are filed to remove any projecting burrs; if these were not removed, the insulation of the coil might be torn when a coil is driven into the slot and cause grounds and short circuits in the winding.

Operations Before Placing Coils on the Core.

The core is mounted in a winding lathe, as shown in Fig. 60. If duck blankets are used they should be placed on the shaft before the core is placed in the winding lathe. If a block is used on the rear end of the armature core to shape the coils as they are wound or to protect the cast iron end-bell, the block should be placed on the shaft

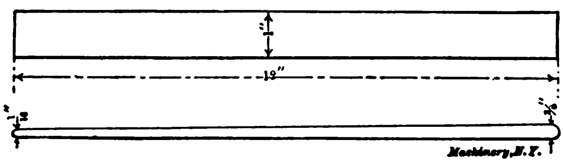

Fig. 58. Steel Drift.

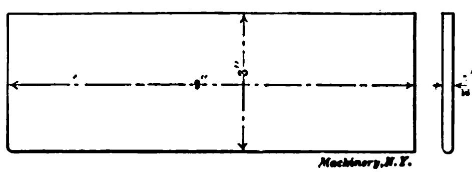

Fig. 59. Fiber Drift.

before mounting in the lathe so that it will not be necessary to remove the core after it is partly wound. The core should be placed in the lathe with the commutator end at the winder's left. The commutator end of an armature may be distinguished by the key-way cut in the shaft next to the core for the commutator key; also on railway armatures the shaft opposite the commutator end is beveled and threaded to fit the pinion as in Fig. 61.

A description of the winding of what is known as a No. 38 B railway motor will be given in detail:

The core of this armature is built on the shaft and has three ventilating ducts parallel to the shaft. There are 45 slots. These slots are relatively narrow as compared with the width of the teeth. It will be seen from Fig. 64 that the end plate of the commutator end fits against a shoulder turned on the shaft. The rear end plate is held in position by a nut which is screwed on to the shaft and held in place by a set-screw. Two duck blankets are used on this armature. They should be placed on the shaft with the wider side of the blanket out and with the seam toward the core. The core should be inspected to

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see whether any of the laminations or fingers project into the slots. The steel drift and rawhide mallet are used to clear the slot of any of these projections.

Cells.

In each slot are placed cells of paraffined express paper. They are made of a width such that when folded and placed in the slots the

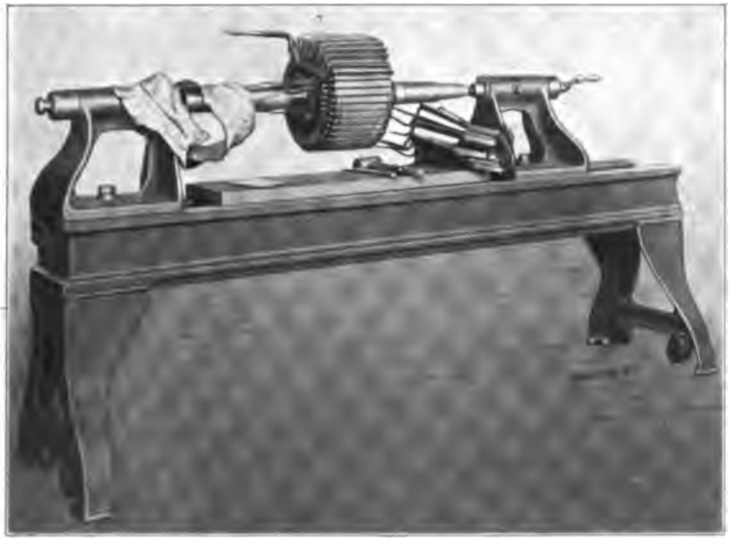

Fig 60. Winding Lathe.

edges will project above the core, and thus protect the coils as they are placed in the slots. The cells are stiff enough so that when bent into the slots they are not easily shaken out, as the armature is revolved in winding. If any cells are longer than the slots they should be cut off so that both ends of the cells will be flush with the ends of the slots.

Coils.

In winding this armature, 45 complete coils are used. Each coil, *i. e.*, complete coil (See Fig. 65), is made by assembling in a cell three

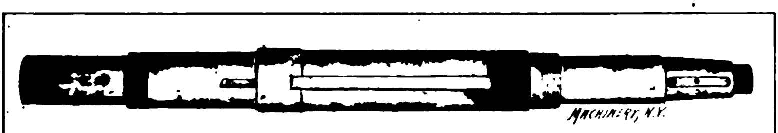

Fig. 61. Armature Shaft.

individual coils each consisting of two turns of No. 9, double cotton covered wire. Each slot contains one side of each of two different coils. One side of a coil is put in the bottom of one slot and the other side in the top of another slot. Three wires or leads are brought out from

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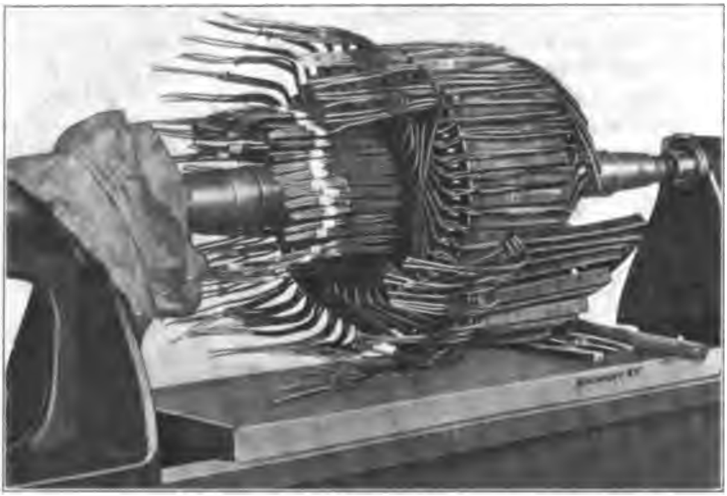

Fig. 62. Inserting the Coils.

each side of the coil on the under side. This type of coil is known as a "three-lead coil."

Taping.

The middle lead of the three coming from the bottom side of the coil is taped with black tape, the outside lead is taped with white tape, and then all three leads are taped together. The top leads are not taped but are bent up and outward, as shown in Fig. 62.

Putting Coils in the Slots.

Beginning with any slot the bottom of a coil is placed in it so that

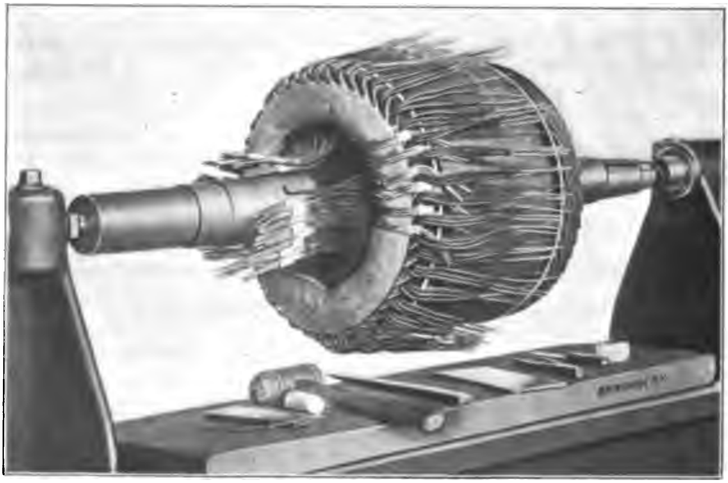

Fig. 63. Fitting the Canvas Blanket.

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each end of the coil is at an equal distance from the core, the top of the coil resting on the core. The bottom of the coil is forced to the bottom of the slot by means of the fiber drift and mallet. Call this slot No. 1, and count toward the top of the coil until slot No. 11 is reached. Start the top of the coil in this slot. This is called a throw of 1 and 11, or simply 11. The tops of the first ten coils are not forced into the slots as they must be taken out when the last ten coils are

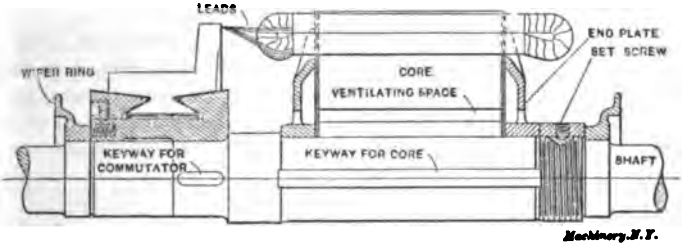

Fig. 64. Section through Armature.

put in place (See Fig. 62). The bottom of the next coil is placed in slot No. 45, and the top in No. 10. After the first eleven coils are in place the tops should also be driven into the slots. Continue in this manner around the armature until slot No. 11 is reached. Beginning with slot No. 45, take out the tops of all the coils up to and including the one in No. 11 slot and bend them away from the armature so that the bottom sides of the last ten coils can be placed in the slots. After the last ten coils have been placed in position, the tops of the coils which were removed to make place for the last ten coils are put in place.

A piece of heavy wire is wrapped around the coils at each end just outside the core and tightened with the pliers as firmly as possible.

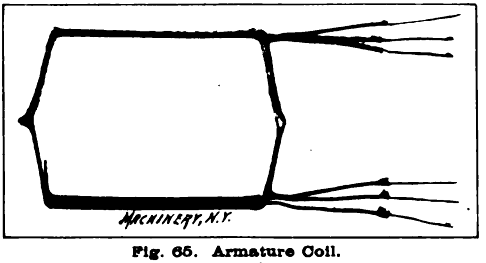

Fig. 65. Armature Coil.

This is to hold the coils in the slots while the winding is being tested, trued and connected. If the upper leads are not bare at the outer ends, the insulation should be scraped from them for about three inches. All the upper leads are then connected by a fine copper wire. Care must be taken that no leads touch the core or shaft as the leads are not required to be insulated sufficiently to withstand the voltage used in the insulation test. This test consists in applying 3,600 volts

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between the winding and the core. If the test shows a ground in any coil, the coil is removed and a new one substituted.

After the armature has passed the insulation test, the tops of the slot cells are cut off even with the core. Then the tops and ends of the coils are trued. To do this the armature is revolved in the lathe and a piece of chalk is held so that in turning the armature it will mark the coils that project. These are then driven down, or the others are brought out even with the high ones. The fronts of the coils are then trued. In Fig. 66 parts of the winding being trued are marked with white chalk. The blankets are next fitted over the front ends and sewed on with a curved needle and wax thread. The thread is passed in under the ends of the coils and brought up through them near the core and tied firmly over the blankets. They should be tied in at least six different places. The blankets are to separate and insulate the

Fig. 66. Truing-up the Winding.

leads from the ends of the coils after the leads are connected to the commutator (See Fig. 63).

The winder then stands on the opposite side of the lathe and takes the bottom lead of any coil and counts seven slots in a clockwise direction facing the commutator. This lead is bent up and across the ends of the coils and held in place by the lead from the seventh slot. Proceeding in the same manner around the armature in a counter-clockwise direction facing the commutator, all of the lower leads are bent like the first one and secured by the upper leads. This finishes the operation of winding. The armature is now ready to have a commutator pressed on.

Pressing on Commutator.

Small commutators are pressed on to the shaft by a hand press. All of the larger commutators are pressed on by means of a power press. In Fig. 67 is shown a hand press. The plate *B* is used in removing old commutators. It is placed back of the commutator as at *x y* with

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the slot *C* over the shaft. Bolts *a b* are passed through the holes *a a* in the plate and secured by nuts. The commutator, can then be forced off the shaft. In pressing on a commutator, a sleeve is placed over the shaft at *O*, and rests against the commutator. The rear end of the shaft is secured so it will withstand the pressure, and the commutator is forced on. The power presses are built on the principle of a hydraulic press. In pressing on a commutator a piece of babbitt metal or soft brass should be painted with white lead before having the commutator pressed on, in order to lubricate the shaft so that the commutator will press on easily. The wiper rings are pressed on after the commutator and then the armature is ready to be connected.

Connecting.

The first operation necessary in connecting is to "lay-off" the commutator. In "laying-off" the upper and lower leads of any coil are

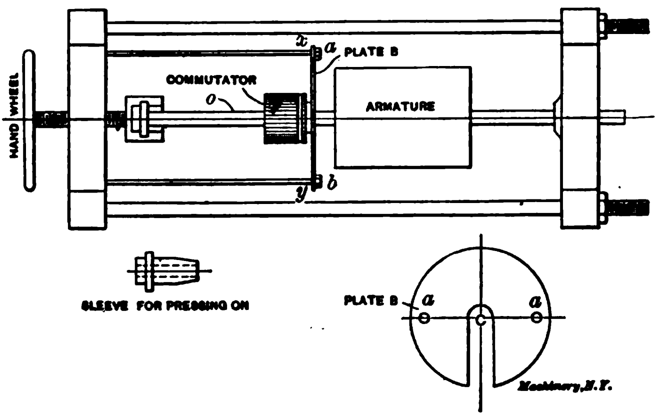

Fig. 67. Press for Forcing on and Removing Commutator.

found by means of a lighting-out set. The slots which contain this coil are marked with chalk. In connecting a No. 38 B railway motor armature the following should be noted: There are 135 bars in the commutator. The throw of coil is 1 and 11 and, as the winding is progressive, the commutator throw equals

$$\frac{\text{number of bars} + 3}{2} = 69, \text{ or } 1 \text{ and } 69.$$

With this commutator throw the center of the throw will be a bar. The throw of a coil is 1 and 11, therefore, the center of a coil throw will be a slot. Hence every slot should line up with a bar. By holding a pencil on the commutator perpendicular to it and sighting along the side of a coil the bar opposite the center of the slot in which the side lies may be located as at *A* in Fig. 68. Mark this bar with a colored pencil. Find the bar opposite the other side of the coil, as at *B*, and mark with the pencil, calling the slot in line with *A*, No. 1.

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Count 20 bars from *A*, in a clockwise direction and mark this bar No. 1. Also count 20 bars from *B* in a counter clockwise direction and mark this bar No. 69. Count from this bar to and including bar No. 1 and there should be 69 bars. Also there should be 29 bars between *A* and *B*. *D B A C* is called the forward throw and *D C* is the back throw. It is seen that the back throw is 66 or three less than the forward, as it always will be in a four-pole, progressive wave-wound armature. If an armature is wound retrogressively the forward and back throws differ by one. If, in laying-off, the center of the slot does not come in line with a bar, find one that will line up with a bar and proceed as above.

The 38B armature has three leads on each side of a coil and as

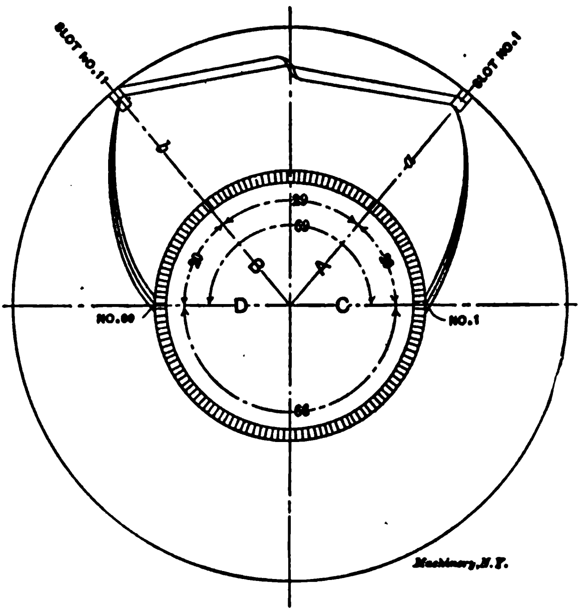

Fig. 68. Armature Correction Diagram.

there are 135 bars, there is no idle coil in this winding. Place the middle lead of the three coming from the bottom of slot No. 1 in bar No. 1, the outside lead in bar No. 135 and the inside lead in bar No. 2. Next take the lower leads from slot No. 2 and place them in bars No. 3, 4, and 5. The insulation should be removed from the leads where they are to be soldered to the commutator necks. They are driven to the bottom of the slot by means of a tool similar to the wedging tool only much thinner. The lower leads are all placed in the commutator and then they are "lighted-out."

Lighting-out.

The purpose of lighting-out is to see that there are no grounds or short circuits between the bars or coils, and to see if the leads are con-

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nected to the proper bars. The lighting-out set consists of two terminals connected in series with a 110-volt incandescent lamp to the 110-volt service lines.

One terminal of the lighting-out set is placed on bar No. 1 and the other on the middle lead coming from the top of same coil. The lamp should light. Next move the terminal on commutator bar No. 1 to bar No. 2 and if the lamp lights it shows a short-circuit between bars or between coils. If the lamp does not light the upper terminal is moved to the next lead counter-clockwise, when the lamp should light; if not, find the bar on which it will light and bring the wire connected to that bar to the proper bar. Continue in this manner around the commutator. After the winding is lighted-out, the ends of the leads projecting out over the commutator beyond the neck are cut off and saved, as they are to be used again.

Two layers of friction cloth are then wound over the lower leads and then the upper leads may be connected. The center lead from slot 11 is connected to bar No. 69, the outside lead is connected to

Fig. 69. The Completed Armature.

bar 70, the inside lead from slot No. 12 is connected to bar No. 71, and so on around the armature. After the leads are all placed in the slots in the commutator necks, they are driven to the bottom of the slots. The lower leads which were cut off are known as "dummies." These are driven into the tops of the slots until the slots are full. After putting in the dummies, all projecting ends are cut off and the armature is tested for grounds and short circuits. The leads are then soldered in the slots and the armature is then ready for banding.

Banding.

Tinned steel wire is used in banding. The bands on the core are insulated with mica and fullerboard while on the coils they are insulated with Japanese paper and tape. The insulation is made wide enough so that it projects one-eighth of an inch on each side of the bands. The bands on the core and lead are five-eighths of an inch wide, while the ones on the ends of the coils are made as wide as possible. In putting on the bands the armature is rotated in a lathe and the steel wire is wound on under tension. Clips are placed under

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the band wires and after sufficient turns have been wound on, the clips are bent over the wires and soldered to them, so that the band wires are held firmly together. After the bands are all on, they are heated with a soldering iron and solder run around each band. Thus the wire and clips are all held firmly in place.

Seven strips or bands are placed on the armature, four on the core, one on each end of the coils and one to hold the leads in place. These are shown in Fig. 69. The two bands on the rear end of the coils are connected to the last band on the core by means of three anchor clips spaced equally around the armature. This is done so there will be no danger of the outer bands slipping. After the armature is banded it is tested for short-circuits or grounds, given a coat of insulating paint and is then ready for assembling with the other motor parts.

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TABLES AND FORMULAS FOR SHOP AND DRAFTING-ROOM

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GENERAL TABLES AND FORMULAS.

The Use of Formulas.

Knowledge of algebra is not necessary for the use of formulas in solving simple problems; all the mathematical knowledge necessary is an understanding of the fundamental rules of arithmetic. The letters in formulas simply stand in place of the figures which are applied for specific cases or problems.

We know, for instance, that in spur gearing the pitch diameter times the pitch equals the number of teeth. Then, if $D =$ pitch diameter, $P =$ pitch, and $N =$ number of teeth, the formula expressing this rule is

$$D \times P = N.$$

The rule expressed in this formula says that whatever actual numbers may express the diameter and the pitch, these numbers multiplied will give the number of teeth. Assume that the diameter is 6 inches, and the pitch 4. Then if these numbers are substituted in the formula, we have

$$6 \times 4 = 24.$$

The number of teeth, N, then equals 24. In the same way, if $D = 16$, and $P = 8$, then the number of teeth, N, equals 128. In formulas, each letter stands for a certain dimension or quantity. When solving a problem by the use of a formula, simply replace the letters in the formula by the figures given in a certain problem, and find the result as in a regular arithmetical problem.

In all formulas and other mathematical expressions the various operations should be carried out in the order given, except that all multiplications should be carried out before additions, subtractions, and divisions; and divisions should be carried out before additions and subtractions.

Rules for Mensuration.

Triangle.—Area equals one-half the product of the base and the altitude.

Parallelogram.—Area equals the product of the base and the altitude.

Trapezoid.—Area equals one-half the sum of the parallel sides multiplied by the altitude.

Irregular figure bounded by straight lines.—Divide the figure in triangles, and find the area of each triangle separately. The sum of the areas of all the triangles equals the area of the figure.

Circle.—Circumference equals diameter multiplied by 3.1416.

Circle.—Diameter equals circumference divided by 3.1416.

Circle.—Area equals diameter squared, multiplied by 0.7854.

Circle.—Diameter equals area divided by 0.7854, and the square root extracted of the quotient.

Circular arc.—Length equals the circumference of the circle, multiplied by the number of degrees in the arc, divided by 360.

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Circular sector.—Area equals the area of the whole circle multiplied by the quotient of the number of degrees in the arc of the sector divided by 360.

Circular segment.—Area equals area of circular sector formed by drawing radii from the center of the circle to the extremities of the arc of the segment, minus area of triangle formed by the radii and the chord of the arc of the segment.

Prism.—Volume equals the area of the base multiplied by the altitude.

Cylinder.—Volume equals the area of the base circle times the altitude.

Pyramid or Cone.—Volume equals the area of the base times one-third the altitude.

Sphere.—Area equals the square of the diameter multiplied by 3.1416.

Sphere.—Volume equals the cube of the diameter times 0.5236.

Spherical sector.—Volume equals two-thirds of the square of the radius of the sphere multiplied by the height of the contained spherical segment multiplied by 3.1416.

Spherical segment.—Volume equals the radius of the sphere less one-third of the height of the segment, multiplied by the square of the height of the segment, multiplied by 3.1416.

MATHEMATICAL SIGNS

$+$	Plus (sign of addition)	$\sqrt{}$	Square root
$-$	Minus (sign of subtraction)	$\sqrt[3]{}$	Cube root
$\pm$	Plus or Minus	$\sqrt[4]{}$	Fourth root
$\times$	Times (multiplication sign)	$\sqrt[n]{}$	*n*th root
$\div$	Divided by (division sign)	a^2	*a* square (2nd power of *a*)
$:$	Divided by (ratio, proportion)	a^3	*a* cube (3d power of *a*)
$=$	Equals	a^4	fourth power of *a*
$\cong$	Approximately equals	a^n	*n*th power of *a*
$>$	Greater than	$\frac{1}{n}$	Reciprocal value of *n*
$<$	Less than	*log*	Logarithm
$\geqq$	Greater than or equal to	log_e / *hyp log*	Hyperbolic or natural logarithm
$\leqq$	Less than or equal to	$\angle$	Angle
$::$	Equals, (in proportion)	*sin*	Sine
$\therefore$	Therefore	*tan*	Tangent
∞	Infinity	*cos*	Cosine
$^\circ$	Degree	*cot*	Cotangent
$'$	Minute	*sec*	Secant
$''$	Second	*cosec*	Cosecant
π	(pi) 3.1416	δ	Delta (used for angles)
g	Acceleration due to gravity	ϕ	Phi (used for angles)
i	Imaginary quantity ($\sqrt{-1}$)	ω	Omega (used for angles)
α	Alpha (used for angles)		
β	Beta (used for angles)		
γ	Gamma (used for angles)		

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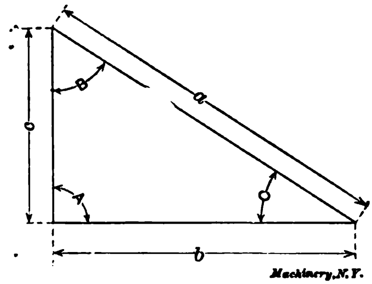
Trigonometrical Functions.

a = hypotenuse,
b = opposite side, to angle B.
c = adjacent side, to angle B.



	Rule.	Formula.
sine of B	= opposite side divided by hypotenuse.	$\sin B = \frac{b}{a}$
cosine of B	= adjacent side divided by hypotenuse.	$\cos B = \frac{c}{a}$
tangent of B	= opposite side divided by adjacent side.	$\tan B = \frac{b}{c}$
cotangent of B	= adjacent side divided by opposite side.	$\cot B = \frac{c}{b}$
secant of B	= hypotenuse divided by adjacent side.	$\sec B = \frac{a}{c}$
cosecant of B	= hypotenuse divided by opposite side.	$\text{cosec } B = \frac{a}{b}$

Formulas for Right-Angled Triangles.

Suppose, for instance, that we call the three sides in a right-angled triangle a, b, and c, as shown in the figure above, and the angles opposite those sides A, B, and C. The angle A, of course, is a right or 90-degree angle. Then, for all right-angled triangles these formulas hold true:

$$a = \frac{b}{\cos C}; \qquad a = \frac{b}{\sin B};$$
$$a = \frac{c}{\cos B}; \qquad a = \frac{c}{\sin C};$$
$$c = b \tan C; \qquad b = a \sin B;$$
$$c = a \text{ ccs } B; \qquad b = c \cot C;$$
$$b = c \tan B; \qquad c = a \sin C;$$
$$b = a \cos C; \qquad c = b \cot B;$$

It will be remembered that expressions such as $c \cot C$ mean simply $c \times \cot C$.

By means of the formulas given above, and a table of sines and tangents, either of the sides in a right-angled triangle may be found when

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one side and one angle, besides the 90-degree angle, are known. If two sides are known, but no angle outside of the 90-degree angle, the angles are found from the formulas:

$$\sin B = \frac{b}{a}; \qquad \cos B = \frac{c}{a};$$

$$\tan B = \frac{b}{c}; \qquad \cot B = \frac{c}{b};$$

$$\sin C = \frac{c}{a}; \qquad \cos C = \frac{b}{a};$$

$$\tan C = \frac{c}{b}; \qquad \cot C = \frac{b}{c};$$

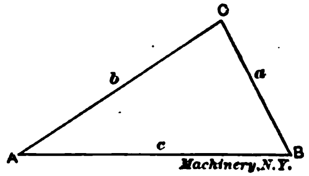
The third side may be found from the formulas

$$a = \sqrt{b^2 + c^2} \qquad b = \sqrt{a^2 - c^2} \qquad c = \sqrt{a^2 - b^2}.$$

The table on page 7 will prove helpful to readers who prefer to use the names for the different sides and angles rather than the letters in the formulas above. This table is arranged so that the required expressions for the parts to be found may be seen at a glance, as soon as the parts given have been located in the column to the left.

Oblique-Angled Triangles.

By means of the tables on pages 8 and 9, any part of an oblique triangle may be found when any three other parts are given. The parts given are found in the left-hand column, where a, b, and c de-



note the sides, and $\angle A$, $\angle B$, and $\angle C$ the angles A, B, and C, respectively, the same as in the triangle in the illustration above. When the line with the parts given has been located in the left-hand column of the table, simply use the formulas in the same line for finding the sides or angles which are not known. In some cases two steps are necessary for the solution; for example, if the sides a and b and the angle A are given, to find the side c; the formula for c is

$$c = \frac{a \sin C}{\sin A},$$

but angle B must first be found from the formula given for the sine for this angle, and then angle C is found from the formula

$$C = 180° - (A + B).$$

First after these operations are completed can the formula for c be applied.

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TABLE FOR SOLVING RIGHT-ANGLE TRIANGLES.

Parts Given	Parts to be Found				
	Hypotenuse (Hyp.)	Adjacent Side (Adj.)	Opposite Side (Opp.)	Angle	Opposite Angle
Hyp. and Adj.	——	——	$\sqrt{\text{Hyp.}^2 - \text{Adj.}^2}$	$\text{Cos} = \frac{\text{Adj.}}{\text{Hyp.}}$	$\text{Sin} = \frac{\text{Adj.}}{\text{Hyp.}}$
Hyp. and Opp.	——	$\sqrt{\text{Hyp.}^2 - \text{Opp.}^2}$	——	$\text{Sin} = \frac{\text{Opp.}}{\text{Hyp.}}$	$\text{Cos} = \frac{\text{Opp.}}{\text{Hyp.}}$
Hyp. and Angle	——	Hyp. × Cos	Hyp. × Sin	——	90° − Angle
Adj. and Opp.	$\sqrt{\text{Adj.}^2 + \text{Opp.}^2}$	——	——	$\text{Tan} = \frac{\text{Opp.}}{\text{Adj.}}$	$\text{Cot} = \frac{\text{Opp.}}{\text{Adj.}}$
Adj. and Angle	$\frac{\text{Adj.}}{\text{Cos.}}$	——	Adj. × Tan	——	90° − Angle
Opp. and Angle	$\frac{\text{Opp.}}{\text{Sin.}}$	Opp. × Cot	——	——	90° − Angle

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SOLUTION OF OBLIQUE ANGLE TRIANGLES.

Parts Given	Parts to be Found: $a =$	$b =$	$c =$	$\angle A$	$\angle B$	$\angle C$
a–b–c				$\frac{b^2+c^2-a^2}{2bc} = \cos A$	$\frac{a^2+c^2-b^2}{2ac} = \cos B$	$\frac{a^2+b^2-c^2}{2ab} = \cos C$
b–c–$\angle A$	$\sqrt{b^2+c^2-2bc\cos A}$				$\frac{b \sin A}{c - b\cos A} = \tan B$	$\frac{c \sin A}{b - c\cos A} = \tan C$
a–c–$\angle B$		$\sqrt{a^2+c^2-2ac\cos B}$		$\frac{a \sin B}{c - a\cos B} = \tan A$		$\frac{c \sin B}{a - c\cos B} = \tan C$
a–b–$\angle C$			$\sqrt{a^2+b^2-2ab\cos C}$	$\frac{a \sin C}{b - a\cos C} = \tan A$	$\frac{b \sin C}{a - b\cos C} = \tan B$	
a–b–$\angle A$			$\frac{a \sin C}{\sin A}$		$\frac{b \sin A}{a} = \sin B$	$180° - (A + B)$
a–b–$\angle B$			$\frac{b \sin C}{\sin B}$	$\frac{a \sin B}{b} = \sin A$		$180° - (A + B)$
a–c–$\angle A$		$\frac{a \sin B}{\sin A}$			$180° - (A + C)$	$\frac{c \sin A}{a} = \sin C$
a–c–$\angle C$		$\frac{c \sin B}{\sin C}$		$\frac{a \sin C}{c} = \sin A$	$180° - (A + C)$	
b–c–$\angle B$	$\frac{b \sin A}{\sin B}$			$180° - (B + C)$		$\frac{c \sin B}{b} = \sin C$

See page 6 for illustration and explanatory notes.

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Parts Given	Parts to be Found					
	$a =$	$b =$	$c =$	$\angle A$	$\angle B$	$\angle C$
b–c–$\angle C$	$\frac{c \sin A}{\sin C}$			$180° - (B + C)$	$\frac{b \sin C}{c} = \sin B$	
a–$\angle A$–$\angle B$		$\frac{a \sin B}{\sin A}$	$\frac{a \sin C}{\sin A}$			$180° - (A + B)$
a–$\angle A$–$\angle C$		$\frac{a \sin B}{\sin A}$	$\frac{a \sin C}{\sin A}$		$180° - (A + C)$	
a–$\angle B$–$\angle C$		$\frac{a \sin B}{\sin A}$	$\frac{a \sin C}{\sin A}$	$180° - (B + C)$		
b–$\angle A$–$\angle B$	$\frac{b \sin A}{\sin B}$		$\frac{b \sin C}{\sin B}$			$180° - (A + B)$
b–$\angle A$–$\angle C$	$\frac{b \sin A}{\sin B}$		$\frac{b \sin C}{\sin B}$		$180° - (A + C)$	
b–$\angle B$–$\angle C$	$\frac{b \sin A}{\sin B}$		$\frac{b \sin C}{\sin B}$	$180° - (B + C)$		
c–$\angle A$–$\angle B$	$\frac{c \sin A}{\sin C}$	$\frac{c \sin B}{\sin C}$				$180° - (A + B)$
c–$\angle A$–$\angle C$	$\frac{c \sin A}{\sin C}$	$\frac{c \sin B}{\sin C}$			$180° - (A + C)$	
c–$\angle B$–$\angle C$	$\frac{c \sin A}{\sin C}$	$\frac{c \sin B}{\sin C}$		$180° - (B + C)$		

See page 6 for illustration and explanatory notes.

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Formulas for Falling Bodies.

s = space in feet which the body passes through in the time t,
u = space in feet which the body falls in the tth second,
v = velocity in feet per second of falling body at the end of the time t,
t = time in seconds the body is falling,
g = acceleration due to gravity = 32.2 feet per second.

$$v = g\,t = \frac{2\,s}{t} = \sqrt{2\,g\,s} = 8.02\sqrt{s}$$

$$s = g\frac{t^2}{2} = \frac{vt}{2} = \frac{v^2}{2g} = \frac{v^2}{64.4}$$

$$t = \frac{v}{g} = \frac{2s}{v} = \sqrt{\frac{2\,s}{g}} = \frac{\sqrt{s}}{4.01} = \frac{u}{g} + \frac{1}{2}$$

Horse-Power Formula for Steam Engines.

P = mean effective steam pressure in pounds per square inch,
D = diameter of cylinder piston in inches,
L = length of stroke in feet,
N = number of double strokes per minute,
$H\,P$ = horse-power of engine,
A = area of piston in square inches,

$$H\,P = \frac{238\,PLD^2N}{10{,}000{,}000} = \frac{PLAN}{33{,}000}.$$

General Horse-Power Formula.

P = power in foot-pounds per second,

$$H\,P = \frac{P}{550}.$$

Relation Between Circumferential Velocity and Revolutions per Minute.

v = circumferential velocity per second,
R = radius of revolving body,
n = number of revolutions per minute,

$$v = \frac{2\pi\,Rn}{60} = 0.1047\;Rn, \qquad n = \frac{9.55\,v}{R}.$$

Formulas for Centrifugal Force.

F = centrifugal force in pounds,
W = weight of revolving body in pounds,
v = velocity of revolving body in feet per second,
R = radius of circle in which body revolves, in feet,
n = number of revolutions per minute.

$$F = \frac{W\,v^2}{32.2\,R} = \frac{W\,R\,n^2}{2933};\; W = \frac{2933\,F}{R\,n^2};\; R = \frac{2933\,F}{W\,n^2};\; n = \sqrt{\frac{2933\,F}{W\,R}}$$

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ULTIMATE STRENGTH OF COMMON METALS: POUNDS PER SQUARE INCH.

Material	Tension	Compression	Shear	Modulus of Elasticity
Aluminum	15,000	12,000	12,000	11,000,000
Brass, cast	24,000	30,000	36,000	9,000,000
Bronze, gun-metal	32,000	20,000	——	10,000,000
Bronze, manganese	60,000	120,000	——	——
Bronze, phosphor	50,000	——	——	14,000 000
Copper, cast	24,000	40,000	30,000	10,000,000
Copper Wire, annealed	36,000	——	——	15,000,000
Copper Wire, unannealed	60,000	——	——	18,000,000
Iron, cast	15,000	80,000	18,000	12,000,000
Iron Wire, annealed	60,000	——	——	15,000.000
Iron Wire, unannealed	80,000	——	——	25,000.000
Iron, wrought	48,000	46,000	40,000	27,000,000
Lead, cast	2,000	——	——	1,000,000
Steel Castings	70,000	70,000	60,000	30,000,000
Steel, plow	268,000	——	——	——
Steel, structural	60,000	60,000	50,000	29,000,000
Steel Wire, annealed	80,000	——	——	29,000,000
Steel Wire, unannealed	120,000	——	——	30,000,000
Steel Wire, crucible	180,000	——	——	30 000,000
Steel Wire, susp. bridge	200,000	——	——	30,000,000
Steel Wire, piano	300,000	——	——	——
Tin, cast	3,500	6,000	——	4,000,000
Zinc, cast	5,000	20,000	——	13,000,000

AVERAGE STRENGTH OF COMMON MATERIALS OTHER THAN METALS.

Material	Compression	Tension
Bricks, best hard	12,000	400
Bricks, light red	1,000	40
Brickwork, common	1,000	50
Brickwork, best	2,000	300
Cement, Portland, one month old	2,000	400
Cement, Portland, one year old	3,000	500
Concrete, Portland	1.000	200
Concrete, Portland, one year old	2,000	400
Hemlock	4,000	6,000
Pine, shortleaf yellow	6,000	9,000
Pine, Georgia	8,000	12,000
Pine, White	5,500	7,000
White Oak	7,000	10,000

FACTORS OF SAFETY.

Material	Steady Load	Load Varying from Zero to Maximum in one Direction	Load Varying from Zero to Maximum in both Directions	Suddenly Varying Loads and Shocks
Cast Iron	6	10	15	20
Wrought Iron	4	6	8	12
Steel	5	6	8	12
Wood	8	10	15	20
Brick	15	20	25	30
Stone	15	20	25	30

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WORKING STRENGTH OF BOLTS.*

Diameter of Bolt, inches.	Area at Root of Thread, square inches.	Working Section, square inches.	Strength of Bolt, 5,000 pounds Stress.	Strength of Bolt, 6,000 pounds Stress.	Strength of Bolt, 7,000 pounds Stress.	Strength of Bolt, 8,000 pounds Stress.	Strength of Bolt, 10,000 pounds Stress.	Strength of Bolt, 12,000 pounds Stress.
$\frac{5}{8}$	0.202	0.044	220	264	308	352	440	528
$\frac{3}{4}$	0.302	0.113	565	678	791	904	1,130	1,356
$\frac{7}{8}$	0.420	0.200	1,000	1,200	1,400	1,600	2,000	2,400
1	0.550	0.298	1,490	1,788	2,086	2,384	2,980	3,476
1 $\frac{1}{8}$	0.694	0.411	2,055	2,466	2,877	3,288	4,110	4,932
1 $\frac{1}{4}$	0.893	0.578	2,890	3,468	4,046	4,624	5,780	6,936
1 $\frac{3}{8}$	1.057	0.710	3,550	4,260	4,970	5,680	7,100	8,520
1 $\frac{1}{2}$	1.295	0.917	4,585	5,502	6,419	7,336	9,170	10,504
1 $\frac{5}{8}$	1.515	1.105	5,525	6,630	7,735	8,840	11,050	13,260
1 $\frac{3}{4}$	1.746	1.305	6,525	7,830	9,135	10,440	13,050	15,660
1 $\frac{7}{8}$	2.051	1.578	7,890	9,468	11,046	12,624	15,780	18,936
2	2.302	1.798	8,990	10,788	12,586	14,384	17,980	21,576
2 $\frac{1}{4}$	3.023	2.456	12,280	14,736	17,192	19,648	24,560	29,472
2 $\frac{1}{2}$	3.719	3.089	15,445	18,534	21,623	24,712	30,890	37,068
2 $\frac{3}{4}$	4.620	3.927	19,635	23,562	27,489	31,416	39,270	47,124
3	5.428	4.672	23,360	28,032	32,704	37,376	46,720	56,064
3 $\frac{1}{4}$	6.510	5.690	28,450	34,140	39,830	45,520	56,900	68,280
3 $\frac{1}{2}$	7.548	6.666	33,330	39,996	46,664	53,328	66,660	79,992

* The figures for the working strength of bolts as given in the table above show the stress to which it is safe to subject the bolt when due allowance has been made for the stresses in the bolt caused by tightening the nut. The table refers specifically to bolts for cylinder covers, receivers containing fluids under pressure, boilers, etc. In work of this character bolts less than ⅝-inch diameter should not be employed.

Formulas for Strength of Materials.

In the following formulas:

A = area of cross-section of material in square inches,

E = modulus of elasticity,

I = moment of inertia of section about an axis passing through the center of gravity,

J = polar moment of inertia of section,

M_b = maximum bending moment in inch-pounds,

M_t = moment of force tending to twist (torsional moment) in inch-pounds,

P = total stress in pounds,

y = distance from center of gravity to most remote fiber,

S = permissible working stress in pounds per square inch,

Z = section modulus (moment of resistance),

e = elongation or shortening in inches,

l = length in inches,

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For tension and compression:

$$P = A \times S; \qquad e = \frac{Pl}{AE}.$$

For shear:

$$P = A \times S.$$

Assume permissible working stress for shear to equal four-fifths the permissible stress in tension.

For bending:

$$M_b = \frac{SI}{y} = SZ$$

For torsion:

$$M_t = \frac{SJ}{y}.$$

The permissible working stress for torsion may be assumed as four-fifths the permissible stress in tension.

Combined bending and torsion:

$$\text{Combined moment} = 0.35\ M_b + 0.65\ \sqrt{M_b^2 + M_t^2}$$

SPECIFIC GRAVITY, FUSING POINT, AND CHARACTERISTICS OF COMMON METALS.

Metal	Melting Point, Degrees F.	Specific Gravity	Color *	Structure **	Electric Conductivity; Silver=100.	Approx. Value per Pound, Dollars	Weight per Cubic Inch, Pounds.
Aluminum	1157	2.56	B-W	M	63.00	0.24	0.0924
Antimony	842	6.71	B-W	B	3.59	0.08	0.2424
Barium	2192	3.75	Y	M	30.61	1025.00	0.1355
Bismuth	485	9.80	G-W	B	1.40	1.60	0.3540
Cadmium	576	8.60	W	M	24.38	1.92	0.3107
Calcium	1472	1.57	Y	M	21.77	2500.00	0.0567
Chromium	4000	6.80	G-W	B	16.00	0.80	0.2457
Cobalt	2932	8.50	P-W	M	16.93	1.60	0.3071
Copper	1929	8.82	P-R	M	97.67	0.14	0.3186
Gold	1913	19.32	Y	M	76.71	288.00	0.6979
Iron, cast	2000	7.20	G-W	B	——	0.01	0.2601
pure	2912	7.80	W	M	14.57	0.01	0.2816
wrought	2800	7.70	G-W	M	——	0.02	0.2780
Lead	618	11.37	B-W	S	8.42	0.04	0.4108
Magnesium	1200	1.74	B-W	M	39.44	4.80	0.0629
Manganese	3452	8.00	G-W	B	15.75	16.00	0.2890
Mercury	—39	13.58	B-W	F	1.75	0.48	0.4909
Nickel	2912	8.80	Y-W	M	12.89	0.45	0.3179
Platinum	3227	21.50	W	M	14.43	324.00	0.7767
Potassium	144	0.87	B-W	S	19.62	3.20	0.0314
Silver	1733	10.53	W	M	100.00	10.40	0.3805
Steel, tool	2532	7.85	W	M	12.00	0.08	0.2837
Tin	446	7.29	W	M	14.39	0.29	0.2634
Tungsten	4000	17.60	W	B	14.00	0.64	0.6353
Vanadium	4278	5.50	W	M	4.95	2500.00	0.1987
Zinc	779	7.15	B-W	M	29.57	0.04	0.2581

* B = blue, G = grey, P = pink, R = red, W = white, Y = yellow.
** B = brittle, F = fluid, M = malleable, S = soft.

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DECIMAL EQUIVALENTS OF FRACTIONS OF AN INCH.

1/64	0.015 625	11/32	0.343 75	43/64	0.671 875
1/32	0.031 25	23/64	0.359 375	11/16	0.687 5
3/64	0.046 875	3/8	0.375	45/64	0.703 125
1/16	0.062 5	25/64	0.390 625	23/32	0.718 75
5/64	0.078 125	13/32	0.406 25	47/64	0.734 375
3/32	0.093 75	27/64	0.421 875	3/4	0.750
7/64	0.109 375	7/16	0.437 5	49/64	0.765 625
1/8	0.125	29/64	0.453 125	25/32	0.781 25
9/64	0.140 625	15/32	0.468 75	51/64	0.796 875
5/32	0.156 25	31/64	0.484 375	13/16	0.812 5
11/64	0.171 875	1/2	0.500	53/64	0.828 125
3/16	0.187 5	33/64	0.515 625	27/32	0.843 75
13/64	0.203 125	17/32	0.531 25	55/64	0.859 375
7/32	0.218 75	35/64	0.546 875	7/8	0.875
15/64	0.234 375	9/16	0.562 5	57/64	0.890 625
1/4	0.250	37/64	0.578 125	29/32	0.906 25
17/64	0.265 625	19/32	0.593 75	59/64	0.921 875
9/32	0.281 25	39/64	0.609 375	15/16	0.937 5
19/64	0.296 875	5/8	0.625	61/64	0.953 125
5/16	0.312 5	41/64	0.640 625	31/32	0.968 75
21/64	0.328 125	21/32	0.656 25	63/64	0.984 375

Metric Conversion Table.

Linear Measure.

1 kilometer = 0.6214 mile
1 meter = { 39.37 inches; 3.2808 feet }
1 centimeter = 0.3937 inch
1 millimeter = 0.03937 inch

1 mile = 1.609 kilometer
1 foot = 0.3048 meter
1 inch = 2.54 centimeters
1 inch = 25.4 millimeters

Square Measure.

1 sq. kilometer = 0.386 sq. mile
1 hectare = 2.47 acres
1 are = 1076.4 sq. feet
1 sq. meter = 10.764 sq. feet
1 sq. centimeter = 0.155 sq. inch
1 sq. millimeter = 0.00155 sq. inch

1 sq. mile = 2.589 sq. kilometers
1 acre = 0.405 hectares
1 sq. foot = 0.0929 sq. meter
1 sq. inch = 6.452 sq. centimeters
1 sq. inch = 645.2 sq. millimeters

Cubic Measure.

1 cub. meter = 35.314 cub. feet
1 cub. centimeter = 0.061 cub. inch
1 liter = 0.2642 U. S. gallon
1 liter = 0.0353 cubic foot.

1 cub. foot = 0.02832 cub. meter
1 cub. inch = 16.383 cub. centimeters
1 cub. foot = 28.317 liters
1 U. S. gallon = 3.785 liters

Weight.

1 metric ton = 2,204.6 pounds
1 kilogram = 2.2046 pounds
1 gram = 15.432 grains troy

1 pound = 0.4536 kilogram
1 ounce = 28.35 grams
1 grain troy = 0.0648 grams

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MILLIMETERS INTO ENGLISH INCHES.

	0	1	2	3	4	5	6	7	8	9
	0.0000	0.0394	0.0787	0.1181	0.1575	0.1969	0.2362	0.2756	0.3150	0.3543
1	0.3937	0.4331	0.4724	0.5118	0.5512	0.5906	0.6299	0.6693	0.7087	0.7480
2	0.7874	0.8268	0.8661	0.9055	0.9449	0.9843	1.0236	1.0630	1.1024	1.1417
3	1.1811	1.2205	1.2598	1.2992	1.3386	1.3780	1.4173	1.4567	1.4961	1.5354
4	1.5748	1.6142	1.6536	1.6929	1.7323	1.7717	1.8110	1.8504	1.8898	1.9292
5	1.9685	2.0079	2.0472	2.0866	2.1260	2.1653	2.2047	2.2441	2.2834	2.3228
6	2.3622	2.4016	2.4409	2.4803	2.5197	2.5591	2.5984	2.6378	2.6772	2.7165
7	2.7560	2.7953	2.8346	2.8740	2.9134	2.9528	2.9921	3.0315	3.0709	3.1102
8	3.1497	3.1890	3.2284	3.2677	3.3071	3.3465	3.3858	3.4252	3.4646	3.5040
9	3.5434	3.5827	3.6221	3.6614	3.7008	3.7402	3.7795	3.8189	3.8583	3.8977
10	3.9371	3.9764	4.0158	4.0551	4.0945	4.1339	4.1733	4.2126	4.2520	4.2914
11	4.3308	4.3701	4.4095	4.4488	4.4882	4.5276	4.5670	4.6063	4.6457	4.6851
12	4.7245	4.7638	4.8032	4.8426	4.8819	4.9213	4.9607	5.0000	5.0394	5.0788
13	5.1182	5.1575	5.1969	5.2363	5.2756	5.3150	5.3544	5.3937	5.4331	5.4725
14	5.5119	5.5512	5.5906	5.6300	5.6693	5.7087	5.7481	5.7874	5.8268	5.8662
15	5.9056	5.9450	5.9843	6.0237	6.0631	6.1024	6.1418	6.1812	6.2205	6.2599
16	6.2993	6.3386	6.3780	6.4174	6.4568	6.4961	6.5355	6.5749	6.6142	6.6536
17	6.6930	6.7324	6.7717	6.8111	6.8505	6.8898	6.9292	6.9686	7.0079	7.0473
18	7.0867	7.1261	7.1654	7.2048	7.2442	7.2835	7.3229	7.3623	7.4017	7.4410
19	7.4805	7.5198	7.5591	7.5985	7.6379	7.6773	7.7166	7.7560	7.7954	7.8347
20	7.8742	7.9135	7.9528	7.9922	8.0316	8.0710	8.1103	8.1497	8.1891	8.2284
21	8.2677	8.3072	8.3466	8.3859	8.4253	8.4647	8.5040	8.5434	8.5828	8.6221
22	8.6614	8.7009	8.7403	8.7796	8.8190	8.8584	8.8977	8.9371	8.9765	9.0158
23	9.0553	9.0946	9.1340	9.1733	9.2127	9.2521	9.2915	9.3308	9.3702	9.4096
24	9.4490	9.4883	9.5277	9.5670	9.6064	9.6458	9.6852	9.7245	9.7639	9.8033
25	9.8427	9.8820	9.9214	9.9608	10.0000	10.0395	10.0789	10.1182	10.1576	10.1970
26	10.2364	10.2757	10.3151	10.3545	10.3938	10.4332	10.4726	10.5119	10.5513	10.5907
27	10.6301	10.6694	10.7088	10.7482	10.7875	10.8269	10.8663	10.9057	10.9450	10.9844
28	11.0238	11.0631	11.1025	11.1419	11.1813	11.2206	11.2600	11.2994	11.3387	11.3781
29	11.4175	11.4568	11.4962	11.5356	11.5750	11.6143	11.6537	11.6931	11.7324	11.7718
30	11.8112	11.8506	11.8899	11.9293	11.9687	12.0080				

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ENGLISH INCHES INTO MILLIMETERS.

Inches	0	1/16	1/8	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16
0	0.0	1.6	3.2	4.8	6.4	7.9	9.5	11.1	12.7	14.3	15.9	17.5	19.1	20.6	22.2	23.8
1	25.4	27.0	28.6	30.2	31.7	33.3	34.9	36.5	38.1	39.7	41.3	42.9	44.4	46.0	47.6	49.2
2	50.8	52.4	54.0	55.6	57.1	58.7	60.3	61.9	63.5	65.1	66.7	68.3	69.8	71.4	73.0	74.6
3	76.2	77.8	79.4	81.0	82.5	84.1	85.7	87.3	88.9	90.5	92.1	93.7	95.2	96.8	98.4	100.0
4	101.6	103.2	104.8	106.4	108.0	109.5	111.1	112.7	114.3	115.9	117.5	119.1	120.7	122.2	123.8	125.4
5	127.0	128.6	130.2	131.8	133.4	134.9	136.5	138.1	139.7	141.3	142.9	144.5	146.1	147.6	149.2	150.8
6	152.4	154.0	155.6	157.2	158.8	160.3	161.9	163.5	165.1	166.7	168.3	169.9	171.5	173.0	174.6	176.2
7	177.8	179.4	181.0	182.6	184.2	185.7	187.3	188.9	190.5	192.1	193.7	195.3	196.9	198.4	200.0	201.6
8	203.2	204.8	206.4	208.0	209.6	211.1	212.7	214.3	215.9	217.5	219.1	220.7	222.3	223.8	225.4	227.0
9	228.6	230.2	231.8	233.4	235.0	236.5	238.1	239.7	241.3	242.9	244.5	246.1	247.7	249.2	250.8	252.4
10	254.0	255.6	257.2	258.8	260.4	261.9	263.5	265.1	266.7	268.3	269.9	271.5	273.1	274.6	276.2	277.8
11	279.4	281.0	282.6	284.2	285.7	287.3	288.9	290.5	292.1	293.7	295.3	296.9	298.4	300.0	301.6	303.2
12	304.8	306.4	308.0	309.6	311.1	312.7	314.3	315.9	317.5	319.1	320.7	322.3	323.8	325.4	327.0	328.6
13	330.2	331.8	333.4	335.0	336.5	338.1	339.7	341.3	342.9	344.5	346.1	347.7	349.2	350.8	352.4	354.0
14	355.6	357.2	358.8	360.4	361.9	363.5	365.1	366.7	368.3	369.9	371.5	373.1	374.6	376.2	377.8	379.4
15	381.0	382.6	384.2	385.8	387.3	388.9	390.5	392.1	393.7	395.3	396.9	398.5	400.0	401.6	403.2	404.8
16	406.4	408.0	409.6	411.2	412.7	414.3	415.9	417.5	419.1	420.7	422.3	423.9	425.4	427.0	428.6	430.2
17	431.8	433.4	435.0	436.6	438.1	439.7	441.3	442.9	444.5	446.1	447.7	449.3	450.8	452.4	454.0	455.6
18	457.2	458.8	460.4	462.0	463.5	465.1	466.7	468.3	469.9	471.5	473.1	474.7	476.2	477.8	479.4	481.0
19	482.6	484.2	485.8	487.4	488.9	490.5	492.1	493.7	495.3	496.9	498.5	500.1	501.6	503.2	504.8	506.4
20	508.0	509.6	511.2	512.8	514.3	515.9	517.5	519.1	520.7	522.3	523.9	525.5	527.0	528.6	530.2	531.8
21	533.4	535.0	536.6	538.2	539.7	541.3	542.9	544.5	546.1	547.7	549.3	550.9	552.4	554.0	555.6	557.2
22	558.8	560.4	562.0	563.6	565.1	566.7	568.3	569.9	571.5	573.1	574.7	576.3	577.8	579.4	581.0	582.6
23	584.2	585.8	587.4	589.0	590.5	592.1	593.7	595.3	596.9	598.5	600.1	601.7	603.2	604.8	606.4	608.0

For transforming English inches into millimeters, when only a rough estimate is required, it is sufficient to assume one inch = 25 millimeters. As it is very easy to multiply mentally by 25, any number of inches can be transformed approximately without tables or calculations. When transforming millimeters to inches, the number of millimeters is, of course, simply divided by 25 to obtain an approximate result.

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GEARING.

Formulas for Spur Gearing.

In the following formulas,

P = diametral pitch,

P_1 = circular pitch,

D = pitch diameter,

D_1 = outside diameter,

N = number of teeth in one gear,

n = number of teeth in mating gear,

A = addendum,

T = thickness of tooth at pitch line,

E = full depth of tooth,

C = distance between centers,

F = clearance.

$$P_1 = \frac{3.1416}{P} \qquad T = \frac{P_1}{2} \qquad P = \frac{N+2}{D_1}$$

$$P = \frac{3.1416}{P_1} \qquad T = \frac{1.5708}{P} \qquad D = D_1 - \frac{2}{P}$$

$$D = \frac{N}{P} \qquad E = \frac{2.157}{P} \qquad D = \frac{D_1 \times N}{N+2}$$

$$A = \frac{1}{P} \qquad E = 0.6866\, P_1 \qquad N = P \times D$$

$$D_1 = \frac{N+2}{P} \qquad C = \frac{N+n}{2P} \qquad N = (D_1 \times P) - 2$$

$$F = \frac{0.157}{P} \text{ or } \frac{A}{8} \qquad P = \frac{N}{D} \qquad D_1 = D + \frac{2}{P}$$

According to the system for cutting gear teeth adopted by the Brown & Sharpe Mfg. Co., any gear of one pitch will mesh into any other gear or into a rack of the same pitch. Eight cutters are required for each pitch. These eight cutters are adapted to cut from a pinion of twelve teeth to a rack, and are numbered respectively, 1, 2, 3, etc. The number of teeth and the pitch for which a cutter is adapted is always marked on each.

No. 1	will cut	wheels from	135	teeth to	a rack.	
" 2	"	"	55	"	134	teeth.
" 3	"	"	35	"	54	"
" 4	"	"	26	"	34	"
" 5	"	"	21	"	25	"
" 6	"	"	17	"	20	"
" 7	"	"	14	"	16	"
" 8	"	"	12	"	13	"

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TABLE OF DIAMETRAL AND CIRCULAR PITCH.

Diametral into Circular Pitch.				Circular into Diametral Pitch.			
Diametral Pitch	Circular Pitch	Diametral Pitch	Circular Pitch	Circular Pitch	Diametral Pitch	Circular Pitch	Diametral Pitch
1 1/4	2.513"	11	0.286"	2"	1.571	7/8"	3.590
1 1/2	2.094	12	0.262	1 7/8	1.676	13/16	3.867
1 3/4	1.795	14	0.224	1 3/4	1.795	3/4	4.189
2	1.571	16	0.196	1 5/8	1.933	11/16	4.570
2 1/4	1.396	18	0.175	1 1/2	2.094	5/8	5.027
2 1/2	1.257	20	0.157	1 7/16	2.185	9/16	5.585
2 3/4	1.142	22	0.143	1 3/8	2.285	1/2	6.283
3	1.047	24	0.131	1 5/16	2.394	7/16	7.181
3 1/2	0.898	26	0.121	1 1/4	2.513	3/8	8.378
4	0.785	28	0.112	1 3/16	2.646	5/16	10.053
5	0.628	30	0.105	1 1/8	2.793	1/4	12.566
6	0.524	32	0.098	1 1/16	2.957	3/16	16.755
7	0.449	36	0.087	1	3.142	1/8	25.133
8	0.393	40	0.079	15/16	3.351	1/16	50.266
9	0.349	48	0.065				
10	0.314						

TABLE SHOWING DEPTH OF SPACE AND THICKNESS OF TOOTH IN SPUR GEARS CUT WITH B. & S. MFG. CO.'S CUTTERS.

Pitch of Cutter	Depth to be Cut in Gear	Thickness of Tooth at Pitch Line	Pitch of Cutter	Depth to be Cut in Gear	Thickness of Tooth at Pitch Line
1 1/4	1.726"	1.257"	11	0.196"	0.143"
1 1/2	1.438	1.047	12	0.180	0.131
1 3/4	1.233	0.898	14	0.154	0.112
2	1.078	0.785	16	0.135	0.098
2 1/4	0.958	0.697	18	0.120	0.087
2 1/2	0.863	0.628	20	0.108	0.079
2 3/4	0.784	0.570	22	0.098	0.071
3	0.719	0.523	24	0.090	0.065
3 1/2	0.616	0.448	26	0.083	0.060
4	0.539	0.393	28	0.077	0.056
5	0.431	0.314	30	0.072	0.052
6	0.359	0.262	32	0.067	0.049
7	0.308	0.224	36	0.060	0.044
8	0.270	0.196	40	0.054	0.039
9	0.240	0.175	48	0.045	0.033
10	0.216	0.157			

Formulas for Bevel Gearing.

On pages 19, 20, 21, and 22 are given complete sets of bevel gear formulas, covering the cases when the angle between the two shafts to be connected by the gearing equals 90 degrees, more than 90 degrees, and less than 90 degrees; a set of formulas applicable to the solution of miter gears is also given. These formulas are systematically arranged so that the proper formula to use can be very easily found at a glance; and no knowledge of trigonometry is required, except an understanding of the use of tables of sines and tangents. The formulas, in substantially the same form as presented in the accom-

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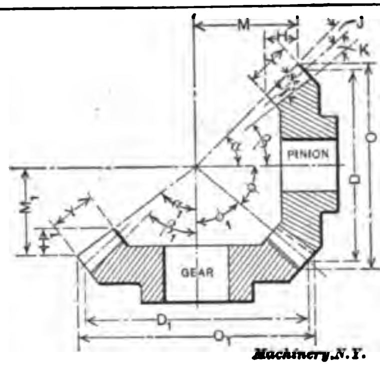
SHAFT ANGLE 90 DEGREES.

P = diametral pitch
C = circular pitch
N = number of teeth in pinion
N_1 = number of teeth in gear
x = shaft angle
N_2 = number of teeth for which to select cutter

$$J = \frac{1}{P} = 0.3183\ C$$

$$K = \frac{1.157}{P} = 0.3683\ C$$



Pinion	Gear
$N = D \times P = \frac{D \times \pi}{C}$	$N_1 = D_1 \times P = \frac{D_1 \times \pi}{C}$
$D = \frac{N}{P} = 0.3183\ C\ N$	$D_1 = \frac{N_1}{P} = 0.3183\ C\ N_1$
$O = D + \frac{2 \cos \phi}{P} = D + 0.6366\ C \times \cos \phi$	$O_1 = D_1 + \frac{2 \sin \phi}{P} = D_1 + 0.6366\ C \times \sin \phi$
$\tan \phi = \frac{N}{N_1}$	$\phi_1 = 90 - \phi$

$$\tan s = \frac{2 \sin \phi}{N}, \quad \tan f = \frac{2.314 \sin \phi}{N}$$

Pinion	Gear
$\alpha = \phi + s$	$\alpha_1 = \phi_1 + s$
$\beta = \phi - f$	$\beta_1 = \phi_1 - f$
$M = \frac{O_1}{2} - \frac{2 \sin \phi}{P} = \frac{O}{2} \times \cot \alpha = \frac{O_1}{2} - 0.6366\ C \times \sin \phi$	$M_1 = \frac{O}{2} - \frac{2 \cos \phi}{P} = \frac{O_1}{2} \times \cot \alpha_1 = \frac{O}{2} - 0.6366\ C \times \cos \phi$
$H = Y \times \cos \alpha$	$H_1 = Y \times \cos \alpha_1$
$N_2 = \frac{N}{\cos \phi}$	$N_2 = \frac{N_1}{\sin \phi}$

panying tables, were originally contributed to MACHINERY by Herman Isler.

While the tables are self-contained, the following remarks will aid in avoiding misunderstanding of the use of the formulas in the table on page 20, for shaft angles more than 90 degrees. If the formula

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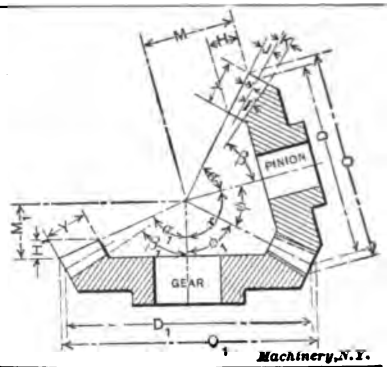
SHAFT ANGLE MORE THAN 90 DEGREES.

P = diametral pitch
C = circular pitch
N = number of teeth in pinion
N_1 = number of teeth in gear
x = shaft angle
N_2 = number of teeth for which to select cutter

$$J = \frac{1}{P} = 0.3183\ C$$

$$K = \frac{1.157}{P} = 0.3683\ C$$



Pinion	Gear
$N = D \times P = \frac{D \times \pi}{C}$	$N_1 = D_1 \times P = \frac{D_1 \times \pi}{C}$
$D = \frac{N}{P} = 0.3183\ C\ N$	$D_1 = \frac{N_1}{P} = 0.3183\ C\ N_1$
$O = D + \frac{2 \cos \phi}{P} = D + 0.6366\ C \times \cos \phi$	$O_1 = D_1 + \frac{2 \cos \phi_1}{P} = D_1 + 0.6366\ C \times \cos \phi_1$
$\tan \phi = \frac{\cos (x - 90^\circ)}{\frac{N_1}{N} - \sin (x - 90^\circ)}$	$\phi_1 = x - \phi$

$$\tan s = \frac{2 \sin \phi}{N}, \quad \tan f = \frac{2.314 \sin \phi}{N}$$

$\alpha = \phi + s$	$\alpha_1 = \phi_1 + s$
$\beta = \phi - f$	$\beta_1 = \phi_1 - f$
$M = \frac{O}{2} \times \cot \alpha$	$M_1 = \frac{O_1}{2} \times \cot \alpha_1$
$H = Y \times \cos \alpha$	$H_1 = Y \times \cos \alpha_1$
$N_2 = \frac{N}{\cos \phi}$	$N_2 = \frac{N_1}{\cos \phi_1}$

for ϕ_1 in any case gives a value greater than 90 degrees, the gear is an internal gear. This form of gearing can be cast, but cannot ordinarily be cut on any commercial machine. In order to avoid the use of an internal gear, the problem may be solved by using, in place of angle x, an angle equal to 180 degrees $- x$, employing then the formula for tan ϕ given in the table for shaft angles less than 90 degrees, on

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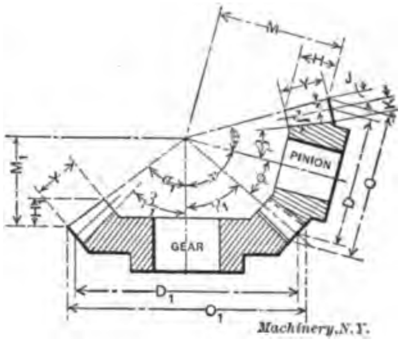
SHAFT ANGLE LESS THAN 90 DEGREES.

P = diametral pitch
C = circular pitch
N = number of teeth in pinion
N_1 = number of teeth in gear
x = shaft angle
N_2 = number of teeth for which to select cutter

$$J = \frac{1}{P} = 0.3183\,C$$

$$K = \frac{1.157}{P} = 0.3683\,C$$



Pinion	Gear
$N = D \times P = \dfrac{D \times \pi}{C}$	$N_1 = D_1 \times P = \dfrac{D_1 \times \pi}{C}$
$D = \dfrac{N}{P} = 0.3183\,C\,N$	$D_1 = \dfrac{N_1}{P} = 0.3183\,C\,N_1$
$O = D + \dfrac{2\cos\phi}{P} = D + 0.6366\,C \times \cos\phi$	$O_1 = D_1 + \dfrac{2\cos\phi_1}{P} = D_1 + 0.6366\,C \times \cos\phi_1$
$\tan\phi = \dfrac{\sin x}{\dfrac{N_1}{N} + \cos x}$	$\phi_1 = x - \phi$

$$\tan s = \frac{2\sin\phi}{N}, \quad \tan f = \frac{2.314\sin\phi}{N}$$

Pinion	Gear
$a = \phi + s$	$a_1 = \phi_1 + s$
$\beta = \phi - f$	$\beta_1 = \phi_1 - f$
$M = \dfrac{O}{2} \times \cot a$	$M_1 = \dfrac{O_1}{2} \times \cot a_1$
$H = Y \times \cos a$	$H_1 = Y \times \cos a_1$
$N_2 = \dfrac{N}{\cos\phi}$	$N_2 = \dfrac{N_1}{\cos\phi_1}$

page 21. This places the gear on the other side of the apex of the pitch cone, and thus requires the shafts to be extended, but avoids the internal bevel gear. This manner of solving the problem is advisable, as the internal gear would be almost impossible of manufacture.

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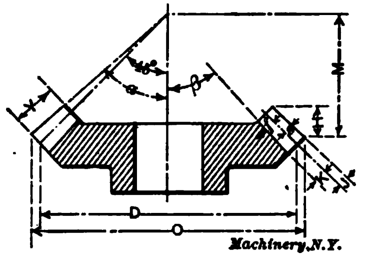
MITER GEARS.*



$$N = D \times P = \frac{D \times \pi}{C}$$

$$D = \frac{N}{P} = \frac{C \times N}{\pi} = 0.3183\ C\ N$$

$$O = D + \frac{1.4142}{P} = D + 0.45\,C$$

$$\tan s = \frac{1.4142}{N}, \quad \tan f = \frac{1.6362}{N}$$

$$\alpha = 45° + s$$

$$\beta = 45° - f$$

$$M = \frac{O}{2} - \frac{1.4142}{P} = \frac{O}{2} - 0.45\ C$$

$$H = Y \times \cos \alpha$$

$$N_2 = \frac{N}{0.7071} = N \times 1.4142$$

*See table on page 21 for meaning of letters in formulas.

Spiral Gearing.

The tooth angle in spiral gearing is the angle made by the helix or spiral of the teeth with the axis of the gear.

The equivalent diameter of a helical gear is found by dividing the number of teeth in the gear by the diametral pitch of the cutter with it is cut.

N_a = No. of teeth in gear *a*.
N_b = No. of teeth in gear *b*.
R = Velocity ratio = $N_a \div N_b$.
P'' = Normal diametral pitch or pitch of cutter.
E = Equivalent diameter (explained above).
D = Pitch diameter.
C = Center distance.
B = Blank or outside diameter.
T = No. of teeth for which cutter is selected.
L = Lead of spiral.
γ = Angle of axes.
α = Angle of tooth with axis.
t = Thickness of tooth on pitch line.
S = Addendum.
$D'' + f$ = Whole depth of tooth.

$$\gamma = \alpha_a + \alpha_b.$$

$$E = \frac{N}{P''}$$

$$E_b + (E_a \times \tan \alpha_a) = 2\,C \times \sin \alpha_a$$

$$D = \frac{E}{\cos \alpha} = E \times \sec \alpha.$$

$$B = D + \frac{2}{P''}$$

$$L = \cot \alpha \times D \times \pi.$$

$$T = \frac{N}{(\cos \alpha)^3}$$

$$t = \frac{1.5708}{P''}.$$

$$S = \frac{1}{P''}.$$

$$D'' + f = \frac{2.1571}{P''}$$

Where subscript letters $_a$ and $_b$ are used, reference is made to two gears *a* and *b*, as for instance, "N_a" and "N_b," where the letter N refers to the number of teeth in gears *a* and *b*, respectively, of a pair of gears *a* and *b*.

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Cutters for Milling Teeth of Spiral Gears.

The cutters used for milling spiral or helical gears are standard spur gear cutters, the number of a cutter and its pitch for a given case being defined by the angle (with the axis) and the normal pitch. The diagram below gives the numbers of the cutters only, the pitch having been previously determined.

The selection of the cutter is fixed by the formula given in the lower right-hand corner of the diagram. The delimiting curves thereon were plotted by the formula, the area between the curves being the

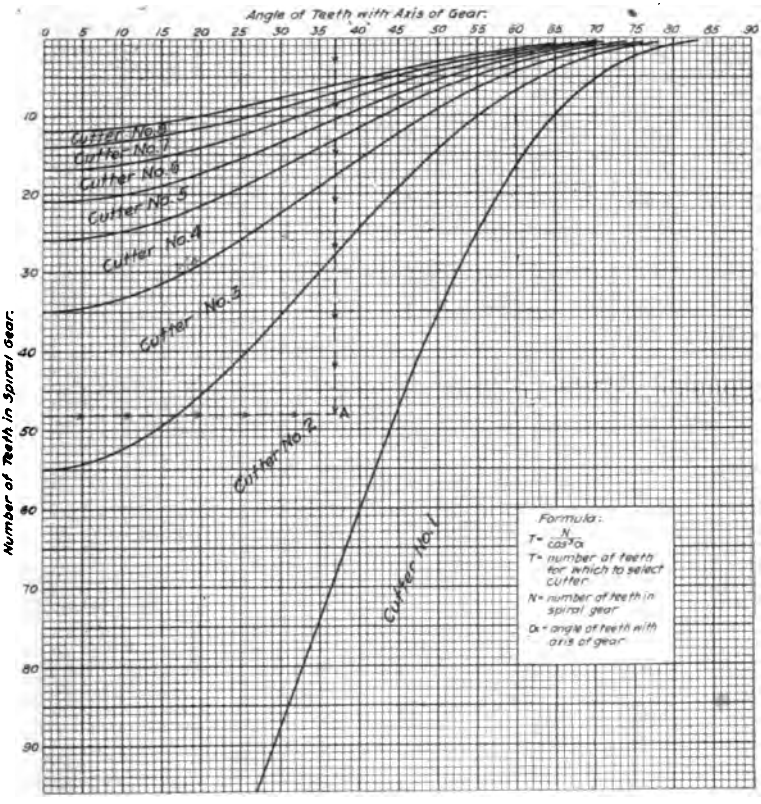

Diagram for Selecting Cutter for Milling Spiral Gear Teeth.

field of intersection of the combinations of angles and numbers of teeth covered by each designated cutter number.

For example, suppose the angle of the teeth of a gear is 37 degrees with its axis, and the number of teeth is 48. The point *A*, at which the horizontal line (representing the tooth number), and the vertical line (representing the angle) intersect, falls within the area marked "Cutter No. 2." Therefore, a No. 2 cutter is required to cut a 48-tooth spiral gear having the teeth at an angle of 37 degrees with its axis.

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WORMS AND WORM GEARING.

C.P. Circular Pitch	N Threads per Inch	D. P. Diametral Pitch	H Tooth above Pitch Line	D Working Depth of Tooth	C Clearance	S Depth of Space below Pitch Line	W. D. Whole Depth of Tooth	T Thickness of Tooth on Pitch Line	W Width of Thread Tool at End	B Width of Thread at Top
C. P. Inches	$N = \frac{1}{C.P.}$	$D.P. = \frac{\pi}{C.P.}$	$H = \frac{1}{D.P.}$	$D = 2 \times \frac{1}{D.P.}$	$C = \frac{1}{8} \times \frac{1}{D.P.}$	$S = H + C$	$W.D. = D + C$	$T = \frac{C.P.}{2}$	$W = 0.3148 \times C.P.$	$B = 0.3354 \times C.P.$
2	1/2	1.5708	0.6366	1.2732	0.0795	0.7161	1.3527	1.0000	0.6296	0.6708
1 3/4	4/7	1.7952	0.5570	1.1141	0.0696	0.6266	1.1837	0.8750	0.5509	0.5869
1 1/2	2/3	2.0944	0.4775	0.9549	0.0596	0.5371	1.0145	0.7500	0.4722	0.5031
1 1/4	4/5	2.5133	0.3979	0.7958	0.0497	0.4476	0.8455	0.6250	0.3935	0.4192
1	1	3.1416	0.3183	0.6366	0.0397	0.3580	0.6763	0.5000	0.3148	0.3354
3/4	1 1/3	4.1888	0.2387	0.4775	0.0298	0.2685	0.5073	0.3750	0.2361	0.2515
2/3	1 1/2	4.7124	0.2122	0.4244	0.0265	0.2387	0.4509	0.3333	0.2098	0.2236
1/2	2	6.2832	0.1592	0.3183	0.0199	0.1791	0.3382	0.2500	0.1574	0.1677
2/5	2 1/2	7.8540	0.1273	0.2546	0.0159	0.1432	0.2705	0.2000	0.1259	0.1341
1/3	3	9.4248	0.1061	0.2122	0.0132	0.1193	0.2254	0.1666	0.1049	0.1118
2/7	3 1/2	10.9956	0.0909	0.1819	0.0113	0.1022	0.1932	0.1429	0.0899	0.0958
1/4	4	12.5664	0.0796	0.1591	0.0099	0.0895	0.1690	0.1250	0.0787	0.0838
2/9	4 1/2	14.1372	0.0707	0.1415	0.0088	0.0795	0.1503	0.1111	0.0699	0.0745
1/5	5	15.7080	0.0637	0.1273	0.0079	0.0716	0.1352	0.1000	0.0629	0.0670
1/6	6	18.8496	0.0531	0.1061	0.0066	0.0597	0.1127	0.0833	0.0524	0.0559
1/7	7	21.9911	0.0455	0.0910	0.0056	0.0511	0.0966	0.0714	0.0449	0.0479
1/8	8	25.1327	0.0398	0.0796	0.0049	0.0447	0.0845	0.0625	0.0393	0.0419
1/9	9	28.2743	0.0354	0.0707	0.0044	0.0398	0.0752	0.0555	0.0349	0.0372
1/10	10	31.4159	0.0318	0.0637	0.0039	0.0357	0.0676	0.0500	0.0314	0.0335
1/12	12	37.6992	0.0265	0.0530	0.0033	0.0298	0.0563	0.0416	0.0262	0.0279
1/14	14	43.9824	0.0227	0.0454	0.0028	0.0255	0.0482	0.0357	0.0224	0.0239
1/16	16	50.2655	0.0199	0.0398	0.0524	0.0223	0.0422	0.0312	0.0196	0.0209
1/18	18	56.5488	0.0176	0.0352	0.0022	0.0198	0.0374	0.0277	0.0174	0.0186

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Formulas for the Design of Worm Gearing.

N = number of teeth in worm-wheel.
n = number of teeth or threads in worm.
P' = circular pitch of wheel and linear pitch of worm.
l = lead of worm.
g = whole depth of worm tooth.
t' = width of the thread tool at the end.
s = addendum or height of worm tooth above pitch line.
o = outside diameter of the worm.
d = pitch diameter of the worm.
b = bottom or root diameter of the worm.
β = helix angle of worm and gashing angle of wheel.
δ = face-angle of worm-wheel.
D = pitch diameter of the worm-wheel.
O = throat diameter of the worm-wheel.
O' = diameter of the worm-wheel to sharp corners.
U = radius of curvature of the worm-wheel throat.
R = velocity ratio.
C = distance between centers.
x = threaded length of worm.

$l = n \times P'$.	Cotangent $\beta = 3.1416 \div l$.
$P' = l \div n$.	$D = N P' \div 3.1416$.
$g = 0.6866\, P'$.	$O = D + 2s$.
$t' = 0.31\, P'$.	$U = \frac{1}{2} o - 2s$.
$s = 0.3183\, P'$.	$O' = 2\ (U - U \cos \delta/2) + O$.
$o = d + 2s$.	$R = N \div n$.
$d = o - 2s$.	$C = (D + d) \div 2$.
$b = o - 2g$.	$d = 2C - D$.

Minimum value of $x = \sqrt{O^2 - (O - 4s)^2}$

The outside diameter of the hob is $0.1\, P'$ larger than the outside diameter of the worm. The root diameter of the hob equals the outside diameter of the worm—$1.2732\, P'$.

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SCREW THREADS.

Abbreviations Used to Denote Different Screw Thread Standards.

U. S. S. = United States Standard Thread.

U. S. F. = United States Standard Form.

V = V-thread.

B. S. W. = Whitworth Thread (British Standard Whitworth).

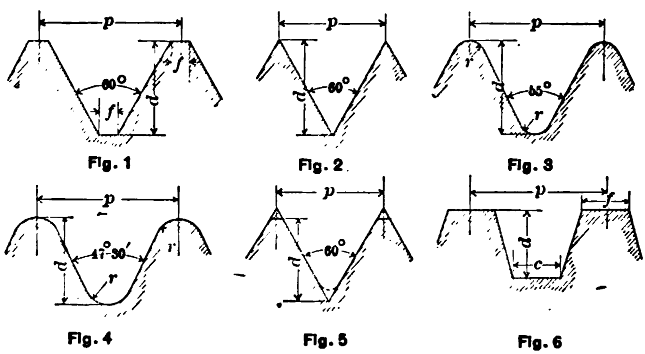
B. S. F. = British Standard Fine Screw Thread.

B. A. S. = British Association Standard Thread.

S. F. = French Standard Thread.

S. I. = International Standard Thread.

A. S. M. E. St'd = Standard Machine Screw Thread, adopted by the American Society of Mechanical Engineers.



Standard Screw Threads.

A. L. A. M. St'd = Fine Screw Thread adopted by the Association of Licensed Automobile Manufacturers.

Formulas for Standard Threads.

$$p = \text{pitch} = \frac{1}{\text{Number of threads per inch.}}$$

UNITED STATES STANDARD THREAD (Fig. 1).

$$d = \tfrac{3}{4} \times p \times \cos 30 \text{ deg.} = 0.64952\, p.$$

$$f = \frac{p}{8}$$

SHARP V-THREAD (Fig. 2.)

$$d = p \times \cos 30 \text{ deg.} = 0.86603\, p$$

WHITWORTH THREAD (Fig. 3.)

$$d = \frac{2}{3} \times \frac{p}{2} \times \cot 27 \text{ deg. } 30 \text{ min.} = 0.64033\, p$$

$$r = 0.1373\, p.$$

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BRITISH ASSOCIATION STANDARD THREAD (Fig. 4.)

$$d = 0.6\,p. \qquad r = \frac{2\,p}{11}.$$

BRIGGS MODIFIED PIPE THREAD AS MADE IN THE UNITED STATES. (Fig. 5.)

$$d = 0.833\,p.$$

$$\text{flat at top} = \frac{p}{26}$$

At the root of the thread, the groove has no flat, but is sharp, the same as a regular V-thread. (The original Briggs pipe thread, as still made in Great Britain, is slightly rounded off at the top and provided with a fillet at the bottom of the same radius, so that the depth of the thread, instead of being equaal to 0.833 p, as above, is only 0.8 p.) .

ACME THREAD. (Fig. 6.)

For Screws.

$$d = \frac{p}{2} + 0.010\text{ inch},$$

$$f = 0.3707\,p,$$

$$c = 0.3707\,p - 0.0052\text{ inch}.$$

For Taps.

$$d = \frac{p}{2} + 0.020\text{ inch},$$

$$f = 0.3707\,p - 0.0052\text{ inch},$$

$$c = 0.3707\,p - 0.0052\text{ inch}.$$

The root diameters of screws and taps are the same. The outside diameter of the tap is 0.020 inch greater than the outside diameter of the screw.

The angle between the sides of the Acme thread is 29 degrees.

French and International Standard Threads.

The form of the thread, and the formulas for the thread dimensions are the same as for the United States Standard thread except that the shape of the thread at the bottom may be flat or rounded, as preferred, as long as clearance at this point is provided. The originators of the thread recommended a rounded profile at the root, but in the United States this thread is made exclusively with a flat, the same as the U. S. Standard thread.

UNITED STATES STANDARD THREAD.

Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch
1/16	64	13/16	10	1 3/4	5	3 3/8	3 1/4
3/32	50	7/8	9	1 13/16	5	3 1/2	3 1/4
1/8	40	15/16	9	1 7/8	5	3 5/8	3 1/4
5/32	36	1	8	1 15/16	5	3 3/4	3
3/16	32	1 1/16	7	2	4 1/2	3 7/8	3
7/32	28	1 1/8	7	2 1/8	4 1/2	4	3
1/4	20	1 3/16	7	2 1/4	4 1/2	4 1/4	2 7/8
5/16	18	1 1/4	7	2 3/8	4	4 1/2	2 3/4
3/8	16	1 5/16	6	2 1/2	4	4 3/4	2 5/8
7/16	14	1 3/8	6	2 5/8	4	5	2 1/2
1/2	13	1 7/16	6	2 3/4	4	5 1/4	2 1/2
9/16	12	1 1/2	6	2 7/8	3 1/2	5 1/2	2 3/8
5/8	11	1 9/16	5 1/2	3	3 1/2	5 3/4	2 3/8
11/16	11	1 5/8	5 1/2	3 1/8	3 1/2	6	2 1/4
3/4	10	1 11/16	5 1/2	3 1/4	3 1/2		

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STANDARD SHARP V-THREAD

Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch
1/16	72	13/16	10	1 3/4	5	3 3/8	3 1/4
3/32	56	7/8	9	1 13/16	5	3 1/2	3 1/4
1/8	40	15/16	9	1 7/8	4 1/2	3 5/8	3 1/4
5/32	32	1	8	1 15/16	4 1/2	3 3/4	3
3/16	24	1 1/16	8	2	4 1/2	3 7/8	3
7/32	24	1 1/8	7	2 1/8	4 1/2	4	3
1/4	20	1 3/16	7	2 1/4	4 1/2	4 1/4	2 7/8
5/16	18	1 1/4	7	2 3/8	4 1/2	4 1/2	2 3/4
3/8	16	1 5/16	7	2 1/2	4	4 3/4	2 5/8
7/16	14	1 3/8	6	2 5/8	4	5	2 1/2
1/2	12	1 7/16	6	2 3/4	4	5 1/4	2 1/2
9/16	12	1 1/2	6	2 7/8	4	5 1/2	2 3/8
5/8	11	1 9/16	6	3	3 1/2	5 3/4	2 3/8
11/16	11	1 5/8	5	3 1/8	3 1/2	6	2 1/4
3/4	10	1 11/16	5	3 1/4	3 1/4		

WHITWORTH STANDARD THREAD.

Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch
1/16	60	13/16	10	1 3/4	5	3 3/8	3 1/4
3/32	48	7/8	9	1 13/16	5	3 1/2	3 1/4
1/8	40	15/16	9	1 7/8	4 1/2	3 5/8	3 1/4
5/32	32	1	8	1 15/16	4 1/2	3 3/4	3
3/16	24	1 1/16	8	2	4 1/2	3 7/8	3
7/32	24	1 1/8	7	2 1/8	4 1/2	4	3
1/4	20	1 3/16	7	2 1/4	4	4 1/4	2 7/8
5/16	18	1 1/4	7	2 3/8	4	4 1/2	2 7/8
3/8	16	1 5/16	7	2 1/2	4	4 3/4	2 3/4
7/16	14	1 3/8	6	2 5/8	4	5	2 3/4
1/2	12	1 7/16	6	2 3/4	3 1/2	5 1/4	2 5/8
9/16	12	1 1/2	6	2 7/8	3 1/2	5 1/2	2 5/8
5/8	11	1 9/16	6	3	3 1/2	5 3/4	2 1/2
11/16	11	1 5/8	5	3 1/8	3 1/2	6	2 1/2
3/4	10	1 11/16	5	3 1/4	3 1/4		

BRITISH STANDARD FINE SCREW THREAD.

Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch
1/4	25	1 1/8	9	2	7	3 3/4	4 1/2
5/16	22	1 3/16	9	2 1/8	7	3 7/8	4 1/2
3/8	20	1 1/4	9	2 1/4	6	4	4 1/2
7/16	18	1 5/16	9	2 3/8	6	4 1/4	4
1/2	16	1 3/8	8	2 1/2	6	4 1/2	4
9/16	16	1 7/16	8	2 5/8	6	4 3/4	4
5/8	14	1 1/2	8	2 3/4	6	5	4
11/16	14	1 9/16	8	2 7/8	6	5 1/4	3 1/2
3/4	12	1 5/8	8	3	5	5 1/2	3 1/2
13/16	12	1 11/16	8	3 1/8	5	5 3/4	3 1/2
7/8	11	1 3/4	7	3 1/4	5	6	3 1/2
15/16	11	1 13/16	7	3 3/8	5		
1	10	1 7/8	7	3 1/2	4 1/2		
1 1/16	10	1 15/16	7	3 5/8	4 1/2		

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BRITISH ASSOCIATION STANDARD THREAD.

British Association Number	Diameter		Pitch		British Association Number	Diameter		Pitch	
	Millimeters	Inches	Millimeters	Inches		Millimeters	Inches	Millimeters	Inches
0	6.0	0.2362	1.0	0.0394	13	1.2	0.0472	0.25	0.0098
1	5.3	0.2087	0.90	0.0354	14	1.0	0.0394	0.23	0.0091
2	4.7	0.1850	0.81	0.0319	15	0.90	0.0354	0.21	0.0083
3	4.1	0.1614	0.73	0.0287	16	0.79	0.0311	0.19	0.0075
4	3.6	0.1417	0.66	0.0260	17	0.70	0.0276	0.17	0.0067
5	3.2	0.1260	0.59	0.0232	18	0.62	0.0244	0.15	0.0059
6	2.8	0.1102	0.53	0.0209	19	0.54	0.0213	0.14	0.0055
7	2.5	0.0984	0.48	0.0189	20	0.48	0.0189	0.12	0.0047
8	2.2	0.0866	0.43	0.0169	21	0.42	0.0165	0.11	0.0043
9	1.9	0.0748	0.39	0.0154	22	0.37	0.0146	0.098	0.0039
10	1.7	0.0669	0.35	0.0138	23	0.33	0.0130	0.089	0.0035
11	1.5	0.0591	0.31	0.0122	24	0.29	0.0114	0.080	0.0031
12	1.3	0.0511	0.28	0.0110	25	0.25	0.0098	0.072	0.0028

ACME STANDARD THREAD.

Threads per Inch	Depth of Thread	Width of Flat at Top of Screw Thread	Width of Flat at Top of Tap Thread and at Bottom of Screw and Tap Thread	Threads per Inch	Depth of Thread	Width of Flat at Top of Screw Thread	Width of Flat at Top of Tap Thread and at Bottom of Screw and Tap Thread
1	0.5100	0.3707	0.3655	5	0.1100	0.0741	0.0689
1⅓	0.3850	0.2780	0.2728	5½	0.1009	0.0674	0.0622
1½	0.3433	0.2471	0.2419	6	0.0933	0.0618	0.0566
2	0.2600	0.1853	0.1801	7	0.0814	0.0530	0.0478
2½	0.2100	0.1483	0.1431	8	0.0725	0.0463	0.0411
3	0.1767	0.1236	0.1184	9	0.0656	0.0412	0.0360
3½	0.1529	0.1059	0.1007	10	0.0600	0.0371	0.0319
4	0.1350	0.0927	0.0875	12	0.0517	0.0309	0.0257
4½	0.1211	0.0824	0.0772				

WHITWORTH STANDARD THREAD FOR GAS AND WATER PIPING, COMMONLY KNOWN AS THE STANDARD GAS THREAD.

Nominal Size of Tube	Actual Outside Size of Tube	No. of Threads per Inch	Nominal Size of Tube	Actual Outside Size of Tube	No. of Threads per Inch	Nominal Size of Tube	Actual Outside Size of Tube	No. of Threads per Inch
⅛	0.385	28	1⅜	1.745	11	2⅝	3.124	11
¼	0.520	19	1½	1.882	11	2¾	3.247	11
⅜	0.665	19	1⅝	2.021	11	2⅞	3.367	11
½	0.822	14	1¾	2.160	11	3	3.485	11
⅝	0.902	14	1⅞	2.245	11	3¼	3.698	11
¾	1.034	14	2	2.347	11	3½	3.912	11
⅞	1.189	14	2⅛	2.467	11	3¾	4.125	11
1	1.302	11	2¼	2.587	11	4	4.339	11
1⅛	1.492	11	2⅜	2.794	11			
1¼	1.650	11	2½	3.001	11			

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BRIGGS STANDARD PIPE THREAD.

Nominal Size of Tube	Actual Outside Size of Tube	No. of Threads per Inch	Nominal Size of Tube	Actual Outside Size of Tube	No. of Threads per Inch	Nominal Size of Tube	Actual Outside Size of Tube	No. of Threads per Inch
1/8	0.405	27	1½	1.900	11½	5	5.563	8
1/4	0.540	18	2	2.375	11½	6	6.625	8
3/8	0.675	18	2½	2.875	8	7	7.625	8
1/2	0.840	14	3	3.500	8	8	8.625	8
3/4	1.050	14	3½	4.000	8	9	9.688	8
1	1.315	11½	4	4.500	8	10	10.750	8
1¼	1.660	11½	4½	5.000	8			

MACHINE SCREW THREADS, AMERICAN SOCIETY OF MECHANICAL ENGINEERS STANDARD.

Number	Diameter	Threads per Inch	Number	Diameter	Threads per Inch	Number	Diameter	Threads per Inch
0	0.060	80	7	0.151	36	18	0.294	20
1	0.073	72	8	0.164	36	20	0.320	20
2	0.086	64	9	0.177	32	22	0.346	18
3	0.099	56	10	0.190	30	24	0.372	16
4	0.112	48	12	0.216	28	26	0.398	16
5	0.125	44	14	0.242	24	28	0.424	14
6	0.138	40	16	0.268	22	30	0.450	14

SPECIAL MACHINE SCREW THREADS, ADOPTED BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

Number	Diameter	Threads per Inch	Number	Diameter	Threads per Inch	Number	Diameter	Threads per Inch
1	0.073	64	7	0.151	32	14	0.242	20
2	0.086	56	7	0.151	30	16	0.268	20
3	0.099	48	8	0.164	32	18	0.294	18
4	0.112	40	8	0.164	30	20	0.320	18
4	0.112	36	9	0.177	30	22	0.346	16
5	0.125	40	9	0.177	24	24	0.372	18
5	0.125	36	10	0.190	32	26	0.398	14
6	0.138	36	10	0.190	24	28	0.424	16
6	0.138	32	12	0.216	24	30	0.450	16

MACHINE SCREW THREADS, OLD STANDARD.

Number	Diameter	Threads per Inch	Number	Diameter	Threads per Inch	Number	Diameter	Threads per Inch
1	0.071	64	8	0.166	32	16	0.272	18
1½	0.081	56	9	0.180	30	18	0.298	18
2	0.089	56	10	0.194	24	20	0.325	16
3	0.101	48	11	0.206	24	22	0.350	16
4	0.113	36	12	0.221	24	24	0.378	16
5	0.125	36	13	0.234	22	26	0.404	16
6	0.141	32	14	0.246	20	28	0.430	14
7	0.154	32	15	0.261	20	30	0.456	14

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INTERNATIONAL SYSTEM STANDARD THREAD.

Diameter		Pitch		Diameter		Pitch	
Milli-meters	Inches	Milli-meters	Inches	Milli-meters	Inches	Milli-meters	Inches
6	0.2362	1.0	0.0394	33	1.2992	3.5	0.1378
7	0.2756	1.0	0.0394	36	1.4173	4.0	0.1575
8	0.3150	1.25	0.0492	39	1.5354	4.0	0.1575
9	0.3543	1.25	0.0492	42	1.6535	4.5	0.1772
10	0.3937	1.5	0.0590	45	1.7716	4.5	0.1772
11	0.4331	1.5	0.0590	48	1.8898	5.0	0.1969
12	0.4724	1.75	0.0689	52	2.0472	5.0	0.1969
14	0.5512	2.0	0.0787	56	2.2047	5.5	0.2165
16	0.6299	2.0	0.0787	60	2.3622	5.5	0.2165
18	0.7087	2.5	0.0984	64	2.5197	6.0	0.2362
20	0.7874	2.5	0.0984	68	2.6772	6.0	0.2362
22	0.8661	2.5	0.0984	72	2.8346	6.5	0.2559
24	0.9449	3.0	0.1181	76	2.9921	6.5	0.2559
27	1.0630	3.0	0.1181	80	3.1497	7.0	0.2756
30	1.1811	3.5	0.1378				

FRENCH SYSTEM STANDARD THREAD.

Diameter		Pitch		Diameter		Pitch	
Milli-meters	Inches	Milli-meters	Inches	Milli-meters	Inches	Milli-meters	Inches
3	0.1181	0.5	0.0197	24	0.9449	3.0	0.1181
4	0.1575	0.75	0.0295	26	1.0236	3.0	0.1181
5	0.1969	0.75	0.0295	28	1.1024	3.0	0.1181
6	0.2362	1.0	0.0394	30	1.1811	3.5	0.1378
7	0.2756	1.0	0.0394	32	1.2598	3.5	0.1378
8	0.3150	1.0	0.0394	34	1.3386	3.5	0.1378
9	0.3543	1.0	0.0394	36	1.4173	4.0	0.1575
10	0.3937	1.5	0.0590	38	1.4961	4.0	0.1575
12	0.4724	1.5	0.0590	40	1.5748	4.0	0.1575
14	0.5512	2.0	0.0787	42	1.6535	4.5	0.1772
16	0.6299	2.0	0.0787	44	1.7323	4.5	0.1772
18	0.7087	2.5	0.0984	46	1.8110	4.5	0.1772
20	0.7874	2.5	0.0984	48	1.8898	5.0	0.1969
22	0.8661	2.5	0.0984	50	1.9685	5.0	0.1969

WOOD SCREW THREAD.

No. of Screw	Diam-eter	Threads per Inch	No. of Screw	Diam-eter	Threads per Inch	No. of Screw	Diam-eter	Threads per Inch
0	0.058	32	11	0.203	12	22	0.347	7
1	0.071	28	12	0.216	11	23	0.361	7
2	0.084	26	13	0.229	11	24	0.374	7
3	0.097	24	14	0.242	10	25	0.387	7
4	0.110	22	15	0.255	10	26	0.400	6
5	0.124	20	16	0.268	9	27	0.413	6
6	0.137	18	17	0.282	9	28	0.426	6
7	0.150	16	18	0.295	8	29	0.439	6
8	0.163	15	19	0.308	8	30	0.453	6
9	0.176	14	20	0.321	8			
10	0.189	13	21	0.334	8			

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TOOLS FOR SQUARE THREAD.

No. of Threads per Inch	Width of Point of Tool: For Taps	For Screws	For Inside Thread Tools for Nuts	No. of Threads per Inch	Width of Point of Tool: For Taps	For Screws	For Inside Thread Tools for Nuts
1	0.4965	0.5000	0.5035	8	0.0615	0.0625	0.0635
1⅓	0.3715	0.3750	0.3785	9	0.0545	0.0555	0.0565
1½	0.3333	0.3333	0.3368	10	0.0490	0.0500	0.0510
1¾	0.2827	0.2857	0.2887	11	0.0444	0.0454	0.0464
2	0.2475	0.2500	0.2525	12	0.0107	0.0417	0.0427
2½	0.1975	0.2000	0.2025	13	0.0375	0.0385	0.0395
3	0.1641	0.1666	0.1691	14	0.0352	0.0357	0.0362
3½	0.1408	0.1428	0.1448	15	0.0328	0.0333	0.0338
4	0.1235	0.1250	0.1265	16	0.0307	0.0312	0.0317
4½	0.1096	0.1111	0.1126	18	0.0272	0.0277	0.0282
5	0.0985	0.1000	0.1015	20	0.0245	0.0250	0.0255
5½	0.0894	0.0909	0.0924	22	0.0222	0.0227	0.0232
6	0.0818	0.0833	0.0848	24	0.0203	0.0208	0.0213
7	0.0699	0.0714	0.0729				

STANDARD WORM THREAD.

Threads per Inch	Depth of Thread	Width of Flat at Top of Thread	Width of Flat at Bottom of Thread	Width of Thread at Pitch Line	Height of Thread Above Pitch Line
1	0.6866	0.3350	0.3100	0.5000	0.3183
1¼	0.5492	0.2680	0.2480	0.4000	0.2546
1½	0.4577	0.2233	0.2066	0.3333	0.2122
2	0.3433	0.1675	0.1550	0.2500	0.1592
2½	0.2746	0.1340	0.1240	0.2000	0.1273
3	0.2289	0.1117	0.1033	0.1666	0.1061
3½	0.1962	0.0957	0.0886	0.1429	0.0909
4	0.1716	0.0838	0.0775	0.1250	0.0796
4½	0.1526	0.0744	0.0689	0.1111	0.0707
5	0.1373	0.0670	0.0620	0.1000	0.0637
6	0.1144	0.0558	0.0517	0.0833	0.0531
7	0.0981	0.0479	0.0443	0.0714	0.0455
8	0.0858	0.0419	0.0388	0.0625	0.0398
9	0.0763	0.0372	0.0344	0.0555	0.0354
10	0.0687	0.0335	0.0310	0.0500	0.0318
12	0.0572	0.0279	0.0258	0.0416	0.0265
16	0.0429	0.0209	0.0194	0.0312	0.0199
20	0.0343	0.0167	0.0155	0.0250	0.0159

LAG SCREW THREAD SYSTEMS IN COMMON USE.

Diameter	Alternate Systems: Threads per Inch	Threads per Inch	Diameter	Alternate Systems: Threads per Inch	Threads per Inch	Diameter	Alternate Systems: Threads per Inch	Threads per Inch
1/4	10	10	1/2	6	6	3/4	4½	5
5/16	9½	9	9/16	5	6	7/8	4½	4
3/8	7	8	5/8	5	5	1	3	4
7/16	7	7	11/16	4½	5			

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FINE SCREW THREAD SYSTEM ADOPTED BY THE ASSOCIATION OF LICENSED AUTOMOBILE MANUFACTURERS.

Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch	Diameter	Threads per Inch
1/4	28	7/16	20	5/8	18	7/8	14
5/16	24	1/2	20	11/16	16	1	14
3/8	24	9/16	18	3/4	16		

TAP DRILLS FOR A. S. M. E. STANDARD AND SPECIAL MACHINE SCREWS.

Standard Sizes are Marked *

No. of Screw and Threads per Inch	Size of Drill	No. of Screw and Threads per Inch	Size of Drill	No. of Screw and Threads per Inch	Size of Drill	No. of Screw and Threads per Inch	Size of Drill
0–80*	56	5–36	42	9–24	30	20–20*	G
1–72*	53	6–40*	35	10–32	23	20–18	F
1–64	54	6–36	36	10–30*	24	22–18*	K
2–64*	50	6–32	38	10–24	28	22–16	I
2–56	51	7–36*	31	12–28*	17	24–18	5/16
3–56*	47	7–32	32	12–24	19	24–16*	N
3–48	48	7–30	33	14–24*	10	26–16*	P
4–48*	43	8–36*	29	14–20	14	26–14	O
4–40	45	8–32	30	16–22*	3	28–16	S
4–36	46	8–30	30	16–20	4	28–14*	R
5–44*	39	9–32*	28	18–20*	A	30–16	V
5–40	40	9–30	28	18–18	1	30–14*	U

TAP DRILLS FOR OLD STANDARD MACHINE SCREWS.

Size of Taps	No. of Threads	Size of Drills	Size of Taps	No. of Threads	Size of Drills	Size of Taps	No. of Threads	Size of Drills
2	48	51	9	28	29	16	24	2
2	56	49	9	30	28	17	16	7
2	64	49	9	32	27	17	18	4
3	40	49	10	24	28	17	20	2
3	48	48	10	28	26	18	16	3
3	56	44	10	30	24	18	18	2
4	32	48	10	32	24	18	20	A
4	36	45	11	24	24	19	16	1
4	40	44	11	28	21	19	18	B
5	30	44	11	30	19	19	20	D
5	32	43	12	20	24	20	16	C
5	36	41	12	22	20	20	18	E
5	40	40	12	24	19	20	20	H
6	30	41	13	20	19	22	16	H
6	32	37	13	24	15	22	18	J
6	36	36	14	20	16	24	14	K
6	40	33	14	22	18	24	16	L
7	28	35	14	24	9	24	18	N
7	30	34	15	18	13	26	14	N
7	32	31	15	20	10	26	16	O
8	24	34	15	24	6	28	14	Q
8	30	30	16	16	13	28	16	S
8	32	30	16	18	10	30	14	T
9	24	30	16	20	6	30	16	V

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TAP DRILL TABLE FOR STANDARD AND FINER PITCHES.

	Number of Threads per Inch.																				
	3	3¼	3½	4	4½	5	5½	6	7	8	9	10	11	12	13	14	16	18	20	22	24
1/4	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	3/16	13/64	13/64
5/16	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	1/4	1/4	17/64	17/64
3/8	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	19/64	5/16	5/16	21/64	21/64
7/16	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	23/64	23/64	3/8	3/8	25/64	25/64
1/2	..	..	..	..	..	..	..	..	..	..	..	..	..	..	13/32	27/64	27/64	7/16	7/16	29/64	29/64
9/16	..	..	..	..	..	..	..	..	..	..	..	..	..	15/32	15/32	31/64	31/64	1/2	1/2	33/64	33/64
5/8	..	..	..	..	..	..	..	..	..	..	..	..	33/64	17/32	17/32	35/64	35/64	9/16	9/16	37/64	37/64
3/4	..	..	..	..	..	..	..	..	..	..	..	5/8	41/64	21/32	21/32	43/64	43/64	11/16	11/16	45/64	45/64
7/8	..	..	..	..	..	..	..	..	..	..	47/64	3/4	49/64	25/32	25/32	51/64	51/64	13/16	13/16	53/64	53/64
1	..	..	..	..	..	..	..	..	..	27/32	55/64	7/8	57/64	29/32	29/32	59/64	59/64	15/16	15/16	61/64	61/64
1 1/8	..	..	..	..	..	..	..	..	61/64	31/32	63/64	1	1 1/64	1 1/32	1 1/32	1 1/32	1 3/64	1 1/16	1 1/16		
1 1/4	..	..	..	..	..	..	..	..	1 5/64	1 3/32	1 7/64	1 1/8	1 9/64	1 5/32	1 5/32	1 11/64	1 11/64	1 3/16	1 3/16		
1 3/8	..	..	..	..	..	..	..	1 11/64	1 13/64	1 7/32	1 15/64	1 1/4	1 17/64	1 9/32	1 9/32	1 9/32	1 19/64	1 5/16	1 5/16		
1 1/2	..	..	..	..	..	..	..	1 19/64	1 21/64	1 11/32	1 23/64	1 3/8	1 25/64	1 13/32	1 13/32	1 27/64	1 27/64	1 7/16	1 7/16		
1 5/8	..	..	..	..	..	..	1 25/64	1 13/32	1 29/64	1 15/32	1 31/64	1 1/2	1 33/64	1 17/32	1 17/32	1 35/64	1 35/64	1 9/16			
1 3/4	..	..	..	..	..	1 1/2	1 33/64	1 35/64	1 37/64	1 19/32	1 39/64	1 5/8	1 41/64	1 21/32	1 21/32	1 43/64	1 43/64	1 11/16			
1 7/8	..	..	..	..	..	1 5/8	1 41/64	1 43/64	1 45/64	1 23/32	1 47/64	1 3/4	1 49/64	1 25/32	1 25/32	1 51/64	1 51/64	1 13/16			
2	..	..	..	..	1 23/32	1 3/4	1 49/64	1 51/64	1 53/64	1 27/32	1 55/64	1 7/8	1 57/64	1 29/32	1 29/32	1 59/64	1 59/64	1 15/16			
2 1/4	..	..	..	..	1 31/32	2	2 1/32	2 1/16	2 3/32	2 3/32	2 1/8	2 1/8	2 9/64	2 5/32	2 5/32	2 11/64	2 11/64				
2 1/2	..	..	..	2 3/16	2 7/32	2 1/4	2 9/32	2 5/16	2 11/32	2 11/32	2 3/8	2 3/8	2 25/64	2 13/32	2 13/32	2 27/64	2 27/64				
2 3/4	..	..	..	2 7/16	2 15/32	2 1/2	2 17/32	2 9/16	2 19/32	2 19/32	2 5/8	2 5/8	2 41/64	2 21/32	2 21/32	2 43/64	2 43/64				
3	..	..	2 41/64	2 11/16	2 23/32	2 3/4	2 25/32	2 13/16	2 27/32	2 27/32	2 7/8	2 7/8	2 57/64	2 29/32	2 29/32	2 59/64	2 59/64				
3 1/4	..	..	2 57/64	2 15/16	2 31/32	3	3 1/32	3 1/16	3 3/32	3 3/32	3 1/8	3 1/8	3 9/64	3 5/32							
3 1/2	..	3 1/8	3 5/32	3 3/16	3 7/32	3 1/4	3 9/32	3 5/16	3 11/32	3 11/32	3 3/8	3 3/8	3 25/64	3 13/32							
3 3/4	3 21/64	3 3/8	3 13/32	3 7/16	3 15/32	3 1/2	3 17/32	3 9/16	3 19/32	3 19/32	3 5/8	3 5/8	3 41/64	3 21/32							
4	3 37/64	3 5/8	3 21/32	3 11/16	3 23/32	3 3/4	3 25/32	3 13/16	3 27/32	3 27/32	3 7/8	3 7/8	3 57/64	3 29/32							

The first Drill Size in each line is for U. S. Standard Tap.

TAP DRILLS FOR PIPE TAPS.

Size of Tap	Drills for Briggs Pipe Taps	Drills for Whitworth Pipe Taps	Size of Tap	Drills for Briggs Pipe Taps	Drills for Whitworth Pipe Taps	Size of Tap	Drills for Briggs Pipe Taps	Drills for Whitworth Pipe Taps
1/8	21/64	5/16	1 1/4	1 15/32	1 15/32	3 1/4	—	3 1/2
1/4	27/64	27/64	1 1/2	1 23/32	1 25/32	3 1/2	3 11/16	3 3/4
3/8	9/16	9/16	1 3/4	—	1 15/16	3 3/4	—	4
1/2	11/16	11/16	2	2 3/16	2 5/32	4	4 3/16	4 1/4
5/8	—	25/32	2 1/4	—	2 13/32	4 1/2	4 11/16	4 3/4
3/4	29/32	29/32	2 1/2	2 9/16	2 25/32	5	5 1/4	5 1/4
7/8	—	1 1/16	2 3/4	—	3 1/32	5 1/2	—	5 3/4
1	1 1/8	1 1/8	3	3 3/16	3 9/32	6	6 5/16	6 1/4

CONSTANTS FOR FINDING DIAMETER AT ROOT OF THREAD.

Threads per Inch	U. S. Standard Thread	Standard V-Thread	Whitworth Standard Thread	Threads per Inch	U. S Standard Thread	Standard V-Thread	Whitworth Standard Thread
2 1/4	0.5774	0.7698	0.5692	18	0.0722	0.0962	0.0711
2 3/8	0.5470	0.7293	0.5392	20	0.0650	0.0866	0.0640
2 1/2	0.5196	0.6928	0.5123	22	0.0590	0.0787	0.0582
2 5/8	0.4949	0.6598	0.4879	24	0.0541	0.0722	0.0534
2 3/4	0.4724	0.6298	0.4657	26	0.0500	0.0666	0.0493
2 7/8	0.4518	0.6025	0.4454	28	0.0464	0.0619	0.0457
3	0.4330	0.5774	0.4269	30	0.0433	0.0577	0.0427
3 1/4	0.3997	0.5329	0.3940	32	0.0406	0.0541	0.0400
3 1/2	0.3712	0.4949	0.3659	34	0.0382	0.0509	0.0377
4	0.3248	0.4330	0.3202	36	0.0361	0.0481	0.0356
4 1/2	0.2887	0.3849	0.2846	38	0.0342	0.0456	0.0337
5	0.2598	0.3464	0.2561	40	0.0325	0.0433	0.0320
5 1/2	0.2362	0.3149	0.2328	42	0.0309	0.0412	0.0305
6	0.2165	0.2887	0.2134	44	0.0295	0.0394	0.0291
7	0.1856	0.2474	0.1830	46	0.0282	0.0377	0.0278
8	0.1624	0.2165	0.1601	48	0.0271	0.0361	0.0267
9	0.1443	0.1925	0.1423	50	0.0260	0.0346	0.0256
10	0.1299	0.1732	0.1281	52	0.0250	0.0333	0.0246
11	0.1181	0.1575	0.1164	56	0.0232	0.0309	0.0229
12	0.1083	0.1443	0.1067	60	0.0217	0.0289	0.0213
13	0.0999	0.1332	0.0985	64	0.0203	0.0271	0.0200
14	0.0928	0.1237	0.0915	68	0.0191	0.0255	0.0188
15	0.0866	0.1155	0.0854	72	0.0180	0.0241	0.0178
16	0.0812	0.1083	0.0800	80	0.0162	0.0217	0.0160

These constants are subtracted from the outside diameter of the tap or screw; the result is the root diameter of the thread.

TAPPING SPEEDS FOR STANDARD TAPS.

Diameter of Tap	Cast Iron	Wrought Iron	Diameter of Tap	Cast Iron	Wrought Iron	Diameter of Tap	Cast Iron	Wrought Iron
3/16	340	265	5/8	117	91	1 3/8	51	41
1/4	295	230	3/4	96	76	1 1/2	46	38
5/16	240	190	7/8	84	65	1 3/4	40	33
3/8	197	152	1	72	57	2	34	28
7/16	170	122	1 1/8	63	50	2 1/4	30	26
1/2	145	114	1 1/4	57	45	2 1/2	26	23

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MISCELLANEOUS TABLES.

TABLE OF DECIMAL EQUIVALENTS OF LETTER SIZE DRILLS.

Letter	Size of Drill in Decimals	Letter	Size of Drill in Decimals	Letter	Size of Drill in Decimals	Letter	Size of Drill in Decimals
Z	0.413	S	0.348	L	0.290	E	0.250
Y	0.404	R	0.339	K	0.281	D	0.246
X	0.397	Q	0.332	J	0.277	C	0.242
W	0.386	P	0.323	I	0.272	B	0.238
V	0.377	O	0.316	H	0.266	A	0.234
U	0.368	N	0.302	G	0.261		
T	0.358	M	0.295	F	0.257		

TWIST DRILL AND STEEL WIRE GAGE.

No.	Size of Drill in Inches	No.	Size of Drill in Inches	No.	Size of Drill in Inches	No.	Size of Drill in Inches
1	0.2280	21	0.1590	41	0.0960	61	0.0390
2	0.2210	22	0.1570	42	0.0935	62	0.0380
3	0.2130	23	0.1540	43	0.0890	63	0.0370
4	0.2090	24	0.1520	44	0.0860	64	0.0360
5	0.2055	25	0.1495	45	0.0820	65	0.0350
6	0.2040	26	0.1470	46	0.0810	66	0.0330
7	0.2010	27	0.1440	47	0.0785	67	0.0320
8	0.1990	28	0.1405	48	0.0760	68	0.0310
9	0.1960	29	0.1360	49	0.0730	69	0.0292
10	0.1935	30	0.1285	50	0.0700	70	0.0280
11	0.1910	31	0.1200	51	0.0670	71	0.0260
12	0.1890	32	0.1160	52	0.0635	72	0.0250
13	0.1850	33	0.1130	53	0.0595	73	0.0240
14	0.1820	34	0.1110	54	0.0550	74	0.0225
15	0.1800	35	0.1100	55	0.0520	75	0.0210
16	0.1770	36	0.1065	56	0.0465	76	0.0200
17	0.1730	37	0.1040	57	0.0430	77	0.0180
18	0.1695	38	0.1015	58	0.0420	78	0.0160
19	0.1660	39	0.0995	59	0.0410	79	0.0145
20	0.1610	40	0.0980	60	0.0400	80	0.0135

CUTTING SPEEDS FOR SHAPER, PLANER AND LATHE TOOLS.

	Bass	Cast Iron	Machine Steel	Tool Steel Annealed
Feet Per Minute	75 to 100	25 to 35	18 to 25	15 to 25

CUTTING SPEEDS FOR MILLING CUTTERS.

	Brass	Cast Iron	Machine Steel	Tool Steel Annealed
Feet per Minute	80 to 120	40 to 60	35 to 45	25 to 35

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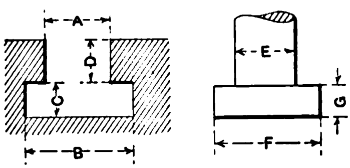
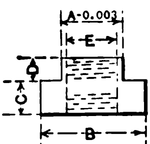
SPEED OF DRILLS

Diameter of Drill Inches	Revolutions per Minute			Diameter of Drill Inches	Revolutions per Minute		
	Wrought Iron and Steel	Cast Iron	Brass		Wrought Iron and Steel	Cast Iron	Brass
1/16	1536	1824	3360	1 1/2	64	76	140
1/8	768	912	1680	1 9/16	61	73	134
3/16	512	608	1120	1 5/8	59	70	129
1/4	384	456	840	1 11/16	57	68	125
5/16	307	365	672	1 3/4	55	65	120
3/8	256	304	560	1 13/16	53	63	116
7/16	219	261	480	1 7/8	51	61	112
1/2	192	228	420	1 15/16	49	59	108
9/16	170	203	373	2	48	57	105
5/8	154	182	336	2 1/16	46	55	102
11/16	139	166	305	2 1/8	45	54	100
3/4	128	152	280	2 3/16	43	52	96
13/16	118	140	258	2 1/4	42	51	93
7/8	109	130	239	2 5/16	41	49	91
15/16	102	122	224	2 3/8	40	48	88
1	96	114	210	2 7/16	39	47	86
1 1/16	90	107	197	2 1/2	38	45	84
1 1/8	85	101	186	2 9/16	37	44	82
1 3/16	81	96	177	2 5/8	36	43	80
1 1/4	77	91	168	2 3/4	35	41	76
1 5/16	73	87	160	2 7/8	33	40	73
1 3/8	70	83	153	3	31	38	70
1 7/16	67	79	146				

T-SLOTS AND T-BOLTS.



T-NUTS.



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Slot A	Slot B	Slot C	Slot D*	Bolt-Head E	Bolt-Head F	Bolt-Head G
1/4	1/2	5/32	3/16	3/16	7/16	1/8
5/16	5/8	5/32	3/16	1/4	9/16	1/8
3/8	11/16	7/32	1/4	5/16	5/8	3/16
7/16	13/16	7/32	9/32	3/8	3/4	3/16
1/2	15/16	9/32	5/16	7/16	7/8	1/4
5/8	1 3/16	13/32	3/8	9/16	1 1/8	11/32
3/4	1 5/16	17/32	1/2	11/16	1 1/4	15/32
7/8	1 5/8	11/16	9/16	3/4	1 1/2	9/16
1	1 7/8	13/16	5/8	7/8	1 3/4	11/16

* Minimum distance possible.
Maximum distance of *D* equals
A + 1/16 for sizes of bolt up to 5/8,
1 for 3/4 size of bolt,
1 1/16 for 7/8 size of bolt,
1 3/16 for 1 size of bolt.

A	B	C	D	E
3/16	7/16	1/8	3/32	1/8
1/4	9/16	1/8	1/8	3/16
5/16	5/8	3/16	1/8	1/4
3/8	3/4	3/16	5/32	5/16
7/16	7/8	1/4	3/16	3/8
9/16	1 1/8	11/32	3/16	1/2
11/16	1 1/4	15/32	1/4	5/8
3/4	1 1/2	9/16	5/16	5/8
7/8	1 3/4	11/16	5/16	3/4

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Rules for Figuring Tapers.

1. If the taper foot is known, the taper per inch is found by dividing the taper per foot by 12.

2. If the taper per inch is known, the taper per foot is found by multiplying the taper per inch by 12.

3. To find the taper per foot, when the diameters at the large and small ends and the length of the taper are given, subtract the small diameter from the large, divide the remainder by the length of the taper, and multiply the result by 12.

4. To find the diameter at the small end when the diameter at the large end, the length of the taper, and the taper per foot are given, divide the taper per foot by 12, multiply the quotient by the length of the taper, and subtract the resulting dimension from the diameter at the large end.

5. To find the diameter at the large end when the diameter at the small end, the length of the taper, and the taper per foot are given, divide the taper per foot by 12, multiply the quotient by the length of the taper, and add the resulting dimension to the diameter at the small end.

6. To find the dimension between two given diameters of a piece of work, when the taper per foot is given, subtract the diameter at the small end from the diameter at the large end, and divide the remainder by the taper per foot divided by 12.

7. To find how much a piece of work tapers in a certain length, when the taper per foot is given, divide the taper per foot by 12, and multiply the quotient by the dimension of the certain length in which the taper is required.

Rules for Setting over the Lathe Tail-Stock for Taper Turning.

1. To find the amount to set over the tail-stock for work tapering for its full length, when the taper per foot and length of the work are known, divide the taper per foot by 12, multiply the quotient by the length of the work, and divide this result, in turn, by 2.

2. To find the amount to set over the tail-stock for work tapering for its full length, when the diameters at the large and small ends are known, subtract the small diameter from the large, and divide the remainder by 2.

3. To find the amount to set over the tail-stock for work partly tapered and partly straight, when the diameters at the large and small ends of the taper, the length of the taper, and the total length of the work are known, subtract the small diameter from the large, divide the remainder by the length of the taper, multiply the quotient thus obtained by the total length of the work, and finally divide by 2.

4. To find the amount to set over the tail-stock for work partly tapered and partly straight, when the taper per foot and the length of the work are known, divide the taper per foot by 12, multiply the quotient by the length of the work, and divide this result, in turn, by 2.

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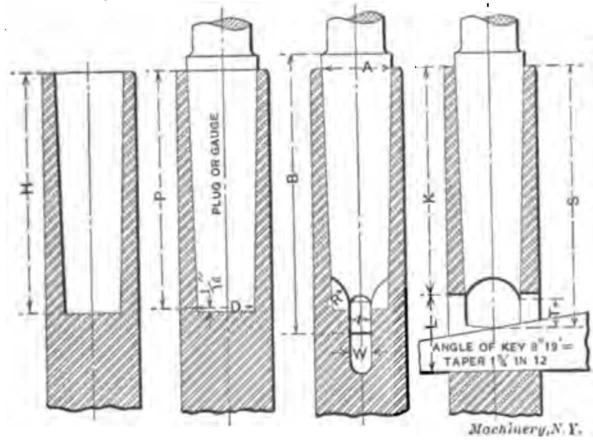
STANDARD TAPER PINS.

No. of Taper Pin	Diameter at Large End of Pin	Approx. Fractional Size at Large End of Pin	Length of Longest Pin of This Size	No. of Taper Pin	Diameter at Large End of Pin	Approx. Fractional Size at Large End of Pin	Length of Longest Pin of This Size
000000	0.0715	5/64	5/8	3	0.219	7/32	1 3/4
00000	0.092	3/32	5/8	4	0.250	1/4	2
0000	0.108	7/64	3/4	5	0.289	19/64	2 1/4
000	0.125	1/8	3/4	6	0.341	11/32	3 1/4
00	0.147	9/64	1	7	0.409	13/32	3 3/4
0	0.156	5/32	1	8	0.492	1/2	4 1/2
1	0.172	11/64	1 1/4	9	0.591	19/32	5 1/4
2	0.193	3/16	1 1/2	10	0.706	23/32	6

MORSE STANDARD TAPERS.



Number of Taper	Diameter of Plug at Small End	Diameter at End of Socket	Standard Plug Depth	Whole Length of Shank	Depth of Hole	End of Socket to Keyway	Length of Keyway	Length of Tongue	Thickness of Tongue	Width of Keyway	Shank Depth	Taper per Foot
	D	A	P	B	H	K	L	T	t	W	S	
0	0.252	0.356	2	2 11/32	2 1/32	1 15/16	9/16	1/4	5/32	0.160	2 7/32	0.625
1	0.369	0.475	2 1/8	2 9/16	2 3/16	2 1/16	3/4	5/16	13/64	0.213	2 3/8	0.600
2	0.572	0.700	2 9/16	3 1/16	2 5/8	2 1/2	7/8	3/8	1/4	0.260	2 7/8	0.602
3	0.778	0.938	3 3/16	3 3/4	3 1/4	3 1/16	1 1/16	7/16	5/16	0.322	3 9/16	0.602
4	1.020	1.231	4 1/16	4 3/4	4 1/8	3 7/8	1 1/4	1/2	15/32	0.478	4 1/2	0.623
5	1.475	1.748	5 3/16	6	5 1/4	4 15/16	1 1/2	5/8	5/8	0.635	5 3/4	0.630
6	2.116	2.494	7 1/4	8 5/16	7 3/8	7	1 3/4	7/8	3/4	0.760	8	0.626
7	2.750	3.270	10	11 5/8	10 1/8	9 1/2	2 5/8	1 3/8	1 1/8	1.185	11 1/4	0.625

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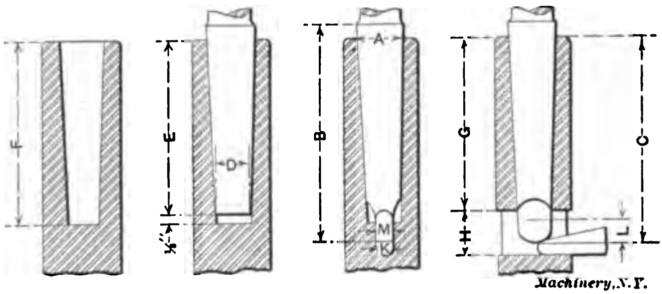
BROWN & SHARPE STANDARD TAPERS.



Number of Taper	Diameter at End of Socket	Whole Length of Shank	Shank Depth	Diameter of Plug at Small End	Standard Plug Depth	Depth of Hole	End of Socket to Keyway	Length of Keyway	Width of Keyway	Length of Tongue	Thickness of Tongue
	A	B	C	D	E	F	G	H	K	L	M
1	0.239	1 9/32	1 3/16	0.200	15/16	1 1/16	15/16	3/8	0.135	3/16	1/8
2	0.299	1 19/32	1 1/2	0.250	1 3/16	1 5/16	1 11/64	1/2	0.166	1/4	5/32
3	0.375	1 31/32	1 7/8	0.312	1 1/2	1 5/8	1 15/32	5/8	0.197	5/16	3/16
3	0.385	2 7/32	2 1/8	0.312	1 3/4	1 7/8	1 23/32	5/8	0.197	5/16	3/16
3	0.395	2 15/32	2 3/8	0.312	2	2 1/8	1 31/32	5/8	0.197	5/16	3/16
4	0.402	1 3/4	1 21/32	0.350	1 1/4	1 3/8	1 13/64	11/16	0.228	11/32	7/32
4	0.420	2 1/16	2 3/32	0.350	1 11/16	1 13/16	1 41/64	11/16	0.228	11/32	7/32
5	0.523	2 9/32	2 3/16	0.450	1 3/4	1 7/8	1 11/16	3/4	0.260	3/8	1/4
5	0.533	2 17/32	2 7/16	0.450	2	2 1/8	1 15/16	3/4	0.260	3/8	1/4
5	0.539	2 21/32	2 9/16	0.450	2 1/8	2 1/4	2 1/16	3/4	0.260	3/8	1/4
6	0.599	2 31/32	2 7/8	0.500	2 3/8	2 1/2	2 19/64	7/8	0.291	7/16	9/32
6	0.635	3 27/32	3 3/4	0.500	3 1/4	3 3/8	3 11/64	7/8	0.291	7/16	9/32
7	0.704	3 1/8	3 1/32	0.600	2 1/2	2 5/8	2 13/32	15/16	0.322	15/32	5/16
7	0.720	3 1/2	3 13/32	0.600	2 7/8	3	2 25/32	15/16	0.322	15/32	5/16
7	0.725	3 5/8	3 17/32	0.600	3	3 1/8	2 29/32	15/16	0.322	15/32	5/16
7	0.767	4 5/8	4 17/32	0.600	4	4 1/8	3 29/32	15/16	0.322	15/32	5/16
8	0.898	4 1/4	4 1/8	0.750	3 9/16	3 11/16	3 29/64	1	0.353	1/2	11/32
8	0.917	4 11/16	4 9/16	0.750	4	4 1/8	3 57/64	1	0.353	1/2	11/32
9	1.067	4 3/4	4 5/8	0.900	4	4 1/8	3 7/8	1 1/8	0.385	9/16	3/8
9	1.077	5	4 7/8	0.900	4 1/4	4 3/8	4 1/8	1 1/8	0.385	9/16	3/8
10	1.260	5 27/32	5 23/32	1.0446	5	5 1/8	4 27/32	1 5/16	0.447	21/32	7/16
10	1.289	6 17/32	6 13/32	1.0446	5 11/16	5 13/16	5 17/32	1 5/16	0.447	21/32	7/16
10	1.312	7 1/16	6 15/16	1.0446	6 7/32	6 11/32	6 1/16	1 5/16	0.447	21/32	7/16
11	1.498	6 23/32	6 21/32	1.250	5 15/16	6 1/16	5 25/32	1 5/16	0.447	21/32	7/16
11	1.531	7 19/32	7 15/32	1.250	6 3/4	6 7/8	6 19/32	1 5/16	0.447	21/32	7/16
12	1.797	8 1/16	7 15/16	1.500	7 1/8	7 1/4	6 15/16	1 1/2	0.510	3/4	1/2
13	2.073	8 11/16	8 9/16	1.750	7 3/4	7 7/8	7 9/16	1 1/2	0.510	3/4	1/2
14	2.344	9 9/32	9 5/32	2.000	8 1/4	8 3/8	8 1/32	1 11/16	0.572	27/32	9/16
15	2.615	9 25/32	9 21/32	2.250	8 3/4	8 7/8	8 17/32	1 11/16	0.572	27/32	9/16
16	2.885	10 3/8	10 1/4	2.500	9 1/4	9 3/8	9	1 7/8	0.635	15/16	5/8

The taper per foot is ½ inch, except for No. 10, where the taper is 0.5161 inch per foot

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TABLE GIVING THE AMOUNT OF TAPER IN A CERTAIN LENGTH, WHEN THE TAPER PER FOOT IS GIVEN.

Length of of Tapered Portion	Taper per Foot.										
	1/16	3/32	1/8	1/4	3/8	1/2	0.600	5/8	3/4	1	1 1/4
1/32	0.0002	0.0002	0.0003	0.0007	0.0010	0.0013	0.0016	0.0016	0.0020	0.0026	0.0033
1/16	0.0003	0.0005	0.0007	0.0013	0.0020	0.0026	0.0031	0.0033	0.0039	0.0052	0.0065
1/8	0.0007	0.0010	0.0013	0.0026	0.0039	0.0052	0.0062	0.0065	0.0078	0.0104	0.0130
3/16	0.0010	0.0015	0.0020	0.0039	0.0059	0.0078	0.0094	0.0098	0.0117	0.0156	0.0195
1/4	0.0013	0.0020	0.0026	0.0052	0.0078	0.0104	0.0125	0.0130	0.0156	0.0208	0.0260
5/16	0.0016	0.0024	0.0033	0.0065	0.0098	0.0130	0.0156	0.0163	0.0195	0.0260	0.0326
3/8	0.0020	0.0029	0.0039	0.0078	0.0117	0.0156	0.0187	0.0195	0.0234	0.0312	0.0391
7/16	0.0023	0.0034	0.0046	0.0091	0.0137	0.0182	0.0219	0.0228	0.0273	0.0365	0.0456
1/2	0.0026	0.0039	0.0052	0.0104	0.0156	0.0208	0.0250	0.0260	0.0312	0.0417	0.0521
9/16	0.0029	0.0044	0.0059	0.0117	0.0176	0.0234	0.0281	0.0293	0.0352	0.0469	0.0586
5/8	0.0033	0.0049	0.0065	0.0130	0.0195	0.0260	0.0312	0.0326	0.0391	0.0521	0.0651
11/16	0.0036	0.0054	0.0072	0.0143	0.0215	0.0286	0.0344	0.0358	0.0430	0.0573	0.0716
3/4	0.0039	0.0059	0.0078	0.0156	0.0234	0.0312	0.0375	0.0391	0.0469	0.0625	0.0781
13/16	0.0042	0.0063	0.0085	0.0169	0.0254	0.0339	0.0406	0.0423	0.0508	0.0677	0.0846
7/8	0.0046	0.0068	0.0091	0.0182	0.0273	0.0365	0.0437	0.0456	0.0547	0.0729	0.0911
15/16	0.0049	0.0073	0.0098	0.0195	0.0293	0.0391	0.0469	0.0488	0.0586	0.0781	0.0977
1	0.0052	0.0078	0.0104	0.0208	0.0312	0.0417	0.0500	0.0521	0.0625	0.0833	0.1042
2	0.0104	0.0156	0.0208	0.0417	0.0625	0.0833	0.1000	0.1042	0.1250	0.1667	0.2083
3	0.0156	0.0234	0.0312	0.0625	0.0937	0.1250	0.1500	0.1562	0.1875	0.2500	0.3125
4	0.0208	0.0312	0.0417	0.0833	0.1250	0.1667	0.2000	0.2083	0.2500	0.3333	0.4167
5	0.0260	0.0391	0.0521	0.1042	0.1562	0.2083	0.2500	0.2604	0.3125	0.4167	0.5208
6	0.0312	0.0469	0.0625	0.1250	0.1875	0.2500	0.3000	0.3125	0.3750	0.5000	0.6250
7	0.0365	0.0547	0.0729	0.1458	0.2187	0.2917	0.3500	0.3646	0.4375	0.5833	0.7292
8	0.0417	0.0625	0.0833	0.1667	0.2500	0.3333	0.4000	0.4167	0.5000	0.6667	0.8333
9	0.0469	0.0703	0.0937	0.1875	0.2812	0.3750	0.4500	0.4687	0.5625	0.7500	0.9375
10	0.0521	0.0781	0.1042	0.2083	0.3125	0.4167	0.5000	0.5208	0.6250	0.8333	1.0417
11	0.0573	0.0859	0.1146	0.2292	0.3437	0.4583	0.5500	0.5729	0.6875	0.9167	1.1458
12	0.0625	0.0937	0.1250	0.2500	0.3750	0.5000	0.6000	0.6250	0.7500	1.0000	1.2500

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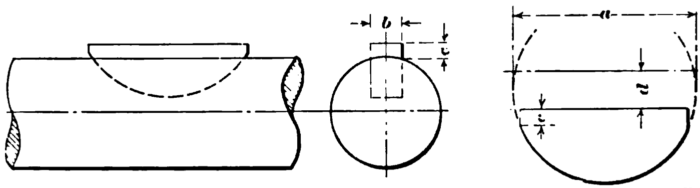
The Jarno Taper.

In the Jarno taper, a definite relation exists between the number of the taper, its length, and the diameters at the large and small ends. Given the number of the taper, the length is as many half inches as expressed by the number, the diameter at the large end as many eighths of an inch, and the diameter as many tenths of an inch as designated by the number of the taper. Thus, a number 7 Jarno taper is seven-halfs or 3½ inches long, ⅞ inch diameter at the large end and 0.7 inch at the small end. The taper per foot of all Jarno tapers is 0.600 inch.

TABLE OF JARNO TAPERS.

No. of Taper	Diameter at Large End	Diameter at Small End	Length	No. of Taper	Diameter at Large End	Diameter at Small End	Length
2	0.250	0.200	1	12	1.500	1.200	6
3	0.375	0.300	1½	13	1.625	1.300	6½
4	0.500	0.400	2	14	1.750	1.400	7
5	0.625	0.500	2½	15	1.875	1.500	7½
6	0.750	0.600	3	16	2.000	1.600	8
7	0.875	0.700	3½	17	2.125	1.700	8½
8	1.000	0.800	4	18	2.250	1.800	9
9	1.125	0.900	4½	19	2.375	1.900	9½
10	1.250	1.000	5	20	2.500	2.000	10
11	1.375	1.100	5½				

SPECIAL WOODRUFF KEYS.



No. of Key	Diameter of Key	Thickness of Key	Depth of Keyway	Center of Stock from which Key is made to Top of Key	Width of Flat	No. of Key	Diameter of Key	Thickness of Key	Depth of Keyway	Center of Stock from which Key is made to Top of Key	Width of Flat
	a	b	c	d	e		a	b	c	d	e
26	2 1/8	3/16	3/32	17/32	3/32	31	3 1/2	7/16	7/32	13/16	3/16
27	2 1/8	1/4	1/8	17/32	3/32	32	3 1/2	1/2	1/4	13/16	3/16
28	2 1/8	5/16	5/32	17/32	3/32	33	3 1/2	9/16	9/32	13/16	3/16
29	2 1/8	3/8	3/16	17/32	3/32	34	3 1/2	5/8	5/16	13/16	3/16
30	3 1/2	3/8	3/16	13/16	3/16						

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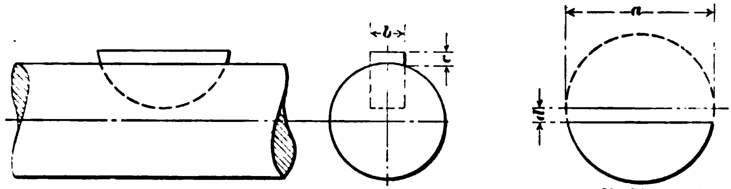
STANDARD WOODRUFF KEYS.



Machinery, N.Y.

No. of Key	Diam. of Key	Thick-ness of Key	Depth of Keyway	Center of Stock from which Key is made to Top of Key	No. of of Key	Diam. of Key	Thick-ness of Key	Depth of Keyway	Center of stock from which key is made to Top of Key
	a	b	c	d		a	b	c	d
1	$\frac{1}{2}$	$\frac{1}{16}$	$\frac{1}{32}$	$\frac{3}{64}$	B	1	$\frac{5}{16}$	$\frac{5}{32}$	$\frac{1}{16}$
2	$\frac{1}{2}$	$\frac{3}{32}$	$\frac{3}{64}$	$\frac{3}{64}$	16	$1\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{5}{64}$
3	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{16}$	$\frac{3}{64}$	17	$1\frac{1}{8}$	$\frac{7}{32}$	$\frac{7}{64}$	$\frac{5}{64}$
4	$\frac{5}{8}$	$\frac{3}{32}$	$\frac{3}{64}$	$\frac{1}{16}$	18	$1\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{5}{64}$
5	$\frac{5}{8}$	$\frac{1}{8}$	$\frac{1}{16}$	$\frac{1}{16}$	C	$1\frac{1}{8}$	$\frac{5}{16}$	$\frac{5}{32}$	$\frac{5}{64}$
6	$\frac{5}{8}$	$\frac{5}{32}$	$\frac{5}{64}$	$\frac{1}{16}$	19	$1\frac{1}{4}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{5}{64}$
7	$\frac{3}{4}$	$\frac{1}{8}$	$\frac{1}{16}$	$\frac{1}{16}$	20	$1\frac{1}{4}$	$\frac{7}{32}$	$\frac{7}{64}$	$\frac{5}{64}$
8	$\frac{3}{4}$	$\frac{5}{32}$	$\frac{5}{64}$	$\frac{1}{16}$	21	$1\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{5}{64}$
9	$\frac{3}{4}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{1}{16}$	D	$1\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{32}$	$\frac{5}{64}$
10	$\frac{7}{8}$	$\frac{5}{32}$	$\frac{5}{64}$	$\frac{1}{16}$	E	$1\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{5}{64}$
11	$\frac{7}{8}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{1}{16}$	22	$1\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{3}{32}$
12	$\frac{7}{8}$	$\frac{7}{32}$	$\frac{7}{64}$	$\frac{1}{16}$	23	$1\frac{3}{8}$	$\frac{5}{16}$	$\frac{5}{32}$	$\frac{3}{32}$
A	$\frac{7}{8}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{16}$	F	$1\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{3}{32}$
13	1	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{1}{16}$	24	$1\frac{1}{2}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{7}{64}$
14	1	$\frac{7}{32}$	$\frac{7}{64}$	$\frac{1}{16}$	25	$1\frac{1}{2}$	$\frac{5}{16}$	$\frac{5}{32}$	$\frac{7}{64}$
15	1	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{16}$	G	$1\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{7}{64}$

STANDARD WOODRUFF KEYS TO USE WITH VARIOUS DIAMETER SHAFTS.

Diameter of Shaft	Number of Keys	Diameter of Shaft	Number of Keys	Diameter of Shaft	Number of Keys
$\frac{5}{16}$—$\frac{3}{8}$	1	$\frac{7}{8}$—$\frac{15}{16}$	6, 8, 10	$1\frac{3}{8}$—$1\frac{7}{16}$	14, 17, 20
$\frac{7}{16}$—$\frac{1}{2}$	2, 4	1	9, 11, 13	$1\frac{1}{2}$—$1\frac{5}{8}$	15, 18, 21, 24
$\frac{9}{16}$—$\frac{5}{8}$	3, 5	$1\frac{1}{16}$—$1\frac{1}{8}$	9, 11, 13, 16	$1\frac{11}{16}$—$1\frac{3}{4}$	18, 21, 24
$\frac{11}{16}$—$\frac{3}{4}$	3, 5, 7	$1\frac{3}{16}$	11, 13, 16	$1\frac{13}{16}$—2	23, 25
$\frac{13}{16}$	6, 8	$1\frac{1}{4}$—$1\frac{5}{16}$	12, 14, 17, 20	$2\frac{1}{16}$—$2\frac{1}{2}$	25

Rules for Calculating Cutting Speeds and Feeds.

1. To find the number of revolutions per minute, when the diameter of work (or drill) in inches and the cutting speed in feet per minute are known, multiply the diameter by 3.14, and divide the result by 12; then divide the cutting speed by the figure thus obtained.

2. To find the cutting speed in feet per minute, when the diameter of the work (or drill) in inches, and the number of revolutions per minute are given, multiply the diameter by 3.14 and divide the result

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by 12; then multiply the quotient thus obtained by the number of revolutions per minute.

3. To find the time required for one complete cut over the work, when the feed per revolution, the total length of the cut, and the number of revolutions per minute are given, divide the total length of the cut by the number of revolutions per minute multiplied by the feed per revolution. If the cutting speed is given, originally, instead of the number of revolutions, find the latter number first from Rule 1.

Rules for Finding Gears for Transmitting Motion between Two Shafts.

1. Place the number of revolutions of the driven shaft in the numerator, and the corresponding number of revolutions of the driving shaft in the denominator of a fraction (or, in general, write the ratio in the form of a fraction), and multiply the numerator and denominator with the same number, until a new fraction is obtained having numerator and denominator expressing suitable numbers of teeth for the gears. The gear represented by the new numerator is the driving gear, and that represented by the new denominator is the driven gear.

2. If compounding of the gears is necessary or advisable, divide up both numerator and denominator in the fraction, giving the ratio, in two factors, and multiply each pair of factors (one factor in the numerator and one in the denominator making "one pair") by the same numbers, until gears with suitable numbers of teeth are found.

Rules for Finding Change Gears for Screw Cutting in the Lathe.

1. To find the number of threads per inch, if the lead of a thread is given, divide 1 by the lead.

2. To find the "screw-cutting constant" of a lathe, place equal gears on spindle stud and lead-screw, and if the gearing is compounded, also equals gears on the intermediate stud; then cut a thread on a piece in the lathe. The number of threads cut with equal gears is called the "screw-cutting constant" of *that particular lathe.*

3. To find the change gears used in simple gearing, when the screw-cutting constant as found from Rule 2, and the number of threads per inch to be cut are given, place the screw-cutting constant of the lathe as numerator and the number of threads to be cut as denominator in a fraction, and multiply numerator and denominator with the same number until a new fraction results representing suitable number of teeth for the change gears. In the new fraction, the numerator represents the number of teeth in the gear on the spindle stud, and the denominator, the number of teeth in the gear on the lead-screw.

4. To find the change gears used in compound gearing, place the screw-cutting constant as found from Rule 2 as numerator, and the number of threads per inch to be cut as denominator in a fraction; divide up both numerator and denominator in two factors each, and multiply each pair of factors (one factor in the numerator and one in the denominator making "a pair") by the same number, until new fractions result representing suitable number of teeth for the change

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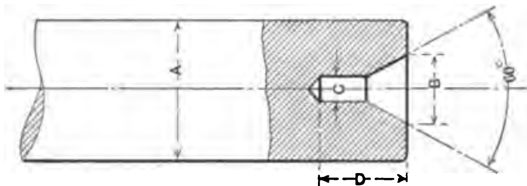
gears. The gears represented by the numbers in the new numerators are driving gears, and those in the denominators are driven gears.

CENTERS FOR REAMERS AND ARBORS.



Machinery, N.Y.

Diameter of Arbor A	Largest Diameter of Center B	Drill C	Depth of Hole D	Diameter of Arbor A	Largest Diameter of Center B	Drill C	Depth of Hole D
1/4	1/8	55	5/32	2 3/8	43/64	H	27/32
5/16	5/32	52	3/16	2 1/2	11/16	J	27/32
3/8	3/16	48	7/32	2 5/8	45/64	K	7/8
7/16	7/32	48	1/4	2 3/4	23/32	L	29/32
1/2	1/4	39	5/16	2 7/8	47/64	M	29/32
9/16	9/32	33	11/32	3	3/4	N	15/16
5/8	5/16	30	3/8	3 1/8	49/64	N	31/32
11/16	11/32	29	13/32	3 1/4	25/32	O	31/32
3/4	3/8	25	7/16	3 3/8	51/64	O	1
13/16	13/32	20	1/2	3 1/2	13/16	P	1
7/8	7/16	17	17/32	3 5/8	53/64	Q	1 1/16
15/16	15/32	12	9/16	3 3/4	27/32	R	1 1/16
1	1/2	8	19/32	3 7/8	55/64	R	1 1/16
1 1/8	33/64	5	5/8	4	7/8	S	1 1/8
1 1/4	17/32	3	21/32	4 1/8	57/64	T	1 1/8
1 3/8	35/64	2	21/32	4 1/4	29/32	T	1 1/8
1 1/2	9/16	1	11/16	4 3/8	59/64	U	1 3/16
1 5/8	37/64	A	23/32	4 1/2	15/16	V	1 3/16
1 3/4	19/32	B	23/32	4 5/8	61/64	V	1 3/16
1 7/8	39/64	C	3/4	4 3/4	31/32	W	1 1/4
2	5/8	E	3/4	4 7/8	63/64	X	1 1/4
2 1/8	41/64	F	25/32	5	1	X	1 1/4
2 1/4	21/32	G	13/16				

Rules for Finding Change Gears for Cutting Spirals in the Milling Machine.

First find the "lead" of the milling machine. To find the lead of a milling machine, place equal gears on the worm stud, and on the feed-screw, and multiply the number of revolutions made by the feed-screw in order to produce one revolution of the index head spindle, by the lead of the feed-screw.

To then find the change gears to be used in a simple train of gearing, when cutting spirals on a milling machine, place the lead of the spiral in the numerator and the lead of the milling machine in the denominator of a fraction, and multiply the numerator and denominator with the same number, until a new fraction is obtained in which the numerator and denominator give suitable numbers of teeth. If

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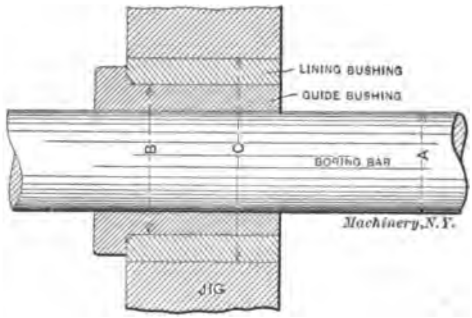
compounding of the gears is necessary, follow the same rule as regards dividing up numerator and denominator in factors, as given on page 44 for compound gearing in the lathe.

Rule for Size of Pulleys and Speed of Shafts.

The number of revolutions of one shaft multiplied with the diameter of the pulley on the same shaft, divided by the number of revolutions of the second shaft, gives the diameter of the pulley of the second shaft.

DIMENSIONS OF BORING BARS AND JIG BUSHINGS.



Diameter of Finished Hole, Reamed	Diameter of Drill	Diameter of Rough Bore	Diameter of Finished Bore	Diameter of Boring Bar A	Diameter of Guide Bushing B	Diameter of Lining Bushing C
0.4875	3/8	0.420	0.430	0.3125	0.5625	0.750
0.500	7/16	0.485	0.495	0.375	0.625	0.875
0.5625	1/2	0.547	0.557	0.4375	0.750	1.000
0.625	9/16	0.610	0.620	0.500	0.875	1.125
0.6875	19/32	0.670	0.682	0.500	0.875	1.125
0.750	5/8	0.730	0.745	0.500	0.875	1.125
0.8125	11/16	0.790	0.807	0.625	1.000	1.250
0.875	3/4	0.855	0.870	0.625	1.000	1.250
0.9375	13/16	0.925	0.940	0.750	1.125	1.4375
1.000	7/8	0.975	0.993	0.750	1.125	1.4375
1.0625	15/16	1.040	1.055	0.8125	1.1875	1.500
1.125	1	1.100	1.118	0.875	1.3125	1.750
1.1875	1 1/16	1.160	1.180	0.9375	1.375	1.750
1.250	1 1/8	1.220	1.243	1.000	1.500	1.875
1.375	1 3/16	1.345	1.368	1.0625	1.500	1.875
1.500	1 5/16	1.470	1.493	1.125	1.625	2.000
1.625	1 7/16	1.595	1.618	1.250	1.8125	2.1875
1.750	1 9/16	1.720	1.743	1.3125	1.9375	2.375
1.875	1 5/8	1.840	1.867	1.375	2.125	2.625
2.000	1 3/4	1.960	1.992	1.500	2.250	2.750
2.250	2	2.210	2.242	1.750	2.500	3.000
2.500	2 3/16	2.460	2.492	2.000	2.750	3.250
2.750	2 3/8	2.710	2.742	2.125	3.000	3.625
3.000	2 5/8	2.960	2.992	2.375	3.375	4.125

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No. 11. BEARINGS.—Design of Bearings; Causes of Hot Bearings; Alloys for Bearings; Ball Bearings; Friction of Roller Bearings.

No. 12. MATHEMATICS OF MACHINE DESIGN.—Compiled with special reference to shafting and efficiency of hoisting machinery.

No. 13. BLANKING DIES.—Making Blanking Dies; Blanking and Piercing Dies; Construction of Split Dies; Novel Ideas in Die Making.

No. 14. DETAILS OF MACHINE TOOL DESIGN.—Cone Pulleys and Belts; Strength of Countershafts; Tumbler Gear Design; Faults of Iron Castings.

No. 15. SPUR GEARING.—First Principles of Gearing; Formulas for Spur Gearing; Design and Calculation of Gear Wheels; Strength of Gear Teeth.

No. 16. MACHINE TOOL DRIVES.—Speeds and Feeds of Machine Tools; Geared or Single Pulley Drives; Drives for High Speed Cutting Tools.

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No. 19. USE OF FORMULAS IN MECHANICS.—Mathematical Signs and Formulas; Strength of Materials; Graphical Methods; Levers; Center of Gravity.

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The foregoing books, up to and including No. 26, were published and in stock in November, 1908. The remainder will go to press as rapidly as practicable. The complete plan of the series, as stated, is to cover the whole field of mechanical practice, and the editors are preparing the additional titles, which will, from time to time, be announced in MACHINERY.

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No. 36

IRON AND STEEL

PRINCIPLES OF MANUFACTURE, STRUCTURE, COMPOSITION AND TREATMENT

CONTENTS

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CHAPTER I.

PRINCIPLES OF IRON AND STEEL MANUFACTURE.

The principles of iron and steel manufacture as outlined in the present chapter were originally given in an article by Mr. George Schuhmann in *The Pilot*, and republished in the August, 1907, issue of MACHINERY.

Commercial iron and steel are metallic mixtures, the chief ingredient of which is the element "iron," that is, pure iron, of which they contain from 93 per cent to over 99 per cent. The difference between iron and steel is principally due to the composition and proportion of the remaining ingredients.

Iron ore is an oxide of iron (iron rust) containing from 35 per cent to 65 per cent of iron; the balance is oxygen, phosphorus, sulphur, silica (sand), and other impurities. The ore is charged in a blast furnace, mixed with limestone as a flux, and melted down with either charcoal, coke, or anthracite coal as fuel; the resulting metal is what is commercially known as pig iron, containing about 93 per cent of pure iron, 3 to 5 per cent of carbon (pure coal), some silicon, phosphorus, sulphur, etc. This pig iron is used in foundries for the manufacture of iron castings, by simply remelting it in a cupola without materially changing its chemical composition; the only result is a closer grain and somewhat increased strength.

The Puddling Process.

In the manufacture of wrought iron the pig iron is remelted in so-called puddling furnaces, by charging about ½ ton in a furnace; while in a molten state, the iron is stirred up with large iron hooks by the puddler and his helper, and kept boiling, so as to expose every part of the iron bath to the action of the flame in order to burn out the carbon. The other impurities will separate from the iron, forming the puddle cinder.

The purer the iron the higher is its melting point. Pig iron melts at about 2,100 degrees F., steel at about 2,500 degrees, and wrought iron at about 2,800 degrees. The temperature in the puddling furnace is high enough to melt pig iron, but not high enough to keep wrought iron in a liquid state; therefore, as soon as the small particles of iron become purified they partly congeal (come to nature), forming a spongy mass in which small globules of iron are in a semi-plastic state, feebly cohering with fluid cinder filling the cavities between them. This sponge is divided by the puddler into lumps of about 200 pounds each; these lumps or balls are taken to a steam hammer or squeezer, where they are hammered or squeezed into elongated blocks (blooms), and while still hot, rolled out between the puddle rolls into bars 3 to 6 inches wide, ¾ inch thick, and 15 to 30 feet long. These

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bars are called puddle bars or muck bars, and, owing to the large amount of cinder still contained therein, they have rather rough surfaces. The muck bars are cut up into pieces from 2 to 4 inches long, and piled on top of each other in so-called "piles" varying from 100 to 2,000 pounds, according to the size product desired. These piles are heated in heating furnaces, and when white hot, are taken to the rolls to be welded together and rolled out into merchant iron in the shape of either sheets, plates, bars, or structural shapes, as desired. When cold, this material is sheared and straightened, and is then ready for the market.

After leaving the puddling furnace, wrought iron does not undergo any material change in its chemical composition, and the only physical change is an expulsion of a large portion of the cinder; the small cinder-coated globules of iron are welded together and the subsequent rolling back and forth will elongate these globules, giving the iron a fibrous structure, and the reheating and rerolling will drive these fibers closer together, thus increasing the strength and ductility of the metal.

Classes and Kinds of Steel.

The word steel, nowadays, covers a multitude of mixtures which are very different from each other in their chemical as well as physical qualities. The ingredient that exerts most influence on these variations is carbon. High grade razor steel contains about 1¼ per cent of carbon, springs 1 per cent, steel rails from ½ to ¾ per cent, and soft steel boiler plate may go as low as 1/16 per cent of carbon. Steel which is very low in carbon can easily be welded, but it cannot be tempered; when carbon is above 1/3 per cent, welding is more difficult and can only be done by the use of borax or some other flux, or by electric or thermit welding. Steel with carbon above ¾ per cent can be tempered, that is, when heated to red heat and then quenched in water or other liquid, it becomes very hard and can be used for tools of various kinds, such as saws, files, drills, chisels, cutlery, etc. In tool steel other ingredients are sometimes used to influence its hardness, such as nickel, manganese, chrome, tungsten, etc., the last named playing an important part in so-called "high-speed steels," that is, tool steels that will cut metal at a high speed without losing their temper or hardness.

As stated above, pig iron and cast iron contain about 4 per cent of carbon, and wrought iron only a trace of it, while steel is between these two extremes. The manufacture of steel, therefore, refers principally to getting the right proportion of carbon. One method is to take pig iron and burn the carbon out of it, as in the Bessemer and open-hearth processes, and the other method is to take wrought iron and add carbon to it, as in the cementation and crucible processes.

The Bessemer Process.

In the Bessemer process the molten pig iron is put into a large pear-shaped vessel, called the converter, the bottom of which is double, the inner one being perforated with numerous holes, called tuyeres,

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to admit air to be forced in under pressure. The molten iron (from 10 to 15 tons at a time) is poured into the converter while the latter is lying on its side; then the compressed air is turned into the double bottom as the converter rises to a vertical position. The air has sufficient pressure (about 20 pounds per square inch) to prevent the molten metal from entering the tuyeres. The air streams pass up through the molten metal (piercing it like so many needles), burning out the carbon, silicon, etc., accompanied by a brilliant display of sparks and a flame shooting out of the mouth of the converter. The 15 tons of molten pig iron contain nearly ¾ of a ton of carbon, and since this carbon is all burned out in less than ten minutes, this rapid rate of combustion increases the heat of the metal very much; it does not cool it, as one would suppose at first thought. The flame, therefore, at first red, becomes brighter and brighter, until it is finally so white that it can sacrcely be looked at with the naked eye. A "blow" generally lasts about nine to ten minutes, when the sudden dropping of the flame gives notice that the carbon is all burned out. The metal in the converter is then practically liquid wrought iron, the converter is then laid on its side again, the blast shut off and a certain amount of spiegeleisen or ferromanganese is added in a liquid form so as to give the steel the proper amount of carbon and manganese to make it suitable for the purpose desired. The liquid steel is then poured out into so-called "ingot molds," and the resulting "ingots," while still hot, but no longer liquid, are rolled out into blooms, billets, or rails without any additional reheating except a short sojourn in so-called "soaking pits." In some steel works, where the molten pig iron is taken in large ladle cars direct from the blast furnace to the converter, it is possible to produce rails without adding any fuel to that contained in the molten pig iron, so that the red-hot rail just finished still contains some of the heat given it by the coke in the blast furnace.

The Open-hearth Process.

The open-hearth process, sometimes called "the Siemens-Martin process," is similar to the puddling process, but on a much larger scale. The furnaces generally have a capacity of from 40 to 50 tons of molten metal (in some exceptional cases as high as 200 tons); they are heated by gas made from bituminous coal (oil and natural gas have also been used). The gas and the air needed for its combustion are heated to a high temperature (over 1,000 degrees) before entering the combustion chamber, by passing them through so-called regenerative chambers. Owing to this preheating of the gas and the air, a very high temperature can be maintained in the furnace, so as to keep the iron liquid even after it has parted with its carbon. The stirring up of the molten metal is not done by hooks as in the puddling furnace, but by adding to the charge a certain proportion of ore, iron scale, or other oxides, the chemical reaction of which keeps the molten iron in a state of agitation. While in the Bessemer process only pig iron is used, in the open-hearth furnace it is practicable to

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use also scrap of wrought iron or steel, as the high temperature in the furnace will readily melt it. When the pig iron or scrap contains too much phosphorus, burnt lime is added to the charge; the resulting slag will absorb the phosphorus, thus taking it out of the metal. This dephosphorization by means of burnt lime is called the basic process in contradistinction to the acid process, where no lime is used, but where care must be taken that the metal charged is low in phosphorus. In this country, the basic process is at present used only in connection with open-hearth furnaces, while in Europe it is also used in many Bessemer plants producing the so-called "basic Bessemer steel."

Producing Tool Steel.

Crucible steel or tool steel, formerly called cast steel, is made by using high grade, low phosphorus wrought iron and adding carbon to it. The oldest method is the so-called "cementation process" in which the iron bars are packed in air-tight retorts, with powdered charcoal between the bars. The filled retorts are put into a cementation furnace, where they are heated to a red heat and kept at that temperature for several days, during which time the iron will absorb about 1½ per cent of its own weight of carbon. The process is similar to the case-hardening process familiar to many blacksmiths. The carbonized bars, called " blister steel," are then cut into small pieces, remelted in a crucible, and from there poured into molds, forming small billets, which are afterward hammered or rolled into the desired shapes. The newer method is to put the small pieces of wrought iron direct into an air-tight crucible mixed with the proper amount of powdered charcoal, and melt down; the iron will absorb the carbon much quicker while in a molten state than when only red-hot, as in the cementation furnace. The other ingredients, such as chrome, tungsten, etc., are also added in the crucible.

Malleable and Steel Castings.

Malleable castings are produced in the reverse way from the blister steel referred to above, that is, instead of taking wrought iron and adding carbon, castings made of cast iron are made malleable by extracting the carbon. The castings are packed into retorts similar to the cementation retorts, but, instead of charcoal, an oxide of iron, generally in the shape of hematite ore, is packed with them, and kept in a red-hot state for several days. The oxygen of the ore will absorb the carbon in the iron, giving the latter a somewhat steely nature.

Steel castings used to be produced in the same manner, but now, steel castings are cast direct from the ladle containing molten steel, which is generally melted in an open-hearth furnace, although small Bessemer converters are also sometimes used for this purpose.

Difference between Wrought Iron and Low Carbon Steel.

While chemically there is not much difference between wrought iron and low carbon steel, there is considerable difference in their physical structures. Owing to the globules of pure iron being coated with

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cinder in the puddling furnace, the subsequent rolling and reworking, while expelling a large portion of this cinder, always leaves a trace of it behind which gives wrought iron the fiber. Steel having been produced in a liquid form, where the cinder all floated to the top and was removed, the metal is homogeneous, that is, without any grain or fiber. When subjected to many vibrations, or strains due to frequent expansion and contraction, wrought iron will generally yield gradually and give warning to the inspector, while steel is more liable to snap off suddenly. Wrought iron being composed of many fibers, the fibers can break one at a time without directly affecting its neighbor (like the strings in a rope), while a rupture once started in steel will extend more rapidly. Wrought iron will also resist corrosion and pitting longer than steel, no doubt due to higher resisting power of the enclosed cinder, which also causes the acid to deflect endwise, thus weakening its action by diffusing it over a larger area and preventing deep pitting. Stay bolts and boiler tubes for locomotives have proved more satisfactory when made of wrought iron than of steel. Thin sheets, tin plate, corrugated iron covering, wire fencing, pipes, oil well casings, etc., have also proved much more durable when made of wrought iron than when made of steel. On the other hand, in rails, tires, guns, armor plate, etc., steel has proved far superior to iron, owing to its greater strength and hardness; corrosion is also here of minor importance, owing to the rails, etc., generally being worn out long before corrosion has a chance to affect them seriously. When structural steel or iron is used for bridges, etc., it is necessary to protect the metal from serious corrosion by frequent and careful painting; in the skeletons of high office buildings and other skyscrapers, when completely covered with concrete, etc., so as to thoroughly exclude air or moisture, steel as well as iron will last indefinitely.

Where material is buried in the ground, or exposed to the weather without the careful protection of paint, or where moisture has access to it by other channels, as in the interior of pipes, for instance, wrought iron will outlast steel by a good margin.

Graphical Illustration of the Metallurgy of Iron.

The diagram Fig. 1 illustrates graphically the metallurgy of iron from the mine to the market and affords an interesting means of tracing out the different processes and showing the kind of steel or iron which each process produces. What has been said in the previous part of this chapter is, in a way, summarized in this diagram. Thus we see that the ore may, by the direct process, be changed at once to wrought iron in which form it is placed upon the market. The ore may go direct to the blast furnace or, if volatile substances are contained in the ore, it is first roasted, by which method these substances are removed and the ore made ready for the blast furnace. In the blast furnace the ore is changed to pig iron of various grades which may be placed directly upon the market or it may be then treated by any one of several processes. If treated by the Bessemer

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process the pig iron is changed to ingot iron, in which form it is placed upon the market. If treated by the open-hearth process it is also changed to ingot iron. If, however, the pig iron is sent to the foundry it is made into cast iron and placed upon the market in the form of castings. In the puddling furnace the pig iron is changed to marketable wrought iron or it may be treated by the cementation process, in which it is changed to blister steel, from which, by the crucible process, we obtain tool steel.

Uniform Nomenclature of Iron and Steel.

At the Brussels Congress of the International Association for Testing Materials held in September, 1906, a report was presented on "The Uniform Nomenclature of Iron and Steel." The following definitions of the most important forms of iron and steel are given:

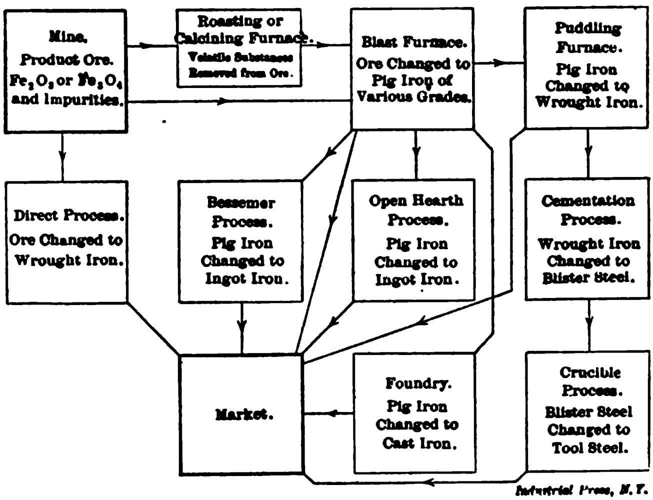

Fig. 1. Chart Illustrating the Metallurgy of Iron.

Alloy cast irons: Iron which owe their properties chiefly to the presence of an element other than carbon.

Alloy steels: Steels which owe their properties chiefly to the presence of an element other than carbon.

Basic pig iron: Pig iron containing so little silicon and sulphur that it is suited for easy conversion into steel by the basic open-hearth process (restricted to pig iron containing not more than 1.00 per cent of silicon).

Bessemer pig iron: Iron which contains so little phosphorus and sulphur that it can be used for conversion into steel by the original or acid Bessemer process (restricted to pig iron containing not more than 0.10 per cent of phosphorus).

Bessemer steel: Steel made by the Bessemer process, irrespective of carbon content.

Blister steel: Steel made by carburizing wrought iron by heating it in contact with carbonaceous matter.

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Cast iron: Iron containing so much carbon or its equivalent that it is not malleable at any temperature. The committee recommends drawing the line between cast iron and steel at 2.20 per cent carbon.

Cast steel: The same as crucible steel; obsolete, and to be avoided because confusing.

Converted steel: The same as blister steel.

Charcoal hearth cast iron: Cast iron which has had its silicon and usually its phosphorus removed in the charcoal hearth, but still contains so much carbon as to be distinctly cast iron.

Converted steel: The same as blister steel.

Crucible steel: Steel made by the crucible process, irrespective of carbon content.

Gray pig iron and gray cast iron: Pig iron and cast iron in the fracture of which the iron itself is nearly or quite concealed by graphite, so that the fracture has the gray color of graphite.

Malleable castings: Castings made from iron which when first made is in the condition of cast iron, and is made malleable by subsequent treatment without fusion.

Malleable iron: The same as wrought iron.

Malleable pig iron: An American trade name for the pig iron suitable for converting into malleable castings through the process of melting, treating when molten, casting in a brittle state, and then making malleable without remelting.

Open-hearth steel: Steel made by the open-hearth process irrespective of carbon content.

Pig iron: Cast iron which has been cast into pigs direct from the blast furnace.

Puddled iron: Wrought iron made by the puddling process.

Puddled steel: Steel made by the puddling process, and necessarily slag-bearing.

Refined cast iron: Cast iron which has had most of its silicon removed in the refinery furnace, but still contains so much carbon as to be distinctly cast iron.

Shear steel: Steel, usually in the form of bars, made from blister steel by shearing it into short lengths, piling, and welding these by rolling or hammering them at a welding heat. If this process of shearing, piling, etc., is repeated, the product is called "double shear steel."

Steel: Iron which is malleable at least in some one range of temperature, and in addition is either (*a*) cast into an initially malleable mass; or, (*b*) is capable of hardening greatly by sudden cooling; or, (*c*) is both so cast and so capable of hardening.

Steel castings: Unforged and unrolled castings made of Bessemer, open-hearth, crucible or any other steel.

Washed metal: Cast iron from which most of the silicon and phosphor have been removed by the Bell-Krupp process without removing much of the carbon, so that it still contains enough carbon to be cast iron.

Weld iron: The same as wrought iron; obsolete and needless.

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White pig iron and *white cast iron:* Pig iron and cast iron in the fracture of which little or no graphite is visible, so that their fracture is silvery and white.

Wrought iron: Slag-bearing, malleable iron, which does not harden materially when suddenly cooled.

CHAPTER II.

STEEL CASTINGS.

The present chapter consists of an abstract published in the June, 1903, issue of MACHINERY of an article which originally appeared in the journal of the *American Society of Naval Engineers*, February, 1903.

The raw materials that usually enter into the making of steel castings are steel scrap, pig iron and iron ore. The scrap consists of the crop ends of plates, shapes and forgings and the borings and turnings from the machine shop. The bulk of the furnace charge is scrap, the proportion of pig being about one-fifth at the beginning of a run —that is, immediately after a furnace has been rebuilt—and increasing up to nearly three-tenths at the end of the run, when the furnace lining and brick work generally are getting so slagged and burnt out as to require renewal. These proportions are for acid steel, basic steel using larger quantities of pig. The amount of iron in the ore is a secondary consideration, the ore being used chiefly for its oxygen, which comes into play in oxidizing the metalloids carbon, silicon, sulphur, and phosphorus. The proportions of ore required in a charge depends upon the character of the other ingredients; ordinarily in an acid furnace from one to two, or two and a half per cent would be used. There have been cases where scrap could not be procured, and the charge has been made up, of necessity, entirely of pig and ore, over three-fifths being pig. Hematite ore is the variety most used, and is obtained in large quantities in the Lake Superior region in this country and Canada; much of it is also imported from Cuba, Spain and elsewhere.

The amount of carbon combined with iron makes one difference between wrought iron and steel and between steel and cast iron. A second and equally important difference is the method of manufacture and the resulting properties and character. Wrought iron is soft and fibrous; cast iron is hard, crystalline and brittle; steel comes in anywhere between.

Basic steel is used in making castings, but not so generally as the acid product. Cheaper raw materials can be used in making basic steel, and phosphorus, the element chiefly objected to, can be nearly eliminated. It is more expensive than acid steel, however, and fewer

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heats per run of furnace can be turned out. With acid steel the number of heats will reach nearly three for each twenty-four hours, depending upon the size of furnace and character and quantity of work.

Open-hearth Furnaces.

The raw materials are melted down in a reverberatory furnace with gaseous fuel distilled from special bituminous coals in gas producers. Under each end of the furnace is a pair of regenerators—one for air, one for gas—which communicate with the furnace on one side and on the other with flues leading to the sources of supply of gas and air to the chimney. Reversing valves are located at the point where the flues meet, and about every twenty minutes, while the furnace is in operation, the valves are shifted and the currents of air and gas turned in the opposite direction. Each regenerator is nearly filled with fire brick built up in such an open checker-board manner that the air and gas find their way among and through them, absorbing heat from them on their way to the hearth, and, when spent, giving up heat to those in the opposite regenerators.

With this type of furnace a temperature of 4,000 degrees Fahrenheit can be attained, but as the supply of gas and air is at all times under control, the temperature can be made whatsoever may be desired. The process in the furnace will be acid or basic according to the character of the furnace lining—basic linings and additions being used in one case, and an acid lining, such as ordinary fire brick and fire clay, in the other.

Tests on Steel Castings.

Castings requiring annealing are placed in annealing furnaces, where they are gradually and uniformly heated up to temperatures depending upon the composition of the metal and varying between 1,200 and 1,600 degrees Fahrenheit, kept soaking at the maximum temperature for a time determined by their size, and allowed gradually to cool without exposure to the air. When cold the necessary test specimens are cut from them and machined accurately to required size. These specimens are then broken or bent in an approved testing machine according to the specifications prescribed. The Bureau of Steam Engineering of the Navy Department prescribes the following regarding the testing of steel castings: Sound test pieces shall be taken in sufficient number to thoroughly exhibit the character of the metal in the entire piece from each of the following castings, *viz.:* Shaft struts or brackets, main cylinder or valve-chest liners, main pistons and followers, eccentric, reversing and rocker shaft arms, crossheads, bedplates, columns of main engines and main air pumps, shaft couplings, and all large castings weighing over 200 pounds. All other castings may be tested by lots, as follows: A lot shall consist of all castings from the same heat, annealed in the same furnace charge. From each lot two or more tensile and one or more bending test pieces shall be taken, and the lot passed or rejected on the results shown by the tests. Large castings shall be suspended and hammered all

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over with a hammer weighing not less than 7½ pounds. No cracks, flaws, defect or weakness shall appear after such treatment.

The tensile strength of high-class steel castings varies from 65,000 pounds to 80,000 pounds per square inch. The percentage of elongation in two inches varies from 15 to 18 per cent, and the reduction of area from 20 to 25 per cent.

Uses of Steel Castings.

Steel castings are used for cylinder and valve-chest covers, for pistons, crosshead guides and slippers, bearing caps and shoes, eccentric sheaves and straps, rocker arms, thrust-bearing boxes and collars, bedplates and housings and other parts of main and auxiliary machinery; for boiler headers, manifolds, drum ends, dry pipes, manhole and handhole doors and other parts of boilers; anchors, anchor davits, hawse pipes, chocks, mooring and towing bitts, stems, stern posts, stern tubes, shaft brackets, manhole covers and other parts of ships' hulls, gun mounts, parts of dynamos and motors. Their use in ship construction and aboard ship is thus seen to be a large and important matter.

The cast-steel girders for the 16-inch U. S. army gun carriage measure 33 feet by 17 feet by 5 feet, and will weigh 100,000 pounds apiece; the carriage presents problems in transportation from the foundry to the arsenal at Watervliet on account of size and weight. A large casting turned out in the eastern part of Pennsylvania for a hydraulic forging press to be set up in the western part of the same State required about 320,000 pounds of metal from six open-hearth furnaces to pour it.

Reliability of Steel Castings.

A forged or rolled object is worked down from a billet which previously was hammered or pressed down from an ingot or part of an ingot, and during these stages of manufacture the metal is more or less thoroughly squeezed and pressed and caused to flow upon itself in various directions, and all parts, inside and out, receive some heat and power treatment, so that the impression grows in the minds of those who manipulate the forgings and of those who witness the manipulation that the accepted objects are free from weakening defects; the assurance of their trustworthiness is positive.

In the case of castings no such certainty or confidence is created. A steel casting may come out of the final cleaning process a thing of beauty, the physical and chemical tests may gladden the heart, the required machining may not show any flaws, yet the fear remains that below its surface somewhere a treacherous cavity or other weakness may some day show up—a day when most dependence is necessarily placed upon the casting, when most damage may result from its failure to do its duty.

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CHAPTER III.

STEEL HARDENING METALS.

In the 1904 issue of "Mineral Resources of the United States," published by the U. S. Geological Survey, a paper appeared written by Mr. Joseph Hyde Pratt, on the Steel Hardening Metals. An abstract of this was published in the May, 1905, issue of MACHINERY.

There are included under the head of steel-hardening metals, nickel and cobalt, chromium, tungsten, molybdenum, vanadium, titanium, and uranium, which are named in the order of the importance of their production and use for steel-hardening purposes.

The special steels resulting from these additions vary among themselves, having individual properties of tensile strength and elastic limit, of conductivity for heat and electricity, of magnetic capacity and of resistance to impact, whether as shell or as armor plate. It was only about twenty years ago that the first of these metals, nickel, began to be used to any extent for the purpose of hardening steel, but since their introduction their use for this purpose has continued to increase steadily. Experiments are still being carried on with some of these metals in order to determine their actual commercial value with regard to the qualities that they impart to steel. In the arts it is the ferro-alloy of these various metals that is first prepared and is then introduced in the required quantity into the manufactured steel, but this ferro-alloy is never added to the molten mass during the manufacture of the steel. All these metals give characteristic and distinct properties to steel, but in all cases the principal quality is the increase in the hardness and the toughness of the resulting steel. Some of the metals—as nickel, chromium and tungsten—are now entirely beyond the experimental stage and are well established in the commercial world as definite steel-hardening metals, and new uses are being constantly devised for the different steels, which are causing a constant increase in their production. Others, as molybdenum and vanadium, though they have been proved to given certain positive values to steel, have not been utilized to any large extent as yet in the manufacture of molybdenum or vanadium steel, partly on account of the high cost of the ores containing these metals. Titanium and uranium are still in the experimental stage, and, although a good deal has been written as to the value of titanium as an alloy with steel, there is at the present time very little if any of it used in the manufacture of a commercial steel.

Since the introduction of the electric furnace and the consequent methods that have been devised for reducing ores, it has become possible to obtain these ferro-alloys directly from the ores by reducing them in the electric furnace, and hence experiments have been conducted on a much larger scale than formerly.

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Manganese Steel.

Besides the use of ferromanganese for the chemical effect which it produces in the manufacture of steel in eliminating injurious substances, it is also used in the production of a special steel which possesses to a considerable degree combined hardness and toughness. Such steel contains from 0.8 to 1¼ per cent of carbon and about 12 per cent of manganese and is known as "Hadfield manganese steel." If only 1.5 per cent of manganese is added, the steel is very brittle, and the further addition increases this brittleness until the quantity of manganese has reached 4 to 5.5 per cent, when the steel can be pulverized under the hammer. With a further increase, however, of the quantity of manganese, the steel becomes ductile and very hard, reaching its maximum degree of these qualities with 12 per cent of manganese. The ductility of the steel is brought out by sudden cooling, a process the opposite of that used for carbon steel. These properties of manganese steel make it especially adapted for use in the manufacture of rock-crushing machinery, safes, and mine car wheels.

Nickel Steel.

Nickel finds its largest use in the manufacture of special nickel and nickel-chromium steels, and the use of these steels for various purposes in the arts is constantly increasing. The greatest quantity of nickel steel is used in the manufacture of armor plate, either with or without the addition of chromium. There is probably no armor or protective deck-plate made which does not contain from 3 up to 5 per cent of nickel. Nickel steel is also used for the manufacture of ammunition hoists, communication tubes, and turrets on battleships, and for gun shields and armor.

The properties of nickel steel or nickel-chromium steel that make it especially adapted for these purposes are its hardness and great tensile strength, combined with great ductility and a very high limit of elasticity. One of the strongest points in favor of a nickel steel armor plate is that when it is perforated by a projectile it does not crack. The Krupp steel, which represents in composition about the universal armor-plate steel, contains, approximately, 3.5 per cent of nickel, 1.5 per cent of chromium, and 0.25 per cent of carbon.

Another use for nickel steel that is gradually increasing is the manufacture of nickel steel rails. During 1903 there were over 11,000 tons of these rails manufactured, which were used by the Pennsylvania, the Baltimore & Ohio, the New York Central, the Bessemer & Lake Erie, the Erie, and the Chesapeake & Ohio railroads. These orders for nickel steel rails resulted from the comparison of nickel steel and carbon-steel rails in their resistance to wear during the five months' trial of the nickel steel rails that were used on the Horseshoe Curve of the Pennsylvania Railroad. The advantages that are claimed for the nickel steel rail are its increased resistance to abrasion and its higher elastic limit, which increases the value of the rail as a girder. On sharp curves it has been estimated that a nickel steel rail will outlast four ordinary rails.

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Nickel steel has also been largely adopted for forgings in large engines, particularly marine engines, and it is understood that this is now the standard material for this purpose in the United States navy. There is now a very great variety of these forgings and drop forgings, including the axles and certain other parts of automobiles, shafting and crank-shafts for government and merchant-marine engines and stationary engines, and locomotive forgings, the last including axles, connecting-rods, piston-rods, crank-pins, link-pins, and pedestal cap bolts, and for sea-water pumps.

Another important application that is being tried with nickel steel is in the manufacture of wire cables, and during the last years such cables have been made by the American Steel and Wire Co., but no comparison can as yet be made between them and the ordinary carbon-steel cables with respect to their wearing qualities.

In the manufacture of electrical apparatus nickel steel is beginning to be used in considerable quantity. The properties of this steel which make its especially valuable for such uses are, first, its high tensile strength and elastic limit, and, second, its high permeability at high inductions. Thus steel containing from 3 to 4 per cent of nickel has a lower permeability at low inductions than a steel without the nickel, but at the higher inductions the permeability is higher. A notable instance of the use of this material is in the field rings of the 5,000 H. P. generators built by the Westinghouse Electric and Manufacturing Co. for the Niagara Falls Power Co. These field rings require very high tensile strength and elastic limit, and in order to reduce the quantity it is desirable that they have high permeability at high inductions. This result was secured by using a nickel steel containing approximately 3.75 per cent of nickel. Steel containing approximately 25 per cent of nickel is nonmagnetic and has a very low resistance temperature coefficient. This property is occasionally of value where a nonmagnetic material of very high tensile strength is required. The high electrical resistance of nickel steel of this quality, together with its low temperature coefficient, makes it valuable for electrical resistance work where a small change in the resistance due to change in temperature is desirable. The main objection to using nickel steel for this purpose is the mechanical defects that are often found in wire that is drawn for this quality of nickel steel.

For rock drills and other rock-working machinery nickel steel is used in the manufacture of the forgings which are subjected to repeated and violent shocks. The nickel content of the steel used in these forgings is approximately 3 per cent, with about 0.40 per cent of carbon. The rock drills or bits are made for the most part of ordinary crucible cast steel which has been hardened and tempered. There is a field for investigation here in respect to the value of some of the special drills in the manufacture of rock-drill steels or bits. A nickel-chrome steel is now being made which is used to some extent in the manufacture of tools.

Nickel steel in the form of wire has been used quite extensively and for many purposes—for wet mines, torpedo defense netting, elec-

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tric lamp wire, umbrella wire, corset wire, etc.—where a non-corrosive wire is especially desired. When a low coefficient of expansion is desired—as in the manufacture of armored glass, in the mounting of lenses, mirrors, level tubes, balances for clocks, weighing machines, etc.—nickel steel gives good satisfaction. For special springs, both in the form of wire and flats, a high carbon nickel steel has been introduced to a considerable extent. Nickel steel is also being used in the manufacture of dies and shoes for stamp mills, for cutlery, table ware, harness mountings, etc.

Nickel steels containing from 25 to 30 per cent nickel are used abroad to a considerable extent for boiler and condenser tubes and are now being introduced into this country. The striking characteristic of these steels is their resistance to corrosion either by fresh, salt, or acid waters, by heat, and by superheated steam. The first commercial manufacture of high nickel steel tubes began in France in 1898, and was followed in Germany in 1899; but it was not until February, 1903, that these tubes were made in the United States. Since then, however, Mr. Albert Ladd Colby states:

"The difficulties of their manufacture have been so thoroughly overcome that the 30 per cent nickel-steel, seamless, cold-drawn marine boiler tubes, now a commercial proposition, are made in practically the same number of operations, and with but a slightly greater percentage of discard than customary in the manufacture of ordinary seamless tubes, and, furthermore, the finished 30 per cent nickel-steel tube will stand all the manipulating tests contained in the specifications of the Bureau of Steam Engineering, United States Navy Department, for the acceptance of the carbon-steel seamless cold-drawn marine boiler tubes now in use. In addition, the nickel-steel tubes have a much greater tensile strength."

Although the first cost of the nickel steel tubes for marine boilers is considerably in excess of the carbon-steel tubes, yet, on account of the longer life of the nickel-steel tubes, they are in the end cheaper than the others. At the present time 30 per cent nickel-steel tubes cost from 35 cents to 40 cents per pound, as compared with 12 cents to 15 cents per pound for the corresponding mild carbon-steel tubes. Thus their initial cost, when used in the boilers of torpedo-boat destroyers, is 2.13 times as great as the other kind and 2.43 times as great when used in the boilers of battleships, but the nickel steel tubes will last two-and-one-third times longer than those made of the carbon steel, and when finally taken from the boilers they can be sold not only for the market price of steel-tubing scrap, but also at an additional price of 20 cents per pound for their nickel content. Thus it is seen that 30 per cent nickel-steel boiler tubes are really more economical to purchase than carbon-steel boiler tubes.

In addition to marine boilers, high nickel-steel tubes can be used to advantage for stationary boilers, automatic boilers, and locomotive safe ends. It is the higher elastic limit of the 30 per cent nickel-steel boiler tubing that will prevent the leaks that are constantly being formed where the mild carbon-steel tube is used. The leaks are due

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to the expansion of the flue-sheets when heated, which compress the tubes at the points where they pass through the flue-sheets, and cause in the case of the mild carbon-steel tube a permanent deformation. This results in leakage and necessitates the frequent expanding of the tubes. In the high nickel-steel tubes this difficulty is overcome by their higher elastic limit. This deformation and the resulting leakage are especially true of locomotive boilers. For automobile tubular boilers a 23 to 25 per cent nickel-steel tubing is used, each coiled section being made from one long piece of nickel-steel tubing, which, by a special heat treatment, is enabled to withstand this bending without cracking.

Nickel-steel tubing containing 12 per cent of nickel has been used in France since 1898 in the manufacture of axles, brake beams, and carriage transoms for field artillery wagons, and the desired result in the reduction of weight has been obtained without loss of strength and stiffness of the wagons. A 5 per cent nickel-steel tubing has been used in the manufacture of bicycles since 1896.

Chromium Steel.

The largest use of chromium is in the manufacture of a ferro-chromium alloy which is used in the manufacture of chrome steel. In the manufacture of armor plate ferro-chrome plays a very important part, and, although it is sometimes used alone for giving toughness and hardness to the armor plate, it is more commonly used in combination with the nickel, making a nickel-chromium-steel armor plate. Other uses of chrome steel are in connection with five-ply welded chrome steel and iron plates for burglar-proof vaults, safes, etc., and for castings that are to be subjected to unusually severe service, such as battery shoes and dies, wearing plates for stone crushers, etc. A higher chromium steel which is free from manganese will resist oxidation and the corrosive action of steam, fire, water, etc., to a considerable extent, and these properties make it valuable in the manufacture of boiler tubes. Chromium steel is also used to some extent as a tool steel, but for high-speed tools it is being largely replaced by tungsten steel, which is especially adapted to this purpose.

The percentage of chromium that is used in the chromium steels varies from 2.5 to about 5 per cent and the carbon from 0.8 to 2 per cent. The hardness, toughness and stiffness which are obtained in chromium steel are very essential qualities, and are what make this steel especially beneficial for the manufacture of armor-piercing projectiles as well as of armor plate. For projectiles, chromium steel has thus far given better satisfaction than any of the other special steels, and is practically the only steel that is used for this purpose. The value of chromium steel for this purpose is well brought out by Mr. R. A. Hadfield, manager of the Hecla Works, Sheffield, England, who states that a 6-inch armor-piercing shot made by this firm was fired at a 9-inch compound plate, which it perforated unbroken. It was then fired again from the same gun and perforated a second plate of the same thickness, the shot still remaining unbroken.

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Tungsten Steel.

Tungsten steel is used to some extent more generally abroad than in the United States, in the manufacture of armor plate and armor-piercing projectiles. For this purpose it is used in combination either with nickel or chromium, or with both of these metals. The use, however, for which tungsten steel is best adapted is in the manufacture of high-speed tools and magnet steels. The property that tungsten imparts to the steel is that of hardening in the air after forging and without recourse to the usual methods of tempering, such as immersion in oil, water, or some special solution. For high-speed tools tungsten steel is especially adapted, as it retains its hardness and cutting edge even at the temperature developed in the use of these high-speed tools. The value of tungsten steel for permanent magnets is on account of it retaining comparatively strong magnetism and of the permanence of this magnetism in the steel. This property makes the tungsten steel particularly desirable in instrument work where the calibration of the instrument depends upon the permanence of the magnet used. For compass needles, tungsten steel has been used with entire satisfaction.

Molybdenum.

The use of molybdenum steel continues to increase, and hence there is an increasing demand for the ores of this metal. The main use of ferromolybdenum is in the manufacture of tool steel. The properties which molybdenum gives to steel are very similar to those given by tungsten, the main difference being that it requires a smaller quantity of molybdenum than of tungsten to acquire the same results. Ferromolybdenum is produced, like ferrotungsten, by reducing it from the ore in an electric furnace. There are now two molybdenum-nickel alloys being produced, one of which contains 75 per cent molybdenum and 25 per cent nickel, and the other 50 per cent molybdenum and 50 per cent nickel. Besides these constituents the alloy contains from 2 to 2.5 per cent iron, 1 to 1.5 per cent carbon, and 0.25 to 0.50 per cent silicon. The molybdenum steel which is made from these alloys is recommended for large cranks and propeller-shaft forgings, for large guns, rifle barrels, and for wiring and for boiler plates. The molybdenum increases the elongation of steel very considerably, and for wire drawing such an increase at a comparatively small cost is important.

Vanadium Steel.

On account of the extremely high price and scarcity of vanadium ores, the metal has thus far been employed very little in the manufacture of ferrovanadium for use in the production of vanadium steel. It is claimed by many that the beneficial properties imparted to steel by vanadium exceed those of any of the other steel hardening metals. These are exaggerated statements, but it may be found that smaller quantities of vanadium will give in some cases the same results that are obtained by comparatively large quantities of the other metals. One property claimed for vanadium steel is that it acquires its maxi-

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mum of hardness not by sudden cooling, but by annealing at a temperature of from 1,300 to 1,470 degrees F. This property would be particularly advantageous for high-speed tool steel and for points of projectiles.

Titanium.

The actual commercial value of titanium as a steel-hardening metal has not been thoroughly demonstrated. Experiments have shown that from 0.5 to 3 per cent of titanium increases the transverse strength and the tensile strength of steel to a very remarkable degree. Until the development of the electric furnace it was practically impossible to produce either titanium or an alloy of iron and titanium, but since the introduction of this furnace, ferrotitanium can be produced directly from the ores. It is to the manufacture of a special cast iron that ferrotitanium seems to be especially adapted. The titanium in the iron gives greater density to the metal, greatly increases its transverse strength, and gives a harder chill or wearing quality to a wheel made from such an iron. For the manufacture of car wheels it would seem that the titanium iron would be especially useful.

CHAPTER IV.

DEVELOPMENT AND USE OF HIGH SPEED STEEL.

The following discussion on high-speed steel and tools made from this material was published in MACHINERY in the December, 1904, issue, and is an abstract of a paper read by Mr. J. M. Gledhill before the Iron and Steel Institute, of Great Britain, October, 1904.

The high-speed steels of the present day are combinations of iron and carbon with: (1) Tungsten and chromium, (2) Molybdenum and chromium, (3) Tungsten, molybdenum and chromium.

Influence of Carbon.

A number of tool steels were made by the Armstrong Whitworth Co. with the carbon percentage varying from 0.4 per cent to 2.2 per cent, and the method of hardening was to heat the steel to the highest possible temperature without destroying the cutting edge, and then rapidly cooling in a strong air blast. By this simple method of hardening it was found that the greatest cutting efficiency is obtained where the carbon ranges from 0.4 per cent to 0.9 per cent, and such steels are comparatively tough. Higher percentages are not desirable because great difficulty is experienced in forging the steels, and the tools are inferior. With increasing carbon contents the steel is also very brittle, and has a tendency to break with unequal and intermittent cutting.

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Influence of Chromium.

Having thus found the best carbon content to range from 0.4 per cent to 0.9 per cent, the next experiments were made to ascertain the influence of chromium varying from 1.0 per cent to 6.0 per cent. Steels containing a low percentage are very tough, and perform excellent work on the softer varieties of steel and cast-iron, but when tried on harder materials the results obtained were not so efficient. With an increased content of chromium the nature of the steel becomes much harder, and greater cutting efficiency is obtained on hard materials. It was observed that with an increase of chromium there must be a decrease in carbon to obtain the best results for such percentage of chromium.

Mention may here be made of an interesting experiment to ascertain what effect would be produced in high-speed steel by substituting vanadium for chromium. The amount of vanadium present was 2.0 per cent. The steel readily forged, worked very tough, and was hardened by heating to a white heat and cooling in an air blast. This tool when tried on medium steel stood well, but not better than the steel with the much cheaper element of chromium in it.

Influence of Tungsten.

This important element is contained in by far the greater number of the present high-speed steels in use. A number of experiments were made with the tungsten content ranging from 9.0 per cent to 27.0 per cent. From 9.0 per cent to 16.0 per cent the nature of the steel becomes very brittle, but at the same time the cutting efficiency is greatly increased, and about 16.0 per cent appeared to be the limit, as no better results were obtained by increasing the tungsten beyond this figure. Between 18.0 per cent and 27.0 per cent it was found that the nature of the steel altered somewhat, and instead of being brittle, it became softer and tougher, and whilst such tools have the property of cutting very cleanly, they do not stand up so well.

Influence of Molybdenum.

The influence of this element at the present time is under investigation, and the experiments with it have so far produced excellent results; it has been found that where a large percentage of tungsten is necessary to make a high-speed steel, a considerably less percentage of molybdenum will suffice. A peculiarity of these molybdenum steels is that in order to obtain the greatest efficiency they do not require such a high temperature in hardening as do the tungsten steels, and if the temperature is increased above 1,800 degrees F. the tools are inferior, and the life shortened.

Influence of Tungsten with Molybdenum.

It was found that the presence of from 0.5 per cent to 3.0 per cent molybdenum in a high tungsten steel slightly increased the cutting efficiency, but the advantage gained is altogether out of proportion to the cost of the added molybdenum.

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Influence of Silicon.

A number of high-speed steels were made with silicon content varying from a trace up to 4.0 per cent. Silicon sensibly hardens such steels, and the cutting efficiency on hard materials is increased by additions up to 3.0 per cent. By increasing the silicon above 3.0 per cent, however, the cutting efficiency begins to decline. Various experiments were made with other metals as alloys, but the results obtained were not sufficiently good by comparison with the above to call for comment.

An analysis of one of the best qualities of high-speed steels produced by the author's firm (Armstrong, Whitworth Co.) is as follows: "A.W." Steel.—Carbon, 0.55 per cent; Chromium, 3.5 per cent; Tungsten, 13.5 per cent.

What may be said to determine a high-speed steel, as compared to an ordinary tool steel, is its capability of withstanding the higher temperatures produced by the greatly increased friction between the tool and the work due to the rapid cutting. An ordinary carbon steel containing, say, 1.20 per cent carbon when heated slightly above the critical point and rapidly cooled by quenching in water becomes intensely hard. Such a steel gradually loses this intense hardness as the temperature of friction reaches, say, 500 degrees F. The lower the temperature is maintained the longer will be the life of the tool, so that the cutting speed is very limited. With rapid cutting steels the temperature of friction may be greatly extended, even up to 1,100 degrees F. or 1,200 degrees F., and it has been proved by experience that the higher the temperature for hardening is raised above the critical point and then rapidly cooled, the higher will be the temperature of friction that the tool can withstand before sensibly losing its hardness. The high degree of heating (almost to the melting point, in fact) which is necessary for hardening high-speed steel, forms an interesting study in thermal treatment and is indeed a curious paradox, quite inverting all theory and practice previously existing. In the case of hardening ordinary carbon steels very rapid cooling is absolutely necessary, but with high-speed steels the rate of cooling may take a considerably longer period, the intensity of hardness being increased with the quicker rate of cooling.

Heat Treatment of High-Speed Steel.

Turning now to some points in the heat treatment of high-speed steel, one of the most important is the process of thoroughly annealing it after working into bars. Accurate annealing is of much value in bringing the steel to a state of molecular uniformity, thereby removing internal strains that may have arisen, due to casting and tilting, and at the same time annealing renders the steel sufficiently soft to enable it to be machined into any desired form for turning tools, milling cutters, drills, taps, threading dies, etc. The annealing of high-speed steel is best carried out in muffle furnaces designed for heating by radiation only, a temperature of 1,400 degrees F. being maintained from twelve to eighteen hours according to the section of the bars of steel dealt with. Further advantage also results from

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careful annealing by minimizing risks of cracking when the steel has to be reheated for hardening. In cases of intricately-shaped milling tools having sharp square bottom recesses, fine edges, or delicate projections, and on which unequal expansion and contraction are liable to operate suddenly, annealing has a very beneficial effect toward reducing cracking to a minimum. Increased ductility is also imparted by annealing, and this is especially requisite in tools that have to encounter sudden shocks due to intermittent cutting, such as planing and slotting tools, or others suddenly meeting projections or irregularities on the work operated on.

In preparing high-speed steel ready for use the process may be divided principally into three stages: forging, hardening, and grinding. It is, of course, very desirable that high-speed steel should be capable of attaining its maximum efficiency and yet only require treatment of the simplest kind, so that an ordinarily skilled workman may easily deal with it, otherwise the preparation of tools becomes an expensive and costly matter, and materially reduces the advantages resulting from its use. Fortunately, the treatment of high-speed steel as produced by leading firms is of the simplest; simpler in fact than of ordinary carbon steels or of the old self-hardening steels. Great care has to be exercised in the heating of the latter steels, for if either are heated above a blood-red heat, say 1,600 degrees F., the danger of impairing their efficiency by burning is considerable; whereas with the high-speed steel, heating may be carried to a much higher temperature, even to the melting point, it being practically impossible to injure it by burning. The steel may be raised to a yellow heat for forging, say 1,850 degrees F., at which temperature it is soft and easily worked into any desired form, the forging proceeding until the temperature lowers to a good red heat, say 1,500 degrees F., when work on it should cease and the steel be reheated.

In heating a bar of high-speed steel preparatory to forging (which heating is best done in a clear coke fire) it is essential that the bar be heated thoroughly and uniformly, so as to ensure that the heat has penetrated to the center of the bar, for if the bar be not uniformly heated, leaving the center comparatively cold and stiff, while the outside is hot, the steel will not draw or spread out equally, and cracking will probably result. A wise rule in heating is to "hasten slowly."

It is not advisable to break pieces from the bar while cold, the effect of so doing tending to induce fine end cracks to develop which ultimately may extend and give trouble; but the pieces should be cut off while the bar is hot, then be reheated as before and forged to the shape required, after which the tool should be laid in a dry place until cold.

The temperature for hardening high-speed steel varies somewhat according to the class of tool being dealt with. When hardening turning, planing, or slotting tools, and others of similar class, the point or nose of tool only should be gradually raised to a white melting heat, though not necessarily melted; but no harm is done even if the point of the tool becomes to a greater or less extent fused or melted.

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The tool should then be immediately placed in an air blast and cooled down, after which it only requires grinding and is then ready for use. Another method, which may be described, of preparing the tools is as follows: Forge the tools as before, and when quite cold grind to shape on a *dry* stone or *dry* emery wheel, an operation which may be done with the tool fixed in a rest and fed against the stone or emery wheel by a screw, no harm resulting from any heat developed at this stage. The tool then requires heating to a white heat, but just short of melting, and afterward complete cooling in the air blast. This method of first roughly grinding to shape also lends itself to cooling the tools in oil, which is specially efficient where the retention of a sharp edge is a desideratum, as in finishing tools, capstans and automatic lathe

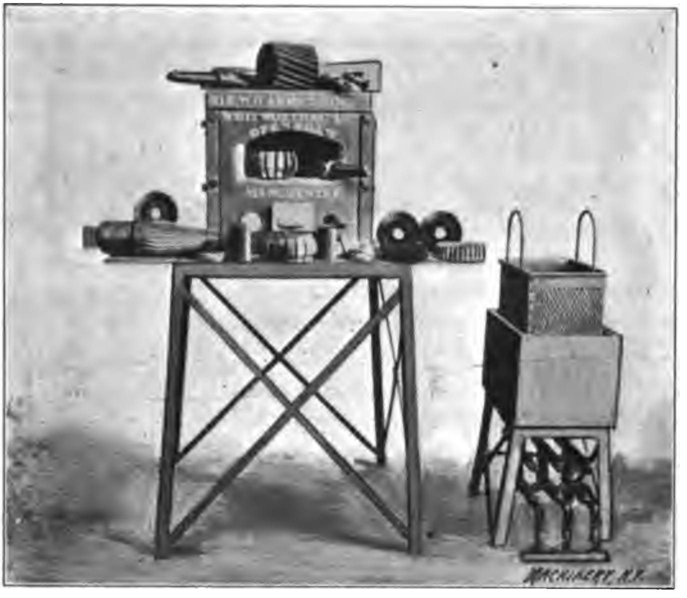

Fig. 2. Muffle Furnace for Hardening Milling Cutters made of High-Speed Steel; also Tank and Dipping Cage for Tempering them in Oil.

tools, brass-workers' tools, etc. In hardening where oil cooling is used, the tools should be first raised to a white heat, but without melting, and then cooled down either by air blast or in the open to a bright red heat, say 1,700 degrees F., when they should be instantly plunged into a bath of rape or whale oil, or a mixture of both.

Referring to the question of grinding tools, nothing has yet been found so good for high-speed steels as the wet sandstone, and the tools ground thereon by hand pressure, but where it is desired to use emery wheels it is better to roughly grind the tools to shape on a dry emery wheel or dry stone *before* hardening. By so doing the tools require but little grinding after hardening, and only slight frictional heating occurs, but not sufficient to draw the temper in any way, and thus the

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cutting efficiency is not impaired. When the tools are ground on a wet emery wheel and undue pressure is applied, the heat generated by the great friction between the tool and the emery wheel causes the steel to become hot, and water playing on the steel while in this heated condition tends to produce cracking.

With regard to the hardening and tempering of specially formed tools of high-speed steel, such as milling and gear cutters, twist drills, taps, threading dies, reamers, and other tools that do not permit of being ground to shape after hardening, and where any melting or fusing of the cutting edges must be prevented, the method of hardening is as follows:

A specially arranged muffle furnace heated either by gas or oil is employed, and consists of two chambers lined with fire-clay, the gas and air entering through a series of burners at the back of the furnace, and so under control that a temperature up to 2,200 degrees F. may be steadily maintained in the lower chamber, while the upper chamber is kept at a much lower temperature. Before placing the cutters in the

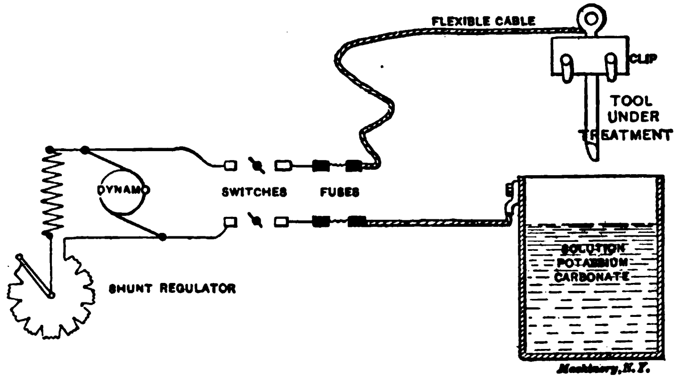

Fig. 3. Apparatus for Hardening Tools Electrically in a Bath of Potassium Carbonate.

furnace it is advisable to fill up the hole and keyways with common fire-clay to protect them. The cutters are first placed upon the top of the furnace until they are warmed through, after which they are placed in the upper chamber, Fig. 2, and thoroughly and uniformly heated to a temperature of about 1,500 degrees F., or, say, a medium red heat, when they are transferred into the lower chamber and allowed to remain therein until the cutter attains the same heat as the furnace itself, *viz.*, about 2,200 degrees F. and the cutting edges become a bright yellow heat, having an appearance of a glazed or greasy surface. The cutter should then be withdrawn while the edges are sharp and uninjured, and revolved before an air blast until the red heat has passed away, and then while the cutter is still warm—that is, *just* permitting of its being handled—it should be plunged into a bath of tallow at about 200 deg. F. and the temperature of the tallow bath then raised to about 520 degrees F., on the attainment of which the cutter should be immediately withdrawn and plunged in cold oil.

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Of course there are various other ways of tempering, a good method being by means of a specially arranged gas-and-air stove into which the articles to be tempered are placed, and the stove then heated up to a temperature of from 500 degrees F. to 600 degrees F., when the gas is shut off and the furnace with its contents allowed to slowly cool down.

Heating Steel by Electrical Means.

Another method of heating tools is by electrical means, by which very regular and rapid heating is obtained, and where electric current is available, the system of electric heating is quick, reliable, and economical. A brief description of this kind of heating may be of interest. One method adopted of electrically heating the points of tools, and the arrangement of apparatus, is shown in Fig. 3. It consists of a cast-iron tank, of suitable dimensions, containing a strong solution of potassium carbonate, together with a dynamo, the positive cable from which is connected to the metal clip holding the tool to be heated, while the negative cable is connected direct on the tank. The tool to be hardened is held in a suitable clip to ensure good contact.

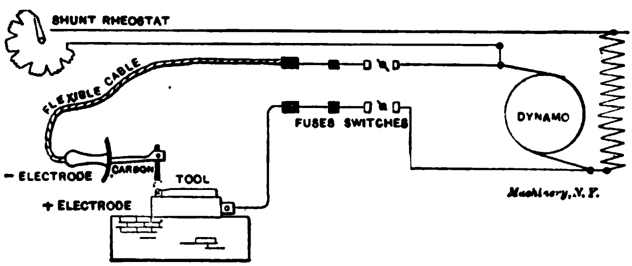

Fig. 4. Apparatus for Heating Tools by the Electric Arc.

Proceeding to harden the tool the action is as follows: The current is first switched on, and then the tool is gently lowered into the solution to such a depth as is required to harden it. The act of dipping the tool into the alkaline solution completes the electric circuit and at once sets up intense heat on the immersed part. When it is seen that the tool is sufficiently heated the current is instantly switched off, and the solution then serves to rapidly chill and harden the point of the tool, so that no air blast is necessary.

Another method of heating the point of tools is by means of the electric arc, the heating effect of which is also very rapid in its action. The general arrangement and form of the apparatus here employed is as illustrated in Fig. 4. The tool under treatment and the positive electrode are placed on a bed of non-conducting and non-combustible material and the arc started gradually at a low voltage and steadily increased as required, by controlling the shunt rheostat, care being taken not to obtain too great a heat and so fuse the end of the tool. The source of power in this case is a motor generator consisting of a continuous-current shunt-wound motor at 220 volts, coupled to a continuous-current shunt-wound dynamo at from 50 to 150 volts. Arcs

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from 10 to 1,000 amperes are then easily produced and simply and safely controlled by means of the shunt rheostat.

Tempering.

Electricity is also a very efficient and accurate means of tempering such forms of tools as milling, gear, hobbing and other similar cutters, also large hollow taps, hollow reamers, and all other hollow tools made of high-speed steel, where it is required to have the outside or cutting portion hard, and the interior soft and tenacious, so as to be in the best condition to resist the great stresses put upon the tool by the resistance of the metal being cut, and which stresses tend to cause disruption of the cutter if the hardening extends too deep. By means of the apparatus illustrated in Fig. 5 this tempering or softening of the interior can be perfectly and quickly effected, thus bringing the cutter into the best possible condition to perform rapid and heavy work.

Tempering of hollow cutters, etc., is sometimes carried out by the insertion of a heated rod within the cutter and so drawing the temper,

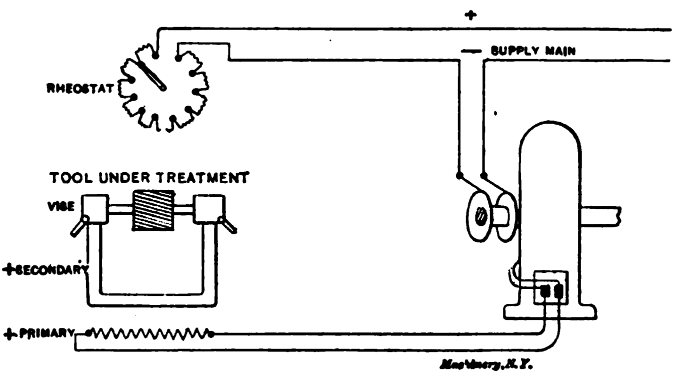

Fig. 5. Apparatus for Tempering Milling Cutters Electrically.

but this is not entirely satisfactory, or scientific, and is liable to induce cracking by too sudden heat application, and further because of the difficulty of maintaining the necessary heat and temperature required, and afterward gradually lowering the heat until the proper degree of temper has been obtained. In electrical tempering these difficulties are overcome, as the rod is placed inside the cutter quite cold, and the electric current gradually and steadily heats up the rod until the correct temperature is reached. Then it can be held at such temperature as long as is necessary, and the current can be gradually reduced until the articles operated on are cold again, and consequently the risk of cracking by too sudden expansion and contraction is reduced very greatly. The apparatus used is very simple, as will be seen by reference to Fig. 5. It consists of a continuous-current shunt-wound motor directly coupled to a single-phase alternating-current dynamo of the revolving field type giving 100 amperes at 350 volts, 50 cycles per second, the exciting current being taken from the works supply main. The power from the alternator is by means of a stepdown transformer,

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reduced to current at a pressure of 2 volts, the secondary coil of the transformer consisting of a single turn of copper of heavy cross-section, the extremities of which are attached to heavy copper bars carrying the connecting vises holding the mandrel upon which the cutter to be tempered is placed. The secondary induced current, therefore, passes through a single turn coil, through the copper bars and vises and mandrel. Although the resistance of the complete circuit is very low, still, owing to the comparatively high specific resistance of the iron mandrel, the thermal effect of the current is used up in heating the mandrel which gradually attains the required temperature, slowly imparting its heat to the tool under treatment until the shade of the

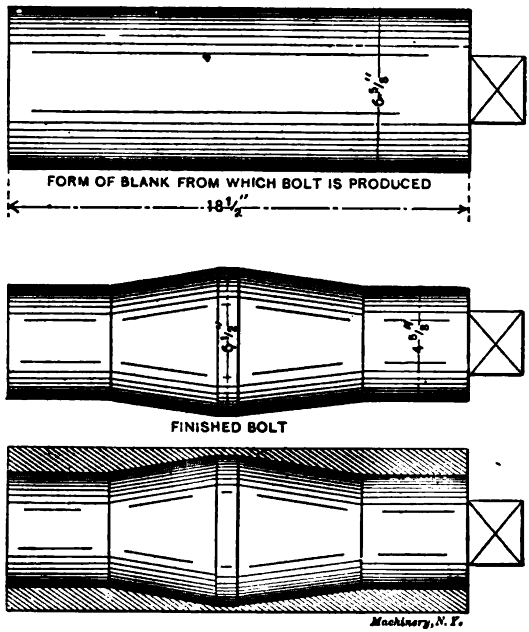

Fig. 6. Armor Bolt Turned at a Cutting Speed of 160 Feet per minute; Feed, 1-32 inch; Mean Depth of Cut, 3-4 inch.

oxide on the tool satisfies the operator. The method adopted to regulate the heat of the mandrel is by varying the excitation current of the alternator by means of the rheostat. An extremely fine variation and perfect heat control is easily possible by this arrangement.

Some Results of the Use of High-Speed Steel.

That great economy is effected by the use of high-speed steel is beyond all doubt, from whichever point of view the question is looked at; for it is not only rapidity of cutting that counts, but the output of machines is correspondingly increased, so that a greater production is obtained from a given installation than was possible when cutting at low speeds with the old tool steel, and the work is naturally produced

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at a correspondingly lower cost, and of course it follows from this that that in laying down new plant and machines the introduction and use of high-speed steel would have considerable influence in reducing expenditure on capital account. It has also been proved that high-speed cutting is economical from a mechanical standpoint and that a given horse-power will remove a greater quantity of metal at a high speed than at a low speed, for although more power is naturally required to take off metal at a high than at a low speed (by reason of the increased

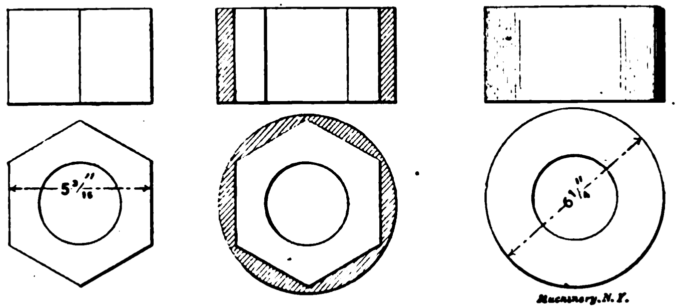

Fig. 7. Examples of Work Milled with a Cutting Speed of 150 Feet per Minute; Maximum Depth of Cut, 1 1-2 inch.

work done) the increase of that power is by no means in proportion to the large extra amount of work done by the high-speed cutting, for the frictional and other losses do not increase in anything like the same ratio as a high-cutting speed is to a low-cutting speed. A brief example of this may be given in which the power absorbed in the lathe was accurately measured, electrically.

Cutting on hard steel, with 3/16-inch depth of cut, 1/16-inch feed and speed of cutting 17 feet per minute, a power of 5.16 horse-power was

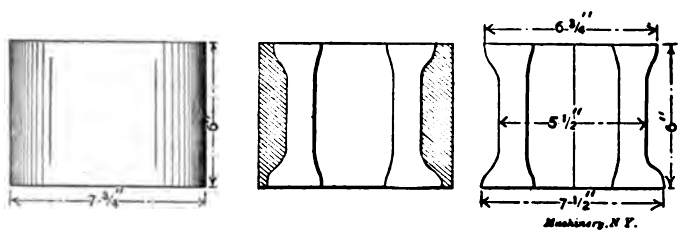

Fig. 8. Sleeves for Armor Bolts Turned with a Cutting Speed of 160 Feet per Minute; Feed, 1-32 inch; Maximum Depth of Cut, 1 1-8 inch.

absorbed, and increasing the cutting speed to 42 feet per minute, the depth of cut and feed being the same, there was a saving in power of 19 per cent for the work being done. Another experiment with depth of cut 3/8 inch and traverse 1/16 inch compared with 1/16 inch traverse and 3/16 inch depth of cut, showed a saving in power of as much as 28 per cent, and still proceeding with a view of increasing the weight of metal removed in a given time the feed was doubled (other conditions being the same) and a still further saving of power resulted.

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In a word, as in the majority of things, so it is with rapid cutting, the more quickly work can be produced the cheaper the cost of production will be.

Again, as regards economy, there is not only a saving effected on the actual machine work, but since the advent of high-speed cutting it is now possible, in many instances, to produce finished articles from plain rolled bars, instead of following the old practice of first making expensive forgings and afterward finishing them in the machine. By this practice not only is the entire cost of forging abolished, but the machining on the rolled bar can be carried out much quicker and cheaper in suitably arranged machines, quicker even than the machining of a forging can be done.

A remarkable sample of the gain resulting from the use of high-

Fig. 9. Making Hexagon Nuts from Rolled Bars with Cutters made from High-Speed Steel. Ninety Nuts Produced in a Day of Ten Hours.

speed cutting from rolled bars is illustrated in Fig. 6, the articles in this case being securing-bolts, made by the author's firm, for armor plates. Formerly where forgings were first made and then machined with ordinary self-hardening steel, a production of eight bolts per day of ten hours was usual. With the introduction of high-speed steel, forty similar bolts from the rolled bar are now produced in the same time, thus giving an advantage of five to one in favor of quick cutting, and also in addition abolishing the cost of first rough forging the bolt to form; in fact, the cost of forging one bolt alone amounted to more than the present cost of producing to required form twelve such bolts by high-speed machining. The cutting speed at which these bolts are turned is 160 feet per minute, the depth of cut and feed being respectively 3/4 inch and 1/32 inch, the weight of metal removed from each

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bolt being 62 pounds, or 2,480 pounds in a day of ten hours, the tool being only ground once during such period of work; from such an example as this it will be at once apparent what an enormous saving in plant and costs results. On the same principle the sleeves (see Fig. 8) of these bolts are produced from bars, sixty being made in one day of ten hours, this being even a greater saving on the old system than the bolt example shows.

Equally remarkable results are obtained by operating on stock bars with high-speed milling cutters, and one example among many, may be cited, which is shown in Fig. 7. Here hexagon nuts for 3⅝ inches diameter bolts are made from rolled bars, the cutting speed of milling being 150 feet per minute, giving a production of ninety nuts per day, against thirty formerly. More than ninety nuts could have been produced had the machine been more powerful.

CHAPTER V.

HARDENING STEEL.

Every shop has one or more men who are considered authorities on hardening. In many cases the man is really an expert, is careful, and uses good judgment in heating the steel and in quenching in the bath: and if the piece is of sufficient size, he is sure to take the strains out by reheating directly after taking from the bath. In some cases, however, the success of one operation is measured by the failure of others. Then if the work passes through the fiery ordeal with enough of it left intact to do the work it is considered a *successful operation;* if not, the fault *must* be in the steel. A manufacturing concern once changed the brand of tool steel they were using three times in less than a year, because the man doing the hardening reported adversely on each make, after attempting to harden it. The article furnished was from three of the leading makers of tool steel. After receiving repeated complaints in regard to the man's inability to harden the steel successfully, one of the makers advised the manufacturers to let some expert in hardening try the steel. Some milling machine cutters were made from each brand of the rejected article and sent to the steel makers. They all came back hard enough, without cracks, proving that the trouble was not in the steel.

Kind of Steel Used.

An expensive steel is not necessarily a satisfactory investment, and a "cheap" brand may be *very expensive.* It is necessary to understand just what is needed in a steel for a given purpose. Some makers have different grades of steel for different purposes—one for taps and similar tools, another for milling machine cutters, etc.—while others put out a steel that is very satisfactory for most purposes. Each has a good

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argument in favor of his particular method of manufacture. In some shops it is thought advisable to use a grade of steel adapted to each individual class of tool; while in other shops, where detail is not followed as closely, this would cause no end of confusion. That part of the subject must be left to the judgment of the individual shop. But the treatment of the steel in the fire and the bath, in order to be successful, must be along certain lines. The successful hardener is he who finds out what particular quality is needed in the piece he is to harden; whether extreme hardness, toughness, elasticity, or a combination of two of these qualities. Then he must know the method to use in order to produce the desired result. The shape of the piece, the nature of the steel, the use to be made of the article, must all be taken into consideration. He must also be governed somewhat by the kind of fire he is to use.

Heating the Steel.

Some brands of steel will not stand, without injury, the range of heat that others will; some require more heat than others in order to harden at all. When hardening, no steel should be heated hotter than is necessary to produce the desired result. With some brands that give off their surface carbon very readily it is not advisable to heat them in an open fire, exposed to the action of the blast and outside air, as the products of combustion extract the carbon to such an extent that the surface will be soft even when the interior is extremely hard. While this might not materially affect a tool that is to be ground, it would spoil a tap, a formed cutter, or similar article, whose outside surface could not be removed. In hardening anything of this nature in an open fire, it should be placed in a piece of tube or some receptacle, so that the fire cannot come in contact with it while heating. There are a number of gas and gasoline hardening furnaces made which have a muffler to receive the work. The fire circulates around the muffler but does not come in contact with the steel. Very excellent results may be obtained when one of these furnaces is used. The front can be closed by means of a door, thus keeping all outside air away from the work. It will be found a great advantage if several large holes are drilled in the door, these being covered with isinglass, to enable the operator to see the work without opening the door.

Taking carbon from the steel is not the only injury done to a high grade of steel when heated in an ordinary blacksmith's forge by a careless operator. Most inexperienced men are apt to use a small fire, particularly if they find one ready built. It may be mostly burned out, but the operator will not care to take the time to get fresh coal, and get the fire to the proper heat; so he puts on the blast and endeavors to heat the work. After a time the piece has all kinds of heats, ranging from a low red to a white heat. The operator thinks it *averages* well, and dips it in the bath. If it comes out in one piece he is fortunate.

Heating in a small fire is dangerous business, as the work not only comes in contact with the surrounding air, but with the cold air from the blast, which will cause minute surface cracks, making the steel

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look as though full of hairs. It will also fill the steel with "strains," causing ends of projections to crack and drop off in the bath.

If obliged to use the blacksmith's forge, use plenty of good charcoal. Make a large, high fire if the piece to be hardened is of any size; keep it up well from the blast inlet, using only blast enough to keep the fire lively, and bring the piece to the proper heat, burying it well in the fire to keep it from the air. The lowest heat that will give the desired result should be used. This varies in different makes of steel, and must also be varied somewhat according to size and shape of the work. The teeth of a milling machine cutter will harden at a lower heat than a solid piece of the same size made from the same bar. Most steelmakers in their instructions advise to harden at a low cherry red. To the average man this is a very uncertain degree; his cherries may be of a different hue from some other fellow's. Most of the leading brands of tool steel in small sizes, however, give the best results when hardened just after the black has disappeared from the center of the piece, provided we were heating slowly so as to get a uniform heat. In no case should steel be dipped when there is a trace of black in it.

The higher a piece of steel is heated—to a certain degree—the harder it will be; but if it is heated higher than to this degree the grain is opened, making it coarse and brittle, and it will be very liable to flake off under strain. For this reason, in the case of cutting tools, it is best to harden at as low a heat as possible. If the work gets too hot, yet not to a point where it is burned, it is always best to allow it to cool until the red has entirely disappeared, then reheat to the proper degree and harden, and the grain will be fine; but if allowed to cool to the proper hardening heat and dipped, it would be as coarse as if hardened at the high heat, and would also be very liable to crack.

Annealing.

In hardening, a great deal depends on the annealing. It is as necessary to understand how to anneal properly as it is to know how to harden right. As generally understood, the purpose of annealing is to soften the steel, which is all right, so far as the party is concerned who works it to shape; but its relation to hardening is another matter. It removes all strains in the steel, incident to rolling and hammering in the steel mill and forging in the blacksmith shop. Experience teaches the hardener that it is necessary to anneal any odd-shaped piece or one with a hole or impression in it, after it has been blocked out somewhere near to shape, a hole somewhat smaller than finished size being drilled in it, and all surface scale being removed. The most satisfactory method to pursue is to pack in an iron box with granulated charcoal, not allowing any of the pieces to come within one inch of the box at any point. This box should then be placed in the furnace and kept at a bright red heat for a length of time dependent on the size of the steel. Pieces one inch in diameter should be kept at a red heat for one hour after the box is heated through; larger pieces should be kept hot correspondingly longer, allowing the work to cool off as slowly as possible. An annealing heat should be higher than a heat for hardening the same piece. The proper heat for annealing, in order that

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all strains may be overcome, should be nearly as high as for forging the same piece; in other words, the work should be heated to a bright red and kept so long enough to overcome any strain or tension liable to manifest itself when the piece is hardened. Tool steel for annealing should never be packed in cast-iron chips or dust, as this extracts the carbon to such an extent that there will be trouble when hardening is attempted. Packing too near the walls of the annealing box will have the same effect to a less extent, but will be more troublesome, as the carbon will be extracted from the surfaces nearest the box, and not affected anywhere else, making the hardening very uneven.

If not situated so that this method can be used, very satisfactory results may be obtained by heating in a large charcoal fire to a uniform forging heat. Put two or three inches of ashes in the bottom of an iron box; on this place a piece of soft wood board, put the work on it, cover with another piece of board, and fill the box with ashes. The boards will char and smolder, keeping the work hot for a long time. Some blacksmiths use a box of cold ashes, while others use cold lime; either way is liable to chill the piece, making it harder than if allowed to cool in the air, and if either material is used it should be hot to get good results. Excellent results may be obtained by heating in a muffler oven, as a very uniform heat of any degree may thus be obtained. It can be run any length of time, but when a piece is heated through in this way it takes a long time to cool.

Hardening Baths.

Hardening a piece of steel is generally accomplished by heating to a low red, and plunging in some cooling bath. As so much depends on the bath, it is quite necessary to understand the effects of the use of the different kinds. The one most commonly used is clear cold water, though many use salt and water or brine. For hardening small articles that must be extremely hard, the following will be found very satisfactory: One pound citric acid crystals dissolved in one gallon of water. For very thin articles a bath of oil is necessary. For hardening springs, sperm oil is very satisfactory; when hardening cutting tools, raw linseed oil is excellent. There are hundreds of formulas for hardening compounds, some of which are excellent for certain classes of work. Some hardening solutions are poisonous, and are dangerous to have around; but for ordinary work the ones mentioned are sufficient.

Many successful hardeners use water that has been boiled, claiming better results from its use than from fresh water. Small odd-shaped pieces are not so liable to crack nor to harden unevenly when the water is slightly warmed.

Examples of Hardening

We will now consider a few pieces of work to be hardened by the open-fire method. If we have a muffler furnace, so much the better, as with this it is easier to get certain results; but with care very satisfactory work can be done when the blacksmith forge is used. If it is a small tap, reamer, counterbore, or similar article we are to harden,

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it is best to heat it in a tube, bring it to a low red, and plunge it in slightly warm water, or in the citric acid solution. If it is a hollow mill, with a hole running part way through it, we should dip it in the bath with the hole up, or the steam will keep the water from entering the hole, leaving the inside walls soft. The steam would also have a tendency to crack the piece; but with the hole up when dipping, by working the piece up and down well in the bath, the steam can escape, and the water can get at the work. Much bother may be saved the hardener if attention is paid to the steam likely to be generated, and some way provided to prevent its keeping the water from the work. Brine does not steam as readily as clear water; neither do the different acid solutions used by many.

In hardening a milling machine cutter, it is best to have a large high fire, to bury the cutter well in the fire, and to use only blast enough to bring the work to the required heat, which should be uniform throughout. If the piece has not been annealed after drilling a hole through it, remove it from the fire when red hot, then allow it to cool off slowly until the red has entirely disappeared, when it can be again placed in the fire and slowly brought to the required heat; it is then plunged in a bath of tepid water or brine and worked around well until it stops "singing." At this point it should be removed and instantly plunged in an oil bath, and left there until it is cool, when the strain should be removed by holding it over the fire until it is warm enough to snap when touched with the moistened finger. It can then be laid aside, and the temper drawn at leisure. In hardening punch press dies we can treat them the same; if there are any screw holes for stripper or guide screws they should be plugged with fire clay or graphite.

Metal slitting saws can be hardened nicely between iron plates whose surfaces are kept oiled. The saws should be heated in such a manner that the fire does not come in contact with them. It is best to heat on a flat plate, as the tendency to warp is much less than if laid on an uneven surface. When the saw is properly heated, place it on the lower oiled plate, placing the other one on it as quickly as possible: hold the upper plate down hard until the saw is cool. If there are many such pieces to harden, a fixture can be made so that one man can handle the saws and fixture alone; otherwise it requires two operators.

If there is no other means of drawing temper, the work may be brightened and drawn by color; but, if possible, do the drawing to temper in a kettle or crucible of oil over the fire, gaging the heat by a thermometer. Much more satisfactory results can be obtained by this latter method; and if very many pieces are to be done, it will be found much cheaper. A very light yellow is 430 degrees; a straw color is 460 degrees; a brown yellow, 500 degrees; a light purple, 530 degrees. A milling machine cutter for ordinary work should be drawn to 430 degrees; a punch press die to 500 degrees; the punch to 530 degrees, and metal-slitting saws to 530 degrees.

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CHAPTER VI.

CASE-HARDENING.

The present chapter contains an abstract published in MACHINERY, August, 1905, of a paper read by Mr. David Flather before the Cycle Engineers' Institute, Birmingham, England.

The term "case-hardening" naturally implies the hardening of the skin of an article, and in order to fully understand the process and its object we must briefly consider the facts and laws upon which it is founded. Carbon has a very great affinity for iron and combines with it at all temperatures above faint red heat. Advantage is taken of this fact in the production of steel by cementation—in fact, the process of case-hardening is in reality incomplete cementation followed by water or oil hardening.

For many purposes in machine work we require articles to have a perfectly hard surface and yet be of such a nature that there is no chance of their breaking in use. In many instances this result can be obtained with high-class crucible steel, but for axles, cups, cones, and many similar parts, it is extremely difficult to obtain perfect hardness combined with great resistance to torsional, shearing, or bursting strains. For such purposes nothing can meet these requirements so fully as articles which have been case-hardened. The greatest risks in the employment of all steel often occur during its treatment by the consumer, and whether it be the finest cast steel or only common Bessemer, it is of first importance that it should be carefully and properly treated with a view to the work it has to do.

Both iron and mild steel have been employed as material for case-hardening; but this is the "steel age," and iron has long passed its day. The steel employed should be prepared, selected, and controlled from the beginning with the object of suiting it to its requirements. There are, of course, many points relating to its composition and treatment by the producer which can only be gained by long experience and by study of the requirements. Suffice it to say that the steel used should be low in carbon and capable of absorbing more carbon with great uniformity when heated under proper conditions; it should contain a minimum of deleterious impurities, and be perfectly sound and free from mechanical faults or weaknesses caused by overheating during the manufacturing processes.

The Case-Hardening Furnace and Muffles.

The furnace should be so constructed as to be capable of being raised to a full orange heat (1,830 degrees F.), and maintained at that heat with great regularity. It should be so constructed that neither the fuel nor the direct flame can come in contact with the charge. The flames should uniformly impinge on the sides and roof of the muffle

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in such a manner as to raise them to a high temperature, thus heating the contents of the muffle by radiated and not by direct heat. A furnace designed on this principle not only gives the best result but is also most economical in the matter of fuel. The muffle chamber and flues must, of course, be constructed of firebrick, and the doors should fit closely and also be lined with firebrick. It is important that there should be a small peep-hole in the door, with a cover plate; a hole 1½ inch diameter is quite large enough. This latter is really a most important detail, as it provides against the need of opening the doors in order to judge the heat, and is indeed the most accurate means of estimating the temperature by the eye. The furnace must be fitted with a reliable damper plate or other effectual means of controlling the draft.

Fig. 10 shows a furnace which may be useful as a guide for the erection. The upper chamber in this furnace is not necessary for case-hardening, but it may be found useful to have such a chamber and employ it for annealing small articles while case-hardening is being done. This will add only very slightly to the amount of fuel used.

Hardening pots are made in both cast and wrought iron, the former being cheaper in first cost, but the latter bear reheating so many times that they are cheaper in the end. The pots should not be of too large dimensions, or there is great risk of articles in the middle of a charge not being carbonized to a sufficient depth. No pot should be above 18 by 12 by 11 inches for such parts as cycle axles, pedal pins, and the like; while for small articles like cups, cones, etc., 12 by 10 by 8 inches is large enough. The pots should each have a plate-lid fitting closely inside.

The carbonizers in general use at the present day are animal charcoal, bones, and one or two other compositions sold under various names, consisting of mixtures of carbonaceous matter and certain cyanides or nitrates. For very slight hardening, cyanides alone are still found very useful, but no great depth of casing is ever attempted with these. Theoretically, the perfect carbonizer should be a simple and pure form of carbon, and good charred leather gives the most certain and satisfactory results. Care should be taken to avoid poorly charred leather or that made from old boots, belting, etc.

Clay.

As clay must be used for a luting around the pot lid, and is also frequently used for stopping off portions to be left soft, it is important to see that a good clay is used, and that it is free from grease. Clay contaminated with grease in any way will cause irregularity in the product.

Reheating Muffles.

As all case-hardened articles have to be reheated before quenching, it is important that a suitable furnace should be employed for the purpose. It is not advisable that the reheating should be done in the case-hardening muffle, unless it is run specially for the purpose and

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at a lower heat. If possible a small gas muffle should be used for reheating, and indeed for all hardening work. A properly-constructed gas muffle can be regulated with great exactness, and this is very important in all hardening.

Packing the Muffles.

The carbonizer having been thoroughly dried and reduced to a fine powder, a layer of not less than 1½ inch in depth is placed in the

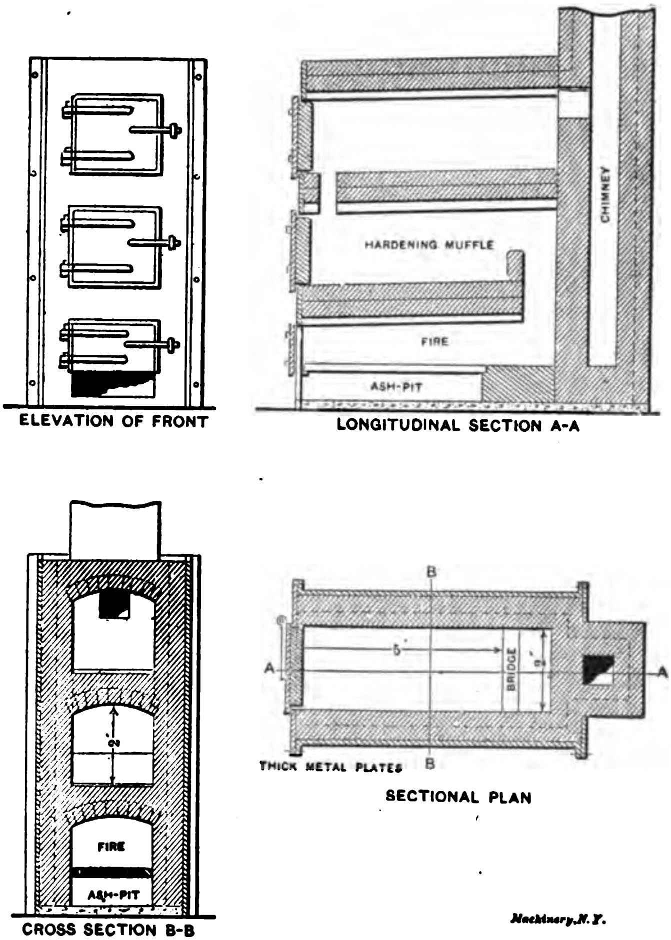

Fig. 10. Plan and Elevation of Flather Case-hardening Furnace.

hardening pot and well pressed down. Upon this are placed the articles to be hardened. Care must be taken to leave sufficient space all around each piece to prevent its touching the others or the walls of the pot; a space of 1½ inch should be sufficient. Another layer of carbonizer is then put in and well pressed down, taking care not to dis-

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place any of the articles already packed, continuing until the pot is nearly full, and then finishing off with another layer of 1½ inch at the top. The object in view must be to make the contents of the pot as compact as possible, consistent with a sufficiency of carbonizer in contact with the articles. The more solidly a pot is packed the more complete is the exclusion of air. The lid is then put on, and the joint all around well luted with clay. By the time the proper number of pots have been filled, the furnace must have been raised steadily to the full working heat.

Furnace Heat.

The proper heat for case-hardening is about 1,800 degrees F., or a full orange heat and this should be maintained with great regularity throughout the operation. The length of time occupied in carbonizing is regulated by the depth of casing required, and indirectly by the dimensions of the article. At the close of the carbonizing period the pot is withdrawn from the furnace and placed in a dry place, where it is allowed to become quite cold. It is then opened, the articles taken out and brushed over to remove all adhering matter. If the pot has been properly packed, and luted up, the articles should be quite white, or at least have only a slight film or bloom of a deep blue color; the denser and more inclined to redness is the surface, the more imperfect has been the packing and sealing of the pot.

Reheating and Hardening.

The carbonized articles are now placed in a muffle furnace and steadily raised to a good cherry red (1,470 degrees F.), and then quenched in cold or tepid water or oil, according to the purpose of the articles required. They should remain in the cooling liquid until they are quite cold right through the body of the metal, thus completing the process.

Although the proper temperature for case-hardening is about 1,830 degrees F., this temperature may be modified to suit the purpose in view. The absorption of the carbon commences when the steel reaches a low cherry-red heat (1,300 degrees F.); it begins, of course, at the outer surface and gradually spreads until the whole of the steel is carbonized. The length of time this requires depends upon the thickness of the metal being treated. The percentage of carbon absorbed is governed by the temperature, and although the increase of carbon is not in uniform proportion to the rising temperature throughout, it is perhaps sufficient for our present purpose to note that at 1,300 degrees F., iron, if completely saturated, can contain no more than about 0.50 per cent carbon; at 1,650 degrees F., about 1.5 per cent carbon; and at 2,000 degrees F., about 2.5 per cent. These results, however, are only obtainable when the whole section of the iron has received all the carbon it is capable of absorbing at the given temperature, and is therefore in a state of equilibrium. From this it will be seen that if the process is stopped before the action is complete, the central parts of the iron must contain less carbon than the outside, and upon this fact the process of casehardening is founded.

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If we take two pieces of ⅝ inch diameter round mild steel, and heat one of them with a carbonizer at a cherry-red heat, and the other at a bright orange heat, for six hours, the first will be cased to a depth of about 1-32 inch, and the other to a depth of nearly 1-16 inch, while the amount of carbon taken up will be about 0.50 and 0.80 per cent respectively. So that, so far as regards the hardness of the skin, the piece carbonized at the higher temperature gives the best result. From this we learn that a temperature of 1,830 degrees F. will give us sufficient hardness of case.

We have next to find which temperature has the least harmful effect on the mild steel core, and this can best be found by heating pieces of the mild steel at varying temperatures at and above the selected one for the same length of time, using lime or other inert substance in the pot instead of a carbonizing material, and afterward reheating and quenching in water. Suppose, for example, we take three pieces, heating at 1,830, 2,370 and 2,730 degrees F., or full orange, white and bright white respectively. We shall find that those at 2,370 and 2,730 degrees break very short and have lost nearly all their original tenacity, while that at 1,830 degrees appears tougher and altogether stronger than before.

Having arrived at a knowledge of the right temperature, it remains now to inquire as to the length of time requisite to yield a sufficient depth of case. At a full orange heat a bracket cup of ordinary dimensions should in two hours be hardened 1-32 inch deep, and a bracket axle 11-16 inch diameter in 6 hours would have a case 1-16 inch deep. From this it will be seen that the speed of penetration is not in exact proportion to the time of heating.

Results of Hardening Without Reheating.

We now arrive at that part of the process where a most important improvement has been made—*i. e.*, the final hardening by quenching in water. It formerly was customary at the end of the carbonizing period to open the pot and fling the contents headlong into a tank of cold water. Here and there some of the more careful workers took each article separately, but direct from the pot, and plunged it into water. These latter obtained better results, but even they had a great deal of trouble in the way of breakages and want of regular hardness. Finding that axles taken singly from the pot and quenched were better than those quenched in bulk, and that if allowed to cool down to cherry red they were better still, an application of the old rule to harden on a rising heat led to the now established principle of allowing the pot and its contents to become quite cold, afterward reheating to cherry red and quenching with water. By this means we obtain a case of great hardness with a very tough core—that is, of course, provided a suitable steel is employed.

To understand the reason of this improved method of working we must remember that the exterior of the steel is now of about 0.80 per cent carbon, and that steel of all kinds raised to and maintained at the high temperature employed for case-hardening will, unless subjected

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to mechanical work, show evidence of overheating, being very brittle and liable to easy fracture; and though quenched in water, and consequently hardened, the metal has little or no cohesion and readily wears away. Steel so hardened breaks with a very coarse crystalline fracture, in which the limits of the case are badly defined. It is known that when steel is gradually heated there is a certain point at which a great molecular change takes place, and that perfect hardness can only be obtained by quenching at this critical point. If quenching takes place below the critical temperature, the steel is not sufficiently hard; if above, though full hardness may be obtained, strength and tenacity are lost in part or completely, according as the critical temperature is exceeded by much or by little. This critical point lies between 1,380 and 1,470 degrees F., or cherry-red color heat. It may be asked why it is not sufficient, when taking the article out of the pot, to allow it to cool down to cherry red and then quench it. To this the answer is that the high temperature has already created a coarsely crystalline condition in the steel, and that until it has become quite cold and has again been heated up to the critical temperature, a suitable molecular condition cannot be obtained. When steel is cooled, whether slowly or not, it bears in its structure a condition representative of the highest heat it was last subjected to. From this it will be quite clear that in case-hardening, as in all other methods of hardening, steel must be quenched on a rising heat.

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CHAPTER VII.

THE BRINELL METHOD OF TESTING THE HARDNESS OF METALS.

The method of testing the hardness of metals devised by Mr. J. A. Brinell has received very favorable attention from metallurgists in this, as well as in other countries. In 1900 Mr. Brinell, then chief engineer and technical manager of the Fagersta Iron and Steel Works in Sweden, first made public his method of testing the hardness of iron and steel, by submitting it to the Society of Swedish Engineers in Stockholm. At the meeting of the *Congrès International des Méthodes d'Essai des Matériaux de Construction* in Paris the same year the method attracted general attention, and its merits were duly acknowledged by awarding the inventor with a personal *Grand Prix* at the Paris Exposition. The method was first described in the English language by Mr. Axel Wahlberg in a paper before the Iron and Steel Institute in 1901. Since then, the practical value of this method has been amply substantiated on various occasions by means of comprehensive tests and investigations undertaken by several distinguished scientists in different countries. In working out his method, Brinell kept in view the necessity of taking into account the requirements that the method must be trustworthy, must be easy to learn and apply, and capable of being used on almost any piece of metal, and particularly, to be used on metal without in any way being destructive to the sample.

Principle of Method for Testing Hardness of Metals.

The Brinell method consists in partly forcing a hardened steel ball into the sample to be tested so as to effect a slight spherical impression, the dimensions of which will then serve as a basis for ascertaining the hardness of the metal. The diameter of the impression is measured, and the spherical area of the concavity calculated. On dividing the amount of pressure required in kilogrammes for effecting the impression by the area of the impression in square millimeters an expression for the hardness of the material tested is obtained, this expression or number being called the *hardness numeral.* In order to render the results thus obtained by different tests directly comparable with one another, there has been adopted a common standard as well with regard to the size of ball as to the amount of loading. The standard diameter of the ball is 10 millimeters (0.3937 inch) and the pressure 3,000 kilogrammes (6,614 pounds) in the case of iron and steel, while in the case of softer metals a pressure of 500 kilogrammes (1,102 pounds) is used. Any variation either in the size of the ball or the amount of loading will be apt to occasion more or less confusion without there being any advantage to compensate for such inconvenience. Besides, making any comparisons between results thus obtained

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in a different manner would be more or less troublesome, and complicated calculations would be required.

The diameter of the impression is measured by means of a microscope of suitable construction, and the hardness numeral may be obtained without calculation directly from the table given herewith, worked out for the standard diameter of ball and pressures mentioned. The formulas employed in the calculation of this table are as follows:

$$y = 2\pi r\,(r - \sqrt{r^2 - R^2}) \qquad (1)$$

$$H = \frac{K}{y} \qquad (2)$$

in which formulas

r = radius of ball in millimeters,
R = radius of depression in millimeters,
y = superficial area of depression in square millimeters,
K = pressure on ball in kilogrammes,
H = hardness numeral.

Suppose, for instance, that the radius of the ball equals 5 millimeters (0.1968 inch), and that the test is undertaken on a piece of steel, the pressure consequently applied being 3,000 kilogrammes (6,614 pounds). Assuming that we found the radius of the depression equal to 2 millimeters (0.07874 inch) by measurement, we have:

$$2\,\pi \times 5\,(5 - \sqrt{25 - 4}) = 13.13 = y,$$

and

$$\frac{3{,}000}{13.13} = 228 = H,$$

which as we see agrees with the figure given in our table for a 4 millimeters diameter of impression.

Relation Between Hardness of Materials and Ultimate Strength.

It has been pointed out by Mr. Brinell himself that this method of testing hardness of metals offers a most ready and convenient means of ascertaining within close limits the ultimate strength of iron and steel. This, in fact, is one of the most interesting and important results of this method of measuring hardness. In order to determine the ultimate strength of iron and steel, it is only necessary to establish a constant coefficient determined by experiments which serves as a factor by which the hardness numerals are multiplied, the product being the ultimate strength. Rather comprehensive experiments were undertaken with a considerable number of specimens of annealed material obtained from various steel works for the purpose of establishing the coefficient by the present director of the Office for Testing Materials of the Royal Technical Institution at Stockholm. The results obtained were as follows:

For hardness numerals below 175, when the impression is effected transversely to the rolling direction, the coefficient equals 0.362; when the impression is effected in the rolling direction, the coefficient equals 0.354.

For hardness numerals above 175. when the impression is effected

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TABLE OF HARDNESS NUMERALS.

Steel ball of 10 millimeters diameter.

Diameter of Impression, mm.	Hardness Numeral. Pressure, kg.		Diameter of Impression, mm.	Hardness Numeral. Pressure, kg.		Diameter of Impression, mm.	Hardness Numeral. Pressure, kg.		Diameter of Impression, mm.	Hardness Numeral. Pressure, kg.		Diameter of Impression, mm.	Hardness Numeral. Pressure, kg.	
	3000	500		3000	500		3000	500		3000	500		3000	500
2.00	946	158	3.00	418	70	4.00	228	38	5.00	143	23.8	6.00	95	15.9
2.05	898	150	3.05	402	67	4.05	223	37	5.05	140	23.3	6.05	94	15.6
2.10	857	143	3.10	387	65	4.10	217	36	5.10	137	22.8	6.10	92	15.3
2.15	817	136	3.15	375	63	4.15	212	35	5.15	134	22.3	6.15	90	15.1
2.20	782	130	3.20	364	61	4.20	207	34.5	5.20	131	21.8	6.20	89	14.8
2.25	744	124	3.25	351	59	4.25	202	33.6	5.25	128	21.5	6.25	87	14.5
2.30	713	119	3.30	340	57	4.30	196	32.6	5.30	126	21	6.30	86	14.3
2.35	683	114	3.35	332	55	4.35	192	32	5.35	124	20.6	6.35	84	14
2.40	652	109	3.40	321	54	4.40	187	31.2	5.40	121	20.1	6.40	82	13.8
2.45	627	105	3.45	311	52	4.45	183	30.4	5.45	118	19.7	6.45	81	13.5
2.50	600	100	3.50	302	50	4.50	179	29.7	5.50	116	19.3	6.50	80	13.3
2.55	578	96	3.55	293	49	4.55	174	29.1	5.55	114	19	6.55	79	13.1
2.60	555	93	3.60	286	48	4.60	170	28.4	5.60	112	18.6	6.60	77	12.8
2.65	532	89	3.65	277	46	4.65	166	27.8	5.65	109	18.2	6.65	76	12.6
2.70	512	86	3.70	269	45	4.70	163	27.2	5.70	107	17.8	6.70	74	12.4
2.75	495	83	3.75	262	44	4.75	159	26.5	5.75	105	17.5	6.75	73	12.2
2.80	477	80	3.80	255	43	4.80	156	25.9	5.80	103	17.2	6.80	71.5	11.9
2.85	460	77	3.85	248	41	4.85	153	25.4	5.85	101	16.9	6.85	70	11.7
2.90	444	74	3.90	241	40	4.90	149	24.9	5.90	99	16.6	6.90	69	11.5
2.95	430	73	3.95	235	39	4.95	146	24.4	5.95	97	16.2	6.95	68	11.3

transversely to the rolling direction, the coefficient equals 0.344; when the impression is effected in the rolling direction, the coefficient equals 0.324.

If the hardness numerals are multiplied by these coefficients, the result obtained will be the ultimate tensile strength of the material in kilogrammes per square millimeter. It is evident that coefficients can easily be worked out so that if the hardness numerals be multiplied by these the strength could be obtained in pounds per square inch.

Suppose, for instance, that a test of annealed steel by means of the Brinell ball test gave an impression of a diameter of 4.6 millimeters. Then the hardness numeral, according to our table, would be 170, and the ultimate tensile strength consequently $0.362 \times 170 = 61.5$

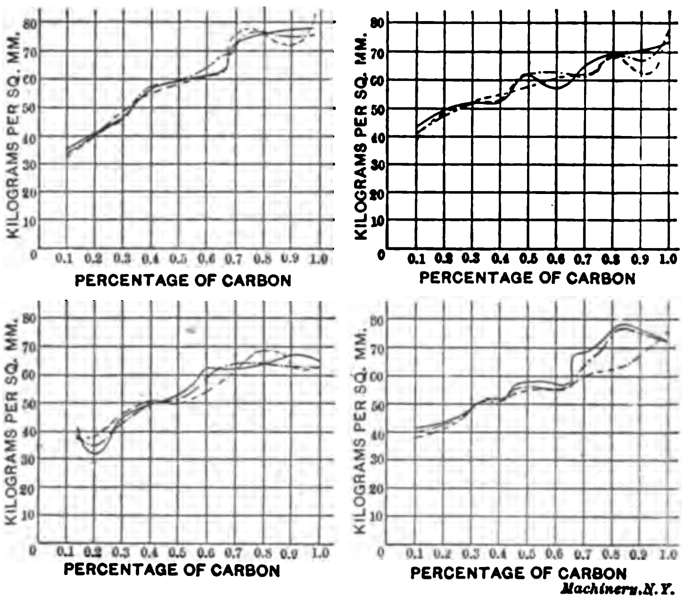

Fig. 11. Diagrams Showing Relation Between Results Obtained by Various Methods for Ascertaining the Ultimate Strength of Materials.

kilogrammes per square millimeter, provided the impression was effected transversely to the rolling direction.

In Fig. 11 are shown a number of diagrams which indicate the results obtained at the tests undertaken to ascertain the coefficients given. In these diagrams the full heavy line indicates the tensile strength of the material, as calculated from the ball tests in the rolling direction. The dotted lines indicate the strength as calculated from the ball tests in a transversal direction, and the "dash-dotted" lines show the actual tensile strength of the material as ascertained by ordinary methods for ascertaining this value. It is interesting to note how closely the three curves agree with one another, and considering

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the general uncertainty and variation met with when testing the same kind of material for tensile strength by the ordinary methods, it is safe to say that the ball test method comes nearly as close to the actual results as does any other method used. Especially within the range of the lower rates of carbon, or up to 0.5 per cent, or in other words, within the range of all ordinary construction materials, the coincidents are, in fact, so very nearly perfect as to be amply sufficient to satisfy all practical requirements.

In the case of any steel, whether it be annealed or not, that has been submitted to some further treatment of any other kind than annealing, such as cold working, etc., or in the case of any special steel, there would be other coefficients needed which would then also be ascertained by experiments. The same coefficient, however, will hold true for the same kind of material having been subjected to the same treatment. Thus, the ball testing method for strength is equally satisfactory, and far more convenient, in all cases where the rupture test would be applied. One of the greatest advantages of the Brinell method is that in the case of a large number of objects being required to be tested, each one of the objects can be tested without demolition, and without the trouble of preparing test bars.

Application of the Brinell Ball Test Method.

Summarizing what has been said in the previous discussion, and adding some other important points, we may state the various uses for which the Brinell ball test method may be applied, outside of the direct test of the hardness of construction materials and the calculation from this test of the ultimate strength of the materials, as follows:

1. Determining the carbon content in iron and steel.
2. Examining various manufactured goods and objects, such as rails, tires, projectiles, armor plates, guns, gun barrels, structural materials, etc., without damage to the object tested.
3. Ascertaining the quality of the material in finished pieces and fragments of machinery even in such cases when no specimen bars are obtainable for undertaking ordinary tensile tests.
4. Ascertaining the effects of annealing and hardening of steel.
5. Ascertaining the homogeneity of hardening in any manufactured articles of hardened steel.
6. Ascertaining the hardening power of various quenching liquids, and the influence of temperature of such liquids on the hardening results.
7. Ascertaining the effect of cold working on various materials.

Machines used for Testing the Hardness of Metals by the Brinell Method.

The method of applying the Brinell ball test was at first only possible in such establishments where a tensile testing machine was installed. As these machines are rather expensive, the use of the ball test method was limited. For this reason a Swedish firm, Aktiebolaget Alpha, Stockholm, Sweden, has designed and placed on the market a

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compact machine specially intended for making hardness tests. This machine, as shown in Fig. 12, consists of a hydraulic press acting downward, the lower part of the piston being fitted with a 10-millimeter steel ball *k* by means of which the impression is to be effected in the surface of the specimen or object to be tested. This object is placed on the support *s* which is vertically adjustable by means of the

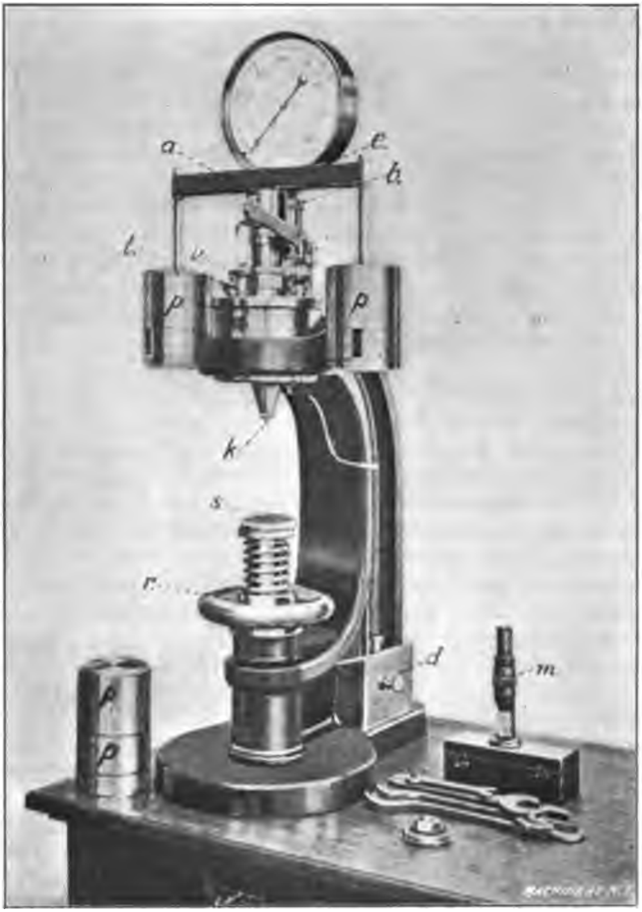

Fig. 12. Aktiebolaget Alpha's Machine for Testing Hardness of Materials.

hand-wheel *r*, while at the same time it can be inclined sideways when this is needed on account of the irregular shape of the part tested. The whole apparatus is solidly mounted on a cast iron stand. The pressure is effected by means of a small hand pump, and the amount of pressure can be read off directly in kilogrammes on the pressure gage mounted at the top of the machine.

In order to insure against any eventual non-working of the mano-

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meter, this machine is fitted with a special contrivance purporting to control in a most infallible manner the indications of that apparatus, while at the same time serving to prevent any excess of pressure beyond the exact amount needed according to the case. This controlling apparatus consists of a smaller cylinder, *a*, directly communicating with the press-cylinder. On being loaded with weights corresponding to the amount of pressure required, the piston in this cylinder will

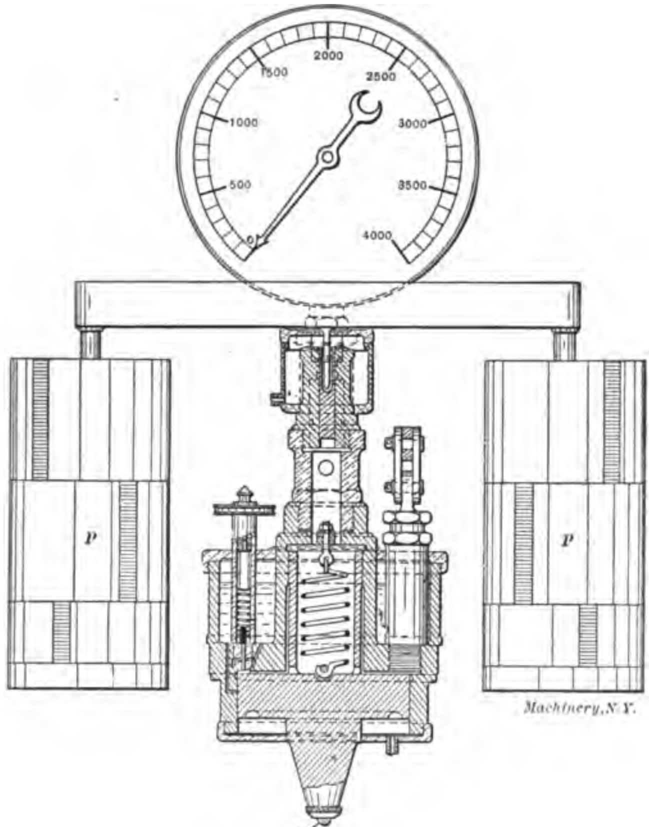

Fig. 13. Section of Press Cylinder of Machine in Fig. 12.

be pushed upward by the pressure effected within the press-cylinder at the very moment when the requisite testing pressure is attained. Owing to this additional device, there can thus be no question whatever of any mistake or any errors as to the testing results, that might eventually be due to the manometer getting out of order.

Method of Performing the Ball Test.

The test specimen must be perfectly plane on the very spot where the impression is to be made. It is then placed on the support *s*, Fig.

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12, which, as mentioned, is adjusted by means of the hand-wheel *r* so as to come into contact with the ball *k*. A few slow strokes of the hand pump will then cause the pressure needed to force the ball downward, and a slight impression will be obtained in the object tested, but as soon as the requisite amount of pressure has been attained, the upper piston is pushed with the controlling apparatus upward, as previously described. On testing specimens of iron and steel, the pressure is maintained on the specimen for 15 seconds, but in the case of softer materials for at least half a minute. After the elapse of this time, the pressure is released, and the contact between the ball and the sample will cease. A spiral spring fitted within the cylinder, and being just of sufficient strength to overcome the weight of the press piston, pulls the same upward into its former position, while forcing the liquid back into its cistern. The diameter of the impression effected by the ball is then measured by the microscope *m*, which is specially constructed for this purpose, the results obtained by this measurement being exact within 0.05 millimeter (0.002 inch). Fig. 13 shows a cross-section through the cylinder and piston part of the machine. Another type of machine is designed for special tests in which very high pressures are required. The ball in this machine is 19 millimeters (0.748 inch) in diameter, and the pressures employed vary from 3 to 50 tons. The construction and operation are otherwise exactly the same as that of the smaller machine in Fig. 12.

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MACHINERY'S REFERENCE SERIES

EACH NUMBER IS ONE UNIT IN A COMPLETE LIBRARY OF MACHINE DESIGN AND SHOP PRACTICE REVISED AND REPUBLISHED FROM MACHINERY

No. 37

BEVEL GEARING

By RALPH E. FLANDERS

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NUMBER 37

BEVEL GEARING

By RALPH E. FLANDERS

CONTENTS

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CHAPTER I

BEVEL GEAR RULES AND FORMULAS

Bevel gearing, as every mechanic knows, is the form of gearing used for transmitting motion between shafts whose center lines intersect. The teeth of bevel gears are constructed on imaginary pitch cones in the same way that the teeth of spur gears are constructed on imaginary pitch cylinders. In Fig. 1 is shown a drawing of a pair of bevel gears of which the gear has twice as many teeth as the pinion. The latter therefore revolves twice for every revolution of the gear. In Fig. 2 is shown (diagramatically) a pair of conical pitch surfaces driving each other by frictional contact. The shafts are set at the same center angle with each other, as in Fig. 1, and

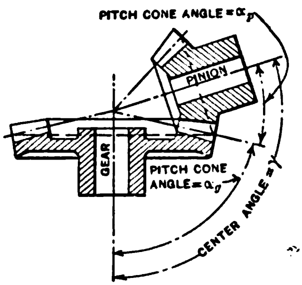

Fig. 1. Bevel Gear and Pinion

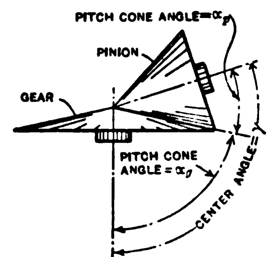

Fig. 2. Pitch Cones of Gears Shown in Fig. 1

the base diameter of the gear cone is twice that of the pinion cone, so that the latter will revolve twice to each revolution of the former. This being the case, the cones shown in Fig. 2 are the pitch cones of the gears shown in Fig. 1. We may therefore define the term "pitch cone" as follows: The pitch cones of a pair of bevel gears are those cones which, when mounted on the shafts in place of the bevel gears, will drive each other by frictional contact in the same velocity ratio as given by the bevel gears themselves.

The pitch cones are defined by their pitch cone angles, as shown in Fig. 2. The sum of the two pitch cone angles equals the center angle. the latter being the angle made by the shafts with each other, measured on the side on which the contact between the cones takes place. The center angle and the pitch cone angles of the gear and the pinion are indicated in Fig. 1.

Different Kinds of Bevel Gears

In Fig. 3 is shown a pair of bevel gears in which the center angle (γ) equals 90 degrees, or in other words, the figure shows a case of right angle bevel gearing. To the special case shown in Fig. 4 in which the number of teeth in the two gears is the same, the term

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mitre gearing is applied; here the pitch cone angle of each gear will always equal 45 degrees.

When the pitch cone angle is less than 90 degrees we have acute angle bevel gearing, as shown in Fig. 5. When the center angle is greater than 90 degrees, we have obtuse angle bevel gearing, shown in Fig. 6 and also in Fig. 1. Obtuse angle bevel gearing is met with occasionally in the two special forms shown in Figs. 7 and 8. When the

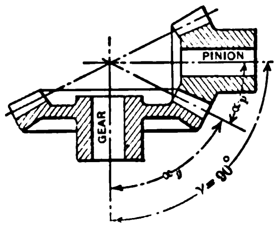

Fig. 3. Right Angle Bevel Gearing

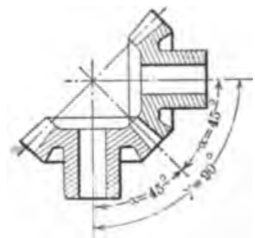

Fig. 4. Mitre Gearing

pitch cone angle a_g equals 90 degrees, the gear *g* is called a crown gear. In this case the pitch cone evidently becomes a pitch plane, or disk When the pitch cone angle of the gear is more than 90 degrees, as in Fig. 8, this member is called an internal bevel gear, and its pitch cone when drawn as for Fig. 2, would mesh with the pitch cone of the pinion on its internal conical surface. These two special forms of gears are of rare occurrence.

Bevel Gear Dimensions and Definitions

In Fig. 9, which shows an axial section of a bevel gear, the pitch lines show the location of the periphery of the imaginary pitch cone.

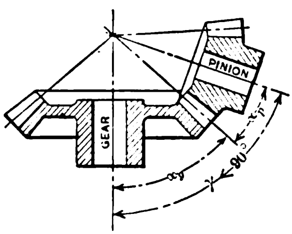

Fig. 5. Acute Angle Bevel Gearing

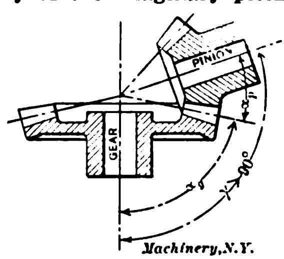

Fig. 6. Obtuse Angle Bevel Gearing

The pitch cone angle is the angle which the pitch line makes with the axis of the gear. The pitch diameter is measured across the gear drawing at the point where the pitch lines intersect the outer edge of the teeth. The teeth of bevel gears grow smaller as they approach the vertex *O* of the pitch cone, where they would disappear if the teeth were cut for the full length of the face. In speaking of the pitch of a bevel gear we always mean the pitch of the larger or outer ends of the teeth. Diametral and circular pitch have the same meaning as in the case of spur gears; the diametral pitch being the

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number of teeth per inch of the pitch diameter, while the circular pitch is the distance from the center of one tooth to the center of the next, measured along the pitch diameter at the back faces of the teeth. The addendum is the height of the tooth above the pitch line at the large end. The dedendum (the depth of the tooth space below the pitch line) and the whole depth of the tooth are also measured at the large end.

The pitch cone radius is the distance measured on the pitch line from the vertex of the pitch cone to the outer edge of the teeth. The width of the face of the teeth, as shown in Fig. 9, is measured on a line parallel to the pitch line. The addendum, whole depth and thickness of the teeth at the small or inner end may be derived from the corresponding dimensions at the outer end, by calculations depending on the ratio of the width of face to the pitch cone radius. (See *s*, *w* and *t* in Fig. 12.)

The addendum angle is the angle between the top of the tooth and the pitch line. The dedendum angle is the angle between the bot-

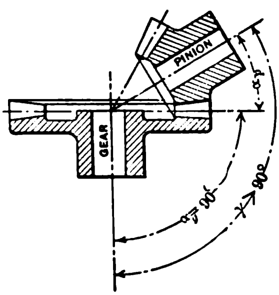

Fig. 7. Crown Gear and Pinion

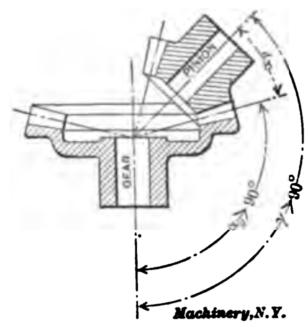

Fig. 8. Internal Bevel Gear and Pinion

tom of the tooth space and the pitch line. The face angle is the angle between the top of the tooth and a perpendicular to the axis of the gear. The edge angle (which equals the pitch cone angle) is the angle between the outer edge and the perpendicular to the axis of the gear. The latter two angles are measured from the perpendicular instead of from the axis, for the convenience of the workman in making measurements with the protractor when turning the blanks. The cutting angle is the angle between the bottom of the tooth space and the axis of the gear.

The angular addendum is the height of tooth at the large end above the pitch diameter, measured in a direction perpendicular to the axis of the gear. The outside diameter is measured over the corners of the teeth at the large end. The vertex distance is the distance measured in the direction of the axis of the gear from the corner of the teeth at the large end to the vertex of the pitch cone. The vertex distance at the small end of the tooth is similarly measured.

The shape of the teeth of a bevel gear may be considered as being the same as for teeth in a spur gear of the same pitch and style of

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tooth, having a radius equal to the distance from the pitch line at the back edge of the tooth to the axis of the gear, measured in a direction perpendicular to the pitch line. This distance is dimensioned $\frac{D'}{2}$ in Fig. 12. The number of teeth which such a spur gear would have, as determined by diameter D' thus obtained, may be called the "number of teeth in equivalent spur gear," and is used in selecting the cutter for forming the teeth of bevel gears by the formed cutter process.

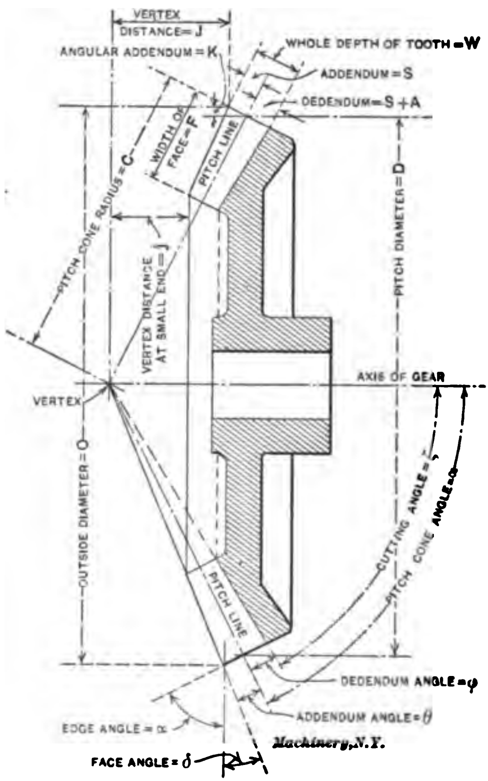

Fig. 9. Dimensions, Definitions and Reference Letters for Ordinary Bevel Gear

In two special forms of gears, the crown gear, Fig. 10, and the internal bevel gear, Fig. 11, the same dimensions and definitions apply as in regular bevel gears, though in a modified form in some cases. In the crown gear, for instance, the pitch diameter and the outside diameter are the same, and the pitch cone radius is equal to ½ the pitch diameter. The addendum angle and the face angle are also the same. The angular addendum becomes zero, and the vertex distance is equal to the addendum. The number of teeth in the equivalent spur gear becomes infinite, or in other words, the teeth are shaped like those of a rack.

When the pitch cone angle is greater than 90 degrees, so that the gear becomes an internal bevel gear, as in Fig. 11, the outside diameter (or edge diameter as it is better called in the case of internal gears) becomes less than the pitch diameter. Otherwise the conditions are the same although many of the dimensions are reversed in direction.

Rules and formulas for calculating the dimensions of bevel gears are given on pages 7, 9, 11, and 13. The following reference letters are used:

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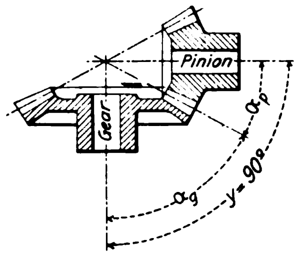
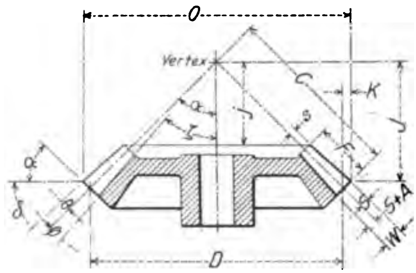
CHART FOR SOLUTION OF BEVEL GEAR PROBLEMS.—I

Bevel Gears with Shafts at Right Angles.





Note: α_p = Pitch Cone Angle of Pinion
α_g = Pitch Cone Angle of Gear
N_p = Number of Teeth in Pinion, etc.

Use Rules and Formulas 1-21 in the order given.

No.	To Find		Rule	Formula
1	Pitch Cone Angle (or Edge Angle) of Pinion		Divide the number of teeth in the pinion by the number of teeth in the gear to get the tangent	$\tan \alpha_p = \frac{N_p}{N_g}$
2	Pitch Cone Angle (or Edge Angle) of Gear		Divide the number of teeth in the gear by the number of teeth in the pinion to get the tangent	$\tan \alpha_g = \frac{N_g}{N_p}$
3	Proof of Calculations for Pitch Cone Angles		The sum of the pitch cone angles of the pinion and gear equals 90 degrees	$\alpha_p + \alpha_g = 90°$
4	Pitch Diameter		Divide the number of teeth by the diametral pitch; or multiply the number of teeth by the circular pitch and divide by 3.1416	$D = \frac{N}{P} = \frac{NP'}{\pi}$
5	These dimensions are the same for both gear and pinion	Addendum	Divide 1.0 by the diametral pitch; or multiply the circular pitch by 0.318	$S = \frac{1.0}{P} = 0.318P'$
6		Dedendum	Divide 1.157 by the diametral pitch; or multiply the circular pitch by 0.368	$S+A = \frac{1.157}{P} = 0.368P'$
7		Whole Depth of Tooth Space	Divide 2.157 by the diametral pitch; or multiply the circular pitch by 0.687	$W = \frac{2.157}{P} = 0.687P'$
8		Thickness of Tooth at Pitch Line	Divide 1.571 by the diametral pitch; or divide the circular pitch by 2	$T = \frac{1.571}{P} = \frac{P'}{2}$
9		Pitch Cone Radius	Divide the pitch diameter by twice the sine of the pitch cone angle	$C = \frac{D}{2 \times \sin \alpha}$
10		Addendum at Small End of Tooth	Subtract the width of face from the pitch cone radius, divide the remainder by the pitch cone radius and multiply by the addendum	$s = S \times \frac{C-F}{C}$
11		Thickness of Tooth at Pitch Line at Small End	Subtract the width of face from the pitch cone radius, divide the remainder by the pitch cone radius and multiply by the thickness of the tooth at the pitch line	$t = T \times \frac{C-F}{C}$
12		Addendum Angle	Divide the addendum by the pitch cone radius to get the tangent	$\tan \theta = \frac{S}{C}$
13		Dedendum Angle	Divide the dedendum by the pitch cone radius to get the tangent	$\tan \phi = \frac{S+A}{C}$

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N = number of teeth,
P = diametral pitch,
P' = circular pitch,
π = 3.1416, (pi),
α = pitch cone angle and edge angle, (alpha),
γ = center angle, (gamma),
D = pitch diameter,
S = addendum,
$S + A$ = dedendum (A = clearance),
W = whole depth of tooth space,

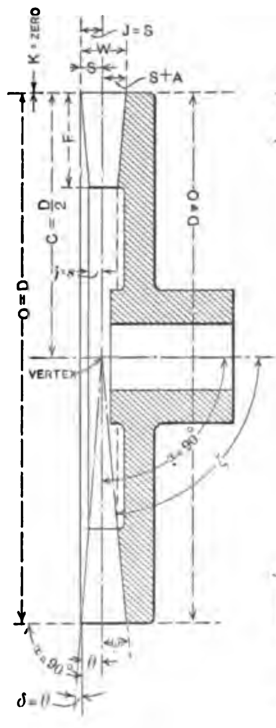

Fig. 10. Dimensions for Crown Gear

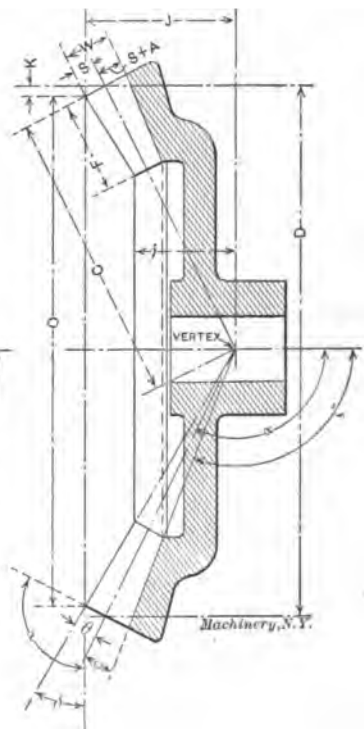

Fig. 11. Dimensions for Internal Bevel Gear

T = thickness of tooth at pitch line,
C = pitch cone radius,
F = width of face,
s = addendum at small end of tooth,
t = thickness of tooth at pitch line at small end,
θ = addendum angle, (theta),
ϕ = dedendum angle, (phi),
δ = face angle, (delta),
ζ = cutting angle, (zeta),
K = angular addendum,

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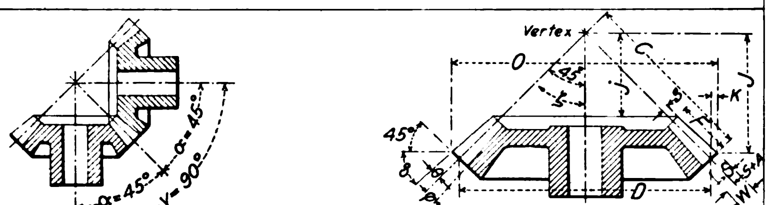
CHART FOR SOLUTION OF BEVEL GEAR PROBLEMS.—II

Bevel Gears with Shafts at Right Angles (Continued).

No.	To Find	Rule	Formula
14	Face Angle	Subtract the sum of the pitch cone and addendum angles from 90 degrees	$\delta = 90° - (\alpha + \theta)$
15	Cutting Angle	Subtract the dedendum angle from the pitch cone angle	$\zeta = \alpha - \varphi$
16	Angular Addendum	Multiply the addendum by the cosine of the pitch cone angle	$K = S \times \cos\alpha$
17	Outside Diameter	Add twice the angular addendum to the pitch diameter	$O = D + 2K$
18	Apex Distance	Multiply one-half the outside diameter by the tangent of the face angle	$J = \frac{O}{2} \times \tan\delta$
19	Apex Distance at Small End of Tooth	Subtract the width of face from the pitch cone radius; divide the remainder by the pitch cone radius and multiply by the apex distance	$j = J \times \frac{C-F}{C}$
20	Number of Teeth in Equivalent Spur Gear	Divide the number of teeth by the cosine of the pitch cone angle	$N' = \frac{N}{\cos\alpha}$
21	Proof of Calculations by Rules Nos. 9, 12, 14, 16 and 17	The outside diameter equals twice the pitch cone radius multiplied by the cosine of the face angle and divided by the cosine of the dedendum angle	$O = \frac{2C \times \cos\delta}{\cos\theta}$

Mitre Bevel Gearing.



Use Rules and Formulas 22, 4-8, 23, 10-13, 24-26, 17-19, 27 and 21 in the order given. All dimensions thus obtained are the same for both gears of a pair

No.	To Find	Rule	Formula
22	Pitch Cone Angle	Pitch cone angle equals 45 degrees	$\alpha = 45°$
23	Pitch Cone Radius	Multiply the pitch diameter by 0.707	$C = 0.707D$
24	Face Angle	Subtract the addendum angle from 45°	$\delta = 45° - \theta$
25	Cutting Angle	Subtract the dedendum angle from 45 degrees	$\zeta = 45° - \varphi$
26	Angular Addendum	Multiply the addendum by 0.707	$K = 0.707\,S$
27	Number of Teeth in Equivalent Spur Gear	Multiply the number of teeth by 1.41	$N' = 1.41N$

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O = outside diameter (edge diameter for internal gears),
J = vertex distance,
j = vertex distance at small end,
N' = number of teeth in equivalent spur gear.
Sub $_p$ refers to dimensions applying to pinion (α_p, N_p, etc.)
Sub $_g$ refers to dimensions applying to gear (α_g, N_g, etc.)

It will be noted that directions for the use of these rules are given for each of the six cases of right angle bevel gearing, mitre bevel gearing, acute angle and obtuse angle bevel gearing, and crown and

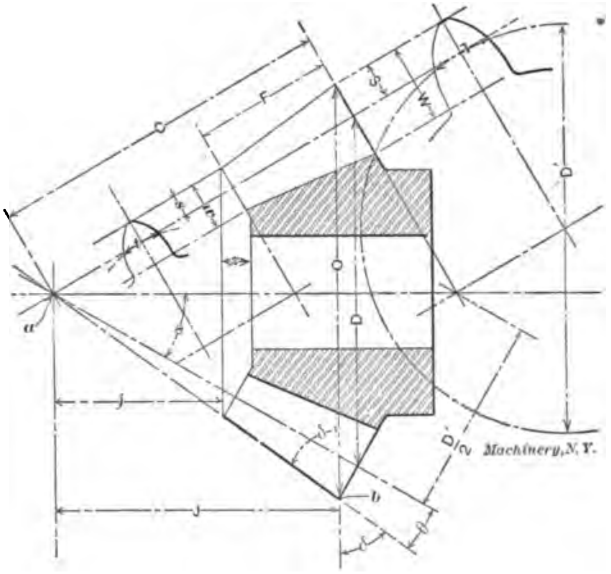

Fig. 12. Diagram Explaining Certain Calculations Relating to Bevel Gears

internal bevel gears. Further instruction as to their use can be obtained from the examples given in Chapter II.

Rules and Formulas for Bevel Gear Calculations

The derivation of most of these formulas is evident on inspection of Figs. 1-12 inclusive, for anyone who has a knowledge of elementary trigonometry. It is not necessary to know how they were derived to use them, however, as all that is needed is the ability to read a table of sines and tangents.

Formulas 5, 6, 7 and 8 are the same as for Brown & Sharpe standard gears. The dimensions at the small end of the tooth given by Formulas 10, 11 and 19 evidently are to the corresponding dimensions at the large end, as the distance from the small end of the tooth to the vertex of the pitch cone is to the pitch cone radius. This relation is expressed by these formulas. The derivation of Formula 20 may be understood by reference to Fig. 12:

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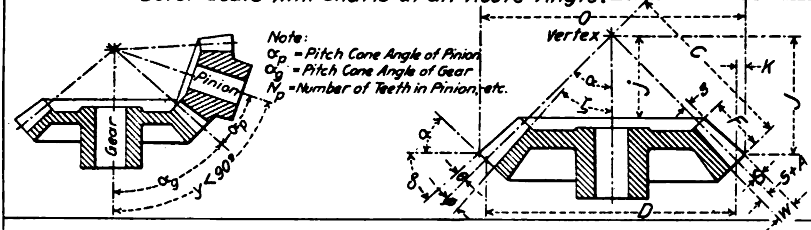
CHART FOR SOLUTION OF BEVEL GEAR PROBLEMS.—III

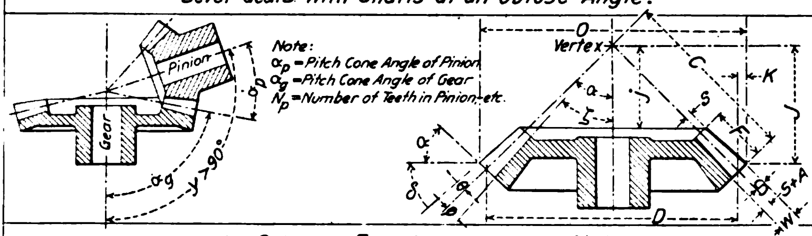
Bevel Gears with Shafts at an Acute Angle.



Use Rules and Formulas 28-30, and 4-21 in the order given.

No.	To Find	Rule	Formula
28	Pitch Cone Angle (or Edge Angle) of Pinion	Divide the sine of the center angle by the sum of the cosine of the center angle and the quotient of number of teeth in the gear divided by the number of teeth in the pinion; this gives the tangent	$\tan \alpha_p = \frac{\sin y}{\frac{N_g}{N_p} + \cos y}$
29	Pitch Cone Angle (or Edge Angle) of Gear	Divide the sine of the center angle by the sum of the cosine of the center angle and the quotient of the number of teeth in the pinion divided by the number of teeth in the gear; this gives the tangent	$\tan \alpha_g = \frac{\sin y}{\frac{N_p}{N_g} + \cos y}$
30	Proof of Calculations for Pitch Cone Angles	The sum of the pitch cone angles of the pinion and gear equals the center angle	$\alpha_p + \alpha_g = y$

Bevel Gears with Shafts at an Obtuse Angle.



Use Rules and Formulas 31 and 32 as directed below.

No.	To Find	Rule	Formula
31	Pitch Cone Angle (or Edge Angle) of Pinion	Divide the sine of 180 degrees minus the center angle by the difference between the quotient of the number of teeth in the gear divided by the number of teeth in the pinion and the cosine of 180 degrees minus the center angle; this gives the tangent	$\tan \alpha_p = \frac{\sin(180° - y)}{\frac{N_g}{N_p} - \cos(180° - y)}$
32	Whether Gear is a Regular Bevel Gear, a Crown Gear, or an Internal Bevel Gear	Add 90 degrees to the pitch cone angle of the pinion. If the sum is greater than the center angle use rules and formulas 33, 30 and 4-21 in the order given. If the sum equals the center angle see rules and formulas for crown gear. If the sum is less than the center angle see rules and formulas for internal bevel gear.	
33	Pitch Cone Angle (or Edge Angle) of Gear	Divide the sine of 180 degrees minus the center angle by the difference between the quotient of the number of teeth in the pinion divided by the number of teeth in the gear and the cosine of 180 degrees minus the center angle; this gives the tangent	$\tan \alpha_g = \frac{\sin(180° - y)}{\frac{N_p}{N_g} - \cos(180° - y)}$

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$$D' = \frac{D}{\cos\alpha} = \frac{N}{P \times \cos\alpha}, \text{ also } D' = \frac{N'}{P}$$

$$\text{therefore } \frac{N'}{P} = \frac{N}{P \times \cos\alpha}, \text{ or } N' = \frac{N}{\cos\alpha}$$

Formula 21 for checking the calculations will also be understood from Fig. 12, where it will be seen that

$$O = 2\,ab \times \cos\delta, \text{ also that } ab = \frac{C}{\cos\theta}$$

$$\text{therefore } O = \frac{2C \times \cos\delta}{\cos\theta}$$

Formulas 22-27 inclusive are simply the corresponding Formulas 1, 9, 14, 15, 16 and 20 when $\alpha = 45$ degrees.

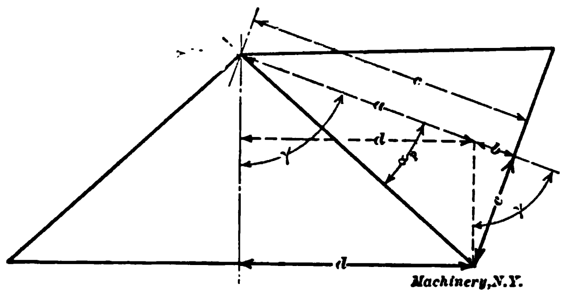

Fig. 13. Diagram for Obtaining Pitch Cone Angle of Acute Angle Gearing

Formula 28 is derived as shown in Fig. 13.

$$c = \frac{e}{\tan\alpha_p}, \text{ also, } c = a + b = \frac{d}{\sin\gamma} + \frac{e}{\tan\gamma}$$

$$\text{therefore, } \frac{e}{\tan\alpha_p} = \frac{d}{\sin\gamma} + \frac{e}{\tan\gamma}$$

Solving for $\tan\alpha_p$, we have: $\tan\alpha_p = \dfrac{e\,(\sin\gamma \times \tan\gamma)}{d\tan\gamma + e\sin\gamma}$

Dividing both numerator and denominator by $e \tan\gamma$, we have:

$$\tan\alpha_p = \frac{\sin\gamma}{\dfrac{d}{e} + \dfrac{\sin\gamma}{\tan\gamma}}$$

Since $d = \dfrac{N_g}{2P}$ and $e = \dfrac{N_p}{2P}$, and since $\dfrac{\sin}{\tan} = \cos$, we have:

$$\tan\alpha_p = \frac{\sin\gamma}{\dfrac{N_g}{N_p} + \cos\gamma}$$

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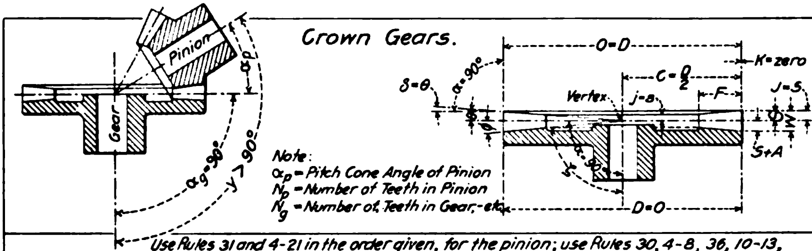
CHART FOR SOLUTION OF BEVEL GEAR PROBLEMS.—IV

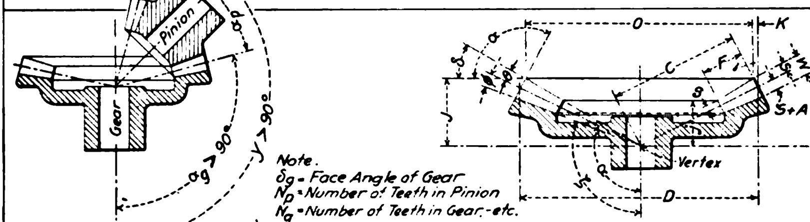
Crown Gears.



Use Rules 31 and 4-21 in the order given, for the pinion; use Rules 30, 4-8, 36, 10-13, 37, 15 and 38 in the order given for the crown gear; if dimensions for crown gear are known, to find center angle and dimensions of pinion, use rules and formulas 34, 35 and 4-21 in the order given

No.	To Find	Rule	Formula
34	Pitch Cone Angle (or Edge Angle) of Pinion	Divide the number of teeth in the pinion by the number of teeth in the gear, to get the sine	$\sin\alpha_p = \frac{N_p}{N_g}$
35	Center Angle	Add 90 degrees to the pitch cone angle of the pinion	$y = 90° + \alpha_p$
36	Pitch Cone Radius	Divide the pitch diameter by 2	$C = \frac{D}{2}$
37	Face Angle of Gear	The face cone angle of the gear equals the addendum angle	$\delta_g = \theta$
38	Number of Teeth in Equivalent Spur Gear	The teeth are equivalent in form to rack teeth	N'_g = infinity

Internal Bevel Gears.



Use Rules and Formulas 31 and 4-21 inclusive for the pinion; use Rules and Formulas 39, 30, 40, 41, 15, 42, 43, 18, 19, 44 and 21 in the order given for the gear

No.	To Find	Rule	Formula
39	Pitch Cone Angle (or Edge Angle) of Gear	Divide the sine of 180 degrees minus the center angle, by the difference between the cosine of 180 degrees minus the center angle and the quotient of the number of teeth in the pinion divided by the number of teeth in the gear; subtract the angle whose tangent is thus found from 180 degrees	$\tan\alpha_a = \frac{\sin(180-y)}{\cos(180-y) - \frac{N_p}{N_a}}$ $\alpha_g = 180 - \alpha_a$
40	Pitch Cone Radius	Divide the pitch diameter by twice the sine of 180 degrees minus the pitch cone angle	$C = \frac{D_g}{2\sin(180-\alpha_g)}$
41	Face Angle of Gear	Subtract 90 degrees from the sum of the pitch cone angle and the addendum angle	$\delta_g = \alpha_g + \theta - 90°$
42	Angular Addendum of Gear	Multiply the addendum by the cosine of 180 degrees minus the pitch cone angle	$K_g = S \times \cos(180-\alpha_g)$
43	Outside (or Edge) Diameter of Gear	Subtract twice the angular addendum from the pitch diameter	$O_g = D_g - 2K_g$
44	Number of Teeth in Equivalent Internal Spur Gear	Divide the number of teeth by the cosine of 180 degrees minus the pitch cone angle	$N'_g = \frac{N_g}{\cos(180-\alpha_g)}$

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Formula 29 is derived by the same process for the other gear. Formula 31 (and likewise 33) is derived from Fig. 14, using the following fundamental equation:

$$\frac{e}{\tan \alpha_p} = \frac{d}{\sin(180° - \gamma)} - \frac{e}{\tan(180° - \gamma)}$$

When solved for $\tan \alpha_p$, this gives formula 31.

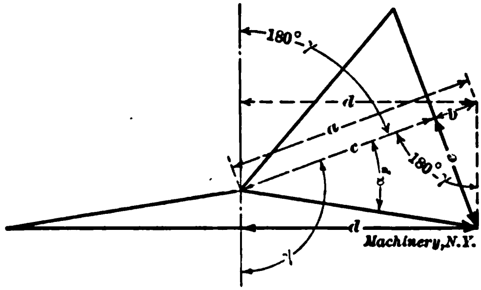

Fig. 14. Diagram for Obtaining Pitch Cone Angle of Obtuse Angle Gearing

Rule No. 32, of course, simply expresses the operation of finding out whether the pitch cone angle of the gear is less, equal to or greater than 90 degrees. The derivation of Formula No. 34 is shown in Fig. 15:

$$\sin \alpha_p = \frac{e}{d} = \frac{N_p}{N_g}$$

Since in a crown gear the dimension $\frac{D'}{2}$ in Fig. 12 is to be measured

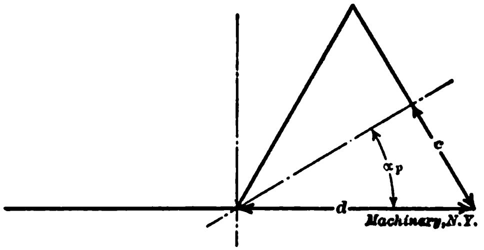

Fig. 15. Diagram for Obtaining Pitch Cone Angle of Pinion to Mesh with Crown Gear

parallel to the axis, and will therefore be of infinite length, the form of the teeth will correspond to those of a spur gear having a radius of infinite length, that is to say, to a rack. This accounts for Formula No. 38.

Formulas 39, 40, 42 and 44 are simply the corresponding Formulas 33, 9, 16 and 20 changed to avoid the use of negative cosines, etc., which occur with angles greater than 90 degrees. These negative functions might possibly confuse readers whose knowledge of trigonometry is elementary. The other formulas for internal gears are readily comprehensible from an inspection of Fig. 11.

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CHAPTER II

EXAMPLES OF BEVEL GEAR CALCULATIONS

A number of examples of calculations are here given for practice, covering all the various types shown in Figs. 3 to 8 inclusive. The conditions of the various examples differ from each other only in the center angle. While such great accuracy is not required in the work itself, it will be found convenient in the calculations to use tables of sines and tangents which give readings for minutes to five figures. This permits accurate checking of the various dimensions by Rules and Formulas 3, 21, etc.

Shafts at Right Angles

Let it be required to make the necessary calculations for a pair of bevel gears in which the shafts are at right angles; diametral pitch = 3, number of teeth in gear = 60, number of teeth in pinion = 15, and width of face = 4 inches.

$$\tan \alpha_p = 15 \div 60 = 0.25000 = \tan 14° \; 2' \quad \text{(1)}$$

$$\tan \alpha_g = 60 \div 15 = 4.00000 = \tan 75° \; 58' \quad \text{(2)}$$

$$\gamma = 14° \; 2' + 75° \; 58' = 90° \quad \text{(3)}$$

$$D_p = 15 \div 3 = 5.000'' \quad \text{(4)}$$

$$S = 1 \div 3 = 0.3333'' \quad \text{(5)}$$

$$S + A = \frac{1.157}{3} = 0.3856'' \quad \text{(6)}$$

$$W = \frac{2.157}{3} = 0.7190'' \quad \text{(7)}$$

$$T = \frac{1.571}{3} = 0.5236'' \quad \text{(8)}$$

$$C = \frac{5}{2 \times 0.24249} = 10.3097'' \quad \text{(9)}$$

$$s = 0.3333 \times \frac{6.31}{10.31} = 0.2040'' \quad \text{(10)}$$

$$t = 0.5236 \times \frac{6.31}{10.31} = 0.3204'' \quad \text{(11)}$$

$$\tan \theta = \frac{0.3333}{10.3097} = 0.03233 = \tan 1° \; 51' \quad \text{(12)}$$

$$\tan \phi = \frac{0.3856}{10.3097} = 0.03740 = \tan 2° \; 9' \quad \text{(13)}$$

$$\delta = 90° - (14° \; 2' + 1° \; 51') = 74° \; 7' \quad \text{(14)}$$

$$\zeta = 14° \; 2' - 2° \; 9' = 11° \; 53' \quad \text{(15)}$$

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$$K = 0.3333 \times 0.97015 = 0.3234'' \quad \ldots\ldots (16)$$

$$O = 5.000 + 2 \times 0.3234 = 5.6468'' \quad \ldots\ldots (17)$$

$$J = \frac{5.6468}{2} \times 3.51441 = 9.9225'' \quad \ldots\ldots (18)$$

$$j = 9.9225 \times \frac{6.31}{10.31} = 6.0726'' \quad \ldots\ldots (19)$$

$$N' = \frac{15}{0.97015} = 15.4 \quad \ldots\ldots (20)$$

$$5.6468'' = \frac{20.6194 \times 0.27368}{0.99948} \cong 5.6461'' \quad \ldots\ldots (21)$$

This gives all the data required for the pinion. Rules 5-13 inclusive apply equally to the gear and the pinion, so we have only calculations by Rules and Formulas 4 and 14 to 21 to make, though it is well to calculate Formula 9 a second time as a check for the same calculation for the pinion.

$$D = \frac{60}{3} = 20.000'' \quad \ldots\ldots (4)$$

$$C = \frac{20}{2 \times 0.97015} = 10.3077'' \quad \ldots\ldots (9)$$

$$\delta = 90 - (75^\circ\ 58' + 1^\circ\ 51') = 12^\circ\ 11' \quad \ldots\ldots (14)$$

$$\zeta = 75^\circ\ 58' - 2^\circ\ 9' = 73^\circ\ 49' \quad \ldots\ldots (15)$$

$$K = 0.3333 \times 0.24249 = 0.0808'' \quad \ldots\ldots (16)$$

$$O = 20 + 2 \times 0.0808 = 20.1616'' \quad \ldots\ldots (17)$$

$$J = \frac{20.1616}{2} \times 0.2159 = 2.1764'' \quad \ldots\ldots (18)$$

$$j = 2.1764 \times \frac{6.31}{10.31} = 1.3320'' \quad \ldots\ldots (19)$$

$$N' = \frac{60}{0.24249} = 247 \quad \ldots\ldots (20)$$

$$20.1616'' = \frac{20.6154 \times 0.97748}{0.99948} \cong 20.1615'' \quad \ldots\ldots (21)$$

This gives the calculations necessary for this pair of gears, which are shown drawn and dimensioned in Fig. 20. There are two or three other dimensions, such as the over-all length of the pinion, etc., which depend on arbitrary dimensions given the gear blank. Directions for calculating these are given in the text in connection with Fig. 20.

Acute Angle Bevel Gearing

Let it next be required to calculate the dimensions of a pair of bevel gears whose center angle is 75 degrees, the number of teeth in the pinion 15, the number of teeth in the gear 60, the diametral pitch 3, and the width of face 4 inches. This is the same as the first exam-

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ple, except for the center angle. Following the directions given in the chart we have:

$$\tan \alpha_p = \frac{0.96593}{\frac{60}{15} + 0.25882} = 0.22681 = \tan 12^\circ 47' \quad (28)$$

$$\tan \alpha_g = \frac{0.96593}{\frac{15}{60} + 0.25882} = 1.89837 = \tan 62^\circ 13' \quad (29)$$

$$\gamma = 12^\circ 47' + 62^\circ 13' = 75^\circ \quad (30)$$

Formulas 4-8 as in the first example; also, $C = 11.2989''$, $s = 0.2154''$, $t = 0.3382''$, $\theta = 1^\circ 41'$, $\phi = 1^\circ 57'$, $\delta = 75^\circ 32'$, $\zeta = 10^\circ 50'$, $K = 0.3251''$, $O = 5.6502''$, $J = 10.9501''$, $j = 7.0748''$, and $N' = 15.3$, also,

$$5.6502'' = \frac{22.598 \times 0.24982}{0.99957} \cong 5.6483'' \quad (21)$$

For the gear, the additional calculations give: $C = 11.303''$, $\delta = 26^\circ 6'$, $\zeta = 60^\circ 16'$, $K = 0.1553''$, $O = 20.3106''$, $J = 4.9748''$, $j = 3.2142''$, $N' = 129$.

$$20.3106'' = \frac{22.606 \times 0.89803}{0.99957} \cong 20.3096'' \quad (21)$$

The above calculations are not all given in full, as most of them are merely re-duplications of formulas previously used.

Crown Gear

Suppose it is required to make a crown gear and a pinion for the same number of teeth, pitch and face as in the previous example. What are the additional calculations necessary? Following the proper formulas in the order given by the chart, we have:

$$\sin \alpha_p = \frac{15}{60} = 0.25000 = \sin 14^\circ 29' \quad (34)$$

$$\gamma = 90^\circ + 14^\circ 29' = 104^\circ 29' \quad (35)$$

The other calculations are similar to those already given.

Internal Bevel Gear

Let it be required to design a pair of bevel gears of the same number of teeth, pitch and face in which the center angle is 115 degrees. This being an example of obtuse angle gearing, we use Formula 31.

$$\tan \alpha_p = \frac{0.90631}{\frac{60}{15} - 0.42262} = 0.25334 = \tan 14^\circ 13' \quad (31)$$

Thus, according to Rule 32, we find that

$$14^\circ 13' + 90 = 104^\circ 13' < 115^\circ \quad (32)$$

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showing that the gear is an internal bevel gear. Applying the rules and formulas for internal bevel gearing, we have:

$$\tan \alpha_s = \frac{0.90631}{0.42262 - \frac{15}{60}} = 5.25032 = \tan 79^\circ\ 13'$$

$$180^\circ - 79^\circ\ 13' = 100^\circ\ 47' \quad \ldots\ldots\ldots\ldots (39)$$

$$\gamma = 100^\circ\ 47' + 14^\circ\ 13' = 115^\circ \quad \ldots\ldots\ldots\ldots (30)$$

$$C = \frac{20}{2 \times 0.98234} = 10.1797'' \quad \ldots\ldots\ldots\ldots (40)$$

$$\delta = 100^\circ\ 47' + 1^\circ\ 53' - 90^\circ = 12^\circ\ 40' \quad \ldots\ldots\ldots\ldots (41)$$

$$\zeta = 98^\circ\ 37', \text{ and } K = 0.0624''$$

$$O = 20 - 2 \times 0.0624 = 19.8752'' \quad \ldots\ldots\ldots\ldots (43)$$

$$N' = \frac{60}{0.1871} = 320 \text{ (internal)} \quad \ldots\ldots\ldots\ldots (44)$$

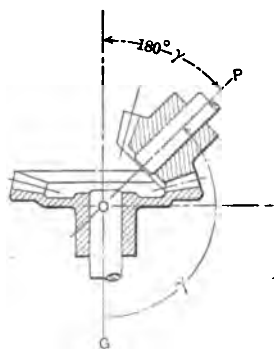

Fig. 16. Internal Bevel Gearing

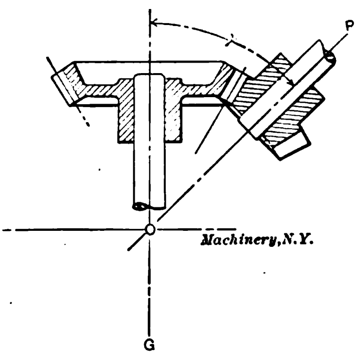

Fig. 17. Acute Bevel Gearing Used as Substitute for Internal Gearing in Fig. 16

The calculatons for the pinion and the other calculations for the gear are similar to those already given.

Obtuse Angle Bevel Gearing

Let it be required to calculate the dimensions of the same set of gears but with the center angle of 100 degrees. This being an example of obtuse angle gearing, we apply Formula 31 as follows:

$$\tan \alpha_p = \frac{0.98481}{\frac{60}{15} - 0.17365} = 0.25738 = \tan\ 14^\circ\ 26' \ldots\ldots\ldots (31)$$

and thus discover that it is an example of regular obtuse angle gearing. since

$$14^\circ\ 26' + 90^\circ = 104^\circ\ 26' > 100^\circ \ldots\ldots\ldots\ldots (32)$$

The remaining calculations for the angles are as follows:

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$$\tan\alpha = \frac{0.98481}{\frac{15}{60} - 0.17365} = 12.8986 = \tan\ 85°\ 34' \quad \ldots\ldots\ldots\ldots (33)$$

$$\gamma = 14°\ 26' + 85°\ 34' = 100° \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (30)$$

and the calculations for the other dimensions as per the table.

How to Avoid Internal Bevel Gears

When Rule 32, in any given case, shows that the large gear will be an internal bevel gear, such as shown in Fig. 16, this construction may be avoided without changing the position of the shafts, the numbers of the teeth in the gear, the pitch of the teeth, or the width of face. This is done simply by subtracting the given center angle from 180 degrees, and using the remainder as a new center angle in calculating a set of acute angle gears by Rules and Formulas 28, 29, 30, etc. A pair of bevel gears calculated on this basis corresponding to those in Fig. 16 is shown in Fig. 17. It will be seen that the contact takes place on the other side of the axis *OP* of the pinion.

It is necessary to avoid internal bevel gears as it is practically impossible to cut them. It may be that some forms of templet planing machines will do this work, if the pitch cone angle is not too great, but no form of generating machine will do it. It is rather doubtful if any one has ever cut a pair of internal bevel gears, though the writer has seen occasional examples of cast gears of this type.

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CHAPTER III

SYSTEMS OF TOOTH OUTLINES USED FOR BEVEL GEARING

Five systems of tooth outlines are commonly used for bevel gearing. They are the cycloid, the standard 14½-degree involute, the 20-degree involute and the 15- and 20-degree octoid.

The Cycloidal System

The cyloidal form of tooth is obsolete for cut bevel gears, and is rarely met with nowadays for cast gears even. It requires very careful workmanship, and is difficult or impossible to generate. It is also a bad shape to form with a relieved cutter, as the cutting edge tends to drag at the pitch line, where for a short distance the sides of the teeth are nearly or quite parallel. For spur gearing it has a few points of advantage over the involute form of tooth, but in the case of bevel gearing these are nullified by the impossibility of generating the teeth in practicable machines. The cycloidal form of tooth need not be seriously considered for bevel gears.

Involute and Octoid Teeth

Most bevel gears are made on the involute system, of either the standard 14½ degree pressure angle, or the 20-degree pressure angle. In spur gear teeth the pressure angle may be defined as the angle which the flat surface of the rack tooth makes with the perpendicular to the pitch line. The 20-degree tooth is consequently broader at the base and stronger in form than the 14½-degree tooth. This same difference applies to bevel gears. Most bevel gears that are milled with formed cutters are made to the 14½-degree standard, as cutters for this shape are regularly carried in stock. The planed gears, made by the templet or generating principles, are nowadays often made to the 20-degree pressure angle, both for the sake of obtaining stronger teeth, and for avoiding undercutting of the flanks of the pinions as well. This undercutting is due to the phenomenon of "interference," as it is called, which is minimized by increasing the pressure angle.

If you ask the manufacturer to plane a pair of involute bevel gears for you on the Bilgram, Gleason or other similar generating machine, he will not give you involute teeth, but something "just as good." This just as good form was invented by Mr. Bilgram, and was named "Octoid" by Mr. Geo. Grant. In generating machines the teeth of the gears are shaped by a tool which represents the side of the tooth of an imaginary crown gear. The cutting edge of the tool is straight line, since the imaginary crown gear has teeth whose sides are plane surfaces. It can be shown that the teeth of a true involute crown

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gear have sides which are very slightly curved. The minute difference between the tooth shapes produced by a plane crown tooth and a slightly curved crown tooth is the minute difference between the octoid and involute forms. Both give theoretically correct action. The customer in ordering gears never uses the word "octoid," as it is not a commercial term; he calls for "involute" gears.

Formed Cutters for Involute Teeth

For 14½-degree involute teeth, the shapes of the standard cutter series furnished by the makers of formed gear cutters are commonly used. There are 8 cutters in the series, to cover the full range from the 12-tooth pinion to a crown gear. The various cutters are numbered from 1 to 8, as given in the table below:

No. 1 will cut wheels from 135 teeth to a rack.
No. 2 will cut wheels from 55 teeth to 134 teeth.
No. 3 will cut wheels from 35 teeth to 54 teeth.
No. 4 will cut wheels from 26 teeth to 34 teeth.
No. 5 will cut wheels from 21 teeth to 25 teeth.
No. 6 will cut wheels from 17 teeth to 20 teeth.
No. 7 will cut wheels from 14 teeth to 16 teeth.
No. 8 will cut wheels from 12 teeth to 13 teeth.

It should be remembered that the number of teeth in this table refers to the number of teeth in the equivalent spur gear, as given by Rule 20, which should always be used in selecting the cutter used for milling the teeth of bevel gears. Thus for the gear in the first example in Chapter II, the No. 1 cutter should be used. The standard bevel gear cutter is made thinner than the standard spur gear cutter, as it must pass through the narrow tooth space at the inner end of the face. As usually kept in stock, these cutters are thin enough for bevel gears in which the width of face is not more than one-third the pitch cone radius. Where the width of face is greater, special cutters have to be made, and the manufacturer should be informed as to the thickness of the tooth space at the small end; this will enable him to make the cutter of the proper width.

Special Forms of Bevel Gear Teeth

In generating machines (such as the Bilgram and the Gleason) it is often advisable to depart from the standard dimensions of gear teeth as given by Rules and Formulas 1 to 44. For instance, where the pinion is made of bronze and the gear of steel, the teeth of the former can be made wider and those of the latter correspondingly thinner, so as to somewhere nearly equalize the strength of the two. Again, where the pinion has few teeth and the gear many, it may be advisable to make the addendum on the pinion larger and the dedendum correspondingly smaller, reversing this on the gear, making the addendum smaller and the dedendum larger. This is done to avoid interference and consequent undercut on the flanks of pinions having a small number of teeth. Such changes are easily effected on generating machines and instructions for doing this for any case will be furnished by the makers.

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CHAPTER IV

STRENGTH AND DURABILITY OF BEVEL GEARS

The same materials are used in general for making bevel gears as for spur gears and each has practically the same advantages and disadvantages for both cases. In general, the strength of different materials is roughly proportional to the durability.

The Materials Used for Making Bevel Gears

Cast iron is used for the largest work, and for smaller work which is not to be subjected to heavy duty. In cases where great working stress or a sudden shock is liable to come on the teeth, steel is ordinarily used. Such gears are made from bar stock for the smallest work, from drop forgings for intermediate sizes made on a manufacturing basis, and from steel castings for heavy work. The softer grades of steel are not fitted for high speed service, as this material abrades more rapidly than cast iron. This objection does not apply to hardened steels, such as used in automobile transmission gears.

As in the case of spur gearing it is quite common to make the gear and pinion of different materials. This is advantageous from the standpoint of both efficiency and durability, since two dissimilar metals work on each other with less friction than similar metals, as is well known. Cast iron and steel, and steel and bronze are common combinations. In general, the pinion should be made of the stronger material, since it is of weak form; and it should be made of the more durable material, as it revolves more rapidly and each tooth comes into working contact more times per minute than do those of the larger mating gear. In a steel-cast iron combination, then, the pinion should be of steel, while the gear is of cast iron. In a steel-bronze combination, the pinion should be of steel and the gear of bronze, though this is more costly than when the materials are reversed.

A wide range of physical qualities is now available in steel, both for parts small enough to be made from bar stock, and for those made from drop forgings. Recent improvements have also given almost as much flexibility in the choice of steel castings. Gears made from high grade steels may be subjected to heat treatments which increase their durability and strength amazingly.

Raw-hide and fiber are quite largely used for pinion blanks in cases where it is desired to run gearing at a very high speed and with as little noise as possible. There is a little more difficulty in building up a raw-hide blank properly for a bevel gear than for a spur gear. Fiber, which is used in somewhat the same way, has the merit of convenience and comparative inexpensiveness, as it may be purchased in a variety of sizes of bars, rods, tubes, etc., ready to be worked up into pinion blanks at short notice. It is not so strong as

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rawhide, and is difficult to machine owing to its gritty composition. For light duty at high speed it does very well. For large, high-speed gearing it was formerly a common practice to use inserted wooden teeth on the gear, meshing with a solid cast iron pinion. This construction is seldom used for cut gearing.

Strength of Bevel Gear Teeth

The Lewis formula is the one generally used in this country for calculating the strength of gears. Mr. Myers, who had an article on the "Strength of Gears" in the December, 1906, issue of MACHINERY, gives Mr. Barth's adaptation of this formula for calculating the strength of bevel gears. The rules and formulas on the next page are condensed from the method given in the article referred to.

The factors to be taken into account are the pitch diameter of the gear, the number of revolutions per minute, the diametral pitch (or circular pitch as the case may be) the width of face, the pitch cone radius, the number of teeth in the gear and the maximum allowable static fiber stress for the material used. From this we may find the maximum allowable load at the pitch line, and the maximum H. P. the gear should be allowed to transmit.

The reader familiar with the Lewis formula will note that rule and formula No. 47 is the same as for spur gears with the exception of the additional factor $\frac{C - F}{C}$. This factor is an approximate one which expresses the ratio of the strength of a bevel gear to that of a spur gear of the same pitch and number of teeth, the decrease being due to the fact that the pitch grows finer toward the vertex. This factor is approximate only and should not be used for cases in which F is more than 1/3 C; but since no bevel gears should be made in which F is more than 1/3 C, the rule is of universal application for good practice. As the width of face is made greater in proportion to the pitch cone radius, the increase of strength obtained thereby grows proportionately smaller and smaller, as may be easily proved by analysis and calculation. Actually the advantage of increasing the width of face is even less than is indicated by calculation, since the unavoidable deflection and disalignment of the shaft is sure at one time or another to throw practically the whole load on the weak inner ends of the teeth, which thus have to carry the load without help from the large pitch at the outer ends.

Rules and Formulas for the Strength of Bevel Gears

The reference letters, rules and formulas on the next page, for the strength of bevel gear teeth, are self-explanatory. As an approximate guide for ordinary calculations, 8,000 pounds per square inch may be allowed for the static stress of cast iron and 20,000 pounds for ordinary machine steel or steel castings. Where the gearing is to be subjected to shock, 6,000 pounds for cast iron and 15,000 pounds for steel are more satisfactory figures. The wide range of materials offered the designer, however, makes any fixed tabulation of fiber

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STRENGTH OF BEVEL GEARS

List of Reference Letters.

D = pitch diameter of gear in inches.

R = revolutions per minute.

V = velocity in ft. per min. at pitch diameter.

S_s = allowable static unit stress for material.

S = allowable unit stress for material at given velocity.

F = width of face.

N′ = No. of teeth in equivalent spur gear (See diagram).

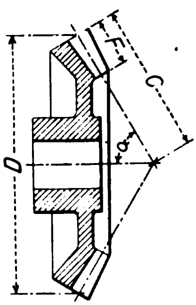
Y = outline factor (see table below).

P = diametral pitch (if circular pitch is given, divide 3.1416 by circular pitch to obtain diametral pitch).

C = pitch cone radius.

W = maximum safe tangential load in pounds at pitch diameter.

H.P. = maximum safe horse power.



$N' = \frac{\text{Number of teeth}}{\text{Cos } \alpha}$

(Rule No. 20)

Table of Outline Factors (Y) for 14½° and 20° Involute

N′	Outline Factor = Y		N′	Outline Factor = Y	
	14½° Involute (Std.)	20° Involute		14½° Involute (Std.)	20° Involute
12	0.210	0.245	27	0.314	0.349
13	0.220	0.261	30	0.320	0.358
14	0.226	0.276	34	0.327	0.371
15	0.236	0.289	38	0.336	0.383
16	0.242	0.295	43	0.346	0.396
17	0.251	0.302	50	0.352	0.408
18	0.261	0.308	60	0.358	0.421
19	0.273	0.314	75	0.364	0.434
20	0.283	0.320	100	0.371	0.446
21	0.289	0.327	150	0.377	0.459
23	0.295	0.333	300	0.383	0.471
25	0.305	0.339	Rack	0.390	0.484

Use rules and formulas 45–48 in the order given

No.	To Find	Rule	Formula
45	Velocity in ft. per min. at the pitch diameter	Multiply the product of the diameter in inches and the number of revolutions per minute, by 0.262	$V = 0.262\ DR$
46	Allowable unit stress at given velocity	Multiply the allowable static stress by 600 and divide the result by the velocity in feet per minute plus 600	$S = S_s \times \frac{600}{600 + V}$
47	Maximum safe tangential load at pitch diameter	Multiply together the allowable stress for the given velocity, the width of face, the tooth outline factor and the difference between the pitch cone radius and the width of face; divide the result by the product of the diametral pitch and the pitch cone radius	$W = \frac{SFY(C-F)}{PC}$
48	Maximum safe Horse Power	Multiply the safe load at the pitch line by the velocity in feet per minute, and divide the result by 33,000	$HP = \frac{WV}{33{,}000}$

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stress impracticable. An example showing the use of these rules and formulas is given herewith.

Calculate the maximum load at the pitch line which can be safely allowed for the bevel gears in Fig. 20, if the maximum allowable static stress for the pinion is 20,000 pounds, and for the gear, 8,000 pounds per square inch; the pinion runs at 300 revolutions per minute. The calculations for the pinion are as follows:

$$V = 0.262 \times 5 \times 300 = 400 \text{ feet per minute (about)} \qquad (45)$$

$$S = 20{,}000 \times \frac{600}{600 + 400} = 12{,}000 \text{ pounds per square inch} \qquad (46)$$

$$W = \frac{12{,}000 \times 4 \times 0.289 \times 6.3}{3 \times 10.3} = 2{,}830 \text{ pounds} \qquad (47)$$

For the gear, the velocity is the same as for the pinion. The necessary calculations are as follows:

$$S = 8{,}000 \times \frac{600}{600 + 400} = 4{,}800 \text{ pounds per square inch} \qquad (46)$$

$$W = \frac{4{,}800 \times 4 \times 0.470 \times 6.3}{3 \times 10.3} = 1{,}840 \text{ pounds} \qquad (47)$$

The gear is, therefore, the weaker of the two, and thus limits the allowable tooth pressure. The maximum horse power this gearing will transmit safely is found as follows:

$$H.P = \frac{1840 \times 400}{33{,}000} = 22.3 \qquad (48)$$

Durability is practically of as much importance as strength in proportioning bevel gears, but unfortunately no data is as yet available for making satisfactory comparisons of durability, so the usual procedure is followed of designing the gears for strength alone, trusting to providence that they will not wear out within the lifetime of the machine in which they are used.

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CHAPTER V

DESIGN OF BEVEL GEARING

So far we have dealt with design as relating to calculations. In this chapter will be discussed the application of the calculated dimensions, the determination of the factors left to the judgment, and the recording of the design in the drawing.

Bevel Gear Blanks

Various forms may be given to the blanks or wheels on which bevel gear teeth are cut, depending on the size, material, service, etc., to be provided for. The pinion type of blank is shown in Fig. 12 and elsewhere. It is used mostly, as indicated by the name, for gears of a small number of teeth and small pitch cone angle. Where the diameter of the bore comes too near to the bottoms of the teeth at the

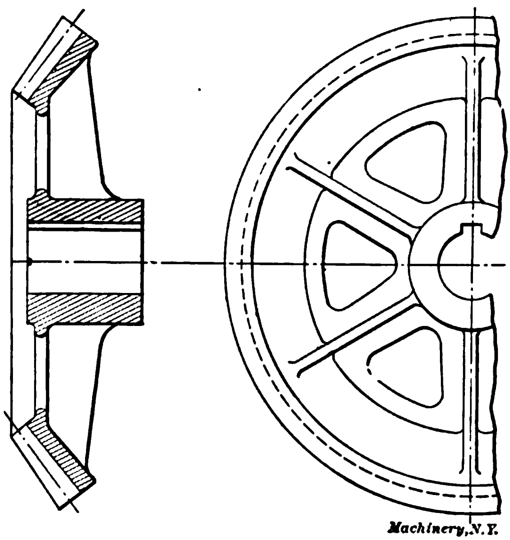

Fig. 18. T-arm Style of Bevel Wheel for Heavy Work

small end, it is customary to omit the recess indicated by dimension z, and leave the front face of the pinion blank as in the case shown in Fig. 20.

For gears of a larger number of teeth, the web type shown in Fig. 9 and elsewhere is appropriate. This does not require to be finished all over, as the sides of the web, the outside diameter of the hub, and the under side of the rim may be left rough if desired.

A steel gear suitable for very heavy work is shown in Fig. 18. Here the web is reinforced by ribs. The web may be cut out so that

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the rim is supported by T-shaped arms, as shown. This makes a very stiff wheel and at the same time a very light one, when its strength is considered. Where the pitch cone angle is so great that the strengthening rib would be rather narrow at the flange, it may be given the form shown in Fig. 20 in place of that shown in Fig. 18.

General Considerations Relating to Design

The performance of the most careful designed and made bevel gears depends to a considerable extent on the design of the machine in which they are used. When the shafts on which a pair of bevel gears are mounted are poorly supported or poorly fitted in their bearings, the pressure of the driving gear on the driven causes it to climb up on the latter, throwing the shafts out of alignment. This in turn causes the teeth to bear with a greater pressure at one end of the face (usually on the outer end) than the other, thus making the tooth more liable to break than is the case where the pressure is more evenly distributed. It is important, therefore, to provide rigid shafts and bearings and careful workmanship for bevel gearing.

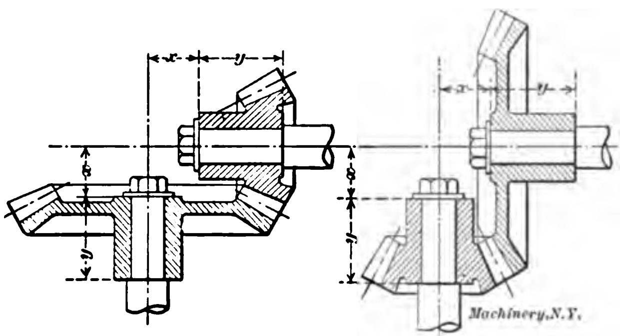

Fig. 19. Interchangeable Bevel Gearing

The question of alignment of the shafts should be considered in deciding on the width of face of the gear. Making the width of the face more than one-third of the pitch cone radius adds practically nothing to the strength of the gear even theoretically, since the added portion is progressively weaker as the tooth is lengthened, as has been explained. In addition to this, there is the danger that through springing of the shafts or poor workmanship, the load will be thrown onto the weak end of the tooth, thus fracturing it. For this reason it may be laid down as a definite rule that there is nothing to be gained by making the face of the bevel gear more than one-third of the pitch cone radius, as required by Rules 45 to 48.

The Brown & Sharpe gear book gives a rule for the maximum width of face allowable for a given pitch. The width of face should not exceed five times the circular pitch or 16 divided by the diametral pitch. This rule is also rational since the danger to the teeth from the misalignment of the shaft increases both with the width of face and with the decrease of the size of the tooth, so that both of these should be

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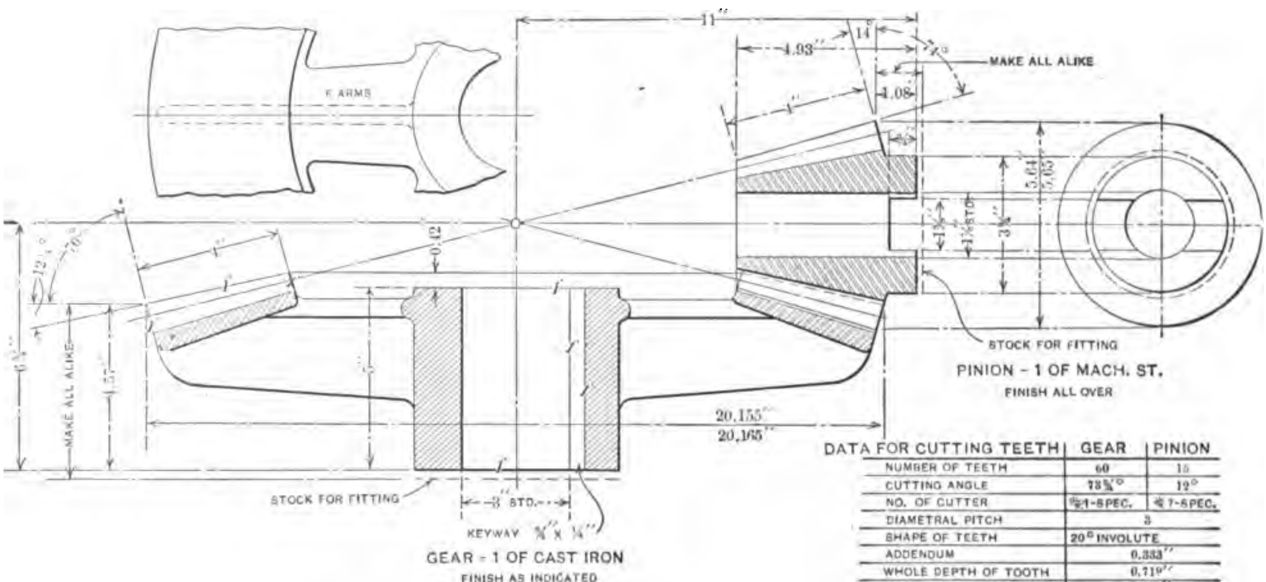

DATA FOR CUTTING TEETH	GEAR	PINION
NUMBER OF TEETH	60	15
CUTTING ANGLE	73⅝°	12°
NO. OF CUTTER	[illegible]1-SPEC.	[illegible]7-SPEC.
DIAMETRAL PITCH	3	
SHAPE OF TEETH	20° INVOLUTE	
ADDENDUM	0.333″	
WHOLE DEPTH OF TOOTH	0.719″	
TOOTH THICKNESS	0.524″	
ADDENDUM AT SMALL END	0.204″	
TOOTH THICKNESS " "	0.320″	

Machinery, N.Y.

Fig. 20. Model Working-Drawing for Bevel Gears to be Cut with Formed Milling Cutter

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limited. In designing gearing it is well to check up the width of face from the rule relating to the pitch cone radius and that relating to the pitch as well, to see that it does not exceed the maximum allowed by either.

Interchangeable Bevel Gearing

It is not generally realized that bevel gears may be used for change gearing in cases where a small number of ratios are desired and where their use will save an extra shaft and bearings. The conditions for interchangeable bevel gears are shown in Fig. 19. The same gears are shown in place at the right and the left of that figure, but reversed in position thus giving two ratios. To make this possible, dimensions x and y must be the same for each of the sets of gears. Any other pair of bevel gears in which dimensions x and y are the same may be used, provided sufficient clearance is furnished in the machine. This arrangement, though sometimes permitting the simplification of the mechanism, has the disadvantage that bevel gears are less convenient to store than spur gears, and that one of the shafts must be slipped back in its bearings to remove the gear.

Model Bevel Gear Drawing

It is not enough for the designer to carefully calculate the dimensions of a set of bevel gearing. In addition to this he has the important task of recording these dimensions in such a form that they will be intelligible to an intelligent workman, and will plainly furnish him every point of information needed for the successful completion of the work without further calculation. A drawing which practically fills these requirements is shown in Fig. 20. The arrangement of this drawing and the amount and kind of information shown on it are based on the drafting-room practice of the Brown & Sharpe Mfg. Co., as described by Mr. Burlingame in the article "Figuring Gear Drawings" in the August, 1906, issue of MACHINERY. Some changes and additions have been made in the arrangement of the dimensioning, however, so that firm cannot be held responsible for all that appears on the engraving.

In general, the dimensions necessary for turning the blank have been given on the drawing itself, while those for cutting the teeth are given in tabular form. All the dimensions were calculated from Rules 1 to 21 inclusive and may be checked for practice by the reader. It will be noticed that limits are given for the important dimensions. This should always be done for manufacturing work which is inspected in its course through the shop. It ought to be done even when a single gear is made, as it is exceedingly difficult to properly set a gear if the workman does not work close enough. There is no sense, however, in asking him to work to thousandths of an inch on blanks like these, so he should be given some notion as to the accuracy requirements by limits such as shown.

It is assumed that the gears are to be cut with rotary cutters. It is unusual to do this with pitches as coarse as this, though there are machines on the market capable of handling such work. In gear cut-

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ting machines using form cutters, the blanks are located for axial position by the rear face of the hub. It is necessary also to leave stock at this place for fitting the gears in the machine. It will be seen that the dimension for bevel gears and pinions from the outside edge of the blank to the rear face of the hub is marked "make all alike." This means that the same amount of stock should be left on all the gears in a given lot so that after the machine is set for one of them, it will not be necessary to alter the adjustment for the remainder.

There are one or two dimensions which are not given directly by rules 1 to 21. One of these is the distance 4.57 inches from the outside edge of the teeth to the finished rear face of the hub of the gear. This dimension is commonly scaled from an accurate drawing, but it may be calculated by subtracting the vertex distance from the distance between the pitch cone vertex O and the rear face of the hub. This gives $6\frac{3}{4} - 2.1764$ equals 4.57 inches (about) as dimensioned. Another dimension not directly calculated is the over-all length of the pinion. This may be obtained by subtracting the vertex distance at the small end (j) from the distance between the vertex and the rear face of the hub, giving 4.93 inches as shown.

In the tabular dimensions for cutting the teeth, most of the figures are self-explanatory. The fact that in this particular case a 20-degree form of tooth has been adapted to avoid the undercut in small pinion (see Chapter III on Systems of Tooth Outlines used for Bevel Gearing) is indicated in the table.

The number of cutter is selected from the table on page 21 in accordance with the number of teeth (N') in equivalent spur gear, as determined by Rule 20. This is 15.4 for the pinion, and 247 for the gear, giving a No. 1 and No. 7 cutter respectively. These cutters are marked special, owing to the fact that they are 20 degrees involute instead of 14½ degrees. They would be special under any circumstances, however, since the width of face for these gears (4 inches) is more than 1/3 the pitch cone radius, which figures out to 10.3097 inches. Standard bevel gear cutters are only made thin enough to pass through the teeth at the small end when the width of face is not more than 1/3 the pitch cone radius. For this reason cutters thinner than the standard would have to be used.

In bevel pinions of the usual form, such as shown in Fig. 12, dimension z there given has to be furnished. This may be scaled from a carefully made drawing, or may be calculated by subtracting the length of the bore of the pinion from the over-all length, the latter being obtained as described for the pinion in Fig. 20. Such dimensions do not need to be given in thousandths on moderately large work. It is also not necessary to give the angles any closer than the quarter degree, as few machines are furnished with graduations which can be read finer than this. In order to check the calculations carefully, however, it is wise, as previously described, to make them with considerable accuracy, using tables of sines and tangents which read to five figures. After the dimensions are calculated, they may be put in more approximate form for the the drawing.

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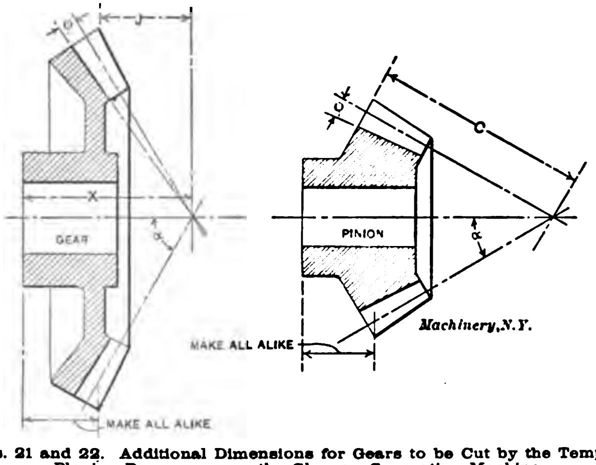
The gear drawing in Fig. 20 is dimensioned more fully, perhaps, than is customary, especially in shops having a large gear cutting department, where the foreman and operators are experienced and have access to tables and records of data for bevel gear cutting. Every dimension given is useful, however, and it is a good plan to include them all, especially on large work.

Dimensioning Drawings for Gears whose Teeth are to be Planed

The machine on which the teeth of a gear are to be cut determines to some extent the dimensions which the workman needs, so this should be taken into account in making the drawing. For gears which are to be cut on a templet planing machine, the dimension given in Fig. 20 may be followed in general. Further dimensions are needed, however, to set the blank so that the vertex of the pitch cone corresponds with the central axis of the machine. For gears with pitch cone angle greater than 45 degrees, this may be obtained from dimension *X*, as given in Fig. 21, or, better, from dimension *J*.



Figs. 21 and 22. Additional Dimensions for Gears to be Cut by the Templet Planing Process, or on the Gleason Generating Machine

For pinions and any gears smaller than 45 degrees, dimension *C* (Fig. 22) may be given.

There are two commercial forms of gear generating machines in general used in this country for planing the teeth of bevel gears. These are the Gleason and Bilgram machines. Since the methods of supporting the gears are different, the drawings should be dimensioned to suit if it is known beforehand how they are to be cut. For the Gleason machine the dimensioning shown in Figs. 21 and 22 should be given, in addition to that shown in Fig. 20. The angles a and ϕ, the pitch cone angle and dedendum angle respectively, may well be put in the table of dimensions instead of on the drawing. The distance from the outside corner of the teeth to the rear face of the hub should be made alike for all similar gears in the lot, the same as for gears which are to be cut by the form cutters or the templet process. The cutting angle may be omitted from the drawing.

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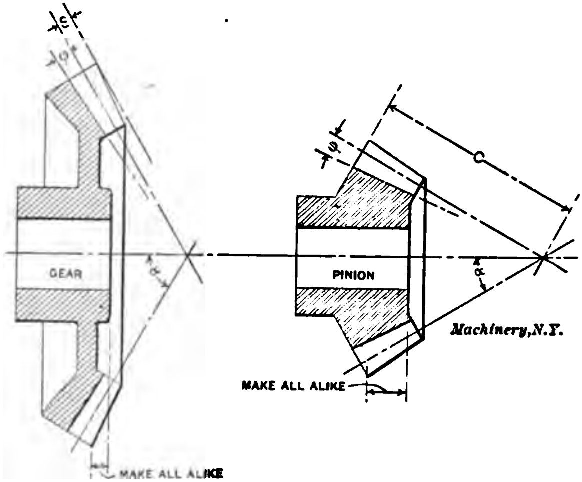
The method of dimensioning for the Bilgram gear planer is shown in Figs. 23 and 24. Angles a and ϕ should be given in the table as before. Dimension S is used for setting on gears of large pitch cone angle, and dimension C or the pitch cone radius for those of small



Figs. 23 and 24. Additional Dimensions for Gears to Be Cut on the Bilgram Generating Machine

pitch cone angle (less than 45 degrees). It is a good idea to give both of these dimensions for both gear and pinion, so that the setting of each may be checked by two different methods. In this machine the gears are located from the front face of the hub, so that the dimension to be marked "make all alike" should be given as shown.

CHAPTER VI

MACHINES FOR CUTTING BEVEL GEAR TEETH

While a very large number of machines have been placed on the market, first and last, for cutting the teeth of bevel gears, the number of designs in common use in this country is small, it being possible, practically, to number them with the fingers of one hand. A brief discussion will here be given of the principles and mechanism of the more commonly used of these machines.

Spherical Basis of the Bevel Gear; Tredgold's Approximation

The principles in common use for cutting teeth of bevel gears are identical with those for cutting the teeth of spur gears, but they are modified in their application to correspond with the spherical basis of the bevel gear. Fig. 25 shows two bevel gears and a crown gear with axis *OC*, *OB*, and *OA* respectively. Fig. 26 shows their pitch surfaces, all of which converge at vertex *O*. These pitch surfaces are formed

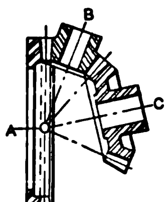


Fig. 25

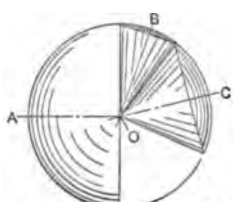


Fig. 26

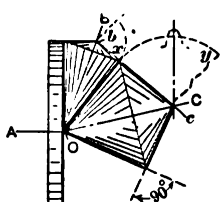


Fig. 27

Illustrating the Spherical Basis of Bevel Gears, and Tredgold's Approximation for Developing the Outlines of the Teeth on a Plane Surface

of cones, cut from a sphere as shown, whose center is at the vertex *O*. The pitch surface of the crown gear becomes the plane face of the hemisphere at the left of Fig. 26. To study the action of these gears the same way as we do that of spur gears when their teeth are drawn on the plane surface of the drawing board, the corresponding lines for the bevel gears would have to be drawn on the surface of the sphere from which the pitch cones were cut. The various pitch circles would be struck from centers located at the points where the axes *OA*, *OB*, and *OC* break with the surface of the sphere. The method of drawing would be identical with that for spur gears. It should be noted that straight lines, on spherical surfaces, are represented by great circles—that is to say, by the intersection of the surface with planes passing through the center of the sphere.

Owing to the impracticability of the sphere as a drawing board, a process, known as "Tredgold's Approximation," is usually followed for laying out the teeth of bevel gears. This is shown in Fig. 27 applied to the same case as in the two preceding figures. The teeth are

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drawn and the action studied on surfaces of cones complementary to the pitch cones—that is, on the cones with vertices at *c* and *b*. The surfaces of these cones can be developed on a flat piece of paper, as shown on axes *OB* and *OC*. In these cases the pitch line becomes *xy* and *xz*, as there illustrated. Teeth drawn on this pitch line as for a spur gear may be laid out on the conical surface and used as the outlines of bevel gear teeth. Teeth so drawn are identical with those of the equivalent spur gear illustrated in Fig. 12, as will be seen when comparing it with Fig. 27. For the crown gear, rack teeth are wrapped around the surface of the cylinder.

Principles of Action of Bevel Gear Cutting Machinery

There are three principles of action commonly used for cutting the teeth of bevel gears, namely, the form tool, the templet and the molding-generating principles. There are two machines used to some extent

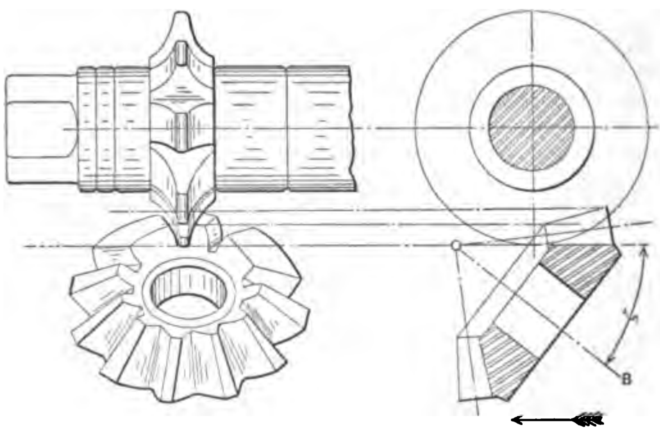

Fig. 28. Shaping the Teeth of a Bevel Gear by the Formed Cutter Process

in Europe which employ a fourth, that known as the odontographic principle. It is not in use in this country, so it will not be described here.

The *formed tool principle* is illustrated in Fig. 28, where a form cutter is shown shaping one side of the tooth of a bevel gear. The gear blank is tipped up to cutting angle ζ and fed beneath the cutter in the direction of the arrow. It will be immediately seen from an examination of the figure that the form tool process is by necessity approximate. It is evident that the right-hand side of the cutter is reproducing its own unchanging outline along the whole length of the face of the tool at the right. This form should not be unchanging for, as has been explained, the teeth and the space between them grow smaller towards the apex of the pitch cone, where they finally vanish, so it is evident that the outline of a tooth at the small end should be the same as that at the large end, but on a smaller scale—not a portion of the exact outline at the large end, as produced by the

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formed tool process and as shown in the figure. The method of adjusting the cuts to approximate the desired shape is described in the next chapter.

The Templet Principle: This principle is illustrated in Fig. 29, in skeleton form only. A former or templet is used which has the same outline as would a tooth of the gear being cut, if the latter were extended as far from the apex of the pitch cone as the position in which the former is placed. The tool is carried by a slide which reciprocates it back and forth along the length of the tooth in a line of direction (*OX*, *OY*, etc.) which passes through the vertex *O* of the pitch cone. This slide may be swiveled in any direction and in any plane about this vertex, and its outer end is supported by the roller on the former. With this arrangement, as the slide is swiveled inward about the vertex, the roll runs up on the templet, raising the slide and the tool so as to reproduce on the proper scale the outline of the former on the tooth being cut. Since the movement of the tool is

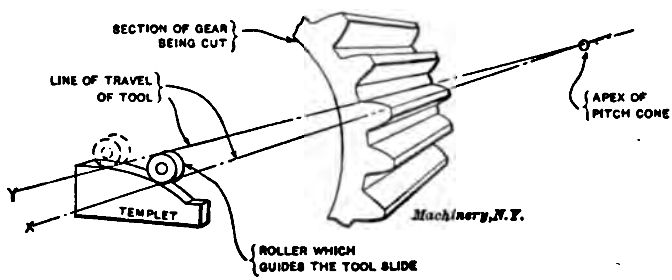

Fig. 29. Illustrating the Templet Principle for Forming the Teeth of Bevel Gears

always toward the vertex of the pitch cone, the elements of the tooth vanish at this point and the outlines are similar at all sections of the tooth, though with a gradually decreasing scale as the vertex is approached—all as required for correct bevel gearing.

The arrangement thus shown diagrammatically is modified in various ways in different machines, but the movement imparted to the tool in relation to the work is the same in all cases where the templet principle is employed, no matter what the connection between the templet and the tool may be.

The Molding-Generating Principle: Suppose we have a bevel gear blank made of some plastic material, such as clay or putty. By transposing Formula 84, $\sin \alpha_p = \frac{N_p}{N_g}$ to read $N_g = \frac{N_p}{\alpha_p}$, it is evidently possible to make a crown gear which will mesh properly with any bevel gear, such as the one we wish to form. If this crown gear and the plastic blank are properly mounted with relation to each other and rolled together, the tooth of the crown gear will form tooth spaces and teeth of the proper shape in the blank. This is the foundation principle of the molding-generating method.

In practice we have blanks of solid steel or iron to machine instead

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of putty or clay, so the operation has to be modified accordingly. Fig. 30 shows in diagrammatic form an apparatus for using the shaping or planing operation with the molding-generating principle. Here the crown gear is of larger diameter than is required to mesh with the gear being cut, and it engages a master gear keyed to the same shaft as the gear being cut, and formed on the same pitch cone. If the teeth of the crown gear, instead of being comparatively narrow as shown, were extended clear to the vertex *O*, they would mesh properly with the gear to be cut. The tooth is provided as shown having a line of movement such that the point of the tooth travels in line *OX*, which is the corner of a tooth of an imaginary extension of the crown gear. This crown gear has a plane face (see reference to "odontoid" form of tooth on page 20) and the cutting edge of the tooth is straight and set to mesh the face of the tooth. As it is reciprocated by suitable mechanism (not shown) the cutting edge represents a face of the imaginary crown gear tooth. If now, the master gear and crown gear are rolled together and the tool reciprocating starts in at one side of the

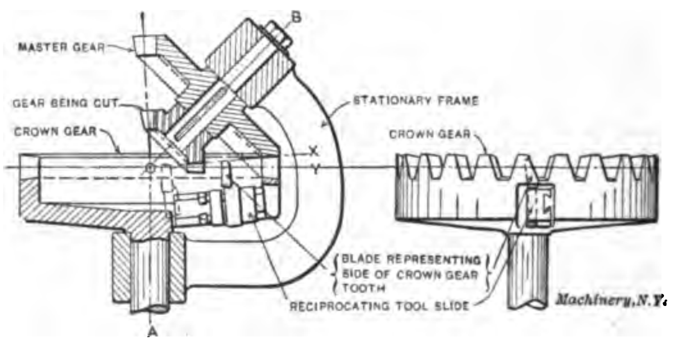

Fig. 30. Model Illustrating the Planing or Shaping Operation Applied to the Molding-Generating Principle of Forming Teeth of Bevel Gears

gear to be cut and passing out at the other, the straight cutting edge of the tooth will generate one side of a tooth in the gear to be cut in the same way as if the extended tooth of the crown gear were rolling its shape on one side of the tooth of a plastic blank. This simple mechanism has, of course, to be complicated by provisions for cutting both sides of the tooth, and for indexing the work from one tooth to the other so as to complete the entire gear. Arrangements have to be made also to make the machine adjustable for bevel gears of all angles, numbers of teeth and diameters within its range.

The use of the three principles illustrated in Figs. 28, 29 and 30 is not limited to the cutting operation shown for each case. In Fig. 28, for instance, a formed planer or shaper tool may be used as well as a formed milling cutter. Templet machines have been made in which a milling cutter is used instead of a shaper tool. This is true also of the molding-generating principle shown in Fig. 30.

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Machines for Cutting the Teeth of Bevel Gears by the Formed Tool Process

A very common method of using the formed tool for cutting bevel gears makes use of the ordinary plain or universal milling machine and adjustable dividing head. Cutting bevel gears by this method is described in the next chapter, so it will not be described here.

Most builders of automatic gear cutting machines furnish them, if desired, in a style which permits the swiveling of the cutter slide or of the work spindle to any angle from 0 to 90 degrees, thus permitting the automatic cutting of bevel gears by the formed cutter process.

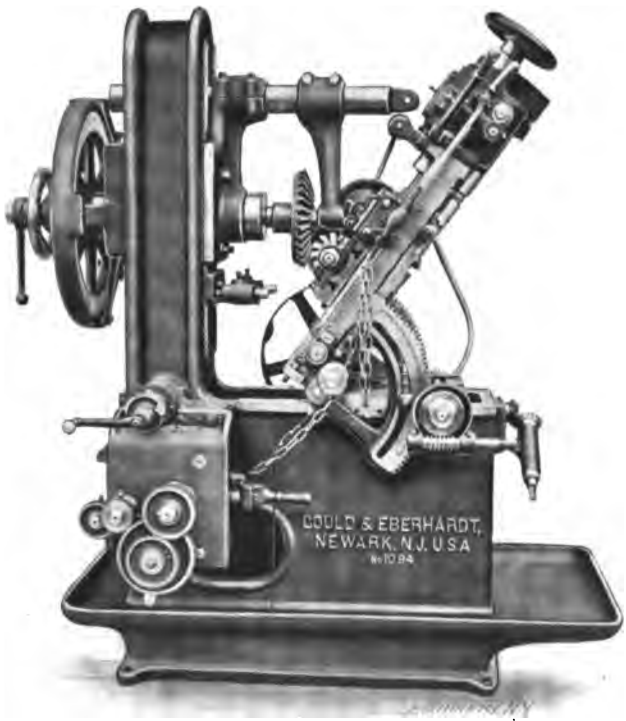

Fig. 31. Gould & Eberhardt Automatic Machine Cutting a Bevel Gear

An example of such a machine is shown in Fig. 31. Here the cutter slide is mounted on an adjustable swinging sector, as may be seen. As explained in the next chapter, it is necessary when cutting bevel gears, to cut first one side of the teeth all around and then the other. Between the two cuts the relation of the work and cutter to each other, as measured in a direction parallel to the axis of the cutter spindle, has to be altered. In the automatic machine this is effected by shifting the cutter spindle axially when the second cut around on the other side of the teeth is taken. Suitable graduations are provided for the angular and longitudinal adjustments.

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Bevel Gear Templet Planing Machines

The templet planing machine most commonly used in this country is shown in one of the smaller sizes in Fig. 32. The tool is carried by a holder reciprocated by an adjustable, quick-return crank motion. The slide which carries this tool-holder may be swung in a vertical plane about the horizontal axis on which it is pivoted to the head, which carries the whole mechanism of tool-holder, slide, crank, driving gearing, etc. This head, in turn, may be swung in a vertical axis about a pivot in the bed. The circular ways which guide this movement are easily seen in the illustration. The intersection of the vertical and horizontal axes of adjustment (which takes place in mid-air in front of the tool slide) is the point *O* in Fig. 29 where the templet

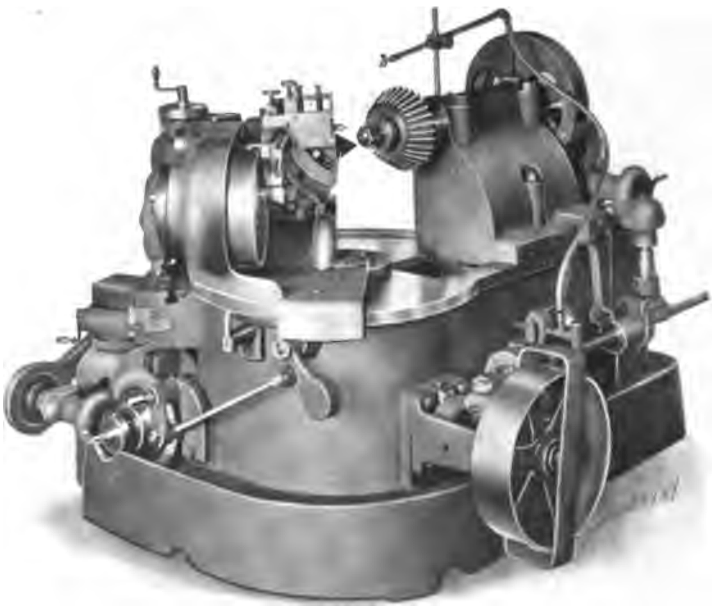

Fig. 32. Gleason Templet-Controlled Bevel-Gear Planing Machine

principle is shown in diagrammatic form. The blank is mounted on a spindle carried by a head which is adjustable in and on the top of the bed of the machine so that the apex of the cone of the gear may be brought to point *O* by means of the gages which are a part of the equipment of the machine.

Three templets are used, mounted in a holder attached to the front of the bed, on the further side in the view shown. The first of these templets is for "stocking" or roughing out the tooth spaces. It guides the tool to cut a straight gash in each tooth space, removing most of the stock. After each tooth space has been gashed in this fashion, the templet holder is revolved to bring one of the formed templets into position, and a tool is set in the holder so that its

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point bears the same relation to the shape of the tooth desired as the cam roll does to the templet. The head is again fed in by swinging it around its vertical axis, during which movement the roll runs up on the stationary templet, swinging the tool about its horizontal axis in such a way as to duplicate the desired form on the tooth of the gear. One side of each tooth being thus shaped entirely around, the holder is again revolved to bring the third templet into position. This has a reverse form from the preceding one adapted to cutting the other side of the tooth. A tool with a cutting point facing the other way being inserted in the holder, each tooth of the gear has its second side formed automatically, as before, completing the gear. The swinging

Fig. 33. The Bilgram Bevel Gear Generating Machine

movement for feeding the tool and the indexing of the work are taken care of by the mechanism of the machine without attention on the part of the operator.

Bevel Gear Generating Machines

The mechanism illustrated in outline in Fig. 30 is one that has been employed in a number of interesting and ingenious machines. The first application of this principle was made by Mr. Hugo Bilgram of Philadelphia, Pa. This form of machine in the hand-operated style has been used for many years. An example of a more recently developed automatic machine of the same type is shown in Fig. 33. The movements operate on the same principle as in Fig. 30, though in a modified form. Instead of rotating the crown gear and master gear

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together, the imaginary crown gear and, consequently, the tool, remain stationary so far as angular position is concerned, while the frame is rotated about the axis of the crown gear, thus rolling the master gear on the latter and rolling the work in proper relation to the tool. Instead of using crown and master gears, however, a section of the pitch cone of the master gear is used, which rolls on a plane surface, representing the pitch surface of the crown gear. The two surfaces are prevented from slipping on each other by a pair of steel tapes, stretched so as to make the movement positive. A still further change consists in extending the work arbor down beyond center *O* in Fig. 30, mounting the blank on the lower side of the center so that the tool, being also on the lower side, is turned the other side up from that shown in the diagram. All these movements can be followed in Fig. 33. As explained, a tool with a straight edge is used, representing

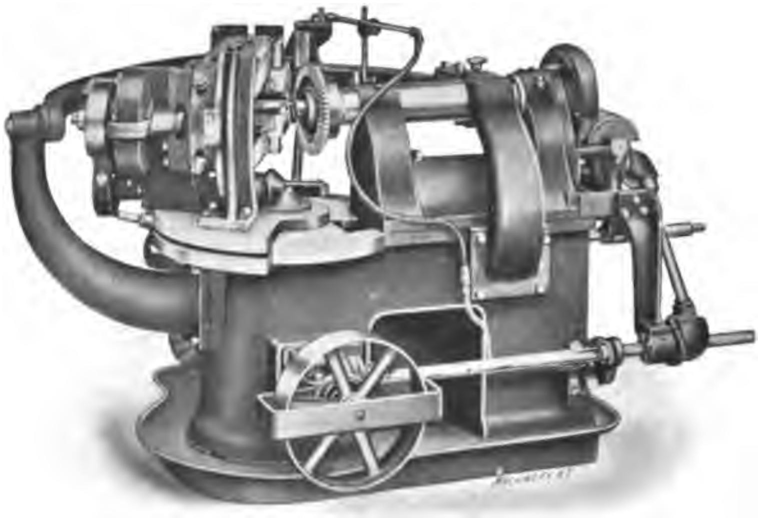

Fig. 34. Gleason Bevel-Gear Generating Machine

the side of a rack tooth, and this tool is reciprocated by a slotted crank, adjustable to vary the length of the stroke, and driven by a Whitworth quick return movement. The feed of the machine is effected by swinging the frame in which the work spindle and its supports are hung, about the vertical axis of the imaginary crown gear.

As stated, the machine is automatic. The operator sets the machine and places a previously-gashed blank on the work spindle and starts the tool in operation. The mechanism provided will, without further attention, complete one side of all the teeth. The machine may be then readjusted and the tool set for cutting the other side, which will be finished in the same automatic fashion. The mechanism does not operate on the principle of completing one side of one tooth before going to the next. It follows the plan of indexing the work for each

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stroke of the tool, the rolling action being progressive with the indexing so as to finish all the teeth at once.

The Gleason generating machine is shown in Fig. 34. It differs from the previous machine in employing two tools, one on each side of the tooth. The construction is identical with the mechanism in Fig. 30, in having the axes of the tool slides and of the blank fixed in relation to each other during the operation, the tool-holders and the blank rocking about their axes to give the rolling movement for cutting. The rocking is effected by means of segments of an actual crown gear and master gear. The segment of the crown gear is permanently attached to the face of the rear of the cutter slide frame, while the segment of the master gear (of which there are several furnished with the machine, the one used being chosen to agree with the angle of the gear to be cut) is clamped to the semi-circular arm pivoted at the outer end of the machine at one side, and fastened to the work spindle sleeve on the other. This arm is rocked by a cam mechanism and slotted link on the side opposite that shown in illustration.

The cycle of operations is as follows: The machine being adjusted properly in its preliminary position, the tool slide and the head on which it is mounted are swung back about the vertical axis so that the tools clear the work. The blank being set in the proper position, a cam movement swings the cutter slide head inward until the reciprocating tools reach the proper depth. The cam movement first mentioned now rocks upward the semi-circular arm extending around the front of the machine, rolling the blank and (through the segmental crown and master gears) the slide, until the tools have been rolled out of contact in one direction, partially forming the teeth as they do so. The arm is then rolled back to the central position and along downward to the lower position, until the tools are rolled out of contact with the tooth in this direction, completing the forming of the proper shape as they do so. The cam then rocks the arm back to the central position, where the cutter-slide head is swung back to clear the tooth, and the work is indexed, after which this cycle of operations is continued for the next tooth. It will be seen that by starting from the central position, going to each extreme and returning, all parts of each tooth are passed over twice, giving a roughing and a finishing chip. The machine is entirely automatic.

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CHAPTER VII

CUTTING THE TEETH OF BEVEL GEARS

Special directions for operating are furnished by the makers of molding-generating and templet planing machines. As these directions are usually adequate, and apply only to the particular machines for which they are given, this chapter will be confined to giving instructions for cutting teeth by the formed tooth method only, as performed on standard machine tools.

The Practicability of the Formed Tool Process

The first piece of instruction to be given in cutting bevel gears with a milling cutter is—don't do it. There are exceptions a-plenty to

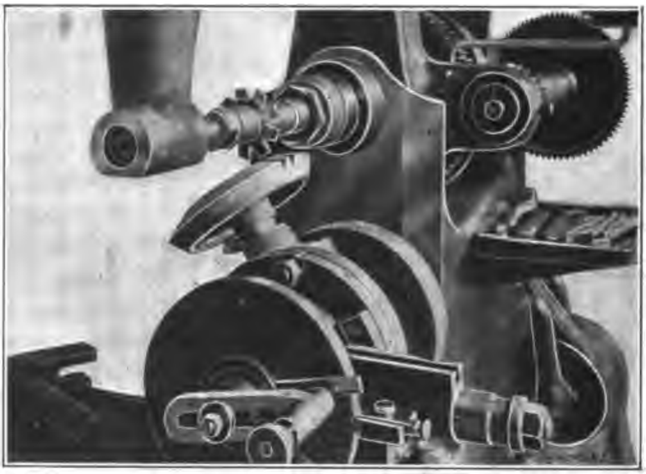

Fig. 35. Milling Machine Set up for Cutting a Bevel Gear

this rule, of course. For instance, gears too small to be cut on any commercial planing machine may be milled with a formed cutter; in general, it is not considered advisable to plane gears having teeth finer than 12 to 16 diametral pitch. It is allowable, also, to mill gears of coarser pitch which are to run at slow speeds or which are to be used only occasionally—such, for instance, as the bevel gears used for driving the elevating screws of a planer cross-rail, or those used in connection with any hand-operated mechanism. It is impracticable under ordinary conditions to mill teeth of bevel wears having teeth coarser than 3 diametral pitch, no matter what the service for which they are to be used.

Cutting Bevel Gears in the Milling Machine

The first requirement for setting up the milling machine to cut bevel gears, is an accurate blank. It is next to impossible to cut satisfactory

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teeth if the face and outside diameter of the gear vary from the proper dimensions. After the settings have been correctly determined for a properly-made blank, the accuracy of the rest of a lot of gears does not cut quite so much figure. Other points which have to be looked out for are the truth of the arbor on which the gear is mounted, and the selection of the proper cutter, as described on page 21.

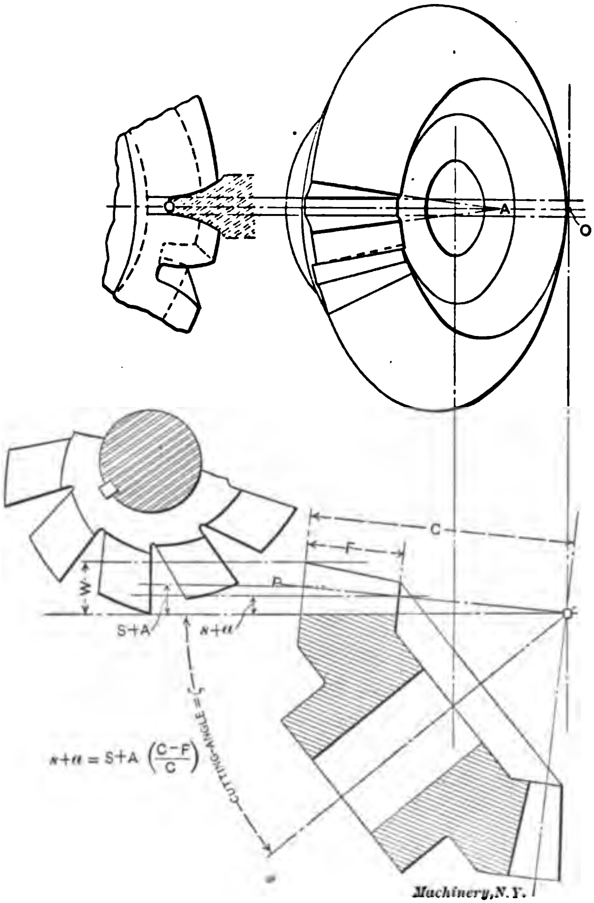

Fig. 36. Relative Positions of the Formed Cutter and the Blank when Taking a Central Cut

Fig. 35 shows the machine set up for cutting a bevel gear, and Fig. 36 shows in diagram form the relative positions of the cutter and the work. The spindle of the dividing head is set at the cutting angle, as shown, and the cutter (which has been centered with the axis of the work-spindle) is sunk into the work to the whole depth *W*, as given by the working drawing.

The centering may be done by mounting a true hardened center in the taper hole of the spindle, and lining up its point with the mark

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which will be found inscribed either on the top or on the back face of the tooth of the commercial gear cutter. A more accurate method is described in MACHINERY's Shop Operation Sheet, No. 1. Setting the cutter to the whole depth *W* is effected by passing the work back and forth under the revolving cutter and slowly raising it until the teeth of the cutter just bite a piece of tissue paper laid over the edge of the blank. This must be done after centering. The dial on the elevating screw shaft is set at zero in this position, and then the knee is raised an amount equal to the whole depth of the tooth, reading the dial from zero. This is evidently not exactly right, since the measurement should be taken in the direction of the back edge of the tooth, which inclines from the perpendicular an amount equal to the dedendum angle, as shown in Fig. 36. In practice, the slight difference in the value for the whole depth thus obtained is negligible.

Having thus mounted the work at the proper angle and having thus centered the cutter and set it to depth, two tooth spaces should next be cut, with the indexing set by the tables furnished with the dividing head to give the number of teeth required for the gear. Cutting these two spaces leaves a tooth between them on which trial cuts are made until the desired setting is obtained. The relative positions of the cutter and the work and the shape of the cuts thus produced are shown in the upper part of Fig. 36. It will be seen at once that this does not cut the proper shape of tooth. As explained in the first paragraph in Chapter VI, all the elements of the bevel gear tooth vanish at *O*, the vertex of the pitch cone—that is to say, the outer corners of the tooth space should converge at *O* instead of at *A*, and the sides of the tooth spaces at the bottom, instead of having the parallel width given them by the formed cutter, should likewise vanish at *O*. Our next problem is that of so re-setting the machine that we can cut gear teeth as nearly as possible like the true tooth-form in which the elements converge at *O*.

Offsetting and Rolling the Blank to Approximate the Shape of Tooth

There are a number of ways of approximating the desired shape of bevel gear tooth. Of these we have selected as most practicable the one in which the sides of the tooth at the pitch line converge properly toward the vertex of the pitch cone. Gears cut by this process will show, of course, the proper thickness at the pitch line when measured by the gear tooth caliper at either the large or the small ends. This method of approximation produces tooth spaces which, at the small end, are somewhat too wide at the bottom and too narrow at the top, or, in other words, the teeth themselves at the small end are too narrow at the bottom and too wide at the top. To make good running gears they must be filed afterward by hand, as described later. When so filed they are better than milled gears cut by other methods of approximation which omit the hand filing.

In the upper part of Fig. 37 is shown a section of the gear in Fig. 36, taken along the pitch cone at *PO*. It will be seen that the teeth at the

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large and the small ends, rotate the tooth against the cutter and take another cut until the proper thickness at either the large or small end has been obtained. If the thickness comes right at both ends the amount of set-over is correct. If it is right at the large end and too thick at the small end, the set-over is too much. If it is right at the small end and too thick at the large end the set-over is not enough. The recommended trial set-over (5 or 6 per cent of thickness of the tooth at the pitch line at the large end) will probably not be enough, so two or three cuts will have to be taken on each side of the trial tooth, as described, before the proper amount is found.

Having found the proper set-over, the cross feed screw is set to that

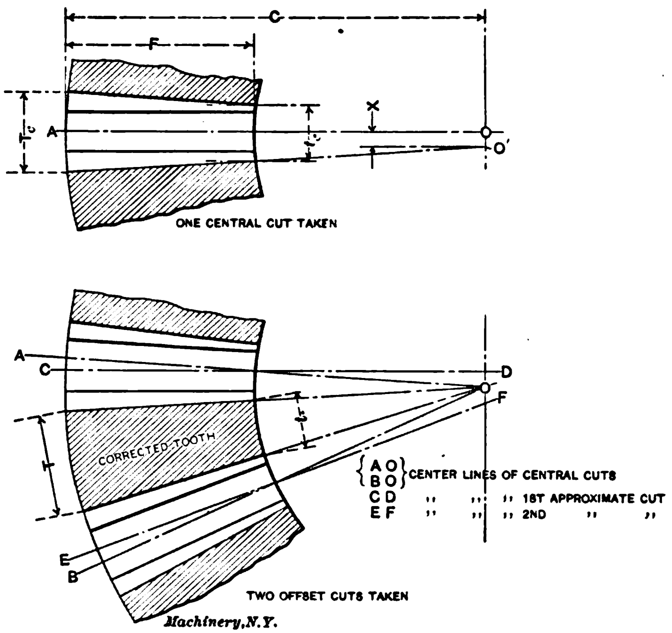

Fig. 37. Section on Pitch Cone Surface P-O of Fig. 36 Showing Central and Offset Cuts

amount and the cut is taken clear around the gear. Then the cross feed screw is set to give the same amount of set-over the other side of the center line and the work is rotated until the cutter matches the teeth spaces already cut at the small end and is run through the work. The tooth will generally be found too thick, and the work spindle is rotated still more until the tooth is of the proper thickness, when the gear is again cut clear around on this second cut.

The number of holes it was necessary to move the index pin on the dividing plate circle between the first and the second cuts to get the proper thickness of tooth, should be recorded. On succeeding gears it will thus only be necessary to take a first cut clear around with the work set over by the required amount on one side of the center line, and then a second cut around with the work set over on the other

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side of the center line, and rotate the index crank the number of holes necessary to give the proper thickness of tooth.

It will be noted that the shifting of the blank by the index crank is only used for bringing the thickness of tooth to the proper dimension. In some cases, particularly in gears of fine pitch and large diameter. this adjustment will not be fine enough—that is to say, one hole in the index circle will give too thick a tooth and the next one too thin a tooth. To subdivide the space between the holes, the little device shown in Fig. 38 may be used. This is mounted with the central plug entering the nearest hole in the outer row of circles in the index plate. The little disk may be rotated to bring any one of the holes in the spiral row in line with the stop pin in the dividing head. A number of subdivisions of the outer row of circles is thus provided. The plug in the disk does not enter the hole far enough to interfere with the pitch line converge, but meet at a point considerably beyond the vertex *O*. What we have to do is to move the cutter off the center, so that it will cut a groove one side of which would pass through *O* if extended that far. The amount by which the cutter is set off the center is known as the "set-over." We may take, for instance, for trial a set-

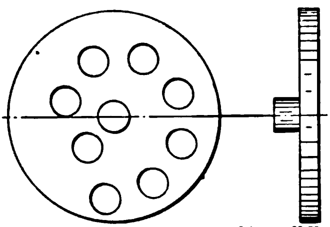

Fig. 38. Device for Subdividing the Space between the Holes on the Index Plate

over equal to 5 or 6 per cent of the thickness of the tooth at the large end. Move the face of the trial tooth away from the cutter by the amount of this trial set-over, having first, of course, run the cutter back out of the tooth space. Now rotate the dividing head spindle to bring this tooth face back to the cutter again, stopping it where the cutter will about match with the inner end of the space previously cut. Take a cut through in this position.

Next index the work to bring the cutter into the second tooth space and move the blank over to a position the other side of the central position by an amount equal to the same set-over, thus moving the opposite face of the trial tooth away from the cutter. Rotate the dividing head spindle again to bring this face toward the cutter until the latter matches the central space already cut at the inner end of the teeth. Take the cut through in this position.

Now with vernier gear tooth calipers or with fixed gages machined to the proper dimensions measure the thickness of the tooth at the pitch line at both large and small ends (the values for the addendum and the thickness of the pitch line at both ends of the totth are given by rules 5, 8, 10 and 11). If the thickness is too great at both the

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plunger on the index crank. Many index heads are provided with a fine adjustment which makes this device unnecessary. Every manufacturer should provide such an adjustment.

In large gears it is best to take the central cuts shown in Fig. 36 clear around every blank before proceeding with the approximate cuts. This gives the effect of roughing and finishing cuts, and produces more accurate gears. The central cuts may be made in a separate operation with a roughing or stocking cutter if desired. It might also be mentioned that it is common practice to turn up a wooden blank for making the trial cuts shown in Figs. 36 and 37, to avoid the danger of spoiling the work by mistakes in the cut-and-try process.

Filing the Teeth

The method of cutting bevel gears just described requires the filing of the points of the teeth at the small end. This can be done "by the

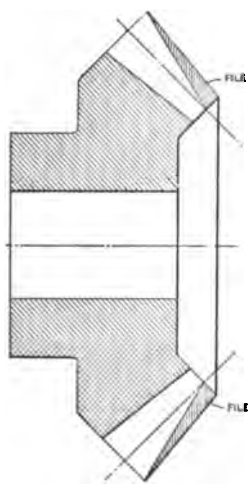

Fig. 39. The Surfaces to be Filed in Fitting Bevel Gears Cut with a Formed Cutter

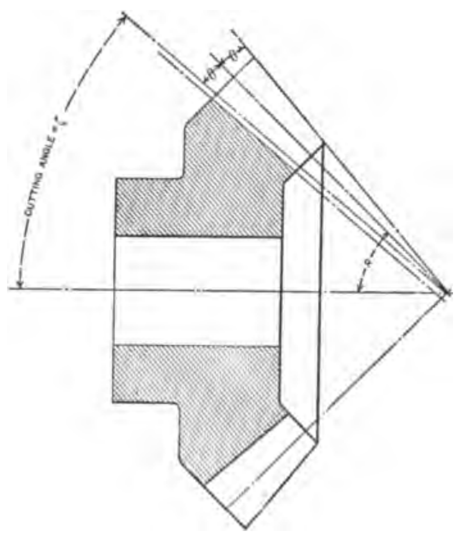

Fig. 40. Cutting Angle and Parallel Clearance, Recommended by Brown & Sharpe for Bevel Gear Cut with a Formed Cutter

eye" very skillfully when the workman is used to it. The operation consists in filing off a triangular area extending from the point of the tooth at the large end to the point at the small end, thence down to the pitch line at the small end and back diagonally to the point at the large end again. This is shown in Fig. 40 by the shaded outline. Enough is taken off at the small end of the tooth so that the edges of the teeth at the top appear to converge at vertex *O*.

The bevel gears may be tested for the accuracy of the cutting and filing by mounting them in place in the machine and revolving them at high speed, or by mounting them in a testing machine made for the purpose. The marks of wear produced by running them together

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under pressure, with the back faces flush with each other, should extend the whole length of the tooth at the pitch line. If it does not, the amount of set-over allowed in cutting them was at fault, being too little if they bear heavily at the large ends, and too much if they bear heavily at the small ends. The bearing area should also be fairly evenly distributed over the sides of the teeth above the pitch line, from the large to the small end. If it is not the filing is at fault The marks of wear will not extend far below the pitch line in a pinion of few teeth.

It is possible to get along without filing by decreasing the amount of set-over so as to make the teeth too thin at the pitch line at the small end, when they are of the right thickness at the large end. This does not give quite as good running gears, however, as when the method just described is followed.

Cutting Bevel Gears on the Automatic Gear Cutting Machine

The directions for cutting bevel gears on the milling machine apply in modified form to the automatic gear cutting machine as well. The set-over is determined in the same way, but instead of moving the work off center, the cutter spindle is adjusted axially by means provided for that purpose. Some machines are provided with dials for reading this movement. The cutter is first centered as in the milling machine, and then shifted—first to the right, and then to the left of this central position.

The rotating of the work to obtain the proper thickness of tooth is effected by unclutching the indexing worm from its shaft (means usually being provided for this purpose) and rotating the worm until the gear is brought to proper position. Otherwise the operations are the same as for the milling machine.

The Brown & Sharpe Co. recommend a somewhat different angle setting for the gear from that shown in Fig. 9, as giving more nearly the true form of tooth than that we have described, and thus requiring less filing. In this case the cutting angle is determined by substracting the *addendum angle* from the pitch cone angle, instead of by subtracting the dedendum angle as in Rule 15. In other words, the clearance at the bottom of the tooth is made uniform, as shown in Fig. 40, instead of tapering toward the vertex. Aside from the fact that the head is set at a different cutting angle, no change in procedure from that described is necessary.

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No. 38

GRINDING AND LAPPING

CONTENTS

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CHAPTER I.

GRINDING AND GRINDING MACHINES.

The development of the grinding machine has made rapid progress during the last few years, and the process of grinding is more and more recognized as having both economical and technical advantages, as compared with the old methods of obtaining finish. This is especially true regarding plain cylindrical grinding, and this is due chiefly to the fact that the machines for this kind of grinding are easier to build, and in general more efficient, than machines for other kinds of grinding.

It has often been claimed, by people who have had long and thorough experience in regard to this subject, and whose testimony, therefore, must be considered as having weight, that time can be saved in finishing a cylindrical piece of work by taking a roughing cut with an ordinary cutting tool, leaving about from 0.008 to 0.010 inch, and grinding off this amount, instead of taking a second cut in the lathe, and finishing the piece by filing. One great advantage of the grinding machine is the closer finish that can be obtained.

In some special cases, when the steel to be finished is so hard that it cannot be cut by means of a cutting tool, the grinding machine has to take the place of the lathe entirely. Of course, the work in this case cannot be done so cheaply as in the case of ordinary kinds of steel, but still it can be done with fair economy. As the piece is taken entirely rough and put up in the grinding machine, there is, considering the errors in casting, about 1/8 inch up to 3/16 inch on the diameter, that has to be ground off. When so large an amount of metal has to be removed by grinding, another problem than that dealt with when only 0.008 to 0.010 inch has to be ground off presents itself. The writer at one time designed three special grinding machines, two for external and one for internal work, all for very heavy duty. Herein are given a few of the conclusions arrived at while designing these machines. Being, as mentioned, mostly used for heavy grinding, the machines may differ some from the common light grinding machines, but the principles remain, in general, the same.

Any machine tool must, of course, be designed heavy enough not only to take all the strains produced by the action of the cutting tool or wheel, but to prevent all, or nearly all, vibration and chattering of the machine itself. This is true of the grinding machine more than of any other machine tool. Rigidity is a very important factor in the efficiency of the machine, both in regard to heavy grinding and grinding for very exact sizes and high finish.

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Influence of Vibrations on Action of Grinding Machines.

The grinding wheel rotating at a high-speed tends to jar its bearings and supports. Vibrations of this kind would result in an oscillating motion of the grinding wheel perpendicular to its own axis of rotation and along the line connecting the center of work with the center of the wheel. The frequency of these vibrations depends entirely upon the weight of the oscillating parts. The cause of the vibrations is that the center of gravity of the rotating parts, grinding wheel, shaft, pulley, etc., is not entirely the same as the center line of rotation. This is partly due to the uneven structure of the material. It is very plain to everybody, that the oscillating grinding wheel cannot cut to its full capacity. The length of the oscillations

TABLE I. SPEED OF EMERY WHEELS.

Diameter of Wheel, inches.	R. P M. for Surface Speed of 5,000 feet	R. P. M. for Surface Speed of 6,000 feet
1	19,099	22,918
2	9,549	11,459
4	4,775	5,730
6	3,183	3,820
8	2,387	2,865
10	1,910	2,292
12	1,592	1,910
14	1,364	1,637
16	1,194	1,432
18	1,061	1,273
20	955	1,146
24	796	955
30	637	764

might not be large, perhaps only one-thousandth of an inch or a fraction thereof, but the cut will be just so much deeper one moment than the next following. Only at one moment, when the wheel is furthest in, will it cut to its full capacity.

It is very important, in order to secure nice running of the wheel, to have the belts in good order, and to have the boxes closely adjusted, even though they run a trifle warm. Because of the high speed of the shaft, the boxes ought to be made with ring-oiling devices. This would allow a closer adjustment, and secure a better running of the shaft. However, as far as the writer knows, there are no grinding machines on the market equipped with ring-oiling boxes. The slides should, for the same reason as the boxes, be adjusted closely, even though they slide hard.

Speed of Grinding Wheels.

The peripheral speed of the grinding wheel should be approximately from 5,000 to 5,500 feet per minute. There are occasionally cases when higher speed is desirable, but with higher speed there is danger of the wheel breaking. The wheel should, however, never be run slower than 5,000 feet per minute, because it becomes less efficient at slower speeds.

Above will be found a table which gives the number of revolutions per minute for specified diameters of wheels to cause them to run at the respective periphery rates of 5,000 and 6,000 feet per minute.

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Experience has shown that for grinding work with fairly large diameter, better results are obtained by using a comparatively small wheel than by using one with too large a diameter. The explanation of this fact is that the wheel of smaller diameter clears itself faster from the work, while the larger one has a larger contact surface, and, therefore, the specific pressure between wheel and work becomes reduced, and the metal removed by the wheel stays too long a time between the wheel and work, and prevents the particles of the wheel from cutting properly into the work. The peripheral speed must, however, be the same for the smaller wheel as for the larger one.

Surface Speed of Work.

The proper surface speed of the work varies somewhat with the material and kind of work to be done. The grinding machine builders recommend 15 to 30 feet as a good average speed range for ordinary kind of work. For cast iron this can be slightly increased. The writer has had experience in grinding a very tough and hard steel (manganese steel), and has found the right surface speed in this special case to be as low as 6 to 8 feet a minute for rough grinding. For finishing grinding, the speed should be somewhat higher than for rough grinding. For delicate work the speed should be slow, because the work could easily be damaged by forced grinding.

As a general rule, for determining the surface speed for a certain kind of material, one can say that a brittle material, as cast iron, takes a high speed, while a tough and hard material, as the best tool steel, takes a slow speed. For grinding close to size and for high finish, the depth of the cut must be small, and higher surface speed can consequently be used.

Many of the grinding machines on the market are built so as to have the work revolving on two dead centers. This is done more for the sake of being able to obtain accuracy than for the sake of increasing the cutting efficiency of the machine.

Traverse Speed of Grinding Wheel.

The traverse speed of the grinding wheel should for ordinary grinding be three-fourths of the width of the wheel, that is, for one revolution of the work the wheel should travel three-fourths of the width of the face. If the wheel be traversed slower, the new cut is overlapping the old one more than necessary, and too large a part of the wheel is idle. It is, however, necessary that the new cut overlaps the old one with about one-fourth of the width of the face, because the edges of the face easily become rounded off, and, if the travel be too rapid, the result is an uneven surface.

The capacity of the wheel, within certain limits, of course, is proportional to the width of the face. A certain specific pressure between wheel and work is required for the highest cutting capacity. A wider wheel requires consequently a larger total pressure. But many of the machines now on the market are not rigid nor heavy enough to stand the pressure needed for a fairly wide wheel, cutting at full load, without vibration and chatter. The grinding machines on the

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market have not, in the writer's opinion, yet reached their full capacity. Wider wheels should be used, and the machines be designed and built heavier in order to take the load of the cutting wheel, without perceptible vibration of the machine.

For the final smooth finish, a slower traverse speed should be used, especially if the face of the wheel is not kept a perfectly straight line. A smoother surface is obtained by using a slower traverse speed. The part of the wheel which is overlapping, while theoretically it does not cut, still wears away the unevenness left from the first cut, and, by this, to some extent polishes the surfaces.

While grinding a plain cylindrical piece of work, the grinding wheel should not be allowed to travel too far past the ends of the piece before reversing; it is only waste of time. The wheel should be reversed when three-fourths of its width is past the end of the work.

Depth of Cut.

The depth of the cut to be taken depends upon the material, kind of wheel, and the work done. It should be deep enough to permit the wheel to do its utmost. This is, of course, true only about pieces that are rigid enough to stand a heavy cut. The grinding operator himself will have to determine the depth of cut for each individual case, judging it by the prevailing conditions of work, machine, and wheel.

When the piece to be ground, owing to the hardness of the material, cannot be roughly finished by a cutting tool before being placed in the grinding machine for the final finish, there is often up to 3/16 inch on the diameter to be removed by grinding. Employing the same principle as when the piece is previously to the grinding operation roughly turned in a lathe, the work should first be put up in a machine equipped with a coarse and wide grinding wheel. A wheel of this kind is capable of removing stock rapidly. The piece should be finished to within 0.005 inch of the finished diameter in this machine, and then moved to a machine equipped with a finer grain wheel, and the final finish given to it.

The Grinding Wheel.

For heavy grinding, the alundum wheel is the best for removing stock rapidly. The carborundum wheel will give a smoother finish, and is to be recommended for the large majority of other classes of grinding. Emery is less abrasive, but gives a higher polish. Most grinding wheel manufacturers recommend their medium grade, *M*.

The question as to what is the very best wheel for finishing any particular piece cannot be definitely answered. On the next page is given a table of wheels which can with advantage be used in the cases mentioned. This table is recommended by one of the largest grinding machine manufacturers.

Grit No. 24 may be too coarse for any but rough classes of work, but if mixed with No. 36 it gives a fair result. No. 30 used sepa-

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rately is capable of a very fair commercial finish, but if mixed with No. 46 will give as fine a finish as is desired by the majority of the grinding machine users, and at the same time it retains the rapid cutting capacity. Nos. 46 and 60 are as fine as is necessary for almost any manufacture, although finer than these are used by some concerns who require a very high gloss finish.

A satisfactory grinding wheel is an important factor in the production of good work. In machine grinding, it is desirable, in order that the cut may be constant, and give the least possible pressure and heat, to break away the particles of the wheel after they have become dulled by the act of grinding. It is the faculty of yielding to or resisting the breaking out of the particles which is called grade. The wheel from which the particles can be easily broken out is called soft, and the one that retains its particles longer is called hard. It is evident that the longer the particles are retained the duller they will become, and the more pressure will be required to make the wheel cut. Retaining the particles too long causes what is familiarly known as glazing. A wheel should cut with the least possible pressure, to effect which it must always be sharp. This is maintained by the breaking out of

TABLE II. GRADE OF WHEEL TO USE FOR DIFFERENT MATERIALS.

Material		Grit No.	Grade
Soft Steel	Ordinary shafts....	24 to 60	Medium.
	Steel tubing or very light shafts......	24 to 60	Two or three grades softer than medium.
Tool steel or cast iron.....		24 to 60	Medium or one grade softer.
Internal grinding		30 to 36	Medium or several grades softer.

particles. Therefore, a wheel of proper grade cutting at a given speed of the work possesses "sizing power," or ability to reduce its size uniformly without breaking away its own particles too rapidly; obviously if the work is revolved at a higher speed, the particles will be torn away too fast, and the wheel will lose its sizing power.

Tho properties of toughness and hardness of the material to be ground has a retarding influence on the grinding because it makes the material stick to or clog the wheel. The ground-off material, instead of being thrown away from the wheel by the centrifugal force, gets in between the particles of the grinding wheel. It is self-evident that this has a greatly retarding effect on the cutting quality of the wheel. A brittle material, on the contrary, does not have the tendency of clogging the wheel, but the stock ground off is immediately thrown away from the wheel, leaving the particles free to cut without the retarding action of undue friction, and generation of more than the due amount of heat. If we take into consideration only these properties of the material to be ground, the tough or leady material requires a soft wheel, because the particles must break away fast enough to prevent the wheel from being clogged. In this case, the particles do not wear enough to become dull, but must break away

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before this. When grinding a brittle or hard material, on the contrary, the wheel is less liable to be clogged, the particles do not need to break away so soon, and, therefore, a harder wheel should be used. However, the wheel must not be so hard that the particles get too dull and become inefficient as cutting agents before they break away.

Importance of Wheel Running True.

In order to obtain the full efficiency of the grinding wheel, it must be run perfectly true; that is, cut evenly all the way around. The grinding wheel detects its own errors. A slight difference in the

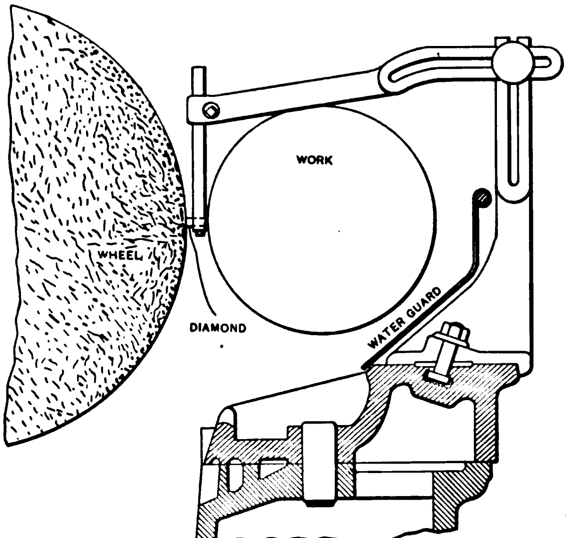

Fig. 1. Fixture for Truing Emery Wheel with Diamond.

sparks indicates that the wheel is out of true. The eccentric wheel has about the same kind of action as the one which is vibrating because of too weak supports. Furthermore, the edge of the grinding wheel should be kept perfectly straight. If the edge be curved, however slightly, a curved cut will be the consequence. Many grinding machines give inefficient results because the edge of the wheel is not kept in a true straight line. The operator seldom appreciates the great importance of this, and, therefore, the foreman should watch the men closely in regard to this point.

The best tool for truing the wheel is the diamond, but, this being rather expensive for shops where not very much grinding is done, the usual emery wheel dresser can be used to good advantage. In truing the wheel, the dressing tool should be kept stationary and rigidly supported, and the wheel should be traversed back and forth, until a true edge is obtained. Fig. 1 shows a fixture and arrangement for wheel truing with a diamond.

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Wet and Dry External Grinding.

Nearly all plain cylindrical grinding is now done wet. There are many reasons why the wet method is to be preferred to the dry. Because of the friction between the grinding wheel particles and the work, as well as between the cut-off material before it leaves the wheel and the work, more or less heat is generated. If this heat is not carried away, the work will be burned. Besides, the edge of the grinding wheel would be highly heated, but the center would still remain comparatively cool, and the outside would expand and there would be danger of the wheel breaking. It is found that the water has a softening effect upon the wheel, therefore a harder wheel is required for wet grinding than for dry.

Machines with Two Grinding Wheels.

The grinding machines on the market are equipped with only one grinding wheel, but there is no reason why two grinding wheels cannot be employed to advantage. In this case one wheel is to operate on each side of the work. As both of the wheels are to throw the sparks and the water down, one of the wheels has to cut with the revolving of the work, that is, the peripheries of the wheel and the work are going downwards. This is, of course, not the ideal condition, but, when the work is revolving at a slow peripheral speed, there is not much difference in the cutting capacity of the two wheels.

It is self-evident that, when employing two wheels, one at each side of the work and just opposite each other, the traverse speed of the wheels must be twice as fast as in the case of only one wheel, or three-fourths of the width of the wheel for one-half revolution of the work. Otherwise one wheel will overlap the cut of the other.

The two machines for external grinding which the writer designed, have two wheels working according to the principle previously described. Fig. 2 gives an idea of the arrangement used on one of these machines. The principal features of the design can be studied direct from the illustration without any further comments.

One new feature of these machines is that each grinding wheel is driven independently by a motor. This mctor is mounted above the wheel spindle, and is belted directly to same. Special attention has been paid to designing the support of the motor in order to prevent the vibrations of the motor from being transferred to the grinding wheel.

Internal Grinding.

The development of internal grinding is not, by far, so advanced as that of external grinding. To be sure, there are a few machines and fixtures on the market that are designed and built for the internal grinding of holes of various kinds, but the machines suffer from lack of rigidity, and some of the most conscientious grinding machine builders do not recommend them very highly, but admit their inefficiency for removing any ccmparatively large amount of stock. It has even gone so far that one man holding a prominent position with one

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of the largest grinding machine manufacturing concerns in the country has said, that in his opinion, the internal grinding machine is a mistake from start to finish, and that it will never be made a success.

This state of affairs has not come about without good reason. As we already have seen, the rigidity of the arrangement for supporting the grinding wheel is a very important factor for all efficient grinding.

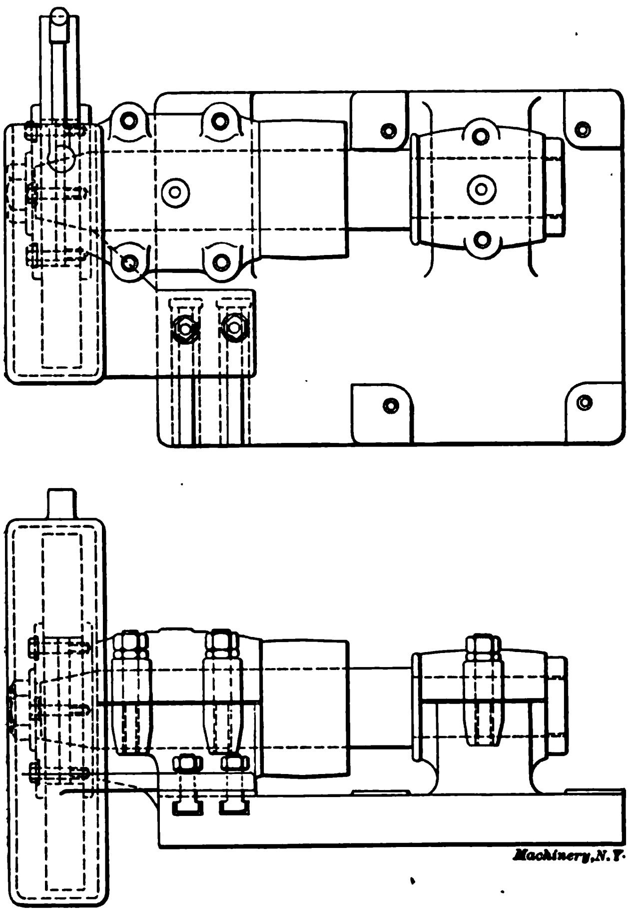

Fig. 2. Grinding Head for External Grinding Machine.

But the internal grinding machine does not very well lend itself to the employment of any rigid and heavy fixtures, and the grinding wheel must necessarily be small, and therefore lacks the strength to stand a heavy cut. The designer, when designing the fixtures for internal grinding, has an entirely different problem to solve than when designing those for external grinding, where it is comparatively easy to obtain ample rigidity. The internal grinding wheel must be mounted at the end of a small spindle which projects past the bearing far enough

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to enable the wheel to reach past the end of the hole to be ground. Such a spindle rotating at a high speed is liable to vibrate, especially if pressure be applied at the end of it, as is here the case.

Sometimes, however, it becomes absolutely necessary to grind, internally, even a comparatively large amount of stock. This is the case, when finishing manganese steel, this material being so hard that it cannot be cut by any kind of tool steel. Take the case of bores of manganese steel car wheels. As the grinding of the bores must be done without any stock having previously been removed from the rough casting, on the average about one-eighth inch of metal must be ground off from the hole. All the errors in the cored hole, as eccentricity in reference to the circumference of the wheel, etc., must be corrected by grinding. A hole cored in a manganese steel casting is always comparatively much rougher than a hole cored in cast iron, and all this must be taken into consideration, when determining the amount of stock to leave for the grinding process.

Design of Heads for Internal Grinding.

The fixture used in the internal grinding machine designed for grinding these wheels is shown in Fig. 3. Internal grinding fixtures gen-

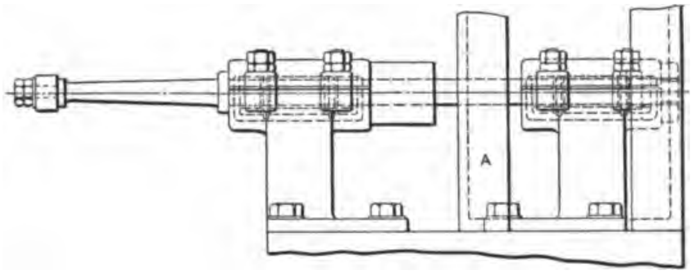

Fig. 3. Grinding Head for Internal Grinding.

erally have a long extension bearing, as shown in Fig. 4. This serves to support the spindle as near to the grinding wheel as possible; but the diameter at the root of this extension, that is, nearest to the box, cannot exceed the diameter of the grinding wheel.

The spindle, shown in Fig. 3, is made solid, and has the largest diameter possible for the size of the grinding wheel. An increased amount of rigidity and a greatly increased simplicity is gained by this design.

When working, the grinding wheel produces, especially in dry grinding, very much dust. When inside a hole the dust cannot very easily get away, but whirls about in the hole. If the spindle has a bearing near to the grinding wheel, the dust will find its way into the journal. This drawback is entirely eliminated by having a large solid spindle without a bearing near to the grinding wheel.

As to the relation between the overhanging part of the spindle and the distance between centers of the boxes, there are many factors that come into consideration in regard to this relation, such as the design

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of the boxes, the diameter of the spindle, how close the spindle can be allowed to run in the boxes, etc. However, the distance between the centers of the boxes should be made as large as the general design conveniently permits.

Fig. 3 shows at *A* the support for the motor. This support is placed on the top of the top rest. The driving pulley is placed between the bearings, so that the support could be made as rigid as possible.

It was found by actual experience with these fixtures, that when the grinding wheel was taking a fairly heavy cut the spindle did not vibrate nearly so much as when the wheel was running idle. The springing quality of the spindle, and the pressure between work and wheel, made the wheel cut without any chattering worth mentioning.

Regarding the peripheral speed of the grinding wheel, what has already been said with reference to external grinding is equally applicable to internal grinding.

Because of the lighter fixtures, the speed of the work should be slower than for external grinding. The writer has found the right cutting speed for hard and tough steel to be, for heavy grinding, about

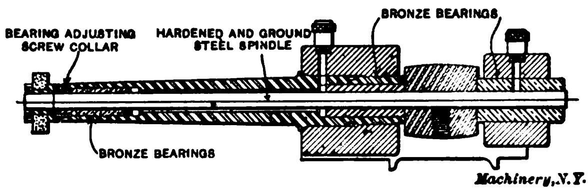

Fig. 4. Common Construction of Grinding Heads for Internal Grinding.

seven feet a minute. For the finishing, the speed can, with advantage, be somewhat higher. The wheel should travel three-fourths of its width for one revolution of the work, the same as for external grinding.

Wet and Dry Internal Grinding.

One point that has been much discussed in regard to internal grinding, is whether it shall be conducted wet or dry. Some grinding machine designers have advanced the opinion that it, by all means, must be done dry, but others claim the wet method to be superior. For light finishing grinding one method might be considered as good as the other, because so small an amount of heat is generated that there is no danger of burning the material or breaking the wheel. But, for heavier grinding, a considerable amount of heat is generated, and it becomes necessary to carry it off by water. At least, such is the writer's own experience on this subject. At a test recently conducted to find out the actual difference between dry and wet internal grinding, it was found that the cutting quality of the grinding wheel was about the same in both cases, but, with a heavy feed and dry grinding, the work was highly heated, and the wheel broke after about half an hour's run, while, with wet grinding, the wheel stood the heavy cut continuously without breaking.

The water can be injected into the hole in a stream about 1/16 inch

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in diameter. In addition to carrying away the heat, the water serves to wash away the removed stock from the hole.

Tests have been undertaken on the above mentioned internal grinding machines, in order to find out the time required to grind the bores of a certain kind of manganese steel car wheels. Two different kinds of wheels were tested. The first one, a 20-inch diameter wheel, had a bore 2⅞ inches in diameter and 5⅞ inches long, and it was to be ground for a press fit. The second one, an 18-inch diameter wheel, had a bore 3¼ inches in diameter and 4½ inches long, and was also to be ground for a press fit. Four wheels of each kind were ground during the course of the test, and it was found that the actual time for the grinding operation, not including the time required for putting up the work in the machine, was, for the first kind of wheels 1 hour and 23 minutes for all four, and for the second kind, 1 hour and 9 minutes for four wheels. Considering that the bores of the wheels were not previously turned, but entirely rough, as the wheels were taken directly from the foundry, and considering the hardness and toughness of the steel, the results obtained were considered good. The time of putting up the work in the machine was about 6 to 8 minutes for each wheel. As the machines work automatically, one man is able to run three machines. Counting 8 minutes for the putting up of each wheel, the man is able to grind one wheel of the first kind in 30 minutes, and one wheel of the second kind in 26 minutes.

The work was revolved at a speed of 7.7 revolutions per minute. This makes a peripheral speed, for the first case, of 5.8 feet per minute, and for the second case, of 6.6 feet per minute. The grinding wheel used was a 2-inch diameter, 1-inch face, No. 46 grit, O grade alundum wheel. It was run at a speed of 4,750 feet per minute.

The traverse speed of the work was as high as 0.84 inch per revolution of work. This allowed the wheel to overlap the old cut by only 0.16 inch, but, as the grinding wheel was trued very carefully, this was found to be all that was required for obtaining a nice smooth surface. The traverse feed was not slowed down, but remained the same while doing the final finishing, and a very satisfactory finished hole was obtained. The test was made throughout with wet grinding.

For heavy cylindrical grinding, which has especially been referred to, the width of the wheel used varies between 1½ and 2½ inches, regardless of the diameter. In some special cases narrower wheels than 1½ inch are used, but these special cases are exceptions to the general practice, and must be recognized as such by the machine builders and users. Although larger wheels are used, there is no doubt that the best range of diameters of wheels is between 12 and 18 inches. For how wide a wheel the grinding machine in the future can be designed, has yet to be decided; but, wider wheels and heavier machines point the direction of the road which the designer and machine builder should follow for the development of the grinding machine.

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CHAPTER II.

THE DISK GRINDER.

In any machine shop or department of a manufacturing plant where tools for manufacturing operations are made, a properly designed and equipped disk grinder should be considered almost indispensable. For a large portion of the operations most commonly done with a file, and many that are considered surface grinder, milling machine or shaper jobs, can be done better and quicker, and at less cost for files, cutters, etc., with a disk grinder.

As a simple example we will take the case of a piece of tool steel needed, say for a box tool, a back rest, a cutter, or a forming tool, to be say, ¼ inch thick, 1 inch wide, 2 inches long, ends and sides straight and square all around. Probably the bar steel ¼ inch × 1 inch will be enough oversize to grind on a disk to exact size, but not enough oversize to work with a milling machine, shaper or surface grinder. Even if larger stock, say 5-16 inch × 1⅛ inch, or a forging, is used,

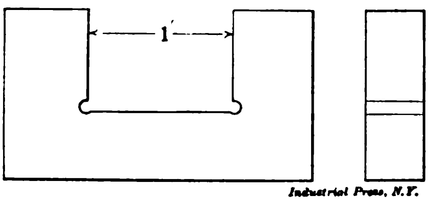

Fig. 5. Snap Gage Finished on Disk Grinder.

it is only necessary to rough one flat side and one edge down fairly close to size and finish all over on a disk grinder. For squaring the ends of one piece like this, and bringing it to exact length, the saving in time over the common way is considerable. Suppose this piece has to be hardened, and after hardening must fit a certain space. It will need truing up after hardening, and here again the disk grinder proves its adaptability.

Regarding the degree of accuracy obtainable with a disk grinder, an example may be of interest. An experienced tool maker was with an exhibit of disk grinders at a fair. Having plenty of time on his hands, he employed a part of it in grinding up six steel pieces, each a one-inch cube. He got the pieces planed roughly in the bar, a little oversize, and sawed off a little long. On his spare time he ground them to one-inch cubes, measuring them with a 1-inch micrometer caliper. When he had finished with them there was no point on any of the cubes that varied more than 0.00025 inch from 1 inch. Packing them together with any combination of sides, the greatest variation from 6 inches as measured with a 6-inch micrometer was 0.0005 inch. All

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the sides of all the cubes were so nearly square with each other that no error could be detected with a hardened steel square. In grinding these cubes no fixture or clamp of any kind was used. They were laid on the swinging table, and against the rib and pressed against the wheel with the fingers.

A few examples of the application of the disk grinder to tool-room work will give a general idea of its application. Suppose a snap gage such as shown in Fig. 5 is to be made. With the disk grinder the

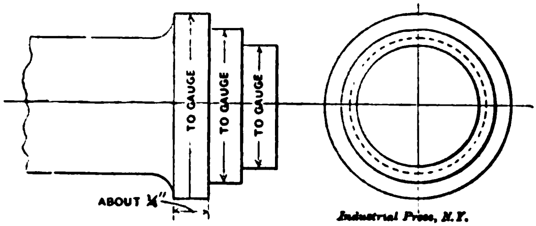

Fig. 6. Work to be Gaged.

gage can be finished all over, sides, edges and ends, and corners beveled or rounded. In hardening the gage springs somewhat, but can easily be squared again on the disk grinder. We are now ready to grind the notch to size. Lay the piece on the swinging table, with the back edge against the rib, the wheel being in the notch. The piece is now ground on both sides without turning it over. This will make the faces of the notch parallel with each other, which they might not be if the piece were turned over. By the use of an end measure gage the snap gage in Fig. 5 is now easily completed.

In a certain shop a job came up to be done in the turret machine.

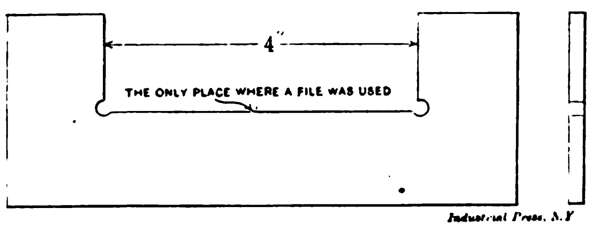

Fig. 7. Snap Gage Ground to Size on Disk Grinder.

A number of cast iron pieces, of the shape shown in Fig. 6, were to be machined. There were eight different sizes of pieces and three dimensions made to gage on each piece, making 24 dimensions in all. The largest dimension on the largest piece was about four inches. The smallest dimension on the smallest piece was about ¾ inch. A few thousand pieces of each size were to be made. Extreme accuracy was not required; a variation of 0.001 inch was allowable. A tool-maker was given the job of making a set of snap gages.

Taking the figures, he made 24 end measure pieces from 5-16-inch round drill rod, hardened them, and marked the size. He then cut

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the gages from ⅛-inch thick sheet steel, as shown in Fig. 7. The working faces were hardened and ground to the end measure pieces on the disk grinder, and the edges squared and the corners rounded in the same machine. The gages were not touched with a file except to smooth off the edge in the bottom of the notch.

The examples given indicate the use of the disk grinder as a tool-

Fig. 8. Form of Hollow Cast Iron Block used for Test.

room machine, This machine, however, is also efficient for removing large amounts of metal in short time. The efficiency of the machine for this purpose depends largely upon the kinds of disks used. Tests were made at the shops of the Gardner Machine Co., Beloit, Wis., to determine the comparative efficiency for grinding cast iron, of differ-

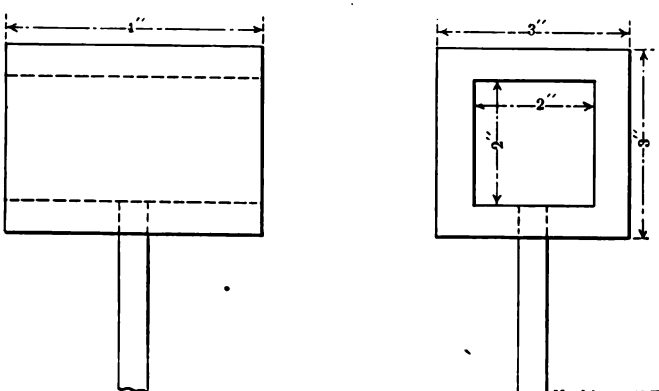

Fig. 9. Hollow Cast Iron Block used for Test.

ent kinds and makes of disks, such as are commonly used in connection with disk grinders. In the following table the different kinds of disks are indicated by figures:

No. 1 indicates the Gardner improved abrasive disk No. 126. No. 5 is the regular No. 24 commercial emery cloth. No. 6 is the same in emery paper. Nos. 2, 3, and 4 are disks of excellent quality as compared to commercial emery cloth.

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The disks tested were all 20 inches in diameter and all excepting Nos. 5 and 6 were No. 16 grain. The grinding was done on the ends of hollow blocks of cast iron, as shown in Figs. 8 and 9. The area ground at the end of blocks was 5 square inches. Reducing the blocks one inch in length indicated the removal of 5 cubic inches of metal. The grinding was all done on the same machine by the same operator.

The micrometer stop at the back of the table was set to grind off a fixed amount, usually 0.050 inch, and the twelve blocks ground to the stop. The stop was then moved back 0.050 inch and the operation repeated until the blocks became too warm for efficient grinding, when

TABLE III. RESULTS OF TEST OF EFFICIENCY OF ABRASIVE DISKS.

Disk Number.	Time Used in Minutes.	Stock Removed in cubic inches.	Number of Times Dressed.	Average Cutting Rate, cubic inches per min.	Cutting Rate, First Half of Time Used.	Cutting Rate, Second Half of Time Used.	Life of Disk, Based on Disk No. 1.	Stock Removed, Based on Disk No. 1.
1	754	349.85	0	0.464	0.442	0.486	100. %	100.0%
2	187	42.13	6	0.307	0.344	0.270	18.1%	12.4%
3	540	113.95	0	0.211	0.238	0.184	71.6%	32.8%
4	68	27.97	2	0.411	0.546	0.276	9.0%	8.0%
5	71	2.41	4	0.034	0.062	0.006	9.4%	0.7%
6	73	12.48	2	0.171	0.273	0.069	9.7%	3.5%

they were cooled, and the time of grinding and the amount of metal removed, noted. This was repeated until the disk was worn out or the blocks all ground up. In the latter case, new blocks were substituted and the operation continued until the disk was worn out. By reversing the blocks they were ground down until the wheel touched the handles on both sides. During this test several hundred pounds of these blocks were converted into cast iron chips.

It will be noted in Table III, that it was necessary to use a Huntington emery wheel dresser on all disks tested except Nos. 1 and 3. The dresser was used whenever the surface of the disk became dull and glazed so that it would not cut cast iron readily. The use of a dresser shortens the life of the disk, but it is absolutely necessary.

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CHAPTER III.

COST OF GRINDING.

To figure, with any degree of accuracy, the cost of commercial wet grinding, requires considerable experience in the use and management of the machine, in order to be as closely approximated as lathe work. There also seems to be a greater difference in operators, due partly no doubt to the fact that the general use of the grinder has not yet become as common. A great many operators seem to be afraid to push their machines, and spend a good deal of time in useless calipering. They seem to forget that if they have several thousandths to take off a piece and are feeding in one or two thousandths at each reversal of the machine, they need not caliper until within one or two thousanths of size, if they will keep in mind the number of reversals the machine has made. And another class seem to think that because grinding is a finishing job, it must be nursed.

As a matter of fact there is no machine which so readily and accurately responds to the touches of an operator as the wet grinding machine. Of course there are delicate pieces and certain shapes which

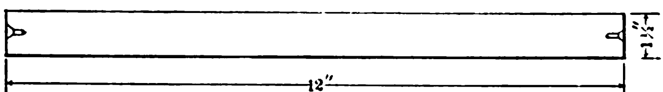

Fig. 10. Plain Cylindrical Piece to be Ground. Machinery, N.Y.

have to be carefully handled, but the usual run of work is so simple that any good apprentice can be put on it and taught in a short time.

As the work usually comes from the lathe, with approximately 1/64 to 1/32 inch stock to be removed, a few reversals of the machine with the work taking nearly the full width of the wheel each revolution and a cut of two to four thousandths until nearly up to size and then a much slower traverse per revolution for finishing, according to the kind of finish desired, and the work is done. To obtain the best speed, the limits required on the lathe must not be made too narrow, from 1/64 to 1/32 inch being admissible for ordinary work, and more on large work; for the facility of the grinder in finishing work is far in excess of the lathe, and the latter must be relieved of all the finishing possible.

To figure the actual time for removing stock on the grinder we must take into account the longitudinal traverse of the wheel for each revolution of the work, the surface speed of the work and the depth of the cut. The latter must be varied according to the nature of the material, greater or less according to whether it is hard or soft; and the traverse per revolution of work is lessened if a fine finish is desired. The shape of the piece also somewhat affects both of these

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points, as long, thin pieces require a slower traverse and lighter cuts.

Take, for instance, the plain piece, Fig. 10; material, hardened steel. For this a work surface speed of 15 feet, or about 37 revolutions per minute would be suitable. Assuming we have a wheel 18 inches in diameter, and 1½ inch face, a traverse of two-thirds the face of the wheel or one inch per revolution of work is usual. This would require 12 revolutions to pass the length of the piece, plus 1 revolution for clearance, or for dwell if there happens to be a shoulder. This would make, roughly, three reversals a minute.

On a medium-sized machine an automatic feed equivalent to a work reduction of about 0.002 inch would be suitable, or a reduction of about 0.006 inch per minute. If the work came with an average allowance of 0.030 inch for grinding, it would require theoretically 5 minutes actual grinding time to rough this piece down. To this must be added the time for handling the work, adjusting the machine and back rests (in this case only one rest would be used), calipering the work and finishing. This time will amount to as much as the grinding time with most operators (most of it being taken up in finishing), which would make the actual time about ten minutes apiece. As a matter of fact, work of this size is actually being ground at the rate of seven or eight pieces per hour.

If a fine finish is desired a higher work speed and slower traverse would be required. For a very fine finish a work speed of 45 feet surface speed and traverse of 1/6 inch per revolution would be suitable for finishing, with, of course, a very much smaller feed. This change in the work and traverse speed could be made when the work is nearly up to size, and would probably require about three minutes. If the piece were of soft steel a deeper cut can be taken and a wider traverse, a cut of 0.003 inch and a traverse nearly up to the width of the wheel being admissible. In grinding long shafts it is necessary to allow proportionately more time for adjusting back rests and for calipering, to insure that the piece be straight. This often takes twice the actual grinding time.

Now let us look at the more complicated piece, Fig. 11. This will have to be done on a larger machine, and the larger machines are slower to handle. This piece is a piston rod of 40 carbon

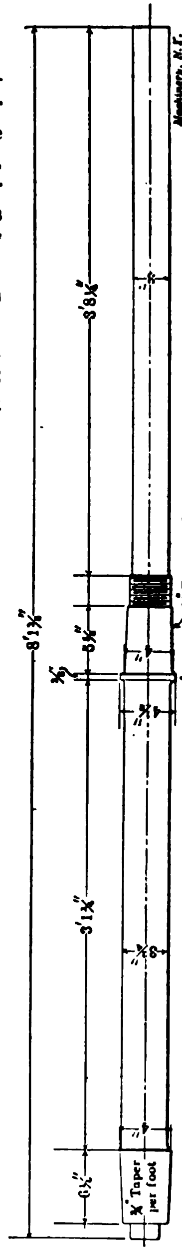

Fig. 11. A more Complicated Piece which is to be Ground.

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steel. We will use for this a 20-inch wheel of 2½ inches face. A suitable traverse for this would be 2 inches per revolution and a surface speed of 15 feet would make about 19 revolutions for the part 3 inches in diameter, and about 15 for the part 3¾ inches. The figures would be about as follows:

Total amount to be removed, 0.060 inch; amount per reversal, 0.004 inch; number of reversals required, 15.

3 inches diameter, to cross once, 1 1/5 minute; total for 15 reversals	18 minutes
3¾ inches diameter, to cross once, 1½ minute; total for 15 reversals	22 minutes
Tapers, both, to cross once, 2/5 minute; total for 15 reversals	6 minutes
Setting up and adjusting	10 minutes
Total	56 minutes

If it be desired to put a radius on the wheel and grind the fillets at shoulder *A*, about 10 minutes more should be allowed; and if there

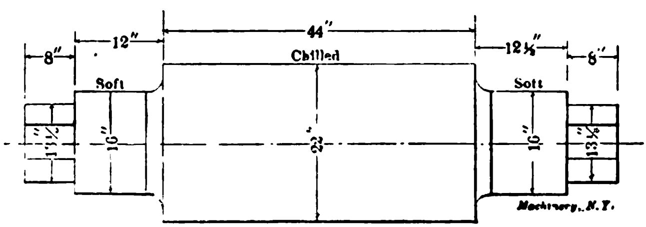

Fig. 12. Chilled Cast Iron Roll Ground from the Rough.

were more than one piece to be done considerable time could be saved in setting for the tapers.

The piece, Fig. 12, is a chilled cast-iron roll to be ground from the rough. This will take the largest machine built, and here the time taken is almost all grinding time. The average reduction for the chilled part is ¼ inch, and a feed of 0.002 inch is about all we can take, with a 30-inch wheel, 3-inch face and about 2-inch traverse per revolution. A work speed of 15 feet would give us about 3 revolutions per minute, making about 7 minutes for one cut across the roll and 14½ hours for the chilled portion of the roll.

For the soft necks of the roll (average reduction ½ inch) we can take a surface speed of 20 feet, equivalent to about 5 revolutions per minute, a feed of 0.004 inch and a traverse of about 2½ inches.

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CHAPTER IV.

THE BURSTING OF EMERY WHEELS.

In 1902 some important tests of the strength of emery wheels were undertaken at the Case School of Applied Science, Cleveland, Ohio, under the direction of Prof. H. Benjamin. Fifteen wheels of various makes were tested to destruction. The results of these tests are given in the following.

Most manufacturers of this class of wheels test them for their own information, but the results are not generally given to the public. At the Norton Emery Wheel Works, all wheels are tested before leaving the shop at a speed double that allowed in regular service, and occasionally wheels are burst to determine the actual factor of safety.

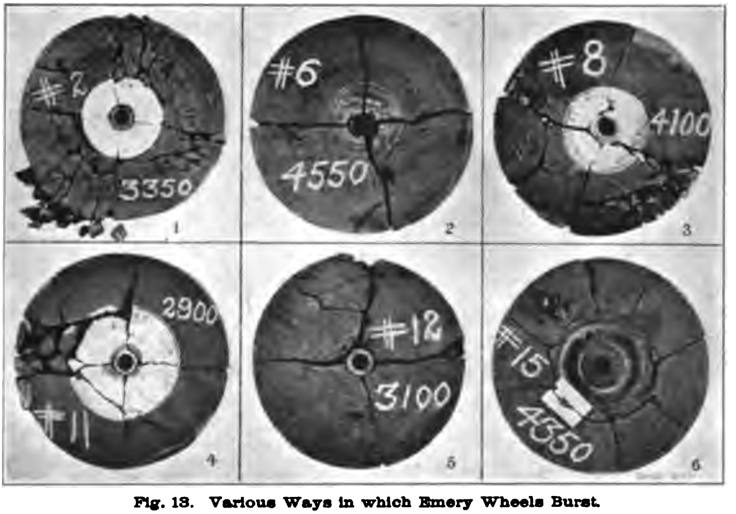

Fig. 13. Various Ways in which Emery Wheels Burst.

Emery-wheel accidents are not uncommon, but can usually be traced to the carelessness of the operator. One common cause of failure is allowing a small piece of work to slip or roll between the wheel and the rest.

The wheels selected for the experiments were all of the same size, being sixteen inches in diameter by one inch thick, and having a hole one and one-half inch in diameter. The object of the experiment being to determine the bursting speed of such wheels as are actually on the market, emery wheels were obtained through various outside parties without indicating to the agents or manufacturers the use to be made of them. In this way wheels of six different makes were obtained, the label on each wheel showing usually the maker's

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name, the grade number or letter, the quality of emery, and the speed recommended for use. As shown in Table IV, giving the results, the working speed varied in the different wheels from 1,150 to 1,400 revolutions per minute, the average being about 1,200 revolutions per minute. For a diameter of sixteen inches this corresponds to a peripheral velocity of about 5,000 feet per minute. The table also shows that the fineness of the emery varied from ten to sixty, the average being about thirty.

The wheels were held between two collars, each six and one-eighth inches in diameter and concaved, so as to bear only on a ring three-fourths of an inch wide at the outer circumference.

Table IV shows the results of the experiments in detail, and needs

TABLE IV. RESULTS OF TEST ON EMERY WHEELS.

No of Test.	Grade Mark.	No. of Emery.	Working Speed.		Bursting Speed.		Speed Ratio	Factor of Safety.
			Revs. per Minute.	Feet per Minute.	Revs per Minute.	Feet per Minute.		
1	4 5	20	1,200	5,080	3,100	13,000	2.58	6.67
2	4.5	20	1,200	5,080	3,200	13,400	2.67	7.14
3	4.5	20	1,200	5,080	3,350	14,020	2.79	7.78
4	Q	30	1,250	5,230	3,750	15,700	3 00	9.00
5	Q	30	1,250	5,230	2 750	11,500	2.20	4.84
6	H	30	1,400	5,870	4 550	19,050	3.25	10.56
7	H	30	1,400	5,870	4,600	19,200	3.28	10 76
8	O	36	1,250	5,230	4,100	17,200	3.28	10.76
9	O	36	1,250	5,230	4,125	17,250	3.30	10.89
10	2 5	60	1,150	4,830	2,750	11,500	2.39	5.71
11	2.5	60	1,150	4,830	2,900	12,100	2.52	6.35
12	M. H.	14	1,200	5,030	3.100	12,970	2.58	6.66
13		24	1,200	5,030	3,800	15,900	3.17	10.00
14	H	10–12	1,200	5,030	4.100	17,200	3.42	11.70
15	H	10–12	1,200	5,030	4,350	18,200	3.62	13.10

Tests 6 and 7; wheels made with wire netting; tests 14 and 15, with vulcanized rubber.

but little explanation. The illustrations in Fig. 13 show characteristic fractures, and the appearance of various wheels after bursting. Wheels numbered 1, 2, and 3 in table were of one make, and show a remarkable uniformity in strength. Nos. 4, 5, 8, and 9 were all made by one firm; the two latter wheels were of finer grain than the others, and show a correspondingly greater strength. Nos. 6 and 7 contained a layer of brass wire netting imbedded in the emery, and were about one-third stronger than the average of the ordinary wheels. The wheels numbered 10 and 11 were the weakest among those tested, but have an apparent factor of safety of between five and six. Nos. 12 and 13, of still another make, burst at about the average speed. Wheels Nos. 14 and 15 were so-called vulcanized wheels, containing rubber in the bond, and intended for particularly severe service. These showed, as was expected, rather more than the average strength.

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An examination of the last two columns in the table shows that the wheels burst at speeds varying from two and one-quarter to three and three-quarters the working speed, and accordingly had factors of safety varying from five to thirteen.

It is then apparent that any of these wheels were safe at the speed recommended, and would not have burst under ordinary conditions. At the same time, considering the violent nature of the service and the shocks to which they are exposed, it would seem that the factor of safety for emery wheels should be large. In comparison with those generally used in machines, a factor of eight or ten would seem small enough. It may also be said that such a variation in strength between wheels of the same make and grade, as for instance, that between Nos. 4 and 5, indicates a lack of uniformity which causes distrust. The fractures were in the main radial, as may be seen from Fig. 13, the wheel splitting in three, four or five sectors as might chance. It may be assumed that these radial cracks started from the rim where the velocity and stress were greatest, but it is a fact

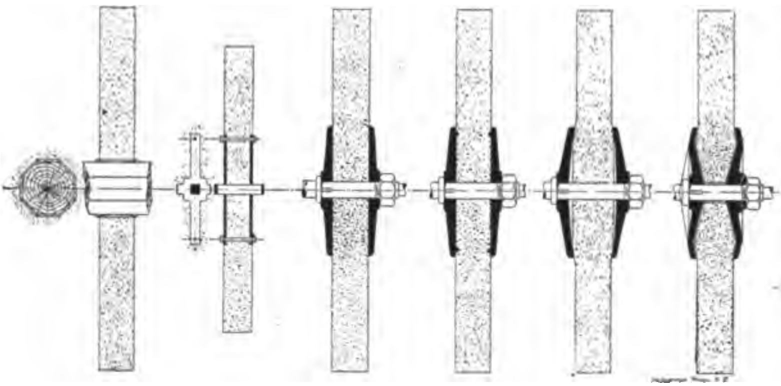

Fig. 14. Fig. 15. Fig. 16. Fig. 17. Fig. 18. Fig. 19.

worthy of notice that in nearly every instance the cracks radiated from points where the lead bushing projected into the body of the wheel.

Fastenings for Emery Wheels and Grindstones.

When an emery wheel or grindstone is revolved, a certain amount of tension is set up and the method of fastening should be carefully considered, as the bursting of emery wheels may be accelerated by improper fastenings. The oldest fastening for this class of tool is that shown in Fig. 14. The stone has a large hole in its center and is carried by a shaft of the same type as that used on old-fashioned water wheels. It is fastened with wooden wedges, which are soaked with water in order to make them swell and tighten as much as possible. The instability of these wedges, which, by alternate wetting and drying, are constantly varying the tension put upon the stone, serves to increase the danger of the bursting of the same. This style of fastening is, therefore, seldom used at present.

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A step in advance was made by the introduction of iron shafts. At first these were square in section. The stone was fastened between two wrought-iron crosses having a square hole at their centers. The ends of the arms were provided with holes through which bolts were passed for the fastening of the stone, as shown in Fig. 15. The crosses were themselves fastened to the shaft by wrought-iron wedges, so that the stone was, in this way, freed from all tension due to its fastenings. It was merely weakened in section by letting the crosses into it and by the holes for the bolts.

An essential improvement was made in the fastenings by the introduction of round shafts as shown in Fig. 16. Here the stone is fastened between two round clamping plates which are drawn together by a nut on the shaft. Through the pressing of these plates the stone is, of course, subjected to a crushing stress; it must, therefore, be admitted that stresses are thus set up in it that extend beyond the outer diameter of the plates. As a matter of fact stones so fastened have been sprung to such an extent that all of the material outside the plates has been fractured. Such clamping plates have also been made with circular ribs, as shown in Fig. 17, thus forming a first-class bursting furrow in the stone. These ribs possess the disadvantage that the section of the stone is dangerously weakened by them.

Another form of fastening is that shown in Fig. 18. Here cone-shaped plates are used for clamping the stone. The hollow cones are brought to bear against it. When these come to a bearing the plate must press equally against the stone throughout its whole circumference. As they are pressed together they are distorted to a greater or less extent on account of their own elasticity; their surfaces will be forced back and the diameter of the rims increased. Through the grinding action on the edge of the stone, the latter will increase in size and this increase will be shared by the plates which will thus set up a radial stress. These cone-shaped plates which seem to be so advantageous are, therefore, detrimental in that they exert a destructive influence on the stone.

The fastening by means of plates in the form of inverted cones, as shown in Fig. 19, is a preferable one. Here, by a tightening of the plates, the rim is drawn in towards the center and a tension towards that point is created. The tensions which are produced by the pressing of the plates together are toward the outside, and counteract each other to a great extent, so that there are no unfavorable stresses set up in the stone by this method of fastening. It will also be of advantage to make the bearing surface of the tightening nut spherical, whereby the plates can be made to better adjust themselves to any inequalities in the stone.

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CHAPTER V.

GRINDING KINKS AND EXAMPLES OF GRINDING.

Grinding a Large Crank-shaft.

A prominent English chainmaker recently sent to the Norton Grinding Co., Worcester, Mass., a rough-turned crankshaft to be ground to the dimensions given in Fig. 20. The conditions given were that the throw must be ½ inch plus or minus 0.001 inch and that the keyway shown in Fig. 20 should line up exactly with the highest point of the eccentric. The keyway was already in the shaft, when received. The following method was pursued in preparing the crankshaft for the grinder:

Two cast-iron blocks, Fig. 22, were planed to the dimensions given, and one side, *E* in Fig. 23, was scraped to a surface-plate. A squaring chip was then taken across a lathe face-plate and the plate was rigged with blocks and parallels as per Fig. 23. The surface *E* of the parallel *B* was also scraped to a surface-plate. When the large hole was bored, the block *A*, Fig. 23, was against parallel *C* and when the small hole, or eccentric hole, was bored, *A* was moved along parallel *B* and block *D* was inserted. Tissue paper was used in both settings to insure actual contact. The large holes were bored 0.015 inch larger than the finished diameter of the crankshaft ends. After boring the small holes, a 1-inch arbor was forced into the small holes and the 60-degree center holes were turned with a lathe tool. The truth of these 60-degree holes was tested by means of a ground cone point and red lead. A tapped hole and setscrew completed each block.

The shaft was now prepared for the blocks by grinding each end a wringing fit for its block. Before doing this, the center holes in the shaft were tested and scraped to a 60-degree cone point, to insure a perfectly round shaft when ground.

The next operation was to correctly locate the keyway. For this, two blocks, *A* and *B*, Fig. 21, were made. *A* is a 1-inch block that tapped lightly into the keyway and projected a short distance, as shown. *B* is a block planed to micrometer gage, and of such a height as to bring the center line of the keyway and the center line of the crankshaft into a plane parallel to the planer surface *C*, Fig. 21. The proper height of *B* was easily found by means of micrometer measurements and deductions. Having made *A* and *B*, Fig. 21, the whole job was taken to a newly planed planer table and the end blocks were placed on the crankshaft. *A* was then placed in the keyway and the crankshaft turned until *A* rested on *B*. With tissue paper under the end blocks *D*, Fig. 21, and between *A* and *B*, adjustments were made until all the papers held fast. The blocks *D* were then made secure by means of the setscrews *E*. After a final test with the tissue papers, the crank-

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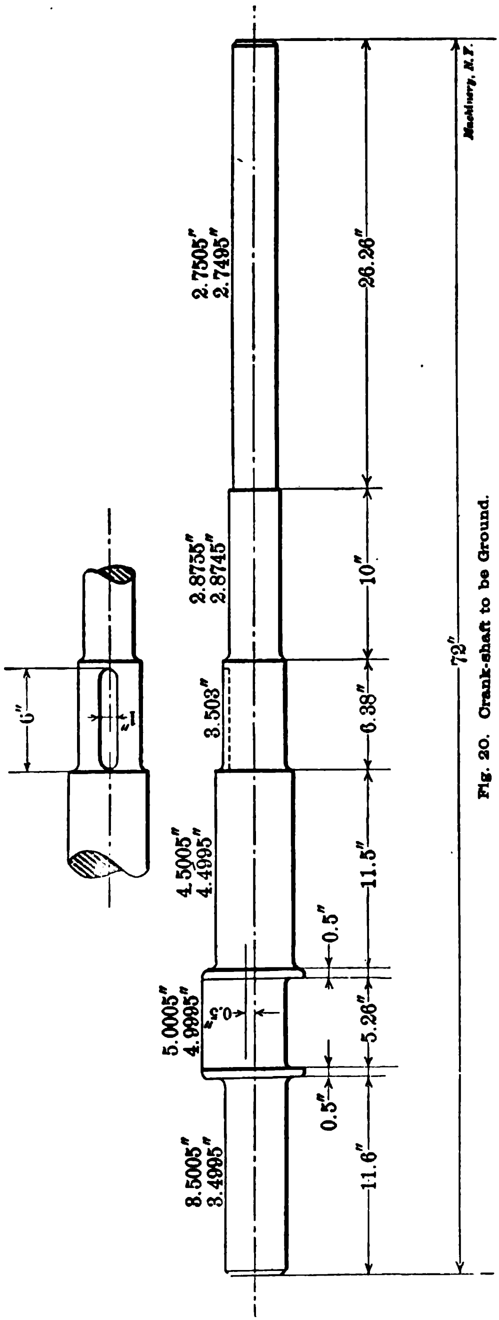
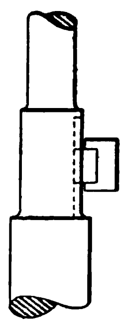


Fig. 20. Crank-shaft to be Ground.



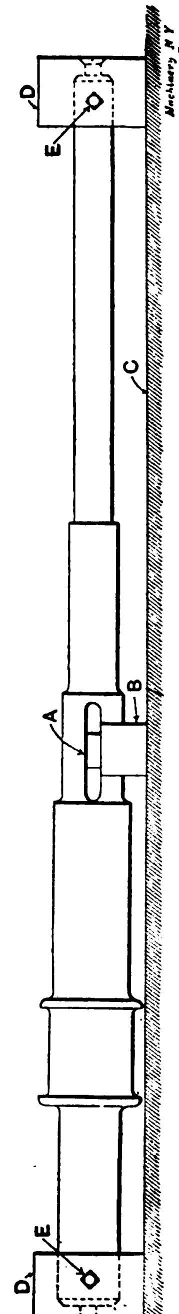


Fig. 21. Method of Mounting Crank-shaft in Fixture.

shaft was ready to have the eccentric ground. This was done on an 18-inch by 96-inch Norton plain grinder. The fillets on the eccentric were also ground at the same time.

The length of throw was tested in the grinder by means of a Bath

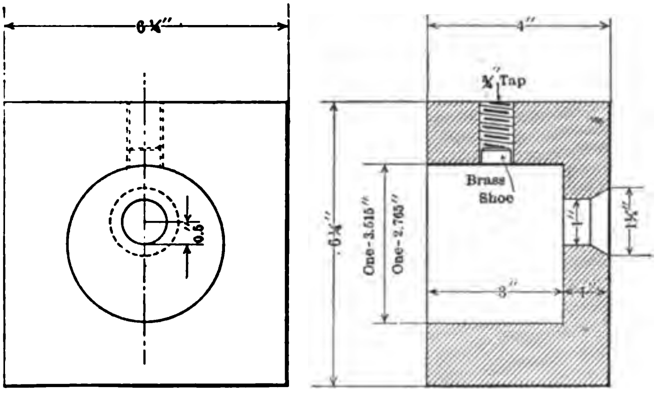


Fig. 22. Fixture for Grinding Crank-shaft.

indicator and a 1-inch B. & S. disk, and found to be within the required limits. When the eccentric was completed, the end blocks were

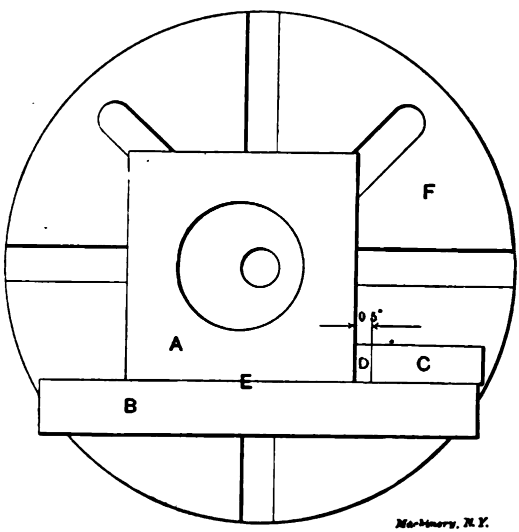


Fig. 23. Method of Boring the Fixture Used for Grinding the Crank-shaft.

removed and the remainder of the crankshaft was ground on its own centers.

Grinding Kinks.

In the following are described some of the kinks used by toolmakers in grinding; these kinks were contributed to MACHINERY by Paul W. Abbott.

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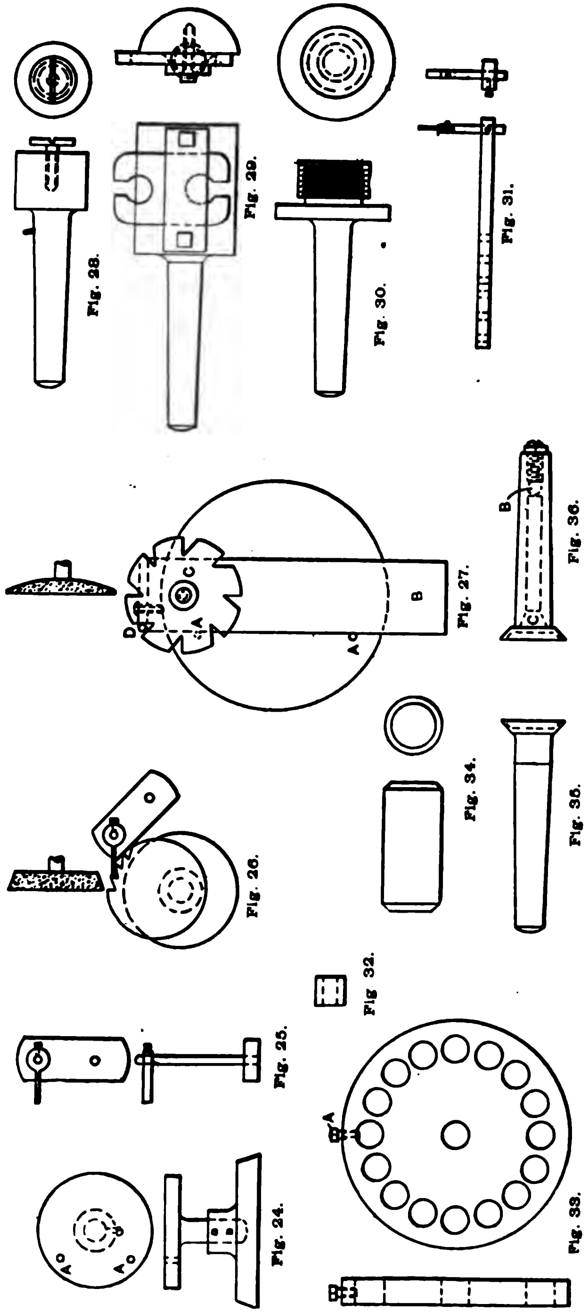

Figs. 24 to 36. Grinding Kinks.

Fig. 24 is a hand grinding rest which is very handy for use on the universal grinder. It is adjustable up and down for height, and is used for hand grinding circular and straight form tools, sharpening metal slotting saws, formed cutters, etc. Fig. 26 shows the application of the hand rest to the grinding of saw teeth in a blank. The tooth rest used in connection with this operation is shown in Fig. 25. These saws are first ground on an arbor, the old teeth being ground off, leaving a perfect circle. The operator then puts on this device. setting the tooth rest so that the teeth will be about ¼ inch apart, and grinds around by hand, not quite bringing each tooth to a sharp point. On the last nine or ten teeth he evens up any inaccuracy in the spacing, the wheel being trued off to the exact shape of tooth space wanted.

Fig. 27 shows a device for accurately sharpening formed cutters up to 3 inches diameter, which is used when the cutter grinder has another job in it, or could be used to advantage where there was no surface or cutter grinder. The device consists of the cast iron slide *B*, at the end of which is a tapped hole *C*, with a small fillister head screw which holds the various sizes of bushings which fit the holes in the cutters. On the same end is the index pin *D*, which is adjustable back and forth. In operation, the hand rest shown in Fig. 1 is also used, and the pins *A* are lined up parallel with the forward travel of the wheel and so that the cutting face of the wheel is on a line with the center of the bushing. The cutter is then slipped on over the bushing and the index pin is set so that the required amount will be ground from the face of the tooth. The operator brings the wheel up to the proper position and then pushes the slide forward until the wheel has reach the bottom of the tooth space; he then withdraws the slide and indexes to the next tooth, and so on, tooth after tooth. It will be noticed that the index pin rests against the back of the tooth, which means that upon the previous milling of the teeth depends the accuracy of the grinding; but on the standard cutters furnished by numerous concerns this spacing will be found accurate enough.

Fig. 28 is a center for the head-stock for holding small forming tools of odd size, or threaded pieces which are to be ground on the periphery. The tools are simply clamped to the face of the center, and trued up by an indicator. Fig. 29 is a device for the tool grinder for grinding snap gages, where there is no surface grinder for this class of work. The shank of this device is made to fit the head-stock, and the gages are clamped to it by a small strap and two screws. This fixture revolves while in use, and the jaws of the gage are ground by feeding a thin wheel in and out by hand. Revolving the device insures perfectly straight gage faces.. Fig. 30 shows a center for the universal grinder for holding a standard line of large end milling cutters with threaded holes, while sharpening. The head-stock is swung around at right angles to the ways, and with a long support for the tooth rest (Fig. 31), which is bolted to the platen, the cutters are ground very handily by throwing in the feed and grinding one tooth, and then, before the wheel comes back, indexing to the next tooth, and so on.

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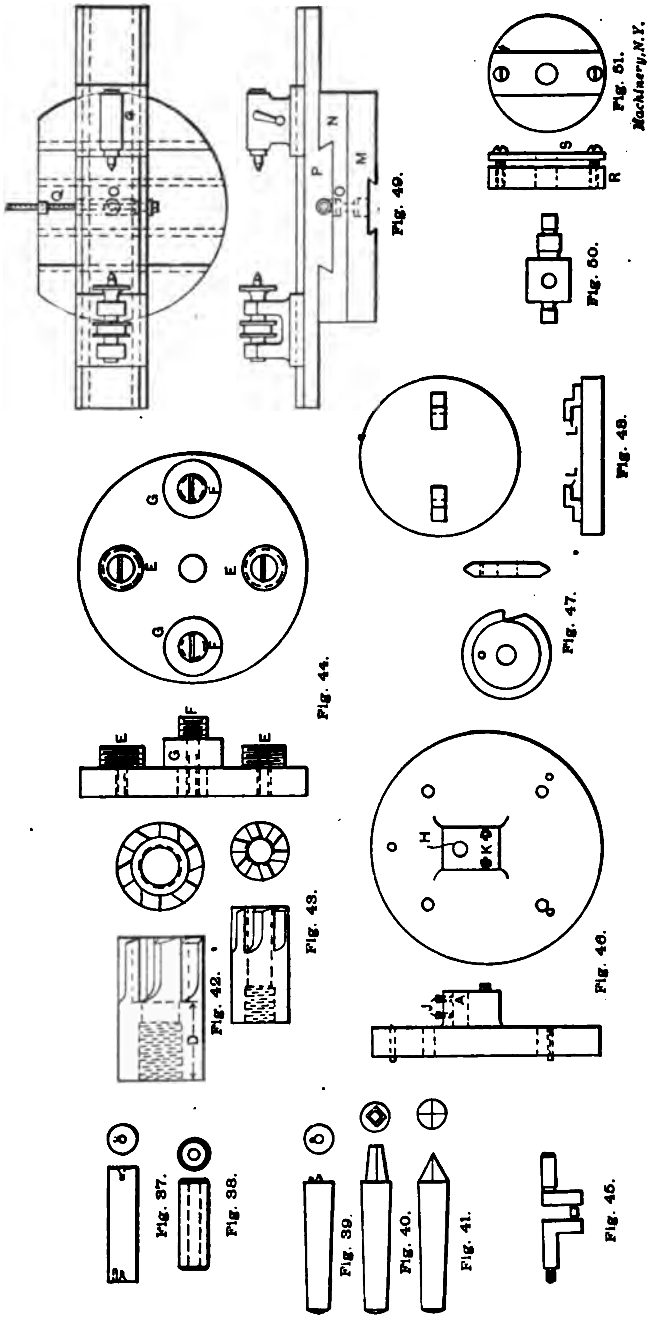

Figs. 37 to 51. Grinding Kinks.

Fig. 32 shows a hardened roller which is ground all over, and Fig. 33 the fixture for the universal grinder for grinding the sides of this roll. This plate was made of cast iron, with both sides ground and with each hole ground to 0.0005 inch over standard size. Each hole has a ¼-inch set-screw, as shown at *A*. In operation, the plate is fastened to the face-plate by a draw-back rod, and the head-stock is swung around at right angles. As the plate revolves, 16 rolls are ground at once, first on one side, and then the plate is turned and the other side ground, the rolls being made to standard length by using a depth gage. The hardened roll shown in Fig. 34, which is used on swaging machines, is held by the centers shown in Fig. 35 and 36, when being ground. Fig. 35 is the head-stock center cupped out on the end to fit the beveled end of the roll. This center drives the roll by friction, the pressure being obtained by the spring tail-stock. Fig. 36 is the tail-center which is in two parts, the inner spindle running with the roll and being adjusted by the screw in the end so that the thrust is taken by the ball *B*, the tapered portions *C* just clearing each other. Other methods of grinding rolls are shown in Figs. 37 to 41. One example of grinding is shown in Fig. 37, and its center in Fig. 39. The roll is driven by a pin on the center, which engages with a corresponding hole in the work. A better method is to center the roll and then in one end drive a square 60-degree punch, using the square center shown in Fig. 41 for driving the work while grinding. Another good method for hollow rolls, such a shown in Fig. 38, is to use a 15-degree square center, such as shown in Fig. 40, the end of which just enters the hole.

Figs. 42 and 43 show two end mills. The smaller one is fastened inside of the larger when in use, and when in position rests against the bottom of the hole and projects outside a definite distance. The length *D* is standard in all these mills. Fig. 44 shows the fixture for grinding two pairs of these mills at a time, so that the same amount will be taken off of both the short and long ones. Threaded bushings *E* fit the larger size mills, and *F*, the smaller. The collars *G* are of such thickness that the cutting face of the smaller mill is brought into the same plane as the larger, and so when grinding, an equal amount is removed from the face of each mill. The plate is held to the face-plate by a draw-back rod. The head-stock is swung at right angles, and with the fixture revolving, the wheel traverses back and forth across the faces of the mills. The mills are then taken to a cutter grinder and backed off.

. Fig. 45 shows a small crank-shaft, and Fig. 46 the fixture for grinding the pin. The bearings are first ground on centers in the usual way. The fixture is of cast iron and is held to the face-plate by screws and dowel pins. In the making of this fixture the hole *H* was ground out to the size of the bearing, and then the fixture was correctly located and doweled to the regular face-plate. The crank, while being ground, is held by the set-screws *J*, and the screws *K* which are set against the crank on either side.

The grinding of formed cutters, similar to the one shown in Fig. 47,

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so that they will be interchangeable, is very interesting. The error limit is 0.00025 inch. The grinder used is a Norton universal tool and cutter grinder. After hardening, the cutters are first ground to a definite thickness. For this operation they are held against the face-plate by a draw-back chuck. The next operation is grinding the beveled sides, which is accomplished by holding the cutters against a small face-plate by a draw-back chuck. The correct angle of bevel is obtained with the protractor, and to get the correct diameter of the bevel sides, and to insure that the bevel sides stand exactly in the same relation to each other, the gage shown in Fig. 48 is used. This gage is hardened and ground all over, and the two gaging points *L* are set a predetermined distance apart and as near the same height from the platen as mechanical means can make them. It is obvious that cutters which are all ground the same thickness, and which will pass through this gage with the beveled sides both touching the gage points with equal pressure, will interchange within pretty close limits. The operator grinds one bevel side at a time, trying the work every little while in this gage; when one side passes through the gage the cutter is turned around and the other bevel ground. For grinding the radius on the periphery and bringing the cutter to the correct diameter, the radius grinding fixture shown in Fig. 49 is used. The dovetailed base *M* is fitted to the platen of the grinder and upon this base is a sliding base *N* which is pivoted to *M* by a bolt *O*. Upon the base *N* there is an auxiliary platen *P* which can be adjusted back and forth by the screw *Q* for getting the proper radius. This auxiliary platen is made the same as the machine platen so that the regular head- and tail-stocks will go on it. A cutter is placed on a special arbor and the platen *P* adjusted to give the correct radius. The wheel is then brought up and the cutter is ground to the correct diameter, the curved face being obtained by swinging the base *N* back and forth by hand in an arc of a circle, with bolt *O* as a center.

Another ingenious scheme is shown in Fig. 51. Three or four pieces similar to the one shown in Fig. 50 were to have the holes ground out. With an independent 4-jawed chuck this would have been easy, but there was no such chuck; and as there would never be any more of these pieces to be ground the fixture for doing the work had to be inexpensive. The face-plate could not be used as the pieces were smaller than the hole in the face-plate. The operator thought awhile, and then hunted around a few minutes and found a large washer *R*, tapped two holes in it, filed up the sheet steel strap *S*, and with a couple of machine screws was ready to begin. The washer was first put in the universal chuck and the outer side ground. One of the pieces was then clamped in place, and after putting on the internal grinding attachment it was ready to be ground.

Grinding Fixture for End Measure Rods.

The fixture shown in Fig. 52 was made to grind spherical end measuring rods 3 inches long, and over. There were three diameters of these rods, ⅜, ½ and ⅝ inch, and three sizes of fixtures were also

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made to suit. Steel sleeves *B* were made of various lengths for each fixture, to stiffen the rods near the end when grinding. These sleeves had a bearing at each end in hardened bushings which were driven into the fixture body. The sleeves were kept from turning and from sliding endways, independently of the gear, by the set-screws shown. These screws, in addition, held the rod being ground. Member *A* was a tight fit in the large bevel gear, which was made separate to allow of cutting the teeth. The flush oilers *C* were placed on opposite sides of the body, so the tendency to throw oil would be done away with as much as possible.

A feature apt to be overlooked is that the number of teeth in the two gears must be prime to each other. If not, every certain number of revolutions, depending on the gears used, the wheel will grind into

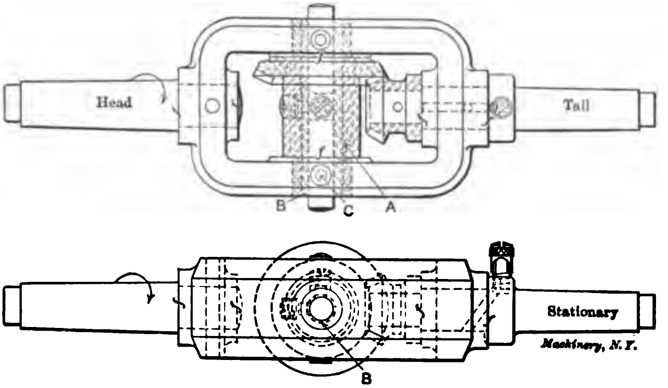

Fig. 52. Fixture for Grinding Spherical End Measure Rods.

one of the previous cuts. The fixtures were arranged to be used on a bench-lathe, with a tool-post grinding fixture.

Selection of Wheel.

The following little hints regarding grinding, taken from a booklet issued by the Norton Co., Worcester, Mass., will prove of value to all who have to do with grinding machines and grinding.

Don't believe that all materials can be ground equally well with one and the same wheel.

Get the proper wheel for the work.

You would not expect to turn all kinds of lathe work with one tool having only one form of cutting edge. The grinding wheel is a tool for cutting.

Different shapes of work, different kinds of metal, require different cutting edges as well when grinding as when turning. Different grades and grains of wheels are required for different kinds of work.

Grinding wheels are numbered from coarse to fine, and graded from soft to hard. The grade is denoted by the letters of the alphabet from E to Z.

Don't decide on the wheel without knowing the work.

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Spindle speed and character of the material, shape of work to be ground, and surface of wheel in contact are prime factors.

In cylindrical grinding, speed of work, diameter of work and depth of cut must all be reckoned with in the selection of the right combination of grain and grade.

The condition of the machine affects the efficiency of the wheel. Heavy machines with large wheel spindles and massive wheel support call for a wheel different from those for lighter machines with smaller spindles.

Don't order a certain grade wheel merely because that grade is used on similar work in another plant.

Don't use a hard wheel to economize—it is production you are after.

A hard wheel is more likely to change the temperature of the work or to become glazed than a soft one; furthermore, it requires more power to do the same amount of work.

It is a common error to assume that a wheel for grinding steel and cast iron, chilled iron and hardened steel, must be as fine as the surface desired. A coarse wheel will produce a fine finish if the proper relations between grade, depth of cut, speed of work, speed of wheel, etc., are observed.

When grinding brass and the softer bronzes, the wheel must be as fine as the finish required. Bronzes with "manganese" or "phosphor" permit the use of coarser wheels.

Don't get a wheel made for soft steel for use on hard steel.

For a fine finish on hard stock, a coarse wheel may be necessary, and the harder the stock the coarser the wheel.

When ordering wheels, don't forget the diameter, width, style of face, arbor holes, description of work, speed of spindle, and the number and letter denoting the combination of grain and grade, if known.

The width of the wheel should be in proportion to the amount of the material to be removed with each revolution of the work.

If you reduce the width of the wheel, you must use a finer feed, and consequently do less work.

Mounting.

Never mount wheels without flanges.

Flanges should be, at least, one-third the diameter of the wheel; one-half is recommended. Flanges should be concave—never straight or convex.

Use fiber or rubber washers a trifle larger than the diameter of the flanges, or flanges with soft metal facings.

Hooded machines are desirable when practicable.

Truing.

Don't start work on a new wheel until you are sure it runs true.

Always have a wheel dresser handy for truing wheels for off-hand grinding.

Never use a dresser on wheels that grind circular work on centers.

For truing wheels used on plain cylindrical and universal grinding machines, cutter and reamer grinders, etc., the diamond is recommended. To obtain the best results it is absolutely necessary.

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Never attempt to true a wheel for circular grinding unless the diamond is held in a rigid tool-post on the table of the machine. You cannot do good work with such a wheel when it is trued "by hand."

To get a truly ground surface, you must keep the face of the wheel true.

The quality of surface finish is dependent on the conditions of the wheel face and depth of cut.

Speed.

Don't start grinding until you know the speed is right—not "near enough," but right.

Even a slight variation in speed may be the cause of success or failure of any wheel.

Failure is sometimes turned into success by merely changing the speed of either the wheel or work.

Speed up the spindle as the diameter of the wheel is decreased. Approximately the same peripheral rate should be maintained as the wheel wears down.

Complaint is sometimes made that wheels appear to be softer toward the center. Usually this is because the same surface rate of speed is not maintained as the wheel is reduced in diameter. This causes the wheel to wear away faster and appear softer. It is also true that while the grade of the wheel may be uniform throughout, yet the smaller line of contact due to the smaller diameter will cause the wheel to appear softer.

Increasing the speed of a grinding wheel gives the effect of a harder wheel; decreasing the speed gives the effect of a softer wheel.

For surface grinding, it is customary to run wheels at a somewhat slower rate of speed than for general grinding. A speed of 4,000 to 5,000 surface feet is usually employed.

Wheels are run in actual practice from 4,000 to 6,000 feet per minute.

General Suggestions.

Transferring a wheel worn down to a small diameter, from a large machine to a small one, is good practice.

Keep the tickets or tags which are sent on the wheels in a record book, so that if a wheel is not satisfactory, reference can be made to order number when making complaint. It is equally valuable as a reference when ordering duplicate wheels.

Don't use the wrong wheel on a job because it will require a few minutes time to change wheels. A stop watch will prove to you that changing wheels is cheaper.

There is seldom a case where one and the same wheel can be used on all work without a greater loss of time than the change of wheel would involve. Many times, the time saved in grinding a single piece more than pays for changing the wheel.

Considerable difference in diameters of work will affect the cutting quality of a wheel on any given material.

A successful wheel on the small diameters may work much slower on the larger diameters.

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The wheel most suitable for work of very large diameter may wear away too fast on work of smaller diameter.

A suitable wheel for small diameters may cause chatter on pieces of large diameters.

Don't grind circular work dry.

A good wheel will grind in water, soda water or oil.

Water keeps the wheel working cool, and increases grinding production.

Soda water keeps the work and the machine from rusting.

Oil in soda water increases the wheel's effectiveness.

The particles from a grinding wheel do not adhere to steel. Don't let any one convince you to the contrary.

Grinding is profitable for removing stock as well as for finishing.

Keep the face of the wheel true and parallel with axis of spindle.

Vibration makes grinding wheels wear.

Keep all rests adjusted close to the wheel, otherwise work is liable to be caught and injury result.

Keep boxes well oiled and adjusted.

When practicable, indicate on each machine the revolution of spindle and size of wheel to be run upon it.

Don't disregard the setting up instructions that go with the grinding machine.

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CHAPTER VI.

LAPPING FLAT WORK AND GAGES.

Tho main essential points of the art of lapping can be described in a book, but, the same as with any other line of mechanical work, it is necessary that the workman shall do considerable lapping before he can become proficient. There are certain motions, touches, sounds, refinements, etc., which the skilled workman acquires by practice, that are impossible of enumeration and description, or of enumeration and description that would be intelligible to an inexperienced man. For instance, ask a carpenter how he knows that he is sawing a board straight, and he will be unable to tell you. Nevertheless, he has acquired a peculiar sense of touch, or such general acuteness

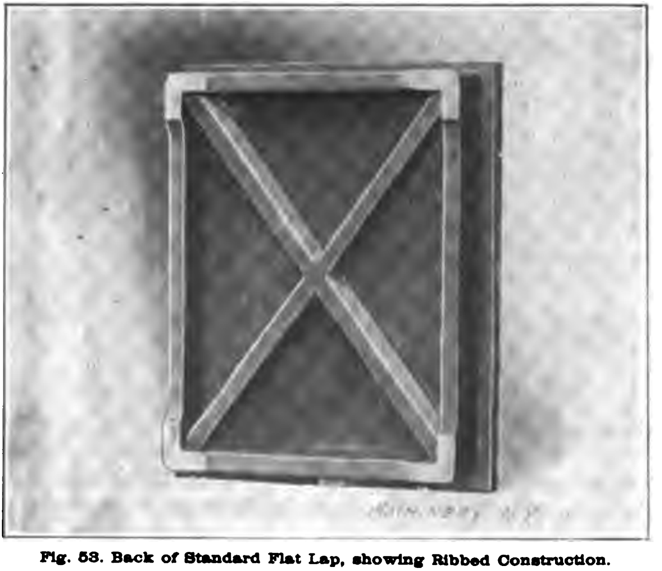

Fig. 53. Back of Standard Flat Lap, showing Ribbed Construction.

of the senses, that he knows instantly when the saw starts to "run out." His mind and arm automatically, as it were, return the saw to a straight line without missing a stroke. It is the same way with a die maker. He can file a die, looking only at the surface line, and can detect the instant when his file "rocks" from a straight line. He will tell you that he "feels" it, but is unable to define what the sensation is. Likewise, one cannot explain some of the finer points in the art of lapping, and can only point out those which are fundamental, and which must be acquired first by the workman unaccus-

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tomed to such operations. He must acquire the refinements by practice and experience.

A Perfect Lap Required for Perfect Lapping.

The first requisite of perfect lapping is a perfect lap, and right here is where the novice will make his first mistake, that is, in the preparation of the lap. To make a surface lap, it should be carefully planed, strains due to clamps being avoided, and then it should be carefully scraped to a standard surface plate. This is done by rubbing the face of the lap on the standard surface plate and scraping down the high spots until a perfect plane surface is obtained. If a standard surface plate is not at hand, a lap can be made level by using three laps that are nicely planed and used alternately as follows. We will number the laps Nos. 1, 2, and 3. Now, rub No. 1

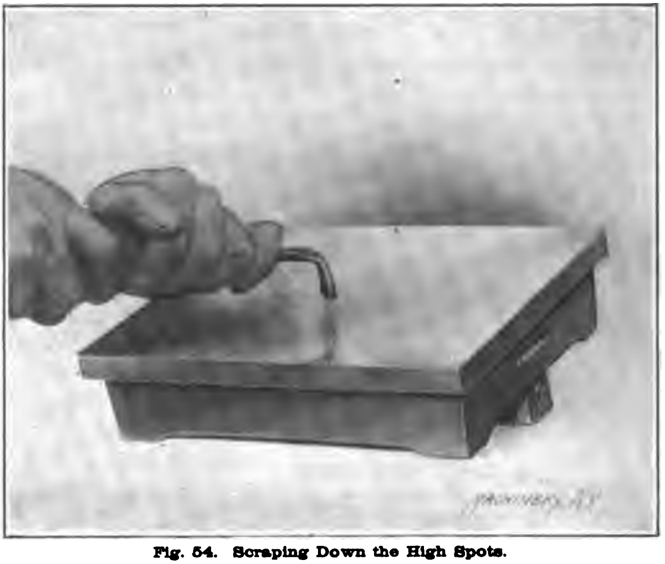

Fig. 54. Scraping Down the High Spots.

and No. 2 together, and scrape the high spots until they fit. Then introduce No. 3 and scrape it down to fit No. 2, and then to fit No. 1, and so on. The third lap eliminates the error that might follow if only two laps were used. For example, it is possible to fit two plates accurately together without making them plane surfaces, one becoming concave and the other convex. The third lap absolutely prevents this and produces a perfect plane surface if time and patience hold out. It is a slow operation, but not so slow as trying to lap a piece true with a lap that is not true.

The Objection to Ground Laps.

The laps may be ground together instead of scraping, but the writer prefers the scraping process, as it is easy to see when the job is done.

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it is also better to scrape them, because it is quicker than attempting to grind them level with the fine grade of emery that is required for nice lapping, and it must be remembered that when ground together the laps *are already charged.* Hence, the necessity of using a fine grade of emery if they are ground together.

Using a Hand Surface Lap.

The writer prefers a cast iron lap, Fig. 53, thoroughly charged, and having all loose emery washed off with gasoline. When lapping, the surface is preferably kept moist with kerosene, although gasoline causes the lap to cut a trifle faster. It evaporates so rapidly, however, that the lap soon becomes dry, and the surface caked and glossy in

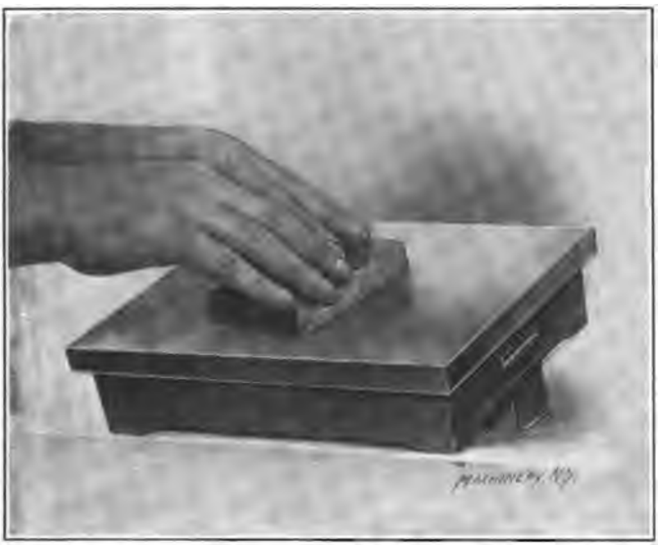

Fig. 55. Charging the Lap, using a Hardened Steel Block.

spots. When in this condition, a lap will not produce true work. The lap should be employed so as to utilize every available part of its surface. Gently push the work all around on its surface, and try not to make two consecutive trips over the same place on the lap.

Do not add a fresh supply of loose emery to a lap, as is frequently done, because the work will roll around on these small particles, which will keep it from good contact with the lap, causing poor results to follow. If a lap is thoroughly charged at the beginning, and is not crowded too hard, and is kept well moistened, it will carry all the abrasive that is required for a long time. This is evident, upon reflection, for if a lap is completely charged to begin with, no more emery can be forced into it. The pressure on the work should only be sufficient to insure constant contact. The lap can be made to cut only so fast, and if excessive pressure is applied, it will become "stripped" in places, which means that the emery which was imbedded

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in the lap has become dislodged, thus making an uneven surface on the lap.

Causes of Scratches—Grading Emery.

The causes for scratches are as follows: Loose emery on the lap; too much pressure on the work which dislodges the charged emery; and what is, perhaps, the greatest cause, poorly graded emery. To produce a surface having a high polish free from scratches, the lap should be charged with emery or other abrasive that is very fine. The so-called "wash flour emery," sold commercially, is generally too uneven in grade. It is advisable for those who have considerable high class lapping to do to grade their own emery in the following manner: A quantity of flour emery is placed in a heavy cloth bag, and

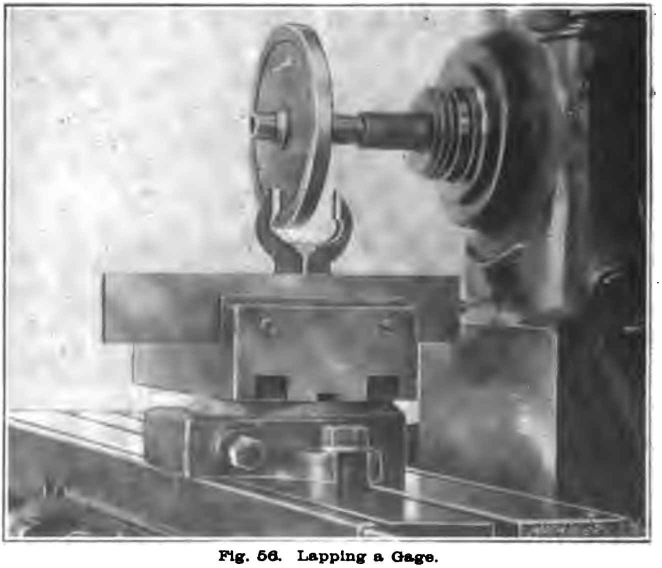

Fig. 56. Lapping a Gage.

the bag gently tapped. The finest emery will work through first, and should be caught on a piece of paper. When sufficient emery is thus obtained it is placed in a dish of lard or sperm oil. The largest particles of emery will rapidly sink to the bottom, and in about one hour the oil should be poured into another dish, care being exercised that the sediment at the bottom of the dish is not disturbed. The oil is now allowed to stand for several hours, say over night, and then is decanted again, and so on, until the desired grade of abrasive is obtained.

For the information of those not well acquainted with grading abrasives, it may be said that the grade of diamond dust known as "ungraded" is obtained in about five minutes, while it requires about three weeks to obtain the grade known as No. 5, which is very fine. But, even at the end of three weeks there still will be small particles in the oil that have not settled, due to the viscosity of the oil.

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To lap true and free from scratches, one must have skill and be thoroughly conversant with the peculiar sounds, touches, and motions spoken of above. For a high polish on work, a rapid motion and slight pressure are necessary for success. It is also necessary that the lap is properly charged with properly graded abrasive.

Lapping Gage Jaws.

Fig. 56 shows the best method that has come to the writer's notice for lapping the jaws of gages. The lap is made of cast iron and is relieved as shown, leaving only a thin edge or flange on each side to bear against the jaws. As the machine table is worked back and forth, the lap passes over the entire surface of the jaw, grinding it down in the same manner as would be done with a cup emery wheel.

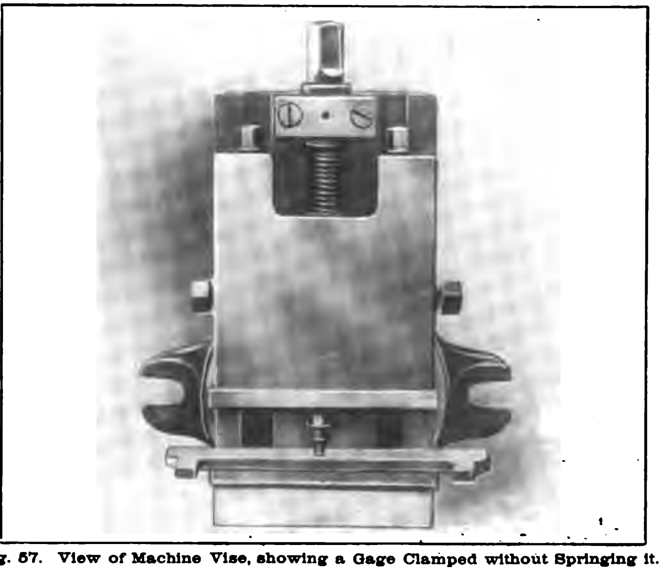

Fig. 57. View of Machine Vise, showing a Gage Clamped without Springing it.

Care must be taken to clamp the gage in the vise so as not to spring it. Fig. 57 illustrates an approved method for holding a gage so that the vise jaws will not deflect it. Should the gage be sprung, it is clamped at the center only, leaving the ends free. Snap gages are now mostly made of machine steel and pack-hardened. Made in this way they do not change much, as the interior of the gage is left soft, and whatever change occurs can be easily remedied, but in any case the method illustrated is the safest one to follow, for it leaves no doubt as to the gage being held free from spring during the lapping operation.

A lap should be turned on the arbor on which it is to be used, for it is almost impossible to put a lap back on an arbor after it has been removed, and have it run true. Therefore, the lap should be recessed quite deeply, as shown, to allow for truing up each time

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the lap is placed on the arbor. Perhaps when the lap is mounted on an arbor in the milling machine, it will be found to run out not more than 0.001 inch, but that means that it is touching the work in only one spot, and the result can be hardly better than if a fly-cutter was used for a lap. Fig. 58 shows the operation of truing the lap. A keen cutting tool is clamped in the vise and in this way the lap can be trued as nicely as though it were done in the lathe. In fact, it is superior, for there is absolutely no change in the alignment of the lap with the work spindle after it is turned, which might easily happen should it be turned in the lathe and then mounted in the milling machine spindle. With a perfectly true lap, a perfect contact between the lap and gage is insured for its entire circumference.

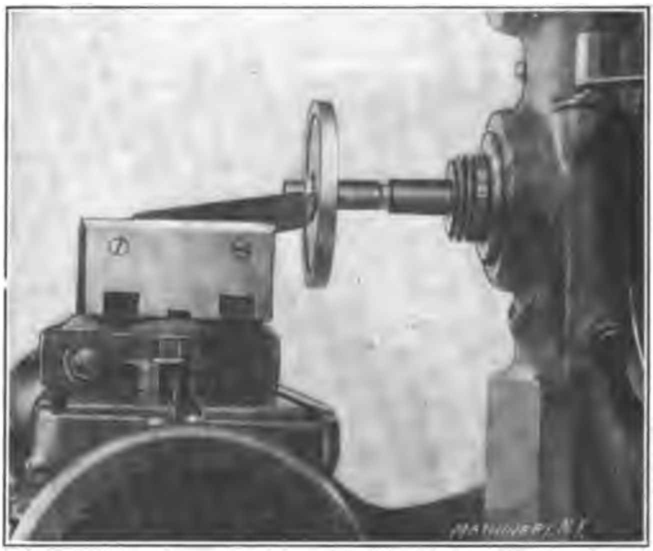

Fig. 58. Truing the Lap with a Tool held in the Vise.

Both sides of the lap should be turned at the same setting on the arbor.

Fig. 59 shows the operation of charging a circular lap, using a roller mounted in a suitable handle for the purpose. The emery is rolled in under moderate pressure. It is good practice to make the roller of hardened steel, and after charging the lap, all the surface emery should be thoroughly washed off.

The next step is to square up the jaws of the gage. Do not depend on the zero marks of the vise. The jaws of the gage may have sprung a little in hardening, and if the zero marks of the vise are depended upon to square the work, there possibly will not be sufficient stock on the jaws to clean up. Be very careful to set the gage by the surface of the jaws and to clamp it in the vise as previously noted, so that it is under no pressure tending to spring it out of shape.

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When employing a power-actuated lap, the little instrument shown in Fig. 60 is useful in determining the instant when the lap touches the work. By placing the forked end on the work and the wooden part to the ear, the sound is greatly magnified, and it makes it much easier to determine the precise point of initial contact. If one depends upon the naked ear to tell when the lap touches the work, he is liable to crowd the lap too much, and scratch the work or strip the lap. With this instrument the mechanic will know the instant the lap just touches the work, and this is the position where its work should be done. In short, the lap should not work under any appreciable pressure, but should simply touch the work. Hence the desirability of some means of magnifying the sound and not depending on the naked ear.

The workman should avoid the custom of adding a fresh supply

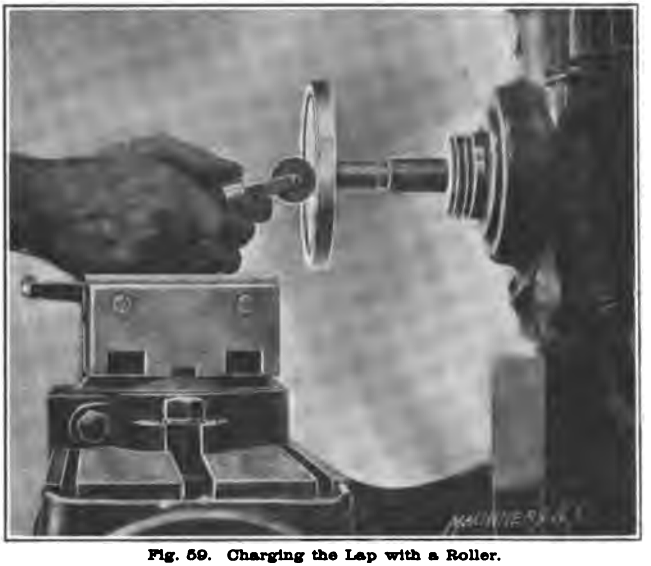

Fig. 59. Charging the Lap with a Roller.

of abrasive to the lap, as it is not only injurious to the character of the product, but it naturally increases the time required for lapping. To illustrate the action, suppose that an arbor is to be ground in a grinding machine, and that it is belted so that it runs with a wheel at the same speed. The consequence will be that no grinding action could take place, as there would be no difference in motion. The condition is very similar when loose emery is placed on a surface lap. The emery simply rolls around between the work and the wheel, and occasionally a piece of emery is imbedded in the lap long enough to scratch the work. While it may look as though the lap was cutting much faster, the truth is that it cuts slower and produces poor work.

In lapping jaws, some workmen round-lap, and then finish by hand,

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but a better job will result when finished in the machine. It is poor practice to rough-lap a gage, using a coarse grade of emery, and then wash the lap and smear it with fine emery. Of course the lap is already charged with a grade of emery last used, and the act of putting on a supply of fine emery on the lap will not produce as good a surface as if the gage were finished without the fresh supply of emery, though the latter is of a finer grade.

Lapping Gages.

Assume that a 1-inch plug gage is to be lapped to size. Such a gage needs only about 0.0015 inch for lapping. The outside lap, shown

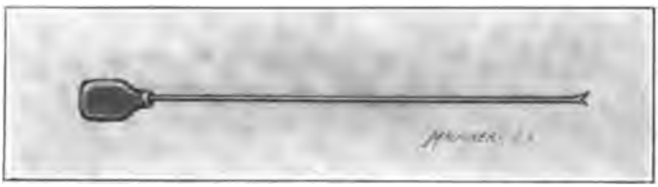

Fig. 60. Sound Magnifier.

in Fig. 62, should be made of cast iron, copper or lead, and the holder *D* should be provided with adjusting screws. The flour emery used should be sifted through a cloth bag to prevent any large particles of emery entering the lap and scratching the gage. After sifting the emery it is mixed with lard or sperm oil to the consistency of a thin paste. The gage is then gripped in the chuck of the lathe by the knurled end and smeared with emery paste. The lap is adjusted to fit snugly on the gage and the lathe is speeded up as fast as possible

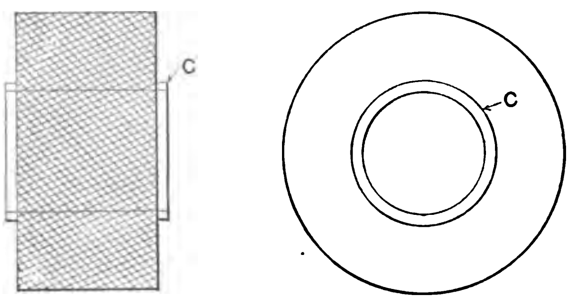

Fig. 61. Ring Gage to be Lapped.

without causing the emery to leave the gage. The lap requires constant adjusting, to take up the wear of the lap, and reduction in size of the gage. When measuring the gage, it should be measured at both ends and in the center to make sure that it is not being lapped tapering. When the gage has been lapped to within 0.0002 inch of the finished size, allow the gage to thoroughly cool and then by hand lap lengthwise of gage to the finished size. By so doing all minute ridges that are caused by circular lapping are removed, thereby leav-

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ing a true surface and also imparting a silvery finish. A gage should never be lapped to size while warm (heated by the friction of the lap), because the gage expands when heated, and if then lapped to size it will contract enough to spoil it.

In grinding out the inside of a ring gage considerable difficulty is experienced in adjusting the grinder so that it will grind straight. One way to prove the straightness of a hole being ground is to move the wheel over to the opposite side of the hole until the wheel will

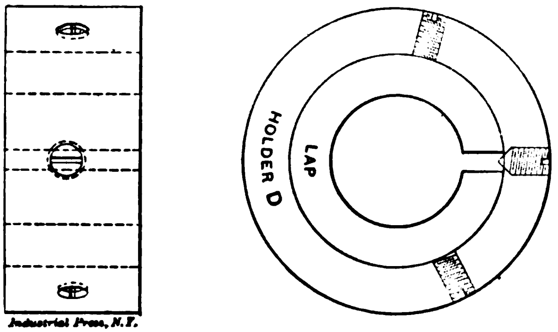

Fig. 62. Outside Lap.

just barely "spark." Then, beginning from the back of the hole, feed out, and if the hole is tapering, the wheel will either cease to spark, or will spark considerably more. Another and better way is by by means of the multiplying indicator gage, Fig. 64. By fastening the indicator to the spindle of the grinder and placing the contact pin of the indicator on the opposite side of the hole and feeding in and out, the pointer will record in thousandths of an inch just what the deflection is.

A ring gage should be made as shown at Fig. 61, the object being

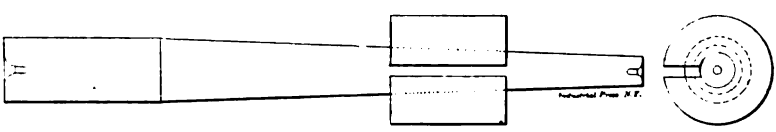

Fig. 63. Inside Lap.

to prevent the gage becoming "bell muzzled" while lapping. After the gage is finished, the thin projecting web *C* is ground off, leaving a good straigth hole. The lap used for inside work is shown at Fig. 63. The lap can be made to always fit the gage by merely forcing the lap further along on the taper arbor; the lap being slotted allows it to expand. In making a ring gage having a taper hole or a taper plug gage, it is necessary to employ a different method of lapping, as it is impossible to lap a taper hole with a taper lap. The facts regarding lapping are these: First, the lap must fit the hole at all times; secondly, the lap must constantly be moved back and forth. Therefore, if a taper lap is made to fit the taper hole it will lock

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and not revolve. If held in one place the lap will quickly assume the uneven surface of the hole. If the operator attempts to lap a taper hole by constantly revolving the gage on a straight lap he will surely dwell longer in one place than another, thereby making a hole that is anything but round. The following method is, therefore,

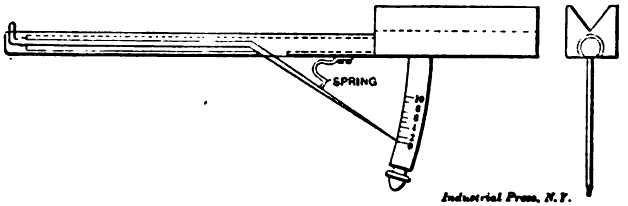

Fig. 64. Indicator for Testing Truth of Holes.

used: Having ground the hole to size, plus allowance for lapping, then, without disturbing position of slide rest or grinder head, change the emery wheel for a lap made of copper—of the same shape as the emery wheel with the exception of having a wider face—and lap in the same manner as the hole was ground, care being taken not to "crowd" the lap.

CHAPTER VII.

THE ROTARY LAP.

In Fig. 65 is shown a rotary lap 24 inches in diameter, intended to run at a speed about 300 revolutions per minute. The engraving will give a clear idea as to the construction. Some men think that if the lap has a true, flat surface, any one can produce true work, but such is not the case; it requires considerable skill, and that skill can be acquired only by long practice. Many machine operations can be shown to another person and the principle grasped readily, but not so with lapping. A great deal of skill is required in lapping thin pieces, small straight-edges or long narrow bars. It is possible, and requires no skill at all, to lap a thin piece of steel, convex or concave, by using a little more pressure in one place than another, and if the surface of the lap is not kept sharp it will soon heat and warp the work out of true. For a rotary lap kerosene and gasoline used together give the best results; but a hand lap should always be used dry. Keeping the surface of the rotary lap straight and true is very important and requires good judgment in using it. The outer edge runs so much faster than the inside that it is obvious that if it is used too much, either in the middle, inside, or outside, hollow places will be worn in the surface, so that it is a good plan to use the lap

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all over and try the surface frequently with a straight-edge, favoring the high places when using it.

The rotary lap is charged by sprinkling carborundum over the surface when not in motion, and then pressing it in by rubbing with a piece of round iron held in both hands. An old pepper box with a perforated top is just the thing to use for sprinkling carborundum.

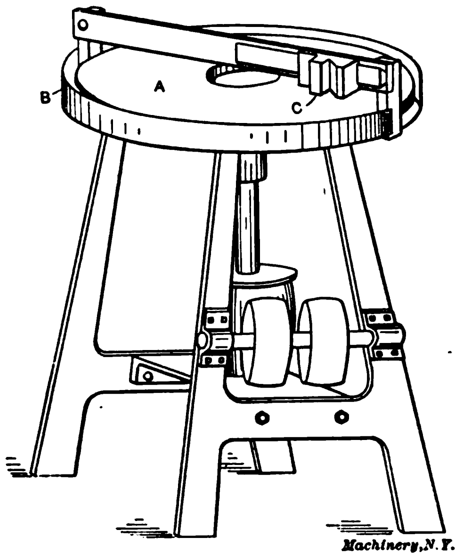

Fig. 65. The Rotary Lap.

The lap in Fig. 65 is made of an ordinary cast iron disk *A* with ribs on the bottom and anchor grooves on the top. It is also provided with a hub and a shaft. The end of the shaft supports the whole weight of the lap and runs on a hardened convex disk. A coating

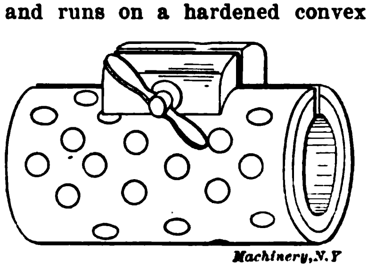

Fig. 66. Lap for Cylindrical Work.

of lead is cast over the surface of the lap, and then hammered to make it compact. A galvanized iron pan *B* is provided, the edges of which project above the surface of the lap to prevent the liquid, or whatever is used, from flying off, and onto everything around. Another handy device on this lap is a bar which is provided with ways and a sliding head *C* which can be pushed from the outer edge to the center of the lap. The bar is fastened to lugs which project

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on opposite sides of the frame, and can readily be removed when not in use. The bar is also provided with adjusting screws to set it parallel with the surface of the lap or to set the sliding head square with the surface. The sliding head has a square corner and an angle groove which can be used for lapping the ends of round or square pieces.

The engraving Fig. 66 shows a good way of making a lap for cylindrical work in the lathe. A piece of wrought iron pipe about 2 inches long, with a number of 5/16-inch holes drilled through will answer the purpose. Face one end of the pipe so that it will stand level on a surface plate; wrap a piece of heavy paper around the outside, using rubber bands to hold it on, then form a lug out of pasteboard and clamp that on any side of the pipe so that several of the holes open into it; secure a mandrel the same size as the piece to be lapped,

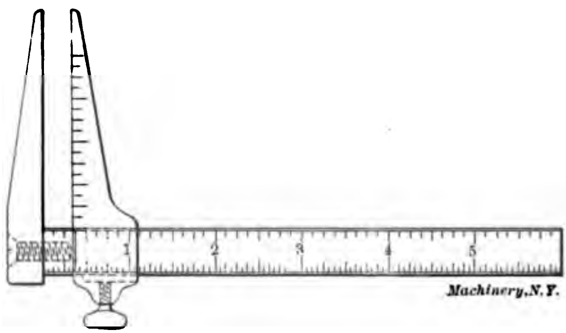

Fig. 67. Tool Used for Lapping when Sizing Duplicate Parts.

twist a piece of string in a spiral around the mandrel, insert it in the center of the pipe, and pour the molten lead in. When cool, drive out the mandrel and proceed to drill and tap a hole for a thumb-screw in the center of the lug; then slit one side through the center of the lug with a hack saw, and file off all sharp edges and burrs so that it will not injure the hands. In lapping, the faster the lap is drawn back and forth over the work the more nearly straight it will be, and as the lap wears and works easy, the thumb-screw is given a slight turn to keep it in contact with the work.

A very handy tool, which is shown in Fig. 67, is almost indispensable in doing work with the rotary and hand laps; it is a home-made caliper square. The taper between the hardened jaws is 0.001 inch in the whole length, and a 6-inch flexible scale is inserted in the beam. One jaw is graduated, which enables one to see how far the piece will slide up in the jaws. This caliper is not used so much for accurate measurements as for accurate sizing for parallelism or duplicating sizes.

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No. 39

FANS, VENTILATION AND HEATING

By CHARLES L. HUBBARD

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CHAPTER I

CENTRIFUGAL AND DISK FANS

There are two types of fans in common use, known as the centrifugal fan or blower, and the disk fan or propeller. The former consists of a number of straight or slightly curved blades extending radially from an axis as shown in Fig. 1. When the fan is in motion the air in contact with the blades is thrown outward by the action of centrifugal force and delivered at the outer circumference or periphery of the wheel. A partial vacuum is thus produced at the center of the wheel, and air from the outside flows in to take the place of that which has been discharged. Fig. 3 illustrates the action of a centrifugal fan, the arrows showing the path of the air. This type of fan is usually enclosed in a steel plate casing of such form as to provide

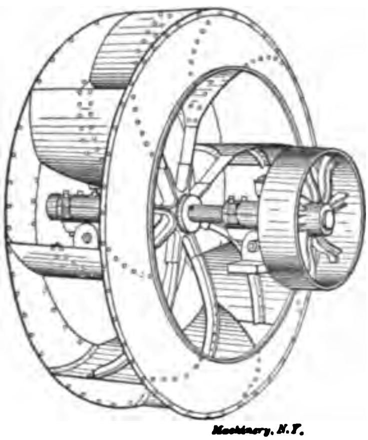

Fig. 1. Centrifugal Fan

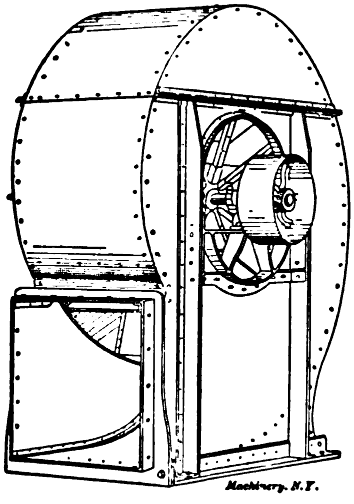

Fig. 2. Centrifugal Fan and Casing

for the free movement of the air as it escapes from the periphery of the wheel. An opening in the circumference of the casing serves as an outlet into the distributing ducts which carry the air to the various rooms to be ventilated, or to the furnaces in the case of mechanical draft

A fan with casing is shown in Fig. 2. The discharge opening can be placed in any position desired, either up, down, top horizontal, bottom horizontal, or at any angle. Where the height of the fan room is limited, a form called the three-quarter housing may be used, in which the lower part of the casing is replaced by a brick pit below

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the floor level (see Fig. 4). Another form of the centrifugal fan is shown in Fig. 6. This is known as the cone fan and is commonly placed in an opening in a brick wall and discharges air from its entire periphery into a room called a plenum chamber with which the various distributing ducts connect. This fan is often made double by placing two wheels back to back and surrounding them with a steel casing in a similar manner to the one shown in Fig. 2.

Cone fans are very efficient and are capable of moving large quantities of air at moderate speeds. Fig. 5 shows a form of small direct-connected exhauster commonly used for ventilating toilet rooms, chemical hoods, etc., and for furnishing a forced draft for forges and small boilers. Centrifugal fans are used almost exclusively for sup-

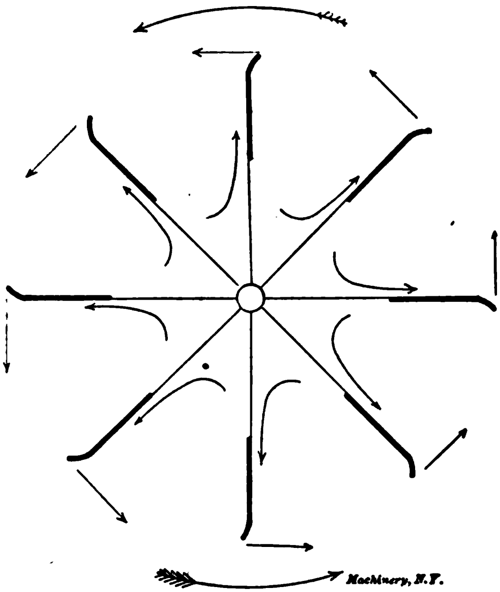

Fig. 3. Direction of Flow in Centrifugal Fan

plying air for the ventilation of buildings, for forced blast heating and for mechanical draft. They are also used as exhausters for removing the air from buildings when the resistance is considerable, and the quantity of air to be handled is large.

The disk fan is similar in construction to the propeller of a vessel and moves the air in lines parallel to its axis. This fan is made in various forms with both flat and curved blades. Fig. 7 shows one of the various designs arranged either for belted or direct-connected motor. This type of fan is light in construction, requires but little power at low speeds and is easily erected. It is especially adapted to exhaust ventilation when the resistance is small, being conveniently placed in the attic or upper part of a building and driven by an electric motor. Disk fans are largely used for the ventilation of public

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toilet rooms, smoking rooms, restaurants, etc., and are often connected with the main vent flues of large buildings, such as schools, halls, churches, theaters, etc. They are especially adapted for use in connection with gravity heating systems where the flow of air through vent flues is apt to be sluggish in mild weather.

Theory of Centrifugal Fans

The action of a fan is affected to such an extent by the various conditions under which it operates that it is impossible to give fixed rules for determining the exact results to be expected in any particular instance. This being the case, it seems best to take up the subject briefly from a theoretical standpoint, and then show what corrections are necessary in the case of a given fan under actual working conditions. As already stated, the rotation of a fan of this type sets in

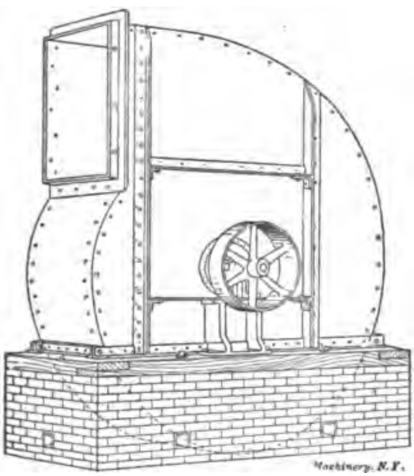

Fig. 4. Fan Partly Encased by Brick Wall

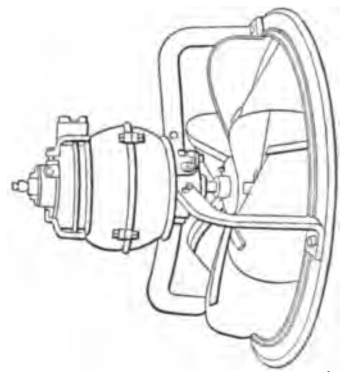

Fig. 5. Ventilator Wheel

motion the air between the blades, which by the action of centrifugal force is delivered at the periphery of the wheel into the casing surrounding it. As the velocity of flow through the discharge outlet depends upon the pressure or head within the casing, and this in turn upon the velocity of the blades, it becomes necessary to examine briefly into the relations existing between these quantities.

If a vessel as shown in Fig. 8 be filled with water, a certain pressure will be exerted upon the bottom, depending upon the depth and temperature of the water. If the weight of a cubic inch of water at a temperature of 50 degrees is 0.036 of a pound (called its *density* at that temperature) and the depth of the water in the vessel is 20 inches, then the pressure upon each square inch of the bottom will be the weight of a column of water having a sectional area of one square inch and a height of 20 inches, which in the above case is 20 ×

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0.036 = 0.72 of a pound; so that for general use we may write $p = h \times d$, or $h = \frac{p}{d}$, in which

h = the height of the column of water, called the head,
d = the density of the water,
p = the pressure produced.

If h is taken in inches and d in ounces per cubic inch, then p will be in ounces per square inch. If h is in feet and d in pounds per cubic foot, p will be in pounds per square foot and so on, depending upon the units taken. When dealing with water pressure it is customary to take such units as will give p in pounds per square inch.

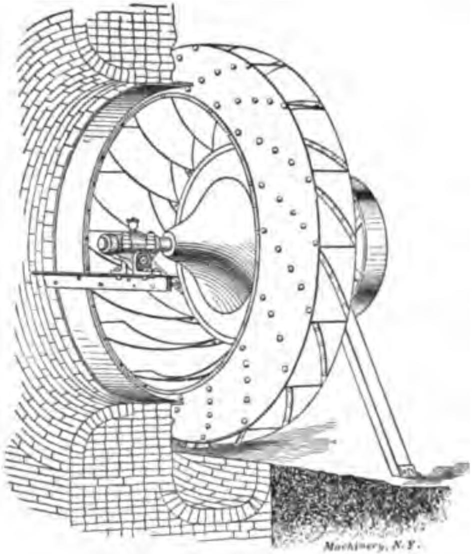

Fig. 6. Cone Fan Placed in Opening in Brick Wall

In the case of air pressure in connection with fans, ounces per square inch is the expression commonly used.

If a pipe be inserted in the side of the vessel (Fig. 9) at any given distance from the surface of the water, and a supply of water be provided sufficient to keep the level constant, a pressure of $h \times d$ will be exerted at the entrance to the pipe, causing the water to flow through it. The height h in this case is called the total head producing flow, and is divided into three parts, as follows: The *entry head*, that required to overcome the resistance to entry into the pipe; the *friction* or *pressure head*, that required to overcome the resistance due to the friction of the water in the pipe, and the *velocity head*, which is that used in giving motion or velocity to the water flowing through the pipe

The entry head depends upon the form of entrance to the pipe, and with smooth rounded edges is inappreciable. The friction head de-

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pends upon the length and size of the pipe, the interior surface, the number of bends and the quantity of water flowing through it. The velocity head is the same as the height through which a body must fall in a vacuum to acquire the velocity with which the water flows into the pipe. This is given by the formula for falling bodies.

$$h = \frac{v^2}{2g} \text{ in which}$$

h = the head in feet,

v = the velocity in feet per second,

g = the acceleration due to gravity, which is equal to 32.16.

This may also be written in the form of $v = \sqrt{2gh}$.

In applying this to the flow of air from a fan casing, it is customary to consider only the last two, and if the outlet is short and properly formed, the friction head may be neglected also. When the fan is

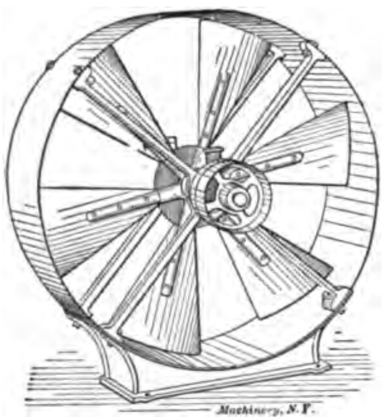

Fig. 7. Disk Fan

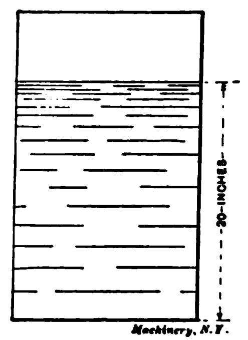

Fig. 8

to be used for moving air through ducts and flues in its practical application to ventilation, the effect of frictional resistance is added and must be provided for as stated later. We have seen that $v = \sqrt{2gh}$, and that $h = \frac{p}{d}$. Substituting $\frac{p}{d}$ for h in the above formula, we have $v = \sqrt{2g\frac{p}{d}}$ in which v is the velocity in feet per second of a liquid flowing from one chamber into another, where the difference in pressure is p and the density is d.

Applying this to the case of dry air at a temperature of 50 deg. F, and allowing for the change in density as it passes from a higher to a lower pressure, the formula becomes for the small differences in pressure employed in ventilating work,

$$v = \sqrt{\frac{1746659 \times p}{235 + p}}$$

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in which v is in feet per second, and p in ounces per square inch.

Table I, computed from the above formula, is taken from "Mechanical Draft," published by the B. F. Sturtevant Co., and gives the velocity of dry air at a temperature of 50 deg. F. flowing into the atmosphere under different initial pressures.

A simple approximate formula giving very nearly the same results for air at 50 deg. is $v = 65.5\sqrt{h}$, in which v is the velocity in feet per

TABLE I. VELOCITY OF DRY AIR AT 50 DEGREES F. TEMPERATURE

Pressure in Ounces per square inch.	Feet per Second.	Feet per Minute.
¼	43.08	2585.0
⅜	52.75	3165.1
½	60.90	3653.8
⅝	68.07	4084.0
¾	74.54	4472.6
⅞	80.50	4829.7
1	86.03	5161.7
1¼	96.13	5768.0

second as before, and h the pressure expressed in inches of water as indicated by the balanced height of a column of that liquid in a water gage. Pressure in inches of water column may be reduced to ounces per square inch by multiplying by 0.58.

Example.—A pressure of ¼ of an ounce will produce a velocity

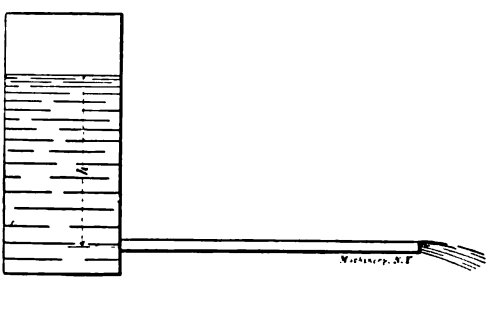

Fig. 9

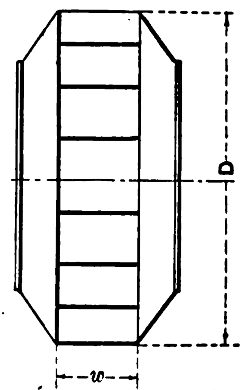

Fig. 10

of 43.08 feet per second, and a pressure of 1 ounce will produce a velocity of 86.03 feet per second, and so on.

The pressure within a fan casing is caused by the air being thrown from the tips of the blades, and varies with the velocity of rotation, that is, the higher the speed of the fan the greater will be the pressure produced.

When the various dimensions of a fan and casing are properly proportioned, the velocity of air-flow through the outlet will be the

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same as that of the tips of the blades, and the pressure within the casing will be that corresponding to this velocity. From this, it is evident that by knowing the diameter and speed of any given fan we can determine the peripheral velocity and find at once from Table I the pressure produced within the casing.

Blast Area

When the outlet from a fan casing is small, the air will pass out with a velocity equal to that of the tips of the blades, and the pressure within the casing will be that corresponding to the tip velocity. Now if the opening be slowly increased, while the speed of the fan remains constant, the air will continue to flow with the same velocity until a certain size is reached. The pressure in the casing will now begin to drop and the velocity of outflow become less than the tip

TABLE II. REVOLUTIONS PER MINUTE PRODUCING A GIVEN PRESSURE

Diam. of Fan Wheel in feet.	Pressure in Ounces per Square Inch.								
	¼	⅜	½	⅝	¾	⅞	1	1¼	1½
2′	411	504	582	650	712	769	822	918	1005
2½′	329	403	465	520	570	615	657	734	804
3′	274	336	388	433	475	513	548	612	670
3½′	235	288	332	372	407	439	469	525	574
4′	206	252	291	325	356	384	411	459	502
4½′	183	224	258	289	316	342	365	408	447
5′	164	202	232	260	285	308	329	367	402
6′	137	168	194	217	238	256	274	306	335
7′	117	144	166	186	203	220	235	262	287
8′	103	126	146	163	178	192	205	230	251
9′	92	112	129	144	158	171	183	204	223
10′	82	101	116	130	142	154	164	184	201

or peripheral velocity. The effective area of outlet at the point when this change begins to take place is called the *capacity area* or *blast area* of the fan. This varies somewhat with different types and makes of fans, but for the common form of blower it is approximately one-third of the projected area of the fan opening at the periphery, that is $\frac{D\,w}{3}$, in which D is the diameter of the fan wheel and w its width at the circumference (see Fig. 10).

Table II gives the speed of fans of different diameters necessary to maintain various pressures over an effective area equal to, or less than the blast area of the fan. The speeds given are in revolutions per minute, and are taken from "Mechanical Draft."

As a matter of fact the outlet of a fan casing is always made larger than the blast area, so that in actual practice the figures given in the following table must be corrected by certain factors as explained later.

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Theoretical Capacity

If we assume the effective outlet area of a fan to be equal to the blast area, its capacity at any given speed can be computed as shown in the following example. A fan 5 feet in diameter has a width of 2 feet at the tips of the blades. What quantity of air will it discharge at a speed of 200 revolutions per minute?

Taking the blast area as $\frac{D\,w}{3}$, we find it to be $\frac{5 \times 2}{3} = 3.33$ square feet.

At a speed of 200 revolutions the tip velocity of the fan will be $3.1416 \times 5 \times 200 = 3{,}141$ feet per minute.

Therefore the air delivered by the fan is $3.33 \times 3{,}141 = 10{,}459$ cubic feet per minute.

What will be the capacity of the same fan working under a pressure of ½ ounce, and what will be the required speed? Looking at Table I we find the velocity corresponding to ½ ounce pressure to be 3,653.8 feet per minute; therefore, the capacity of the fan is $3.33 \times 3{,}653.8 = 12{,}167$ cubic feet per minute.

TABLE III. VOLUME OF AIR DISCHARGED AND HORSE-POWER REQUIRED

Pressure in Ounces per Square Inch	Cubic feet of Dry Air at 50 degrees Temperature which will be Discharged through an Orifice having an Effective Area of 1 square inch.	Horse Power required to move the Given Volume of Air under the Given Conditions.
1/4	17.95	.00122
3/8	21.98	.00225
1/2	25.37	.00346
5/8	28.36	.00483
3/4	31.06	.00635
7/8	33.54	.00800
1	35.85	.00978
1¼	40.06	.01366

The required speed may be taken directly from Table II, where it is found to be 232 revolutions per minute.

Power Required to Move Air

The work done by a fan in moving air is represented by the pressure exerted multiplied by the distance through which it acts. This is expressed in foot-pounds by the equation $W = PAV$, in which

W = work done, in foot-pounds per minute,

P = pressure at discharge opening, in pounds per square foot,

A = area in square feet, over which pressure P is exerted,

V = the velocity of flow, through discharge outlet, in feet per minute.

The horse-power required for moving air through any given area of discharge is given by the formula

$$\text{H. P.} = \frac{d\,a\,v^3}{5{,}100{,}480} \text{ in which}$$

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d = density of the air at the given temperature,
a = effective area of discharge outlet, in square inches,
v = velocity of flow, in feet per second.

Table III (from "Mechanical Draft") gives the volume of air in cubic feet, which will be discharged per minute through an effective area of 1 square inch under different pressures; also the H. P. required for moving these quantities of air under the different conditions. This table gives only the power necessary for moving the air, and does not take into consideration the friction of the air in passing through the fan nor that of the fan itself. The additional power required to offset these losses will be taken up later.

Example: The effective area of a fan outlet is 480 square inches, and the pressure within the casing is ½ ounce per square inch. What volume of air will be discharged per minute, and what H. P. will be required, neglecting friction?

From Table III we find that for ½ ounce pressure, 25.37 cubic feet of air will be discharged per minute through an area of 1 square inch, at an expenditure of 0.00346 H. P. Therefore, the total quantity discharged will be $480 \times 25.37 = 12{,}177$ cubic feet, requiring $480 \times 0.00346 = 1.66$ H. P.

Relation Between Volume, Pressure and Power

It can be shown mathematically that the following relations are true in the case of an ideal fan, and tests have shown them to be approximately correct for fans in actual operation. (1) The volume of air delivered varies directly as the speed of the fan, that is, doubling the number of revolutions doubles the volume of air delivered. (2) The pressure varies as the square of the speed; for example, if the speed is doubled, the pressure is increased $2 \times 2 = 4$ times. (3) The power required to run a fan varies as the cube of the speed; that is, if the speed is doubled the power required is increased $2 \times 2 \times 2 = 8$ times.

The value of a knowledge of these relations may be illustrated by the following example. Suppose for any reason it was desired to double the volume of air delivered by a certain fan. At first thought we might decide to use the same fan and run it twice as fast; but when we come to consider that the power would have to be increased eight times, it is probable that it would be much cheaper in the end to use a larger fan and run it at a lower speed.

Effect of Temperature

All computations and tables given thus far have been based on a temperature of 50 degrees F.

Raising the temperature of air causes it to expand and therefore reduces its density, or weight per unit of volume. This fact is of much importance where fans are used for induced draft, as the temperature of the gases commonly ranges from 300 degrees to 600 degrees. Table IV, also from "Mechanical Draft," shows the effect on the speed and power of a fan when the temperature of the air is increased. In the following example it is assumed for simplicity that

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the effective outlet area is equal to the blast area in each of the fans considered.

Example: From Table II we see that a 4-foot fan running at a speed of 411 revolutions per minute, will produce a pressure of 1 ounce, and looking in Table III we find that it will discharge 35.85 cubic feet of air per minute at a temperature of 50 degrees through an effective area of 1 square inch, with an expenditure of 0.00978 H. P. If the width of the fan is 18 inches at the tips of the blades, the blast area may be taken as $\frac{48 \times 18}{3} = 288$ square inches, from which the delivery will be $35.85 \times 288 = 10{,}324$ cubic feet per minute, requiring $0.00978 \times 288 = 2.8$ H.P.

Let us now assume the air to be heated to a temperature of 500 degrees and see what conditions are necessary to handle the same weight per minute. Looking in Table IV for a temperature of 500 degrees we find that the volume becomes $10{,}324 \times 1.88 = 19{,}409$ cubic feet, and the speed of fan necessary to move this quantity is $411 \times 1.88 = 772$ revolutions per minute, requiring an expenditure of $2.8 \times 3.56 = 9.97$ H.P.

TABLE IV. EFFECT OF TEMPERATURE ON SPEED AND POWER

Temperature in Degrees F.	Volume for same Weight.	Relative Velocity due to the same Pressure.	Speed of Fan to Handle same Weight.	Speed to Produce same Pressure.	Power for Speed Required to Handle same Weight.	Power Required to Handle same Weight at same Pressure with a Properly Proportioned Fan.
1	2	3	4	5	6	7
50	1.00	1.00	1.00	1.00	1.00	1.00
100	1.10	1.05	1.10	1.05	1.21	1.10
150	1.20	1.09	1.20	1.09	1.43	1.20
200	1.29	1.14	1.29	1.14	1.67	1.29
250	1.39	1.18	1.39	1.18	1.93	1.39
300	1.49	1.22	1.49	1.22	2.22	1.49
350	1.59	1.26	1.59	1.26	2.51	1.59
400	1.68	1.30	1.68	1.30	2.84	1.68
450	1.78	1.34	1.78	1.34	3.18	1.78
500	1.88	1.37	1.88	1.37	3.56	1.88
550	1.98	1.41	1.98	1.41	3.92	1.98
600	2.08	1.43	2.08	1.44	4.32	2.08

Suppose the above fan to be used for supplying a forced draft of 1 ounce to a battery of boilers, and it is desired to change to induced draft where the gases are to pass through the fan at a temperature of 500 degrees, what size and speed of fan will be required to produce the same intensity of draft (suction in this case) and what horsepower will be required to run it?

It is evident that the weight of air required will be the same in each case. This, for forced draft, we found to be 10,324 cubic feet per minute at a temperature of 50 degrees, and Table I shows that a

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peripheral velocity of 5,161.7 feet per minute was required to produce the pressure of 1 ounce at this temperature.

Referring to Table IV, columns 2 and 5, we find that after raising the air to a temperature of 500 degrees a volume of $10,324 \times 1.88 = 19,409$ cubic feet per minute is to be handled by the fan, requiring a peripheral velocity of $5,161.7 \times 1.37 = 7,071.5$ feet per minute to maintain the same pressure. As the velocity of flow through the discharge outlet is practically the same as that of the fan tips, the required blast area of fan will be $19,409 \div 7,071.5 = 2.74$ square feet $= 394$ square inches. Assuming a fan 60 inches in diameter, we have $394 \div \frac{60}{3} = 19.7$, or in round numbers, 20 inches as the required width at the periphery. This gives a fan of very nearly the same proportions as the 4-foot fan first used. The circumference of a 6-foot fan is 18.8 feet, therefore, 7,071.5, the required peripheral velocity, divided by $18.8 = 376$ revolutions per minute, the required speed of the fan. The

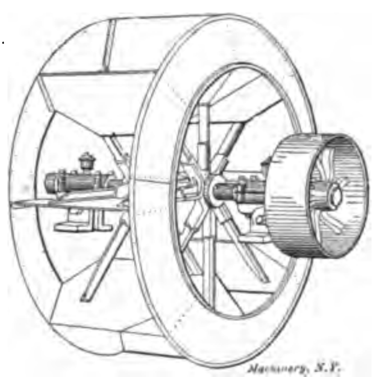

Fig. 11. Construction of Fan Wheel

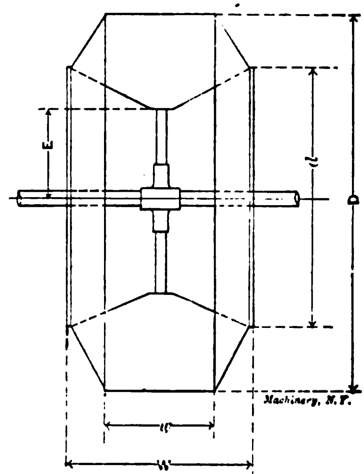

Fig. 12. Diagram of Fan Wheel

H. P. required by the 4-foot fan in handling the same weight of air at 50 degrees was 2.8. Referring to Table IV, column 7, we find that the power required to deliver the same weight of air at the same pressure at a temperature of 500 degrees is 1.88 times as great, or $2.8 \times 1.88 = 5.2$ H. P.

Having taken up the centrifugal fan from a theoretical standpoint and noted its action under ideal conditions, we will now consider it when working under the requirements of actual practice, and show what corrections must be made to the various rules and formulas previously given.

General Proportions

The general form of a fan wheel is shown in Fig. 11, which represents a double spider wheel with straight blades. Those over 4 feet

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in diameter usually have two spiders, while fans of large size are often provided with three or more. The number of blades or floats commonly varies from six to twelve, depending upon the size of fan. They are made both curved and straight; the former, it is claimed, run more quietly, but if curved too much will not work so well against a high pressure as the latter form. Fig. 12 represents a section

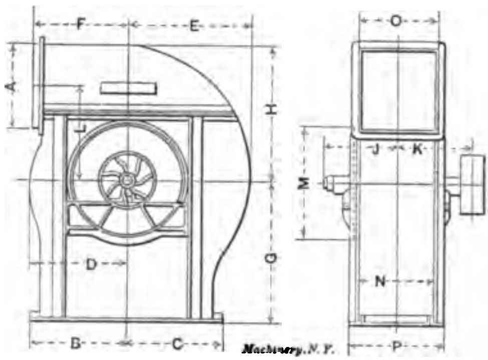

Diagram of Full Housing, for Use with Table V

through a fan wheel and shows the principal dimensions to be considered.

The following proportions are averages taken from fans of different sizes as made by several manufacturers for general ventilating and similar work and will be found to vary slightly from the proportions given by any one maker.

The diameter of the inlet (*d*) usually varies from 0.66 to 0.7 of the

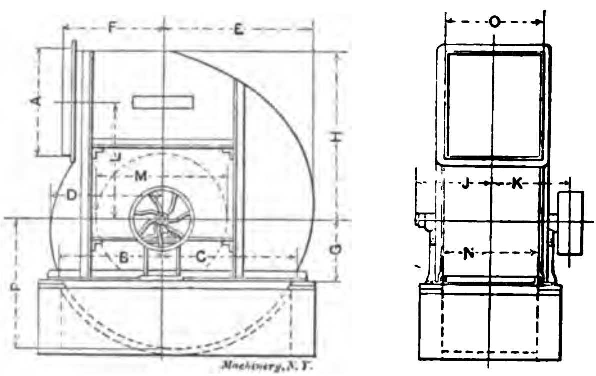

Diagram of Three-quarter Housing, for Use with Table VI

diameter of the wheel (*D*); 0.68 has been used in the following tables as a fair average. The distance from the center to the heel of the blades (*E*) is generally made 0.25 of the diameter of the wheel.

The width *W* varies somewhat in fans designed for different purposes. In the makes examined it averaged from 0.50 to 0.54 of the diameter, and 0.52 has been used in the tables following.

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TABLE V. DIMENSIONS OF FULL HOUSING FANS.

Size of Wheel.	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P
2½	18¾	20½	23½	21	25	24	28⅛	27	16¾	17⅛	17⅝	21	15	18¾	19¾
3	22¾	22	26	24	30	26¼	33¾	33	20¼	20⅜	21⅝	27½	18	22¾	22½
3½	25¾	24½	29½	27¼	34¼	28¾	39¼	37¾	22½	23⅜	24⅞	31⅞	21	25¾	27
4	28⅝	31	31	31	39	30½	41¼	43	22⅞	23½	29	34	24	29⅝	30¼
4½	31¼	34½	34½	34	44	34¼	41¼	49	24¾	25¾	34	38⅞	28	29½	34¼
5	33½	35½	35½	37	49	36⅛	47	55	27	28	39	42⅜	32	33½	38¼
5½	37¼	40½	40½	40½	53½	40½	51	60	29	30¼	42	47	36	37¼	44
6	42½	46	46	44	58	41½	55	65	33	34½	44	52¼	42	42½	51
7	48½	49	49	50	68	45½	64	77	36¼	38½	58	60½	48	48½	57
8	54½	57	63	57	77	49½	72	87	38	39¾	60	68½	48	48½	59
9	60¾	63	78	65	87	57½	80	98	41	42¾	68	77	54	54¾	65
10	72¾	66	84	72	96	68½	88	108	44⅜	47	72	85½	60	60¾	71

TABLE VI. DIMENSIONS OF THREE-QUARTER HOUSING FANS.

Size of Wheel.	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P
4	28⅝	29½	36	31	39	30½	15	43	22⅞	23½	29	34	24	24⅝	35
4½	31¼	32	40	34	44	34¼	18	49	24⅞	25¾	34	38⅞	28	29½	39
5	33½	34½	44	37	49	36⅛	20	55	26¾	27¾	39	42⅜	32	33½	43
5½	37¼	38¼	48¾	40½	53½	40½	20	60	29	30¼	42	47	36	37¼	47
6	42½	40¾	52	44	58	41½	24	65	33	34½	44	52¼	42	42½	51
7	48½	47	62	50	68	45½	26¼	77	36¼	38½	58	60½	48	48½	59
8	54½	52½	69	57	77	49½	32¼	87	38	39¾	60	68½	48	48½	67
9	60¾	60	78	65	87	57½	36	98	41	42¾	68	77	54	54¾	76
10	72¾	66	85	72	96	68½	40	108	44⅜	47	72	85½	60	60¾	84

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The width at the periphery should be, theoretically, such that the area of outlet around the entire wheel will be the same as the sum of the openings between the blades at the inlet, but in actual practice it is made somewhat greater, averaging from 0.7 to 0.8 of the width of the wheel. For convenience the relations between the different parts of the wheel may be expressed by the following equations:

$d = 0.68\,D,$ $\quad W = 0.52\,D,$

$E = 0.25\,D,$ $\quad w = 0.8\,W,$

in which

D = diameter of wheel,
d = diameter of inlet to wheel,
E = distance from center of fan to heel of blades,
W = width of fan at inlet,
w = width of fan at periphery.

These proportions, as already stated, do not represent those of any particular make, nor follow any fixed rule, but are general averages

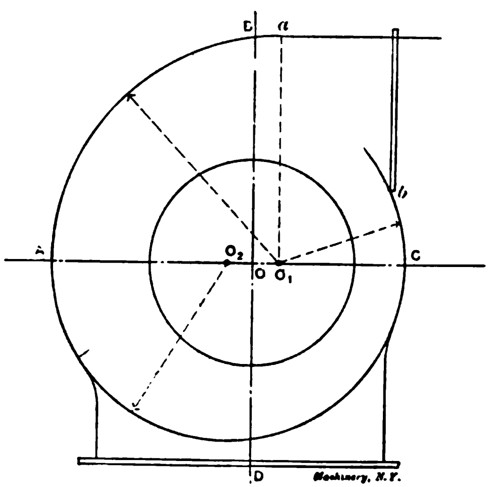

Fig. 13. Layout of Scroll Casing

as found from the catalogue dimensions of several well-known manufacturers.

Fans are made both with double and single inlets; the former being called "blowers" and the latter "exhausters."

The dimensions of the casing or housing vary somewhat with different makers and can be obtained from their catalogues. The dimensions given in Tables V and VI are taken from the catalogue of the B. F. Sturtevant Co. and will be found useful in the design of ventilating systems in approximating the space required. In case any particular make of fan is to be used, the exact dimensions should be obtained from the manufacturers, especially if the available space is limited.

The scroll of the casing is usually made up of the arcs of three

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circles of different radii, and the following method will be found useful in laying out a fan casing to scale. (See Fig. 13.)

First draw the center lines and lay off the distances *OA*, *OB*, *OC* and *OD* from the catalogue dimensions of the fan to be used. Then with a radius R, equal to *OB*, and a center at O_1 draw the arc *Aa*; with the same center and a radius R_1 equal to O_1C draw the arc *bC*; lay off $O\,O_2 = O\,O_1$ and with a radius R_2 equal to O_2A and a center at O_2 draw the arc *AC*. The other bounding lines can then be drawn in from dimensions taken from the catalogue table. The width of the casing is practically the width of the wheel with a small allowance for clearance.

Form of Orifice

The form of the opening through which the air passes when under pressure has a certain effect upon the quantity discharged, and makes it less than the theoretical amount which the size of opening and difference in pressures would indicate.

This reduction is due to two causes. First, to a contraction of the stream within, or just beyond the opening, depending upon its form; and second, to a certain amount of friction which tends to reduce the velocity somewhat.

The ratio between the actual quantity of air discharged and the theoretical quantity is called the "coefficient of discharge" and may be taken as about 0.8 for the short outlet from a fan casing, and as 0.56 for the inlet.

Example: The pressure within a fan casing is ½ ounce per square inch, and the area of the outlet is 4 square feet. What will be the actual discharge in cubic feet per minute?

From Table I the velocity corresponding to a pressure of ½ of an ounce is found to be 3,653.8 feet per minute, which gives a theoretical discharge of $3{,}653.8 \times 4 = 14{,}615.2$ cubic feet per minute. To find the actual quantity we must multiply this result by the coefficient of discharge, which gives us $14{,}615.2 \times 0.8 = 11{,}692.16$ cubic feet per minute.

The quantity of air discharged through any given opening, divided by the velocity of flow, is called the effective area. In the preceding example $11{,}692.16 \div 3{,}653.8 = 3.2$ square feet, which is the effective area of the outlet, while 4 square feet is the actual area.

Sometimes the conditions of the problem are such that it is stated as follows: The pressure within a fan casing is ½ ounce, and the outlet is 4 square feet, how much must the pressure be increased to make the actual discharge equivalent to the theoretical at the original pressure? It can be shown mathematically that the ratio of the theoretical pressure to the actual pressure required is $\frac{1}{K^2}$, in which K is the coefficient of discharge. Taking this as 0.8 for a fan outlet, the ratio becomes $\frac{1}{0.64} = 1.56$.

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In the above example the theoretical discharge is 3,653.8 × 4 = 14,615.2 cubic feet per minute, in which it is assumed that 4 square feet is the effective area of the outlet. In order to make the actual discharge equal the theoretical, using the same sized outlet, it will be necessary to increase the pressure by 1.56, which gives us ½ × 1.56 = 0.78 of an ounce.

Blast Area

While the blast areas of fans of different diameters are slightly different for varying proportions, the formula $\frac{D w}{3}$ applied to standard fans will be found sufficiently accurate for ordinary use, in view of the approximations which must be made later in the assumption of pressures to be operated against, due to the friction of the air in ducts and flues.

Assuming $w = 0.8\ W$, and substituting for W its equivalent, 0.52 D, we have for the blast area $A = D \times 0.8 \times 0.52\ D \times 0.33 = 0.14\ D^2$.

TABLE VII

Dia. of Fan in Feet.	Blast Area, in sq. ft.
3	1.26
3½	1.72
4	2.24
4½	2.84
5	3.50
6	5.04
7	6.86
8	8.96
9	11.34
10	14.00

Table VII gives the blast areas for fans of different diameters, computed by the above method, which will be found to correspond very closely with those calculated by more complex methods for fans of approximately the same proportions.

Actual Capacity

In the examples given under *Theoretical Capacity*, it was assumed that the effective area of outlet was equal to the blast area, so that the velocity of outflow could be taken the same as the tip velocity. In actual practice the effective area of outlet is always made greater than the blast area and consequently the actual volume of air discharged is greater than the theoretical. On the other hand, the pressure drops below that due to the tip velocity and the velocity of flow through the outlet is correspondingly less.

The size of discharge outlet varies somewhat for different makes, but for a large number of fans examined it was found to average about 2.23 times the blast area as computed by the preceding method. Assuming a coefficient of discharge of 0.8, it gives as the effective area of discharge, 0.8 × 2.23 = 1.78 times the blast area.

A series of carefully conducted tests made some time ago upon an enclosed fan of practically the proportions taken, showed the pressure producing the flow of air through the outlet to be about 0.7 of that due

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to the peripheral velocity, when the effective area of outlet was made 1.78 times the blast area as computed above. Calculations based upon tests made by one of the leading manufacturers of fans of similar proportions give practically the same result.

We have seen that the velocity corresponding to any given pressure, or in other words, the peripheral velocity necessary to produce any desired pressure may be found by the formula:

$$v = 65.5\sqrt{h}$$

for air at 50 degrees temperature, when the effective area of outlet is equal to or less than the blast area.

If increasing the effective outlet area to 1.78 times the blast area causes the pressure to drop to 0.7 that due to the peripheral velocity, we must, in order to again bring the pressure up to its original point, increase the velocity of the fan tips to a speed given by the equation

$$v = 65.5\sqrt{\frac{h}{0.7}}$$

Assuming an original pressure (h) of 1, it is found that the tip velocity must be multiplied by 1.2 in order to produce this result.

This may be made clearer by an illustration: A 6-foot fan running

TABLE VIII. VOLUME OF AIR DISCHARGED PER MINUTE IN CUBIC FEET

Dia. of Fan.	¼ oz.	⅜ oz.	½ oz.	⅝ oz.	¾ oz.	⅞ oz.	1 oz.	1¼ oz.
3	5 690	6,960	4,380	8,980	9,840	10,600	11,350	12,700
3½	7,750	9,490	10,950	12,250	13,400	14 500	15,500	17,300
4	10.350	12.650	14,600	16,350	17,900	19,300	20.650	23,100
4½	12,950	15,850	18,250	20,400	22,350	24,150	25,800	28,850
5	16,050	19,600	22,650	25,500	27,750	29,950	32,000	35,750
6	23,250	28,600	32,900	36,750	40,250	43.450	46,450	51,900
7	31,550	38,600	44.600	49,800	54,550	58,950	62,950	70,350
8	40 850	50.000	57,750	64,550	70,650	76.300	81,550	91,150
9	53,200	63,950	73 750	82,500	90,350	97,560	104 250	116,500
10	64,100	78,500	90,600	101,300	110,950	119.800	128,000	143,050

at a speed of 194 revolutions per minute produces a pressure of ½ ounce with a discharge outlet having an effective area equal to the blast area. If the effective discharge outlet is made 1.78 times the blast area, at what speed must the fan be run to maintain the same pressure, that is ½ ounce?

$$194 \times 1.2 = 233 \text{ revolutions per minute.}$$

Table VIII gives the cubic feet of air discharged per minute by fans of different diameters when run at such speeds as will produce the pressures indicated at the head of each column. These results were obtained by assuming the effective area of discharge outlet equal to 1.78 times the blast area, and multiplying this area in square inches by the quantities for the corresponding pressures as given in Table III. The results are given to the nearest ten, for quantities less than 10,000, and to the nearest fifty for those above 10,000.

Table IX gives the speeds of fans of different sizes necessary to

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maintain various pressures over effective discharge areas equal to 1.78 times the blast areas. These results are obtained by multiplying the speeds given in Table II by 1.2.

Horse-power Required

The power required for moving a given quantity of air under different conditions is given in Table III. This, however, does not include that necessary for overcoming the friction of the fan or the passage of the air through it.

The efficiency of a fan varies with the speed, the size of outlet, and the pressure against which it is working. Under favorable conditions, properly proportioned fans should have an efficiency of about 40 per cent, although they often fall considerably below this. The horse-power given in Table X for different sized fans is obtained by multiplying the effective area of outlet, in square inches (blast area × 1.78) by

TABLE IX. REVOLUTIONS PER MINUTE OF CENTRIFUGAL FANS

Dia. of Fan.	¼ oz.	⅜ oz.	½ oz.	⅝ oz	¾ oz.	⅞ oz.	1 oz.	1¼ oz.
3	328	403	465	531	570	615	657	734
3½	282	345	398	446	488	526	562	630
4	247	302	349	390	427	460	493	550
4½	219	268	309	346	379	410	438	489
5	196	242	278	312	342	369	394	440
6	164	201	232	260	285	307	328	367
7	140	172	199	223	243	264	282	314
8	123	151	175	195	213	230	246	276
9	110	134	154	172	189	205	219	244
10	98	121	139	156	170	184	197	220

the quantities given in column three of Table III, and dividing the result by 0.4 (the efficiency).

Effect of Resistance

The effect of adding resistance to the flow of air from a fan by connecting it with a series of ventilating ducts or the furnace of a boiler is the same as would result from partially closing the discharge outlet.

Carefully conducted tests upon this type of fan have shown that the reduction of air flow is very nearly in proportion to the reduction of discharge area. That is, if the outlet of the fan is closed to one-half its original area the quantity of air discharged will be practically one-half that delivered by the fan with a free opening. Tests also show that the required horse-power varies approximately as the quantity of air discharged. The effect of attaching a fan to the ventilating flues of a building like a schoolhouse, church, or hall, where the ducts have easy bends and the velocity of air flow through them is not over 1,000 to 1,200 feet per minute, is about the same as closing the outlet from 10 to 20 per cent. For factories, with deep heaters and smaller ducts, where the velocity of air flow runs up to 1,500 or 1,800 feet per minute, the effect is equivalent to closing the outlet from 20 to 30 per cent, or even more in very large buildings.

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For the average mechanical draft plant, the effect is about the same as for factory ventilation, that is, a reduction of area from 20 to 30 per cent.

In buildings similar to schoolhouses and churches, it has been found in practice that fans of the blower type, having curved floats, operate quietly and give good results when run at a speed corresponding to one-half ounce pressure at the discharge outlet; this gives a speed of about 3,600 feet per minute at the circumference of the wheel. Higher speeds are accompanied with greater expenditure of power and are likely to produce a roaring noise or cause vibration. A much lower speed does not provide sufficient pressure to give proper control of the air distribution during strong winds. For factories and similar buildings, a higher pressure of three-fourths ounce or more is generally employed. In the case of mechanical draft, a pressure of three-fourths to one ounce is usually required for forced draft, and from one-half to three-fourths ounce for induced draft.

Example: A schoolhouse requires an air supply of 52,000 cubic feet

TABLE X. HORSE-POWER REQUIRED FOR CENTRIFUGAL FANS

Dia. of Fan.	H. P. Required for Different Pressures.							
	¼ oz.	⅜ oz.	½ oz.	⅝ oz.	¾ oz.	⅞ oz.	1 oz.	1¼ oz.
3	1.0	1.9	2.8	3.8	5.0	6.4	7.9	10.8
3½	1.5	2.5	3.8	5.4	7.0	8.8	10.8	15.0
4	1.8	3.3	5.0	6.9	9.1	11.5	14.0	19.6
4½	2.3	4.0	6.8	8.8	11.5	14.5	17.8	24.8
5	2.8	5.0	7.8	10.8	14.3	17.9	21.9	30.5
6	3.9	7.8	11.1	15.8	20.4	25.8	35.1	44.0
7	5.3	10.0	15.8	21.3	27.9	35.1	42.3	60.0
8	7.0	13.0	20.0	28.0	36.7	46.4	56.8	81.0
9	8.8	16.1	25.0	34.8	45.6	57.5	70.4	98.5
10	11.0	20.3	31.1	43.5	57.1	72.0	88.0	122.9

of air per minute. What size and speed of fan will be required, and what will be the necessary horse-power of engine, assuming the resistance to be equal to reducing the discharge outlet 20 per cent?

As the effect of the resistance is the same as reducing the discharge opening, and consequently the air quantity, 20 per cent, we must look in Table VIII for a fan capable of delivering $\frac{52,000}{0.8} = 65,000$ cubic feet of air per minute. In the column headed ⅝ ounce pressure we find that an 8-foot fan will deliver 64,550 cubic feet, and is the size we should use.

Table IX shows us that a fan of this size must be run at a speed of 195 revolutions per minute to produce a pressure of ⅝ ounce. In Table X we find that 28 horse-power is required to run the fan at a speed to produce this pressure when discharging into free air. Under the conditions of the problem, with the outlet restricted 20 per cent, the required power would be $28 \times 0.8 = 22.4$ H. P.

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Actually the pressure is increased slightly by restricting the outlet at constant speed, but this is seldom taken into account in ventilating work as air volume and power required are the quantities sought.

Mechanical Draft

Theoretically, about 12 pounds of air are required to burn one pound of average coal or coke. In practice, however, with natural draft, it is necessary to supply about twice this amount, to secure complete combustion, owing to the difficulty of bringing the air into contact with the entire body of coal. With mechanical draft and deeper fires, this quantity may be reduced to about 18 pounds per pound of coal, which is equivalent in round numbers to 230 cubic feet at a temperature of 50 degrees F. Assuming a coal consumption of 5 pounds per horse-power per hour, calls for an air supply of $5 \times 230 = 1{,}150$, or practically 1,200 cubic feet of air per boiler horse-power per hour.

The size of fan and power required for supplying a given quantity of air at a stated pressure may be determined in the same manner for forced draft as illustrated in the preceding example, except that the resistance may be taken as equivalent to a reduction of about 25 per cent instead of 20 per cent in air volume and horse-power, at a given speed or pressure.

In the case of induced draft a correction must be made for the effect of higher temperature of gases passing through the fan. The proper size and speed of fan in this case, together with the power of engine, may be found by first computing them for an air temperature of 50 degrees and then making corrections as shown below.

Let us assume the average temperature of the gases from the furnace to be 550 degrees. Referring to Table IV, we find that the volume of a given weight of air at this temperature becomes 1.98 times that at 50 degrees (column 2), and that the peripheral speed of fan required to produce a given pressure is 1.41 times as great (column 5), also that the necessary power is increased 1.98 times when a properly proportioned fan is used (column 7). The horse-power can evidently be obtained at once by multiplying that already computed by 1.98.

As we wish to keep the pressure the same, and to move the same weight of air, it will be necessary to increase the peripheral velocity of the fan by 1.41; the speed required for doing this can be fixed after determining the size of fan. From column 3 we find that the relative velocity of flow due to the same pressure at this higher temperature is 1.41. Therefore, if the pressure is to remain constant, the volume of air moved 1.98 times as great, and the velocity of flow through the outlet 1.41 times that of air at 50 degrees, the effective area of outlet must be $\frac{1.98}{1.41} = 1.4$ times that of the fan used for delivering the air at the lower temperature.

We may then select a fan having approximately this size of outlet and obtain the necessary speed in revolutions by dividing the required peripheral velocity by the circumference of the fan wheel.

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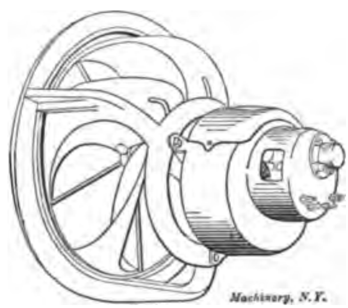
For other temperatures, the method will be the same, although the ratios from Table IV will vary.

Example: Determine size and speed and power required for driving a fan used in producing an induced draft of ¾ ounce for a battery of boilers aggregating 670 horse-power. We find first 670 × 1,200 = 804,000 cubic feet of air per hour, or 804,000 ÷ 60 = 13,400 cubic feet per minute. Assuming the resistance as equivalent to restricting the area of outlet 0.25, we must look in Table VIII under ¾ ounce pressure for a fan delivering 13,400 ÷ 0.75 = 17,866 cubic feet, where we find that a 4-foot fan will deliver 17,900 cubic feet.

Table IX shows us that a speed of 427 revolutions per minute is required to produce this pressure, and Table X gives the necessary power as 9.1, which multiplied by 0.75 is 6.8, or in round numbers, 7 horse-power. Assuming the temperature of the gases to be 550 de-



Figs. 14 and 15. Types of Disk Fans

grees, and referring to Table IV, we find the following ratios or multipliers: Volume, 1.98; velocity due to same pressure, 1.4; peripheral speed to produce same pressure, 1.4; power, 1.98.

The effective area of outlet of a 4-foot fan is 2.24 × 1.78 = 4 square feet; therefore the required area is $\frac{1.98}{1.4} \times 4 = 5.6$ square feet. The fan having an outlet area corresponding most nearly to this is a 5-foot, whose outlet is 3.5 × 1.78 = 6.2 square feet. Although this is slightly larger than called for, we can depend upon the automatic regulator to keep the speed at the required point to supply the necessary air volume. The required peripheral speed is 427 × 12.6 (circumference of 4-foot wheel) × 1.4 = 7,532 feet per minute. The circumference of a 5-foot fan is 15.7 feet; therefore, 7,532 ÷ 15.7 = 480 revolutions per minute, which is the required speed of the larger fan. The final horse-power is 7 × 1.98 = 13.86, or 14 in round numbers.

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Disk Fans

The capacity of disk fans varies greatly with the type and conditions under which they operate. The rated capacities given in catalogues are for fans revolving in free air; that is, mounted in an opening without being connected with ducts or working against a resistance. Disk fans of the type shown in Figs. 14 and 15, when working against a low resistance such as is commonly encountered in ventilating work where the air is drawn or forced through ducts of medium length at velocities not exceeding 600 or 800 per minute, propel the air in a direction parallel to the shaft, a distance equal to about 0.6 of the diameter for each revolution. From this we have the equation $C = 0.6\,D \times R \times A$, in which

C = cubic feet of air delivered per minute,
D = diameter of fan in feet,
R = revolutions per minute,
A = area of fan in square feet.

TABLE XI

Dia. of Fan, Inches	Cubic ft. per Revolution
12	0.5
18	1.7
24	4.0
30	7.8
36	13.0
42	21.0
48	32.0
60	62.0
72	108.0

In order to obtain the best results the linear velocity of air-flow through the fan should not average over 1,000 feet per minute, nor exceed 1,200 feet as a maximum.

Table XI gives the volume of air delivered per revolution for fans of different diameter based upon the above formula.

TABLE XII

Velocity through Fan in feet per minute	Horse-power required per 1,000 cubic feet of air removed
600	0.12
700	0.20
800	0.30
900	0.35
1,000	0.40
1,100	0.45
1 200	0.55

The power required per cubic foot of air delivered, depends upon the velocity of flow through the fan.

Table XII gives the horse-power required per thousand cubic feet of air for different velocities moved through the fan.

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CHAPTER II.

HEATERS FOR HOT BLAST AND VENTILATION

The best type of heater for any particular case will depend upon the volume and final temperature of the air, the steam pressure and the available space. When the air is to be heated to a high temperature for both warming and ventilating a building as in the case of a shop or mill, or for drying purposes, heaters of the general form shown in Figs. 16, 18, and 19 are used. These may also be adapted to all classes of work by varying the proportions as required. They can be made shallow and of large superficial area for the comparatively

Fig 16. Sturtevant Miter Type Heater

low temperatures used in purely ventilating work, or deeper, with less height and breadth, as higher temperatures are required.

Description of Types of Heaters

Fig. 18 shows the general construction of the standard hot blast heater of the B. F. Sturtevant Company. This consists of several sectional cast iron bases with loops of wrought iron pipe connected as shown. The steam enters the upper part of the bases or headers and passes up one side of the loops, then across the top and down on the other side, where the condensation is taken off through the return drip, which is separated from the inlet by a partition. These heaters are made up in sections of 2 and 4 rows of pipes each, and can be made any depth desired by adding more sections. The height varies

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from 3½ to 9 feet and the width from 3 feet to 7 feet in the standard sizes. They are usually made up of one-inch pipe, although 1¼ inch is commonly used in the larger sizes.

For convenience in estimating the approximate dimensions of a heater, Table XIII is given. The standard heaters made by different manufacturers vary somewhat, but the dimensions given in the table represent average practice. Column 3 gives the square feet of heating surface in a single row of pipes of the dimensions given in columns 1 and 2, and column 4 gives the free area between the pipes.

In calculating the total height of the heater add 1 foot for the base. These sections are made up of 1-inch pipe except the last, or 7-foot sections, which are made of 1¼-inch pipe.

Fig. 16 shows the miter type of the Sturtevant heaters, with single-chambered inlet and outlet sections. This arrangement provides abso-

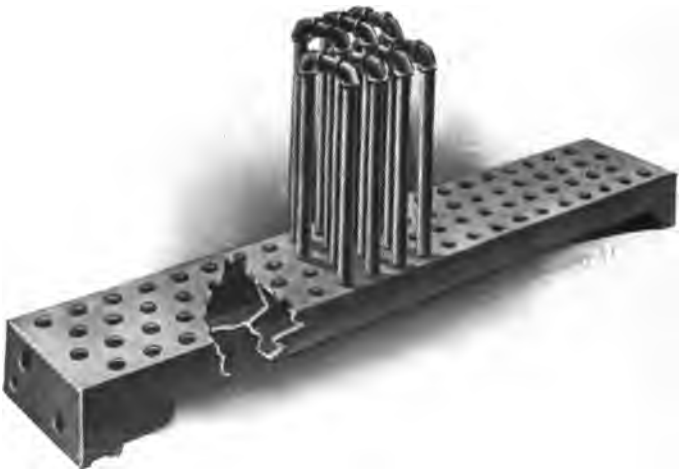

Fig. 17. American Blower Co.'s Hot Blast Heater, Four-pipe Section

lute freedom of expansion and perfect circulation. Steam is admitted at the top of the inlet section, and the drips removed from the end of the outlet section. Heaters of this type are usually enclosed in a steel casing as shown in Fig. 29, although brick walls are often used for heaters of large size.

Fig. 17 illustrates the construction of a 4-pipe section of the heater made by the American Blower Company, and Fig. 19 the same heater complete, without its steel plate casing. This heater is similar in appearance to the one just described, but differs somewhat in its construction. The base is divided lengthwise by an inside partition, so that the two pipes or legs of each loop connect with different chambers, one of which connects with the steam supply and the other with the return.

Fig. 20 shows a special form of heater particularly adapted to venti-

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lating work where the air does not have to be raised above 75 or 80 degrees. It is made up of 1-inch wrought iron pipe connected with supply and return headers; each section contains 14 pipes, that is, 2 pipes wide and 7 pipes deep, and they are usually made up in groups of 5 sections each. These coils are supported upon T-irons resting upon a brick foundation. Heaters of this form are usually made to

Fig. 18. Sturtevant Hot Blast Heater

extend across the side of a room with brick walls at the sides instead of being encased in steel housings.

Figs. 21, 22, and 24 show the "Vento" cast iron hot blast heaters made by the American Radiator Company. This type of heater is to be used under the same conditions as the pipe heaters already described. Fig. 21 shows a group of sections and illustrates the general construction

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and method of connection. Fig. 22 shows the sections arranged in a stack, five rows deep; and Fig. 24 the same stack with its steel casing and the supply and return connections.

Cast iron indirect radiators of the pin pattern shown in Fig. 23 are well adapted for use in connection with mechanical ventilation, and also for heating where the air volume is large and the temperature not too high, as in churches and halls. They make a convenient form of heater for schoolhouse and similar work, for being shallow, they can be supported upon I-beams at such an elevation that the con-

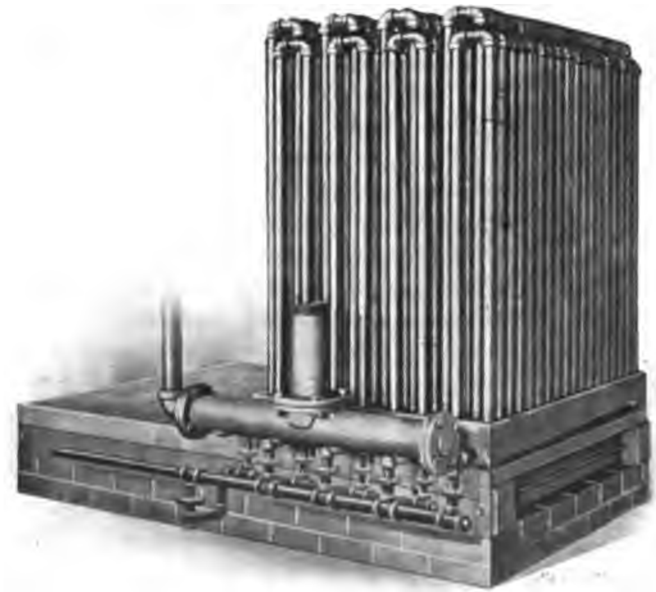

Fig. 19. American Blower Company's Heater Complete without Casing

densation may be returned to the boilers by gravity. In the case of vertical pipe heaters the bases are below the water-line of the boilers, and the condensation must be returned by the use of traps and pumps.

Efficiency of Pipe Heaters and Calculation of Sizes Required

The efficiency of the heaters used in connection with forced blast varies greatly, depending upon the temperature of the entering air, its velocity between the pipes, the temperature to which it is raised, and the steam pressure carried in the heater. The general method in which the heater is made up is also an important factor.

In designing a heater of this kind, care must be taken that the free area between the pipes is not contracted to such an extent that an excessive velocity will be required to pass the given quantity of air through it. In ordinary work it is customary to assume a velocity of 800 to 1,000 feet per minute; higher velocities call for a greater pressure on the fan which is not desirable in ventilating work.

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TABLE XIII

Width of Section	Height of Pipes Ft.	In.	Heating Surface, Square Feet	Free Area through Heater, Square Feet
3	3	6	20	4.2
3	4	0	22	4.8
3	4	6	25	5.4
3	5	0	28	6.0
4	4	6	34	7.2
4	5	0	38	8.0
4	5	6	42	8.8
4	6	0	45	9.6
5	5	6	52	11.0
5	6	0	57	12.0
5	6	6	62	13.0
5	7	0	67	14.0
6	6	6	75	15.6
6	7	0	81	16.8
6	7	6	87	18.0
6	8	0	92	19.2
7	7	6	98	21.0
7	8	0	103	22.4
7	8	6	109	23.8
7	9	0	116	25.2

In the heaters shown, about 0.4 of the total area is free for the passage of air; that is, a heater 5 feet wide and 6 feet high would

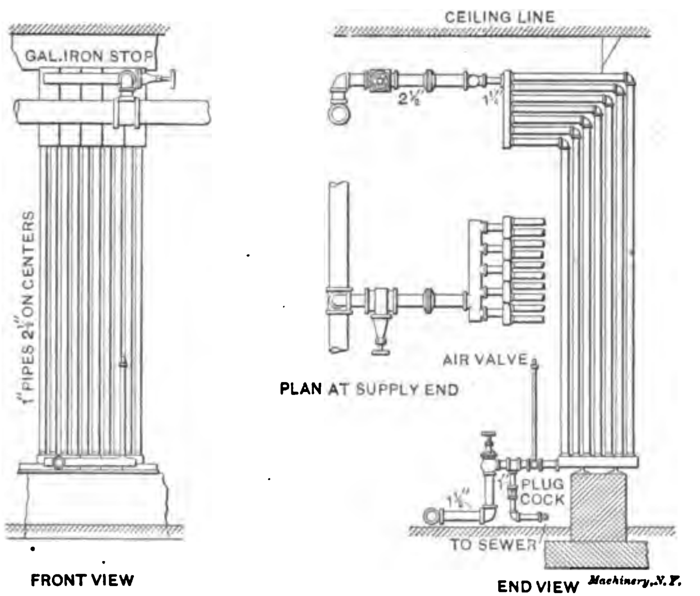

Fig 20 Special Heater used in Ventilating Work

have a total area of $5 \times 6 = 30$ square feet, and a free area between the pipes of $30 \times 0.4 = 12$ square feet. The depth or number of rows of pipe does not affect the free area, although the friction is increased and additional work is thrown upon the fan. The efficiency in any

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given heater will be increased by increasing the velocity of the air through it, but the final temperature will be diminished, that is, a larger quantity of air will be heated to a lower temperature in the second case, and while the total heat given off is greater, the air quantity increases more rapidly than the heat quantity, which causes a drop in temperature.

Increasing the number of rows of pipe in a heater with a constant

Fig. 21. "Vento" Cast Iron Heater

air quantity increases the final temperature of the air but diminishes the efficiency of the heater, because the average difference in temperature between air and steam is less. Increasing the steam pressure in the heater (and consequently its temperature) increases both the final temperature of the air and the efficiency of the heater. Table XIV has been prepared from different tests and may be used as a guide in computing probable results under ordinary working conditions. In this table it is assumed that the air enters the heater at a

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temperature of zero and passes between the pipes with a velocity of 800 feet per minute. Column 1 gives the number of rows of pipe in the heater and columns 2, 3, and 4 the final temperature of the air for different steam pressures. Columns 5, 6, and 7 give approximately the corresponding efficiency of the heater.

Example: Air passing through a heater 10 pipes deep and carrying 20 pounds pressure will be raised to a temperature of 90 degrees and

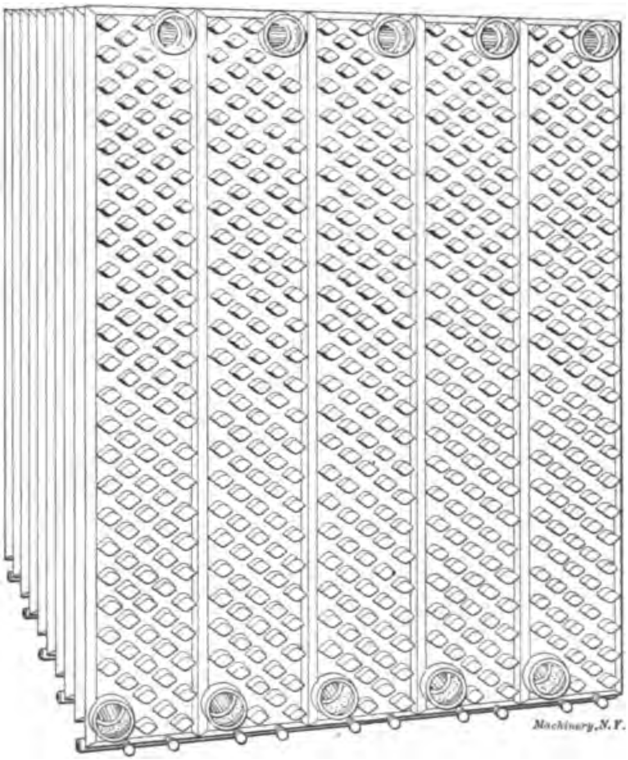

Fig. 22. "Vento" Cast Iron Heater

the heater will have an efficiency of 1,650 B. T. U. per square foot of surface per hour.

For a velocity of 1,000 feet, multiply the *temperatures* given in the table by 0.9 and the *efficiencies* by 1.1.

Example: How many square feet of radiation will be required to raise 600,000 cubic feet of air per hour from zero to 80 degrees, with a velocity through the heater of 800 feet per minute and a steam pressure of 5 pounds? What must be the total area of the heater front and how many rows of pipes must it have?

The B. T. U. required is found by multiplying the volume of air by the desired rise in temperature and dividing the result by 55; hence $600{,}000 \times 80 \div 55 = 872{,}727$ B. T. U. are required.

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Referring to Table XIV we find that for the above conditions a heater 10 pipes deep is required, and that an efficiency of 1,500 B. T. U. will be obtained. Then 872,727 ÷ 1,500 = 582 square feet of surface is required, which may be taken as 600 in round numbers. 600,000 ÷ 60 = 10,000 cubic feet of air per minute, and 10,000 ÷ 800 = 12.5 square feet of free area required through the heater. If we assume

TABLE XIV

Temperature of entering air, zero.
Velocity of air between the pipes, 800 feet per minute.

Rows of Pipe Deep	Temp. to which the Air will will be raised from zero. Steam Pressure in Heater.			Efficiency of the Heating Surface in B. T. U. per sq. ft. per hour. Steam Pressure in Heater.		
	5 lbs.	20 lbs.	60 lbs.	5 lbs.	20 lbs.	60 lbs.
4	30	35	45	1600	1800	2000
6	50	55	65	1600	1800	2000
8	65	70	85	1500	1650	1850
10	80	90	105	1500	1650	1850
12	95	105	125	1500	1650	1850
14	105	120	140	1400	1500	1700
16	120	130	150	1400	1500	1700
18	130	140	160	1300	1400	1600
20	140	150	170	1300	1400	1600

0.4 of the total heater front to be free for the passage of air, then 12.5 ÷ 0.4 = 31.25 square feet, total area required.

The general method of computing the size of heater for any given

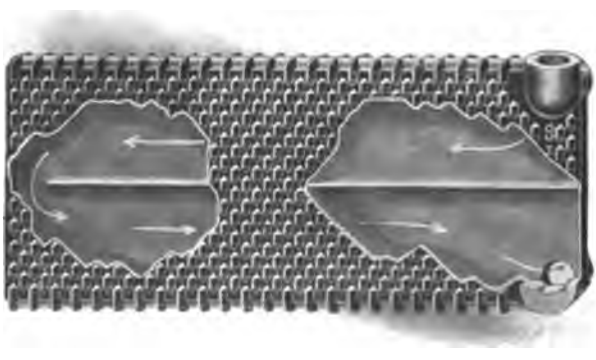

Fig. 23. Pin Type Heater

building which is to be both ventilated and warmed by a hot-blast system, is the same as in the case of indirect heating. First obtain the B. T. U. required for ventilation, and to that add the heat loss through walls, etc., and divide the result by the efficiency of the heater under the given conditions.

Example: An audience hall is to be provided with 400,000 cubic feet of air per hour. The heat loss through walls, etc., is 250,000 B. T.

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U. per hour in zero weather. What will be the size of heater, and how many rows of pipe deep must it be, with 20 pounds steam pressure?

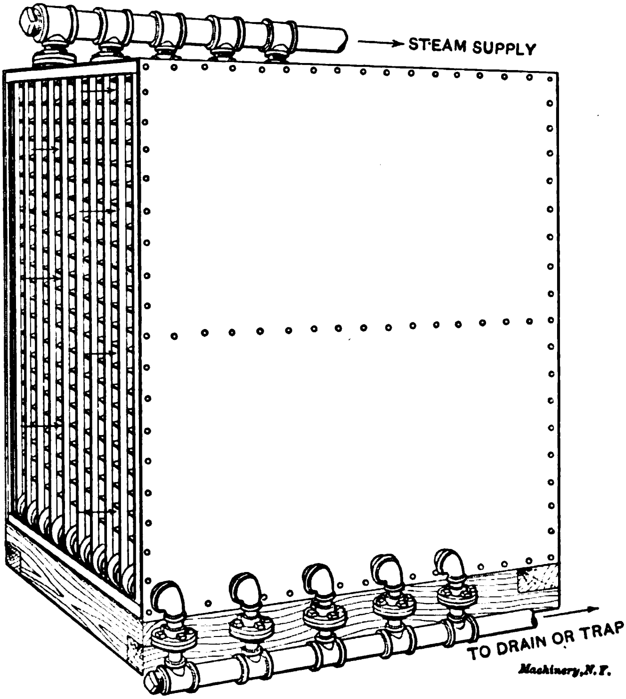

Fig. 24. "Vento" Cast Iron Heater

400,000 × 70 ÷ 55 = 509,090 B. T. U. for ventilation. Therefore 250,000 + 509,090 = 759,090 B. T. U., total to be supplied.

We must next find to what temperature the entering air must be

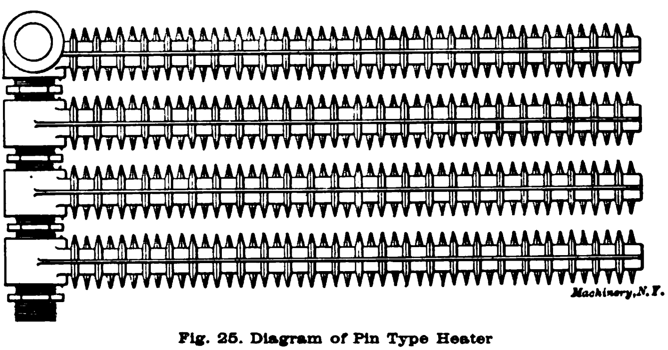

Fig. 25. Diagram of Pin Type Heater

raised in order to bring in the required amount of heat, so that the number of rows of pipe in the heater may be obtained and its corresponding efficiency determined. We have entering the room for

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purposes of ventilation, 400,000 cubic feet of air every hour at a temperature of 70 degrees, and the problem now becomes, to what temperature must this air be raised to carry in 250,000 B. T. U. additional for warming?

We know that 1 B. T. U. will raise 55 cubic feet of air 1 degree.

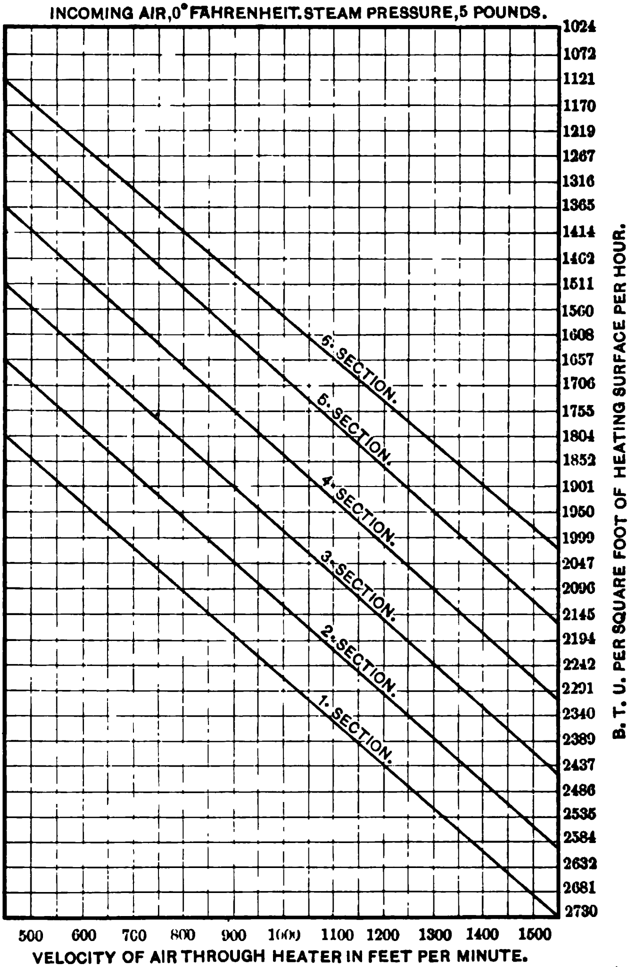

Fig. 26. Condensation Chart

Then 250,000 B. T. U. will raise 250,000 × 55 cubic feet or air 1 degree. Thus 250,000 × 55 ÷ 400,000 = 34 degrees, required excess temperature. The air in this case must then be raised to 70 + 34 = 104 degrees to provide for both ventilation and warming. Referring to Table XIV we find that a heater 12 pipes deep will be required, and

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that the corresponding efficiency of the heater will be 1,650 B. T. U. Then 759,090 ÷ 1,650 = 460 square feet of surface required.

Heating Surface Required for Factories

The proportional heating surface for factory heating is generally expressed in the number of cubic feet in the building for each linear foot of 1-inch steam pipe in the heater. On this basis, in factory practice, with all of the air taken from out of doors, there are generally allowed from 100 to 150 cubic feet of space per foot of pipe,

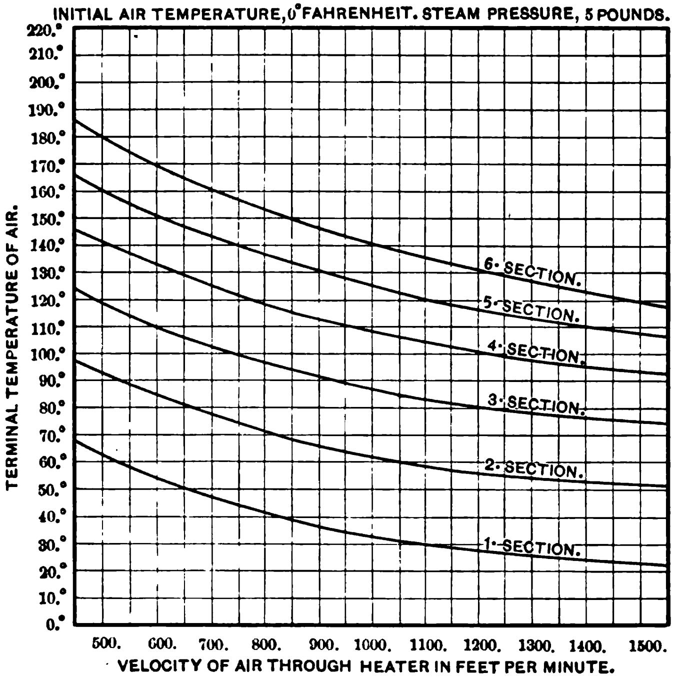

Fig. 27. Temperature Chart

according to whether exhaust or live steam is used, live steam here indicating steam of about 80 pounds pressure. If practically all of the air is returned from the buildings to the heater, these figures may be raised to about 140 as a minimum, and possibly 200 as a maximum, per foot of pipe.

Temperature and Condensation Charts

The accompanying "temperature" and "condensation" charts, Figs. 26 and 27, show the results obtained with the "Vento" cast iron heater, and the data given therein correspond to that found in Table XIV for pipe heaters. These charts explain themselves and require no further description.

Indirect Pin Radiators

Heaters made up of indirect pin radiators of the usual depth have an efficiency of at least 1,500 B. T. U. with steam at 5 pounds pressure,

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and are easily capable of warming air from zero to 80 degrees or over when computed on this basis. The free space between the sections bears such a relation to the heating surface that ample area is provided for the flow of air through the heater without producing an excessive velocity.

Pipe Connections

Hot blast heaters, commonly called main heaters, are usually divided into several sections, the number depending upon their size, and each provided with a separate valve in the supply and return. In

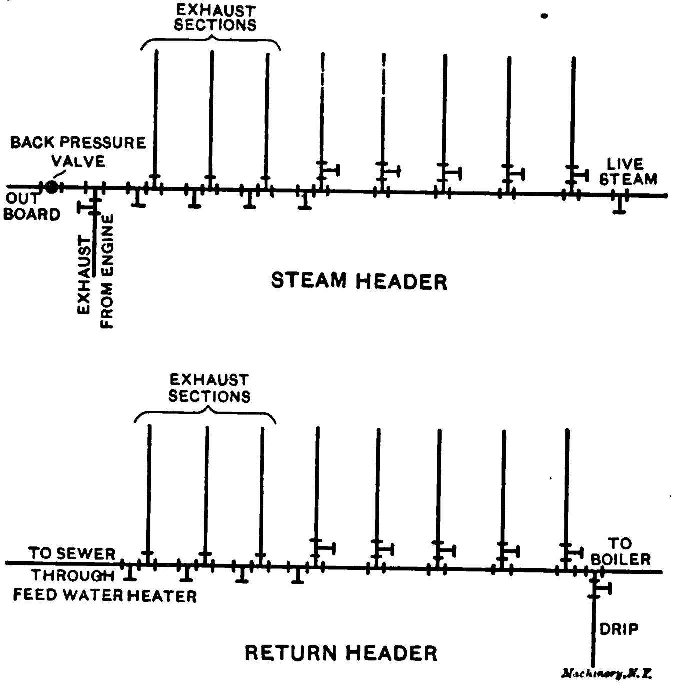

Fig. 28. Diagram of Heater Pipe Connections

making these divisions, special care should be taken to arrange for as many combinations as possible.

Example: A heater, 10 pipes deep, may be made up of three sections, one of 2 rows and two of 4 rows each. By means of this division, 2, 4, 6, 8 or 10 rows of pipe can be used at one time, as the outside weather conditions may require.

In making the pipe connections to a heater of this kind, a main or header is usually run along one side, from which branches of the proper size are carried to the different sections. The arrangement of the returns should correspond in a general way with the supplies. The main header should be properly drained, and the condensation from the heater tapped to a receiving tank, or returned to the boilers by gravity if the heater is overhead. If possible, the return from

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each section should be provided with a water-seal two or three feet in depth. This is because condensation is greater in the outer sections, resulting in a slight difference in pressure which causes the

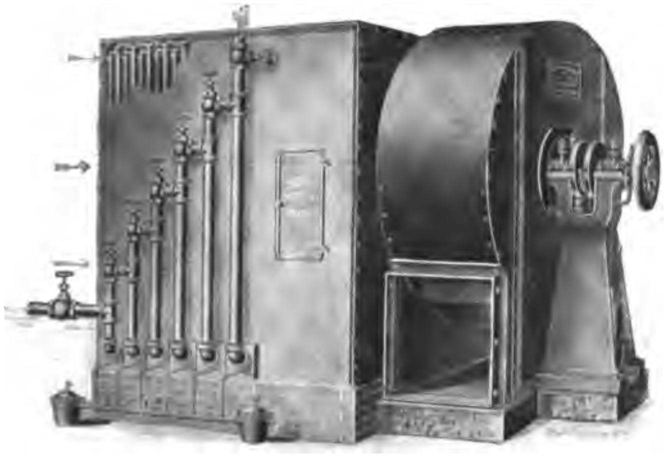

Fig. 29. Buffalo Forge Company's Heater

return water from the inner sections to be drawn into the outer ones, thus producing water-hammer and imperfect circulation of steam.

In the case of overhead heaters, the returns may be sealed by the water-line of the boiler or by the use of a special water-line trap, but vertical pipe heaters resting on foundations near the floor are usually provided with siphon loops, extending into a pit. If this arrangement is not convenient, a separate trap should be placed on the return from

TABLE XV

Square feet of Surface	Diameter of Steam Pipe	Diameter of Return
150	2"	1¼"
300	2½"	1½"
500	3"	2"
700	3½"	2"
1000	4"	2½"
2000	5"	2½"
3000	6"	3"

each section. The main return, in addition to its connection with the boilers or pump receiver, should have a connection with the sewer for blowing out when steam is first turned on. Sometimes each section is provided with a connection of this kind.

Large automatic air valves should be connected with each section, and it is well to supplement these with a hand pet-cock, unless individual blow-off valves are provided as described above. If the fan is driven by a steam engine, provision should be made for using the exhaust in the heater, and part of the sections should be so valved that they may be supplied with either exhaust or live steam as desired.

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Fig. 28 shows in diagram a method of making the connections for a heater in which three of the sections may be used in this way. Another way of accomplishing the same result is shown in Fig. 29, which shows a heater made by the Buffalo Forge Company. In this arrangement all of the sections are interchangeable.

The sizes of the mains and branches are often fixed by the tapping of the heater sections. Table XV, based on experience, has been found to give satisfactory results where the apparatus is near the boilers.

From 50 to 60 square feet of radiating surface should be provided in the exhaust portion of the heater for each engine horse-power, and should be divided into at least three sections, so that it can be proportioned to the requirements of different outside temperatures.

The condensation from the exhaust sections contains oil from the engine and should not be returned to the boilers; much of its heat, however, can be saved by passing it through a feed water heater. A simple heater for this purpose may be made of a piece of 8-inch pipe, 7 or 8 feet in length, with flanged heads, and containing a coil made up of four lengths of 1-inch brass pipe. The feed to the boilers is made to pass through the coil, while the space around it is filled with hot condensation. A similar heater is sometimes placed in the exhaust pipe from the engine, for use when exhausting outboard in mild weather. After passing through the feed water heater the condensation should be trapped to the sewer.

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CHAPTER III.

HEATING AND VENTILATING MACHINE SHOPS

Methods of Heating

The older method of heating a shop was by means of steam coils, either run along the walls under the windows, or supported overhead as most convenient. This arrangement necessitates a large amount of heating surface together with an extended system of supply and return piping, thus greatly increasing the liability to leaks and freezing. This method provides no fresh air for ventilation, and the distribution of heat is not of the best. When the coils are placed along the walls, under benches, it is uncomfortably warm for those working near them, and if supported overhead, the heat rises directly to the ceiling or roof, thus leaving the lower portion of the room too cold.

The most satisfactory arrangement is where the heating is done by hot air properly distributed through suitable ducts and flues. The heating surface in this case is very compact, only about one-fifth of that required for direct heating being necessary; and as the surface is grouped in a single heater, even in buildings of large size, long runs of piping are avoided. In the largest plants, or where the buildings are more or less detached, it becomes necessary to increase the number of units, but even then the pipe runs are simple compared with those necessary for direct heating. A better distribution of heat is obtained, resulting in a more uniform temperature throughout the rooms. As heating systems of this kind are usually arranged for taking a portion of their air supply from out of doors, it is possible to secure any degree of ventilation required.

General Arrangement

The location of the fan and heater and the general arrangement of the distributing ducts will depend largely upon the construction and plan of the building. One of the simplest arrangements for a building of small size is that shown in Fig. 31. In this case a single galvanized iron uptake is carried from the mouth of the fan directly upward through the different stories of the building. At each floor the requisite number of outlets are provided at or near the ceiling level, and the air discharged toward the outer walls. In the case of a larger building it would be necessary to extend the distributing ducts horizontally from the main uptake, as shown in Fig. 33.

Another typical arrangement is that shown in Fig. 30, which represents the plan and elevation of the heating and ventilating system installed in the shops of the Ashcroft Manufacturing Company, of Bridgeport, Conn. In this arrangement the fan and heater are cen-

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trally located in the basement near one of the side walls. Main distributing ducts are carried in both directions near the floor, and from these, vertical risers are taken off at frequent intervals and carried up to the different stories. The air is discharged into the rooms horizontally at an elevation of about eight feet from the floor. Regulating dampers are provided in each uptake for proportioning the air flow through each outlet. The fan is of the centrifugal type, with bottom horizontal discharge. The wheel is 6½ feet in diameter, and is driven by a 7-inch by 7-inch vertical direct-connected engine. The heater is made of 1-inch pipe, twenty-two rows deep. The sections are 6½ feet wide by 8 feet high. The building is warmed by air-rotation, no connection being made with the outside air.

Figs. 34 and 35 show plan and section elevation of a different arrangement, as installed in the shops of the Houston, Stanwood & Gamble Co., at Covington, Ky. In this case the fan and heater are placed in one corner of the building, and the air carried across one

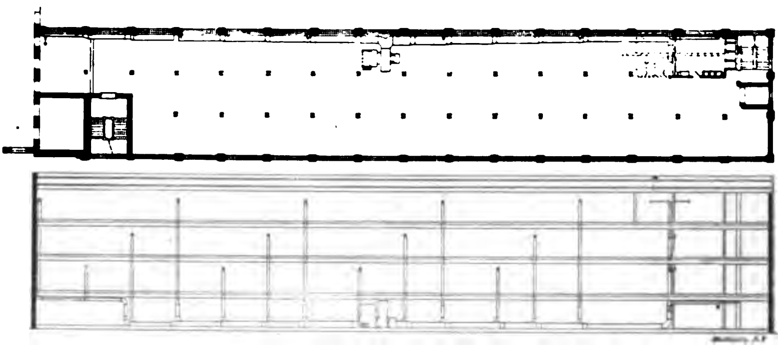

Fig. 30. Heating and Ventilating System of the Ashcroft Manufacturing Company, Bridgeport, Conn.

end in an underground concrete duct. Two uptakes connect with this, running up beside columns supporting the roof; horizontal branches are then carried the entire length of the building, one on each side, passing through the roof trusses as shown. The air is discharged through short mouthpieces set at an angle to give it a downward direction, and arranged to deliver it both to the central portion and the side aisles or bays. This particular arrangement of the distributing ducts is made necessary on account of the traveling cranes which pass through the entire length of the building in each of the three sections as indicated by tracks at the sides in Fig. 34. This is a typical illustration of an overhead distribution, the air being discharged at an elevation of about 18 feet above the floor. The apparatus in this case also takes its air supply from the building, leakage being depended upon for ventilation. The fan is of the three-quarter housed type, with a bottom horizontal discharge connecting with the underground duct. The wheel is 9 feet in diameter. The heater is made up of two groups, with a supply and return header

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at each side. It is 20 pipes deep, and has an exposed front 8 feet high by 12½ feet wide.

Fig. 36 illustrates a somewhat similar arrangement, although in this case two overhead units are used, each made up of two fans and a heater, and the air is carried downward to a point about 8 feet from the floor before being discharged. The system is installed in the machine and erecting shops of the Pennsylvania Railroad Company at Trenton, N. J. All parts of the system are symmetrically arranged.

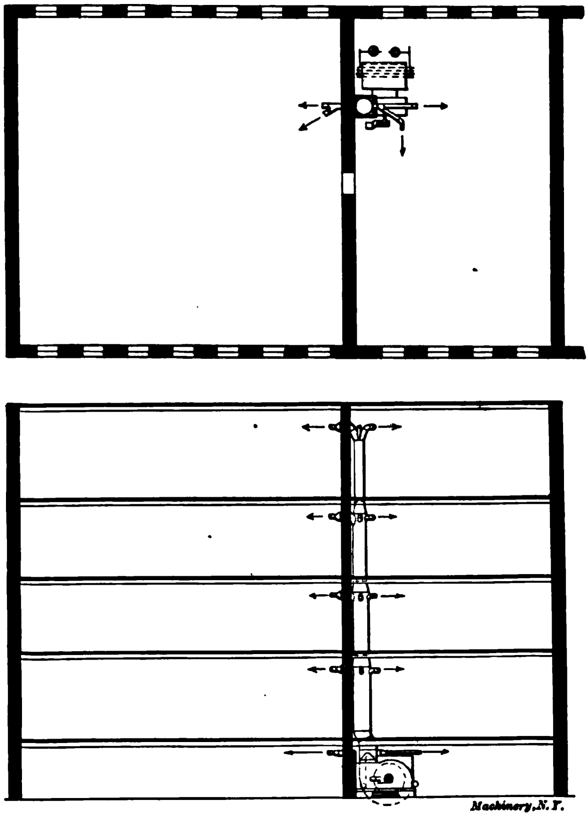

Fig. 31. Simple Arrangement of Heating Installation in a Small Building

which gives practically an equal resistance to the flow of air in each of the four main distributing ducts.

A plan and elevation of the fan and heater of one of the units is shown in Fig. 32. Two double inlet fans are used, with wheels 8½ feet in diameter attached to a common shaft and driven by a belted motor. The heater is made up with a double header, as in the previous layout. It is 20 pipes deep, 13½ feet wide and 10 feet high. The air is taken from the building, but is forced through the heater in-

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stead of being drawn through by suction, as in the other arrangements mentioned.

These five buildings illustrate the more common methods of arranging the distributing systems in the heating of machine shops of modern construction.

Material Used for Ducts and Flues

The airways are either constructed of brick or galvanized iron. In brick buildings where the heating system is planned before the build-

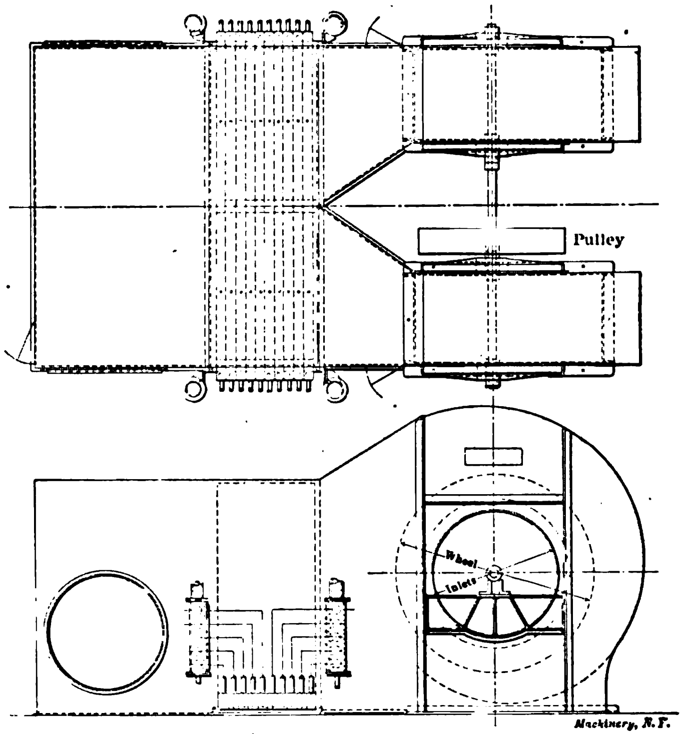

Fig. 32. Fan and Heater of Installation shown in Fig. 36

ing is constructed the flues may be most readily and economically built in the walls as the building is erected. When this is done, care should be taken to give them as smooth an interior as possible by removing all projecting motor from between the bricks.

Underground ducts are built either of brick or concrete for the larger sizes, and generally of glazed tile for the branches. In buildings of wooden construction, and also those of brick when erected before the heating system is laid out, it is customary to use galvanized iron. This is easily worked into the required form, is light in weight, and takes up a minimum of space for a given area.

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Construction of Ducts and Flues

Great care should be taken in the design and construction of a system of ducts and flues. When a change in the direction of flow is necessary, a gracious curve should be provided. For 90-degree turns, the elbow should be made with at least five pieces, and the radius of the inner side of the elbow should not be less than the diameter of the pipe. This relation between the radius of curvature and the size of pipe should hold in the case of rectangular ducts as well.

When a branch is taken off from a straight run of pipe, it should be given an angle of 45 degrees at the point of connection, and the remaining change in direction made by an easy turn. The main run of pipe is commonly reduced at each branch or take-off by an easy taper, about 28 inches in length, which can be made from a sheet of iron of standard width, which is 30 inches. Whenever the duct or pipe branches, the construction should be such as to divide the air volume into the required proportions, giving to each branch an easy

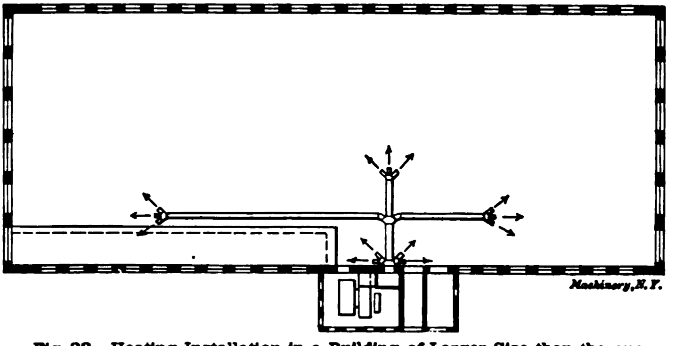

Fig. 33. Heating Installation in a Building of Larger Size than the one shown in Fig. 31

change in direction, when possible. While due regard should be given to the proper proportioning of the pipe areas, it is not possible to get a sufficiently accurate distribution of air without the use of dampers and deflectors. In the case of a large number of small outlets from a main duct, the best results are usually obtained by the use of adjustable dampers in each outlet. Where there are several branches of considerable size leading from the main, it is well to place adjustable deflectors at the junction of the ducts, so that the air volume can be deflected into the branches in such quantities as may be desired. In the case of brick or concrete underground ducts, the same points relating to curves, dampers, etc., should be observed as described above for galvanized iron.

Size of Ducts and Flues

The sectional area of ducts and flues is based upon the velocity of the air flow through them. It is a well-known fact that the frictional resistance to the flow of air through pipes increases as the

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square of the velocity, hence if the power required for driving the fan was an important factor, very low velocities would be required. As a matter of fact, this is generally neglected in practice, and velocities are based upon the most desirable speed of fan for this class of work, which is about 5,000 linear feet per minute tip velocity. This results in a velocity through the fan outlet of about 3,500 feet per minute. The size of the main duct is made such that the velocity commonly runs from 2,000 to 3,000 feet per minute, although some engineers make a practice of starting with a duct the same size as the fan outlet, but making a small increase in the size of the branches, so that their aggregate area shall be from 30 to 40 per cent greater than the area of fan outlet, thus bringing the velocity down to from 2,500 to 2,800 feet per minute.

It is frequently possible in shop practice to secure satisfactory circulation of the air with a limited extent of ducts by discharging it at a high velocity, as noted above, thus compelling it to continue its

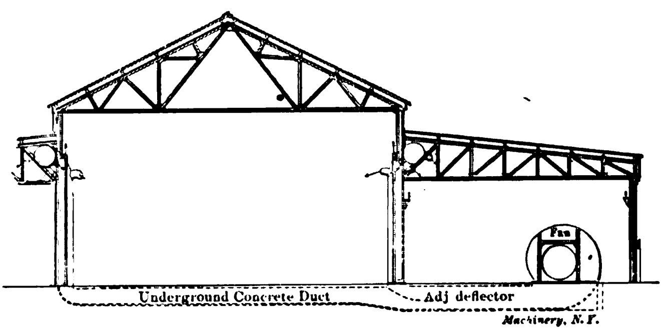

Fig. 34. End Elevation of Installation shown in Fig. 35

direction of movement for a considerable distance without the use of conducting pipes. It is not uncommon in such cases to force the air 100 feet or more from the outlets at a velocity of 2,000 to 3,000 feet per minute.

Weight of Iron Used

Table XVI gives the gage of iron commonly used for pipes of different diameter. All sizes above 60 inches are made of No. 16 gage. If the pipe is made much lighter, particularly in the larger sizes, it will not keep its shape when supported horizontally, which results in loosening the joints and also decreasing the area of the pipe. The common practice is to make rectangular pipes of the same gage as round pipes having the same sectional area, but under certain conditions, as in the case of a thin, flat pipe, bracing is necessary to prevent sagging, even with heavy gages. When braces are used, lighter iron may be used than given in the table.

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Heaters

The subject of heaters for shop heating was quite thoroughly discussed in the previous chapter. A few of the results noted there will

TABLE XVI

Diameter of Pipe.	Gage of Iron
Less than 9 inch	28
9 inch to 14 inch	26
15 inch to 20 inch	25
21 inch to 26 inch	24
27 inch to 35 inch	22
36 inch to 46 inch	20
47 inch to 60 inch	18
61 inch and above	16

be given here together with some special reference to heating by air rotation. In shop practice, the amount of heating surface is generally expressed in linear feet of one-inch pipe for a given space to be heated. This, for average conditions, may be taken as follows. With all of the air taken from out of doors, there is generally allowed 100 cubic feet of space for one foot of pipe when exhaust or low-pressure steam is used, and 150 cubic feet with steam at 80 pounds pressure. When the building is heated by air rotation, the aboxe figures may be raised

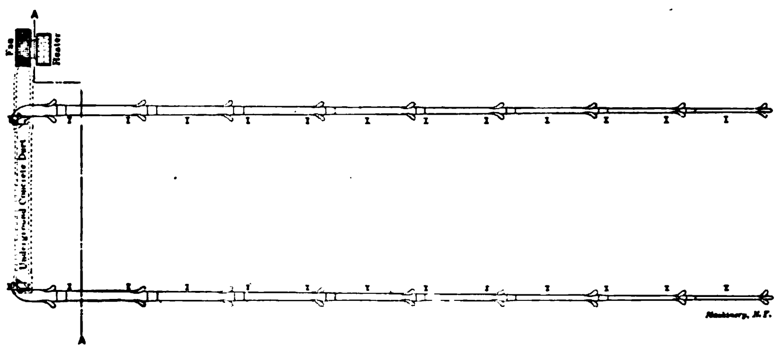

Fig. 35. Plan of Heating and Ventilating System of the Houston, Stanwood & Gamble Company, Covington, Kentucky

to about 140 and 200 for low-pressure and high-pressure steam, respectively. The heater is generally made about twenty pipes deep under ordinary conditions. Heaters of this type have an efficiency of about 1,300 heat units per square foot of surface for steam at 5 pounds pressure, and an efficiency of 1,600 for 60 pounds pressure.

Volume of Air Required

When the air is taken from out of doors for the purpose of ventilation, it may be based upon the number of occupants or upon a given number of air changes per hour. Usually the cubic contents is large per occupant and may vary considerably in different shops, so that under ordinary conditions it is best to use the former method. The air supply per occupant may be taken as about 25 or 30 cubic feet per minute, unless the building is very openly constructed, in which

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case the air volume may be reduced and leakage depended upon to a considerable extent. In many shops the heating is done entirely by air rotation, and leakage is depended upon entirely for ventilation. This is made possible because of the large enclosed space in proportion to the number of occupants and the thorough mixture of the inleaking air with that which is in rotation. When this method of heating is used, the air simply becomes the medium for transferring the heat to the different parts of the building, and the volume required will depend upon the amount of heat to be transferred and the temperature to which the air is raised.

Suppose the air is returned to the heater at a temperature of 60 degrees and delivered at a temperature of 140 degrees, the total rise being 80 degrees. In cooling one degree, one cubic foot of air gives out 1/55 of a heat unit, or in cooling 80 degrees will give out $1/55 \times 80 = 80/55$, or 1.4 heat unit. Therefore, if we divide the total amount of heat to be supplied in a given time, expressed in heat units, by 1.4, it will give the volume of air to be rotated in that time, assuming, of course, that it is cooled through 80 degrees during its passage through the room.

The heat loss from the building may be computed by any of the common methods in use, or the size of the heater may first be computed by the method already given, and the heat given off taken as the equivalent of the heat lost. Referring to Table XIV we find that a heater twenty pipes deeps with steam at 5 pounds pressure will raise the temperature of air from 0 to 140 degrees, and has an efficiency of 1,300 heat units. Steam at 5 pounds pressure has a temperature of 227 degrees. The average temperature of the air passing through the heater is $\frac{0 + 140}{2} = 70$ degrees, hence the difference in temperature between the steam and air is $227 - 70 = 157$ degrees. In case the air is rotated, its average temperature is $\frac{60 + 140}{2} = 100$ degrees, and the difference between the steam and air is $227 - 100 = 127$ degrees. The efficiency of a heater varies directly as the difference between the temperature of the steam and air; hence, in the second case, with the air rotated, the efficiency would be $157 : 127 = 1{,}300 : x$, and x, the efficiency in this case, would be approximately 1,100 heat units. Then the square feet of surface in the heater multiplied by 1,100 will be the heat given off per hour, and this divided by 1.4 will give the cubic feet of air to be moved per hour by the fan.

Size of Fan

The required size of fan for moving any given volume of air may be taken from Table XVII, which also gives the approximate speed and the horse-power required for driving the fan.

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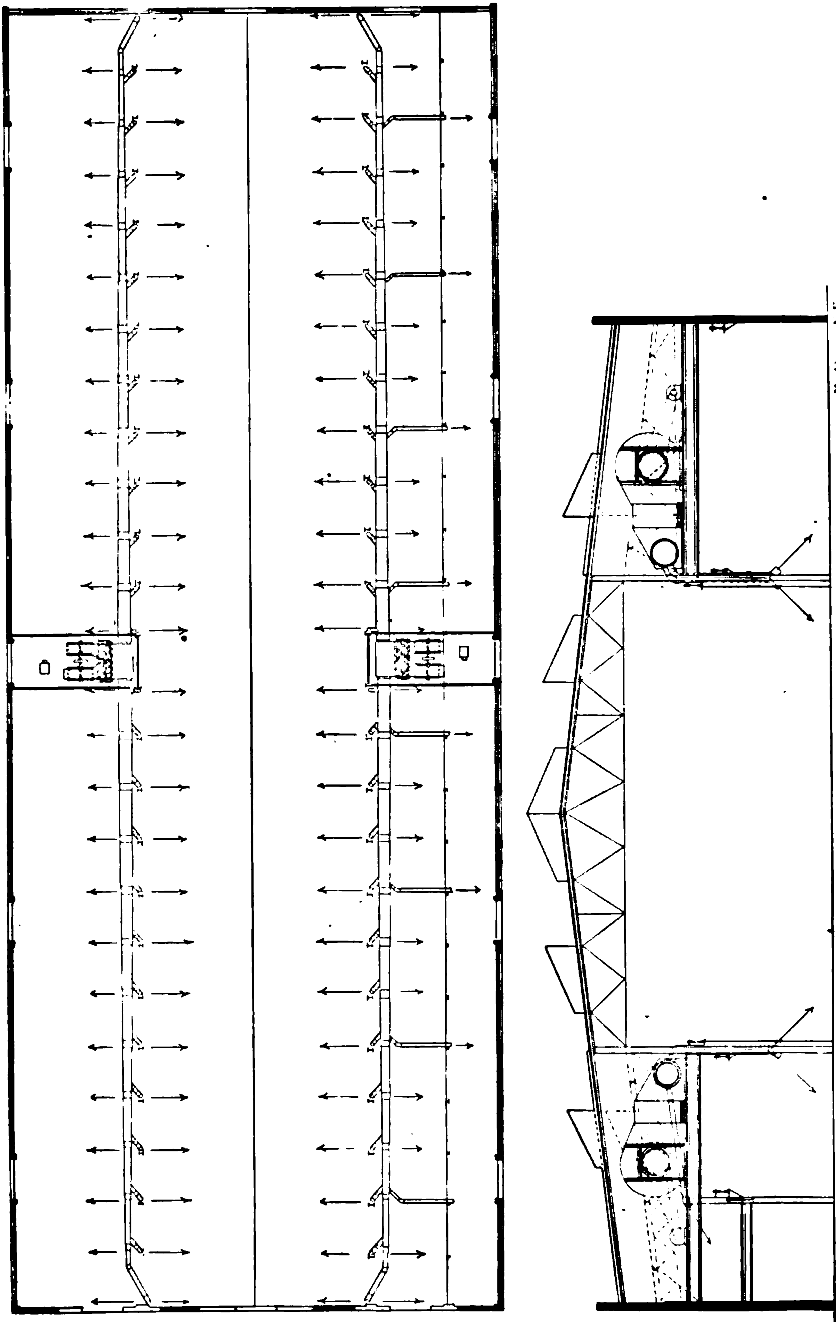

Fig. 36. Heating System of the Pennsylvania Railroad Company's Machine Shop at Trenton, N. J.

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TABLE XVII

Normal Size of Fan, Height of Housing in Inches	Diameter of Fan Wheel in Inches	Width of Housing in Inches	Ordinary Speed Giving ½ Ounce Pressure	Cubic Feet of Air Delivered per Minute	Horse-power of Engine to Drive the Fan
30	18	9	870	1000	½
40	24	12	580	1600	1
50	30	15	465	2600	1
60	36	18	390	4500	2
70	42	21	333	6000	2½
80	48	24	293	8000	2½
90	54	28	260	11000	4
100	60	32	233	12500	4
120	72	43	195	21500	7
140	84	48	167	28600	9
160	96	48	147	31800	10
	108	54	130	40400	13
	120	60	117	51000	16

The speeds given in the table are for ½ ounce pressure; should it be desired to deliver the air under a higher pressure, in order to force it a long distance from the outlets, it would be necessary to increase the speed of the fan somewhat, depending upon local conditions.

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NUMBER 40

FLY-WHEELS

CONTENTS

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CHAPTER I

FLY-WHEELS, THEIR PURPOSE, CALCULATION AND DESIGN

The object of all fly-wheels is to equalize the energy exerted and the work done, and thereby prevent great or sudden changes of speed. The extent to which speed changes may take place is the determining factor in all fly-wheel design. The application and use of fly-wheels will, however, be more readily perceived if we examine some concrete practical examples.

In a shear press or punch the energy of the driving belt remains practically constant, but the work done by the jaws varies from practically nothing at the return stroke to the full power of the machine when cutting. In an engine, on the other hand, the work done by the belt wheel may be constant, or nearly so, while the energy exerted by the steam on the piston varies throughout each stroke. Furthermore, in any engine using a connecting-rod and crank, the energy exerted on the crank pin, tending to turn the shaft, varies from nothing at the end of the stroke to its greatest value near mid-stroke. A steam engine without a fly-wheel would, therefore, be useless and could hardly make one single revolution.

Now in what way can a fly-wheel help to overcome this difficulty? If such a wheel of large diameter and with a heavy rim were mounted on ball bearings to reduce the friction, it would even then be found difficult to set it in motion or increase its speed suddenly, on account of its inertia. To give to it any particular increase of speed would require a certain amount of work or energy put into it. Having once acquired speed it would be capable of doing the same amount of work on any opposing body by virtue of this stored energy.

So the fly-wheel of the engine or press, as long as the energy exerted is greater than the work being done, will turn faster, the inertia of its rim absorbing excess of energy. On the other hand, if the energy becomes less than the work, the wheel turns slower and slower, giving up its stored energy to supply the deficiency. The heavier the rim, and the greater its velocity, the less the change of speed for a given storage of energy.

In the steam engine these changes of speed occur twice in every revolution, the wheel moving most slowly near one-quarter stroke and most rapidly near three-quarters stroke, the exact times being dependent on the point of cut-off and the connecting-rod ratio. The use of the fly-wheel is often confused with that of the governor. The fly-wheel can only average the speed during one revolution and prevent violent changes in that time. It has no control over the number of revolutions per minute. On the other hand, no ordinary governor

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works quickly enough to regulate the speed in one revolution. The governor prevents any permanent change in speed by adapting the amount of steam admitted to the amount of work to be done. A governor will prevent an engine from running away; a fly-wheel cannot.

Elementary Calculations of Fly-wheels for Steam Engines

In order to determine the weight of rim of a fly-wheel, it is necessary to know the probable excess and deficiency of energy in each stroke and the per cent of variation in speed that can be tolerated. The earlier the cut-off of the engine the greater the variation in energy and the larger the fly-wheel that will be required. The weight of the reciprocating parts and the length of the connecting-rod also affect the variation. The following table from Rankine's "Steam Engine" shows about what may be expected.

TABLE I. CONDENSING ENGINES

Fraction of stroke at which steam is cut off	1/3	1/4	1/5	1/6	1/7	1/8
Factor of energy excess	0.163	0.173	0.178	0.184	0.189	0.191

TABLE II. NON-CONDENSING ENGINES

Steam cut off at	1/2	1/3	1/4	1/5
Factor of energy excess	0.160	0.186	0.209	0.232

To obtain the excess of energy from this table it is only necessary to find the average work in foot-pounds done by the engine in one revolution and multiply this by the decimal given in the table. We will call this excess of energy E. The allowable variation in speed depends upon the use to which the engine may be put. In modern engines an allowance of from 1 to 2 per cent is usual.

The following formula is used for computing the weight of the fly-wheel rim:

Let W = weight of rim in pounds,
D = mean diameter of rim in feet,
N = number of revolutions per minute,
$\frac{1}{n}$ = allowable variation in speed (from 1/50 to 1/100),
E = excess and deficiency of energy in foot-pounds,
c = factor from Tables I and II,
$H.P.$ = indicated horse-power.

Then, if the indicated horse-power is given:

$$W = \frac{387{,}587{,}500 \times cn \times H.P.}{D^2 N^3} \qquad (1)$$

If the work in foot-pounds is given, then:

$$W = \frac{11{,}745\, n E}{D^2 N^2} \qquad (2)$$

E is calculated as before explained. From these formulas it will be seen that increasing the diameter or the speed of a wheel diminishes

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the necessary weight of rim very rapidly. To make clear the use of these formulas, we will work out two examples, such as might arise in practice.

Example 1.—A non-condensing engine of 150 indicated horse-power makes 200 revolutions per minute, with a variation of 2 per cent. The average cut-off is at one-quarter stroke, and the fly-wheel is to have a mean diameter of 6 feet. Required the necessary weight of rim in pounds.

From Table II we find $c = 0.209$, and from the data given we evidently have:

$$H.P. = 150;\quad N = 200;\quad \frac{1}{n} = 1/50 \text{ or } n = 50;\quad D = 6.$$

Substituting these values in equation (1) we have:

$$W = \frac{387{,}587{,}500 \times 0.209 \times 50 \times 150}{36 \times 200 \times 200 \times 200}$$

or $W = 2{,}110$ pounds, nearly.

Example 2.—A condensing engine, 24 x 42 inches, cuts off at one-third stroke and has a mean effective pressure of 50 pounds per square inch. The fly-wheel is to be 18 feet in mean diameter and make 75 revolutions per minute with a variation of 1 per cent. Required, weight of rim.

The work done on the piston in one revolution is equal to the pressure on the piston multiplied by the distance traveled or twice the stroke in feet. The area of the piston in this case is 452.4 square inches, and twice the strcke is 7 feet. The work done on the piston in one revolution is, therefore: $452.4 \times 50 \times 7 = 158{,}340$ foot-pounds. From Table I, $c = 0.163$, and therefore:

$E = 158{,}340 \times 0.163 = 25{,}810$ foot-pounds.

From the data given we have: $n = 100$; $D = 18$; $N = 75$. Substituting these values in equation (2):

$$W = \frac{11{,}745 \times 100 \times 25{,}810}{18 \times 18 \times 75 \times 75} = 16{,}650 \text{ pounds, nearly.}$$

Dimensions of Rim

In the above formulas, D, the mean diameter of the rim, is really twice the so-called radius of gyration, and would be found by squaring the outer and inner diameters, adding them together, dividing by two and extracting the square root. In symbols this would read:

$$D = \sqrt{\frac{D_1^2 + D_2^2}{2}}$$

It is usually accurate enough to take $D = \dfrac{D_1 + D_2}{2}$, or the arithmetical mean.

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To illustrate, we will assume $D_1 = 8$, $D_2 = 9$, then:

$$D = \sqrt{\frac{64 + 81}{2}} = 8.514$$

But the mean of 8 and 9 is 8.5, which is accurate enough in practice.

The number of cubic feet in the rim may be found by dividing the weight in pounds by 450 for cast iron or 480 for steel.

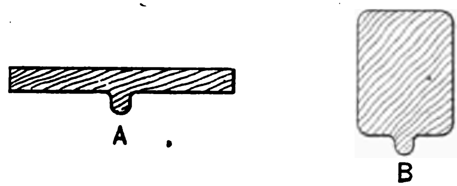

Fig. 1. Types of Fly-wheel Rim Sections

For instance, the rim in the first example given above would contain:

$$\frac{2,110}{450} = 4.69 \text{ cubic feet of cast iron,}$$

and the one in the second example would contain:

$$\frac{16,650}{450} = 37 \text{ cubic feet.}$$

The area of the cross section of the rim may be found approximately by dividing the cubic contents by the circumference corresponding

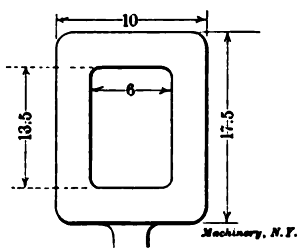

Fig. 2. Hollow Fly-wheel Rim

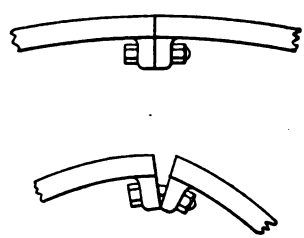

Fig. 3. Manner in which Bolts in Fly-wheel Joints are Stressed

to the mean diameter D. This area, in the first example, would be:

$$\frac{4.69}{6 \times 3.1416} = 0.2487 \text{ square feet} = 35.83 \text{ square inches,}$$

and in the second example:

$$\frac{37}{18 \times 3.1416} = 0.654 \text{ square feet} = 94.2 \text{ square inches.}$$

The shape of the cross-section is determined by the use to be made of the wheel. If it is a belt wheel the width of rim is determined by the width of belt, and the section is usually something like *A*, Fig. 1.

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If the wheel is to be simply a fly-wheel, it is better to adopt a stronger form of section and one easier to fasten at the joints. A common form in such cases is like *B*. In very large wheels it is better to make the rim hollow, as it is easier to cast and easier to put together. To illustrate the above principles, we will assume that the wheel in the first example is to carry a double leather belt sufficiently wide to transmit the desired horse-power. We will say that under the given conditions a belt 18 inches wide would be sufficient.

We may then assume that the rim should be a little wider than this, or say 19 inches.

The thickness will then be:

$$t = \frac{35.83}{19} = 1.88 \text{ inch.}$$

On the other hand, if we assume the wheel in the second example to be used solely as a fly-wheel, we can take any proportions which

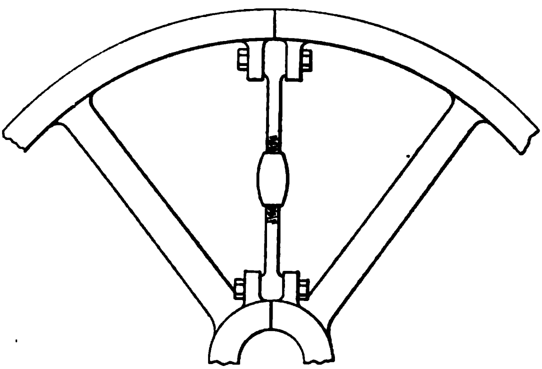

Fig. 4. Fly-wheel Joint Reinforced by Tension Rod

may be convenient. If a width of eight inches is chosen, the depth will be:

$$\frac{94.2}{8} = 11\tfrac{3}{4} \text{ inches, nearly,}$$

and the wheel will be about 19 feet in outside diameter.

If it is decided to make the rim hollow and a thickness of two inches is adopted, the proportions would be about as in Fig. 2.

Joints in Rims

Wheels less than 8 feet in diameter are usually cast solid. Wheels from 8 to 16 feet in diameter may be cast in halves to facilitate transportation. Wheels larger than 16 feet are usually cast in several pieces, the hub being a separate piece. Each arm may have a segment of the rim cast with it, but in the larger sizes of wheels the segments of the rim are bolted to the arms as well as to each other. The bolts must be kept as snug to the rim and as far from the lower edge of the flange as possible.

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Until quite recently it has been customary to join the segments of the rims of belt pulleys by internal flanges and bolts, the joint coming midway between the arms. Theory and experiment both show this to be a very unsafe arrangement, the strength of the joint being only from one-fifth to one-third that of the solid rim. When the wheel is running at a high speed, the pressure of the centrifugal force bends the joint out and opens it as shown in Fig. 3, thus throwing a great additional stress upon both bolts and flanges. If a joint of this type is adopted, it should be strengthened by wrought iron or steel tension rods, running from the flanges to the hub, and preventing the rim from bending at the weak point. (See Fig. 4.) Mr. James Stanwood has suggested placing such a joint at a point one-quarter way from the arm, where the bending is practically nothing. Most engine builders of late have put the joints in the rim directly over the arm. In any case, the bolts should be located as close to the rim as practicable.

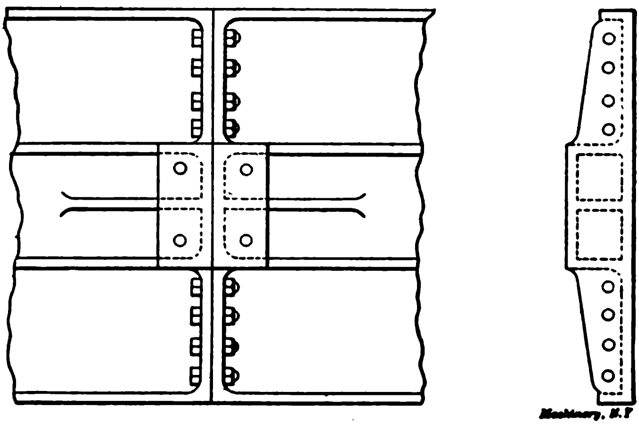

Fig. 5. Joint in Fly-wheel with Wide Face

Fig. 5 shows a joint in a belt wheel rim as made by a well-known firm of engine builders. This is a wheel 24 feet in diameter, with a 64-inch face intended for two belts. The flanges are well braced by the internal ribs, and the bolts which hold the arms also assist in strengthening the rim joint. If a rim of this kind is very wide, the centrifugal force tends to stretch the outer edges of the rim and to crack the cross flanges near the arms. In such cases it were better to use two sets of arms or two separate wheels.

A much safer and better rim, where it can be used, is the narrow and deep form. The tendency of bending between the arms, due to centrifugal force, is then resisted by the great depth of metal. The links or bolts which form the joint can be placed nearer the center of the rim, where the bending will not affect them. Two common forms of such joints are shown in Figs. 6 and 7. The links or prisoners, as they are sometimes called, are let in on the two opposite sides of the rim and occasionally on the inside face as well. They are

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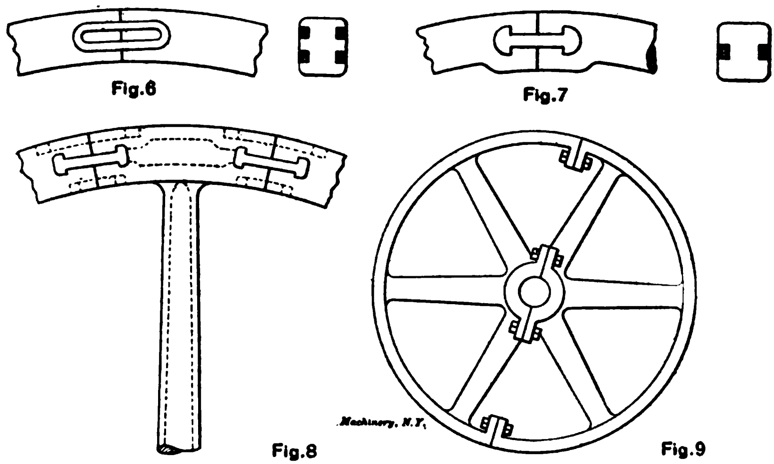
usually put in hot and allowed to shrink into place, drawing the joint firmly together.

Experiments made at the Case School of Applied Science, Cleveland, Ohio, have shown that a joint of this kind may have a strength two-thirds that of the solid rim. If the section of the rim is slightly increased where the link is inserted, the strength will be greater. (See Fig. 7.)

In a paper read before the American Society of Mechanical Engineers, Mr. John Fritz has called particular attention to the advantages of the hollow rim and arms in fly-wheel construction. Fig. 8 is a sketch of a joint in such a wheel. The coring of the rim insures a sounder casting and also makes it possible to get a stronger joint, by thickening the metal at that point. The links are made of different lengths so that the heads will not all come at the same point.



Figs. 6 to 9. Types of Fly-wheel Rim Joints

It is not possible to give any definite rule for determining the width and thickness of fly-wheel arms, there being so many disturbing factors. The thickness of the metal should be as uniform as possible, and where the arm joins the rim, if cast on, the change of thickness should be gradual, to avoid cooling strains. Increasing the number of arms will strengthen the rim by diminishing the bending between the arms. Under ordinary circumstances the stresses on the arms of a fly-wheel are slight, but when started or stopped suddenly, as in rolling-mill work, it is difficult to calculate how great they may be.

Experiments made on ordinary belt pulleys have shown that the bending due to belt tension is not evenly distributed among the arms, but is almost twice as great on the arm which happens to be nearest to the tight side of the belt at any instant. This difference is probably less in fly-wheels, on account of their stiffer rims, but even here it would be preferable to design the arms for about twice the average moment.

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Examples

Example 3.—It is required to design an internal flange joint for a fly-wheel rim 3 inches thick and 18 inches wide, the wheel being 10 feet in diameter and built in two halves.

A simple design in such a case is to part the wheel on a diameter which shall pass near two of the arms, as in Fig. 9. The distance from the joint to the center of the arm should not be more than one-quarter the space between the arms. The flanges should be of approximately the same thickness as the rim. We will use steel bolts having a tensile strength of 75,000 pounds per square inch. If the metal in the rim has a tensile strength of 18,000 pounds per square inch, the

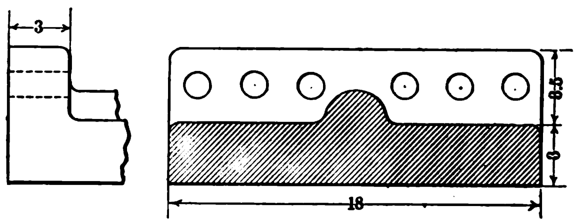

Fig.10

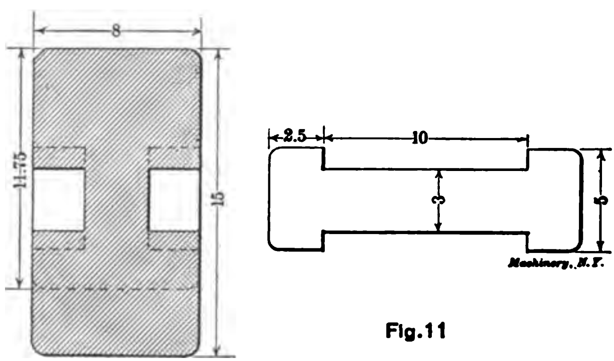

Fig.11

Figs. 10 and 11. Fly-wheel Rim Joints

total tensile strength of the rim is $3 \times 18 \times 18{,}000 = 972{,}000$ pounds. It will be found difficult to make the strength of joint more than one-third this, or 324,000 pounds.

$$\frac{324{,}000}{75{,}000} = 4.32 \text{ square inches of bolt area required.}$$

Six 1¼-inch bolts will have a combined area at the root of thread of $6 \times 0.89 = 5.34$ square inches.

With bolts even as large as this, the flange will probably break before the bolts. The joints would have the appearance shown in Fig. 10. The bolts should be kept as snug to the rim and as far from the lower edge or flange as possible.

Example 4.—Let it be required to design a link joint for the rim of

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the wheel in Example 2, supposing the rim to be solid, 8 inches wide and 11.75 inches deep.

Assuming the tensile strength of the cast iron as 18,000 pounds per square inch, the total strength of rim is:

$$8 \times 11.75 \times 18{,}000 = 1{,}692{,}000 \text{ pounds.}$$

If the tensile strength of the steel used for links is taken as 75,000 pounds per square inch, and we try to make the joint two-thirds as strong as the rim, we shall need:

$$\frac{2}{3} \times \frac{1{,}692{,}000}{75{,}000} = 15 \text{ square inches.}$$

If we adopt the form of joint shown in Fig. 7 and use two links,

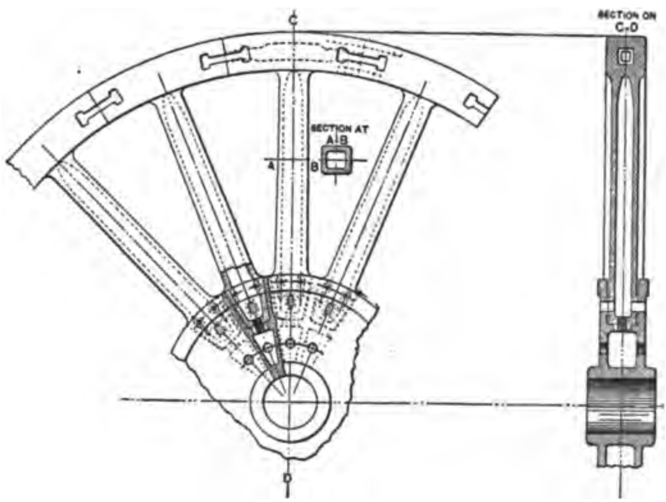

Fig. 12. Fly-wheel for Rolling Mill Service

the area of cross-section of each link will be 7.5 square inches, or 2.5 × 3 inches. Fig. 11 shows the proportions of such a joint. As the heads of the links will remove 2 × 2.5 × 5, or 25 square inches of metal from the cross-section of the rim, it will be necessary to add this amount by increasing the depth at the joint to about 15 inches.

Types of Fly-wheels

The ordinary fly-wheel used in this country is built of cast iron; many serious accidents from the bursting of these wheels have occurred because of bad design, hidden flaws, or because the wheels were run at higher than their designed speed.

The fly-wheel recommended by Mr. John Fritz, and already referred to, is shown in detail in Fig. 12. This wheel is the outgrowth of the demands of rolling-mill practice, where the duty of fly-wheels is exceptionally severe.

Referring to the illustration, it will be seen that the segment is cast hollow, and also the arms, which are made at the ends to compare

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in thickness to the segment, so as to relieve them of strains which might occur if the segments were cast solid. The holes in the segments are small at the ends, so as to make up for the metal taken out for the tees. The links, or tees, are of different lengths, so that the strain on the segments will not come all at one place, and by using oil-tempered steel in the links, or double tees, the rim will be practically as strong at the joints as it is elsewhere. There are no abrupt changes in the thickness of the castings, thus avoiding as much as possible the liability of strains.

It will be noticed that there is a space in the center of about one-quarter inch in the front and rear side of each arm. This is filled with oakum and driven hard, after the wheel is finished and in place, to keep the arm from yielding in the direction of the strain, and at the same time it greatly lessens the work of fitting up the wheel. The one and three-quarter inch round holes through the center and arm are reamed out and steel pins made and turned so that they will drive in snugly.

Sixty-ton Fly-wheel

A 19-foot fly-wheel of special and interesting construction, which was built in 1905 by the Nordberg Engineering Co., Milwaukee, Wis., for a mine pumping engine operated by the Calumet & Hecla Mining Co., is shown in detail in Fig. 13. The wheel weighs 120,000 pounds and is designed to run at 107 revolutions per minute which means a peripheral speed of nearly 6,400 feet per minute. A reasonable factor of safety for this speed requires a construction considerably stronger than possible with the usual plain form of cast iron fly-wheel. The nominal safe speed limit for cast iron wheels is put at about 5,000 feet per minute but the jump to 6,400 feet means that the bursting stress is increased in the ratio of 2.5 to 4.1. It might be argued that reversing rolling-mill fly-wheels which are subjected to tremendous shocks by reason of constant reversals are made of plain cast iron construction and stand up to the work with very few failures, but this argument would be made without taking into consideration the great increase of centrifugal force incident to increasing the speed even as little as 10 per cent. Between the reversing rolling-mill fly-wheel running at say 4,000 feet per minute and this wheel running at 6,400 feet per minute, the centrifugal stress, which increases as the square of the velocity, as is evident in the formula for centrifugal force, $F = 0.000341\,W R N^2$, changes the factor of disruptive stress in the ratio of 1.6 to 4.1. The stress on the reversal does not directly affect the integrity of the rim, but does throw a heavy bending stress on the spokes, hence the part of a reversing fly-wheel which is most affected at reversing is the spokes and not the rim. Therefore, the comparison between the plain reversing mill fly-wheel and the reinforced wheel forming the subject of this description should be made on the basis of speed alone and not with reference to the effect of reversal.

The Calumet & Hecla wheel is made up of two cast iron segments forming the wheel center. These segments are held together by two

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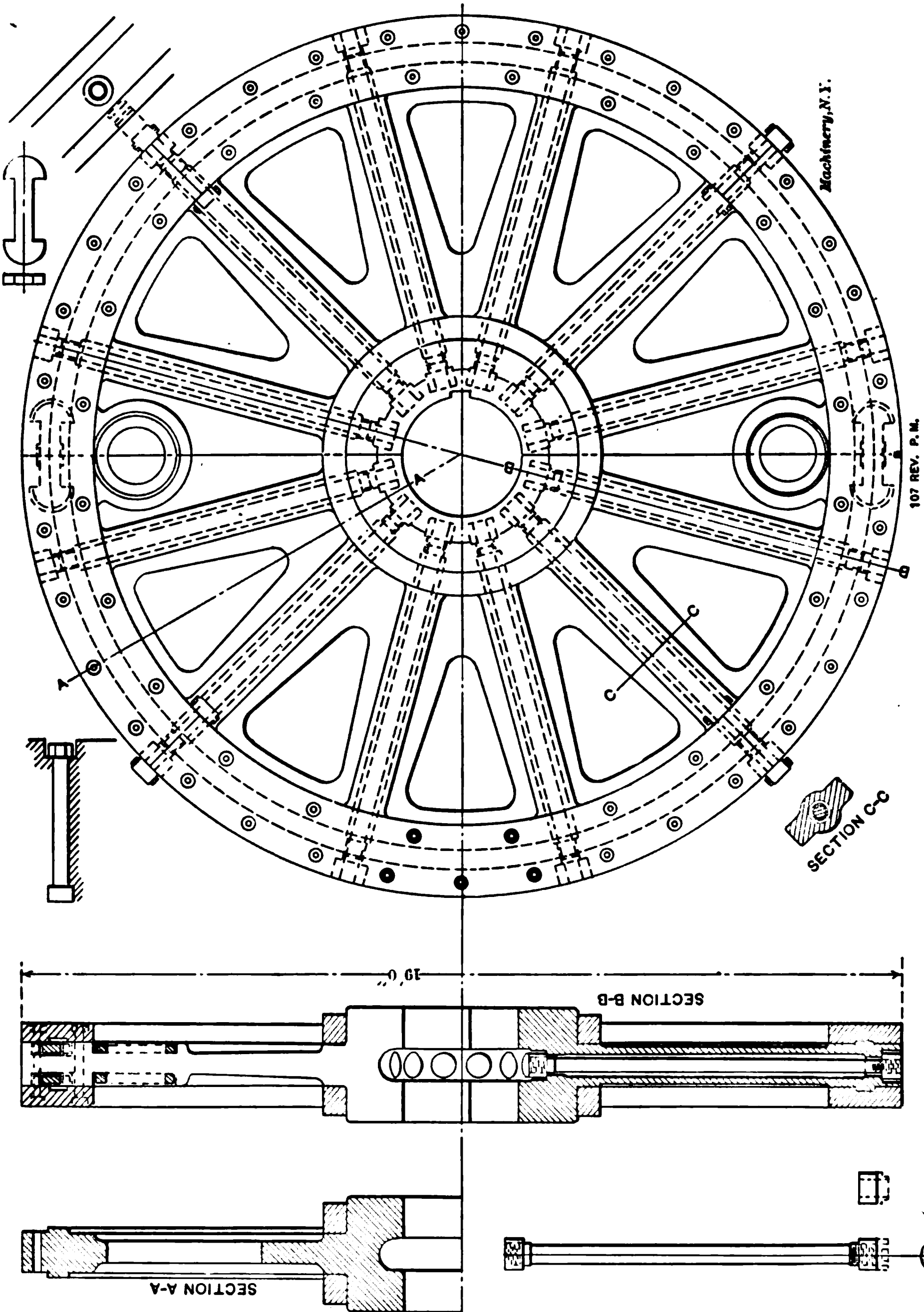

Fig. 13. A Sixty-ton Fly-wheel

steel shrink rings on the hub, four shrink rings under the rim and two steel rim rings made in halves and secured to the sides of the cast iron rim of the wheel center by 58 bolts 2 inches diameter and 18 inches long. In addition the halves of the wheel center are bound together by four T-head steel links set in the pockets underneath the rim rings and shrunk on in the usual manner. The spokes are cast hollow and 12 open-hearth steel bolts 5 inches diameter are set radially therein, being secured at the ends by round nuts fitting in countersunk seats. These bolts are warmed up before being put in place and the nuts are screwed up tightly before cooling, thus getting a heavy compression effect on the spokes due to the contraction of the bolts, the intention of the designer being to relieve the cast iron parts of all tensional strain due to centrifugal force. The spokes of the cast iron

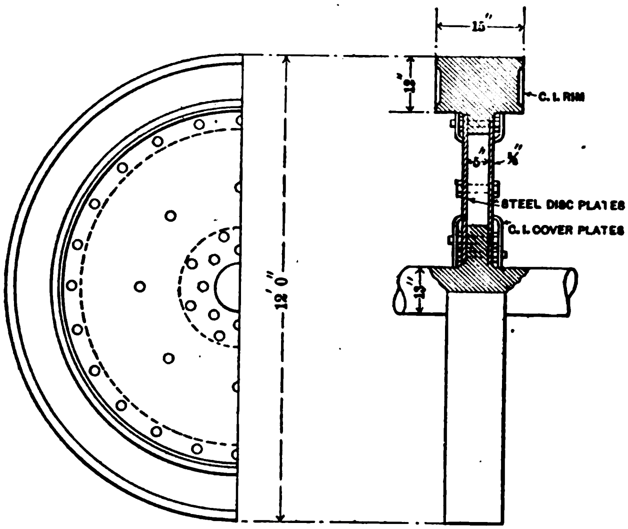

Fig. 14. An Unusual Fly-wheel Design

wheel center have an open space between each pair save at the junction of the halves where the web is made solid, having a thickness of 7 inches next the hub. A boss is cast on each side under the rim in which an annular seat is machined with a boring bar for four steel shrink rings which bind the two halves together, as mentioned. The rim rings are steel castings and provided with lugs for bolting together for the boring and facing operations in the boring mill. After being bolted to the sides of the cast iron center, the lugs are chipped off, leaving a smooth surface.

Fig. 14 shows a novel type of wheel suitable for the severe work of a traction plant. It is used on a 600 kilowatt set for power and lighting in the works of Messrs. Workman, Clark & Co., Ltd., Belfast. The diameter is 12 feet and the advantages claimed for it, which seem correct, are as follows:

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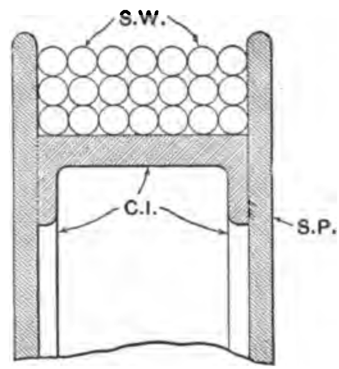
The rim is continuous, and the strength maintained, therefore, practically to the full. The number of bolts in the rim being much more numerous than spokes, the stresses that occur, due to the bending of the rim between the points of support, are correspondingly less. The steel disks connecting the rim with the hub are made very strong to resist the great torques of sudden changes of speed, a very important matter in a fly-wheel for electric traction. It is exceedingly cheap to make. The stresses in the arms, due to the cooling and shrinkage of a cast iron wheel are absent from a wheel of this type.

If a larger wheel of this type were made it could be made with the rim in sections, when all the above advantages would obtain, except the first. Another type of wheel, claimed to have been originated by Prof. Sharp, is shown to the left in Fig. 15.

Steel wire of great tensile strength is wound around the periphery



CROSS SECTION
S.W. = Steel Wire
S.P. = Steel Plates
C.I. = Cast Iron

Fig. 15. Fly-wheel having a Steel Wire Rim

of the wheel. With a well-made wheel of this type it is practically safe to run it at three times the velocity of an ordinary cast iron wheel. Hence it would store nine times the energy for the same weight, at the same radius of gyration, as a cast iron wheel.

A wheel of this type is used at the Mannsmans Tube Works. About 70 tons of steel wire was wound on the wheel with a tension of about 50 pounds. The fly-wheel was 20 feet in diameter and ran at 240 revolutions per minute, equal to a peripheral speed of about 250 feet per second. The speed of a cast-iron wheel of the same diameter would be about 100 feet per second.

Danger of Fly-wheels Bursting

As regards the danger of fly-wheels bursting, Professor Barr states that it is not realized how dangerous they are, and mentions a case in point. It was on an experimental engine. The makers of the fly-wheel, on which an experimental brake was used, had, contrary to his

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wishes, and entirely on their own responsibility, made the fly-wheel with a hub solid with the arms and rim. One evening, during the run, a report like a gunshot was heard and the observers noticed that the flywheel was running out of true. The rim of the wheel was just warm, about as warm as one's hand. The engine was stopped and they found three of the arms out and six broken. The middle one was open about 3/32 of an inch. There must therefore have been an enormous initial stress in the arms. There were two fly-wheels on the engines and the makers were told that they must replace both. They said they would replace the broken one with a new one having a split boss and cut the boss of the other wheel in two. They were warned as to what would happen, but they put the wheel in a slotting machine, and before they had cut half way through one side of the boss, the stresses of the wheel completed the job in a manner astonishing to the man running the slotter.

Great care must be taken regarding test specimens, as a test specimen cast from the same melting as the wheel does not necessarily indicate the same strength as that in the wheel. Test specimens vary also according to the way they are cast, so that a high factor of safety must be allowed in all cast wheels—say from 12 to 15.

Mr. C. A. Matthey, Scotland, says that, considering the ultimate strength of British cast iron as 16,000 pounds, it is safe to assume a factor of safety of 8, with a speed of 140 feet per minute, the arms to be cast with the rim but without the hub, thus avoiding cooling stresses, the hub being conscientiously fitted afterwards. This involves entering the arms sideways and not radially into pockets in the hub. He thinks that the attachment of the arms to the rim, when separate from solid rims, should be such as to drive the rim around without pulling it in toward the center. Let the rim support itself by its own tensile strength without radial pressures at a number of points.

The strength necessary in the arms of a fly-wheel has little if anything to do with the centrifugal force, and their sections should be proportioned to the driving moments they exert in storing up energy in the rim and in re-delivering it to the shaft. In certain kinds of engine service a good rule is to make the fly-wheel arms strong enough to pull up the wheel from full speed to a dead stop in one revolution.

From Mr. Marshall Downie's paper in *Transactions of the Institute of Engineers and Shipbuilders* (Scotland), the following is quoted: "The combined cross-sectional area of the arms in well designed wheels of the type used for electric traction, etc., is generally from two to three times the cross section of the rim. The strength of the arms as beams, fixed at the inner end and loaded at the outer end, with the force required to produce an acceleration, either plus or minus, in the mass of one segment, while changing the velocity through the limits specified in the time elapsing between two consecutive points of coincidence of actual and mean crank effort lines, should also be considered; and this, together with the resistance to shearing by the same load, should not tax the material above one-eighth of its ultimate load."

The fixing of the arms to the hub is usually by means of bolts or

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cotters and their strength in double shear should be equal to that of the arm in shear or tension, whichever is greater.

Fly-wheel Rim Joints

Several forms of rim joints are in use for fly-wheels. Among the principal ones are the following: (*a*) flanged and bolted; (*b*) dowel plate and cotters; (*c*) arrow-headed bolts; (*d*) links and lugs. As illustrated in Fig. 16, the following points must be observed:

(*a*.) In flanged and bolted joints, the bolts should be as near the rim as possible, consistently with getting a deep flange. The bolts should be carefully fitted at each end and cleared in the center, so that the stress on them should be tensile rather than shearing. They should all be initially stressed by screwing up, if possible to the same amount.

(*b*.) The accurate machining and fitting of the dowel plate and cotter joint is most important. It should be so designed that the strength of the cast-iron, cotters and portion of the dowel plate in shear is equal to the strength of the portion of the dowel plate in tension. The accuracy with which the initial stress in this form of joint can be adjusted is an important feature in its favor.

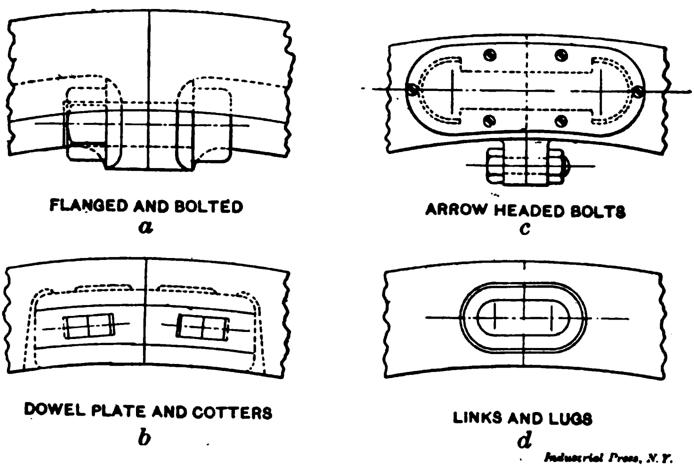

Fig. 16. Additional Types of Rim Joints

(*c*.) The arrow-headed bolt joint is a shrunk joint, and is open to the objection that the initial stress on the bolts due to the shrinkage is a more or less unknown quantity and the ultimate stress, therefore, indeterminate. The points to be attended to in its construction are accurate machining between the lugs on the bolts and rim, and provision for clearance at the center, for the same reason as noted in (*a*).

(*d*.) The link and lug joint is also a shrunk joint and subject to the same objections as (*c*) on that score. If made with the lug projecting, as shown in the figure, it has the advantage that the section of the rim is not diminished at the joint. The increase of weight, however,

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which such a form necessitates, is a good reason for removing the position of the joint nearer one arm. From the experiments of Prof. C. H. Benjamin reported in the *Proceedings of the American Society of Mechanical Engineers* and from the workings of the engines of the Metropolitan Street Railway, Dr. Downie has drawn the following conclusions:

A good average value for the energy necessary to be stored in fly-wheels for electric lighting purposes is 2.9 foot-tons per electric horse-power; and in traction plants, 4 foot-tons.

Where practicable, cast-iron fly-wheels should have one-piece rims, but when jointed the best form is the link and lug type, where such can be adopted without inconvenience, and the next best is the dowel plate and cotter. Flanged and bolted joints should be avoided and the best place for a joint is near one arm.

One-piece rim cast iron fly-wheels may be run at a peripheral speed of 100 feet per second with the certain knowledge that the factor of safety is not under 12, and link-jointed wheels may also be run at that speed and have a factor of safety of 8. A lower factor of safety should not be used, and flange-jointed wheels should not be run above 70 to 75 feet per second. Built steel wheels may be run up to 130 feet per second. Arms should be joined to rim with large fillets and their fixing to the hub should be carefully fitted.

The best material of its kind should be used in the construction and homogeneity should be insured as far as practicable by having test bars from each segment cast and examined.

Stresses in Fly-wheel Rims

The stress tending to cause rupture in a fly-wheel rim depends solely on the rim velocity, and is independent of the radius of the fly-wheel. This can be proved in the following manner:

The sum of the centrifugal (radial) forces of the whole rim of a fly-wheel is

$$F = \frac{W v^2}{g R} = \frac{4 W \pi^2 R r^2}{3600 g} = 0.000341 W R r^2,$$

where F = centrifugal force, in pounds,

W = weight of rim in pounds,

v = velocity of rim in feet per second,

g = 32.16,

R = mean radius of rim in feet,

r = revolutons per minute.

The resultant of half of this force tends to disrupt one-half of the rim from the other half. The rupture is resisted by the two sections of the rim at each end of the diameter. The resultant of half the radial forces is to the sum of half of the radial forces as the diameter of the fly-wheel is to half its circumference, or

$$\frac{\text{resultant}}{\text{sum of half the radial forces}} = \frac{1}{\frac{1}{2}\pi};$$

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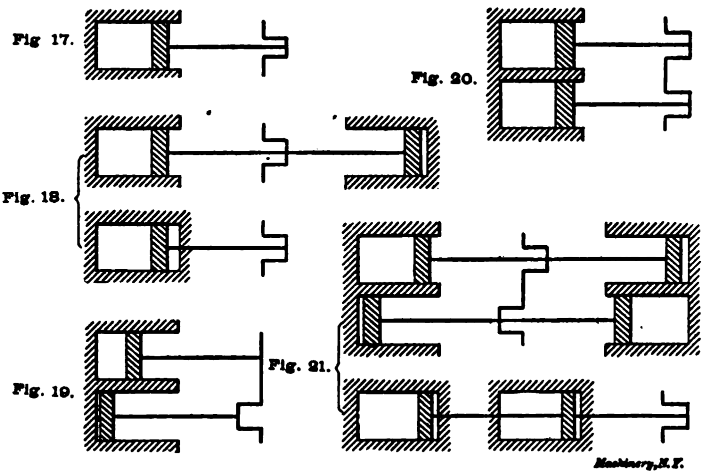
$$\text{resultant} = \frac{2}{\pi} \times \text{sum of half of the radial forces} = \frac{2}{\pi} \times \frac{0.000341\, W R r^2}{2}$$

$$= 0.00010854\, W R r^2.$$

As this resultant force is resisted by the section at each end of the diameter, each section must resist a force

$$S = \frac{0.00010854\, W R r^2}{2} = 0.00005427\, W R r^2.$$



Figs. 17 to 21. Diagrams of Different Types of Gas Engines

The weight of a rim of cast iron, one square inch in section, is $2\pi R \times 3.125 = 19.635\, R$ pounds, R being in feet. Consequently

$$S = 0.00005427 \times 19.635\, R \times R r^2 = 0.0010656\, R^2 r^2.$$

But as $v = \dfrac{2\pi R r}{60}$, and $v^2 = \dfrac{4\pi^2 R^2 r^2}{3600}$, we have

$$S = \frac{0.0010656\, v^2 \times 3600}{4\pi^2} = 0.0972\, v^2.$$

Thus the stress S in the fly-wheel rim is independent of the radius and depends only on the rim velocity.

Formula for Gas Engine Fly-wheels

The following formula for the calculation of fly-wheels for gas engines is applied by Mr. R. E. Mathot to all classes of engines. If, in the formula,

P = the weight of the rim (without arms or hub) in tons;
D = diameter of the center of gravity of the rim in meters;
a = the amount of allowable variation;
n = the number of revolutions per minute;
N = the number of brake horse-power;
K = coefficient varying with the type of engine:

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then, $P = K \dfrac{N}{D^2an^3}$.

If D is transformed to feet, the formula will read:

$$P = K \frac{10.75\,N}{D^2an^3}.$$

The coefficient K, which varies with the type of engine, is determined as follows:

$K = 44{,}000$ for Otto-cycle engines, single-cylinder, single-acting, (Fig. 17.)

$K = 28{,}000$ for Otto-cycle engines, two opposite cylinders, single-acting, or one cylinder double-acting. (Fig. 18.)

$K = 25{,}000$ for two cylinders single-acting, with cranks set at 90 degrees. (Fig. 19.)

$K = 21{,}000$ for two cylinders, single-acting. (Fig. 20.)

$K = 7{,}000$ for four twin opposite cylinders, or for two tandem cylinders, double-acting. (Fig. 21.)

The factor a, the allowable amount of variation in a single revolution of the fly-wheel is as follows:

For ordinary industrial purposes......................	1/25 to 1/30
For electric lighting by continuous current.............	1/50 to 1/60
For spinning mills and similar machinery.............	1/120 to 1/130
For alternating current generators in parallel..........	1/150

The total weight of the fly-wheel may be considered as equal to $P \times 1.4$.

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CHAPTER II

FLY-WHEEL TESTS

Fly-wheels do not often break from faults of construction or material, but from a sudden increase of speed due to some accident. Most fly-wheels that fail are really sound and safe under ordinary conditions. The fact that a defective wheel may run for years and then suddenly burst, shows that the immediate cause was not the defects, but some change from the normal condition under which the wheel had been running. Wheels do not often fail from torsional stress or from twisting action in pulling their load, because enough material can be put in the wheel to resist successfully any load required of the engine. There is, however, no possible way to overcome the centrifugal force due to speed. Increasing the thickness of the rim of the wheel does not strengthen it so far as centrifugal force is concerned, because the weight added also increases the centrifugal force, leaving the wheel no stronger than before. There is, therefore, a definite speed at which any wheel, however sound, will explode, regardless of the amount of material it contains.

For cast iron wheels sound theory borne out by good practice has fixed *a mile a minute* as the danger limit for the rim speed. Most wheels are run at or near this speed. The normal speed, however, is only of incidental consideration, as an accident may at any moment allow the speed to get beyond the normal. As the stress in the rim due to centrifugal force increases with the *square of the speed*, disruption quickly follows even a slight increase in speed. If the speed should be accidentally tripled, for example, the stress in the rim would become *nine times* as great as before, and the wheel would have exploded long before it had attained that speed. As a matter of fact, the percentage of fly-wheels that explode is 33 per cent greater than the percentage of boilers that explode.

Fly-wheel Tests at the Case School of Applied Science

For several years tests have been conducted at the Case School of Applied Science, Cleveland, Ohio, to find the relative strength of fly-wheels of different designs and proportions, and the results of these form the best data we have upon the strength of such wheels at the present time. The tests were made upon small model wheels, 15 inches to 2 feet in diameter, run at enormously high speeds by means of a steam turbine, until they finally burst. Apparatus was provided for recording the speed at the time of bursting. At the annual meeting of the American Society of Mechanical Engineers in 1898, Prof. C. H. Benjamin gave the results of the tests made up to that time and drew the following conclusions:

1.—Fly-wheels with solid rims, of the proportions usual among engine

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builders, and having the usual number of arms, have a sufficient factor of safety at a rim speed of 100 feet per second, if the iron is of good quality and there are no serious cooling strains. In such wheels the bending due to centrifugal force is slight and may be safely disregarded.

2.—Rim joints midway between the arms are a serious defect, and reduce the factor of safety very materially. Such joints are as serious mistakes as would be a joint in the middle of a girder under a heavy load.

3.—Joints made in the ordinary manner, with internal flanges and bolts, are probably the worst that could be devised for the purpose. Under the most favorable conditions they have only about one-fourth the strength of the solid rim and are particularly weak against bending. In several joints of this character on large fly-wheels, calculation shows a strength less than one-fifth that of a solid rim.

4.—The type of joint having the rim held together with links is probably the best that could be devised for narrow rimmed wheels not intended to carry belts, and possesses, when properly designed, a strength about two-thirds that of the solid rim.

At the 1901 meeting of the society, Prof. Benjamin gave some additional data, deduced from experiments conducted since 1898. Wheels with solid rims were again tested, to afford a standard for comparison by which wheels with jointed rims of various designs could be judged. These burst at a rim speed of 395 feet per second, corresponding to a centrifugal tension of about 15,600 pounds per square inch.

Four wheels were tested with joints and bolts inside the rim, after the familiar design ordinarily employed for band wheels, but with the joints located at points one-fourth of the distance from one arm to the next, these being the points of least bending moment, and, consequently, the points at which the deflection due to centrifugal force would be expected to have the least effect. The tests, however, did not bear out this conclusion. The wheels burst at a rim speed of 194 feet per second, corresponding to a centrifugal tension of about 3,750 pounds per square inch. These wheels, therefore, were only about one-quarter as strong as the wheels with solid rims, and burst at practically the same speed as wheels in the previous series of tests in which the rim joints were midway between the arms. This is doubtless due to the fact that the heavy mass of the flanges and bolts locates the bending moment near them. In these wheels the combined tensile strength of the bolts in the flange joints was slightly less than one-third the strength of the rim, which is about the maximum ratio possible with this style of joint.

Another type of wheel with deep rim, fastened together at the joints midway between the arms by links shrunk into recesses, after the manner of fly-wheels for massive engines, gave much superior results. This wheel burst at a speed of 256 feet per second indicating a centrifugal tension of about 6,600 pounds per square inch.

Tests were made on a band wheel having joints inside the rim, midway between the arms, and in all respects like others of this

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design previously tested, except that tie rods were used to connect the joints with the hub. It burst at a speed of 225 feet per second, showing an increase of strength of 30 to 40 per cent over similar wheels without the tie rods. Several wheels of special design, not in common use, were also tested, the one giving the greatest strength being an English wheel, with solid rim of I-section, made of high-grade cast iron and with the rim tied to the hub by steel wire spokes. These spokes were adjusted to have a uniform tension by "tuning," and the wheel gave way at a rim speed of 424 feet per second, which is slightly higher than the speed of rupture of the solid rim wheels with ordinary style of spokes.

The Bursting of a Four-foot Fly-wheel

At the December, 1904, meeting of the American Society of Mechanical Engineers, Prof. Benjamin read a paper regarding further fly-wheel tests. This time the tests were made on fly-wheels four feet in diameter. To insure safety to the students and to the building of the Case School of Applied Science, these tests were conducted outdoors, in consequence of which they nearly proved disastrous to the neighbors. The fly-wheels were run in a casing of steel castings, located in a pit lined with brick. The flanges of the lower half rested on brick piers and were bolted in place. The entire upper half of the casing could be hoisted up, giving access to the interior for hoisting or removing the wheels. Two wheels were broken successfully, but the third one burst through its bounds and carried the casing with it many feet in the air. Fortunately, every precaution for safety had been taken, all the observers being located far away from the plane of rotation of the wheel.

In carrying out these experiments, the shaft supporting the wheel to be tested turned in bearings bolted to angle irons on the lower halves of the side plates, and was connected to the driving mechanism just inside the building by a flexible sleeve coupling. After the wheel was in place, the casing was lined with wooden blocks to absorb the momentum of the flying fragments. Instead of using a steam turbine as in former experiments, the fly-wheel shaft was speeded up by means of a Reeves variable speed countershaft, interposed between line-shaft and driving-shaft.

The first wheel to be experimented on was a well-proportioned cast-iron pulley, such as is used on shafting for transmitting power. This pulley was 48 inches in diameter, had six arms and weighed 194 pounds. The rim was whole and was 8½ inches wide and about ⅜ inch thick, finished on the outside. The arms were elliptical in section, 3⅛ inches by 1 1/16 inch at the hub, and 2 inches by ¾ inch at the rim. On the whole the wheel was well-designed and showed no signs of shrinkage strains. It had, however, been balanced in the customary manner by riveting a cast-iron washer inside the rim at the lighter side, and this proved its undoing. The combination of a thin place in the rim, a rivet hole and heavy mass of cast iron is enough to wreck any wheel.

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As has been shown by previous experiments on whole rim wheels of cast-iron, a bursting speed of 400 feet per second may be reasonably expected. The circumference of a four-foot wheel being about 12½ feet, such a wheel should burst at about 32 revolutions per second, or 1,920 revolutions per minute. The pulley in question burst at 1,100 revolutions per minute, as recorded by a tachometer connected to the driving-shaft. The balance weight weighed 3½ pounds, and its center was approximately 23 inches from the axis of rotation. At 1,000 revolutions per minute the centrifugal force of the balance weight alone would be 2,760 pounds. Add this radial pressure at a weak point between the arms to that due to the weight of the rim itself, and the low bursting speed is easily accounted for. The linear speed of the rim at rupture was 230 feet per second. As 100 feet per second is considered the limit for belt speed, this pulley would have a working factor of safety of $(2.3)^2$ or 5.3. But suppose the rim had been a little thinner and consequently a bigger weight had been put on with a larger rivet?

Wheel No. 2 was a cast-iron pulley of the same general style and dimensions as No. 1, but with a split hub and rim. The balance-weight was present here as in the former case, but was obliged to yield the palm to its rival, the flanged joint. The wheel had been cast in one piece, as is usual in such cases, with cavities cored at the joints of rim and hub. After finishing, it had been broken apart by wedges, making a fracture joint. The flanges, being located midway between the arms and bolted at some little distance inside the rim, were in the worst possible position to withstand the bending action due to centrifugal force, and their own weight only aggravated the difficulty. The flanges weighed with their bolts 7½ pounds. This wheel burst at less than 700 revolutions per minute, the tachometer not recording below this speed. It was estimated that the speed was only about 600 revolutions per minute. At this speed the centrifugal force of the flanges on one side would have been 1,680 pounds. At 600 revolutions per minute the linear speed of rim would be only 125 feet per second. At the very common belt speed of 4,500 feet per minute the factor of safety would be but 2.8, which is altogether too low, considering the nature of the material and the shocks to which a pulley may be exposed.

It was reserved for wheel No. 3 to develop the most dramatic series of incidents of any yet experimented upon, big or little. This wheel measured 49 inches in external diameter and weighed about 900 pounds. The rim was 6¾ inches wide and 1⅛ inch thick, and was built of ten segments, the material being steel casting. Each joint was secured by three prisoners of an I-section on the outside face, by link prisoners on each edge, and by a dove-tailed bronze clamp on the inside, fitting over lugs on the rim. The arms were of phosphor bronze, twenty in number, ten on each side, and were a cross in section. These arms came midway between the rim joints and were bolted to plane faces on the polygonal hub. The rim was further reinforced by a system of diagonal bracing, each section of the rim being sup-

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ported at five points on each side, in such a way as to relieve it almost entirely from bending. The braces, like the arms, were of phosphor bronze. and all bolts and connecting links of steel. This wheel was designed by a Baltimore firm as a model of a proposed 30-foot fly-wheel.

On account of the excessive air resistance it was found necessary to enclose the wheel at the sides between sheet-metal disks, before any great speed could be attained. Even then repeated trials failed to reach a speed of more than 800 or 900 revolutions per minute on account of the great inertia of the wheel, and the consequent slipping of belts. By putting on more and wider belts, and by a liberal use of "Cling-Surface" and with the aid of a 7½ horse-power electric motor belted on in parallel, it was found possible to get a speed of 1,650 revolutions per minute, and after the wheel had been run at this speed it was stopped and examined.

The inspection showed fracture of several of the I-shaped prisoners on the outer surface of the joints and a slight opening of the joints themselves, to the extent of perhaps one or two hundredths of an inch. On June 2, 1903, the casing was closed for the last time, and the combination of driving mechanisms set to work. The observers were all well protected by the thick piers of the building, while other spectators were kept at a safe distance and well away from the plane of rotation. Two of the observers watched the pointer of the tachometer through opera glasses, another kept the time, while a fourth manipulated the driving levers.

As the hand of the speed counter reached and slowly passed the 1,600 mark, the feeling of suspense on the part of those watching reached the acute stage. The pointer crept slowly on and when it quivered on the mark of 1,775, there was a sudden crash, a sound of rending and tearing, and the observers saw the countershaft inside writhing on the floor like a wounded snake. On stepping outside they were saluted by a shower of falling splinters and fine debris, and were surprised—putting it mildly—to note the disappearance of the greater part of casing and wheel.

The steel rim of the casing was broken about six inches below one of the flanges, and the entire upper half weighing half a ton was projected about 75 feet into the air and landed some hundred feet away on the campus. On its way up it carried away part of the cornice of the building, and this collision was probably what caused it to deviate so much from a vertical path. The hub and main spokes of the wheel remained nearly in place, but parts of the rim were found two hundred feet away, while one large fragment landed on the roof of the building.

This sudden failure of the rim casing was unexpected, as it was thought the flange bolts were the parts to give way first. The tensile strength of the casing at the point of fracture was about 1,200,000 pounds, or about four times the strength of the wheel rim at a solid section. Examination of the break in the casing showed a clean, bright fracture, with almost no imperfections.

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The failure of the wheel itself was due to a gradual opening of the joints, occasioned by the fracture of the outside prisoners, and to flaws in the bronze castings of the arms near their junction with the rim. On putting the pieces of the wheel together in their original order it was easy to locate the joint which first gave way, on account of the symmetry of the breaks either side of a diameter through this point. It is but fair to the builders of the wheel to say that the fractures showed uniformity of strength and of workmanship, since there was hardly a member or a joint which did not fail in one part of another of the wheel.

One thousand seven hundred and seventy-five revolutions per minute means a linear speed of rim of 22,300 feet per minute, or 372 feet per second would be 645,000 foot-pounds. Further assuming that none of rim of good design, but it is greater than the speed of any sectional or jointed rim which has been tested. The tensile stress due to the centrifugal force at this speed is 13,800 pounds per square inch. This shows that the joints were much weaker than the solid rim. On the whole, the test of this particular wheel was disappointing, since its strength was not sufficient to repay one for the expense of the design.

It is interesting to compare the kinetic energy of the rim of the wheel at the recorded speed with the work of destruction. Assuming the rim with its lugs, flanges, etc., to weigh 300 pounds, which is a reasonable estimate, the kinetic energy at a speed of 372 feet per second would be 645,000 foot-pounds. Further assuming that none of the energy was dissipated in heat, and that the combined mass of wheel and casing projected into the air weighed 1,500 pounds, we find the height of projection to be 430 feet.

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CHAPTER III

SAFE SPEED FOR FLY-WHEELS

The following is an abstract from an article by Mr. William H. Boehm, in the *Monthly Bulletin of the Fidelity and Casualty Company.* It is a well understood fact that while an engine pulley or fly-wheel can be designed to successfully resist the torsional stresses due to any load required of the engine, there is no possible way to overcome the centrifugal force due to speed. For a given material there is a definite speed at which disruption will occur regardless of the amount of material used. This is not an uncertain theory, but a mathematical truth easily demonstrated, and is expressed by the following formula:

$$V = 1.6\sqrt{\frac{S}{W}}$$

in which V is the velocity of the rim of the wheel in feet per second at which disruption will occur, W the weight of a cubic inch of the material used, and S the tensile strength per square inch of the material.

In words the formula means that if we divide the tensile strength of the material by its weight per cubic inch, extract the square root of the quotient and then multiply this by 1.6, the result will be the rim speed in feet per second at which disruption will occur. If instead of the ultimate strength of the material we take its safe strength, the result will be the rim speed in feet per second at which the wheel may be run with safety; the supposition so far being, however, that the rim is made solid in one piece of homogeneous material and free from shrinkage strains.

Applying the formula to determine the safe rim speed for cast iron wheels made in one piece, we would assume that, if the ultimate strength of small test bars were 20,000 pounds per square inch, we could depend upon having 10,000 pounds in large castings. Using a factor of safety of 10 on this would give 1,000 pounds per square inch as the safe strength of this material. The weight of a cubic inch of cast iron is approximately 0.26 pounds, so that we have for cast iron wheels:

$$V = 1.6\sqrt{\frac{S}{W}} = 1.6\sqrt{\frac{1000}{0.26}} = 99.2$$

per second; so that 100 feet per second may be regarded as a safe rim speed for cast-iron wheels made in one piece. This corresponds to about 1.15 miles per minute, but as such wheels are likely to contain shrinkage strains, it is not considered good practice to run them faster than a mile a minute.

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If the wheel is made in halves, or sections, the efficiency of the rim joint must be taken into consideration. For belt wheels with flanged and bolted rim joints located between the arms, the joints average only one-fifth the strength of the rim, and no such joint can be designed having a strength greater than one-fourth the strength of the rim. If the rim is thick enough to allow the joint to be reinforced by steel links shrunk on, as in heavy balance wheels, one-third the strength of the rim may be secured in the joint, but this construction cannot be applied to belt wheels having thin rims.

Applying the formula to wheels made of steel casting having an ultimate strength of 60,000 pounds per square inch, or a safe strength of 6,000 pounds per square inch, and weighing 0.28 pound per cubic inch, we have:

$$V = 1.6\sqrt{\frac{S}{W}} = 1.6\sqrt{\frac{6000}{0.28}} = 234.3$$

per second; so that a steel casting wheel made in one piece and free from shrinkage strains could be run with perfect safety at a rim speed of 234 feet per second, corresponding to 2.66 miles per minute.

It will perhaps surprise some mechanics to learn that wheels made of wood may be run at a higher speed than those made of cast iron. Wood, however, is one of the very best materials that can be used for fly-wheel construction, and many large wheels have been constructed of this material and are giving satisfactory results. Applying the formula to hard maple having a tensile strength of 10,500 pounds per square inch, and weighing 0.0283 pound per cubic inch, we have, using a factor of safety of 20, and remembering that the strength is reduced one-half because the rim is built up of segments,

$$V = 1.6\sqrt{\frac{S}{W}} = 1.6\sqrt{\frac{262.5}{0.0283}} = 154.1$$

per second; so that a well-made maple wheel may be run with perfect safety at a rim speed of 154 feet per second, which corresponds to 1.75 miles per minute. Or comparing two wheels of the same diameter, one of cast iron, tne other of maple, the number of revolutions per minute for the maple wheel may be 54 per cent greater than for the cast iron wheel. One hundred and fifty-four feet per second would not, however, be a safe rim speed for the wood wheel if made in halves or sections, on account of the weakness of rim joints.

Of late years wheels for large electric plants have been built up of steel plates riveted together, and wheels for special work or unusually high speed have been specially designed. It is questionable, however, whether the complicated built-up steel construction is profitable for wheels of standard steam engines as commercially built. When an engine runs fast enough to burst a well-made cast iron wheel it is doubtful whether anything would save it. The small amount of time required for the additional acceleration necessary to burst a steel wheel at that stage would be little, and when the crash did come, it would be all the more disastrous.

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From the above formulas it will be seen that the stress in the rim of a wheel increases with the square of the speed, or, to put it in other words, the factor of safety on speed is always the square root of the factor of safety on strength. If the speed be tripled, for example, the stress in the rim becomes nine times as great as before; that is, with a factor of safety of nine on strength, there is a factor of safety of only three on speed. It will be understood from this that the stress increases enormously for even a slight increase in speed.

Let us consider the usual cast iron, sectional, belt wheel having flanged and bolted rim joints located between the arms. As pointed out above, such joints average a strength of only one-fifth the strength of the rim, and no joint of this kind can be designed that will have a strength greater than one-fourth the strength of the rim. If this wheel had at normal speed a factor of safety of 12 in the rim, then with joints of maximum strength the factor of safety in the joint would be only 3 on strength or 1.73 on speed. That is, an increase in speed of 73 per cent would burst the wheel. The wide gulf in this case between the apparent factor of safety of 12 on strength and the real factor of safety of 1.73 on speed is appalling. This is, however, only another warning that things are not always as they seem.

As a matter of fact, few wheels have a margin of safety of 73 per cent on speed. In the accident of the Amoskeag Mills, in which a 30-foot wheel wrecked the building, killed two girls and badly injured the assistant engineer, the evidence proved that an increase in speed of only 20 per cent caused the disaster. Many wheels in use to-day are running on a narrower margin than this. It will now be understood why racing is so frequent a cause of fly-wheel accidents. Some slight accident to the governor or valve gear of the engine occurs, and away goes the wheel, causing a costly if not fatal wreck. The stress in the rim increases so rapidly with increase of speed that sound wheels amply safe at normal speed, go to pieces without warning, and apparently without cause.

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CHAPTER IV

SIZE, WEIGHT AND CAPACITY OF FLY-WHEELS FOR PUNCHES

In this chapter will be given a method of determining the size, weight and capacity of a fly-wheel to punch a given size hole through a given thickness of metal.

Effect of Relative Size of Punch and Die, and Shape of Punch

To begin with, there are a number of things which affect the effort that is required to punch a certain size hole through a given thickness of metal. In Fig. 22, *P* is the punch, *A* is the diameter of the punch, and $A + x$ is the diameter of the hole in the die. For the regular run of work, and for a ¾-inch punch, the hole in the die would be about 1/32 inch larger than the punch. If we reduce the

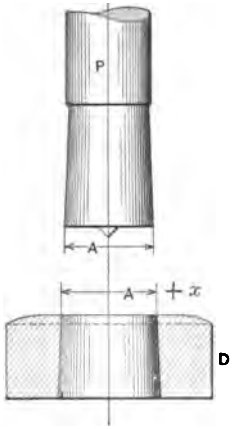

Fig. 22

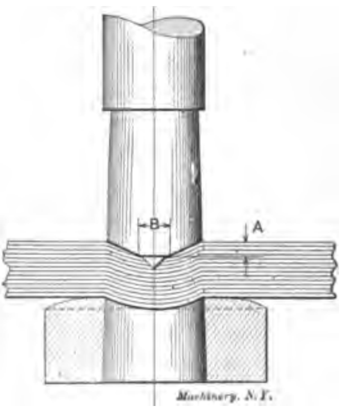

Fig. 23

size of the hole, the effort necessary to punch the hole will be greatly increased, and the life of the punch will be short, but if we increase the size of the hole, within certain limits, the effort required to punch the hole will be less, and the life of the punch will be greatly increased. The use of a large hole in the die causes a cone-shaped hole in the sheet, which is always more or less objectionable, and, therefore, one cannot get too far away from the standard proportions used by punch makers. The punching effort required will also be decreased by the use of a punch which has something of a shearing action, as shown at *A*, Fig. 23. The flat portion, *B*, enters the sheet first and probably presents no more than one-fourth the total cutting circumference of the punch. By the time the whole punch has entered into the sheet, which would represent the greatest effort required, the

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metal under *B* is nearly sheared away. Through the remainder of the stroke there is a shading off of the effort required to remove the metal. The shape of the punch with reference to the diameter of the end and of the body also has some effect upon the effort.

Fig. 24 shows a regular flat punch. The sides of *S* are tapered off gradually from ¾ inch at the bottom to 11/16 inch at the top. Fig. 25 shows a similar punch with the sides parallel, but flaring off at the bottom for a distance of 3/16 inch. There is little difference in the effort required in using either of these punches when both are new. But when they become worn the side pressure against the punch is considerable. It is this wearing off of the sides which causes the greatest trouble in punching. The style shown in Fig. 25 is used a

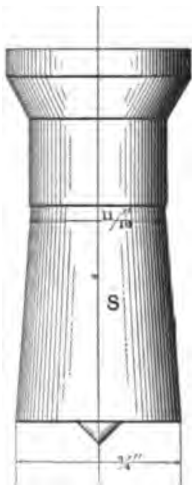

Fig. 24

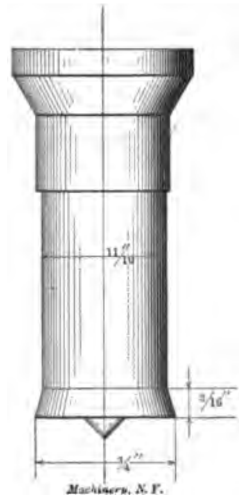

Fig. 25

great deal in structural work, and seems to give less trouble from side friction than the punch shown in Fig. 24.

Punching Effort Proportional to Area Sheared

In calculating the size fly-wheel which will be necessary to punch a given hole, a flat punch only will be considered, and it will be assumed that the punches are kept in fairly good condition. Also, the calculations will be based upon punching wrought iron and steel, such as boiler plate, angles, tees, bars, etc.

The area sheared off in punching a 1-inch hole through a ¾-inch plate is the circumference of a 1-inch circle, times the thickness of the sheet. The circumference of a 1-inch circle is 3.1416 inches.

Let A = area to be sheared = $3.1416 \times \frac{3}{4} = 2.3562$ square inches, or say, for all practical purposes, = 2.36 square inches.

For ordinary run of work, we will use a shearing resistance stress of 60,000 pounds per square inch. In working with harder or softer material, this shearing stress will have to be taken higher or lower, depending upon the shearing stress of different metals.

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Let $P =$ the push required to punch the hole, or the shearing effort,
$S =$ shearing stress per unit of area $=$ 60,000 pounds per square inch.

We then have

$P = A \times S$, and for the case considered $= 2.36 \times 60{,}000 = 141\,600$ pounds $=$ effort required to punch a 1-inch hole through 3/4 inch plate.

In order to punch such a hole, a large amount of energy will be required for a brief period of time, as one can infer from the crank circle shown in Fig. 26, in which the punching is represented as being all done through the small portion *T* of the circumference. This distance represents the distance that the crank-pin passes through while removing the metal, *D* being the the diameter of the crank-pin circle. It will be seen from the case shown that *T* represents about one-tenth of the crank circle. The energy required for punching would have to be given out in about one-tenth revolution of the eccentric

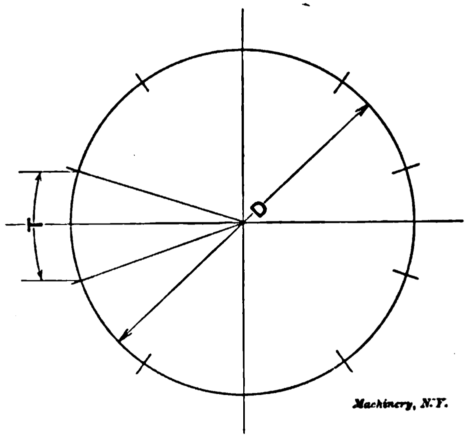

Fig. 26. Diagram of Crank-Pin Circle

shaft. During the meantime the machine can pick up energy through the other nine parts of the circumference. If the fly-wheel is properly proportioned, and if the energy applied to the machine is sufficient, the fly-wheel will pick up through these nine parts of the circumference sufficient energy to do the punching while the crank-pin is passing through the tenth part of the circumference.

Design of Fly-wheel and its Function

A good design of fly-wheel is shown in Fig. 27. The ledge *L* inside the fly-wheel extends from arm to arm, which makes very strong connection between the arm and rim. The outside diameter *D* of the fly-wheel as well as the sides are machined. The hub *H* should never be less than two diameters of the shaft. A good deal depends upon the strength of this hub, and as the extra metal required to increase the size of the hub is small in proportion to the size of the fly-wheel, it is good practice to make the hub, say, from 2½ to 3 times the diameter of the shaft.

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In order that the fly-wheel shall give out energy, it must slow down in speed. If the fly-wheel is not large enough, the energy required will be greater than the capacity of the fly-wheel, and the change in speed will be great. In some cases a machine might even be stopped owing to the fly-wheel not having energy enough. If a fly-wheel is properly designed it will perform its work and slow down in speed a certain percentage, but this must not be so great that the machine cannot pick up again for the next stroke. The amount that the fly-wheel can be slowed down by taking its energy away from it is a matter of experiment. For ordinary punch and shear work we can take this drop in speed to be about 20 per cent while the machine is doing the work. This would have to be regained through the belt or

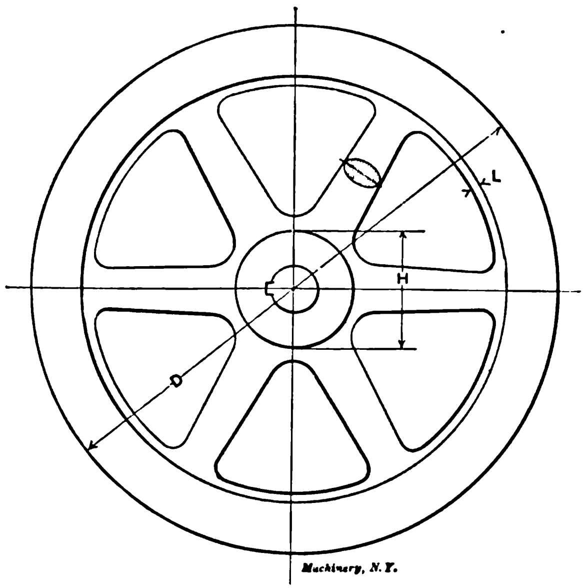

Fig. 27. Design of Fly-wheel

through the motor during the remaining portion of the stroke so that the fly-wheel would be up to speed again for punching the next hole.

There are many belted punches which are running along and doing their work satisfactorily which are not at all up to this standard of requirement. The reason for this is that these machines punch a hole only "once in a while." The drop in speed is very much greater than one-fifth, being probably one-third. If one should take such a machine with the rated capacity of 1 inch through ¾-inch plate, and punch one hole after the other without missing a stroke, the machine would stop. In this connection, therefore, it will be noted that there is a chance for a great variation in the size of fly-wheel and the horse-power required to drive a punch. In these calculations the fly-wheel will be so proportioned as to punch its rated capacity for every stroke for continuous working.

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To Calculate the Potential Energy of a Fly-wheel for a Given Reduction of Velocity

Let V = velocity of center of gravity of fly-wheel rim at normal speed before punching, in feet per second,

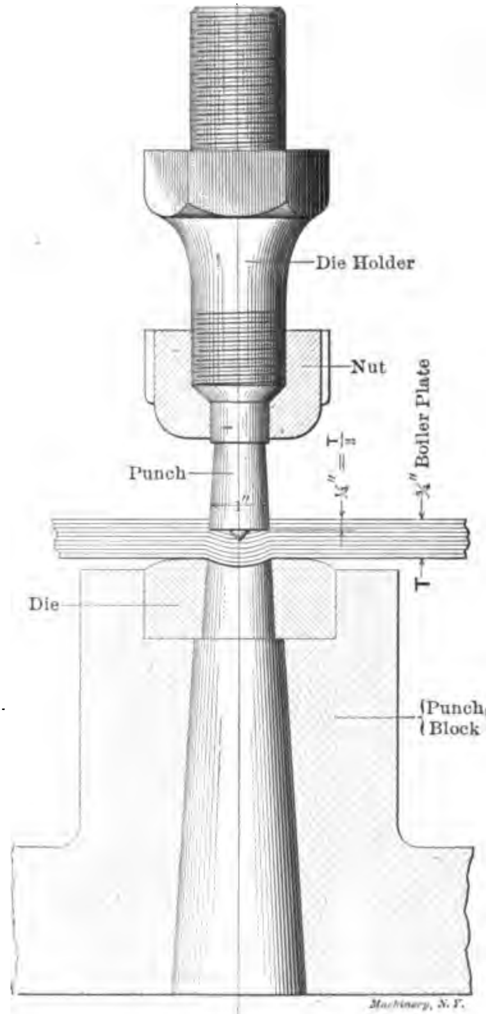

Fig. 28. Diagram Illustrating Part of Stroke Offering Maximum Resistance to Punching

E = the energy delivered to the fly-wheel or given out by the fly-wheel for one stroke.

W_r = weight of the rim,

W_a = weight of the arms,

g = 32 (approximately) = acceleration due to gravity,

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V_1 = velocity of mean periphery of fly-wheel rim after punching, in feet per second.

$$\text{Then } E = (W_r + \tfrac{1}{3} W_a)\left(\frac{V^2 - V_1^2}{2g}\right) \qquad (1)$$

In this expression W_a represents the weight of the arms. This is a very small percentage of the total weight of the fly-wheel, and for all purposes we can neglect this item.

Neglecting item 1/3 W_a we have for (1)

$$E = W_r \frac{V^2 - V_1^2}{2\,g}$$

$$= W_r \frac{V^2 - V_1^2}{64} \qquad (2)$$

To Calculate the Weight of the Fly-wheel

E also equals the energy necessary to punch a 1-inch hole through a ¾-inch plate. Experiments show that when a punch has entered about one-third way through the sheet, see Fig. 28, the material is all sheared off, or in other words, when the punch has passed one-third way through the sheet, the hole is punched, and it then only remains to push the punching out through the die.

Let T = thickness of plate = ¾-inch; we then have

$$E = P \times \frac{1/3\,T}{12}$$

$$= \frac{P \times 1/3 \times 3/4}{12}$$

$$= \frac{141{,}600}{4 \times 12}$$

= 2,950 foot-pounds = energy required per stroke.

By transposing equation (2), we have

$$W_r = \frac{E \times 64}{V^2 - V_1^2} \qquad (3)$$

In order to determine the size of the fly-wheel, we must know the speed of the fly-wheel, and we must assume a diameter which in our judgment would be approximately correct. We will take for the present case a single-ended punch, as shown in Fig. 29, with bottom drive, with tight and loose pulleys and with a single fly-wheel F running at a normal speed of 175 R. P. M. before punching and falling off 20 per cent during the actual punching operation. This machine should take a fly-wheel about 36 inches outside diameter, or say about 30 inches diameter at center of gravity of rim. The velocity in feet per second would be

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$$V = \frac{\text{dia.} \times \pi}{12} \times \frac{175}{60}$$

$= 23$ feet. Substituting in (3) we get

$$W_r = \frac{E \times 64}{V^2 - V_1^2} = \frac{2950 \times 64}{23^2 - 18.4^2}$$

$= 992$ pounds, weight of fly-wheel.

This fly-wheel would be made of cast iron and the section of the rim would be obtained thus:

Let $B =$ the face of the fly-wheel (see Fig. 30) $= 6\frac{3}{4}$ inches,

$H =$ the average thickness of the rim. We then have

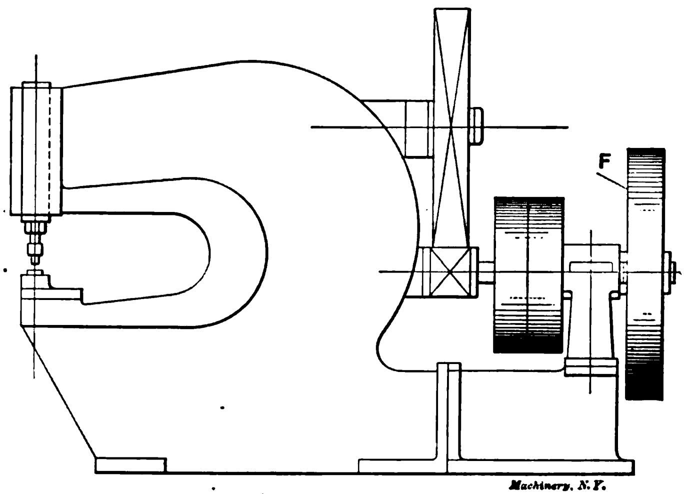

Fig. 29. Single-ended Punch

$W_r = 6\frac{3}{4} \times H \times 30 \times \pi \times 0.26$, and transposing

$$H = \frac{W_r}{6\frac{3}{4} \times 30 \times \pi \times 0.26}$$

$$= \frac{992}{6\frac{3}{4} \times 30 \times \pi \times 0.26}$$

$= 6$ inches depth of rim.

The fly-wheel, therefore, should be 36 inches outside diameter with a rim $6\frac{3}{4}$ inches face by 6 inches thick.

Effect of Frame Elasticity in Reducing Efficiency

There is another thing which should be mentioned in connection with the size of a fly-wheel which would be required to do a certain amount of work. If the machine is not stiff in the frame or shafting, a large amount of energy will disappear, and there is apparently nothing to show for it. This can best be explained by referring to Fig. 31, which shows a double-ended punch. If the shaft S is small

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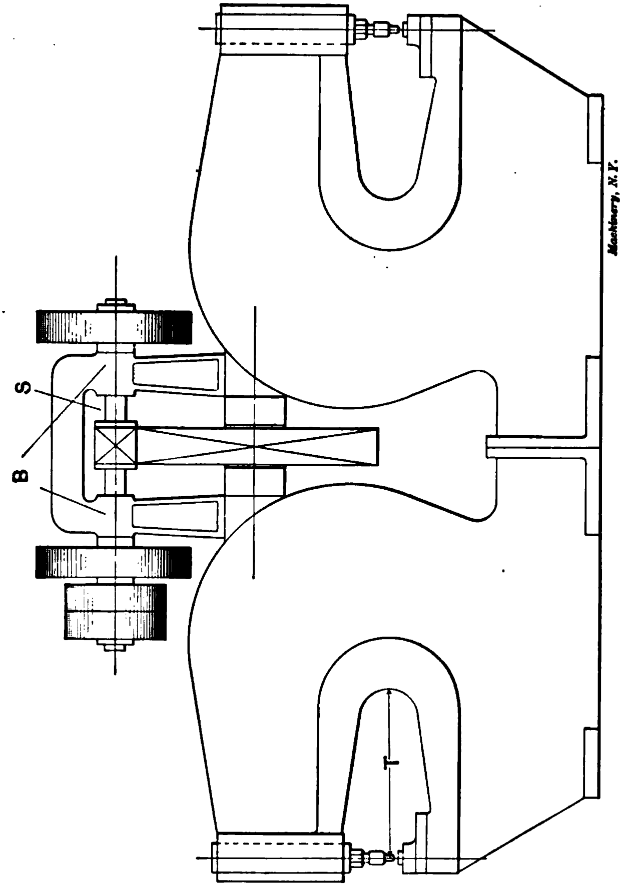

Fig. 31. Double-ended Punch

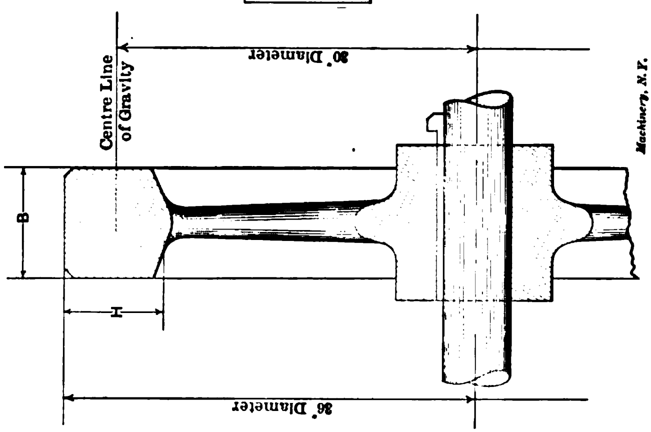

Fig. 30

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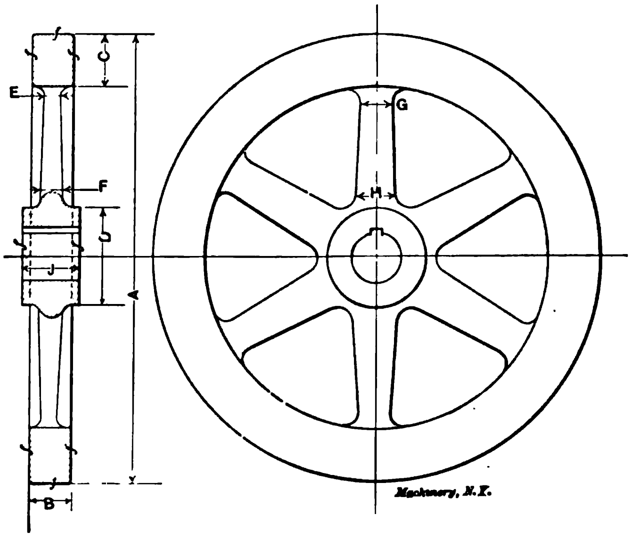
in diameter, or if the distance between the bearings *B* and *B* is great, this shaft will spring, and the result or the effect of the fly-wheel is "deadened." Also, if the eccentric shaft is very long and is small in diameter, it will have the same effect, hence the great importance of a solid machine for punching. It is remarkable what capacity the upright punching press has, but this is largely due to the very solid construction. The metal in the upright is in direct tension, therefore the spring or stretch is small. With a regular punching machine,

TABLE III. DIMENSIONS OF FLY-WHEELS FOR PUNCHES



A	B	C	D	E	F	G	H	J	Max. R.P.M.
24	3	3½	6	1¼	1⅛	2¾	3¼	3½	955
30	3½	4	7	1⅜	1¼	3	3¾	4	796
36	4	4½	8	1½	1½	3¼	4¼	4½	637
42	4¼	4¾	9	1⅝	2	3½	4½	5	557
48	4½	5	10	1¾	2	3¾	4¾	5½	478
54	4¾	5½	11	2	2¼	4	5	6	430
60	5	6	12	2¼	2½	4½	5½	6½	382
72	5½	7	13	2½	2¾	5	6½	7	318
84	6	8	14	3	3½	5½	7½	8	273
96	7	9	15	3¼	4	6	9	9	239
108	8	10	16½	3¾	4½	6½	10½	10	212
120	9	11	18	4	5	7½	12	12	191

however, there are a number of chances for spring, and each cuts down the fly-wheel effect.

With a short throat punch there is not much spring in the frame, but with a deep throat punch the spring is considerable in amount. A spring of 1/8 to 3/16 inch at the dies is a very common thing. A deep throat machine will punch far beyond its rated capacity if the tie-

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rods are placed close up to the head. This stiffens the machine and concentrates the work of the fly-wheel on the metal being punched. A short throat punch is usually rated higher in capacity than a deep throat punch of the same pattern. In figuring the size fly-wheel, therefore, it should be made large enough to do the work of a short throat punch.

When a double-end punch is required, as in Fig. 31, one or two fly-wheels may be used. Frequently, on account of the limited space, two fly-wheels must be used. This wheel or wheels, as the case may be, should be calculated to do the continuous work of both ends of the machine. It will be noted in equation (2) that E varies with the square of the velocity of the fly-wheel; we can take advantage of this fact sometimes, where a punch has a fly-wheel that is somewhat too light. The machine can be speeded up, which will give the fly-wheel more energy, and in this way will punch up to the capacity of the machine.

Limitations of Fly-wheel Size and Speed

In practise there are a number of things which limit the diameter and speed of a fly-wheel, and in such cases the weight must be gotten by either increasing the face and thickness of the rim or else putting on two fly-wheels. Table III, on the previous page, gives the dimensions of fly-wheels. The last column gives the maximum R. P. M. at which a cast iron fly-wheel should be run. There are cases where very high speeds of fly-wheels cannot be avoided, but as far as possible the tendency is to use a heavy fly-wheel at moderate speed and one or two runs of heavy gears.

If a punch is fitted with a proper size fly-wheel, and the motor or pulleys are too small when running on continuous work, the machine will slow down and stop. In the case of a belted machine, the belt will break or slide off the pulley, and in the case of motor drive, the motor probably will be so overloaded as to cause it to burn out after running awhile.

Calculation of Horse-Power Required for a Punch, and Width of Belt

We can determine the horse-power necessary to run a punch in the following manner: Take the case of a 1-inch diameter by ¾-inch punch, running 30 strokes per minute; we have

$E = 2{,}950$ foot-pounds energy per stroke,

Let $H.P. =$ horse-power,

$N =$ number of strokes per minute.

We then have

$$H.P. = \frac{E \times N}{33{,}000} \qquad (4)$$

$$= \frac{2{,}950 \times 30}{33{,}000}$$

$= 2.7$ H. P. for a single machine, or $2 \times 2.7 = 5.4$ H. P. for a double machine with both sides running continuously.

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A machine of this size would most likely be run with a single belt which would be considered to exert a pull of 40 pounds per inch width of belt. We will assume a certain diameter for the pulley, and figure the face to suit the required horse-power.

Let D = the diameter of the pulley in inches = 20 inches,
x = face in inches,
n = 175 R. P. M. of pulley,

$$H.P. = \frac{D \times \pi}{12} \times \frac{40 \times x \times n}{33{,}000}, \text{ and transposing we get}$$

$$x = \frac{H.P \times 12 \times 33{,}000}{D \times \pi \times 40 \times n} \text{ for single machine} \qquad (5)$$

$$= \frac{2.7 \times 12 \times 33{,}000}{20 \times \pi \times 40 \times 175} = 2.45 \text{ inches belt width,}$$

= say, 3 inches belt face of pulley for single punch.

For a double punch we would require twice the power, or assuming 30 inches diameter for the pulley and substituting in (5), we get

$$x = \frac{H.P. \times 12 \times 33{,}000}{D \times \pi \times 40 \times n}$$

$$= \frac{5.4 \times 12 \times 33{,}000}{30 \times \pi \times 40 \times 175} = 3.25 \text{ inches belt width.}$$

= say 3¾ inches belt face of pulley for double machine.

If these machines were to be motor driven, the single machine would require at least a 3-horse-power motor and the double machine from 5 to 7½ horse-power motor. A 5-horse-power motor would in all probability be all right, as a double machine would hardly be run so as to use every stroke. It is always best, however, to have a motor that is a little larger than is required, as punching is very severe work on the motor, especially when the motor is geared to the fly-wheel shaft through cut spur gears. The variation in speed jars the motor, and this tells on the windings, etc. The variation of the speed in the fly-wheel has less effect on the motor if it is belted, or if it is connected to the machine through a slip gear or a friction clutch.

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CHAPTER V

SIMPLIFIED METHODS FOR FLY-WHEEL CALCULATIONS

In the previous chapter the customary methods and formulas have been given relating to the design of fly-wheels and the size of motor required for giving out a certain amount of energy per stroke of the machine under consideration. In this chapter a method of calculation will be given, whereby the work of finding the desired results may be considerably shortened.

In shears of large size cutting short pieces, where the maximum effort may be required almost continuously, it is of great importance that motor and fly-wheel be of sufficient capacity to perform their work properly. Since the amount of energy to be given out by the fly-wheel depends upon the size of the motor, this should always be determined first. Let

E = total energy required per stroke,
E_1 = energy given up by motor during cut,
E_2 = energy given up by fly-wheel,
T = time in seconds per stroke,
T_1 = time in seconds in which E_1 is given up,
T_2 = time in seconds in which E_2 is restored to fly-wheel,
V_1 = initial velocity of fly-wheel in feet per second,
V_2 = velocity after cut in feet per second,
R_1 = initial revolutions per minute of fly-wheel,
R_2 = revolutions per minute after cut,
R_3 = revolutions per minute after n cuts,
W = weight of fly-wheel rim in pounds,
D = mean diameter of fly-wheel rim in feet,
H_1 = horse-power required to cut every stroke,
H_2 = horse-power actually used,
a = width of fly-wheel rim,
b = depth of fly-wheel rim,
g = 32.16,
n = number of cuts shear will make for a total given reduction in speed.

In the previous chapter this formula for the horse-power required was given:

$$H.P. = H_1 = \frac{EN}{33{,}000},$$

and since $N = \frac{60}{T}$ we have

$$H_1 = \frac{E}{550T} \qquad (1)$$

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$$H_1 = \frac{E_1}{550 T_1}$$

$$E_1 = 550\, T_1\, H_1 = \frac{550\, E\, T_1}{550\, T} = \frac{E\, T_1}{T}$$

$$E_2 = E - \frac{E\, T_1}{T} = E\left(1 - \frac{T_1}{T}\right) \qquad (2)$$

Having now the energy that must be given out by the fly-wheel, we can proceed as follows:

We know that $E_2 = \frac{W}{2g}\,(V_1^2 - V_2^2)$ and that

$$V_1^2 = \left(\frac{D \times \pi \times R_1}{60}\right)^2 = 0.00274\, D^2\, R_1^2$$

$$V_2^2 = \left(\frac{D \times \pi \times R_2}{60}\right)^2 = 0.00274\, D^2\, R_2^2$$

$$V_1^2 - V_2^2 = 0.00274\, D^2\, (R_1^2 - R_2^2)$$

$$E_2 = \frac{W}{64.32} \times 0.00274\, D^2\, (R_1^2 - R_2^2)$$

$$E_2 = 0.0000426\, W D^2\, (R_1^2 - R_2^2) \qquad (3)$$

$$W = \frac{E_2}{0.0000426\, D^2\, (R_1^2 - R_2^2)} \qquad (4)$$

Making $0.0000426\, (R_1^2 - R_2^2) = C R_1^2$ we have

$$E_2 = C W D^2 R_1^2 \qquad (5)$$

$$W = \frac{E_2}{C D^2 R_1^2} \qquad (6)$$

In cast iron fly-wheels it is usual not to exceed a speed which represents a fiber stress of more than 1,000 pounds per square inch of rim cross section. The stress in pounds due to centrifugal force equals $0.0972\, V_1^2$ for cast iron, and for fly-wheels having a maximum stress of 1,000 pounds per square inch, we can develop the following formulas:

$0.0972\, V_1^2 = 1{,}000;\ V_1 = 101.5.$

But $V_1 = \frac{D \pi R_1}{60}$, therefore we have

$$101.5 = \frac{D \pi R_1}{60},$$

$$R_1 = \frac{101.5 \times 60}{D \pi} = \frac{1{,}940}{D} \qquad (7)$$

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$$D = \frac{1,940}{R_1} \qquad (8)$$

Squaring (7) we have $R_1^2 = \frac{1,940^2}{D^2}$

Substituting this in (6) we have

$$W = \frac{E_2}{C D^2 \frac{1940^2}{D^2}} = \frac{E_2}{1940^2 C}$$

Making $1,940^2 C = C_1$, and $\frac{1}{C_1} = C_2$ we have

$$W = \frac{E_2}{C_1} = C_2 E_2 \qquad (9)$$

The following are the values of C, C_1, and C_2 for different reductions in speed:

Per cent. Reduction.	C	C_1	C_2
2½	0.00000213	8.00	0.1250
5	0.00000426	16.00	0.0625
7½	0.00000617	23.20	0.0432
10	0.00000810	30.45	0.0328
12½	0.00001000	37.60	0.0266
15	0.00001180	44.50	0.0225
20	0.00001535	57.70	0.0173

Size of Rim

Let us assume that the depth of rim equals 1.22 times the width. We have then these formulas for size of rim:

$$a = \sqrt{\frac{W}{12 D}} \qquad (10)$$

$$b = 1.22\, a \qquad (11)$$

These two formulas can be changed to suit any required ratio of depth to width of rim.

Let y = required ratio,

$$a = \sqrt{\frac{1.22\, W}{12\, Dy}} \qquad (12)$$

$$b = ya \qquad (13)$$

Effect of Changing Size of Motor

Let us now suppose that we do not wish to use a motor large enough to cut continuously, and desire to find how many cuts the machine would make continuously without drifting down more than a certain percentage of the original speed. Transposing (3) we have

$$R_1^2 - R_2^2 = \frac{E_2}{0.0000426\, WD^2}$$

Let $\frac{E_2}{0.0000426\, WD^2} = K.$

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$K = R_1^2 - R_2^2$, and

$R_2 = \sqrt{R_1^2 - K}$

$$R_3 = \sqrt{R_1^2 - nK + (n-1)K\frac{H_2}{H_1}} \qquad (14)$$

After several reductions we have

$$n = \frac{\dfrac{H_1(R_1^2 - R_3^2)}{K} - H_2}{H_1 - H_2}$$

and since $K = R_1^2 - R_2^2$ we have

$$n = \frac{\dfrac{H_1(R_1^2 - R_3^2)}{R_1^2 - R_2^2} - H_2}{H_1 - H_2} \qquad (15)$$

The time now required to bring the fly-wheel up to full speed again after n cuts will be

$$T_2 = \frac{E_2}{550\,H_2} \qquad (16)$$

Examples

We will now work out some examples illustrating the use of these formulas.

Example 1.—A hot slab shear is required to cut a slab 4 × 15 inches which, at a shearing stress of 6,000 pounds per square inch, gives a pressure between the knives of 360,000 pounds. The total energy required for the cut will then be $360{,}000 \times \frac{4}{12} = 120{,}000$ foot-pounds. The shear is to make 20 strokes per minute, and with a six-inch stroke the actual cutting time is 0.75 seconds, and the balance of the stroke is 2.25 seconds.

The fly-wheel is to have a mean diameter of 6 feet 6 inches and is to run at a speed of 200 R. P. M.; the reduction in speed to be 10 per cent per stroke when cutting.

$$H_1 = \frac{120{,}000}{3 \times 550} = 72.7 \text{ horse-power.}$$

$$E_2 = 120{,}000 \times \left(1 - \frac{0.75}{3}\right) = 90{,}000 \text{ foot-pounds.}$$

$$W = \frac{90{,}000}{0.0000081 \times 6.5^2 \times 200^2} = 6570 \text{ pounds.}$$

Assuming a ratio of 1.22 between depth and width of rim,

$$a = \sqrt{\frac{6{,}570}{12 \times 6.5}} = 9.18 \text{ inches,}$$

$b = 1.22 \times 9.18 = 11.2$ inches,

or size of rim, say, 9 × 11½ inches.

Example 2.—Suppose we wish to make the fly-wheel in Example 1 with a stress of 1,000 pounds, due to centrifugal force, per square inch of rim section.

C_2 for 10 per cent $= 0.0328$,

$W = 0.0328 \times 90{,}000 = 2{,}950$ pounds,

$$R_1 = \frac{1940}{D}. \text{ If } D = 6 \text{ ft., } R_1 = \frac{1940}{6} = 323 \text{ R. P. M.}$$

$$a = \sqrt{\frac{2950}{12 \times 6}} = 6.4 \text{ inches}$$

$b = 1.22 \times 6.4 = 7.8$ inches,

or size of rim, say, $6\frac{1}{4} \times 8$ inches.

Example 3.—Let us now suppose that in Example 1 we wish to use a 50 H. P. motor, and wish to find how many cuts the shear will make continuously without drifting down more than 20 per cent in speed? And what time must be allowed for the motor to restore the fly-wheel to its original speed?

$$R_1^2 - R_3^2 = 200^2 - 160^2 = 14400$$

$$R_1^2 - R_2^2 = 200^2 - 180^2 = 7600$$

$$n = \frac{\dfrac{72.7 \times 14400}{7600} - 50}{72.7 - 50} = 3.86 \text{ cuts}$$

Allowing the shear to make 4 cuts we have

$$R_3 = \sqrt{200^2 - 4 \times 7600 + 3 \times 7600 \times \frac{50}{72.7}} = 159 \text{ R. P. M.}$$

$$E_2 = 0.0000426 \times 6570 \times 6.5^2 \times (200^2 - 159^2) = 175{,}000 \text{ foot-pounds, about.}$$

$$T_2 = \frac{175000}{550 \times 50} = 6.4 \text{ seconds.}$$

Example 4.—Let us now suppose that in Example 2 we wish to use a 50 H. P. motor under the same conditions as in Example 3.

$$R_1^2 - R_3^2 = 323^2 - 258^2 = 37750$$

$$R_1^2 - R_2^2 = 323^2 - 291^2 = 19650$$

$$n = \frac{\dfrac{72.7 \times 37750}{19650} - 50}{72.7 - 50} = 4 \text{ cuts, nearly.}$$

$$E_2 = 0.0000426 \times 2950 \times 6^2 \times (323^2 - 258^2) = 170{,}000 \text{ foot-pounds, about.}$$

$$T_2 = \frac{170{,}000}{550 \times 50} = 6.2 \text{ seconds.}$$

These examples show the possibilities of the formulas as time-savers for the designer, by reducing the calculations to the smallest possible number, and at the same time reducing the possibility of error.

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CHAPTER VI

FLY-WHEELS FOR MOTOR-DRIVEN PLANERS

The question of motor drive for high-speed planing machines brings forward many interesting problems, among which the ascertaining of the correct dimensions for the flywheel is not the least. The primary function of a fly-wheel is here not so much the preservation of a constant speed as the relieving of the motor from excessive shock at the instant of reversal.

A shunt-wound motor tends to keep the same speed at all loads, but must necessarily slow down for a moment, however large the fly-wheel at the instant of reversal, thus tending to spark. Of course, the larger the motor, the greater the store of energy in the armature, consequently the smaller the drop in speed and less tendency to sparking. A compound-wound motor, on the other hand, will drop slightly in speed under heavy loads, the percentage of drop, of course, depending upon the amount of compounding. It is this property of the compound motor which enables the fly-wheel to perform its work satisfactorily. A correctly designed fly-wheel will, at the moment of reversal, keep up the speed of the motor slightly higher than that corresponding to the load on the motor at that instant, thus eliminating all possibility of sparking.

Now the determining of the dimensions of the fly-wheel before the machine is made, to fulfill these conditions, necessitates close scrutiny of the engineering press, so as to be continually cognizant of tests taken at different times on high-speed planing machines. A better method, where practicable, is to test the machine before deciding upon either the motor or the fly-wheel.

A machine recently tested under the latter condition gave the following: Average horse-power, cutting, 19; average horse-power, backing, 11. At the instant of reversal to backing stroke the ammeter needle jumped to 190 on a 220-volt circuit, showing maximum horse-power to be about 55, and the time taken up from the table striking the dog to the attainment of maximum backing speed was 3 seconds. It was decided to drive this machine by a 30 B. H. P. motor at 500 revolutions per minute, compounded so as to give a maximum variation of about 12 or 14 per cent. Allowing a 40 per cent momentary overload on the motor would bring the maximum horse-power allowable on reversal to 42, and the additional 13 horse-power would have to be supplied by the fly-wheel. The dimensions of the wheel were obtained in the following manner:

As energy in a moving body varies directly as V^2, where V = velocity in feet per second, it is clear that the best place for the fly-wheel is upon the shaft having the greatest number of revolutions per minute, which, of course, is the motor shaft. From the figures given,

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it will be seen that the wheel must be capable of parting with sufficient energy to develop 13 horse-power during the time of reversal, *viz.*, 3 seconds, and its drop in speed must not exceed 10 per cent, so as to keep the actual variation slightly below that allowed by the motor.

$$\text{Energy to be given out by the fly-wheel} = \frac{13 \times 33{,}000 \times 3}{60}.$$

Now assume M to be the store of energy in foot-pounds in this fly-wheel when it makes one revolution per minute; then, as the energy varies as V^2, and V varies as the revolutions per minute, the store of energy in the wheel when making 500 revolutions per minute = $M \times 500^2$.

As the drop of speed of the wheel = 10 per cent of speed of wheel, the speed of the wheel at the end of three seconds = 500 — 10 per cent of 500 = 450 revolutions per minute, and the store of energy then in the wheel = $M \times 450^2$.

Thus the energy given up by the wheel in being reduced from 500 to 450 revolutions per minute = $M\ (500^2 - 450^2)$.

But the energy given up must $= \dfrac{13 \times 33{,}000 \times 3}{60}$, as already shown;

$$\text{therefore } M\ (500^2 - 450^2) = \frac{13 \times 33{,}000 \times 3}{60}.$$

$$M = \frac{13 \times 33{,}000 \times 3}{60\ (500^2 - 450^2)}$$

$$= \frac{13 \times 33{,}000 \times 3}{60 \times 47{,}500}.$$

$$M = 0.45 \text{ foot-pounds.}$$

Therefore, the store of energy in the fly-wheel when making 1 revolution per minute = 0.45 foot-pounds.

The limit of peripheral speed of plate fly-wheels generally allowed in machine tool practice is about 7,000 feet per minute, which quantity will enable us to find the outside diameter of the wheel thus:

$$\frac{7{,}000}{3.1416 \times 500} = 4.4 \text{ feet.}$$

As the energy in a revolving wheel $= \dfrac{W V^2}{2g}$, where V = velocity in feet per second at center of area of rim, $\dfrac{W V^2}{2g}$ must equal 0.45 when the wheel makes 1 revolution per minute. Therefore $\dfrac{W V^2}{2 \times 32.2} = 0.45$.

Velocity of wheel in feet per second when wheel makes 1 revolution

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per minute $= \dfrac{4 \times 3.1416}{60}$; the diameter of the wheel to center of area, it will be seen, is taken as 4 feet, 4.4 feet being the outside diameter; thus

$$\frac{W \times 4 \times 4 \times 3.1416 \times 3.1416}{2 \times 32.2 \times 60 \times 60} = 0.45.$$

$$W = \frac{0.45 \times 2 \times 32.2 \times 60 \times 60}{4 \times 4 \times 3.1416 \times 3.1416};\ W = 660 \text{ pounds}.$$

Thus, knowing the outside diameter and the weight of the wheel, the other dimensions are very easily ascertained.

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