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MEASURING TOOLS

CALIPERS—DIVIDERS—SURFACE GAGES—
MICROMETER MEASURING INSTRUMENTS

THIRD EDITION

MACHINERY'S REFERENCE BOOK NO. 21
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MEASURING TOOLS

THIRD EDITION

CONTENTS

CHAPTER I

HISTORY AND DEVELOPMENT OF STANDARD MEASUREMENTS*

While every mechanic makes use of the standards of length every day, and uses tools graduated according to accepted standards when performing even the smallest operation in the shop, there are comparatively few who know the history of the development of the standard measurements of length, or are familiar with the methods employed in transferring the measurements from the reference standard to the working standards. We shall therefore here give a short review of the history and development of standard measurements of length, as abstracted from a paper read by Mr. W. A. Viall before the Providence Association of Mechanical Engineers.

Origin of Standard Measurements

By examining the ruins of the ancients it has been found that they had standard measurements, not in the sense in which we are now to consider them, but the ruins show that the buildings were constructed according to some regular unit. In many, if not all cases, the unit seems to be some part of the human body. The "foot," it is thought, first appeared in Greece, and the standard was traditionally said to have been received from the foot of Hercules, and a later tradition has it that Charlemagne established the measurement of his own foot as the standard for his country.

Standards Previous to 1800

In England, prior to the conquest, the yard measured, according to later investigations, 39.6 inches, but it was reduced by Henry I in 1101, to compare with the measurement of his own arm. In 1324, under Edward II, it was enacted that "the inch shall have length of three barley corns, round and dry, laid end to end; twelve inches shall make one foot, and three feet one yard." While this standard for measurement was the accepted one, scientists were at work on a plan to establish a standard for length that could be recovered if lost, and Huygens, a noted philosopher and scientist of his day, suggested that the pendulum, which beats according to its length, should be used to establish the units of measurement. In 1758 Parliament appointed a commission to investigate and compare the various standards with that furnished by the Royal Society. The commission caused a copy of this standard to be made, marked it "Standard Yard, 1758," and laid it before the House of Commons. In 1742, members of the Royal Society

* MACHINERY, October, 1897.

of England and the Royal Academy of Science of Paris agreed to exchange standards, and two bars 42 inches long, with three feet marked off upon them, were sent to Paris, and one of these was returned later with "Toise" marked upon it. In 1760 a yard bar was prepared by Mr. Bird, which was afterwards adopted as a standard, as we shall see later.

In 1774 the Royal Society offered a reward of a hundred guineas for a method that would obtain an invariable standard, and Halton proposed a pendulum with a moving weight upon it, so that by counting the beats when the weight was in one position and again when in another, and then measuring the distance between the two positions, a distance could be defined that could at any time be duplicated. The Society paid 30 guineas for the suggestion, and later the work was taken up by J. Whitehurst with the result that the distance between the positions of the weight when vibrating 42 and 84 times a minute was 59.89358 inches. The method was not further developed.

How the Length of the Meter was Established

In 1790, Talleyrand, then Bishop of Autun, suggested to the Constituent Assembly that the king should endeavor to have the king of England request his parliament to appoint a commission to work in unison with one to be appointed in France, the same to be composed of members of the Royal Society and Royal Academy of Science, respectively, to determine the length of a pendulum beating seconds of time. England did not respond to the invitation, and the French commission appointed considered first of all whether the pendulum beating seconds of time, the quadrant of the meridian, or the quadrant of the equator should be determined as a source of the standard. It was decided that the quadrant of the meridian should be adopted and that 0.0000001 of it should be the standard.

The arc of about nine and one-half degrees, extending from Dunkirk on the English Channel to Barcelona on the Mediterranean and passing through Paris, should be the one to be measured. The actual work of measuring was done by Mechain and Delambre according to the plans laid down by the commission. Mechain was to measure about 25 per cent of the arc, the southern portion of it, and Delambre the remainder; the reason for this unequal division was that the northern division had been surveyed previously, and the territory was well-known, whereas the southern part was an unknown country, as far as the measurement of it went, and it was expected that many severe difficulties would have to be surmounted. The Revolution was in progress, and it was soon found that the perils attending the measurement of the northern part were greater than those attending the southern part of the territory. The people looked askance at all things that they did not understand, and Delambre with his instruments was looked upon as one sent to further enthrall them. He was set upon by the people at various times and although the authorities endeavored to protect him, it was only by his own bravery and tact that he was able to do his work and save his life. The Committee of Safety ordered that Mechain and Delambre close their work in 1795, and it was some time afterward before it was resumed.

Having completed the field work, the results of their labors were laid before a commission composed of members of the National Institute and learned men from other nations, who had accepted the invitation that had been extended to them, and after carefully reviewing and calculating the work, the length of the meridian was determined, and from it was established the meter as we now have it. A platinum bar was made according to the figures given, and this furnishes the prototype of the meter of the present time. Notwithstanding all of the care taken in establishing the meter, from work done by Gen. Schubert, of Russia, and Capt. Clarke, of England, it has been shown that it is not 0.0000001 of the quadrant passing through Paris, but of the one passing through New York.

The Standard Yard in England—Its Loss and Restoration

Whether incited by the work of the French or not, we do not know, but in the early part of this century the English began to do more work upon the establishment of a standard, and in 1816 a commission was appointed by the crown to examine and report upon the standard of length. Capt. Kater made a long series of careful observations determining the second pendulum to be 39.1386 inches when reduced to the level of the sea. This measurement was made on a scale made by Troughton—who, by the way, was the first to introduce the use of the microscope in making measurements—under the direction of and for Sir Geo. Schuckburgh. In 1822, having made three reports, after many tests, it was recommended that the standard prepared by Bird in 1760, marked "Standard Yard, 1760," be adopted as the standard for Great Britain.

The act of June, 1824, after declaring that this measure should be adopted as the standard, reads in Sec. III.: "And whereas it is expedient that the Standard Yard, if lost, destroyed, defaced or otherwise injured should be restored to the same length by reference to some invariable natural Standard; and whereas it has been ascertained by the Commissioners appointed by His Majesty to inquire into the Subjects of Weights and Measures, that the Yard, hereby declared to be the Imperial Standard Yard, when compared with a Pendulum vibrating Seconds of Mean Time in the latitude of London, in a Vacuum at the Level of the Sea, is in the proportion of Thirty-six Inches to Thirty-nine Inches and one thousand three hundred and ninety-three ten thousandth parts of an Inch; Be it enacted and declared, that if at any Time hereafter the said Imperial Standard Yard shall be lost, or shall be in any manner destroyed, defaced or otherwise injured, it shall and may be restored by making a new Standard Yard bearing the same proportion to such Pendulum, as aforesaid, as the said Imperial Standard Yard bears to such Pendulum."

It was not long after this act had been passed, if indeed not before, that it became known that the pendulum method was an incorrect one, as it was found that errors had occurred in reducing the length obtained to that at the sea level, and despite the great pains that had been taken, it is doubtful if the method was not faulty in some of its other details.

When the Houses of Parliament were burned in 1834, an opportunity was offered to try the method upon which so much time and care had been spent. A commission was appointed and to Sir Francis Baily was assigned the task of restoring the standard. He did not live to complete the task, dying in 1844. He succeeded in determining the composition of the metal that was best adapted to be used, which metal is now known as Baily's metal.

Rev. R. Sheepshanks constructed a working model as a standard and compared it with two Schuckburg's scales, the yard of the Royal Society, and two iron bars that had been used in the ordnance department. Having determined to his own satisfaction and that of his associates the value of the yard, he prepared the standard imperial yard, known as Bronze No. 1, a bronze bar 38 × 1 × 1 inch, with two gold plugs dropped into holes so that the surface of the plugs passes through the center plane of the bar. Upon these plugs are three transverse lines and two longitudinal lines, the yard being the distance from the middle transverse line—the portion lying between the two longitudinal ones—of one plug, to the corresponding line on the other plug. Forty copies were made, but two of these being correct at 62 degrees Fahrenheit, and these two, together with the original and one other, are kept in England as the standards for reference. In 1855 the standard as made by Rev. Sheepshanks was legalized.

Attempts to Fix a Standard in the United States

The Constitution empowers Congress to fix the standards of weights and measures, but up to 1866 no legal standard length had been adopted. In his first message to Congress Washington said: "A uniformity in the weights and measures of the country is among the important objects submitted to you by the Constitution, and if it can be derived from a standard at once invariable and universal, it must be no less honorable to the public council than conducive to the public convenience."

In July, 1790, Thomas Jefferson, then Secretary of State, sent a report to Congress containing two plans, both based on the length of the pendulum, in this case the pendulum to be a plain bar, the one plan to use the system then existing, referring it to the pendulum as the basis, and the other to take the pendulum and subdivide it, one-third of the pendulum to be called a foot. The whole length was that of one beating seconds of time. He made a table to read as follows:

10 Points make a Line.	10 Decads make a Rood.
10 Lines make a Foot.	10 Roods make a Furlong.
10 Feet make a Decad.	10 Furlongs make a Mile.

Congress did not adopt his system, and as England was then working on the problem, it was decided to await the results of its labors. In 1816, Madison, in his inaugural address, brought the matter of standards to the attention of Congress, and a committee of the House made a report recommending the first plan of Jefferson, but the report was not acted upon. In 1821, J. Q. Adams, then Secretary of State, made a long and exhaustive report in which he favored the metric system, but still advised Congress to wait, and Congress—waited.

What the Standards are in the United States

The standard of length which had generally been accepted as *the* standard, was a brass scale 82 inches long, prepared by Troughton for the Coast Survey of the United States. The yard used was the 36 inches between the 27th and 63d inch of the scale. In 1856, however "Bronze No. 11" was presented to the United States by the British government. This is a duplicate of the No. 1 Bronze mentioned before, which is the legalized standard yard in England. It is standard length at 61.79 degrees F., and is the accepted standard in the United States. A bar of Low Moor iron, No. 57, was sent at the same time, and this is correct in length at 62.58 degrees F. The expansion of Bronze No. 11 is 0.000342 inch, and that of the iron bar is 0.000221 inch for each degree Fahrenheit. While the yard is the commonly accepted standard in this country, it is not the legal standard. In 1866 Congress passed a law making legal the meter, the first and only measure of length that has been legalized by our government. Copies of the meter and kilogram, taken from the original platinum bar at Paris, referred to before, were received in this country by the President and members of the Cabinet, on Jan. 2, 1890, and were deposited with the Coast Survey. By formal order of the Secretary of the Treasury, April 5, 1893, these were denominated the "Fundamental Standards."

The International Bureau of Weights and Measures

After the original meter was established, it was found that copies made by various countries differed to a greater or less extent from the original, and believing that a copy could be made from which other copies could be more readily made than from the end piece meter, and that better provision could be made for the preservation of the standard, France called a convention of representatives from various States using the system, to consider the matter. The United States representatives, or commissioners, were Messrs. Henry and Hildegard, who met with the general commission in 1870. The commissioners at once set at work to solve the problem presented to them, but the Franco-Prussian war put an end to their deliberations. The deliberations were resumed later, and May 20, 1875, representatives of the various countries signed a treaty providing for the establishment and maintenance, at the common expense of the contracting nations, of a "scientific and permanent international bureau of weights and measures, the location of which should be Paris, to be conducted by a general conference for weights and measures, to be composed of the delegates of all the contracting governments."

This bureau is empowered to construct and preserve the international standards, to distribute copies of the same to the several countries, and also to discuss and initiate measures necessary for the determination of the metric system. The commission adopted a form for the standard as shown in Fig. 1. The lines representing the length of the meter are drawn on the plane *A*, which is the neutral plane, and will not change in length should the bar deflect. The bar is made of 90 per cent platinum and 10 per cent iridium, about 250 kilograms having been melted when preparations were made for the first standard, so that all of the copies made from this cast represent the same coefficient of expansion and are subject to the same changes as the original. The French government presented to the bureau the pavilion Breteuil, opposite the Park of St. Cloud, which was accepted and put into order and is now the repository of the originals of the meter and the kilogram. The expense attending the first establishment of the bureau was about $10,000 to the United States, and since then its share of the annual expense has been about $900. The standards in the possession of the United States were received through the international bureau.

The Commercial Value of a Standard

Having at the disposal of the nation a standard of length, the question arises, "What can be made of it commercially, and how do we know when we have a copy of the standard?"

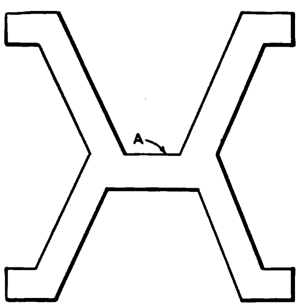


Fig. 1. Form of Bar Adopted for International Standards of Length

In 1893, the Brown & Sharpe Mfg. Co. decided to make a new standard to replace the one they had at that date. Mr. O. J. Beale was detailed to do this work. He prepared steel bars about 40 inches long by 1¼ inch square, and after planing them, they were allowed to rest for several months. At the ends of these bars he inserted two gold plugs, the centers of which were about 36 inches apart, and a little beyond these two others about one meter apart. A bar was placed in position upon a heavy bed. This was so arranged that a tool carrier could be passed over the bar. The tool carrier consisted of a light framework, holding the marking tool. One feature of the marking was that the point of the marking tool was curved and had an angle, so that if dropped it made an impression in the form of an ellipse. In graduations, ordinarily, the line, when highly magnified, is apt to present at its ends an impression less definite than in the center, by reason of the form of the objective. The line made with the tool mentioned is short, and that portion of the line is read which passes, apparently, through the straight line in the eye-glass of the microscope. In order to make these lines as definite as possible, the point was lapped to a bright surface. After being placed in position, the microscope, which could be placed on the front of the tool carrier, was set to compare with the graduation on the standard bar from which the new bar was to be prepared. After such a setting the readings were made by three persons, and by turning the lever the marking tool was dropped, making a very fine line, so fine indeed, that when the authorities in Washington began the examination of the bar later on they declared that no line had been made upon these studs.

After making the first line, the carriage was moved along to compare with the other line on the standard, and after the correction had been made by the use of the micrometer in the microscope, the marking tool was again dropped, giving the second line, which was intended to mark the limit of one yard over-all. The same operation was repeated in the marking of the meter. The whole of this work was done, of course, with the greatest care, and, while the theoretical portion of it appears very simple in detail, it required a great deal of time and patience before the last line had been made. The bar thus marked was taken to Washington, and in Mr. Beale's presence was compared by the attendants with Bronze No. 11 and later with Low Moor bar, No. 57.

In comparing this standard, a method was employed very similar to that used in marking it. The bar, properly supported, was placed upon a box that rested upon rolls, and on this same box was placed the government standard with which the Brown & Sharpe standard was to be compared. The standard was placed in position under the microscope, and after being properly set to the standard, the bar to be measured was placed under the microscope, and by the micrometer screw of the microscope the variation was measured. Three comparisons were made by each of the attendants on each end before determining the reading of the microscope, and after such comparisons and many repetitions of it, the value of the standard No. 2 was found to be 36.00061 inches for the yard, and 1.0000147 meter for the meter

After this work had been done, Mr. Beale prepared a second standard which he called No. 3, and after examining, as shown above, the error was found to be 0.00002 inch for the yard, and 0.000005 meter for the meter. Observing these variations as compared with the standards originally made, we find they are very close, and it is doubtful if many repeated trials would furnish more accurate work, when we remember that out of forty original standards made, but two are correct at 62 degrees Fahrenheit.

After establishing a yard, the problem of obtaining an inch comes next, and this was made by subdividing the yard into two equal parts, these into three, and the three further subdivided into six parts. It should be particularly noted that no mention has been made of a standard inch, as there is none, the standard yard only existing, the subdivision of which falls upon those undertaking standard work. There is a remarkable agreement between at least three leading gage makers of this country and abroad, and each came to the result by its own method of subdividing the standard yard.

Kinds of Measurements and Measuring Tools

The measurements in the shop may, in general, be divided into measurements of length and measurements of angles. The length measurements in turn may be divided into line measurements and end measurements, the former being made by placing a rule or similar instrument against the object being measured, and comparing its length with the graduations on the measuring instruments; the latter are made by comparing the object being measured with the measuring instrument, by bringing the object measured into actual contact with the measuring surfaces of the instrument. Examples of line measurements are the ordinary measurements made with the machinist's rule, and examples of end measurement are those made by the micrometer, measuring machines, and snap gages. Angular measurements can also be divided into two classes; those measured directly by graduations on the instrument, and those measured by comparison with a given angle of the instrument.

Measuring instruments may also be divided into two classes, according to whether they actually are used for measuring, or whether they are principally used for comparing objects with one another. According to this classification all kinds of rules and protractors belong to the first class, whereas all gages belong to the second class. The ordinary instruments for length measurements, the regular machinists' rule, the caliper square, and the ordinary micrometer caliper, are too well known to require any additional explanation. The same is true of the regular bevel protractor for measuring angles. We shall therefore in the following chapters deal principally with special measuring tools, and with such methods in the use of tools which are likely to suggest improvements, or otherwise be valuable to the user and maker of measuring tools.

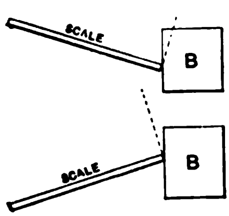
CHAPTER II

CALIPERS, DIVIDERS, AND SURFACE GAGES

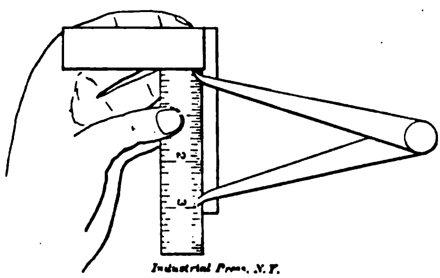
In the present chapter we shall deal with the simpler forms of tools used for measuring, such as ordinary calipers, and their use; surface gages; special attachments for scales and squares, facilitating accurate measuring; and vernier and beam calipers. The descriptions of the tools and methods referred to have appeared in MACHINERY from time to time. The names of the persons who originally contributed these descriptions have been stated in notes at the foot of the pages, together with the month and year when their contribution appeared.

Setting Inside Calipers

It is customary with most machinists, when setting inside calipers to a scale, to place one end of the scale squarely against the face of



Figs. 2 and 3



Industrial Press, N.Y.

Fig. 4

Setting Inside Calipers

some true surface, and then, placing one leg of the caliper against the same surface, to set the other leg to the required measurement on the scale. For this purpose the faceplate of the lathe is frequently used on account of its being close at hand for the latheman. The sides of the jaws of a vise or almost anything located where the light is sufficient to read the markings on the scale are frequently used.

The disadvantages of this method are, first, that a rough or untrue object is often chosen, particularly if it happens to be in a better light than a smooth and true one, and, second, that it is very hard to hold the scale squarely against an object. It is easy enough to hold it squarely crosswise, but it is not so easy a matter to keep it square edgewise. As can be readily seen, this makes quite a difference with the reading of the calipers, particularly if the scale is a thick one.

Figs. 2 and 3 show this effect exaggerated. *B* is the block against which the scale abuts. The dotted line indicates where the caliper leg should rest, but cannot do so, unless the scale is held perfectly square with the block. Fig. 4 shows a method of setting the calipers by using a small square to abut the scale and to afford a surface against which to place the leg of the caliper. The scale, lying flat on the blade of the square, is always sure to be square edgewise, and is easily held squarely against the stock of the square as shown. This method has also the advantage of being portable, and can be taken to the window or to any place where the light is satisfactory. When using a long scale, the free end may be held against the body to assist in holding it in place.*

Shoulder Calipers

In Fig. 5 are shown a pair of calipers which are very handy in measuring work from shoulder to shoulder or from a shoulder to the end of the piece of work. For this purpose they are much handier, and more accurate, than the ordinary "hermaphrodites." The legs

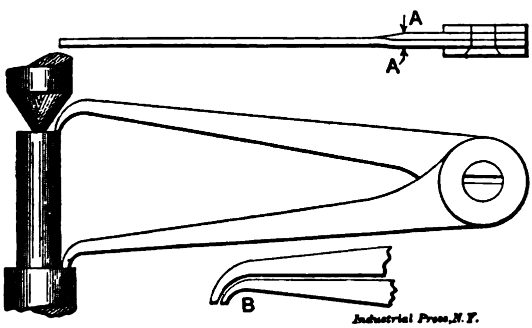


Fig. 5. Shoulder Calipers

are bent at *AA* so as to lie flat and thus bring the point of the long leg directly behind the short one which "nests" into it, as at *B*, so that the calipers may be used for short measurements as well as for long ones.

Double-jointed Calipers to Fold in Tool Box

In Fig. 6 are illustrated a pair of large calipers that can be folded up and put in a machinist's ordinary size tool chest. The usual large caliper supplied by the average machine shop is so cumbersome and heavy that this one was designed to fill its place. It can be carried in the chest when the usual style of large caliper cannot. It is a very light and compact tool. It is a 26-inch caliper, and will caliper up to 34 inches diameter. The top sections are made in four pieces, and the point ends fit between the top half like the blade of a knife, as shown in the engraving. Each side of the upper or top section is made of saw steel 1/16 inch thick, and the lower part or point of steel 1/8 inch thick. The double section makes the tool very stiff and light.

The point section has a tongue *A*, extending between the double section, which is engaged by a sliding stud and thumb nut. The stud

* M. H. Ball, April, 1902.

is a nice sliding fit in the slot, and the thumb nut clamps it firmly in place when in use. *B*, in the figure, shows the construction of the thumb nut. *C* is a sheet copper liner put between the washers at *A*. The dotted lines in the engraving show the points folded back to close up. The large joint washers are 1¾ inch diameter, and a ⅝-inch pin with a ⅝-inch hexagon head screw tightens it up. The forward joints are the same style, but smaller. The main joint has two 1¾-inch brass

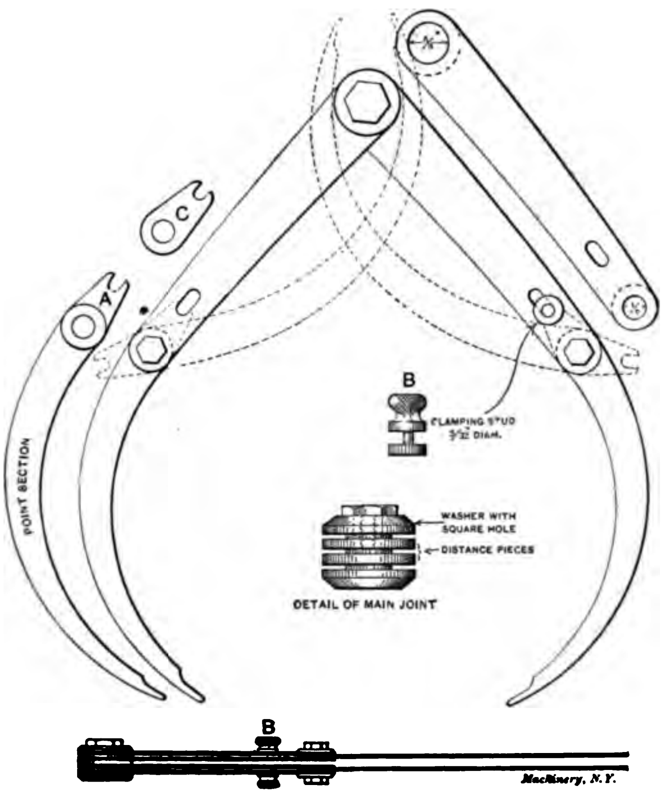


Fig. 6. Large Double-jointed Calipers

distance pieces or washers between the two main washers. The top section is 12½ inches between centers, and the point sections 15 inches from center to point. Closed up, the calipers measure 16 inches over-all.

Kinks in Inside Calipering

Close measurements may be made by filing two notches in each leg of an inside caliper so as to leave a rounded projection between, as shown at *E*, Fig. 7. Then, with an outside caliper, *D*, the setting of the inside caliper, *B*, is taken from the rounded points. The inside caliper can be reset very accurately after removal by this method. A still better way is to have two short pins, *CC'* set in the sides of the inside caliper legs, but this is not readily done as a makeshift. To measure the inside diameter of a bore having a shoulder like the piece *H*, the inside caliper *F* may also be set as usual and then a line marked with a sharp scriber on one leg, by drawing it along the side *G*. Then the legs are closed to remove the caliper, and are reset to the scribed line. Of course, this method is not as accurate as the previous one, and can be used only for approximate measurements.

To get the thickness of a wall beyond a shoulder, as at *K*, Fig. 7, set the caliper so that the legs will pass over the shoulder freely, and

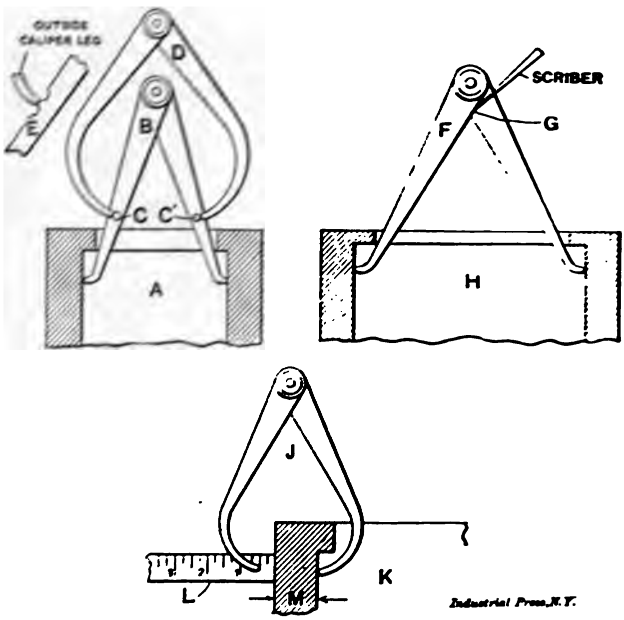


Fig. 7. Methods of Inside Calipering

with a scale measure the distance between the outside leg and the outside of the piece. Then remove the caliper and measure the distance between the caliper points. The difference between these two distances will be the thickness *M*.

Inside Calipers for Close Spaces

In Fig. 8 are shown a pair of inside calipers which are bent so as to be well adapted for calipering distances difficult of access, such as the keyway in a shaft and hub which does not extend beyond the hub, as indicated. With the ordinary inside calipers, having straight legs, and which are commonly used for inside work, it is generally impossible to get the exact size, as the end which is held in the hand comes in contact with the shaft before both points come into the same vertical plane. The engraving plainly shows how calipers for this purpose are made, and how used. Any mechanic can easily bend a common pair to about the shape shown to accommodate this class of work.*

Surface Gage with Two Pointers

Figs. 9 and 10 show a special surface gage, and illustrate an original idea which has been found to be a great saver of time and of milling

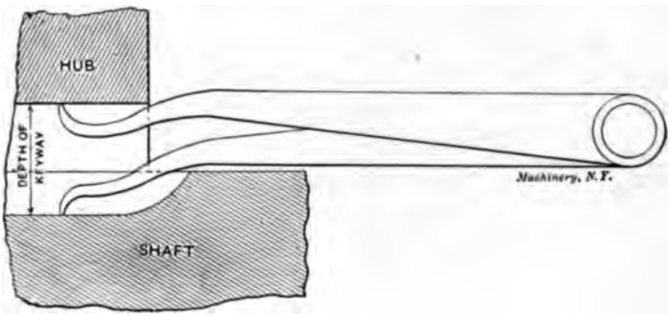
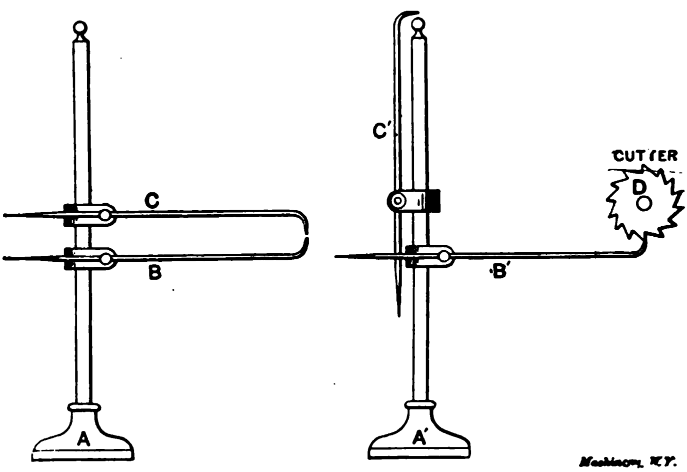


Fig. 8. Inside Calipers for Close Spaces

cutters. It can also be used on the planer or shaper. By its use the operator can raise the milling machine table to the right height without testing the cut two or three times, and eliminate the danger of taking a cut that is liable to break the cutter. This tool is especially valuable



Figs. 9 and 10. Surface Gage with Two Pointers

on castings, as raising the table and allowing the cutter to revolve in the gritty surface while finding the lowest spot is very disastrous to the cutting edges.

To use this surface gage, the pointer marked *C* in Fig. 9 is set to the lowest spot in the casting, and then the pointer *B* is set from it with perhaps 1/32 inch between the points for a cut sufficient to clean up the

* M. H. Ball, February, 1901.

surface. Pointer *C* is then folded up as shown at *C'* in Fig. 10, and the table is raised until the pointer *B* will just touch the under side of the cutter as shown at *B'* in Fig. 10. In this way the table is quickly

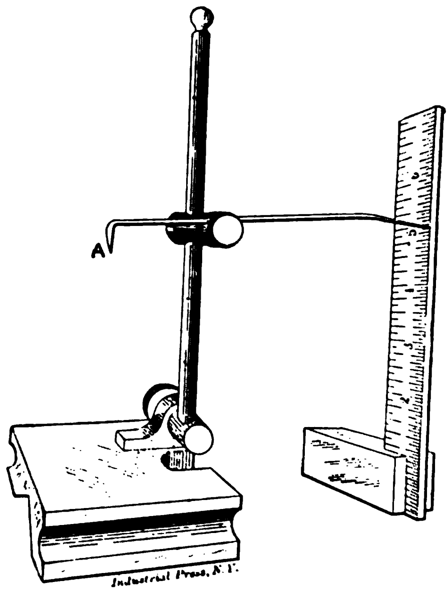


Fig. 11. Method of Adjusting the Needle of a Surface Gage

adjusted to a cut that will clean the casting or other piece being machined, and with no cutting or trying whatever.*

To Adjust the Needle of a Surface Gage

Fig. 11 illustrates a method of adjusting the needle of a surface gage. To set the gage 3¾ inches from the table, get somewhere within ¼

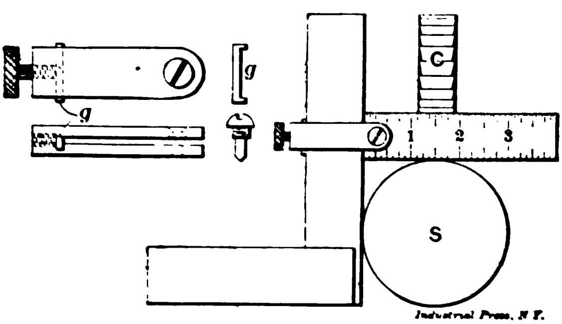


Fig. 12. Scale Attachment for the Square

inch of the mark on the square. With the thumb and forefinger on hook *A*, turn the needle till it reaches the point desired. By turning

* Harry Ash April, 1900.

the needle, it will travel in a circular path, on account of the bend near the point, and thus reach the desired setting.

Scale Attachment for the Square

Fig. 12 shows a device for attaching a scale to a square. This combination makes a very convenient tool to use when setting up work for keyseating, as is illustrated in the engraving, in which *S* is the shaft to be splined and *C* the milling cutter. It is also a very handy tool for truing up work on the boring mill or lathe. At the upper left-hand corner is shown the construction of the parts, which are made of dimensions to suit the size of the scale and the square. For the combination to be successful, it is essential that the blade of the square is the same thickness as the scale.*

Attachment for Machinist's Scale

Fig. 13 shows a very convenient appliance. It will be found very useful in the machine shop for setting inside calipers to any desired

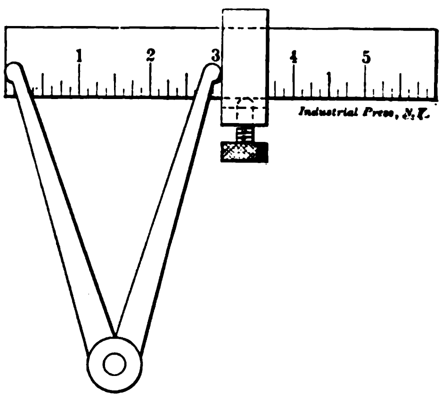


Fig. 13. Convenient Attachment for Machinist's Scale

size. The gage is clamped over the rule wherever desired, and one leg of the calipers set against the gage, the other leg being brought flush with the end of the scale.**

Setting Dividers Accurately

To set dividers accurately, take a 1-inch micrometer and cut a line entirely around the thimble as at *A*, Fig. 14, and then, with the instrument set at zero, make a punch mark *B* exactly one inch from the line on the thimble. If less than one inch is wanted, open out the micrometers and set the dividers to the dot and line so as to give one inch more than the distance wanted. Now with the dividers make two marks across a line, as at *a* and *b*, Fig. 14, and then set the dividers to one inch and mark another line as at *c*. The distance from *c* to *b* is the amount desired, and the dividers can be set to it. Great care must, of course, be exercised, if accurate results are required.

* M. H. Ball, March, 1903.
** Ezra F. Landis, May, 1902.

Combination Caliper and Divider

The combination caliper and divider shown in Fig. 15 is one that is not manufactured by any of the various tool companies. It is, however, one of the handiest tools that can be in a machinist's kit, as it lends itself to so many varied uses, and often is capable of being used where only a special tool can be employed. The illustration suggests its usefulness. The tool can be used as an outside caliper, as an inside

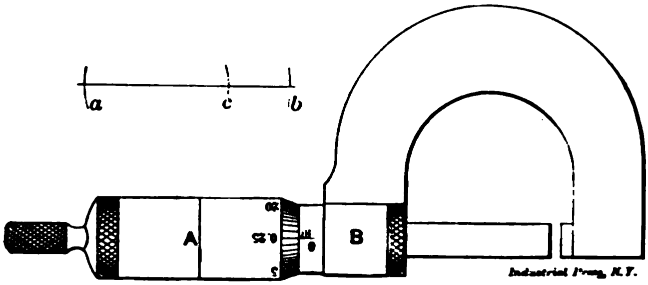


Fig. 14. Method of Setting Dividers Accurately

caliper, and as a divider. The common form of this tool has generally only one toe on the caliper legs, but the double toes save the reversal of the points when changing from outside to inside work. The divider points may be set at an angle, which permits of stepping off readily around the outside of a shaft at angular distances, where the ordinary dividers are useless. A number of other uses could be mentioned, but any intelligent mechanic can readily suggest them for himself.

Attachment for Vernier Calipers

While vernier and slide calipers are very handy shop tools, their usefulness is much more limited than it ought to be for such expensive

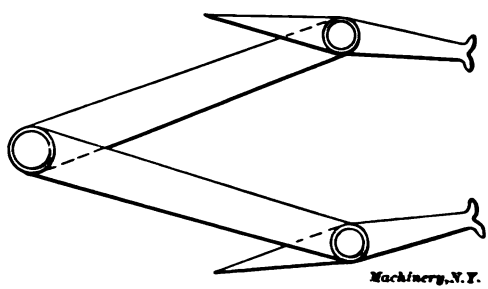


Fig. 15. Combination Caliper and Divider

instruments. In order to increase the usefulness of these tools, the attachments shown in Fig. 16 may be made. In the upper left-hand part of the engraving the details of a useful addition to the caliper are shown. *A* is made of machine steel, while the tongue *B* is of tool steel, hardened and ground and lapped to a thickness of 0.150 inch, the top and bottom being absolutely parallel. This tongue is secured to *A* by the two rivets *CC*. The thumb-screw *D* is used for fastening the attachment to the sliding jaw of the vernier or slide caliper. In the upper part of the engraving is shown the base, which is of machine steel, with the slot *F* milled for the reception of the fixed jaw of the caliper. The set-screws *GGG* are put in at a slight angle so that the caliper will be held firmly and squarely in this base. In the figure to the left these pieces are shown in the position for forming a height gage, for which purpose the attachment is most commonly used. As a test of the accuracy of its construction when the attachment is placed in this position, the tongue *B* should make a perfect joint with the fixed jaw of the caliper, and the vernier should give a reading of exactly 0.150. When it is desirable that the tongue *B* should over-

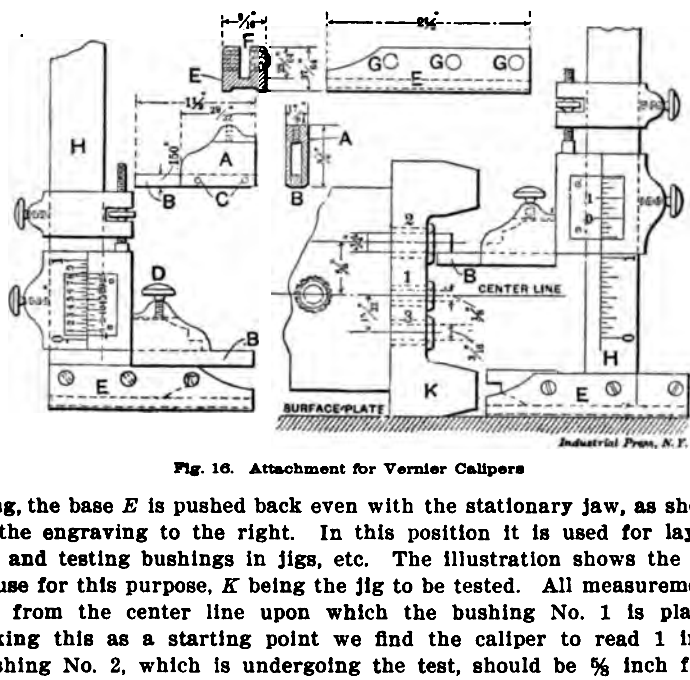


Fig. 16. Attachment for Vernier Calipers

hang, the base *E* is pushed back even with the stationary jaw, as shown in the engraving to the right. In this position it is used for laying out and testing bushings in jigs, etc. The illustration shows the tool in use for this purpose, *K* being the jig to be tested. All measurements are from the center line upon which the bushing No. 1 is placed. Taking this as a starting point we find the caliper to read 1 inch. Bushing No. 2, which is undergoing the test, should be ⅝ inch from this center line. It has a ¼-inch hole, and we therefore insert a plug of this diameter. Now adjust the tongue of the caliper to the bottom of this plug (as shown in the engraving) and the vernier should read 1.625 minus one-half the diameter of the plug, or 1.500, and any variation from this will show the error of the jig. In this case the top surface of *B* was used and no allowance had to be made for its thickness. In case the bottom surface is used, 0.150 must be deducted from the reading of the caliper.

It is very easy to make a mistake in setting a bushing, and such a mistake is equally hard to detect unless some such means of measuring as this is at hand. It often happens that jigs and fixtures are put into use containing such errors, and the trouble is not discovered until many dollars' worth of work has been finished and found worthless. The illustration shows but one of the many uses to which this attachment may be applied. The figures given on the details are correct for making an attachment to be used upon the Brown & Sharpe vernier caliper, but for other calipers they would, of course, have to be altered to suit.*

Improved Micrometer Beam Caliper

In a beam caliper having a sliding micrometer jaw with or without a separate clamping slide, it is necessary to have the beam divided into unit spaces, at which the jaw or slide may be accurately fixed, the micrometer screw then being used to cover the distance between the divisions; but it is difficult to construct a beam caliper of this type with holes for a taper setting pin, at exactly equal distances apart; consequently a plan that is generally followed in making such tools

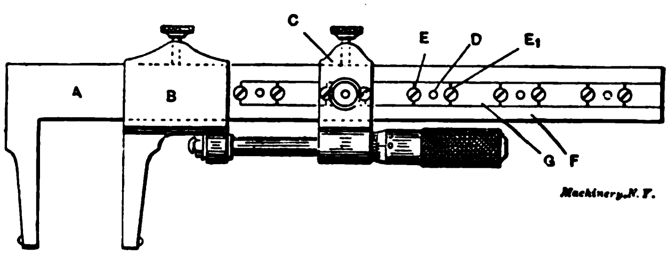


Fig. 17. Improved Micrometer Beam Caliper

is to provide as many holes through the slide and beam as there are inch divisions, each hole being drilled and reamed through both the slide and beam at once. If it were attempted to drill the holes through the beam at exactly one inch apart, having only one hole in the clamping head and using it as a jig for the purpose, it would be found very difficult, if not impossible, to get the holes all of one size and exactly one inch apart. The design of the micrometer beam caliper shown in Fig. 17, which has been patented by Mr. Frank Spalding, Providence, Rhode Island, is such, however, that it is not necessary to drill more than one hole through the clamping slide. The beam *F* is grooved longitudinally, and in the groove are fitted hardened steel adjusting blocks in which a taper hole *D* is accurately finished. Between the blocks are filling pieces *G*, which are brazed or otherwise fastened in the groove. Holes are drilled, tapped, and countersunk between the blocks and the filling pieces *G*, in which are fitted taper head screws EE_1. The construction is thus obviously such that the blocks may be shifted longitudinally by loosening one screw and tightening the other. In constructing the caliper, the holes through the beam are drilled as accurately as possible, one inch apart, and centered in the longi-

* L. S. Brown, March, 1903.

tudinal groove, but are made larger than the holes in the blocks, so as to provide for slight adjustment.

Large Beam Caliper

Fig. 18 shows a large beam caliper designed for machinists and patternmakers. It consists of a beam *MN* and the legs *R* and *S*, made of cherry wood to the dimensions indicated. The legs are secured in position on the beam by means of the thumb screws *A*, which jam against the gibs *C* at the points of the screws. The gibs have holes countersunk for the screws to enter, to hold them approximately in place, and

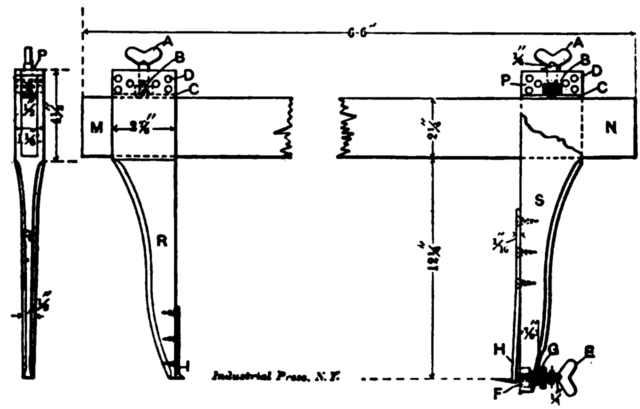


Fig. 18. Large Beam Caliper

the nuts *B* are of brass, fitted into the filling pieces *P* that keep them from turning. The filling pieces are riveted to the legs by means of cherry dowels *D*. One leg *S* is provided with a fine adjustment consisting of flexible steel spring *H*, ending in a point which is adjusted by the thumb screw *E*. This screw is locked in adjustment by the check nut *G* bearing against the brass nut *F*, which is inserted in the leg as shown.*

* C. W. Putnam, October, 1901.

CHAPTER III

MICROMETER MEASURING INSTRUMENTS

Of all measuring instruments used in the shop intended for accurate measurements, those working on the principle of the ordinary micrometer calipers are the most common. In the present chapter we shall describe and illustrate a number of different designs of these tools, intended to be used for various purposes. The instruments shown in Figs. 19 to 23 were built, in leisure hours, by Mr. A. L. Monrad, of East Hartford, Conn.

Micrometer for Snap Gages

Fig. 19 shows a form of micrometer that has proved very handy for measuring snap gages, and thicknesses, and can also be used as a small height gage to measure the distance from a shoulder to the base, as shown in Fig. 20. In measuring snap gages or thicknesses, the

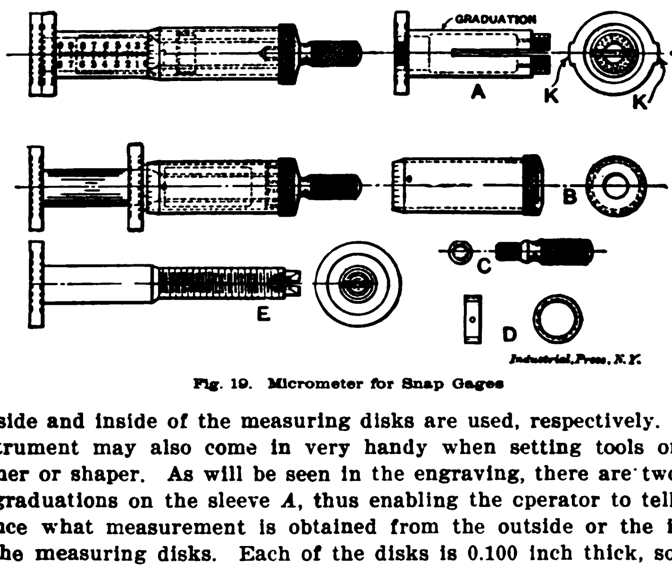


Fig. 19. Micrometer for Snap Gages

outside and inside of the measuring disks are used, respectively. This instrument may also come in very handy when setting tools on the planer or shaper. As will be seen in the engraving, there are two sets of graduations on the sleeve *A*, thus enabling the cperator to tell at a glance what measurement is obtained from the outside or the inside of the measuring disks. Each of the disks is 0.100 inch thick, so that the range of the micrometer is 0.800 and 1.000 inch for the outside and inside, respectively. The details of the instrument are as follows:

The sleeve *A* is composed of the inside measuring disk, the graduated sleeve, and the micrometer nut combined. On the disk are two projections *KK*, which are knurled, thus providing a grip when operating the tool. The sleeve is threaded on the inside of one end, which acts as a micrometer nut, and the outside of this same end is threaded to receive the adjusting nut *D*. The sleeve has two slots, each placed 90 degrees from the graduations, and these provide for compensation for wear. The disk part is hardened by heating in a lead bath, and is finished by grinding and lapping. The barrel *B* is the same as a regular micrometer barrel, and is graduated with 25 divisions. Spindle *E* consists of the outside disk and the micrometer screw, and the barrel *B* fits on its end, which is tapped out to receive the speeder *C*, which serves to hold the barrel in position. The thread is ¼ inch, 40 pitch, and the disk and unthreaded parts are hardened, ground and lapped. To adjust this instrument, loosen the speeder *C* and turn the barrel until the proper adjustment is obtained. Then lock the barrel by tightening the speeder again.*

Micrometer Caliper Square

Fig. 21 shows an assembled view and the details of a micrometer caliper square which, if accurately made, is equal and often preferable

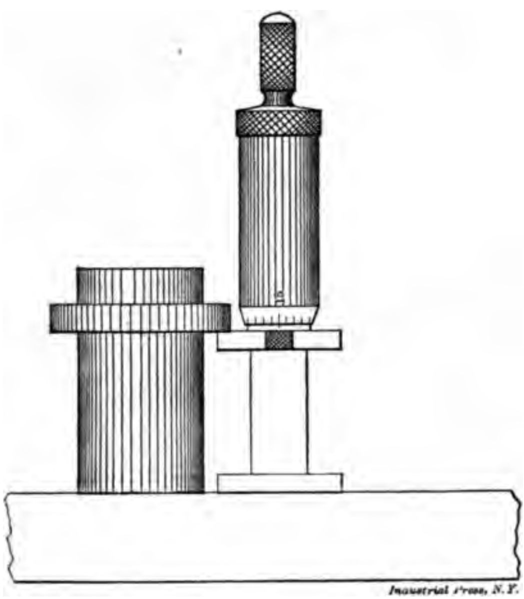


Fig. 20. Micrometer in Fig. 19 used as Height Gage

to the vernier caliper now so generally used. One of its advantages over the vernier is that when the measurement is taken, it can be readily discerned without straining the eyes, and this instrument is as easy to manipulate as the regular micrometer.

In the details, part *A*, which is the main body of the instrument, is made of tool steel, the forward or jaw end being solid with the body. This end is hardened, and the jaw ground and lapped. The body is bored out and two flats milled on the outside, which lighten it up and make it neat in appearance. The jaw end is counterbored out with a 45-degree counterbore to form a bearing for the forward end of

* Jos. M. Stabel, May, 1903.

the micrometer screw. A slot, ⅛ inch in width, extends from the fixed jaw to the other end, and in this slides the movable jaw *C*. There are 44 divisions along the side of this slot, each division being 0.050 inch apart, giving the tool a range of 2.000 inches for outside and 2.200 inches for inside measurements. The screw *B* is the most essential part of this tool, its construction requiring great accuracy. Its diameter is ⅜ inch and it is cut with 20 threads per inch. On its forward end fits the cone *F*, which is hardened and ground, the round part acting as the forward bearing of the screw and fitting in the 45-degree counterbored hole in the body *A*. On its other end fits the graduated barrel *D* and also the speeder *G*.

The barrel is graduated in fifty divisions, each division equaling 0.001 inch. On the inside of the barrel is a 45-degree bearing which rides on the cone *M*, the cone being held stationary on the end of the

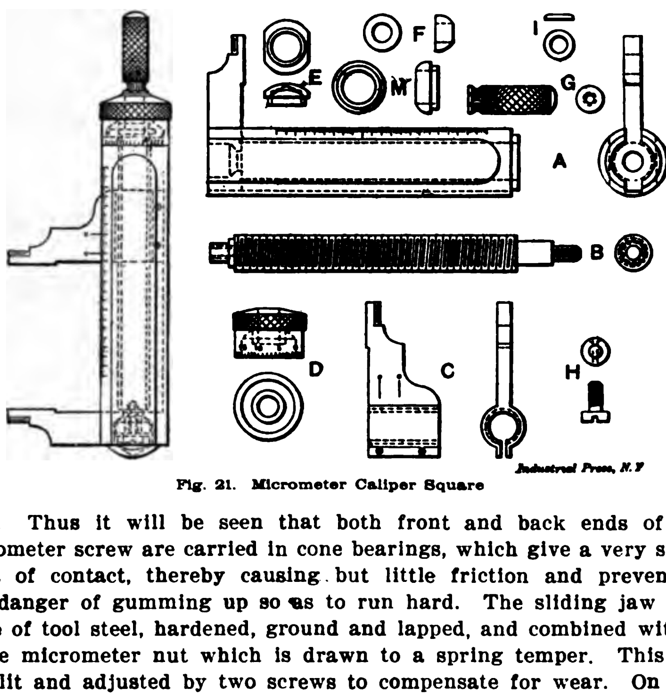


Fig. 21. Micrometer Caliper Square

body. Thus it will be seen that both front and back ends of the micrometer screw are carried in cone bearings, which give a very small point of contact, thereby causing but little friction and preventing any danger of gumming up so as to run hard. The sliding jaw *C* is made of tool steel, hardened, ground and lapped, and combined with it is the micrometer nut which is drawn to a spring temper. This nut is split and adjusted by two screws to compensate for wear. On this jaw are the two zero marks that tell at a glance the outside or inside measurements taken. The screw and washer, marked *H* and *I*, go onto the end of the micrometer screw and take up the end play. To make a neat appearance, the cap *E* is placed in the forward counterbored hole, being held in place by a tight fit. The adjustment of the tool is accomplished by loosening the speeder *G* and turning the barrel on the screw; when the adjustment is made, the speeder is again tightened down and the barrel locked.*

* Jos. M. Stabel, May, 1903.

Micrometer Depth Gage

The depth gage, shown in Fig. 22, has a ½-inch movement of the rod, and may be used with rods of any desired length. These have small 45-degree-on-a-side grooves cut into them at intervals of ½ inch. A small spiral spring, marked *I*, gives the rod a constant downward pressure, so that, when taking a measurement, the base of the tool is placed on the piece of work, and the rod always finds the bottom of the hole; then, by tightening the knurled screw *F* the rod is clamped in position and the tool may be picked up and its measurement read from the dial. The graduations on this instrument are similar to those

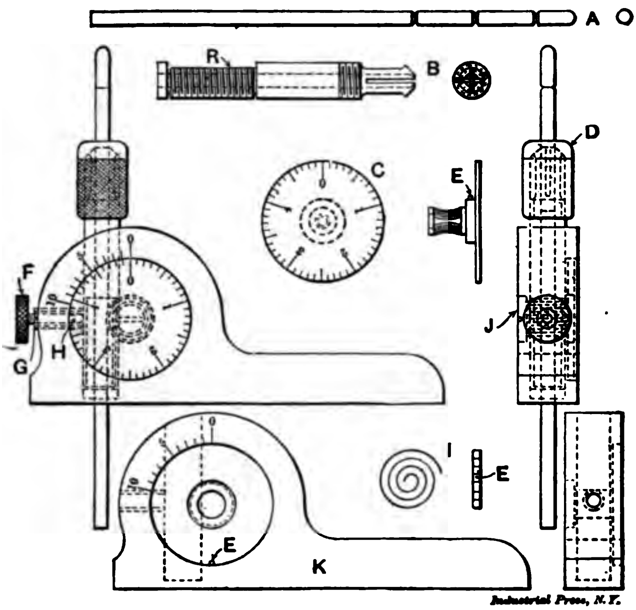


Fig. 22. Micrometer Depth Gage

of the vernier caliper, only they are much plainer, as a half-inch movement of the rod turns the dial one complete revolution. The figures on the dial denote tenths of an inch, and those on the body of the tool thousandths; each graduation on the dial is therefore equal to 0.010, so that to show the depth of a hole to be 0.373 the dial would be revolved around so that the seventh division beyond the 3 mark would be near to 0, and then by looking from the 0 mark toward the left, the third graduation on the body and one on the dial would be in line, thus denoting 0.373.

The most essential part of this tool is the threaded screw *B*, which acts as a rack, and the worm-wheel, solid with the dial *C*. The upper end of the screw forms a split chuck which grips the measuring rods, while the part marked *R* is flatted off, and against this portion bears a threaded sleeve *G*, which acts as a key to keep the screw in position.

This sleeve is threaded, both inside and outside, and screws into the body of the tool, while the binding screw *F* fits into it and binds against a small piece of copper, marked *H*, which in turn holds the screw in position. The thread on *B* is 0.245 inch in diameter and is cut with 40 threads per inch. The worm-wheel which meshes into this screw is solid with the dial, as shown at *C*. It is 0.18 inch in diameter, and requires great accuracy in cutting; it is not hobbed, but the teeth, of which there are twenty, are milled with a circular cutter of the same diameter as the screw *B* plus 0.002 inch. The little studs, marked *EE*, on the dial and on the body *K*, hold the coiled spring in position. Very great accuracy must be attained when locating the holes in *K* that are to receive the screw and dial *B* and *C*. The screw marked *J* fits into

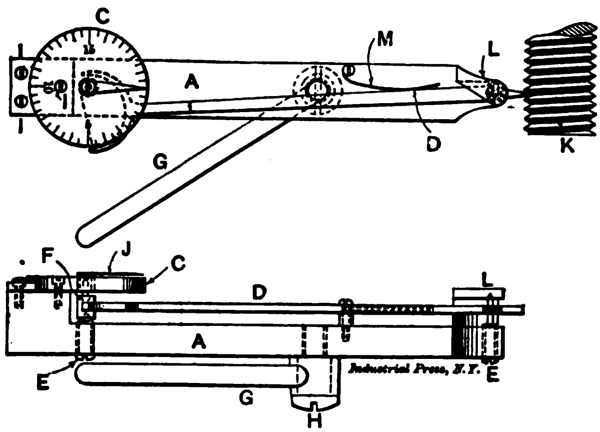


Fig. 23. Indicator for Accuracy of Lead-screws

the dial, where it serves as a bearing and also holds the dial in position. The knurled cap *D* tightens the split chuck in order to hold the measuring rod firmly.*

Indicator for Accuracy of Lead-screws

All of the tools that have been described require an accurately cut screw, and, as very few lathes are capable of producing this, it may be well to illustrate an indicator for testing the accuracy of the lead-screw, and to explain the method by which it is used. This instrument is shown in Fig. 23, where it is applied to a test screw *K*. It consists of a body *A* on one end of which is a projection *L* serving as the upper bearing for the pivoted lever *D*. This lever swings about a small steel pivot which can be adjusted by the screw *E*. The rear end of the lever is forked, and between the prongs is passed a thread making a double turn about the pivot *F* that carries the pointer *J*. Any movement of this lever will, therefore, cause this pointer to revolve about the dial *C*. This dial has 20 divisions, each indicating one-half thousandth of an inch movement of the front end of the lever, so that

* Jos. M. Stabel, May, 1903.

a total revolution of the pointer about the dial would indicate a movement of the front end of the lever of 0.020 inch. The screws *I* serve to hold the dial in place on the body of the indicator, while the spring *M* keeps the pointer normally at the zero mark. The indicator is held in the toolpost by the arm *G*, which can be set at any angle and firmly clamped by the screw *H*.

To use the indicator, remove the screw from a micrometer which is known to be accurate, and, with the aid of a brass bushing, chuck it in the lathe so that the thread end will project. Now gear the lathe to cut 40 threads per inch and apply the indicator. When the lathe is started, the point of the indicator follows along in the thread of the micrometer screw, and any variation in the lead will be noted by a

Fig. 24. Micrometer with Attachment for Reading Ten-thousandths of an Inch

movement of the pointer over the dial. If, on the other hand, no movement takes place, it is an indication that the pitch of the lead-screw is correct.*

Micrometer Attachment for Reading Ten-thousandths of an Inch

Fig. 24 shows an attachment for micrometers designed and made for readings in tenths of thousandths of an inch. With very little fitting it is interchangeable for 1-, 2-, or 3-inch B. & S. micrometers. The idea is simple, as can be seen by the illustration. The diameter of the thimble is increased 3 to 1 by a disk which is graduated with 250 lines instead of 25, making each line represent 0.0001 inch instead of 0.001 inch. A piece of steel is then turned up and bored and cut away so as to form the index blade and a shell to clasp the micrometer frame, the whole thing being made in one piece. The thimble disk being just a good wringing fit, it can be easily adjusted 0 to 0. The attachment can be removed when fine measuring is not required.**

Special Micrometer for Large Dimensions

Fig. 25 shows a 6-inch micrometer caliper designed for measuring from 0 to 6 inches by half-thousandths. The sliding micrometer head

* Jos. M. Stabel, May, 1903.
** P. L. L. Yorgensen, February, 1908.

travels on a cylinder barrel through which a hole is accurately bored to suit three plugs, one, two, and three inches long, as shown in the engraving. These plugs serve to locate the traveling head at fixed distances one inch apart. The micrometer screw itself has a travel of one inch, like any standard micrometer. A locknut is used to hold the screw in any desired position. A thumb screw at the end of the barrel bears against the end plug, and zero marks are provided to bring the screw against the plug with the same degree of pressure at each setting. When the head is clamped by means of the locking nut, it is

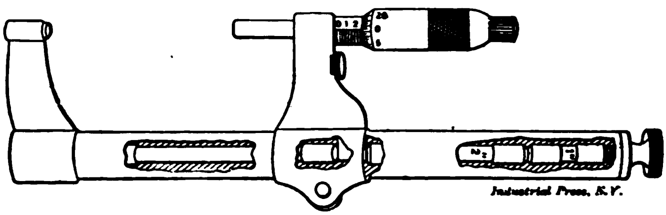


Fig. 25. Special Micrometer for Large Dimensions

as rigid as though it were solid with the barrel, and the faces of the measuring points are thus always parallel.

Combination Micrometer

A combined one- and two-inch micrometer is shown in Fig. 26. One side records measurements up to one inch, and the other side up to two inches. A single knurled sleeve or nut serves to move the double-ended measuring piece one way or the other as desired, this piece

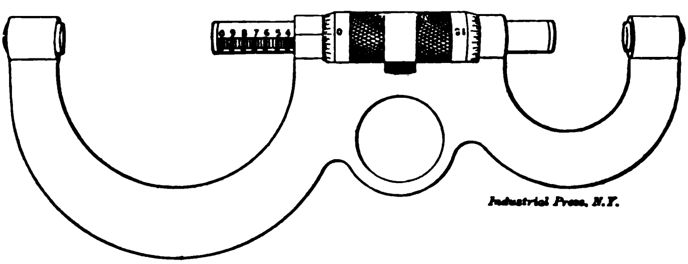


Fig. 26. Combined One- and Two-inch Micrometer

having a travel of one inch. The spindle is non-rotating, so that the faces of the screw and anvil are always parallel. A locking device holds the screw in any position. This tool is convenient for use both in measuring and as a gage, since it can be conveniently held by the finger ring appearing at the back.

Micrometer Stop for the Lathe

Most micrometer lathe stops are limited in their use to work where only a stationary height is required. It is, however, often necessary to use the stop at different heights, to accommodate different lathes; then again, we wish to use it on the right-hand side as well as the left. The form of holder shown in Fig. 27 can be used either right or left, and for various heights, and, by simply taking out the screw *A*, the micrometer may be removed and used in any other form of holder desired.

Both an assembled view and details of the holder are shown in the engraving, so that it can be easily constructed by any one desiring to do so. The micrometer and barrel may be procured from any of the manufacturers of measuring instruments. The swivel *C* is bored out so that the axis of the micrometer screw will be parallel to the body of the holder when it is in place. The swivel is made of tool steel and is fastened to the holder by the screw *A*. It is hardened and lapped to a true bearing surface on the sides and bottom, and so adjusted that it will turn to either side and remain in the desired position without moving the screw. The holder *B* is milled through

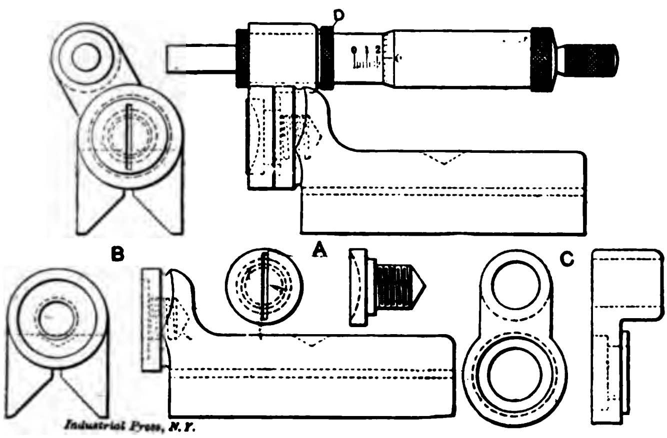


Fig. 27. Micrometer Stop for the Lathe

its entire length with a 90-degree cutter so that it will fit along the ways of the lathe, and the bottom is lapped to a true surface. For a neat appearance, the tool should be color hardened. On top the holder is spotted or countersunk with a drill to form a recess for the C-clamp. A knurled ring *D* is driven onto the micrometer sleeve so that it can be turned around to bring the graduations uppermost when the position of the barrel is changed.*

Micrometer Surface and Height Gage

Fig. 28 shows a form of surface gage that has proved very handy, and which can be used also as a height gage for measuring distances from shoulders to the base. If accurately made it is equal, and often preferable, to the vernier or slide caliper now so generally used with an attachment to the sliding jaw. One of its advantages over the vernier is the readiness with which the graduations are discerned, and it is as easy to manipulate as the ordinary micrometer. The part *B*, which forms the main body of the instrument, is made of tool steel, and one end is fitted into the base where it is held in position by the screw *D*.

* A. L. Monrad, December, 1903.

The remainder is milled to a thickness of ⅛ inch and has graduations of 0.025 inch for a distance of three inches. The screw *A* is the most essential part of the tool, and its construction requires great accuracy. Its diameter is ½ inch, and it is cut with 20 threads per inch. In the upper end of the screw is driven the ball *H* for the sake of giving a neat appearance. The top of the thread is turned off 0.010 inch to allow the scriber *F* to slide freely on the screw. The barrel *I* is used for raising and lowering the slide, but instead of having the gradua-

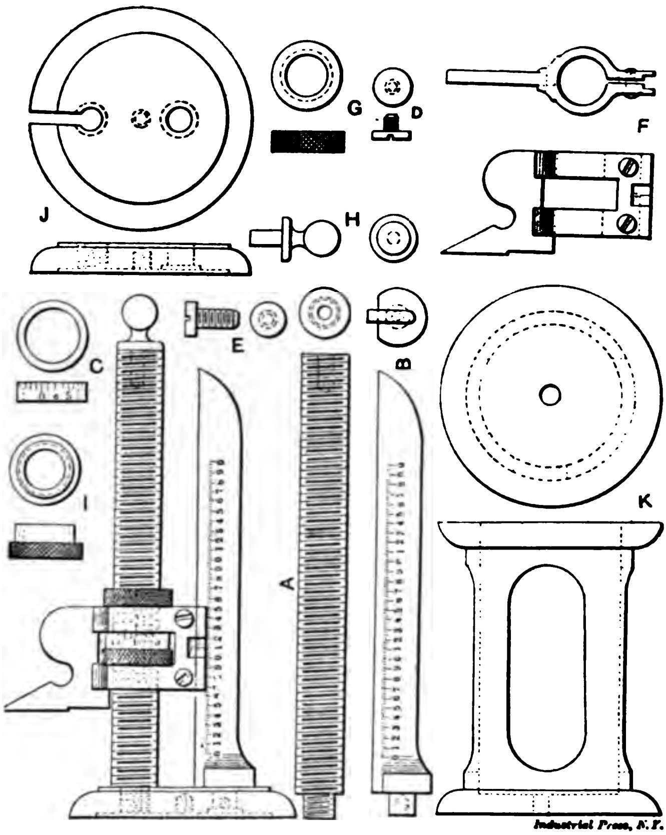


Fig. 28. Micrometer Surface and Height Gage

tions placed directly upon it, they are made upon the sleeve *C*, which fits over a shoulder on the barrel. This allows more easy means of adjustment than would be possible were the graduations placed on the barrel itself. The sleeve is graduated with fifty divisions each equaling a movement of the scriber of 0.001 inch. This sleeve may be turned by means of a small spanner wrench so as to bring the zero line into correct position to compensate for wear. A knurled locking nut is also provided for holding the scriber in any fixed position. The scriber itself is hardened and lapped to a finished surface, the tail end being slotted and provided with two screws to compensate for wear. On the scriber is placed the zero mark which shows at a glance the measurement that is being taken. The block *K* is three inches in height, and by using this block and placing the gage on its top, the range of the gage is increased to six inches. The screw *E* is used for fastening the gage to the top of the block. The center of the block is drilled out and slots cut through the sides in order to make it light and neat in appearance.*

Micrometer of from One- to Five-inch Capacity

Fig. 29 shows a very simple and light five-inch micrometer that can be quickly set to exact position from one to five inches. The round beam is graduated by a series of angular grooves, 1 inch apart, which

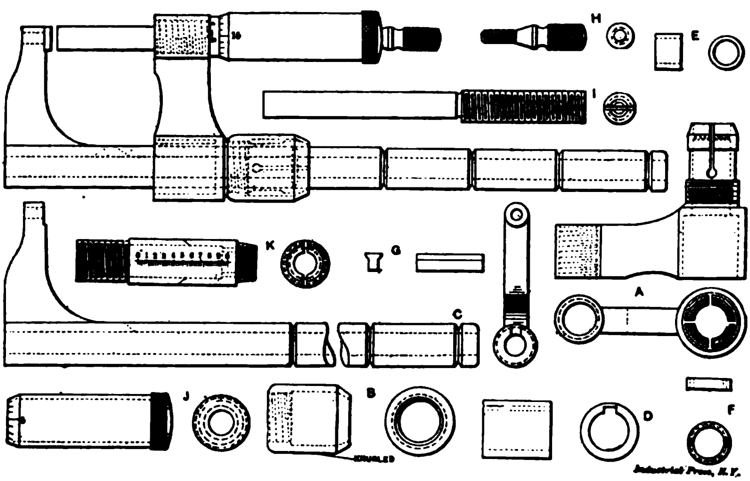


Fig. 29. Micrometer of from One- to Five-inch Capacity

are of such a form and depth that the clamping fingers at the end of part *A* spring in, allowing one inch adjustment of the beam to be quickly and positively made. The sleeve *K* is of tool steel, being counterbored from the forward end for all but one-half inch of its length. For this half inch it is threaded on the inside and acts as a micrometer nut. The outside of the same end is threaded to receive the adjusting nut *F*, and two slots are cut in the sleeve, at 90 degrees with the graduations. These slots, by a movement of the nut *F*, provide a means for compensating for wear. The bushing *E* is hardened and lapped, and fitted tightly in the forward counterbore of this sleeve, where it acts as a guide for the front end of the micrometer screw. The barrel *J* is the same as that of a regular micrometer, and is graduated in 0.025 inch divisions.

The most essential part of the tool is the threaded screw *I*, over the end of which fits the barrel *J*. The end is tapped out to receive the

* A. L. Monrad, December, 1903.

speeder *H*, which serves to hold the barrel in position. The thread is 5/16 inch in diameter, with 40 threads per inch, while the unthreaded part is hardened, ground and lapped. To adjust the instrument, loosen the speeder *H* and turn the barrel until the proper adjustment is obtained; lock the barrel by again tightening the speeder. The beam *C* has a ¼-inch hole drilled throughout its entire length in order to make it light. Small 90-degree grooves are cut into it at intervals of 1 inch, and a ⅛-inch slot is milled through one side to within 1¼ inch of the forward end. The back end of part *A* forms a spring-tempered split chuck, which grips the beam and holds *A* in position, while the exterior is threaded to receive the knurled cap *B* by which the chuck is tightened firmly to the beam. From the front end, toward

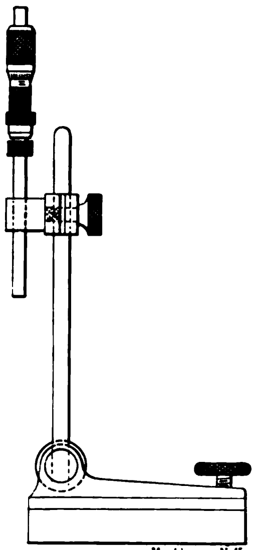


Fig. 30. Method of Setting Calipers from Inside Micrometers

the split chuck, the body is counterbored ⅝ inch and the bushing *D* driven in tight. This bushing has a key *G* fitted into it, which slides in the slot of the beam and prevents the arm from turning. The projecting arm is bored and tapped to receive the sleeve *K*. This gage must be carefully and accurately made to be of value.*

Inside Micrometer for Setting Calipers

Fig. 30 shows an application of inside micrometers which is very handy. The hole for the scriber in the scriber clamp of a surface gage is reamed out to fit the rods used with inside micrometers. This forms a convenient holder for the micrometer when used for setting outside calipers to it. The calipers can be set easily and accurately at the same time, and where extreme accuracy is not necessary this arrangement is more handy than that of using large-sized micrometers.

* A. L. Monrad, December, 1903.

With care and practice an accuracy of within one-quarter of 0.001 inch is obtainable in this way. Mistakes, in fact, are more easily guarded against than is the case when using the micrometers directly.

Micrometer Frame

Fig. 31 shows a micrometer frame used some years ago at the Westinghouse works. The frame is an aluminum casting, and the anvil is simply a tool-steel pin, which fits well in the hole into which it is

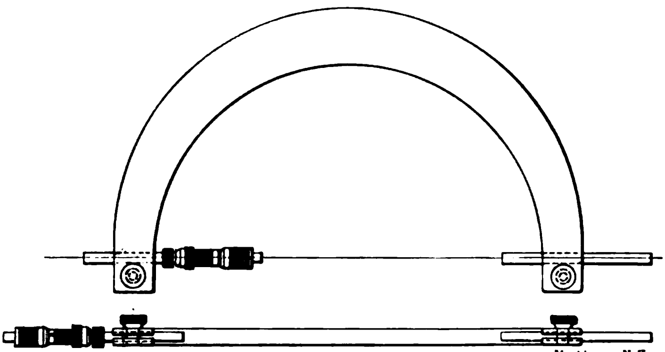


Fig. 31. Useful and Handy Micrometer Frame

inserted, and can be clamped anywhere within the limits of its length. The micrometer end of the frame is supplied with an inside micrometer head. The tool is adjusted to a gage, either to a standard pin gage,

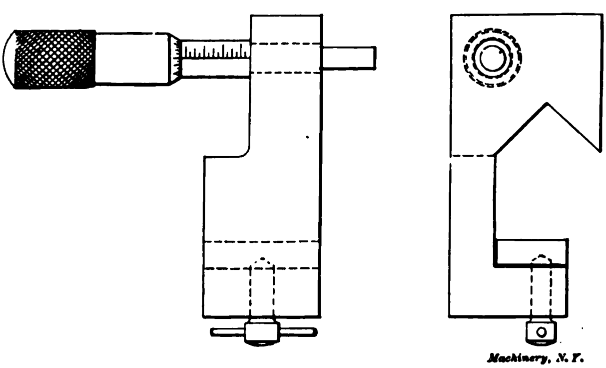


Fig. 32. Micrometer Stop for the Lathe

or to an inside micrometer gage. The capacities of three of these micrometers in a set may be from about 3½ to 7 inches, 6 to 11 inches, and 10 to 15 inches. When the head is turned outward, as shown in the lower view in the cut, the tool is very handy around a horizontal boring machine where a pin gage cannot be used without removing the boring bar.

Micrometer Stop for the Lathe

The simple micrometer stop shown in Fig. 32 is used on the engine lathe for obtaining accurate movements of the lathe carriage. It consists of a micrometer head, which can be purchased from any micrometer manufacturer, and a machine steel body which is bored to fit the micrometer head. This tool is clamped on the front way of the lathe bed, and when the jaw of the micrometer is against the lathe carriage, it can easily be adjusted to a thousandth of an inch. Of course, care should be taken not to bump the carriage against the micrometer.*

Use of Micrometer for Internal Thread Cutting

Fig. 33 illustrates a means of determining the size of internally threaded work. The work shown is intended for a lathe chuck. The outside diameter of the hub on the work is turned to the same size

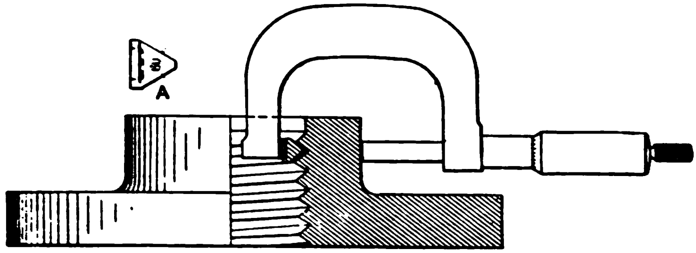
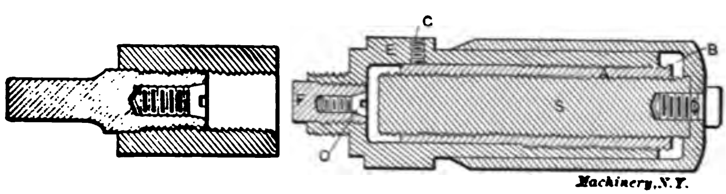


Fig. 33. Method of using Micrometer for Internal Thread Cutting

as the hubs on small faceplates which are furnished with all new lathes. The threaded size is then taken and transferred with a micrometer, over the anvil of which is fitted a 60-degree point as shown enlarged at *A*. In connection with a graduated cross-feed screw this greatly facilitates the work over the usual cut-and-try method.**

Inside Micrometer

The inside micrometer shown in sections in Figs. 34 and 35 is adapted to measuring, by use of extension rods, from 2 inches up to



Figs. 34 and 35. Section of Inside Micrometer

size of hole, and has one inch adjustment of the measuring screw. Referring to the section shown in Fig. 35, the measuring screw *S* is secured to the thimble *B* with the screw *D*, the head of which is hardened and forms the anvil. By loosening this screw *D*, the thimble can be rotated to compensate for wear. The wear of the measuring screw and nut is taken up by screwing the bushing *A* into the frame with the wrench shown in Fig. 37. This bushing is split in three sec-

* J. L. Marshall, February, 1908.
** Charles Sherman, November, 1905.

tions for about two-thirds of its length on the threaded end. The three small lugs on the wrench fit into these slots. The handle end of the wrench is a screw driver which is used for manipulating the set screw *C*. The bushing is made an easy fit in the frame on its plain end and tapered, as shown, on its outside threaded part. This thread

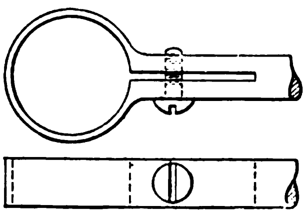


Fig. 36. Handle for Inside Micrometer, Figs. 34 and 35

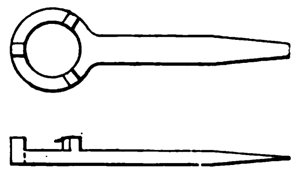


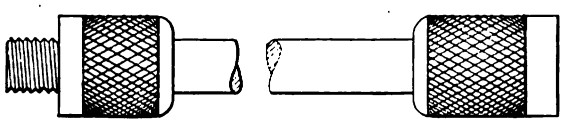
Fig. 37. Wrench used with Inside Micrometer

being the same pitch as the measuring screw, adjustment for wear does not affect the reading of the micrometer. This manner of adjustment brings the nut squarely down on the measuring screw for its whole length, presenting the same amount of wearing surface after adjustment as when new.

The point *F*, which is hardened on its outer end, screws into the frame, and is secured by the taper-headed screw *O*, which screws into and expands the split and threaded end of the point *F*. The handle, Fig. 36, clamps over the knurled part of the frame for use in small, deep holes. The rods, six in number, running from 1 to 6 inches inclusive, are made by screwing a sleeve onto a rod with a hardened point and locking it with a taper-headed screw on its threaded and split end, the same as in the point *F*. The extension pieces, Fig. 38, are adjustable, on their socketed ends, in the same way, and run in lengths of 6, 12, 18 inches, etc.*

Direct Fractional-reading Micrometer

The direct fractional-reading micrometer shown in Fig. 39 is the result of talks with many mechanics in which all agreed that such a feature added to a micrometer would, by making it both a fractional and decimal gage, more than double its practical value. While approximate readings in 64ths, etc., may be obtained by the graduations on the



Industrial Press, N.Y.

Fig. 38. Adjustable Extension Pieces for Inside Micrometer

barrel *B* as on an ordinary inch scale, the exact readings of 64th, etc., may be obtained only by reference to graduations on the movable thimble *A*. There are but eight places on *A* which coincide with the long graduation line on *B* when any 64th, 32d, 16th, or 8th is being

* M. H. Ball, May, 1903.

measured, and each of these eight places is marked with a line, and the 64th, 32d, 16th, or 8th for which that line should be used is marked thereon. (See *a* and *b*, Fig. 40.) The line *a* would be used for 3/32, 7/32, 11/32, etc., and the line *b* for 1/64, 9/64, 17/64, etc. Now suppose we wish to accurately measure 15/32 inch. We first roughly read it off the inch scale on sleeve *B* by turning out thimble *A*. Having

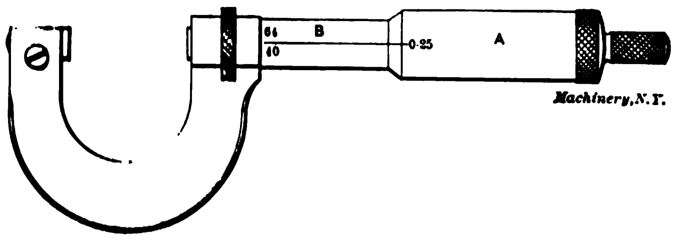


Fig. 40. Direct Fractional-reading Micrometer

secured it closely by drawing edge of *A* over that graduation, we find that the line *a* (Fig. 40) on the movable thimble very nearly or exactly coincides with the long graduation line on *B*. When these lines coincide, we have the exact measurement of 15/32 inch without reference to how many thousandths may be contained in the fraction. Thus all through the scale any fraction may be found instantly. There is no mental arithmetic, use of tables, or memory work in using the tool.

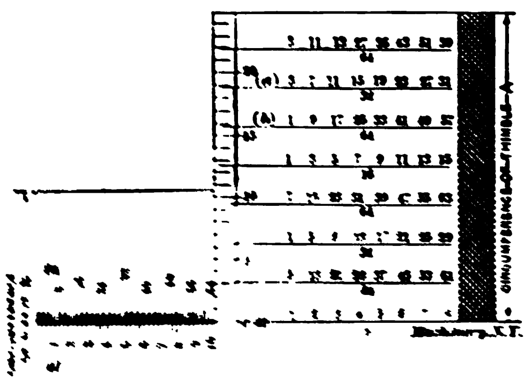


Fig. 41. Graduations on the Fractional-reading Micrometer

The new graduations are independent of the old and may be used [illegible] with or without them.

[illegible] Instead of using [illegible] of a turn [illegible] correspond, as [illegible] etc., on *A* [illegible] and the numerators [illegible] placed at the end instead of under the line. To the ordinary user of the tool, this is all that is necessary for a perfectly clear reading of the fractions.*

Sensitive Attachment for Measuring Instruments

No matter how finely and accurately micrometers and verniers may be made, dependence must in all cases be placed on the sensitiveness of a man's hand to obtain the exact dimensions of the piece to be measured. In order to overcome this difficulty and eliminate the per-

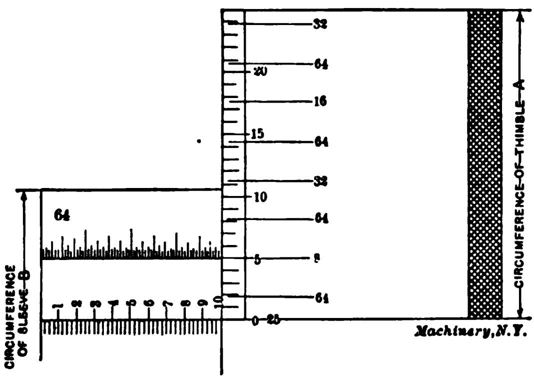


Fig. 41. Another Method of Graduating for Fractional Reading

sonal equation in the manufacture of duplicate and interchangeable parts, the sensitive attachment to the micrometer shown in Fig. 42 may be used, and will be found of much value.

The auxiliary barrel *A* is held to the anvil of the micrometer by means of a thumb screw *B*. At the inside end of the barrel is a secondary anvil *C*, the base of which bears against the short arm of the indicating lever *D*. The action will be clearly seen by reference to the engraving. The micrometer is so set that when a gage, *G*, of exact

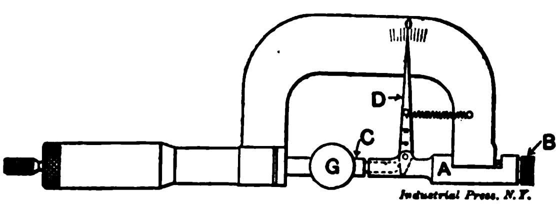


Fig. 42. Sensitive Micrometer Attachment

size, is placed between the measuring points, the long arm of the indicator stands at the 0 mark. If the pieces being calipered vary in the least from the standard size it will be readily noted by the movement of the pointer. Hard rubber shapes turned from rough casting often vary from 0.003 to 0.005 inch after having passed the inspector's test with an ordinary micrometer. With this attachment the inspector's helper can detect very minute variations from the limit size.

* Chas. A. Kelley, May, 1908.

Anything within the limits of the micrometer can be made to show to the naked eye variations as small as a ten-thousandth inch.*

Another Sensitive Micrometer Attachment

When testing the diameters of pieces that are handled in great quantities and are all supposed to be within certain close limits of a standard dimension, the ordinary micrometer presents the difficulty of having to be moved for each piece, and small variations in diameters have to be carefully read off from the graduations on the barrel. Not only does this take a comparatively long time, but it also easily happens that the differences from the standard diameter are not carefully noted, and pieces are liable to pass inspection that would not pass if a convenient arrangement for reading off the differences were at hand. Fig. 43 shows a regular Brown & Sharpe micrometer fitted

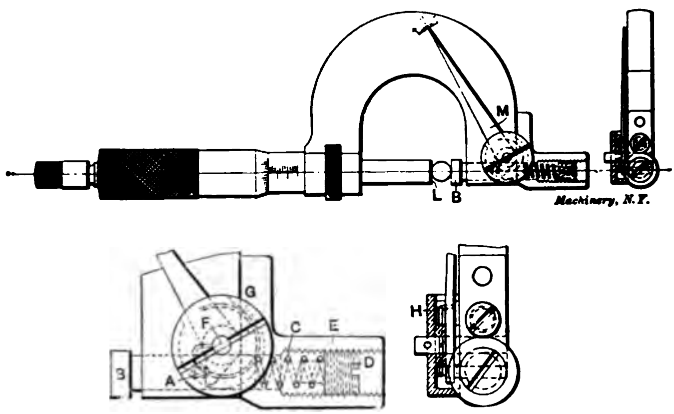


Fig. 43. Another Sensitive Micrometer Attachment

with a sensitive arrangement for testing and inspecting the diameters of pieces which must be within certain close limits of variation. The addition to the ordinary micrometer is all at the anvil end of the instrument. The anvil itself is loose and consists of a plunger *B*, held in place by a small pin *A*. The pin has freedom to move in a slot in the micrometer body, as shown in the enlarged view in the cut. A spring *C* holds the plunger *B* up against the work to be measured, and a screw *D* is provided for obtaining the proper tension in the spring. The screw and the spring are contained in an extension *E* screwed and doweled to the body of the micrometer. A pointer or indicator is provided which is pivoted at *F* and has one extensional arm resting against the pin *A*, which is pointed in order to secure a line contact. At the end of the indicator a small scale is graduated with the zero mark in the center, and as the indicator swings to one side or the other the variations in the size of the piece measured are easily determined. A small spring *G* is provided for holding the pointer up against the pin *A*. The case *H* simply serves the purpose of protecting the spring men-

* H. J. Bachmann, December, 1902.

tioned. As the plunger *B* takes up more space than the regular anvil, the readings of the micrometer cannot be direct. The plunger *B* can be made of such dimensions, however, that 0.100 inch deducted from the barrel and thimble reading will give the actual dimension. Such a deduction is easily done in all cases. In other words, the reading of the micrometer should be 0.100 when the face of the measuring screw is in contact with the face of the plunger; the 0.100 inch mark is thus the zero line of this measuring tool.

When desiring to measure a number of pieces, a standard size piece or gage is placed between the plunger *B* and the face *L* of the micrometer screw, and the instrument is adjusted until the indicator points exactly to zero on the small scale provided on the body of the micrometer. After this the micrometer is locked, and the pieces to be measured are pushed one after another between the face *L* and the plunger *B*,

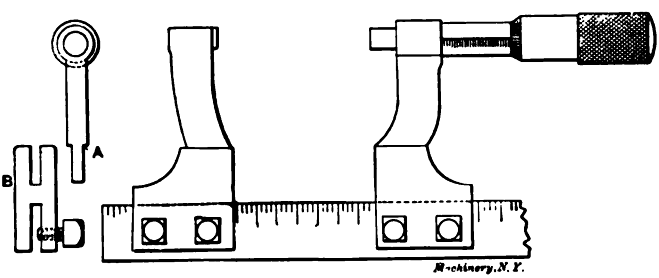


Fig. 44. Micrometer Mounted on Machinist's Scale

the indications of the pointer *M* being meanwhile observed. Whenever the pointer shows too great a difference, the piece, of course, does not pass inspection. All deviations are easily detected, and any person of ordinary common sense can be employed for inspecting the work.

Micrometer Scale

A micrometer, mounted as shown in Fig. 44 is very handy. The micrometer may be used in combination with a 4-, 6-, 9-, or 12-inch scale. It can be adjusted on standard plugs, or one can make a set of gages up to 12 inches, out of 3/16-inch round tool steel wire, and use these for setting. In mounting the micrometer, before cutting it apart, mill the shoulders shown at *A*, and in milling the bottom pieces *B*, use a piece of machine steel long enough for both, cutting the piece in half after milling the slots. In this way one obtains perfect alignment. In a shop where a set of large micrometers is not kept, this arrangement is very useful.*

* Wm. Ainscough, May, 1908.

CHAPTER IV

MISCELLANEOUS MEASURING TOOLS AND GAGES

Among the miscellaneous measuring tools and gages dealt with in this chapter are tools and gages for measuring and comparing tapers, adjustable gages, radius gages, gages for grinding drills, sensitive gages, tools for gaging taper threaded holes, contour gages, etc. Of course, these are offered merely as examples of what can be done in the line of measuring tools for different purposes, and, while having a distinct and direct value to the mechanic, they also have a great indirect value, because they furnish suggestions for the designing and making of tools for similar purposes.

Tool for Measuring Tapers

Fig. 45 shows a tool which has proved very useful. It is a tool for measuring tapers on dowel pins, reamers, drill shanks, or anything

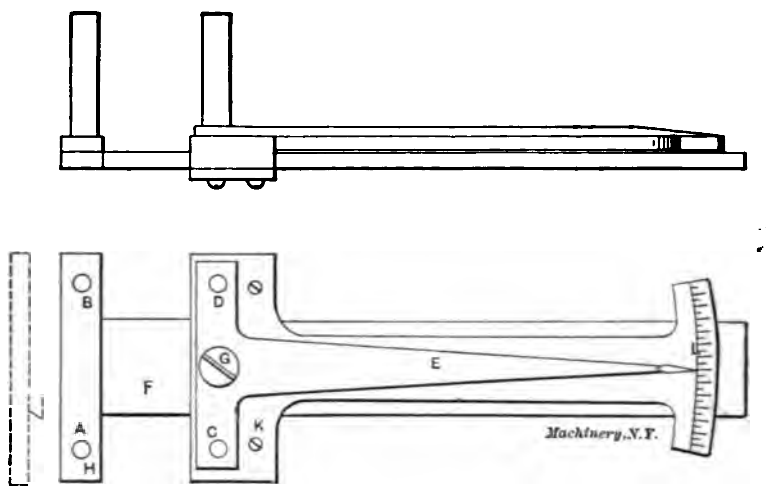


Fig. 45. Taper Measuring Tool

to be tapered. Most machinists know that to find the taper of a shank they must use their calipers for one end and reset them for the other end; or else caliper two places, say, three inches apart, and if, for instance, the difference should be 1/16 inch, they must multiply this difference by four to get the taper per foot. With the tool above mentioned, all this trouble in calipering and figuring is saved. Simply place the shank or reamer to be measured between pins *A*, *B*, *C*, and *D*, and slide *H* and *K* together. Then the taper can be read at once on the graduated scale at *L*. The construction of the tool will be readily understood. The body or base *F* has a cross piece supporting the two pins *A* and *B*. On this slides piece *K*, which has at its right end the graduated segment. The screw *G* is fast to piece *K*, and upon it swivels the pointer *E*, which carries the two pins *C* and *D*. Thus these two pins can be brought into contact with a tapered piece of any diameter within the capacity of the tool, and the swivel screw *G* allows the pins to adjust themselves to the taper of the work and the pointer *E* to move to the left or right, showing instantly the taper per foot.

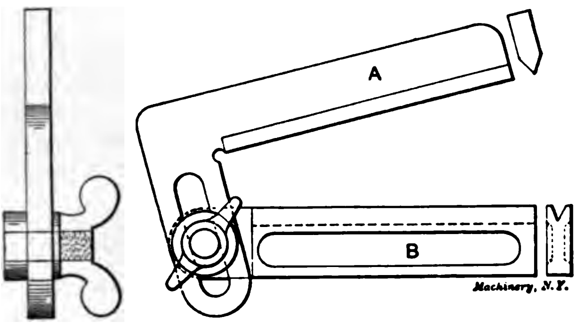


Fig. 46. Handy Taper Gage

As the pins *A* and *B* are 1½ inch apart, which is ⅛ of a foot, and the distance from *G* to *L* is 4½ inches, which is three times longer than the distance between *A* and *B*, the graduations should be 3/64 inch apart, in order to indicate the taper per foot in eighths of an inch.*

Taper Gage

A handy taper gage is shown in Fig. 46. The blades of the gage are made of tool steel. The edge of the blade *A* is V-shaped, and the blade *B* has a V-groove to correspond. The end of *B* is offset so as to make the joint and allow the two blades to be in the same plane. A

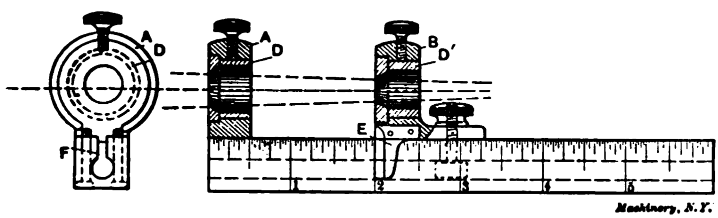


Fig. 47. Test Gage for Maintaining Standard Tapers

strong screw and nut are provided to hold the blades at any setting. The user of this gage looks under the edge of *A*. and is thereby enabled to tell whether the taper coincides with that set by the gage, and also where a taper piece needs touching up to make it true.**

Test Gage for Maintaining Standard Tapers

In steam injector work, accurately ground reamers of unusual tapers are commonly required, and the gage shown in Fig. 47 was designed to

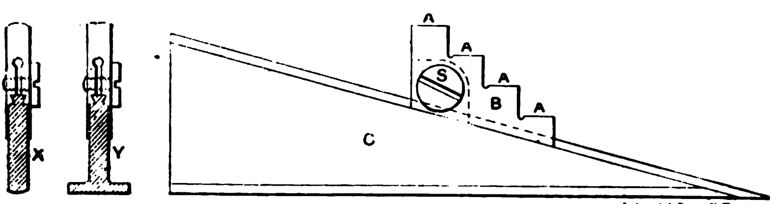
* John Aspenleiter, October, 1900.
** W. W. Cowles, June, 1901.

maintain the prevailing standard. It consists of a graduated bar, 1 inch square, with the slot F running its entire length. The stationary head A is secured in position flush with the end of the bar, and the sliding head B is fitted with a tongue which guides it in the slot. This head may be secured in any desired position by means of a knurled thumb nut. The bushings D and D' are made of tool steel, hardened and ground to a knife edge on the inside flush with the face. All bushings are made interchangeable as to outside diameter.

The head B is fitted with an indicating edge E which is set flush with the knife edge of the bushing. The reading indicates to 0.010 inch the distance the bushings are from each other, and the difference in their diameter being known, it is easy to compute the taper. With this gage it is possible to maintain the standard tapers perfectly correct, each reamer being marked with the reading as shown by the scale.*

Inside and Outside Adjustable Gages

Fig. 48 shows an inside and an outside adjustable gage for accurate work, used in laying out drill jigs, and in setting tools on lathes, shapers, planers, and milling machines. The outside gage is shown



Industrial Press, N.Y.

Fig. 48. Adjustable Gage for Inside and Outside Measurements

in the side view and in the sectional end view marked Y. At X in the same figure is a sectional end view showing how the gage is constructed for inside work. The top and bottom edges are rounded, so ·t the diameters of holes may be easily measured.

› gage consists of a stepped block B, mounted so as to slide upon nclined edge of the block C. There are V-ways upon the upper of the latter, and the block B is split and arranged to clamp over ways by the screw shown at S. All parts of the gage are hard- ‑d and the faces of the steps marked A, are ground and finished so at at any position of the slide they are parallel to the base of the ‹ C. The lower split portion of the block is spring-tempered to breaking under the action of the screw, and also to cause it n when loosened. The gage has the advantage that it can sted to any size within its limits, which does away with In planing a piece to a given thickness, the gage may height with great accuracy by means of a micrometer the planer or shaper tool adjusted down to the gage. way with the "cut-and-try" process, and will bring

‑r, 1904.

the finishing cut within 0.001 inch of the required size. If the piece being planed, or the opening to be measured, is larger than the extreme limit of the gage, parallels may be used. In fitting bushings into bushing holes, the adjustable gage may be moved out to fit the hole, and then, when the bushing is finished to the diameter given by the gage, as determined by a micrometer caliper, a driving fit is ensured.*

Radius Gage

Fig. 49 shows a radius gage which has proved to be very handy for all such work as rounding corners or grinding tools to a given radius.

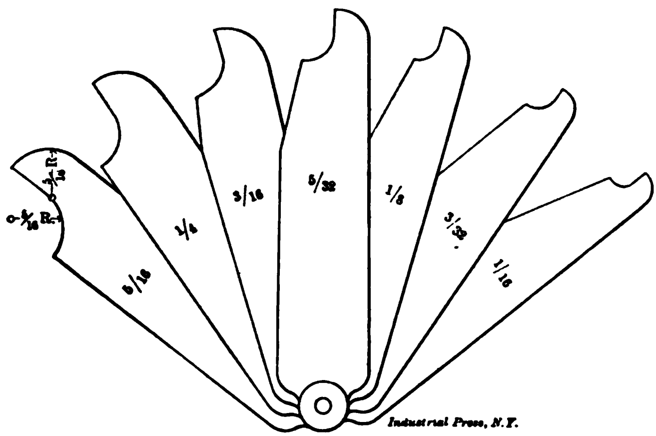


Fig. 49. Radius Gage

The blades are of thin steel, and are fastened together at the end by a rivet, thus forming a tool similar to the familiar screw pitch gage. The right-hand corner of each blade is rounded off to the given radius, while the left-hand corner is cut away to the same radius, thus providing an instrument to be used for either convex or concave surfaces. The radius to which each blade is shaped is plainly stamped upon the side.**

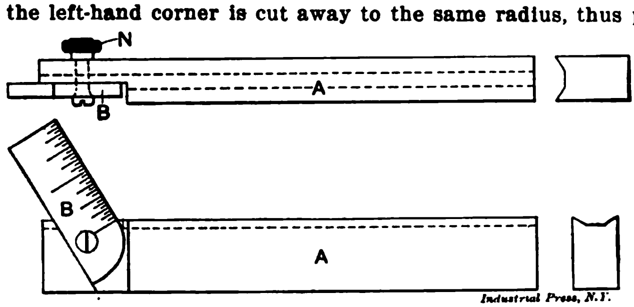


Fig. 50. Gage for Grinding Drills

Gage for Grinding Drills

Fig. 50 shows a gage for use in grinding drills, which has been found very handy and accurate. This gage enables either a large or small

* Geo. M. Woodbury, February, 1902.
** A. Putnam, July, 1903.

economical tool, and it will doubtless be appreciated by those having much work of this kind to do. The hole in which the stud is to be fitted is calipered by filling the threads of the plug with chalk, and

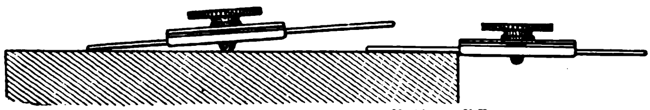


Fig. 53. End View of Contour Gage

then screwing the plug in the hole. When the plug is removed the chalk will show exactly the largest diameter of the hole.*

Contour Gage

Figs. 52, 53 and 54 illustrate a special tool which will be found of great value in certain classes of work. The need of some such device becomes apparent when patterns and core boxes are required to be accurately checked with the drawings of brass specialties, in particular. The tool is applied to the work, and the wires pressed down onto the contour by using the side of a lead pencil. Of course, patterns parted on the center could have their halves laid directly on the drawing without using the contour gage, but some patterns are cored and inseparable. Such a tool proves a relentless check upon the pat-

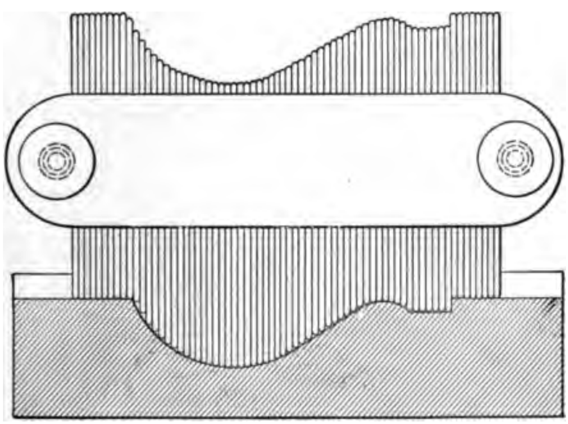


Fig. 54. Testing Core-box with Gage

ternmaker, who, by making the patterns larger than necessary, can cause a considerable loss in a business where thousands of casts are made yearly from the same patterns. As a ready and universal templet it is very useful.**

Testing a Lead-screw

A reliable way for testing the pitch of a lead-screw, at any position of its length, is to procure a micrometer screw and barrel complete,

* F. Rattek, January, 1908.
** Howard D. Yoder, December, 1907.

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 the holder when the screw is in place, as shown in Fig. 55. 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. 56, so that the body of the holder is parallel to the carriage. Adjust the micrometer to one inch when the

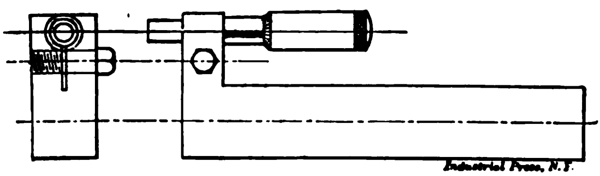


Fig. 55. Micrometer for Testing Lathe Lead-screw

point of the screw bears against the carriage and with a surface gage scribe a line on the outer edge of the faceplate. Now rotate the lathe spindle any number of full revolutions that are required to cause the carriage to travel over the portion of the lead-screw that is being tested, bringing the line on the faceplate 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 car-

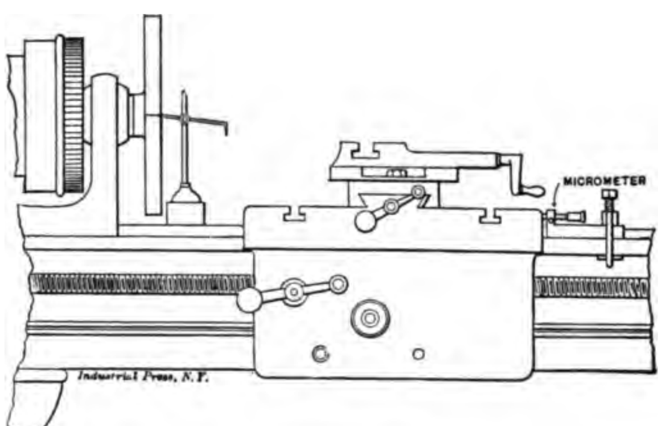


Fig. 56. Testing a Lathe Lead-screw

riage 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 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.*

Simple Tool for Measuring Angles

Fig. 57 shows a very simple, but at the same time, a very ingenious tool for measuring angles. Strictly speaking, the tool is not intended

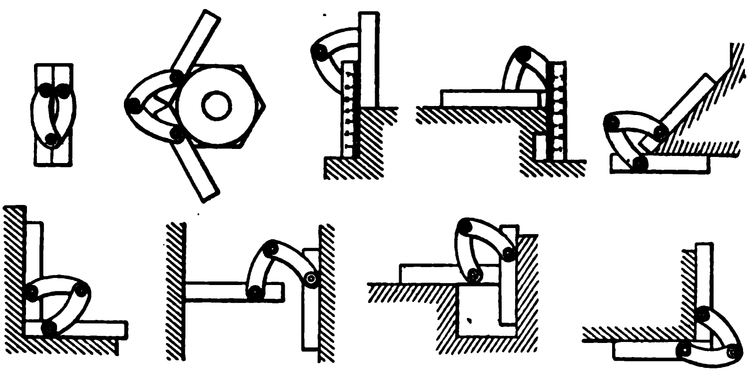


Fig. 57. Special Tool for Measuring Angles

for measuring angles, but rather for comparing angles of the same size. The illustration shows so plainly both the construction and the application of the tool, that an explanation would seem superfluous. It will be noticed that any angle conceivable can be obtained in an instant, and the tool can be clamped at this angle by means of screws

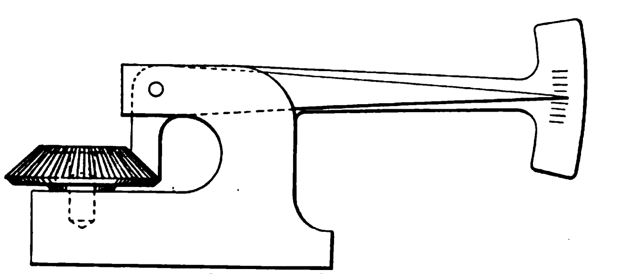


Fig. 58. Sensitive Gear-testing Gage

passing through the joints between the straight and curved parts of which the tool consists. Linear measurements can also be taken conveniently, one of the straight arms of the tool being graduated. As both of the arms which constitute the actual angle comparator are in

* W. Cantelo, July, 1903.

the same plane, it is all the easier to make accurate comparisons. This tool is of German design, and is manufactured by Carl Mahr, Esslingen a. N.

Bevel Gear-testing Gage

In Fig. 58 is shown a sensitive gage for inspecting small bevel gears. The special case shown to which the gage is applied in the engraving is a small brass miter gear finished on a screw machine, in which case some of the holes through the gears were not concentric with the beveled face of the gears, causing the gears to bind when running together in pairs. The gage shown is quite inexpensive, but it indicates the slightest inaccuracy.

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CHAPTER I

THE FACTOR OF SAFETY*

It is the custom among most firms engaged in the designing of machinery to settle upon certain stresses† as proper for given materials in given classes of work. These stresses are chosen as the result of many years of experience on their own part, or of observation of the successful experience of others, and so long as the quality of the material remains unchanged, and the service does not vary in character, the method is eminently satisfactory.

Progress, however, brings up new service, for which precedent is lacking, and materials of different qualities, either better or cheaper, for which the safe working stresses have not been determined, and the designer is compelled to determine the stress proper for the work in hand by using a so-called "factor of safety." The name "factor of safety" is misleading for several reasons. In the first place, it is not a factor at all, from a mathematical point of view, but is in its use a divisor, and in its derivation a product. In order to obtain the safe working stress, we divide the ultimate strength of the material by the proper "factor of safety," and in order to obtain this factor of safety we multiply together several factors, which, in turn, depend upon the qualities of the material, and the conditions of service. So our factor of safety is both a product and a divisor, but it is not a factor. Then again, we infer, naturally, that with a factor of twelve, say, we could increase the load upon a machine member to twelve times its ordinary amount before rupture would occur, when, as a matter of fact, this is not so, at least not in a machine with moving parts, sometimes under load, and sometimes not subjected to working stresses. Still more dangerous conditions are met with when the parts are subjected to load first in one direction, and then in the other, or to shocks or sudden loading and unloading. The margin of safety is, therefore, apparent, not real, and we will hereafter call the quantity we are dealing with the "apparent factor of safety," for the name factor is too firmly fixed in our minds to easily throw it off.

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* MACHINERY, January, 1906.

† Throughout this chapter we will adhere to the following definitions:

A "stress" is a force acting within a material, resisting a deformation.

A "load" is a force applied to a body, from without. It tends to produce a deformation, and is resisted by the stress which it creates within the body.

A "working load" is the maximum load occurring under ordinary working conditions.

A "working stress" is the stress produced by the working load, statically applied.

The "safe working stress" is the maximum permissible working stress under the given conditions.

The "ultimate strength" of a material is its breaking strength in pounds per square inch, in tension, compression, or shearing, as the case may be.

The "total stress" is the sum of all the stresses existing at any section of a body.

Unless a stress is mentioned as a total stress, the number of pounds per square inch of section, sometimes called "the intensity of stress," will be meant.

Formula for Factor of Safety

The apparent factor of safety, as has been intimated, is the product of four factors, which for the purpose of our discussion, we will designate as factors a, b, c, and d. Factors b and c, as will appear later, may be, and often are, 1, but none the less they must always be considered and given their proper values. Designating the apparent factor of safety by F, we have then

$$F = a \times b \times c \times d.$$

The first of these factors, a, is the ratio of the ultimate strength of the material to its elastic limit. By the elasic limit we do not mean the yield point, but the true elastic limit within which the material is, in so far as we can discover, perfectly elastic, and takes no permanent set. There are several reasons for keeping the working stress within this limit, the two most important being: First, that the material will rupture if strained repeatedly beyond this limit; and second, that the form and dimensions of the piece would be destroyed under the same circumstances. If a piece of wire be bent backward and forward in a vise, we all know that it will soon break. And no matter how little we bend it, provided only that we bend it sufficiently to prevent it from entirely recovering its straightness, it will still break if we continue the operation long enough. And similarly, if the axle of a car, the piston rod of an engine, or whatever piece we choose, be strained time after time beyond its limit of elasticity, no matter how little, it will inevitably break. Or suppose, as is the case with a boiler, that the load is only a steady and unremitting pressure. The yielding of the material will open up the seams, allowing leakage. It will throw the strains upon the shorter braces more than upon the others, thus rupturing them in detail. It is absolutely necessary, therefore, excepting in very exceptional cases, that we limit our working stress to less than the elastic limit of the material.

Among French designers it is customary to deal entirely with the elastic limit of the material, instead of the ultimate strength, and with such a procedure no such factor as we have been discussing would ever appear in the make-up of our apparent factor of safety. Although this method is rational enough, it is not customary outside of France, because many of the materials we use, notably cast iron, and sometimes wrought iron and hard steels, have no definite elastic limit. In any case where the elastic limit is unknown or ill-defined, we arbitrarily assume it to be one-half the ultimate strength, and factor a becomes 2. For nickel-steel and oil-tempered forgings the elastic limit becomes two-thirds of the ultimate strength, or even more, and the factor is accordingly reduced to $1\frac{1}{2}$.

The second factor, b, appearing in our equation is one depending upon the character of the stress produced within the material. The experiments of Wohler, conducted by him between the years 1859 and 1870 at the instance of the Prussian government, on the effects of repeated stresses, confirmed a fact already well known, namely, that the repeated application of a load which would produce a stress less than the ultimate strength of a material would often rupture it. But they did more. They showed the exact relation between the variation of the load and the breaking strength of the material under that variation. The investigation was subsequently extended by Weyrauch to cover the entire possible range of variation. Out of the mass of experimental data so obtained a rather complicated formula was deduced, giving the relation between the variation of the load (or rather the stress it produced), the strength of the material under the given conditions (which is generally known as the "carrying strength" of the material) and the ultimate strength. To Prof. J. B. Johnson, we believe, is due the credit of substituting for this formula a much simpler and more manageable one, which perhaps represents the actual facts with almost equal accuracy. Prof. Johnson's formula is as follows:

$$f = \frac{U}{2 - \frac{p'}{p}}$$

where f is the "carrying strength" when the load varies repeatedly between a maximum value, p, and a minimum value, p', and U is the ultimate strength of the material. The quantities p and p' have plus signs when they represent loads producing tension, and minus signs when they represent loads producing compression.

From what has just been said, it follows that if the load is variable in character, factor b must have a value,

$$b = \frac{U}{f} = 2 - \frac{p'}{p}.$$

Let us now see what this factor will be for the ordinary variations in loading.

Taking first a steady, or dead load, $p' = p$ and therefore $\frac{p'}{p} = \frac{1}{1} = 1$, and we have our factor,

$$b = 2 - \frac{p'}{p} = 2 - 1 = 1.$$

In other words, this factor may be omitted for a dead load.

Taking a load varying between zero and a maximum,

$$\frac{p'}{p} = \frac{0}{p} = 0,$$

and we have for our factor,

$$b = 2 - \frac{p'}{p} = 2 - 0 = 2.$$

Again, taking a load that produces alternately a tension and a compression equal in amount,

$$p' = -p \text{ and } \frac{p'}{p} = -1,$$

and we have, for our factor,

$$b = 2 - \frac{p'}{p} = 2 - (-1) = 2 + 1 = 3.$$

A fourth time, taking a load which produces alternately a tension and a compression, the former being three times the latter,

$$p = -3\,p' \text{ and } \frac{p'}{p} = -1/3,$$

and we have for our factor,

$$b = 2 - \frac{p'}{p} = 2 - (-1/3) = 2 + 1/3 = 2\,1/3.$$

Recapitulating our results, we may say that when the load is uniform, factor $b = 1$; when it varies between zero and a maximum, factor $b = 2$; when it varies between equal and opposite values, factor $b = 3$; when the load varies between two values, p and p', of which p' is the lesser factor, $b = 2 - \frac{p'}{p}$.

The experiments which have been made upon the effects of variable loads have almost without exception been made upon mild steel and wrought iron. Designers are in need of data based upon the results obtained with bronze, nickel steel, cast iron, etc.

It has already been noted that a stress many times repeated will rupture a piece when that stress is greater than the elastic limit, but less than the ultimate strength. It is also known that the application of a stress will change the elastic limit of a material, often by a very considerable amount. A material has really two elastic limits, an upper and a lower one, the latter often being negative in value (*i. e.*, an elastic limit in compression). Between these two limits there is a range of stress, which we may call the elastic range of the material, and within which the material is, so far as we can discover, perfectly elastic. It has been assumed, therefore, that under the influence of the varying or repeated load, this elastic range takes on certain limiting values depending on the character of the variation. So long as the variation is confined within these limits, the piece is safe. If, however, the range of variation of the stress exceeds the elastic range of the material under the given conditions, the piece breaks down. In confirmation of this view of the case, it has been found that pieces long subjected to alternating stresses have an elastic limit of one-third their ultimate strength, while pieces subjected to either repeated tensions, or compressions, only, have an elastic limit of one-half their ultimate strength.

From lack of data we cannot speak with authority on this matter, but it is probable that for material whose elastic limit is other than one-half its ultimate strength, Prof. Johnson's formula, and considerations derived from it, no longer hold. It is more than likely that with fuller knowledge of the subject we will find that the facts of the case may be more truly expressed by the formula,

$$f = \frac{nU}{1 - \frac{p'}{p}(1 - n)}$$

where n is the ratio of the elastic limit to the ultimate strength.

The third factor, c, entering into our equation, depends upon the manner in which the load is applied to the piece. A load suddenly applied to a machine member produces twice the stress within that member that the same load would produce if gradually applied. When the load is gradually applied, the stress in the member gradually increases, until finally, when the full load is applied, the total stress in the member corresponds to this full load. When, however, the load is suddenly applied, the stress is at first zero, but very swiftly increases. Since both the load and the stress act through whatever slight distance the piece yields, the product of the average total stress into this distance must equal the product of the load into this same distance. In order that the average stress should equal the load, it is necessary that the maximum value of the stress should equal twice the load. In recognition of this fact, we introduce the factor $c = 2$ into our equation when the load is suddenly applied.

It sometimes occurs that not all of the load is applied suddenly, in which case the factor 2 is reduced accordingly. If one-half the load were suddenly applied, the factor would be properly $1\frac{1}{2}$, and in general, if a certain fraction of the load, $\frac{n}{m}$, is suddenly applied, the factor is $1 + \frac{n}{m}$. Or, again, it may occur that friction, or some specially introduced provision, may prevent the sudden application of the load from having its full effect, in which case, if the amount of the reduction of this effect be known, or if it be possible to compute it, an appropriate reduction may be made in the value of this factor.

Sometimes, however, a load is applied not only suddenly, but with impact. In such a case it is highly desirable to compute the total stress produced by the load, and to substitute it for the load when obtaining the working section. Failing in this, it is necessary to make factor c more than 2, and sometimes as high as 10 or more. As an example of the possibilities arising in ordinary work, we may instance an elevator suspended by a wire rope of one square inch in section, and fifty feet long. If a truck weighing 500 pounds were wheeled over the threshold and allowed to drop two inches onto the elevator platform, a stress of over 10,000 pounds would be produced in the rope. Thus we see that in this very ordinary case arising in elevator service, this factor would need to be as much as 20.

The last factor, d, in our equation, we might call the "factor of ignorance." All the other factors have provided against known contingencies; this provides against the unknown. It commonly varies in value between 1½ and 3, although occasionally it becomes as great as 10. It provides against excessive or accidental overload, against unexpectedly severe service, against unreliable or imperfect materials, and against all unforeseen contingencies of manufacture or operation.

When we can compute the load exactly, when we know what kind of a load it will be, steady or variable, impulsive or gradual in its application, when we know that this load will not be likely to be increased, that our material is reliable, that failure will not result disastrously, or even that our piece for some reason must be small or light, this factor will be reduced to its lowest limit, 1½.

The conditions of service in some degree determine this factor. When a machine is to be placed in the hands of unskilled labor, when it is to receive hard knocks or rough treatment, the factor must be made larger. When it will be profitable to overload a machine by increasing its work or its speed in such a way as to throw unusual strains upon it, we are obliged to discount the probability of this being done by increasing this factor. Or again, when life or property would be endangered by the failure of the piece we are designing, this factor must be made larger in recognition of the fact. Thus, while it is 1½ to 2 in most ordinary steel constructions, it is rarely less than 2½ for a better grade of steel in a boiler. Even if property were not in danger of destruction, and the failure of the piece would simply result in considerable loss in output or wages, as in the case of the stoppage of a factory, it is best to increase this factor somewhat.

The reliability of the material in a great measure determines the value of this factor. For instance, in all cases where it would be 1½ for mild steel, it is made 2 for cast iron. It will be larger for those materials subject to internal strains, for instance for complicated castings, heavy forgings, hardened steel, and the like. It will be larger for those materials more easily injured by improper and unskillful handling, unless we know that the work will be done by skilled and careful workmen. It will be larger for those materials subject to hidden defects, such as internal flaws in forgings, spongy places in castings, etc. It will be smaller for ductile and larger for brittle materials. It will be smaller as we are sure that our piece has received uniform treatment, and as the tests we have give more uniform results and more accurate indications of the real strength and quality of the piece itself.

Of all these factors that we have been considering, the last one alone has an element of chance or judgment in it, except when we make an allowance for shock. In fixing it, the designer must depend on his judgment, guided by the general rules laid down.

Someone may ask at this point, why, if we introduce a factor for the elastic limit, do we also introduce a factor for repeated loads? It may be argued that if we keep the stress within the elastic limit, no harm will be done, no matter how often the load be repeated, and they are right. However, with a dead load acting upon a piece and straining it to its elastic limit, we have as a margin of safety the difference between its elastic limit and its ultimate strength. But when the load is a repeated load, of the same amount as before, the piece has no margin of safety, unless its section be increased, and it does not have the same margin of safety as it had in the first place, until its section is doubled.

Examples of Application of Formula

It remains to illustrate the method outlined for developing an "apparent factor of safety" by some practical examples. Let us take first the piston rod of a steam engine. It will be of forged steel, of simple form and reasonable size. The elastic limit will presumably be slightly more than one-half the ultimate strength, so factor $a = 2$. The rod will be in alternate tension and compression many times a minute and factor $b = 3$. The steam pressure will be applied suddenly (in a great many engines, on account of compression, only a part of this load is applied suddenly) and factor $c = 2$. And since the material is reliable, and the service definite and not excessively severe, factor $d = 1\frac{1}{2}$. Then,

$$F = 2 \times 3 \times 2 \times 1\tfrac{1}{2} = 18.$$

Taking next a steam boiler, our factor $a = 2$ as before. While the load in reality varies between zero and a maximum, since the load is steady in operation, and gradually applied, it is correct to make factor $b = 1$ and factor $c = 1$. Although we have an exceptionally reliable material, corrosion is likely to occur, and failure would be disastrous to life and property, so factor $d = 2\frac{1}{2}$ or 3, depending upon the workmanship. Then,

$$F = 2 \times 1 \times 1 \times 2\tfrac{1}{2} \text{ (or 3)} = 5 \text{ (or 6)}.$$

For our last illustration we will take the rim of a cast-iron flywheel for a steam engine. Factor $a = 2$, factor $b = 1$, and factor $c = 1$, for the load which is due to centrifugal force is constant. However, the material is the most unreliable with which the designer has to deal. It is probably spongy, and has great internal stress resulting from the cooling. It would be easy and profitable to increase both the power of the engine and the strain in the rim, by speeding it up. In ordinary cases we would make factor d equal to 3 or 4, but in this case the stress in the rim increases, not with the speed, but with the square of the speed, and it is entirely proper to make factor $d = 10$. So we have

$$F = 2 \times 1 \times 1 \times 10 = 20.$$

Table of Factors of Safety

The following table may be helpful in assisting the designer in a proper choice of the factor of safety. It shows the value of the four factors for various materials and conditions of service, and will give helpful hints to young designers as to what factors to use under similar circumstances.

CLASS OF SERVICE OR MATERIALS	Factor a	b	c	d	F
Boilers	2	1	1	2¼-3	4½- 6
Piston and connecting-rods for double-acting engines	1½-2	3	2	1½	13½-18
Piston and connecting-rod for single-acting engines	1½-2	2	2	1½	9 -12
Shaft carrying bandwheel, fly-wheel, or armature	1½-2	3	1	1½	6¾- 9
Lathe spindles	2	2	2	1½	12
Mill shafting	2	3	2	2	24
Steel work in buildings	2	1	1	2	4
Steel work in bridges	2	1	1	2½	5
Steel work for small work	2	1	2	1½	6
Cast-iron wheel rims	2	1	1	10	20
Steel wheel rims	2	1	1	4	8
MATERIALS	Minimum Values				
Cast iron and other castings	2	1	1	2	4
Wrought iron or mild steel	2	1	1	1½	3
Oil tempered or nickel steel	1½	1	1	1½	2¼
Hardened steel	1½	1	1	2	3
Bronze and brass, rolled or forged	2	1	1	1½	3

CHAPTER II

WORKING STRENGTH OF BOLTS*

Doubtless most mechanics have heard of the rule in use in many drafting offices, "Use no bolts smaller than ⅝-inch diameter, unless space or weight is limited." Or perhaps they may have heard pretty much the same thing stated in another way, namely, that a man will twist off a ½-inch bolt, trying to make a steam-tight joint. It is a matter of common experience among mechanics that a bolt has to be strained a good deal in order to make a tight packed joint, and that bolts must not only be made large enough to properly sustain the load due to the steam or water pressure, but to sustain this initial stress as well.

Bolts subject to tension are called upon for two different classes of service. Either they serve to hold two heavy and rigid flanges together, metal to metal, or they serve to compress a comparatively elastic packing, in order to make a joint steam-tight. In either case the bolt is under a considerable initial tension, due to the strain of screwing up, and hence the advisability of not making it smaller than ⅝ inch diameter. When the flanges are pressed together iron to iron, they are much more unyielding than the bolts. Hence when the bolts are screwed up, they are stretched a good deal more than the flanges are compressed. If we assume that the flanges are so heavy and unyielding that they cannot be compressed at all, the bolt is virtually a spring, and in order to produce in it a stress greater than the initial stress, we must pull so hard on the flanges as to separate them.

* MACHINERY, November, 1906.

The truth of this statement may be seen by referring to Fig. 1. The bolt shown clamps together the two flanges, and the nut is screwed down so tight that the bolt is stretched 0.001 inch. We will assume that the bolt is of such a size that the stress produced in it by this elongation is 1,000 pounds. If so, the flanges are pressed together with a force of 1,000 pounds. Supposing now that we pull the flanges apart in the manner shown by the arrows, with a force of 500 pounds. We

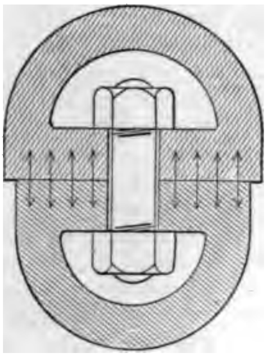


Fig. 1

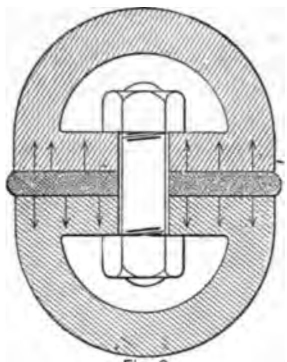


Fig. 2

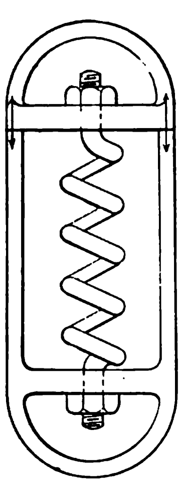


Fig. 3

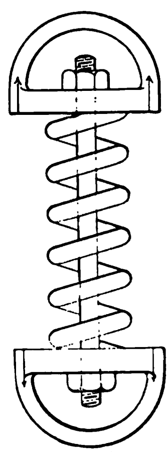


Fig. 4

Machinery, N.Y.

Figs. 1 to 4. Illustrations of Stresses in Bolts

cannot produce a greater stress in the bolt than 1,000 pounds until we stretch it a little more than it is stretched already. We cannot do this unless we separate the flanges, and it will take a pull of over 1,000 pounds to do that. Although the pull of 500 pounds adds nothing to the stress in the bolt, it does diminish the pressure between the flanges, which will be now the pressure holding them together, less the force pulling them apart, or 500 pounds. Exactly the same effects would have been noted had we chosen any other force than 500 pounds, provided it was less than 1,000 pounds. The stress in the bolt would not have been increased, but the pressure between the flanges would have been diminished by exactly the amount of the force applied.

On the other hand, supposing that we apply a force of 2,000 pounds to separate the flanges, we will find that the bolt will stretch under this load 0.002 inch, allowing the flanges to separate by only half that amount, and the pressure between them is nothing. It follows then that the stress in the bolt is now 2,000 pounds. If we had chosen any other force greater than 1,000 pounds, it would have been sufficient to separate the flanges, and the stress in the bolt would have been equal to the force applied. In other words, we find that the stress in the bolt is always either the initial stress, or else the force tending to separate the flanges, and it is always the greater of the two.

If, however, we place a piece of packing between the faces of the flanges, we find it is the packing rather than the bolt that is elastic. On tightening up the nut, the packing will be compressed say 0.010 inch. The stress in the bolt we will again assume to be 1,000 pounds. Applying a force of 500 pounds in the same manner as before, as shown in Fig. 2, we will not stretch the bolt very much in comparison to the amount by which we have already compressed the packing. Hence the packing will maintain its pressure against the flanges with almost undiminished force. We have simply added the 500 pounds to the 1,000 pounds stress already in the bolt. Exactly the same thing occurs when the force is increased to 2,000 pounds. The bolt will not give sufficiently to materially reduce the pressure due to the elasticity of the packing, and the stress in the bolt is the initial stress, plus the stress due to the force tending to separate the flanges.

The principles involved in the above discussion may be more easily understood by a reference to the illustrations, Figs. 3 and 4. The yielding members in Figs. 1 and 2 are represented in Figs. 3 and 4 as springs. A few moments consideration of the forces acting in each case will convince one of the truth of these two rules:

1. When the bolt is more elastic than the material it compresses, the stress in the bolt is either the initial stress or the force applied, whichever is greater.

2. When the material compressed is more elastic than the bolt, the stress in the bolt is the sum of the initial stress and the force applied.

Some experiments were made at the mechanical laboratories of Sibley College, Cornell University, some years ago, to determine the initial stress due to screwing up the bolts in a packed joint in an effort to get it steam-tight. The tests were made with ½-, ¾-, 1-, and 1¼-inch bolts. Twelve experienced mechanics were allowed to select their own wrenches, and tighten up three bolts of each size in the same way as they would in making a steam-tight joint. The bolts were so connected in a testing machine that the stress produced was accurately weighed. The wrenches chosen were from 10 to 12 inches long in the case of the ½-inch bolts, and ranged up to 18 and 22 inches long in the case of the 1¼-inch bolts. Thirty-six tests were made with each size of bolt, and while the results were not very close together in all cases, it was shown that the stress in the bolt due to screwing up varies about as its diameter, and that the stress produced in this way is often sufficient to break off a ½-inch bolt, but never anything larger.

Now since the stress varies about as the diameter of the bolt, and the area varies as the square of the diameter, it is evident that the larger the bolt is, the greater the margin of safety it will have. If the stress in a ½-inch bolt is equal to its tensile strength, the stress in a 1-inch bolt will be about one-half its tensile strength, and in a 2-inch bolt, one-quarter of its tensile strength. These are very low factors of safety, especially in the case of the sizes commonly used. When we come to add the stress due to the force tending to separate the flanges, there is an exceedingly small margin left, which is in many cases absolutely wiped out by any sudden increase of pressure due to water hammer, or some similar cause. If, however, we are to use the same factors of safety in designing the bolting for packed joints as we do in designing the other parts of machinery, we would use nothing smaller than 1¼-inch bolts under any circumstances, and generally bolts ½ inch or so larger. Such a proposition as this seems ridiculous in the light of successful practice, and so the writer was moved some time ago to investigate a great many flanged joints, some successful and some otherwise, with a view to obtain if possible some rule for proportioning the bolts so that they can always be relied upon.

From this investigation it was found that we may take for the "working section" of a bolt in a joint *its area at the root of the thread, less the area of a ½-inch bolt at the root of the thread times twice the diameter of the given bolt, in inches.* This working section must be sufficient to sustain, with a liberal factor of safety, the stress due to the steam load, or other force tending to separate the flanges. The largest unit stress, found by dividing the stress due to the load on the bolt produced by the steam pressure, or other such cause, by the working section of the bolt, is about 10,000 pounds per square inch. Let us take as an example of the application of this rule the case of a 1-inch bolt. Its area at the root of the thread is 0.550 square inch. Twice its diameter in inches is 2. The area of a ½-inch bolt at the root of the thread is 0.126 square inch. If from 0.550 square inch we subtract 2 × 0.126 square inch, the result, 0.298 square inch, is the working section of the 1-inch bolt. At 10,000 pounds to the square inch this bolt will sustain a stress of not quite 3,000 pounds, in addition to the stress due to screwing up.

There is reason, although not very sound, for this allowance. It has already been noted that a ½-inch bolt will sometimes be twisted off in screwing it up to make a steam-tight joint. It has also been noted that a 1-inch bolt will have twice the initial stress due to this cause that a ½-inch bolt will. Therefore if we could divide the area of the 1-inch bolt into two parts, 0.252 square inch of it would be strained to the breaking limit, resisting the initial stress, and the rest of the area, 0.298 square inch, would be free to take the other stresses that might come upon it. As a matter of fact, we cannot so divide the area, so the reasoning is not very sound, but inasmuch as the rule corresponds to the best practice in this regard, while theoretically more perfect rules would give us excessive and undesirable diameters, it seems better to use it than to adopt the familiar method of using a high factor of safety, and paying no attention to the initial stress. The latter method invariably leads one to grief, unless one is familiar

TABLE I. WORKING STRENGTH OF BOLTS

Diameter of Bolt, inches.	Area at Root of Thread, square inches.	Working Section, square inches.	Strength of Bolt, 5000 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.
1/2	.126	0	0	0	0	0	0	0
5/8	.202	.044	220	264	308	352	440	528
3/4	.302	.113	565	678	791	904	1,130	1,356
7/8	.420	.200	1,000	1,200	1,400	1,600	2,000	2,400
1	.550	.298	1,490	1,788	2,086	2,384	2,980	3,476
1 1/8	.694	.411	2,055	2,466	2,877	3,288	4,110	4,932
1 1/4	.893	.578	2,890	3,468	4,046	4,624	5,780	6,936
1 3/8	1.057	.710	3,550	4,260	4,970	5,680	7,100	8,520
1 1/2	1.295	.917	4,585	5,502	6,419	7,336	9,170	10,504
1 5/8	1.515	1.105	5,525	6,630	7,735	8,840	11,050	13,260
1 3/4	1.746	1.305	6,525	7,830	9,135	10,440	13,050	15,660
1 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 1/4	3.023	2.456	12,280	14,736	17,192	19,648	24,560	29,472
2 1/2	3.719	3.089	15,445	18,534	21,623	24,712	30,890	37,068
2 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 1/4	6.510	5.690	28,450	34,140	39,830	45,520	56,900	68,280
3 1/2	7.548	6.666	33,330	39,996	46,664	53,328	66,660	79,992

by long experience with the proper working stress to use with each size of bolt.

It will be found that for ordinary sizes of bolts the above rule works out in about the following form:

$$S = f\ (0.55\ D^2 - 0.25D)$$

where S = the strength of the bolt when used in a packed joint,
D = the diameter of the bolt in inches,
f = the safe working stress in pounds per square inch.

This formula is simple to use, and not difficult to remember. It must be borne in mind that it is only approximate, and not exact. As an example of its use, we will take the case of the 1-inch bolt again. Using a working strength of 10,000 pounds per square inch it will be found that

$$S = 10{,}000\ (0.55 - 0.25) = 3{,}000.$$

As the sizes of the bolts become greater, the formula gives results lower than they should be. It is very nearly correct for the common sizes of bolts, and on the safe side for the uncommon sizes.

Table I on the opposite page has been prepared, giving the diameters, least areas, working sections, and strengths of different sizes of bolts with U. S. standard threads. Thus from the table we find that the area of a 1¼-inch bolt, at the root of the thread, is 0.893 square inch. Its working section is 0.578 square inch, and its strength

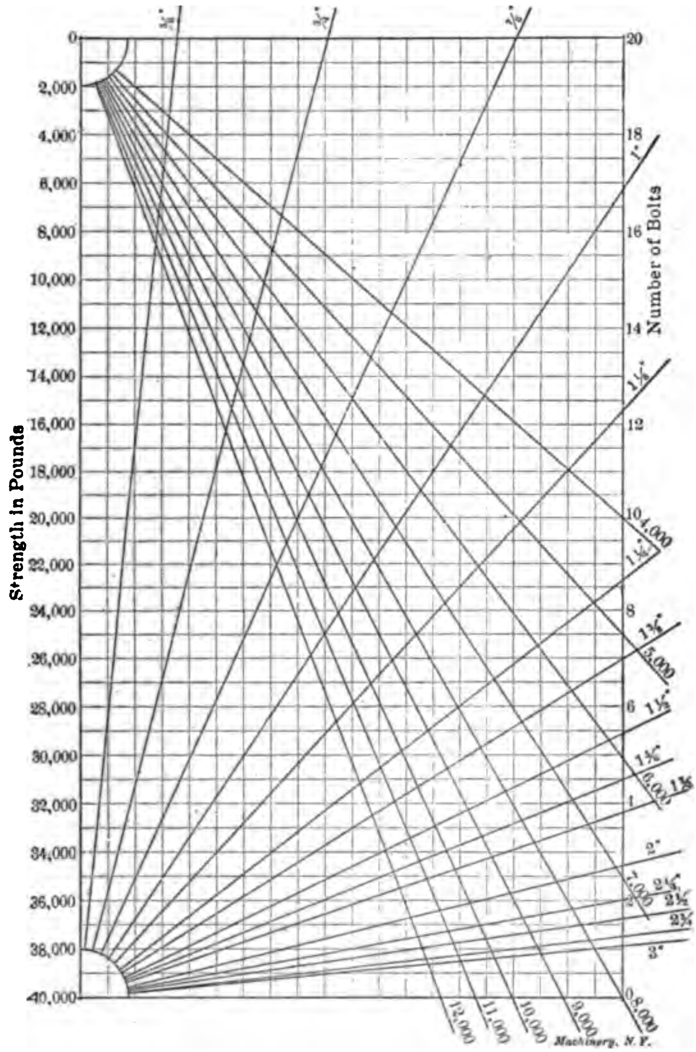


Fig. 5. Diagram of Working Strength of Bolts

at 8,000 pounds per square inch working stress is 4,624 pounds. As an example of the use of the table, let us design the bolting of a valve chest 8 inches wide and 12 inches long. Let us assume that the steam pressure is 100 pounds per square inch, and that ten bolts will be needed. The total load on the ten bolts will then be 8 × 12 × 100, or 9,600 pounds. The load per bolt is 960 pounds. Assuming a working stress of 6,000 pounds, we find that a 7/8-inch bolt is necessary.

The diagram, Fig. 5, gives the strength of any number of bolts, of any given size, with any required working stress when used in a packed joint. Suppose that it is required to find the strength of 20 3/4-inch bolts when used with a working stress of 6,000 pounds to the square inch. Finding the figure "20" at the right-hand side of the chart, we follow horizontally to the left on the heavy line, until we reach the diagonal line marked 3/4 inch. We then descend the vertical line which intersects the line 3/4 inch at the same point as does line 20, until this vertical line intersects the diagonal line marked 6,000. We then follow the horizontal line which intersects line 6,000 at this point, to the left-hand edge of the chart, where the adjacent figures indicate that the answer is 13,500 pounds. If we check the answer from the table we will find that the strength of a 3/4-inch bolt at 6,000 pounds working stress is 678 pounds, and therefore the strength of 20 of them is 13,560 pounds.

In designing flanged joints it must be remembered that an unlimited number of bolts cannot be crowded into a flange. The largest number of bolts that it is possible to use in a flanged joint and still have room to turn the nuts with an ordinary wrench is equal to the diameter of the bolt circle, divided by the diameter of the bolts, both in inches. A greater number of bolts than this can be used if necessary but a special form of wrench must be provided. The number of bolts generally used is about $D - 2\sqrt{D} + 8$, where D is the diameter of the interior of the pipe or cylinder in inches. For ordinary pressures this does not crowd the bolts too closely, although it puts them close enough together so that the flange will not leak under steam. The number of bolts actually taken for any flange is usually the nearest number divisible by four. For instance, for a water chamber of 60 inches diameter, the number of bolts obtained from the formula is $60 - 2\sqrt{60} + 8$, or 52½. The number of bolts actually taken might be 52 or 56, probably 52.

For our last problem let us take a rather extreme case. We will suppose the case of the water chambers of a high-pressure mining pump, 30 inches internal diameter, and subject to a pressure of 500 pounds per square inch. The number of bolts taken will be $30 - 2 \times \sqrt{30} + 8$, or taking the nearest number exactly divisible by four, 28 bolts. The area of the 30-inch circle is 0.7854×30^2, or 706.86 square inches. The total load on all the bolts due to the water pressure is 706.86 × 500, or 353,430 pounds. It will be noted that the diagram which we have already used does not extend above 40,000 pounds strength, but by multiplying both the number of pounds strength and the number of bolts by 10, the effective range can be increased to

400,000 pounds strength and 200 bolts. Taking, then, 35,300 instead of 353,000 at the left-hand edge of the chart, we follow to the right to the intersection with the diagonal line marked 8,000; then ascend the vertical line passing through this intersection till it meets horizontal line 2.8; we find that this point falls between the radial lines marked 1¾-inch and 2 inches, thus indicating that 28 bolts 1¾ inch diameter are not strong enough, and 28 bolts 2 inches diameter are stronger than is necessary. In fact, the vertical line we have been following intersects the line marked 2 inches at the horizontal line 2.4, indicating that 24 2-inch bolts would be required.

Stresses on Bolts Caused by Tightening of Nuts by a Wrench*

An interesting discussion on the stresses thrown upon bolts by the tightening of the nut by a wrench appeared in the *Locomotive*, July, 1905, and it may be considered proper to include the substance of this discussion in this chapter. While it is impossible to make any accurate computation of the tensile stress that is thrown upon a bolt by tightening a nut on its end, says the author of the article referred to, it is possible to obtain a roughly approximate estimate of that stress, when the nut is tightened under given conditions.

Let us suppose that a given screw is provided with a nut, which is to be turned up solidly against some resisting surface, so as to throw a tensile stress on the screw. Let the nut be turned by means of a wrench whose effective length is L inches. When the nut has been brought up pretty well into place, let us suppose that a force of P pounds, when applied to the end of the wrench in the most effective manner, will just move it. The work done by the man at the wrench, per revolution of the nut under these circumstances, is found by multiplying the force P by the circumference of the circle described by the end of the wrench. The wrench being L inches long, the circumference of this circle is $2\pi L$ inches, where $\pi = 3.1416$. Hence the work performed by the workman, per revolution, is $2\pi LP$ inch-pounds. Let us assume, for the moment, that the screw runs absolutely without friction, either in the nut, or against the surface where the nut bears against its seat. Then the work performed by the workman is all expended in stretching the screw, or deforming the structure to which it is attached. Hence, if the screw has n threads per inch of its length, and T is the total tension upon it in pounds, the work performed may also be expressed in the form $T \div n$; for in one turn the screw should be drawn forward $1 \div n$ inch, against the resistance T. Under the assumed conditions of perfection, the two foregoing expressions for the work done must be equal to each other. That is, we should have $2\pi LP = T \div n$, or

$$T = 2\pi nLP,$$

from which we could calculate the tension, T, on the bolt, if the screw were absolutely frictionless in all respects.

We come, now, to the matter of making allowances for the fact that in the real screw the friction is very far from being negligible. The

*MACHINERY, September, 1905.

actual tension that the given pull would produce in the bolt will be smaller than the value here calculated, and the fraction (which we will denote by the letter E) by which the foregoing result must be multiplied in order to get the true result is called the *efficiency* of the screw. The efficiency of screws has been studied both experimentally and theoretically; but the experimental data that are at present available are far less numerous than might be supposed, considering the elementary character and the fundamental importance of the screw in nearly every branch of applied mechanics. In the *Transactions* of the American Society of Mechanical Engineers, Volume 12, 1891, pages 781 to 789, there is a paper on screws by Mr. James McBride, followed by a discussion by Messrs. Wilfred Lewis and Arthur A. Falkenau, to which we desire to direct the reader's attention. In this place Mr. Lewis gives a formula for the efficiency of a screw of the ordinary kind, which appears to be quite good enough for all ordinary purposes, and which may be written in the form

$$E = 1 \div (1 + nd);$$

where d is the external diameter of the screw. If we multiply the value T, as found above, by this "factor of efficiency," the value of T, as corrected for friction, becomes

$$T = \frac{2\pi\, nLP}{1 + nd}$$

As an example of the application of this formula, let us consider the case in which a workman tightens up a nut on a two-inch bolt, by means of a wrench whose effective length is 50 inches, the maximum effort exerted at the end of the wrench being, say, 100 pounds. A standard two-inch bolt has 4.5 threads per inch; so that in this example the letters in the foregoing formula have the following values: $n = 4.5$; $L = 50$ inches; $P = 100$ pounds; $d = 2$ inches; and π stands for 3.1416. Making these substitutions, the formula gives

$$T = \frac{2 \times 3.1416 \times 4.5 \times 50 \times 100}{1 + 4.5 \times 2} = \frac{141{,}372}{10} = 14{,}137 \text{ pounds.}$$

That is, the actual total tension on the bolt, under these conditions, is somewhat over 14,000 pounds, according to the formula. As another example, let us consider a screw 1.5 inch in external diameter, with the nut set up with the same force and the same wrench as before. A standard screw of this size has six threads to the inch, so that the formula gives in this case

$$T = \frac{2 \times 3.1416 \times 6 \times 50 \times 100}{1 + 6 \times 2} = \frac{188{,}496}{13} = 14{,}500 \text{ pounds, about.}$$

Comparative Strength of Screw Threads*

A subject nearly related to the working strength of bolts is the comparative strength of screw threads. There has been considerable discussion from time to time among mechanics as to which of the three

*MACHINERY, October, 1906.

forms of thread, V, square, and Acme, is the strongest against shear. The following report of tests undertaken by C. Bert Padon at the James Millikin University, Decatur, Ill., to settle this question, with the idea of determining as nearly as possible with the means at hand just what relation these styles of thread bear to each other, will, therefore, prove of interest.

Each of the three forms was tested under two different conditions. First, a screw and nut of each form was made with threads all the same outside diameter, 15/16 inch, and with both screw and nut of the same axial length, 17/32 inch, and of the same material, the grade of steel commonly known in the shop as "machine steel." These three samples are shown at *a*, *b*, and *c* in Fig. 6, in which *a* is the V-thread, *b*, the Acme thread, and *c*, the square thread. In the second test all three screws were of the same root diameter, about 5/8 inch, and were

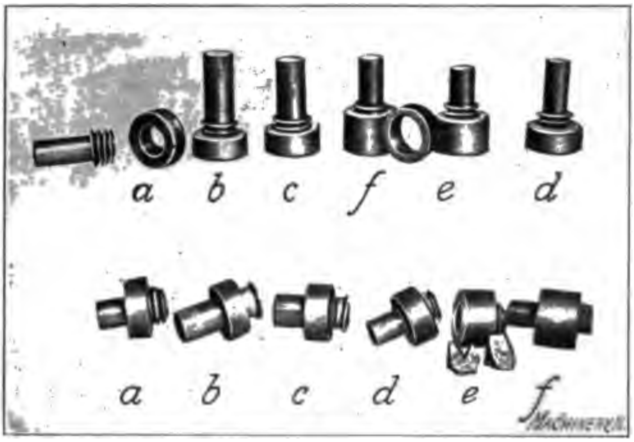


Fig. 6. Test Pieces used for Finding the Comparative Strength of Screw Threads

all made of gray cast iron, while the nuts were of machine steel. The length of the thread helix in each screw was such that each of the samples would present the same shearing area, the assumption being that they would shear at the root diameter of the screw since the screw was made of the weaker material. The different thicknesses of the nuts to suit the length of the helix required for this will be noticed in the halftone at *d*, *e*, and *f*, which show respectively the V-thread, Acme, and square samples. All the threads were made a snug fit, with the threaded length of the screw exactly the same as the thickness of the nut. The diameter of the shank was less than the root diameter of the thread in each case. The screws had all 6 threads per inch.

The upper row in the halftone shows the samples before testing, while the lower row shows the nature of the failure of each sample under test. A 50,000-pound Olsen machine was used. The nuts were supported on the ring shown with sample *f*, to allow room for the screw to drop through the nut when it failed, while pressure was applied at the top of the shank, which was carefully squared. The shank of the Acme thread screw *e* in the second set of three samples was not strong enough to withstand compression, but crushed before the thread gave way, at a pressure of 29,300 pounds. The fragments of the broken shank are shown. The screw was afterwards pushed through with a short piece of steel rod, failing at 29,600 pounds pressure. Table II gives the results of the test. As will be seen from the table, the Acme, or 29

TABLE II. RESULTS OF TESTS OF SHEARING STRENGTH OF SCREW THREADS.

Sample.	Style of Thread.	Material.		Thickness of Nut.	Diameter of Screw.	Breaking Load in pounds.
		Screw.	Nut.			
Threads same outside diameter and all 6 threads per inch.						
a	Sharp V	M. S.*	M. S.	$\frac{17}{32}$	$\frac{15}{16}$	29,980†
b	Acme	"	"	$\frac{17}{32}$	$\frac{15}{16}$	34,090‡
c	Square	"	"	$\frac{17}{32}$	$\frac{15}{16}$	23,880‡
Threads same root diameter, $\frac{5}{8}$ inch, and same area of section to resist shear. All are 6 threads per inch.						
d	Sharp V	C. I.*	M. S.	$\frac{1}{2}$	0.914	20,450‡
e	Acme	"	"	$\frac{13}{16}$	0.792	29,600‡
f	Square	"	"	1	0.792	25,550‡

* M. S. stands for Machinery Steel; C. I. for Cast Iron.
† Threads bent over in both screw and nut.
‡ Sheared at root of thread.

degree thread, makes the best showing in each case. The V-thread sample, *a*, evidently could not have failed in the way described without expanding the nut enough to allow the distorted threads to slip by each other. In this case, then, the thickness and strength of the nut play an important part. If the hole had been tapped in a larger piece of metal, it is difficult to believe that the thread would have failed by shearing, or in any other way, at a pressure less than that sustained by the Acme thread.

CHAPTER III

FLANGE BOLTS*

The calculations required for determining the number and size of bolts necessary to hold down a pillar crane are very instructive. The illustrations herewith, Figs. 7 to 9, show three examples of bolts used in this manner—that is, a series of bolts equally spaced around a circular flange intended to resist overturning. The first shows a pillar crane where the load has a tendency to overturn the pillar; the second, a radial drill where the pressure on the drill has a tendency to overturn the column, and the third a self-supporting chimney, where the wind pressure has an overturning effect.

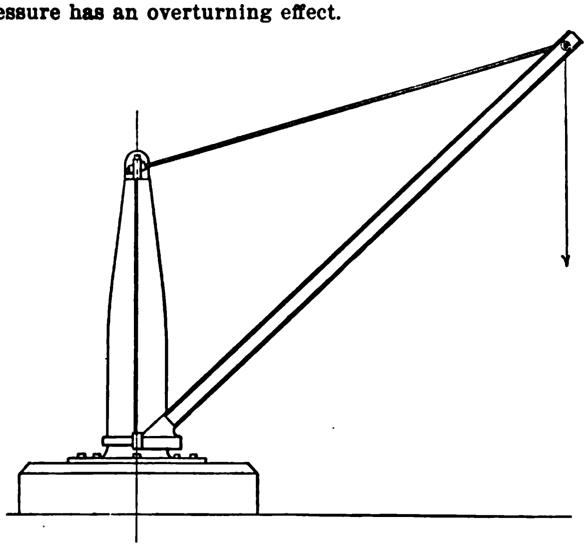


Fig 7. Jib Crane; Load has a Tendency to Overturn

It will be noted that there are two elements—one of tension due to the strain in the bolts, and one of compression due to the compression set up in the foundation. To exaggerate matters, suppose we were to place a layer of soft wood between the flange of the crane and the foundation. It is evident that the load would have a tendency to stretch the bolts on the side opposite the load and also to sink that part of the flange nearest the load, into the wood as in Fig. 10. The neutral axis would be a line drawn through the point where the flange and the foundation separate and at right angles to the direction of the load. On one side of this line we have the compression element due to the foundation, the bolts on this side having no value whatever. Start-

* MACHINERY, December, 1906.

ing at this neutral line and running the other way, we note that each bolt has a different value. To find the total value of the bolts, which constitutes our problem, we must add up these different values, and in consequence must know the position of the neutral axis.

If, instead of coming in contact with the foundation or bed-plate, the flange was supported by studs as shown in Fig. 11, we would have half of the studs in compression and the other half in tension, and the neutral axis would pass through the center of the bolt circle. If the flange had an annular surface inside of the bolts upon which to rest, as in Fig. 12, the neutral axis would lie somewhere inside of the larger circumference of this annular bearing surface as indicated. If conditions were as in Fig. 13, the neutral axis would be somewhere

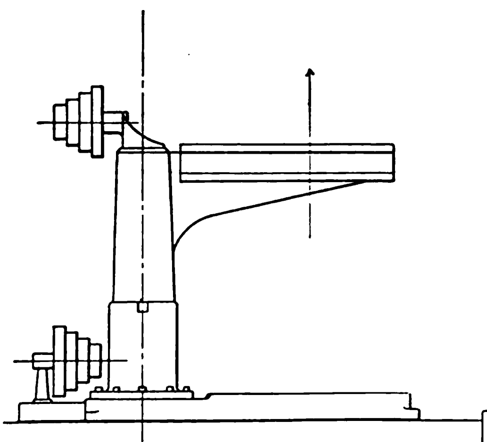


Fig. 8. Radial Drill: Pressure of Feed Tends to Overturn

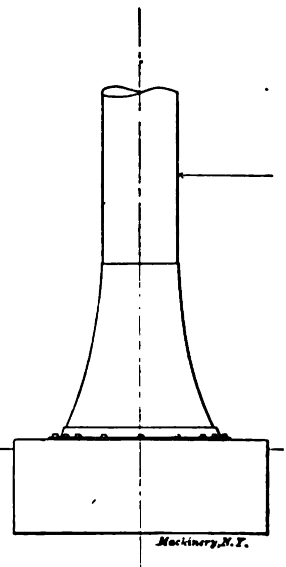


Fig. 9. Wind Pressure Tends to Overturn Chimney

between the bolt circle and the outside circumference of the flange, or possibly tangent to the bolt circle. Let us first determine the total bolt values for certain given positions of the neutral axis, and later look into the factors that control the position of this axis.

Referring to Fig. 10 it will be evident that the amount each bolt is stretched, and therefore the stress it resists, varies directly as its distance from the neutral axis. It will be further noted that the moment of any one bolt as regards the neutral axis is directly proportional to the square of its distance from this axis, because the moment of any bolt is the product of the force it exerts, and the distance through which it acts. Consequently, if we could easily determine the value of the mean square, as we surely can, we will then only have to multiply it by the number of bolts to obtain the sum of the squares.

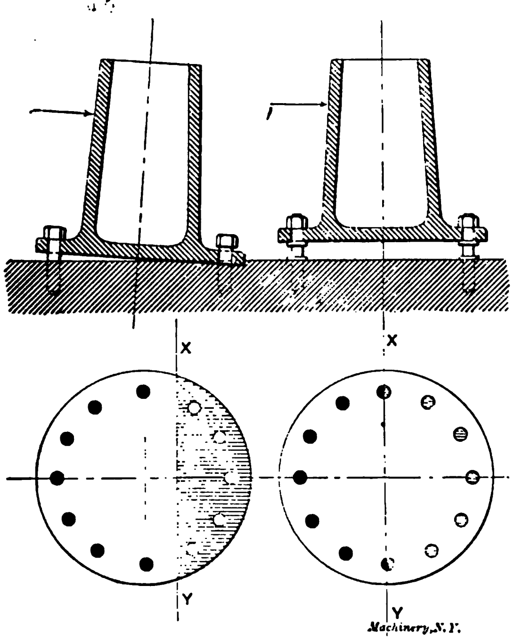
Consider six bolts, as in Fig. 14, spaced equidistant on a circle of radius = 1. Let the maximum stress in any bolt be 8,000 pounds, and take the neutral axis as being tangent to the bolt circle. Hence we have the following:

TABLE III. SIX BOLTS

Bolt No.	Distance.	Square of Distance	Stress.	Moment.
1...........	2.00	4.00	8,000	16,000
2...........	1.50	2.25	6,000	9,000
3...........	.50	.25	2,000	1,000
4...........	....	....	.....	......
5...........	.50	.25	2,000	1,000
6...........	1.50	2.25	6,000	9,000
Totals......	....	9.00	.....	36,000

This gives a value for the mean square 9.00 ÷ 6 = 1.50. If the radius were twice as great, the mean square would, of course, be four times as great. This table, therefore, indicates that the

$$\text{Mean square} = 1.50\,R^2 = \tfrac{3}{8}\,D^2 \qquad (1)$$



Figs. 10 and 11. Location of Neutral Axis under Varying Conditions

The total of these square values represents the moment of inertia of the set of bolts, and if we multiply the sum by the maximum stress and divide it by the distance of the point at which that stress acts, *viz.*,

D, we obtain the moment of resistance just as we do in figuring the strength of a beam in flexure. Hence we have the following:

Moment of inertia = number of bolts × mean square = $1.50\,R^2N = \frac{3}{8}\,D^2N$, and

$$\text{Moment of resistance} = \frac{1.50\,R^2\,N\,S}{D} = \tfrac{3}{8}\,N\,D\,S \qquad (2)$$

where S is the maximum total stress in any bolt.

TABLE IV. TWELVE BOLTS

Bolt No.	Distance.	Square of Distance.	Stress.	Moment.
1..........	2.000	4.000	8,000	16,000.0
2..........	1.866	3.482	7,464	13,928.3
3..........	1.500	2.250	6,000	9 000,0
4..........	1.000	1.000	4,000	4,000.0
5..........	.500	.250	2,000	1,000.0
6..........	.134	.018	536	71.8
7..........	.....	.....	.....	....
8..........	.134	.018	536	71 8
9..........	.500	.250	2,000	1,000.0
10..........	1.000	1.000	4,000	4,000.0
11..........	1.500	2.250	6,000	9,000.0
12..........	1.866	3.482	7.464	13,928.3
Totals......	.....	18.000	.....	72,000.2

TABLE V. TWENTY-FOUR BOLTS

Bolt No.	Distance.	Square of Distance.	Stress.	Moment
1	2.000	4.000	8 000	16,000
2–24	1.966	3.865	7 864	15 461
3–23.......	1.866	3.482	7,464	13.928
4–22.......	1.707	2.914	6,828	11,655
5–21.......	1.500	2.250	6,000	9,000
6–20.......	1.259	1.585	5,036	6,340
7–19.......	1.000	1.000	4,000	4 000
8-18	.741	.549	2,964	2,196
9–17.......	.500	.250	2 000	1,000
10–16.......	.293	.086	1,172	343
11–15.......	.134	.018	536	72
12–14.......	.034	.001	136	5
Totals......		36.000	....	144,000

Applying this to Fig. 14 we have $\frac{3}{8} \times 6 \times 2 \times 8{,}000 = 36{,}000$, which is verified by Table III where the moment of each bolt is computed separately.

Similarly we may take twelve bolts, and considering that the maximum stress on any bolt is 8,000, the distance to, and stress in, each bolt are as given in Table IV.

By equation (2) we have

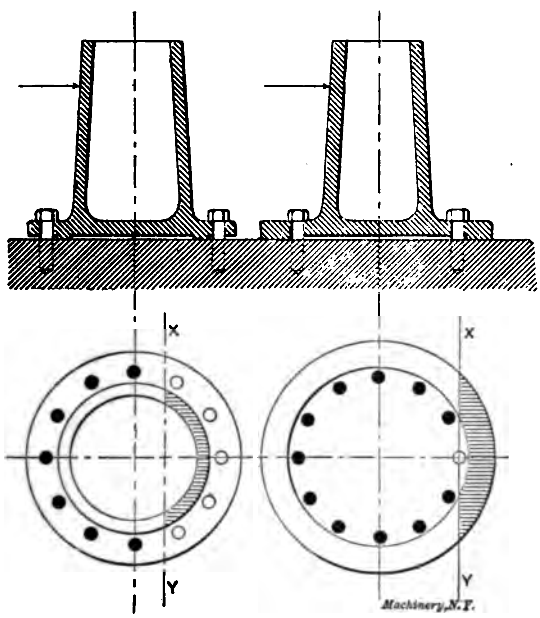
Moment of resistance $= \frac{3}{8}\,NDS = \frac{3}{8} \times 12 \times 2 \times 8{,}000 = 72{,}000$, which

agrees with the result found by computing the moment of each bolt separately, as Table IV shows. The value of the mean square is, by equation (1), equal to 1.5 R^2, and the table gives in this case 18 ÷ 12 = 1½. This table, then, verifies our formulas for both mean square and total moment exerted by the twelve bolts.

For twenty-four bolts the results are the same, and Table V on the previous page is given to show that the formulas are applicable to any number of bolts.

Moment $= \frac{3}{8} NDS = \frac{3}{8} \times 24 \times 2 \times 8{,}000 = 144{,}000.$

Mean square $1.5\, R^2 = \dfrac{36}{24} = 1.5.$



Figs. 12 and 13. Location of Neutral Axis under Varying Conditions

The foregoing applies only where the neutral axis is tangent to the bolt circle, but knowing what the moment of a series of bolts is when the neutral axis is in this position, it is a simple matter to determine the moment for any other known position.

Referring to Fig. 16, let the neutral axis have the position *XY*. It will be evident that the moment depends upon the mean square of a series of distances, which are composed of two parts, *viz.*, a constant ϕ and a variable such as a, b, c, d. Hence for the total of the squares we have

$$(\phi + 0)^2 + (\phi + a)^2 + (\phi + b)^2 + (\phi + c)^2 + \ldots$$

which may be written

$$N\phi^2 + 2\phi\,(a + b + c + \ldots) + a^2 + b^2 + c^2 + \ldots$$

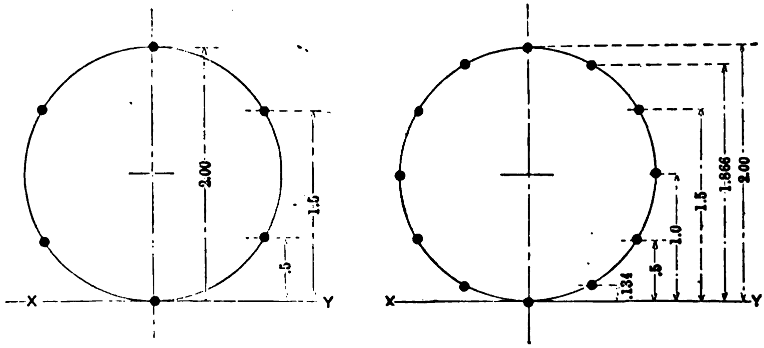
Referring to Fig. 16 it will be seen that the average of 0 and $f =$ radius; a and $e =$ radius; b and $d =$ radius, etc., which means that the sum of $a + b + c \ldots = NR$, which **may be written for the second term of the previous expression. For the third term we may write $a^2 + b^2 + c^2 \ldots = \frac{3}{8} ND^2$ by equation (1) which we have already outlined.**

Hence we may write for the sum of the squares

$$N\phi^2 + N\phi D + \tfrac{3}{8} ND^2.$$

To obtain the moment of resistance we **must divide this by the distance of the point of maximum stress from the neutral axis and multiply it by the maximum stress. Therefore**

$$\text{Moment of resistance} = \frac{N\,(\phi^2 + \phi D + \frac{3}{8} D^2)\,S}{\phi + D} \qquad (3)$$



Figs. 14 and 15. Finding the Stress on the Bolts for Six and Twelve Bolts

When the neutral axis lies inside of the bolt circle we have $(0 - \phi) + (a - \phi) + (b - \phi) + (c - \phi) + \ldots$ which may be written $N\phi^2 - 2\phi\,(a + b + c + \ldots) + a^2 + b^2 + c^2 + \ldots$ and for the moment we have

$$\text{Moment of resistance} = \frac{N\,(\phi^2 - \phi D + \frac{3}{8} D^2)\,S}{D - \phi} \qquad (4)$$

The only remaining factor to determine is the position of the **neutral** axis so that we can apply the above formula. **In the first place it** would be well to point out certain conditions **that render this some**what uncertain. In these, as in most all bolt calculations, the initial strain set up in a bolt by tightening the nut cannot be definitely determined. Then again, the assumption that each bolt is strained directly in proportion to its distance from the neutral axis necessitates that the flange be absolutely rigid. While a heavy cast-iron flange with a large fillet, and possibly a few stiffening ribs, is about as **rigid** as anything we might find in construction work, yet it is not **absolutely** rigid. Finally we might mention the weight of the structure or pillar that is borne by the flange. This factor has a tendency to increase the element of compression and decrease the element of tension to a slight extent.

It is, however, much more practical and advisable to determine the position of the neutral axis as closely as possible than to attempt to determine these several uncertain quantities. The formula will at best give uniformity of results, and if experience points out that our results are correct in one case, they will also be correct for other cases when they apply to similar conditions.

It is an accepted fact that in all cases of flexure the neutral axis passes through the center of gravity of the section. This means that

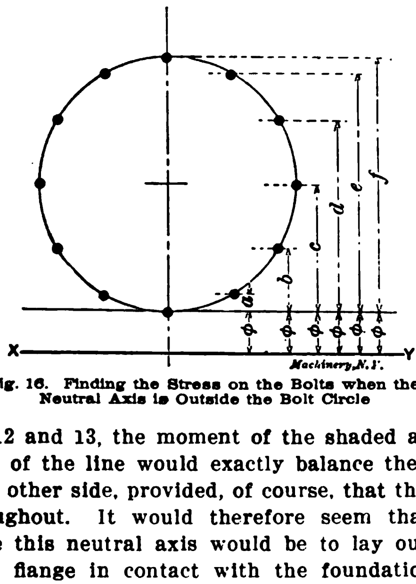


Fig. 16. Finding the Stress on the Bolts when the Neutral Axis is Outside the Bolt Circle

in Figs. 10, 11, 12 and 13, the moment of the shaded area in compression on one side of the line would exactly balance the moment of the bolt areas on the other side, provided, of course, that the same material were used throughout. It would therefore seem that the practical method to locate this neutral axis would be to lay out the bolts and that part of the flange in contact with the foundation and find the center of gravity, making allowance for the fact that the weight per unit of area of tension and compression areas should be taken as proportional to their respective stresses per square inch.

CHAPTER IV

FORMULAS FOR DESIGNING RIVETED JOINTS*

In designing a riveted joint it is first necessary to know the pressure per square inch and the diameter of the cylinder, or the thickness of the metal.

In the following formulas the notation below is used:

t = thickness of the plate,
P = pressure to be resisted by 12 inches of the joint,
D = diameter of the cylinder, in inches,
a = pressure per square inch,
S = ultimate shearing strength of rivet or plate,
p = pitch of rivets,
f = factor of safety = ratio of bursting pressure to working pressure,
T = tensile strength of the plate,
d = diameter of the rivet hole,
B = bearing value of the plate,
l = distance from center of rivet to the edge of the plate,
b = diagonal pitch,
e = efficiency of the joint,
n = number of rows of rivets.

The value of some of the above quantities are as follows:

$S = 0.75$ to 0.80 of the tensile strength of the plate, for a rivet in single shear; a rivet in double shear is taken as 1¾ times one in single shear. As the rivets of a joint are protected from deterioration while the plates are thinned by wear, the shearing strength of a rivet is frequently taken as equal to the tensile strength. Also, in determining the shearing value of a rivet from the tensile strength of the plates, if iron rivets are being used with steel plates, the shearing value of the rivet must be determined from the tensile strength of iron, and not from the tensile strength of steel.

$f = 6$, for cylinders of moderately good materials and workmanship. The following additions should be made for structural defects when they exist, *viz.*, an addition of 25 per cent when the rivets are not good and fair in the girth seams; 50 per cent if the rivets are not good and fair in the longitudinal seams; 100 per cent if the seams are single riveted; and 200 per cent when the quality of materials or workmanship is doubtful or unsatisfactory.

T = for steel plates about 55,000 to 60,000 pounds per square inch; for wrought iron about 45,000 pounds per square inch. The tensile strength of wrought iron plates across the grain is on an average 10 per cent less than along the grain.

$B = \dfrac{3T}{2}$ for ordinary bearing, and, $2T$ for web bearing.

* MACHINERY, April, 1906.

The formulas apply to joints having only one pitch.

If the thickness, t, of the plate is known

$$d = \frac{\sqrt{t \times 92}}{8} + \frac{1}{16} \tag{5}$$

$$p = \frac{d^2 \times 0.7854 \times S \times n}{t \times T} + d \tag{6}$$

A riveted joint is twice as strong against circumferential rupture as against longitudinal rupture. Therefore, a cylinder which requires a double riveted lap joint for the longitudinal seams will only require a single riveted lap joint of the same diameter and pitch for the circular seams.

$$P = 6aD \tag{7}$$

Now choose a trial value, d, for the diameter of the rivet hole; commercial rivets vary by 1/16 inch up to 7/8 inch, more commonly by 1/8 inch; 5/8 inch, 3/4 inch, 7/8 inch, and 1 inch being the most frequently used. Remember that the cold rivet is 1/16 inch less in diameter than the hole, and that the diameter of the hole must be greater than the thickness of the plate, otherwise the punch will not be likely to endure the work of punching.

Substitute the chosen value of d in the following equations until the proper pitch is found. Six diameters of the rivet is the maximum pitch for proper calking, owing to the liability of the plates to pucker up when being calked.

$$p = \frac{9.4248d^2S}{Pf} \tag{8}$$

for single riveted lap joints.

$$p = \frac{18.8496d^2S}{Pf} \tag{9}$$

for double riveted lap joints and single riveted butt joints with two cover plates.

$$p = \frac{37.6992d^2S}{Pf} \tag{10}$$

for double riveted butt joints with two cover plates.

Notice that twice the result found by (8) is equal to the result found by (9), and that four times the result found by (8) is equal to the result found by (10).

Having now the pitch and diameter of the rivet, try the percentage of strength, or efficiency, of the plate, by,

$$e = \frac{p - d}{p}, \tag{11}$$

and if the result is not satisfactory, try a new diameter of rivet and find its corresponding pitch as before.

The strength or efficiency of a well designed single riveted joint may

be 56 per cent; of a double riveted joint 70 per cent; and of a triple riveted joint 80 per cent of that of the solid plate.

In determining the pitch of rivets and the efficiency of joints with punched holes, the larger diameter of the punched hole should be used in determining the efficiency, and the smaller diameter, or the diameter of the rivet, should be used in determining the bearing value, etc., of the rivet.

$$t = \frac{f \times P \times p}{12 \times T(p - d)} \qquad (12)$$

Now check the pitch, diameter and thickness **by substituting these values in (6).**

If the rivet fills the hole, and is well driven, there is no bending moment exerted on it unless it passes through several plates. Practical tests have shown that rivets cannot be made to surely fill the holes if the combined thickness of plates exceeds 5 diameters of the rivets.

Butt joints are generally used for plates over ½ inch in thickness. Where one cover plate is used on a butt joint, its thickness is 1⅛ times the thickness of the plate. Where two cover plates are used each should be about ⅝ of the plate thickness.

Now check the diameter, thickness and pitch for crushing by

$$\frac{12dtB}{P} = \text{or} > Pf \qquad (13)$$

for single riveted joint.

$$\frac{24dtB}{P} = \text{or} > Pf \qquad (14)$$

for double riveted joint.

The distance from the center of the rivet to the edge **of the plate** after being beveled for calking should be $1\frac{1}{2}d + \frac{1}{8}$ **inch. Check by**

$$l = \frac{fPp}{24tS} \qquad (15)$$

and if the result is greater than $1\frac{1}{2}d$, use it, adding ⅛ inch.

The diagonal pitch of rivet of a seam having several rows **of rivets,** all of the same pitch, is generally equal to 0.75 to 0.80 **of the straight** pitch, and should not be less than

$$b = \frac{(p \times 6) + (\text{dia. of rivet} \times 4)}{10} \qquad (16)$$

Diagonal Seams

The ratio of strength, R, of an inclined or diagonal seam **to that of** a straight seam, or ordinary longitudinal seam, may be found by

$$R = \frac{2}{\sqrt{\text{cos of angle of inclination}} \times 3 + 1} \qquad (17)$$

Rivet Material

It is necessary to make the rivets of the same material as the plates to prevent corrosive wasting from galvanic action. That is, iron rivets should be used with iron plates, steel rivets with steel plates, and copper rivets for copper plates.

Elastic Limit of Riveted Joints of Steam Boilers

The riveted seams of a steam boiler should cease to be steam tight for some time before the internal pressure is equal to the elastic limits of the plate. If a boiler were stretched beyond the elastic limit of the material, the rivet holes would become stretched and the joints of the plates would be disturbed, resulting in large leakage from the rivet holes and seams.

The elastic limit of riveted joints of wrought iron and mild steel is as follows:

Best quality of mild steel, 32,000 to 34,000 pounds per square inch.

Ordinary quality of mild steel, 28,000 to 30,000 pounds per square inch.

Best quality of wrought iron, 24,000 to 26,000 pounds per square inch.

Ordinary quality of wrought iron, 20,000 to 22,000 pounds per square inch.

Weight of Seams or Riveted Joints of Cylinders

The weight of seams of cylinders varies according to their proportions and must be calculated in each particular case. A rough approximation of the weight of riveted seams may, however, be obtained by increasing the weight of the cylinder by 1/6, if formed with single riveted circumferential seams and double riveted longitudinal seams; and by 1/5, if formed with double riveted circumferential seams and triple riveted longitudinal seams.

Gripping Power of Rivets

When two plates are fastened together by properly proportioned and well closed rivets, the frictional adhesion of the plates depends upon the longitudinal tension of the rivets. The adhesion of the plates or their resistance to sliding, per square inch of sectional area of the rivets, is in a general way equal to 2/9 of the ultimate tensile strength of the rivet.

Punched Holes

The distressing effect on the plate due to punching may generally be neutralized by countersinking ⅛ inch in width around the rivet hole with a reamer. All rivet holes shall be so accurately spaced and punched that when several parts are assembled together, a rivet 1/16 inch less in diameter than the hole can generally be entered hot into any hole. In the better class of plate work it is now the practice to drill rivet holes in plates after the plates are in place, so that the holes are sure to be fair. In some cases the holes are punched to a smaller diameter, and then drilled out to final size after the plates

are in place. In either case the plates are afterwards separated, and the burr left by the drill removed.

The effect of clearance between the punch and die is to produce a conical hole in the plate. The punched plates are generally arranged with the large ends of the holes outside or the small ends together.

Comparative Strength of Boiler Joints*

An interesting fact about riveted joints, which will prove instructive to discuss more fully, is that the stress in the second row of rivets always amounts to more than that in the first row. This is the case when a triple joint is used, having a narrow outer butt strap and a wide one inside, and when the pitch in the second row is half the pitch of the first, and all rivets have the same diameter. We will here show how to calculate the stress of the shell plate at both rows of rivets. Take the joint shown in Fig. 17, *i. e.*: **shell, ⅝ inch, rivets, 1 1/16**

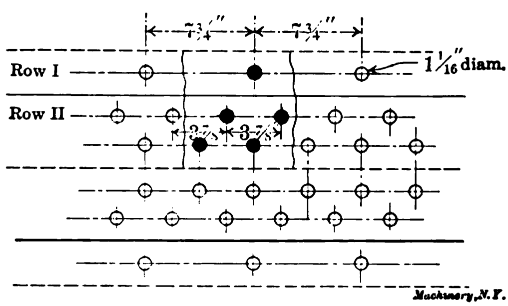


Fig. 17. Joint to be Investigated

inch = 1.06 inch, about; radius of shell, 29 inches; pitch, 7¾ inches; pressure, 200 pounds per square inch.

Row I. Pull along one pitch = 7.75 × 29 × 200 = 45,000 **pounds.**

Length of plate = 7¾ — 1 1/16 = 6.68 inches.

$$\text{Tearing of plate} = \frac{45\,000}{6.68 \times 0.625} = 10{,}780 \text{ pounds per square inch.}$$

$$\text{Shearing of rivets} = \frac{45{,}000}{9 \times \frac{\pi}{4} \times 1.06^2} = 5{,}650 \text{ pounds per square inch.}$$

Row II. Pull in second row of rivets is **45,000 pounds less the amount taken away by rivet in (I); that is, the amount transmitted** in row (I) through one rivet to the butt straps.

$$45{,}000 - 5{,}650 \times \frac{\pi}{4} \times 1.06^2 = 45{,}000 - 5{,}000 = 40{,}000 \text{ pounds.}$$

Length of plate = 7¾ — 2 × 1 1/16 inch = 5⅝ = 5.625 **inches.**

$$\text{Tearing of plate} = \frac{40{,}000}{5.625 \times 0.625} = 11{,}380 \text{ pounds per square}$$

inch or about 5½ per cent more than in row (I).

* Machinery, June, 1907.

To avoid this there are two methods possible; one of them is shown in Fig. 18. Use the same pitch at row (I), but increase the pitch at rows (II) and (III), all rivets remaining the same diameter.

Row I. Pull along one pitch $= 7.75 \times 29 \times 200 = 45{,}000$ pounds.

$$\text{Tearing of plate} = \frac{45{,}000}{6.68 \times 0.625} = 10{,}780 \text{ pounds per square inch.}$$

$$\text{Shearing of rivets} = \frac{45{,}000}{7 \times \frac{\pi}{4} \times 1.06^2} = 7{,}275 \text{ pounds per square inch}$$

$$\text{Factor of safety} = \frac{38{,}000}{7{,}275} = 5.22.$$

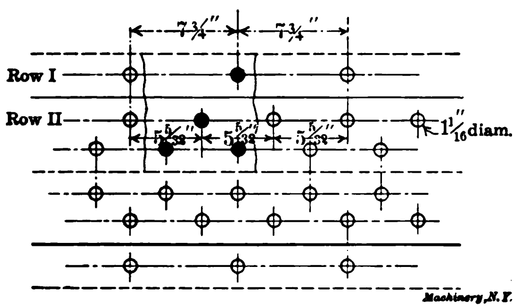


Fig. 18. Joint Re-designed to give Less Stress in Row II than in Row I

Row II. Pull along one pitch $= 45{,}000 - 7.275 \times \frac{\pi}{4} \times 1.06^2 = 38{,}575$ pounds.

Length of plate $= 7.75 - 1.5 \times 1.06 = 7.75 - 1.6 = 6.15$ inches.

$$\text{Tearing of plate} = \frac{38{,}575}{6.15 \times 0.625} = 10{,}050 \text{ pounds per square}$$

inch or 7 per cent less than in row (I).

A second method, shown in Fig. 19, consists in increasing the pitch and diameter of rivets in the first row, or using smaller rivets in the second and third rows. Of course, this is somewhat awkward, on account of it being necessary to change the riveting tools (but on the European continent this is the usual practice) for the two sizes of rivets. If, however, we keep the 1 1/16-inch rivets in the first row, and use 15/16-inch rivets in the second and third rows, we get:

Row I. Pull alone one pitch $= 7.75 \times 29 \times 200 = 45{,}000$ pounds.

$$\text{Area of rivets} = \left(1 \times \frac{\pi}{4} \times 1.06^2\right) + \left(8 \times \frac{\pi}{4} \times 0.94^2\right) =$$

$$0.883 + 5.550 = 6.433 \text{ square inches.}$$

Length of plate $= 7\frac{3}{4} - 1\ 1/16 = 6.68$ inches.

$$\text{Tearing of plate} = \frac{45{,}000}{6.68 \times 0.625} = 10{,}780 \text{ pounds per square inch.}$$

$$\text{Shearing of rivets} = \frac{45{,}000}{6.433} = 7{,}000 \text{ pounds per square inch.}$$

Row II. Pull $= 45{,}000 - 0.883 \times 7{,}000 = 38{,}820$ pounds.

Length of plate $= 7.75 - 2 \times 15/16 = 5.875$ inches.

$$\text{Tearing of plate} = \frac{38{,}820}{5.875 \times 0.625} = 10{,}570 \text{ pounds per square}$$

inch or $1\frac{3}{4}$ per cent less than in row (I).

If, instead of using smaller diameter rivets in the second and third rows, we keep 1 1/16-inch rivets, but increase the diameter of rivets

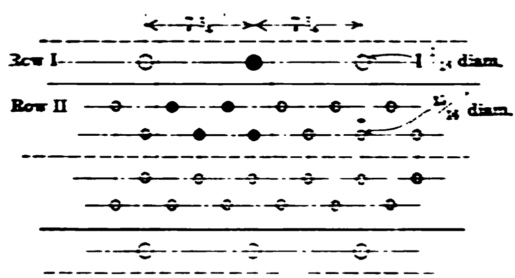


Fig. 19. Joint in which the Stresses are Nearly Equalized

in the first row to 1 3/16 inch, and also the pitch to give the same percentage, similar results would be obtained. In a triple butt joint with straps of equal width, the stress in the second row would always be less than in the first row; on this account, therefore, it is unnecessary to make any calculations of row (II).

English Practice

In England it is customary to use higher working stresses than in the United States; while here plates are used with a tensile strength of 55,000 pounds per square inch, with a factor of safety of 5, they use there plates of not less than 60,000 pounds, allowing a factor of safety of 5 for double butt joints, and a factor of safety of $4\frac{1}{2}$ for triple butt joints. In England they never use iron rivets, but always steel rivets, with a shearing strength of 50,000 pounds per square inch, and a factor of safety of 5, which equals 10,000 pounds per square inch, under pressure. It is also their rule to take the diameter of the steel rivets from $1.1\sqrt{T}$ to $1.2\sqrt{T}$, where T equals thickness of plate in inches; so that in the previous case they would have used $1.2\sqrt{0.625} =$ 15/16 inch for the diameter of the rivets, and the riveting as shown in Fig. 18.

CHAPTER V

CALCULATING THE STRENGTH OF A MOUTHPIECE RING AND COVER*

There are thousands of digesters, vulcanizers and other similar vessels in use working under considerable pressure. Accidents to these, particularly the bursting of the head or of the ring to which it is clamped, are almost as common as boiler explosions, and oftentimes do considerable damage and sometimes result in the loss of life. There are one or two points relating to the problem of designing vessels of this kind which do not always receive proper attention from the men

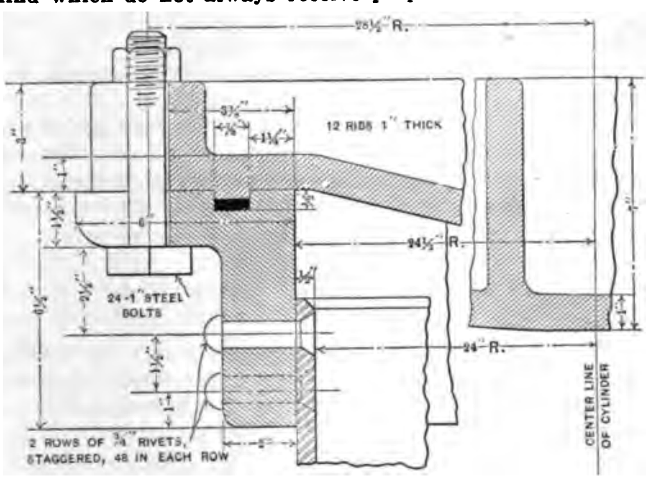


Fig. 20. Design of Mouthpiece Ring and Cover

responsible for the calculations involved, and it is with the object of calling attention to some of these points that we give herewith the calculations made for figuring the strength of a cover and mouthpiece ring.

Fig. 20 shows the essential features of the design. The body of the cylinder itself is a welded steel tube 4 feet in diameter, ½ inch thick, and about 7 feet long. To this is riveted a mouthpiece ring, presumably of cast iron, having slots for 24 one-inch steel bolts by which the cover is made fast. The important dimensions are shown. No other information being at hand, the material of the cover is taken as cast iron, while the shell is supposed to be made of steel having a tensile strength about equal to that of boiler plate. The following data as to the strength of the materials are assumed:

*MACHINERY, November, 1906.

the inner edge, and thus expanding the lower portion of the hub. From this distortion the principal stress is that of tension in the hub. The way in which the part fails under these circumstances is shown in Fig. 22. "Hub cracks" are introduced running from the lower edge up into the body of the ring, sometimes passing through the rivet holes and sometimes avoiding them. The formula given in the *Locomotive* for determining the maximum tensile stress at this point is as follows:

$$F = \frac{(mNE + LD)\,(h - a)}{6.2832\,(I - a^2A)}$$

in which F = the tensile stress per square inch,
m = the distance from the gasket to the bolt circle,
N = the total number of the cover bolts,
E = the excess of the actual tension on each cover bolt above

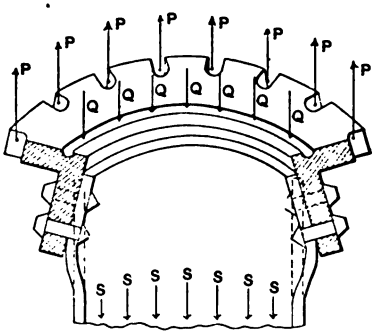
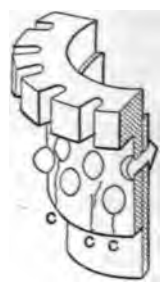


Fig. 21. Stresses on the Ring



Machinery, N.Y.

Fig. 22. Usual Manner of Failure

that due to the steam load (1,200 pounds is suggested in the article referred to),
L = total steam load,
D = the distance from the inner edge of the ring to the bolt circle,
h = height of the ring,
a = the distance from the center of gravity of the ring section to the face of the ring,
I = the moment of inertia of the ring section about axis OX (see Figs. 23 and 24),
A = area of the ring section.

Those letters which refer to dimensions will be found in Fig. 23, where a diagrammatical sketch of the ring section is given. The quantity of the denominator $(I - a^2A)$ amounts to the same as the moment of inertia of the section about the neutral axis. It is put in the form given for convenience in calculating, the issue of the *Locomotive* referred to having a table of moments of inertia of rectangles provided for the purpose. No explanation need be given here of the methods of finding the center of gravity and moment of inertia of a section. This will be found discussed in any text book dealing with the strength of materials.

Drawing the diagram shown in Fig. 23 and for the sake of simplicity risking the leaving out of the gasket groove, we find the following values:

$A = 15\ 1/16$ square inches,

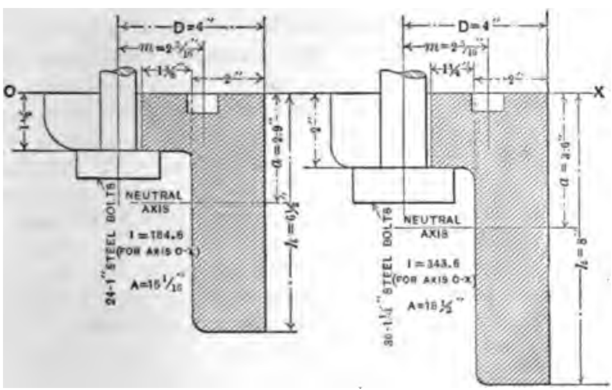
$a = 2.91$ inches,

I (about axis OX) $= 184.6$.

Substituting the known values in the given formula we have

$$F = \frac{(2\ 5/16 \times 24 \times 1200 + 126{,}000 \times 4)\ (6.5 - 2.91)}{6.2832\ (184.6 - 2.91^2 \times 15\ 1/16)} = 5{,}600 \text{ pounds.}$$

Twenty thousand pounds was taken as a safe figure for the tensile strength of cast iron. This is none too high, especially if great care



Machinery, N.Y.

Fig. 23. Data for Original Ring Fig. 24. Suggested Section

is not taken in the selection of the iron and the inspection of the casting after it is completed. With a factor of safety of 5 we have 4,000 pounds as the safe figure for a working tensile strength. The results of our calculation would thus show that the stresses in the ring are high enough to be dangerous. To give the additional strength necessary the section shown in Fig. 24 is suggested. The hub has been made 1½ inch longer, and the thickness of the flange has been increased about ½ inch. This latter change was made both to keep the parts in good proportion so far as looks are concerned, and from the fear, as well, that the ring might fail by breaking at the corner of the gasket groove. The possibility of this would be a rather difficult thing to calculate with assurance, but good judgment would seem to indicate that the casting is none too strong at this point. Repeating the same operation on this enlarged section that we went through in calculating the strength of the smaller section, also now considering 36 bolts instead of 24, as already suggested, we have

$A = 18\frac{1}{2}$ square inches,

$a = 3.6$ inches,

I (about axis OX) $= 343.6$.

Substituting the known values in the equation as before, we have

$$P = \frac{(2\,5/16 \times 26 \times 1200 + 126{,}000 \times 4)\,(8 - 3.6)}{6.2832 \times (343.6 - 3.6^2 \times 18.5)} = 4{,}070 \text{ pounds.}$$

This figure, while a little large, may be considered safe, perhaps, if a good casting from a good quality of iron is used.

The value of E used above is that recommended in the discussion from which the formula was taken, namely 1,200 pounds. This is arbitrarily selected, and although it would seem somewhat low in view of the possibilities for excessive strain afforded by the wrench and pipe combination, the boiler insurance company referred to has found that the formula, as given, is rather on the side of safety. The large bolts suggested for the improved section are favorable for reducing the excess pressure, since the workman is not liable to overstrain a large bolt in the same proportion that he would a smaller one.

It would be unwise to conclude this chapter without some reference to the testing pressure called for on the blueprint previously referred to. All the parts have thus far been figured out for a working pressure of 60 pounds. If this really is to be the *maximum* working pressure, and the parts have been proportioned with this figure in view, it is an exceedingly unwise thing to do to test the vessel at a pressure greatly in excess of this; 75 or 80 pounds at least should never be exceeded in testing the structure. Damage is often done by careless use of excessive pressures in testing, these injuries sometimes not showing at the time, but being disastrous later on. If the pressure in use will occasionally run up to a figure approaching 125 pounds per square inch, that is another matter, and the whole design should be altered to make this possible without straining the parts beyond what they are able to bear.

CHAPTER VI

KEYS AND KEYWAYS*

It is not very common in practice to determine the dimensions of keys by calculation, but rather according to the results of experience, so that great differences between the sizes used by different machine builders are not uncommon. Twenty years ago, however, a collection was made of the various key standards, and a system of average dimensions was founded on this basis. These dimensions, having stood the test of time, can be utilized as a basis for the examination of the strain to which keys are exposed. If we assume that the narrow side of the key alone has to take up the moment of rotation, then the strain of these narrow sides must be about the same as the strain of the material in the shaft itself. The narrow sides are subjected to the specific superficial pressure p, while the tension k in a shaft of the

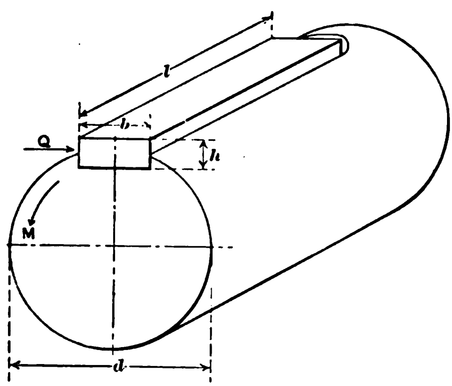


Fig. 25. Shaft with Ordinary Rectangular Key

diameter d is produced by the moment of rotation M. (See Fig. 25.) The lateral surface pressure Q on the key is therefore

$$Q = \frac{M}{\frac{d}{2}} = \frac{\pi}{8} d^2k = 0.4\, d^2k \text{ (approximately).} \qquad (18)$$

This pressure has to be taken up by half the narrow side of the key, and therefore

$$0.4d^2k = \frac{h}{2} lp \qquad (19)$$

The length l of the key is usually about 1 or $1\frac{1}{2}d$, the value $l = d$ being the average minimum. The superficial pressure p should not be allowed to exceed 17,000 pounds per square inch. The strain of rotation k should be taken at a lower value than in the case of shafts exposed to a pure twisting strain, since keyed shafts are almost invari-

* MACHINERY, March. 1907.

center affords us the means of finding the relative distances moved by the points where the force is applied and the resistance overcome.

In Fig. 30 *ad* and *cd* are the arms of the toggle-joint. What we call the instantaneous center is at *o*. It is located at the intersection of the perpendiculars to the lines along which the two ends of the arm *ad* move, this being the arm upon which forces *F* and *P* act. Thus, the end *a* moves in a horizontal line at right angles to line *ao*, and the end *d*, which is guided by the arm *cd*, and travels about the center *c*, moves for the instant at right angles to line *do*. The point of intersection *o* of lines *ao* and *do* is the instantaneous center.

The reason why this point is given the name of "instantaneous center" is because, if we consider the movements of the ends of the arm

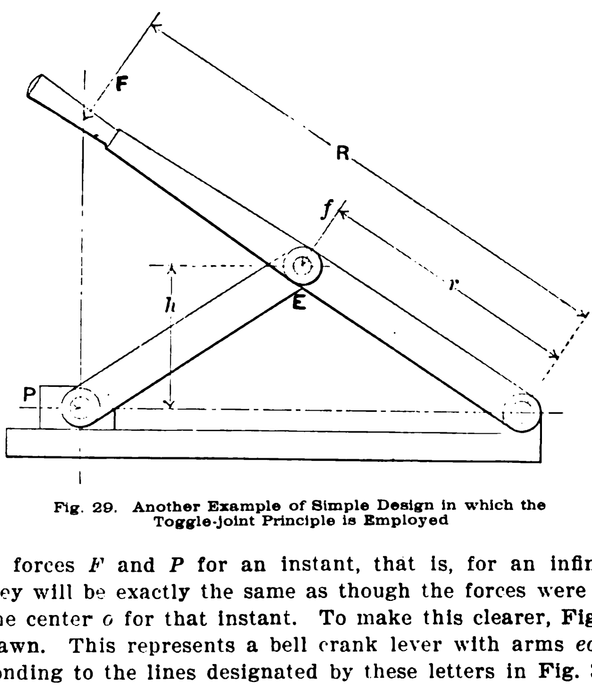
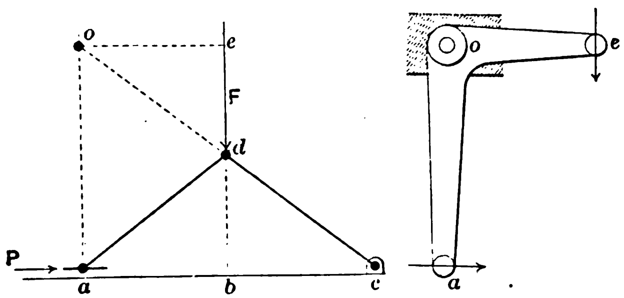


Fig. 29. Another Example of Simple Design in which the Toggle-joint Principle is Employed

and the forces *F* and *P* for an instant, that is, for an infinitesimal time, they will be exactly the same as though the forces were rotating about the center *o* for that instant. To make this clearer, Fig. 31 has been drawn. This represents a bell crank lever with arms *eo* and *ao* corresponding to the lines designated by these letters in Fig. 30. The axis *o* corresponds to the position of the instantaneous center of Fig. 30. Now it is plain, that if the lever be moved an exceedingly small distance about center *o*, the movements of points *e* and *a* will be precisely the same as the movements of forces *F* and *P* in the actual toggle-joint.

For example: Suppose it were found that for the position of the toggle-joint shown in Fig. 30, a downward push of 0.001 inch at *d* produced a movement at *a* of 0.002 inch. Also, suppose the lever in Fig. 31 to be constructed as directed, with the center-lines of its arms corresponding to *eo* and *ao* in Fig. 30. It will then be the case that a downward movement of 0.001 inch at e will move point a 0.002 inch, just as in the toggle-joint.

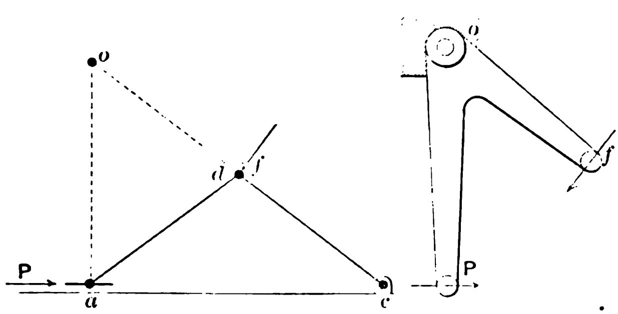
Since the movements of the extremities of the two arms of a lever are proportional to the lengths of the arms, it makes the calculation of any toggle-joint very simple to first find the instantaneous center about which an equivalent lever may be assumed to turn, and then



Figs. 30 and 31. Analysis of Principles Involved in Design Fig. 28

make the calculations as though based upon the lengths of these lever arms.

Basing our calculations, now, upon the respective lengths of the lever arms, it ought to be clear from the reasoning given above, or even without that reasoning, that if the lever in Fig. 31 is in balance, the force at e multiplied by the length of the arm eo will equal the force at a by the length of the arm ao. Returning to Fig. 30, this is



Figs. 32 and 33. Analysis of Principles Involved in Design Fig. 29

equivalent to saying that $F \times eo = P \times ao$. To locate point o conveniently, erect at point a the perpendicular to the direction of force P, and continue cd until it intersects the perpendicular at o.

Transposing our formula, we now have

$$P = \frac{F \times eo}{ao}$$

When, as is often the case, the two arms of the toggle are of equal length, then eo will be equal to one-half ac, or ac and ao will equal twice bd. Substituting h for bd in Fig. 28 and l for ab, we shall then have, for a toggle-joint with equal arms, like that in Fig. 28,

$$P = \frac{F \times l}{2h}$$

Referring to Fig. 29, this case is best solved by first neglecting the handle F, and assuming the toggle-joint to be composed of the linkage afc as in Fig. 32. Here the force f acts at right angles to the arm cd. It rotates about the center o with a radius fo, and P rotates about o with a radius ao, as indicated in Fig. 33. Therefore, $f \times do = P \times ao$, or,

$$P = \frac{F \times do}{ao}$$

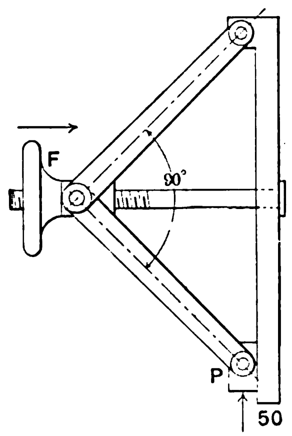


Fig. 34. Toggle-joint Design where Pressure is Exerted by Handwheel and Screw

With equal arms, $do = dc = r$ in Fig. 29 and $ao = 2 \times h$. Hence, for equal arms, as in Fig. 29,

$$P = \frac{f \times r}{2h}$$

Now, taking into account the increased leverage afforded by the handle, with the force acting at F, we have $f \times r = F \times R$. Or,

$$f = \frac{F \times R}{r}$$

Combining this with the equation above, the effect of force F upon P is found to be,

$$P = \frac{F \times R}{2h}$$

Double Toggle-joints

In most presses in which a screw and toggle-joint are used, the latter is usually made in the form of a double toggle-joint, as shown in

Figs. 35 and 36. The question is often asked whether such an arrangement is twice as powerful as a single joint, and to make this point clear let us first take up the joint and screw of Fig. 34.

Assume for illustration that the two arms are of equal length and at an angle of 90 degrees with each other, and that a force F of 100 pounds is applied by means of the handwheel. With the proportions and position assumed, it is evident that a small movement of the joint at F will produce twice as much movement at P, and consequently only half as much resistance, or 50 pounds, can be overcome at P.

In Fig. 35 the same proportions and positions of the parts are used as in Fig. 34. While the action of these different joints can easily be demonstrated, whatever the proportions, it is simpler to take the positions shown, because the relative movements of the parts can be seen at a glance. In Fig. 35 a right- and left-hand screw is used of the

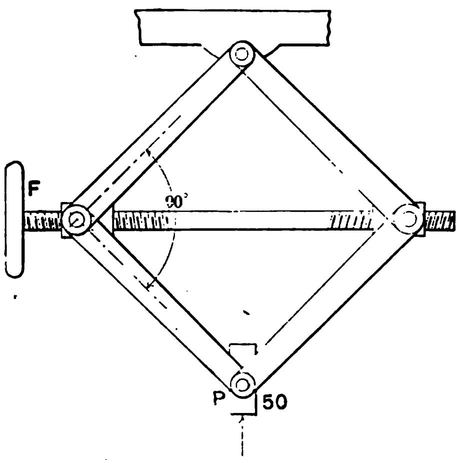


Fig. 35. Double Toggle-joint

same pitch as in Fig. 34, and one turn of the handwheel will therefore advance each one of the toggle joints and also the point P just as far as the corresponding parts were advanced in Fig. 34, and no farther. It will, therefore, take just the same pull on the handwheel to overcome 50 pounds at P as in Fig. 34, but as each joint takes half the strain, there will be only 50 pounds tension in the rod between the joints instead of 100 pounds as before.

In Fig. 36 the case is somewhat different. Here the rod is threaded at one end only, of the same pitch as before, and the handwheel screws on the threaded part, drawing the two parts of the joint together. One turn of the handwheel will advance the handwheel itself a distance, relative to the screw, equal to the pitch. Each side of the toggle-joint will be advanced a distance equal to half the pitch, and point P will be moved twice this amount, or a distance equal to the pitch, or the same distance that the handwheel moves along the screw. Hence, if the handwheel produces a force of 100 pounds, a resistance of 100 pounds can be overcome at *P*, or twice as much as in Fig. 35. The stress in the rod will, of course, be 100 pounds.

To summarize, one inch horizontal movement of the handwheel in

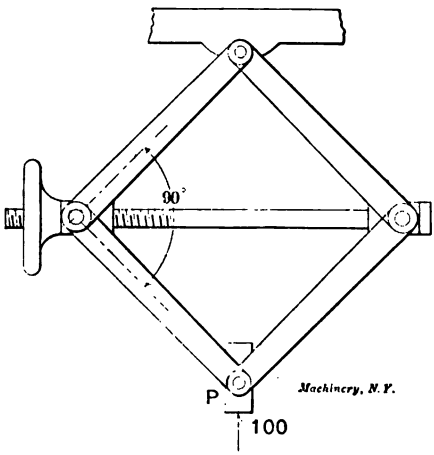


Fig. 36. Alternative Design of Double Toggle-joint

Fig. 34 will produce two inches movement at *P*; one inch movement in Fig. 35 will accomplish the same result, and hence the resistance overcome will be the same; but one inch movement of the wheel in Fig. 36 will produce the same movement, or one inch at *P*, and this form of toggle-joint has twice the power, but half the motion, of the other two.

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CHAPTER I

THE DESIGN OF JIB CRANES

Among the various types of jib cranes employed for different services in the industrial field, the simple underbraced type is most common, and has been selected for analysis in this chapter. In the investigation, the method of design, and all the possible stresses to which this type of crane may be subjected, are considered. The treatise may appear somewhat lengthy for such a simple machine, and although some of the stresses discussed are frequently disregarded in actual practice because of the employment of large factors of safety, yet all stresses should be investigated and provision made for them, especially in cranes of abnormal capacities or proportions, or both, which are frequently met with in practice.

As has often been said, sound judgment is a requisite of a successful designer. No precise rules can ever be formulated to cover all cases as they arise in practice, and the judgment of the designer is called upon repeatedly to decide the correct proceeding where there is no precedent.

The following discussion is of a typical crane, and is treated from a theoretical as well as a commercial standpoint, such as would be followed in the engineering office of a manufacturing company.

The type considered consists essentially of a structure in which *GF*, a mast, rests on a foundation (see Fig. 1), and is supported at the top by a suitable connection. *AE* is a member secured to the mast, and supported at *D* by a strut *DC*, which is bolted or riveted to a gusset plate on the member and mast, or connected to these members either with angles or castings as in Fig. 4. Let us first investigate the stresses produced in these members composing the frame by the external forces acting on the crane. The member *AE*, commonly called the jib, is subjected to stresses produced by the loads concentrated at the wheels of the trolley, and the weight of the members themselves, which stresses we will proceed to find. The trolley carrying the load is supported by four wheels traveling the length of the jib and producing the loads p, p, placed at a distance d from each other. The constant distance d is known as the wheel base. These wheel loads p, p are equal to the sum of the net load to be lifted, P, plus the weight of the trolley, ropes and bottom block, divided by the number of wheels supporting trolley, usually four.

The jib is considered as a beam supported at the joints *A* and *D*, having a cantilever end *DE*, and subjected to axial tensile, eccentric tensile, eccentric compressive, and flexural stresses. The length of the cantilever end from *D* to center line of load, when the load is at extreme outer end of the jib, is frequently made about one-fourth the distance between supports A and D, since, in general, the maximum bending moment produced by loads p, p when at the end of the cantilever, and that produced when the load is midway between A and D are about equal. But, more accurately, this ratio should be proportioned so as to obtain equal maximum fiber stresses in both cantilever and span, and thus a jib having a constant cross-section, such as a rolled beam or channel, can be economically employed. When loads p, p are acting between D and E, the maximum reaction R at D, when the trolley is at the extreme end of the cantilever, is the sum of the products of each of the wheel loads multiplied by the ratio of the long levers AE and AE_1 to the short lever AD. Expressing AE, AE_1, and

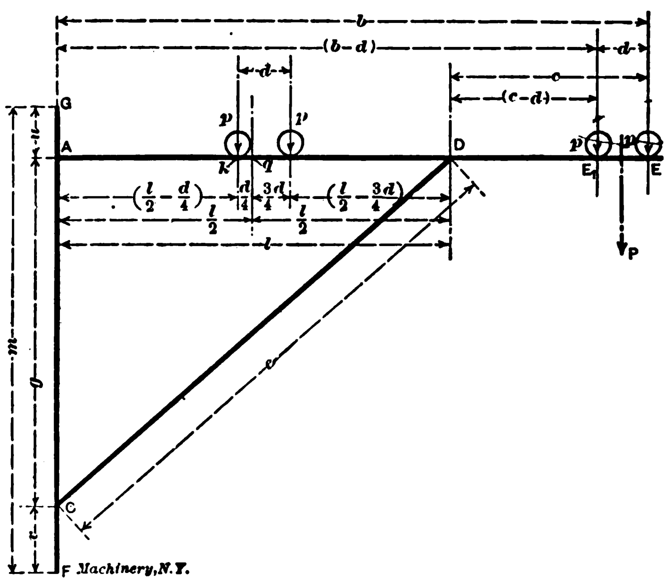

Fig. 1. Diagram of Type of Jib Crane selected for Analysis of Stresses

AD in terms of the dimension letters, we have (see Fig. 1) b, $(b-d)$ and l, respectively. Then taking moments about A, the fulcrum of the lever, we have

$$R = p \times \frac{b}{l} + p \times \frac{(b-d)}{l} \qquad (1)$$

This reaction R produces a direct tensile stress between the points A and D of the jib, and a compressive stress in strut CD.

Let side AC of the triangle ADC in Fig. 1 represent the magnitude of this reaction R; then side AD represents the value of the tensile stress, or

$$\text{Stress in } AD = \frac{\text{side } AD}{\text{side } AC} \times R,$$

and, employing dimension letters l and g, we obtain

$$\text{Stress in } AD = \frac{l}{g} \times R.$$

Substituting the value of R of formula (1) for R, we have

$$\text{Stress in } AD = \frac{pb + p(b - d)}{l} \times \frac{l}{g} = \frac{pb + p(b - d)}{g} \qquad (2)$$

Before the section of the jib can be determined, it is required to find the maximum flexural stresses due to the live and dead load bending moments, and combine them with the axial or direct tensile stresses acting on span AD, when the absolute maximum bending moment occurs, that is when the wheel loads p, p are so placed that the center of the span is midway between the center of gravity of these loads and one of the trolley wheels. They must also be combined with the stresses produced by the eccentric pull of the ropes holding the load.

The direct tensile stress in the jib to be so combined is then not the maximum one just found by formula (2), but that due to the reaction R_1 when the trolley is at the position in the span producing the greatest bending moment, and the value of that reaction R_1 at D is found by taking the moments about support A, or,

$$R_1 = \frac{p\left(l + \frac{d}{2}\right)}{l}$$

Value of R_2 at A is found by taking moments about support D,

$$R_2 = \frac{p\left(l - \frac{d}{2}\right)}{l}$$

To obtain the maximum live load bending moment we take moments about point k under one of the wheels (as shown in Fig. 1); then we have

$$\text{Maximum bending moment} = R_2 \times \left(\frac{l}{2} - \frac{d}{4}\right)$$

But as $R_2 = \dfrac{p\left(l - \frac{d}{2}\right)}{l}$, if we substitute this value of R_2 in the last equation, we find the greatest live load bending moment from

$$\text{Live load bending moment} = \frac{p}{2l}\left(l - \frac{d}{2}\right)^2 \qquad (3)$$

$$\text{Dead load bending moment} = \frac{wl}{8} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad (4)$$

$$\text{Approximate total bending moment} = \frac{p}{2l}\left(l - \frac{d}{2}\right)^2 + \frac{wl}{8} \qquad (5)$$

where d = wheel base,

w = weight of jib between supports A and D, which weight must be assumed,

$l = AD$, or span.

In regard to formula (5) it may be said that the customary approximate method of adding the maximum live load bending moment to the maximum dead load bending moment is incorrect, except in cases where the maximum live load bending moment occurs at the center of the span. The correct method for this case is to add to the maximum live load bending moment its increment of the dead load moment at that point, and not the maximum value which takes place at the center of the span. The usual method is sufficiently correct for practical purposes, however, as it is on the safe side.

The unit-stress f, due to bending, in pounds per square inch is found from

$$f = \frac{\dfrac{p}{2l}\left(l - \dfrac{d}{2}\right)^2 + \dfrac{wl}{8}}{Z} \qquad (6)$$

The unit-stress due to jib reaction R_1 is found from

$$f_1 = \frac{R_1 \times \dfrac{l}{g}}{a} = \frac{p\left(l + \dfrac{d}{2}\right)}{ag} \qquad (7)$$

Unit-stress f_2, due to tension of rope, is found from

$$f_2 = \frac{T}{a} + \frac{Tz}{Z} \qquad (8)$$

where T = tension in rope in pounds,

R_1 = value of reaction at D when greatest live load bending moment occurs,

z = eccentricity or distance between center line of rope and center line of member, in inches,

Z = section modulus of section,

a = area of section of member in square inches.

The maximum compressive stress in top flange of jib section = $f - f_1 + f_2$, or

$$\frac{\dfrac{p}{2l}\left(l - \dfrac{d}{2}\right)^2 + \dfrac{wl}{8}}{Z} - \frac{p\left(l + \dfrac{d}{2}\right)}{ag} + \frac{T}{a} + \frac{Tz}{Z}$$

or combining

$$f - f_1 + f_2 = \frac{\left[\frac{p}{2l}\left(l - \frac{d}{2}\right)^2 + \frac{wl}{8}\right] + Tz}{Z} + \frac{T - p\dfrac{\left(l + \frac{d}{2}\right)}{g}}{a} \qquad (9)$$

The maximum tensile stress in the bottom flange of jib when such flange is opposite to the line of action of the rope (see Fig. 3) $= f + f_1 - f_2$ (f_2 in this case being modified to give tensile stress in bottom flange, due to eccentricity of rope loading), or

$$\frac{\frac{p}{2l}\left(l - \frac{d}{2}\right)^2 + \frac{wl}{8}}{Z} + \frac{p\left(l + \frac{d}{2}\right)}{ag} - \frac{T}{a} + \frac{Tz}{Z}$$

or combining,

$$f + f_1 - f_2 = \frac{\left[\frac{p}{2l}\left(l - \frac{d}{2}\right)^2 + \frac{wl}{8}\right] + Tz}{Z} + \frac{\dfrac{p\left(l + \frac{d}{2}\right)}{g} - T}{a} \qquad (10)$$

These results should not exceed the specified fiber stress for the structure. Before selecting a structural shape to resist these maximum stresses just found, the stresses on the cantilever end should be considered as follows:

f = flexural stress due to bending,
f_1 = tensile stress due to jib reaction,
f_2 = compressive stress due to tension or pull of ropes.

Live and dead load maximum bending moment

$$= (p \times c) + [p \times (c - d)] + \left(w \times \frac{c}{2}\right) \qquad (11)$$

and f, or stress due to bending on cantilever

$$\text{Unit stress } f = \frac{pc + p(c - d) + \frac{wc}{2}}{Z} \qquad (12)$$

This maximum flexural stress takes place at D, and immediately to the left of D, there exists at the same time the direct tensile stress due to the maximum reaction R, when the trolley is at the extreme end of the cantilever producing this bending stress, found in formula (1), which also must be combined with the stress due to the pull of the rope. Therefore the unit-stress at point $D = f + f_1 - f_2$

$$\text{Unit-stress } f_1 = \frac{Rl}{ag} = \frac{pb + p(b - d)}{ag} \qquad (13)$$

Stress due to pull of ropes $= f_2$

$$\text{Unit-stress } f_2 = \frac{T}{a} + \frac{Tz}{Z} \qquad (8)$$

Therefore the maximum fiber stress =

$$f + f_1 - f_2 = \frac{pc + p\,(c - d) + \dfrac{wc}{2}}{Z} + \frac{pb + p\,(b - d)}{ag} - \left(\frac{T}{a} + \frac{Tz}{Z}\right)$$

or combining,

$$f + f_1 - f_2 = \frac{pc + p\,(c - d) + \dfrac{wc}{2} - Tz}{Z} + \frac{\dfrac{pb + p\,(b - d)}{g} - T}{a} \qquad (14)$$

where p = wheel load as before,

w = weight of section of jib from D to its extremity (see Fig. 1).

The compressive stress in strut CD is $R \times \dfrac{\text{side } CD}{\text{side } AC}$, of the triangle ADC, or $R \times \dfrac{e}{g}$

And since R is maximum when the trolley is at the extreme end of the cantilever, or

$$R = \frac{pb + p(b - d)}{l} \qquad (1)$$

then the maximum compressive stress in strut =

$$\frac{pb + p(b - d)}{l} \times \frac{e}{g} \qquad (15)$$

$$\text{Unit-stress in strut} = \frac{[pb + p(b - d)]e}{agl}, \text{ (see Fig. 1).} \qquad (16)$$

where a equals area of cross-section of strut.

The allowable unit-stress per square inch of section of this member is found by the usual Gordon formulas:

$$\text{for structural steel, } f = 17{,}100 - 57\,\frac{l}{r} \qquad (17)$$

$$\text{for yellow pine, } f = 1{,}200 - 18\,\frac{l}{t} \qquad (18)$$

However, a satisfactory reducing formula of the Rankine type, extensively used by bridge companies, and specified by some railroad companies, is recommended. It is as follows:

$$\text{for structural steel, } f = \frac{15{,}000}{1 + \dfrac{l^2}{13{,}500\, r^2}} \qquad (19)$$

$$\text{for yellow pine, } f = \frac{1{,}200}{1 + \dfrac{l^2}{250\, t^2}} \qquad (20)$$

where l = length of strut in inches,
t = thickness of timber in inches,
r = least radius of gyration.

The stress in the strut due to its own weight is neglected as being very small in most practical cases.

Ordinarily the ratio $\frac{l}{r}$ should not exceed 130; however, this ratio is frequently increased if the fiber-stress is well under the one specified, and as long as its departure from straightness will not subject the strut to an appreciable bending moment.

The stresses that may exist in the mast are as follows: (See Fig. 1.)

(1) Axial compression due to reaction R_2 and weight of structure.

(2) Eccentric stress due to R_2 when trolley is at extreme position on jib next to mast for cranes where jib connects to the face of the mast, and not at the center line of gravity of its section.

(3) Eccentric flexural stress due to tension in ropes.

(4) Flexural stress due to direct tension in jib AE, and to the horizontal component of direct compression in the strut DC.

(5) Eccentric flexural stress due to weight of drum and other hoisting machinery. This last stress is usually disregarded, however, except where the jib and hoisting machinery are of abnormally large proportions.

$$\text{Unit-stress } f'_1 = \frac{R_2}{a},$$

but
$$R_2 = \frac{pl + p(l-d)}{l}$$

therefore
$$f'_1 = \frac{\frac{pl + p(l-d)}{l}}{a} = \frac{pl + p(l-d)}{al} \qquad (21)$$

$$\text{Unit-stress } f'_2 = \frac{R_2}{a} + \frac{R_2 z_1}{Z} \text{ (see Fig. 8)},$$

but
$$R_2 = \frac{pl + p(l-d)}{l}$$

therefore
$$f'_2 = \frac{pl + p(l-d)}{l}\left(\frac{1}{a} + \frac{z_1}{Z}\right) \qquad (22)$$

$$\text{Unit-stress } f'_3 = \frac{T}{a} + \frac{Tz_2}{Z}\cos\theta \qquad (23)$$

$$\text{Tension in jib} = H = \frac{pb + p(b-d)}{g} \text{ (see Fig. 2)} \qquad (2)$$

The horizontal component of stress in strut is equal to the tension

H. The mast is then considered as a beam supported by reactions *H* and *r*. (See Fig. 2.)

$$r = \frac{p \times b + p(b - d) + w_1 \times j}{m} \qquad (24)$$

where w_1 = weight of structural frame,

j = distance from center of mast to center of gravity of frame,

m = distance between centers of bearings.

The quantity $w_1 \times j$ may be omitted when the frame is not very large. The maximum bending moment in the mast is then $r \times u$ or

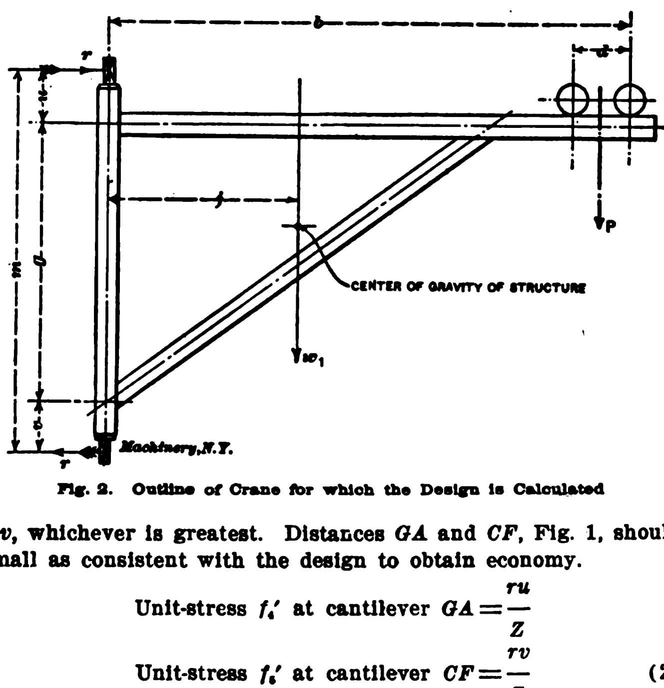

Fig. 2. Outline of Crane for which the Design is Calculated

$r \times v$, whichever is greatest. Distances *GA* and *CF*, Fig. 1, should be as small as consistent with the design to obtain economy.

$$\text{Unit-stress } f_4' \text{ at cantilever } GA = \frac{ru}{Z}$$

$$\text{Unit-stress } f_4' \text{ at cantilever } CF = \frac{rv}{Z} \qquad (25)$$

The axial compressive stress in the mast due to the whole weight of the structure, should be added to the flexural compressive stress f_4' due to bending when the trolley is at the extreme end of the jib, since that part of the mast immediately beneath *C* is subjected to both t the same time under these conditions.

The stress f_4' is not added to the stress f_3' as found by formula (22), because they do not take place at the same time, the maximum bending taking place when the trolley is at the end of the jib, and the maximum eccentric compressive stress when the trolley is close to the mast.

It is sometimes required, when long jib members are necessary, to brace the two shapes composing the jib at some intermediate point in order to reduce the ratio $\frac{l}{r}$, and, at the same time, lessen the tendency of the jib members to spread. This is done by securing structural shapes bent clear over the jib trolley. (See Fig. 4.) The ratio $\frac{l}{r}$ should not exceed that above specified.

The pintles at G and F should be made large enough to resist the bending moment on them, and also designed for a safe bearing pressure per square inch of their projected area. This pressure is the quantity r in formula (24).

The jib end connection is subjected to flexural stresses due to the tension of the rope or ropes, which should be taken into consideration. The connection is treated as a beam, and the pull of the rope or ropes

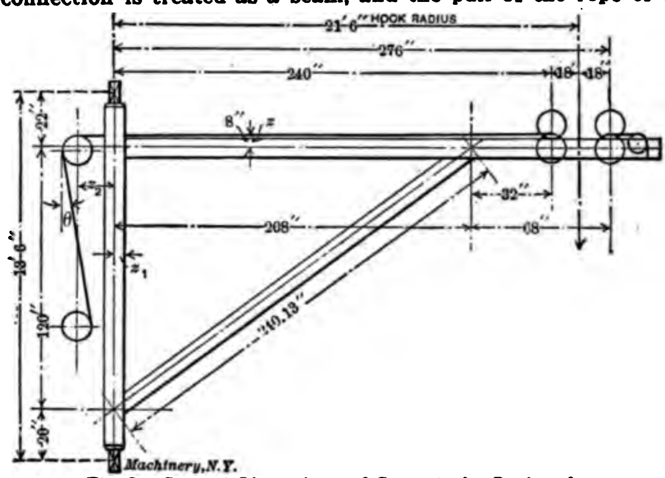

Fig. 3. General Dimensions of Crane to be Designed

as concentrated loads in the middle or at equal distances from the middle, according to the kind of connection employed, the beam in question being supported at both ends.

Example

Required to design a jib crane of the underbraced type to lift a load of 10,000 pounds at a radius of 21 feet 6 inches; distance between underside of roof truss or top support and floor 13 feet 6 inches; jib to be constructed of two structural steel frames composed of standard size channels and connected together (see Fig. 4); trolley mounted on four wheels running on top flanges of jib member. Maximum fibre-stress 13,000 pounds per square inch, which is allowable for hand-power machines. For a load of 10,000 pounds we will use four parts of 7/16-inch—6 strands of 19 wires—plow steel hoisting rope, having a breaking strength of 17,700 pounds, and will give a factor of safety of $\frac{4 \times 17,700}{10,000} = 7.08$, which must also take care of the bending stresses

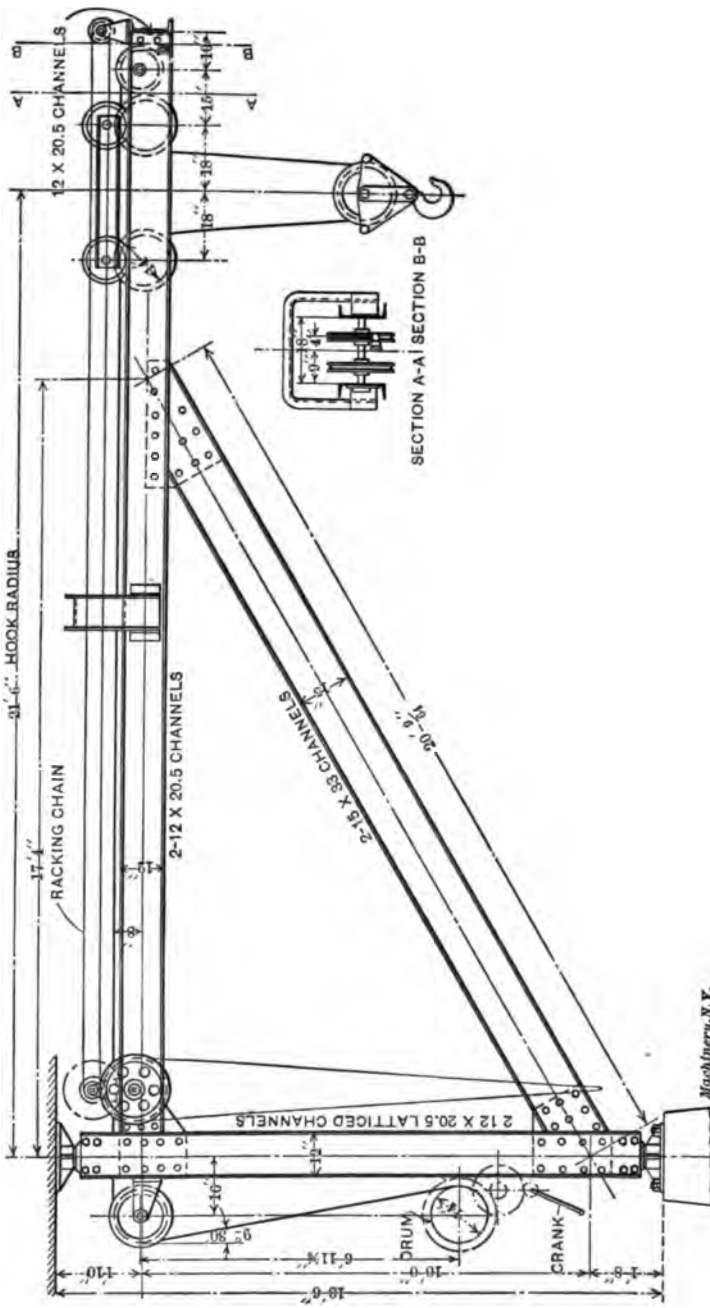

Fig. 4. Crane Calculated to Lift a Load of 10,000 Pounds at a Radius of 21 Feet 6 Inches

in the ropes. This size of rope will require sheaves of 14 inches in diameter, and will allow a wheel base of 36 inches. Two ends of these two lengths of rope will wind on the drum, and the other two ends will be supported at the outer end of the jib by an equalizing beam.

Load to be lifted	10,000 pounds
Approximate weight of trolley, ropes and block	500 pounds
Total	10,500 pounds

which will make the wheel loads $\frac{10,500}{4} = 2,625$ pounds each.

Distance between mast and joint D, Fig. 3, $= 208$ inches.

Distance between jib and joint $C = 120$ inches.

Distance between mast and extreme position of outermost wheels of trolley = effective radius + half the wheel base = 21 feet 6 inches + 1 foot 6 inches = 23 feet = 276 inches.

Let us first assume the trolley at that position in the span AD producing the greatest bending moment (see Fig. 1):

Maximum live load bending moment

$$= \frac{2625}{2 \times 208}\left(208 - \frac{36}{2}\right)^2 = 227,798 \text{ inch-pounds.} \qquad (3)$$

By looking at the table of properties of steel channels in any steel company's handbook, we find that a 12-inch channel weighing 20.5 pounds per foot, with an area of 6.03 square inches, has a section modulus about the axis perpendicular to the web of 21.4, and this value divided into the live load bending moment will give a stress of 10,644 pounds per square inch, which leaves us a margin for the other stresses to be yet considered. Therefore, we will temporarily select the above shape for the purpose of finding the bending-moment due to the uniform weight of the member itself.

$$\text{Weight of channel between } A \text{ and } D = \frac{208}{12} \times 20.5 = 355 \text{ pounds.}$$

$$\text{Dead load bending moment} = \frac{355 \times 208}{8} = 9230 \text{ inch-pounds.} \qquad (4)$$

Approximate total bending moment

$$= 227,798 + 9,230 = 237,028 \text{ inch-pounds.} \qquad (5)$$

Unit-stress due to bending

$$= \frac{\frac{p}{2l}\left(l - \frac{d}{2}\right)^2 + \frac{wl}{8}}{Z} = \frac{237,028}{21.4} = 11,076 \text{ pounds per sq inch.} \qquad (6)$$

Unit-stress due to reaction R_1

$$= \frac{2625 \times \left(208 + \frac{36}{2}\right)}{120 \times 6.03} = 817 \text{ pounds per square inch.} \qquad (7)$$

Tension in ropes $\dfrac{10000}{4} = 2500$ pounds.

Unit-stress due to tension in rope

$$= \frac{2500}{6.08} + \frac{2500 \times 8}{21.4} = 1848 \text{ pounds per square inch.} \qquad (8)$$

Total stress on top flange $= f - f_1 + f_2 = 11{,}076 - 817 + 1{,}348 =$ 11,607 pounds per square inch (9), which stress is under the one specified; the shape tentatively selected may therefore be used for this member of the crane.

Weight of cantilever end of jib $= \dfrac{93}{12} \times 20.5 = 159$ pounds.

Unit-stress due to bending

$$= \frac{2625 \times 68 + 2625 \times 82 + 159 \times \dfrac{93}{2}}{21.4} = 12613 \text{ pounds per sq. inch.} \qquad (12)$$

Unit-stress due to reaction R

$$= \frac{2625 \times 276 + 2625 \times (276 - 86)}{120 \times 6.08} = 1872 \text{ pounds per sq. inch.} \qquad (13)$$

Unit-stress due to pull in rope

$$= \frac{2500}{6.08} + \frac{2500 \times 8}{21.4} = 1848 \text{ pounds per square inch.} \qquad (8)$$

Total unit-stress on top flange of cantilever =

$12{,}613 + 1{,}872 - 1{,}348 = 13{,}137$ pounds per square inch, (14)

which is 137 pounds per square inch more than the specified stress. In practice, this will not be considered of sufficient importance to change the design.

Total length of jib member = 25 feet 1 inch, or 301 inches.

Least radius of gyration of 12×20.5 pounds channel $= 0.81$.

The ratio $\dfrac{\text{length}}{\text{least radius or gyration}} = \dfrac{301}{0.81} = 371$, consequently the channels of the two frames should be braced at least at a point midway between the end connection and the mast. (See Fig. 4.)

Length of strut $DC = \sqrt{120^2 + 208^2} = 240.18$ inches. Selecting a 15×33-pound channel having a cross-sectional area of 9.9 square inches, and least radius of gyration of 0.91, for strut, we have the compressive unit-stress

$$= \frac{[2625 \times 276 + 2625\,(276 - 86)] \times 240.18}{208 \times 120 \times 9.9} = 1816 \text{ pounds per sq. inch.} \qquad (16)$$

$$\text{Allowable stress} = \frac{15000}{1 + \dfrac{240.18^2}{18500 \times 0.91^2}} = 2440 \text{ pounds per sq. inch.} \qquad (19)$$

The ratio of the length of the strut to its least radius of gyration is $\dfrac{240.13}{0.91} = 264$, which is excessive; the maximum unit-stress, however, is very low, only 1,316 pounds per square inch, or hardly more than half of that allowed by the formula (19). As there is not a channel rolled by any mill with a greater "least radius of gyration" than the one we have employed, we may stiffen the strut laterally by riveting an angle to its web in the inside or back of channel. Unless the ratio 130 must be adhered to, the channel should be left as it is as long as the member shows no great deflection under load.

Let us now investigate the stresses existing in the mast, which we assume is composed of two 12 x 20.5-pound channels. The distance from center of mast to nearest wheel when the trolley is at the extreme position next to mast = 11 inches.

$$\text{Then } R_2 = \frac{2{,}625\ (208 - 11) + 2{,}625\ (208 - 11 - 36)}{208} = 4{,}518 \text{ pounds.}$$

As the two vertical shapes composing the mast are latticed together, we will take the two equal reactions R_2 (one which acts on one channel and the other on the opposite one) to be resisted by the two shapes combined, therefore the least radius of gyration of the mast as built is then that perpendicular to the web of the channels, whose value is 4.61.

Then the allowable compressive stress

$$= \frac{15000}{1 + \dfrac{162^2}{13500 \times 4.61^2}} = 13{,}761 \text{ pounds per square inch.} \qquad (19)$$

Unit stress f'_1

$$= \frac{2625\ (208 - 11) + 2625\ (208 - 11 - 36)}{6.03 \times 208} = 749 \text{ pounds per sq. inch.} \qquad (21)$$

Stresses f'_2 do not take place in this gusset-connected frame.

Unit-stress f'_3 due to tension of rope

$$= \frac{2500}{6.03} + \frac{2500 \times 16}{21.4} \times \cos 9 \text{ deg. } 30 \text{ min.} \qquad (23)$$

$$= 2257 \text{ pounds per square inch. (See Fig. 4.)}$$

Horizontal reaction at top and bottom of mast when load is at extreme outside end of jib =

$$r = \frac{2{,}625 \times 276 + 2{,}625\ (276 - 36)}{162} = 8{,}361 \text{ pounds.} \qquad (24)$$

Unit-stress f'_4 due to bending moment at top of mast =

$$\frac{8{,}361 \times 22}{21.4} = 8{,}548 \text{ pounds per square inch.} \qquad (25)$$

Maximum unit-stress immediately beneath point A of mast $= f'_4 + f'_3 = 8{,}548 + 2{,}257 = 10{,}805$ pounds per square inch.

For the end connection of the jib at E we select a 12-inch × 20.5 pound channel for the sake of symmetry, and proceed to investigate the bending stress to which it is subjected, due to the pull of the ropes. The distance between the jib members is 18 inches; the pull on the ropes 2,500 pounds. The section modulus of the channel in consideration about an axis parallel to the web is 1.75. Two ropes, both four inches from the center of connecting channel are used (see Fig. 4).

Maximum bending moment on channel $= 2{,}500 \times (9 - 4) = 12{,}500$ inch-pounds.

$$\text{Unit-stress} = \frac{12{,}500}{1.75} = 7{,}143 \text{ pounds per square inch.}$$

Horizontal reaction on pintles

$$r = 2 \times \frac{2{,}625 \times 276 + 2{,}625\,(276 - 36)}{162} = 16{,}722 \text{ pounds.} \qquad (24)$$

Assuming the pintles to be 6 inches long, and taking moments about a lever arm from the center of the bearing to the support ($= 3$ inches), we have, bending moment $= 16{,}722 \times 3 = 50{,}166$ inch-pounds. Unit-stress on pintle should not exceed 9,000 pounds per square inch for machine steel.

Section modulus of a circular section $= \dfrac{\pi d^3}{32} = 0.098d^3$, where $d =$ diameter of section.

$$\text{Diameter of pintle} = d = \sqrt[3]{\frac{50{,}166}{0.098 \times 9{,}000}} = 3.84 \text{ inches.}$$

The bearing pressure on pintles should not exceed 1,000 pounds per square inch of projected area. Therefore $\dfrac{16{,}722}{1{,}000} = 16.72$ square inches are required. We will make the pintles $3\frac{7}{8}$ inches in diameter × 6 inches in length, which will give a bearing pressure of $\dfrac{16{,}722}{3.875 \times 6} = 719$ pounds per square inch.

CHAPTER II

EXAMPLES OF JIB CRANE CALCULATIONS

The following examples will prove helpful as suggestive of the ordinary procedure in jib crane calculations. Two problems are presented for solution, the first of which may be stated as follows.

Problem 1

The column of the crane, designed as shown in Fig. 5, is of cast iron, has all the appearance of being sound, and is supposed to have ¾ inch thickness of metal. The dimensions are as per sketch. The compression member consists of two 7-inch channels, weighing 11¼ pounds per foot, arranged back to back with a 3-inch space between

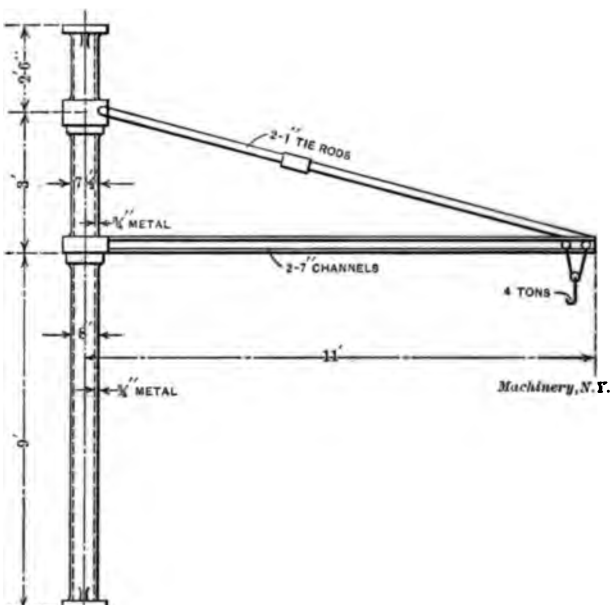

Fig. 5. General Construction of Jib Crane in Problem 1

them for the trolley to operate in. These are fastened together at each end, and the outer ends are supported by two 1-inch rods. The question raised is whether it will be safe to suspend 4 tons from the end of the 11-foot jib.

Calling a ton 2,000 pounds, the force conditions, reduced to simplest terms, will be as shown in Fig. 6. A compound beam with compressive stress, as indicated in the lower part of Fig. 6, would evidently be an equivalent case. Considering the column as a compound beam, the moment diagram will be as shown in Fig. 7. From this it will be seen that the maximum bending moment on the column equals 54,630 foot-pounds, exerted in the axis of the jib, or 5½ feet from the upper end of the column.

To find the maximum fiber unit stress for the case of a beam subject to flexure by transverse loads and also to compression in the

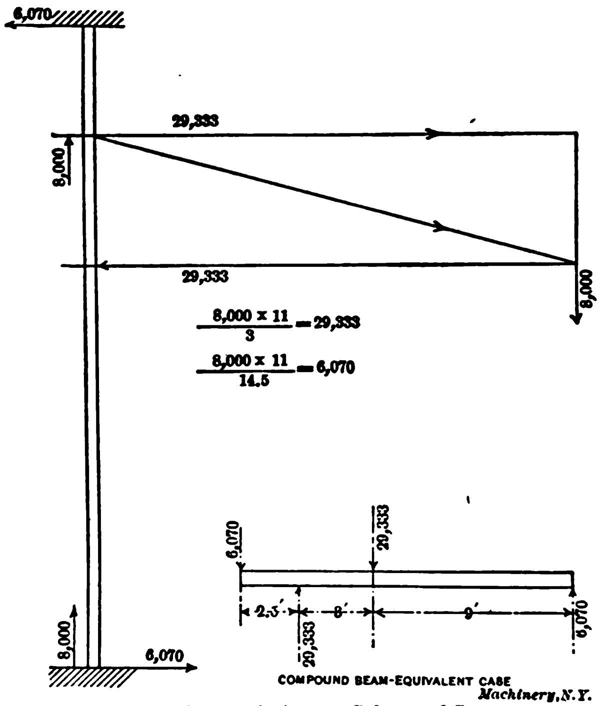

Fig. 6. Forces Acting on Column of Crane

direction of its length, we find in Merriman's "Mechanics of Materials," 10th edition, page 256, a formula which can be reduced to the form

$$S_1 = \frac{M c}{I - \dfrac{n P l^2}{m E}}$$

where S_1 = maximum fiber unit stress,

M = maximum bending moment in inch-pounds,

c = distance from the neutral axis to the remotest fiber,

I = moment of inertia of the cross section,

P = longitudinal compressive force = 8,000 pounds,

E = coefficient of elasticity = 15,000,000 for cast iron,

n and m = numbers depending on design and kind of loading,

l = length of span of the beam, in inches.

In the above, M, the maximum bending moment, $= 54{,}630$ foot-pounds $= 54{,}630 \times 12$ inch-pounds, and $c = 3\frac{3}{4}$ inches.

I, for hollow column, $= 0.0491\ (d^4 - d_1^4)$, where d and d_1 are the external and internal diameters, $7\frac{1}{2}$ and 6 inches, and hence $I = 91.5$. The approximate value of $\dfrac{n}{m}$ is 1-12. The span l, in the equivalent case of the compound beam, is the distance between supports, or in our case, 144 inches. Hence we have

$$S_1 = \frac{54{,}630 \times 12 \times 3\frac{3}{4}}{91.5 - \dfrac{1}{12} \times \dfrac{8{,}000 \times 144^2}{15{,}000{,}000}}$$

$= 27{,}150$ pounds per square inch.

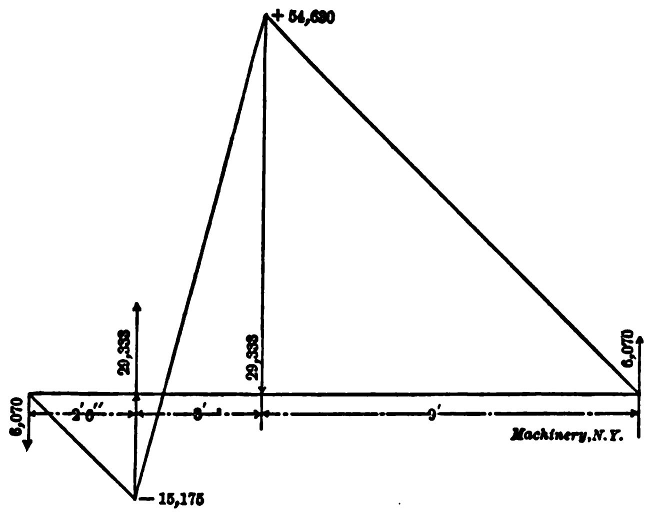

Fig. 7. Moment Diagram for Crane Column

To this should be added $\dfrac{P}{A}$, where A is the cross sectional area of the column, for the maximum compressive unit stress. $A = 15.9$ inches, making $\dfrac{P}{A} = \dfrac{8{,}000}{15.9} = 503$, and hence $S = S_1 + \dfrac{P}{A} = 27{,}650$ pounds per square inch, for compression. For tension, $S = S_1 - \dfrac{P}{A} = 26{,}650$ pounds per square inch.

Since the average ultimate strength of cast iron in tension is 20,000 pounds per square inch, it follows that the column will probably fail when a load of 8,000 pounds is lifted at the end of the jib.

The above method applies also to the discussion of the channels

[illegible]

[illegible]

[illegible]

The following table gives the properties of the [illegible]:

Depth of [illegible] in inches	[illegible]	[illegible]	[illegible]
Weight per foot in pounds	[illegible]	[illegible]	[illegible]
A = area of section in square inches	[illegible]	[illegible]	[illegible]
r = least radius of gyration	.212	[illegible]	[illegible]
I = section modulus, axis perpendicular to web	[illegible]	[illegible]	[illegible]

In addition to [illegible] the following will be used:

M = bending moment.

[illegible] = maximum fiber stress due to bending.

[illegible] = maximum fiber stress due to tension.

[illegible] = maximum fiber stress.

[illegible]

$$M = [illegible] = [illegible]00,000 \text{ inch-pounds.}$$

$$[illegible] = \frac{M}{I} = \frac{[illegible]00,000}{[illegible]} = [illegible] \text{ pounds per square inch.}$$

[illegible]

$$M = \frac{W l}{4} = \frac{[illegible]}{4} = [illegible]0,000 \text{ inch-pounds.}$$

[illegible]

design of framework adopted, it would be well to make *ED* about one-fourth of the length of *DH*. If this is done, the bending moment will be the same whether the load is at *E* or *B*, and, will in either case, be less than the maximum moment we have just found, so that a smaller section could be used.

The vertical reaction at *D* is found thus:

$$R_1 = W \times \frac{a}{l} = 5{,}000 \times \frac{13}{8} = 8{,}125 \text{ pounds.} \qquad (28)$$

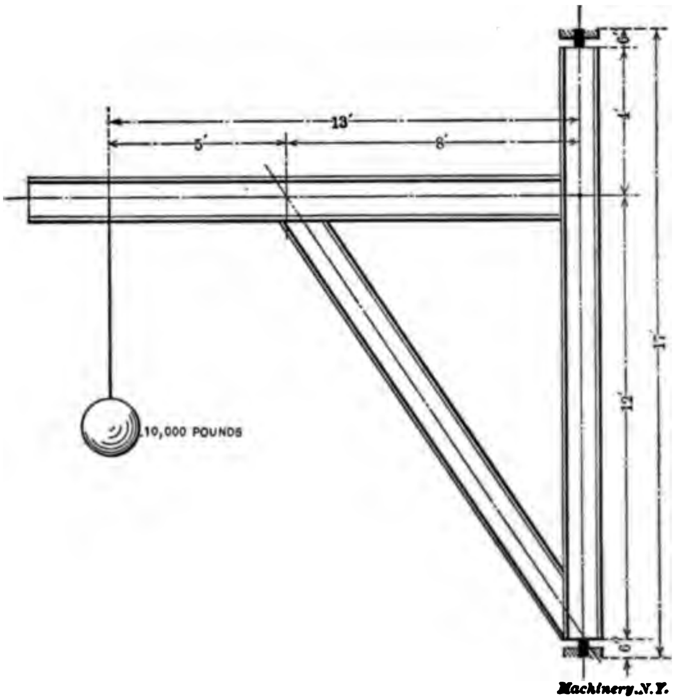

Fig. 8. General Design of Crane in Problem 2

This produces a tensile stress in *DH* which may be found by the parallelogram of forces shown in Fig. 9, or by the following calculation:

$$R_3 = R_1 \times \frac{l}{g} = 8{,}125 \times \frac{8}{12} = 5{,}420 \text{ pounds.} \qquad (29)$$

The stress per square inch in *DH* due to this force is:

$$S_t = \frac{R_3}{A} = \frac{5{,}420}{9.9} = 550 \text{ pounds per square inch} \qquad (30)$$

Adding this to the stress found in (26), we have the total stress in *DH*:

$$S = S_t + S_b = 550 + 7{,}200 = 7{,}750 \text{ pounds per sq. in.} \qquad (31)$$

which is the maximum fiber stress in the yard-arm, occurring just to the right of point D. This is well within the limit of safety, which may be taken as about 13,000 pounds per square inch.

The allowable fiber stress in the brace may be calculated from the following formula based on Rankine's formula for columns:

$$S = \frac{15{,}000}{1 + \dfrac{e^2}{13{,}500 \times r^2}} = \frac{15{,}000}{1 + \dfrac{178^2}{13{,}500 \times 0.805^2}} = 3395 \text{ pounds per sq. inch.} \qquad (32)$$

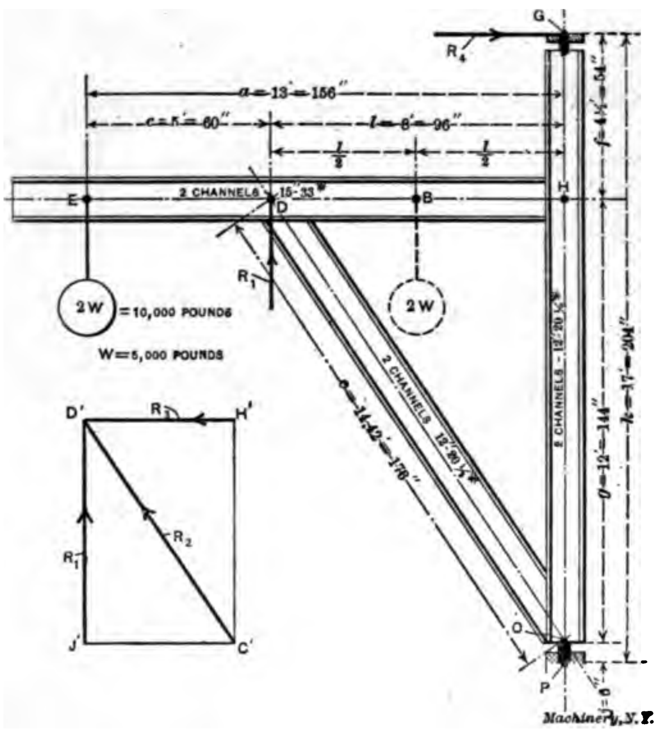

Fig. 9. Calculating the Stresses in the Channels in Crane Fig. 8

The reaction producing compression in CD is found by the force diagram in Fig. 9, or by the following calculation:

$$R_2 = R_1 \times \frac{e}{g} = 8{,}125 \times \frac{173}{144} = 9{,}760 \text{ pounds} \qquad (33)$$

The compressive stress per square inch in the brace is, then,

$$S = \frac{R_2}{A} = \frac{9{,}760}{6.03} = 1{,}620 \text{ pounds per square inch} \qquad (34)$$

ch is, as may be seen, not quite one-half the allowable amount. The f the length to the radius of gyration ($e \div r$) in this strut is so great, being about 215, that it is wise to keep the unit compressive stress down to a very low point.

The mast is most liable to fail by bending at H when the load is at E. To find the bending moment at H, we must first find the horizontal reaction at G:

$$R_4 = W \times \frac{a}{k} = 5{,}000 \times \frac{13}{17} = 3{,}825 \text{ pounds} \qquad (35)$$

The bending moment at H is then:

$$M = R_4 \times f = 3{,}825 \times 54 = 206{,}550 \text{ inch-pounds} \qquad (36)$$

and the maximum fiber stress due to bending at this point is

$$S_b = \frac{M}{Z} = \frac{206{,}550}{21.4} = 9{,}650 \text{ pounds per square inch} \qquad (37)$$

which is well within the limit of safety.

If the next size smaller standard channels had been used for these members, the results would have been as follows: A 12-inch 20½-pound channel for the yard-arm gives a maximum unit stress at D of about 14,000 pounds, which is too much. The unit compressive stress in the brace, if made of 10-inch 15-pound channels, would be about 2,190 pounds. Rankine's formula for this would allow 2,830 pounds, but there is not enough margin of safety with the high ratio of e to r, which is here about 240. The maximum stress in the mast at H would be 15,400 pounds per square inch. It will thus be seen that the sizes we have selected are the commercial sizes best suited for the case in hand.

CHAPTER III

CALCULATIONS FOR THE SHAFT, GEARS, AND BEARINGS OF CRANE MOTORS

To illustrate definitely the use of the table and diagrams in the method of calculation explained in the present chapter the following example will be taken:

Given: A crane motor with 4 poles, 15 H. P., 750 R. P. M. at normal load:

Diameter of armature	**= 9½ inches**
Air gap	**= 3/32 inch**
Area of pole face	**= 29 square inches**
Density in air gap, given in lines of force per square inch	**= 55.000**
Total weight of rotating parts	**= 150 pounds**

From a general layout drawing of the motor we have the dimensions given as in Fig. 10.

Motors for hoisting purposes are usually series wound, and thus run at different speeds under different loads. Therefore, if the motor

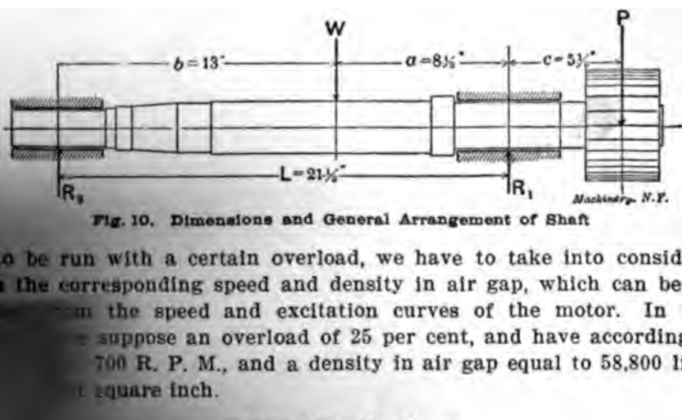

Fig. 10. Dimensions and General Arrangement of Shaft

is to be run with a certain overload, we have to take into consideration the corresponding speed and density in air gap, which can be obtained [illegible] the speed and excitation curves of the motor. In this [illegible] suppose an overload of 25 per cent, and have accordingly: [illegible] 700 R. P. M., and a density in air gap equal to 58,800 lines [illegible] square inch.

Calculating the Gear

[illegible] figuring the gear we suppose that the diametral pitch equals 5 [illegible] the number of teeth equals 18. This gives us a pitch diameter

$= \frac{18}{5} = $ [illegible] Thus, the pitch line speed at 700 R. P. M. =

[illegible] feet per minute.

VALUES OF C_1 FOR 15° INVOLUTE TEETH OF ONE-INCH FACE, WHICH PRODUCE FIBER STRESS OF 1000 POUNDS PER SQUARE INCH

Number of Teeth in Gear.	Circular Pitch.																
	3.14	2.51	2.10	1.80	1.57	1.40	1.25	1.14	1.05	0.96	0.89	0.84	0.79	0.70	0.63	0.57	0.52
	Diametral Pitch.																
	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3½	3¾	4	4½	5	5½	6
12	210	168	140	120	105	93	84	76	70	65	60	56	52	47	42	38	35
13	220	178	147	126	110	98	88	80	73	68	63	59	55	49	44	40	37
14	226	180	151	129	113	100	90	82	75	69	65	60	56	50	45	41	38
15	236	189	157	135	118	105	94	86	79	73	67	63	59	52	47	43	39
16	242	194	161	138	121	107	97	88	81	74	69	64	60	54	48	44	40
17	251	200	167	143	125	112	100	91	84	77	72	67	63	56	50	46	42
18	261	208	174	149	130	116	104	95	87	80	75	70	65	58	52	47	43
19	273	218	182	156	136	121	109	100	91	84	78	73	68	61	54	50	45
20	283	227	189	162	142	126	113	103	94	87	81	75	71	63	57	51	47
21	289	232	193	165	144	128	116	105	96	89	83	77	72	64	58	52	48
23	295	236	197	169	147	131	118	107	98	91	84	79	74	65	59	53	49
25	305	245	203	174	152	136	122	111	102	94	87	81	76	68	61	55	51
27	314	250	210	180	157	140	126	114	104	97	90	84	78	70	63	57	52
30	320	256	213	183	160	142	128	116	106	99	91	85	80	71	64	58	53
34	327	262	218	188	164	146	131	119	109	101	94	88	82	73	66	59	54
38	336	268	224	192	168	148	134	122	112	103	96	90	84	75	67	61	56
43	346	277	230	198	173	154	138	126	115	106	99	92	86	77	69	63	57
50	352	282	235	200	176	156	140	128	117	108	100	94	88	78	70	64	58
60	358	286	238	204	179	159	143	130	119	110	102	95	89	80	71	65	59
75	364	292	243	208	182	162	146	132	121	112	104	97	91	81	73	66	61
100	371	297	247	212	185	164	148	135	124	114	106	99	93	82	74	67	62
150	377	302	251	215	188	167	151	137	126	116	108	100	94	84	75	68	63
300	384	308	256	219	192	171	152	140	128	118	110	102	96	85	77	70	64
Rack	390	312	260	223	195	173	156	142	130	120	112	104	97	87	78	71	65

It is not advisable to use a pitch-line speed exceeding 1,000 feet per minute, on account of noisy running.

Before figuring the width of gear we have to determine the pressure P on the teeth. This is given by the following formula

$$P = \frac{\text{H. P.} \times 33{,}000}{\text{Pitch line speed}}$$

where P is expressed in pounds and pitch line speed in feet per minute Thus for 18.75 H. P. and a pitch line speed of 660 feet per minute

$$P = \frac{18.75 \times 33{,}000}{660} = 940 \text{ pounds, approximately.}$$

The width of gear is given by the formula:

$$w = \frac{P}{f \times C_1 \times C_2} \text{ where}$$

w = width of tooth of gear in inches,

P = pressure on tooth in pounds,

f = permissible fiber stress in thousands of pounds per square inch,

C_1 = coefficient depending on diametral pitch and number of teeth in gear. Its values can be obtained from table given on page 25.

C_2 = coefficient depending on pitch line speed. Its values can be obtained from curve in Fig. 11. If we suppose a gear of steel, we may use a fiber stress of 8,500 pounds per square inch. We therefore get as the width of tooth:

$$w = \frac{940}{8.5 \times 52 \times 0.5} = 4\tfrac{1}{4} \text{ inches, approximately.}$$

Forces Acting on Shaft

Besides the weight of the rotating part and the pressure on the gear, we must, when figuring the shaft and bearings, take into consideration the unbalanced magnetic pull caused by a displacement of the arma- ˙he motor in relation to the poles. If B is the density given in ꞏ per square inch at air gap, A is the area of pole face in and k is a constant which

for 4-pole machines = 2,
6-pole machines = 4.7,
8-pole machines = 7,

$$\text{magnetic pull per pole} = \frac{B^2 \times A}{k \times 72{,}134{,}000} \text{ pounds. This for-}$$

us in our example a magnetic pull per pole $= \dfrac{58{,}800^2 \times 29}{2 \times 72{,}134{,}000}$

roximately. The pull per inch of the circumference

$$\frac{\times \text{ pull per pole}}{\times \text{ pole bore}} = \frac{4 \times 700}{\pi \times 9.69} = 92 \text{ pounds, approxi-}$$

ment of armature of 25 per cent of the

normal air gap, the ratio between air gap and displacement = 4. In the diagram, Fig. 12, reading on the vertical side the pull per inch of the circumference = 92, and on the horizontal side the ratio between air gap and displacement = 4, the line 55 passing through the intersection point of 92 and 4 indicates that the half value of the maximum magnetic pull per inch of circumference of pole bore is 55.

$\frac{M_{max}}{2} = 55$ pounds. Thus $M_{max} = 55 \times 2 = 110$ pounds. In order

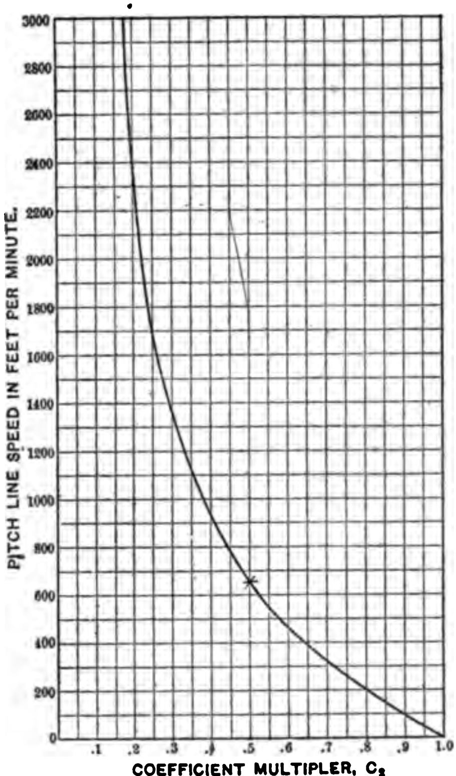

Fig 11

to give the values of $\frac{M_{max}}{2}$ as exactly as possible for a wide range of values of magnetic pull and displacement ratios, the diagram in Fig. 12 contains two sets of lines, one in the lower right corner on a small scale, and one in the upper left corner on a larger scale.

$$\text{The total magnetic pull on armature} = \frac{4 \times \text{radius of armature} \times M_{max}}{\pi}$$

In our example the radius of armature is 4.75 inches, and the M_{max}

is 110 pounds. The total magnetic pull therefore $= \dfrac{4 \times 4.75 \times 110}{\pi} =$ 665 pounds.

When the unbalanced magnetic pull is acting in the same direction as the weight of the rotating part, the shaft is subjected to its worst strain. Therefore, by adding these two forces we get the resulting force

$$W = 665 + 150 = 815 \text{ pounds.}$$

In general, the location of this force W on the shaft will practically lie at the center line of the armature.

The forces R_1 and R_2 acting on the bearings, as shown in Fig. 10, will be found from the following equations:

$$R_1 = \frac{W \times b + P \times (c + L)}{L} = 1{,}680 \text{ pounds.}$$

$$R_2 = W + P - R_1 = 75 \text{ pounds.}$$

Diameter of Shaft

The diameter of the shaft between the bearings (see Fig. 10) must be calculated for the maximum bending moment occurring. The bending moment at W is:

$$M_b = R_2 \times b = 975 \text{ inch-pounds.}$$

The bending moment at R is:

$$M_b = P \times c = 5{,}050 \text{ inch-pounds.}$$

This is, consequently, the maximum bending moment, and the shaft should be calculated accordingly.

The twisting moment for 18.75 H. P. and 700 R. P. M. is

$$M_t = \frac{18.75}{700} \times 63{,}024 = 1{,}690.$$

The combined moment M_c of M_b and M_t is:

If M_b is greater than M_t

$$M_c = 0.975 \times M_b + 0.25 \times M_t;$$

or, if M_b is less than M_t

$$M_c = 0.6 \times M_b + 0.6 \times M_t.$$

our example, where $M_b > M_t$,

$$M_c = 0.975 \times 5{,}050 + 0.25 \times 1{,}690 = 5{,}340, \text{ approximately.}$$

Now, if f = fiber stress in shaft per square inch, and D = diameter f shaft at W in inches, the moment of resistance

$$M_r = 0.1 \times f \times D^3$$

re if we put $M_r = M_c$,

$$D = \sqrt[3]{\frac{M_c \times 10}{f}}$$

fiber stress $f = 8{,}500$ pounds per square inch, we get

$$D = 1\tfrac{7}{8} \text{ inches, approximately.}$$

imum required diameter of the shaft in the bearing at R_1. Of course, ordinarily both bearings are made the same. It is evidently of advantage to have the diameters of the journals somewhat larger than calculated, as strength alone is not the only consideration; the lubrication of the bearing, a low unit pressure per square inch of projected area attained without excessive length of journal, etc., are also important questions to consider. The diameter of the journals would therefore, in this case, be made, say 2¼ inches.*

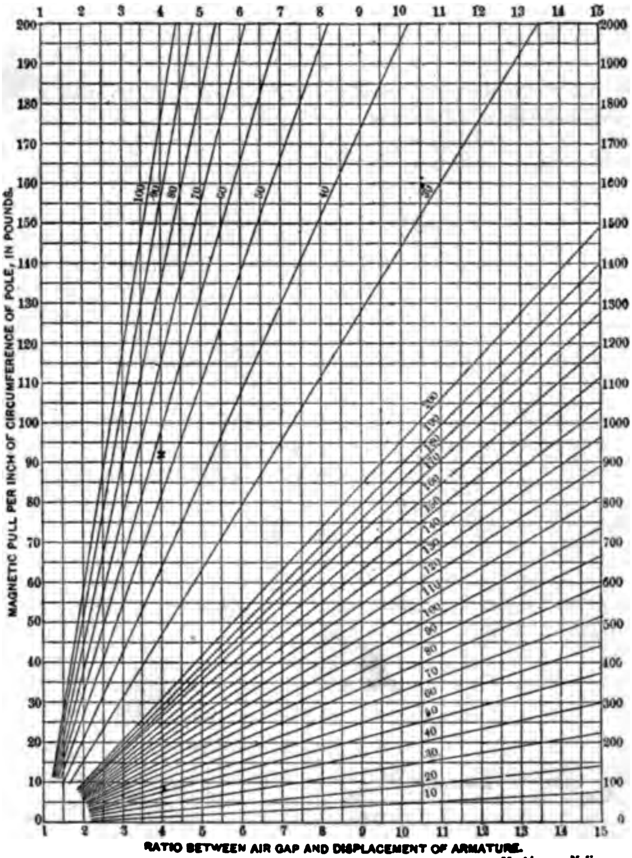

Fig. 12

Now, when the diameters at the journals are made 2¼ inches, evidently the remainder of the shaft at W would not be made as small as calculated, or 1⅞ inch in diameter. The mechanical design requires that this latter diameter be made larger than the journals, say 2½ inches, which diameter we will then use for calculating the deflection, as indicated later.

* For a more thorough discussion of the subject of journals and bearings, see MACHINERY's Reference Series No. 11, Bearings, page 3, The Design of Bearings.

Calculation of Journals

In the bearings, it is not advisable to exceed a pressure per square inch of projected area of 150 pounds, nor should the product of this pressure by the peripheral velocity in feet per minute in the journal be greater than 55,000, when grease lubrication is used. For oil lubrication this product can be somewhat higher. If in our example we assume a pressure of 130 pounds per square inch, with the diameter of shaft at $R_1 = 2\frac{1}{4}$ inches, we obtain

$$\text{length of journal} = \frac{1{,}680}{130 \times 2.25} = 5\frac{3}{4} \text{ inches.}$$

At 700 R. P. M. and diameter of shaft $= 2\frac{1}{4}$ inches, and a pressure of 130 pounds per square inch in journal, the product of pressure by

$$\text{velocity will be} = \frac{130 \times \pi \times 2.25 \times 700}{12} = 53{,}500. \text{ approximately.}$$

Maximum Deflection of Shaft

For calculating the maximum deflection S of the shaft we have the following formula:

$$S = \frac{W \times a \times b \times (2L - a)}{9 \times E \times I \times L} \times \sqrt{\frac{a \times (2L - a)}{3}}$$

where S is in inches and

W = the resulting force acting on the shaft in pounds,

L = distance between centers of journals in inches,

a = shortest distance between center line of one bearing and the acting point of force W,

$b = L - a$ in inches,

E = modulus of elasticity,

= 29,000,000 for steel,

= 27,000,000 for wrought iron,

I = moment of inertia of shaft $= 0.0491 \times D^4$, where D is the diameter of shaft in inches.

In this example we get the maximum deflection,

$$S = 0.0027 \text{ inches, approximately.}$$

Most of the formulas given above are empirical, and give only approximate results, but they are exact enough for practical use.

CHAPTER IV

FORCE REQUIRED TO MOVE CRANE TROLLEYS

In designing crane trolleys and similar constructions the force required to move them is not always calculated to a nicety, and the design then based upon the figures. This may be the conception of the man fresh from college, but it more frequently happens that past experience of a case similar to the one in hand is relied upon entirely.

This is both a safe and quick method, when conditions make it possible, provided good judgment is exercised in allowing for differences between the past construction and the proposed new one. The designer is, however, often confronted by a problem in which he has no past experience to draw upon and for which he has no applicable data at hand, or the design may be of a type similar to that of past experience, but so different as to sizes that he is compelled to calculate from elementary principles. Two troublesome questions then arise: First, what theoretical conditions should be taken into account and what ones may be safely neglected? and second, what values should be assigned to the various constants and assumed factors entering into the calculations? The practicability of his designs will depend almost entirely upon the manner in which the above questions are answered.

Taking up the subject of crane trolleys, of the type in which the load is suspended by ropes passing over sheaves in the trolley and hanging block, as illustrated in Figs. 13, 14, and 15, the above questions may be considered as mutually dependent upon each other, and might be answered as follows:

Take into account journal friction of the trolley wheels, trolley sheaves and hanging block sheaves; also the separate weights of load to be carried, hanging block, and trolley.

Neglect friction of ropes in grooves of sheaves, power necessary to bend ropes over sheaves, and the rolling friction of the trolley wheels on the track, allowing these to be taken care of by the assumed coefficient of journal friction.

Neglect inertia, also, for the usual speeds of crane trolleys, since the difference between the coefficient of rest and of motion is sufficient to produce the necessary acceleration.

In choosing the coefficient of friction, consider the general conditions of lubrication as being poor, and consider that it is the coefficient of rest which is required. Assume this coefficient to be the same for all journals. A fair value is 0.1. Having settled these preliminary considerations, general formulas may be developed.

CASE 1. (See Fig. 13.) The conditions are: Two parts of rope supporting the load, one sheave in hanging block, and two sheaves in trolley.

Let W_l = weight of load to be carried,
W_b = weight of hanging block,
W_t = weight of trolley,
P_f = pull on trolley to overcome friction,
S_b = diameter of sheave in block,
J_b = diameter of journal in block,
S_t = diameter of sheave in trolley,
J_t = diameter of journal in trolley sheaves,
D = diameter of trolley wheels,
A = diameter of trolley axle journals,
C = coefficient of friction,

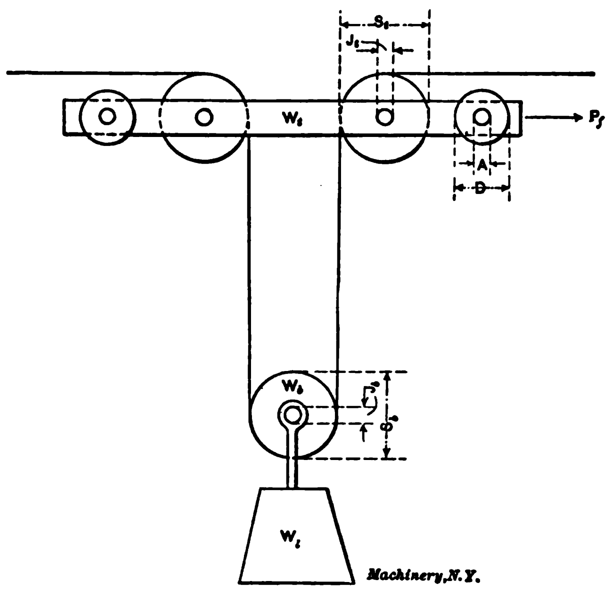

Fig. 13. Trolley with Sheave Suspended by Two Parts of Rope

F_b = friction of hanging block sheave,
= friction of trolley sheaves,
= friction of trolley wheels.
e I,

$$F_b = (W_l + W_b)\, C \frac{J_b}{S_b} \qquad (38)$$

e load being supported by two ropes, the load in each is ½ (W_l +
and the arc of contact of the rope on the trolley sheaves being
rees (a), the resultant pressure on the journals of each of these

$$(W_l + W_b)\, 2 \cos \frac{a}{2} = \frac{1}{2}\,(W_l + W_b)\, 2 \cos 45 \text{ degrees.}$$

aves the resultant pressure amounts to

$$1.4\,(W_l + W_b).$$

From the above we get:

$$F_{ts} = 1.4\,(W_l + W_b)\,C\,\frac{J_t}{S_t} \qquad (39)$$

For the friction of the axle bearings of the trolley wheels, the weight of the load, hanging block, and trolley must be considered, thus:

$$F_{tw} = (W_l + W_b + W_t)\,C\,\frac{A}{D} \qquad (40)$$

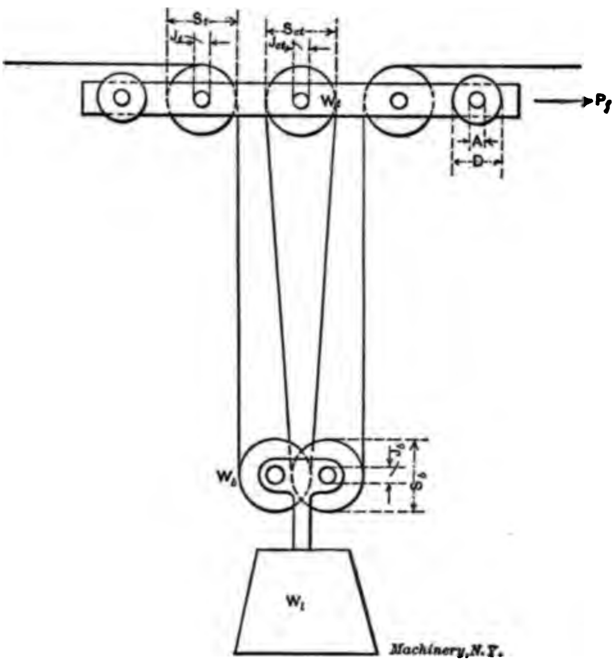

Fig. 14. Trolley with Sheave Suspended by Four Parts of Rope

We have, then, for the total friction

$P_f = F_b + F_{ts} + F_{tw}$, or

$$P_f = C\left[(W_l + W_b)\left(\frac{J_b}{S_b} + 1.4\,\frac{J_t}{S_t}\right) + (W_l + W_b + W_t)\,\frac{A}{D}\right] \qquad (41)$$

Case II. (See Fig. 14.) The conditions are: Four parts of rope supporting the load, two sheaves in the hanging block, and three sheaves in the trolley.

Let the notation be as for Case I with the addition of:

S_{ct} = diameter of sheave at center of trolley,

J_{ct} = diameter of journal for this sheave.

Then, F_b = same as for Case I (equation 38).

F_{tw} = same as for Case I (equation 40).

condition is approximately true, and let R = this ratio, the foregoing formulas for the value of P_f may be reduced to the following form:

$$\text{For case I, } P_f = C\,R\,[8.4\,(W_1 + W_b) + W_t] \qquad (50)$$

$$\text{For case II, } P_f = C\,R\,[3.2\,(W_1 + W_b) + W_t] \qquad (51)$$

$$\text{For case III, } P_f = C\,R\,[3.1\,(W_1 + W_b) + W_t] \qquad (52)$$

It is seen from the above that the friction is nearly the same for the three cases, provided the value of R be the same. Equation (51) being the intermediate condition may then be considered as representative of all.

APPENDIX

CALCULATION OF PILLAR CRANES

The maximum stresses in the different parts of a pillar crane are due to the maximum load lifted (the live load) and the weight (dead load) of the crane parts themselves. Fig. 16 shows a conventional design of a hand pillar crane, and assuming, for an example, the maximum load $Q = 5$ tons, the height $H = 12\frac{1}{2}$ feet, and the radius $A = 13$ feet, the stresses in the different parts of the crane are calculated as shown in the following.

Stresses in the Boom

Fig. 16 shows plainly that the stresses in the boom and tie-bars are not due to the live load only, but that the weight of the eccentric parts of the crane (*i. e.*, boom, tie-bars, hoist, sheave wheels, crane hook and hoisting rope) and the pull of the hoisting rope must also be considered. As it is not possible to determine the dead load accurately before the crane is calculated and designed, it must be assumed. A practical method is to assume the weight of the above mentioned eccentric parts of the crane as half of the maximum live load, and its center of gravity at a distance equal to one-fourth of the radius of the crane from the center line of the pillar. These assumptions expressed in formulas read:

$$Q_1 = \frac{Q}{2}; \text{ or } Q_1 = \frac{10{,}000}{2} = 5{,}000 \text{ pounds.} \qquad (1)$$

$$a = \frac{A}{4}; \text{ or } a = \frac{13}{4} = 3\tfrac{1}{4} \text{ feet.} \qquad (2)$$

[illegible] Q_1 = the weight of the eccentric parts of the crane, and a = [illegible] of the center of gravity of Q_1 from the center of the crane. [illegible] actual figures, determined after the crane is calculated, differ [illegible] from these assumptions, corrections have to be made.

[illegible] step is to determine the height h of the pillar, a practical [illegible] make h about 0.6 of the radius of the crane:

$$[illegible]; \text{ or } h = 0.6 \times 13 = 8 \text{ feet, approximately.} \qquad (3)$$

The frame diagram shown in Fig. 17 can now be drawn. According to the law of equilibrium the sum of moments of the external forces must be equal to the sum of moments of the internal forces about the same center. The moment M_1 of the internal force in the boom is the product of its compressive stress C and its lever arm e (5¾ feet) about center K:

$$M_1 = Ce \quad (4)$$

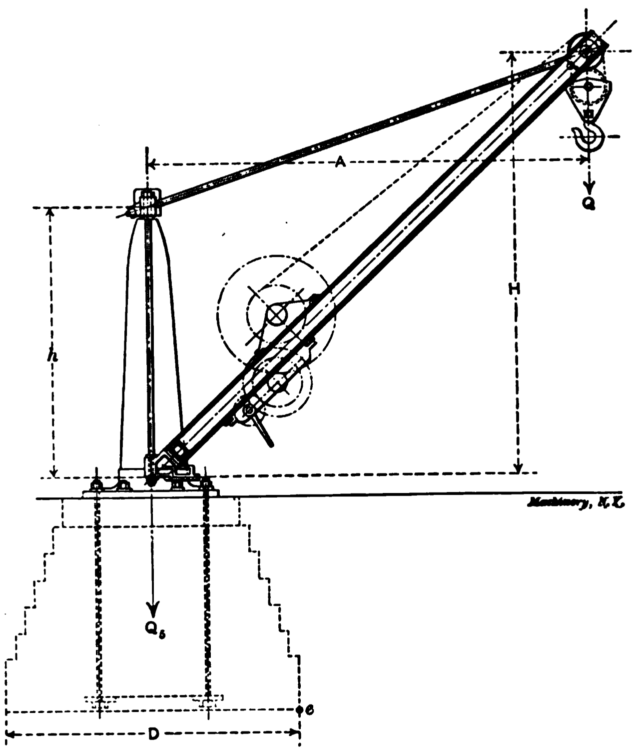

Fig. 16. General Lay-out of Pillar Crane

The moments of the external forces about center K are:

$$\text{The moment of } Q = M = QA \quad (5)$$

$$\text{The moment of } Q_1 = M_1 = Q_1a \quad (6)$$

$$\text{The moment of } Q_2 = M_2 = Q_2b \quad (7)$$

Dimension b is found to be 4 feet by scaling.

Substituting the values:

$M = 10,000 \times 13 = 130,000$ foot-pounds,

$M_1 = 5,000 \times 3\frac{1}{4} = 16,250$ foot-pounds,

$M_2 = 5,000 \times 4 = 20,000$ foot-pounds.

The sum of above moments is:

$$M_s = M + M_1 + M_2 = 130,000 + 16,250 + 20,000 = 166,250 \text{ foot-pounds.} \quad (8)$$

As explained before

$$M_1 = M_s \tag{9}$$

or

$C\,e = M_s$ and transposing

$$C = \frac{M_s}{e} = \text{the compressive stress in the boom.} \tag{10}$$

Substituting the values in above formula

$$C = \frac{166,250}{5\frac{3}{4}} = 28,910 \text{ pounds.}$$

The unsupported length of the boom scales about 17 feet, and as in this case it is made up of two channels the load on each channel is $\frac{C}{2}$ or 14,455 pounds. The inclination of the two channels towards each other need not be taken into consideration as the increase of load is very small. Consulting any handbook of information relating to structural steel we find that a 6-inch × 8-pound channel has a sectional area of 2.38 square inches and a radius of gyration with respect to an axis perpendicular to its web of 2.34 inches. Only this radius of gyration need be considered as the flanges of the two channels are latticed together. For the ratio of the length L of the boom in feet to the radius of gyration r in inches

$$\frac{L}{r} = \frac{17}{2.34} = 7.2,$$

which is not excessive. The elastic limit for soft steel may be taken at 30,000 pounds per square inch. Dividing the load on one channel by its sectional area the actual unit stress will be

$$\frac{14,455}{2.38} = 6,075 \text{ pounds,}$$

which shows that two 6-inch × 8-pound channels are quite sufficient to ... load.

Stresses in the Tie-bars

...sile stress in the tie-bars the same method as just ...used.. Taking K_1 as a center of moments and refer... 17, the moments of the external forces are:

$$\text{Moment of } Q = M = QA, \tag{5}$$

$$\text{Moment of } Q_1 = M_1 = Q_1 a, \tag{6}$$

$$\text{Moment of } Q_2 = M_2 = -Q_2 f. \tag{11}$$

f is found by scaling to be 2½ feet.

... values in these formulas:

... = 130,000 foot-pounds,

... 16,250 foot-pounds.

... = − 12,500 foot-pounds,

...se moments is

$$\ldots 250 - 12,500 = 133,750 \text{ foot-pounds.} \tag{12}$$

The moment M_1 of the internal stress T in the tie-bar is

$$M_1 = Td \tag{13}$$

Dimension d scales 7½ feet.

Since M_1 must be equal to M_s

$$Td = M_s \tag{14}$$

and transposing:

$$T = \frac{M_s}{d} \tag{15}$$

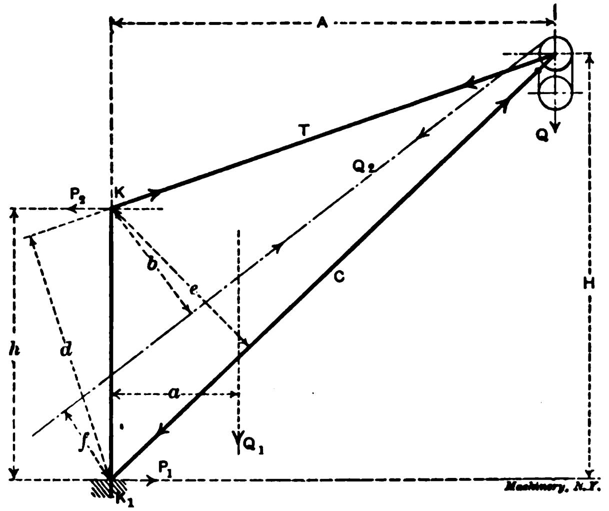

Fig. 17. Diagrammatical View of Pillar Crane

Substituting the values in formula (15) the tensile stress in the tie-bars is: $T = \dfrac{133{,}750}{7\frac{1}{2}} = 17{,}830$ pounds.

As there are two tie-bars the load on one is $\dfrac{17{,}830}{2} = 8{,}915$ pounds.

Using a safe fiber stress of 10,000 pounds per square inch, the area of one bar $= \dfrac{8{,}915}{10{,}000} = 0.892$ square inch with the corresponding diameter of 1 1/16 inch.

Stresses in Pillar

The stresses in the pillar are due to the bending moments of the loads Q and Q_1 and to the direct vertical loads Q and Q_1. The bending moments of Q and Q_1 were found by formulas (5) and (6) to be 130,000 foot-pounds and 16,250 foot-pounds, respectively. The sum of these moments is

$$M_s = M + M_1 = 130{,}000 + 16{,}250 = 146{,}250 \text{ foot-pounds} \tag{16}$$

or 1,755,000 inch-pounds. This bending moment in inch-pounds must be equal to the product of the section modulus of the pillar cross-section, times the safe unit fiber stress. Considering the sectional area of a hollow cylinder for the cast iron pillar, the section modulus is

$$S = \frac{\pi (D^4 - d^4)}{32D} \qquad (17)$$

in which D = the outside diameter of the pillar, and

d = the inside diameter of the pillar.

Using a safe fiber stress $s = 3{,}000$ pounds per square inch, the above mentioned equation reads:

$$M_s = \frac{\pi (D^4 - d^4)}{32D} \times s \qquad (18)$$

Assuming an outside diameter of 24 inches, the inside diameter d is found by transposing the formula (18):

$$d = \sqrt[4]{D^4 - \frac{32 M_s D}{s \pi}} \qquad (19)$$

and substituting the values:

$$d = \sqrt[4]{24^4 - \frac{32 \times 1{,}755{,}000 \times 24}{3{,}000 \times 3.14}} = 20\tfrac{7}{8} \text{ inches.}$$

As mentioned before, not only the bending moment, but also the direct vertical loads Q and Q_1 must be considered. As the bending moment produces a tensile stress on one side of the column and a compressive stress on the other side, the additional vertical loads Q and Q_1 naturally increase the compressive and reduce the tensile unit stress somewhat.

The unit stress in the pillar caused by the vertical loads is

$$s_1 = \frac{Q + Q_1}{\text{Area}} = \frac{15{,}000}{110} = 137 \text{ pounds per square inch.} \qquad (20)$$

110 square inches is the sectional area of the pillar at the dangerous section.

The sectional area of the pillar was calculated for a bending stress of 3,000 pounds. Adding the unit stress for the bending moment and the unit stress for the vertical loads, the actual compressive unit stress is found to be:

$$3{,}000 + 137 = 3{,}137 \text{ pounds per square inch.} \qquad (21)$$

Substracting the unit-stress produced by the vertical loads from the unit bending stress the actual tensile stress in the pillar results:

$$3{,}000 - 137 = 2{,}863 \text{ pounds per square inch.} \qquad (22)$$

Vertical Tie-rods

The vertical tie-rods, connecting the crosshead with the lower end of the boom, receive the vertical component of the tensile stress T in the ties, and the vertical component of the compressive stress C in the boom, or what is the same, the added load Q and Q_1.

$$Q_3 = Q + Q_1 = 15{,}000 \text{ pounds.} \tag{23}$$

in which $Q_3 =$ the stress in the two vertical tie-bars.

Each of the two tie-rods receives half of this load or 7,500 pounds. Using a safe unit fiber stress of 10,000 pounds, the area of one tie-rod is $\frac{7{,}500}{10{,}000} = 0.75$ square inch, with a corresponding diameter of one inch.

Pintle

The reaction P_2 on the pintle (see Fig. 18) is caused by the loads Q and Q_1, whose moments about K must equal the moment of P_2 about the same center:

$$QA + Q_1a = P_2h = P_1h \tag{24}$$

and this formula transposed and the values substituted

$$P_1 = P_2 = \frac{10{,}000 \times 13 + 5{,}000 \times 3\frac{1}{4}}{8} = 18{,}280 \text{ pounds.} \tag{25}$$

The reaction P_2 produces a bending moment on the pintle, and referring to Fig. 18, this bending moment

$$M_b = \frac{P_2L}{2} \tag{26}$$

in which $L =$ the length of the pintle. Assuming the length $L = 1\frac{1}{2}\,D_1$, the formula (26) reads:

$$M_b = \frac{P_2 \times 1\frac{1}{2}\,D_1}{2}.$$

This moment has to be equal to the product of the section modulus of the sectional area of the pintle times the safe unit stress. The section modulus for a circular section being $\frac{\pi}{32}\,D_1^3$, and assuming the safe unit stress $s = 8{,}000$ pounds the equation reads:

$$\frac{P_2 \times 1\frac{1}{2}\,D_1}{2} = \frac{\pi}{32}\,D_1^3\,s \tag{27}$$

and transposing

$$D_1 = \sqrt{\frac{P_2 \times 1\frac{1}{2} \times 32}{2\,\pi\,s}} \text{ or} \tag{28}$$

$$D_1 = \sqrt{\frac{18{,}280 \times 1\frac{1}{2} \times 32}{2 \times 3.14 \times 8{,}000}} = 4\frac{1}{4} \text{ inches approx.}$$

and $L = 1\frac{1}{2} \times 4\frac{1}{4} = 6\frac{3}{8}$ inches.

Besides the bending moment produced by the reaction P_2 the direct vertical loads Q and Q_1 also produce stress, and this stress per square inch is found by dividing the sum of the vertical loads by the sectional area of the pintle in square inches:

$$s_1 = \frac{10{,}000 + 5{,}000}{14.19} = 1{,}060 \text{ pounds per square inch.} \qquad (29)$$

The maximum unit stress on the pintle is then:

$$8{,}000 + 1{,}060 = 9{,}060 \text{ pounds per square inch.} \qquad (30)$$

Foundation Bolts

Considering an axis *A A* in Fig. 19, which shows a plan of the base of the pillar, the sum of the moments of the overturning loads Q and Q_1 about this axis must equal the sum of the resisting moments. The latter are due to the stress in the foundation bolts and to the weight of the pillar. This weight can easily be calculated as the cross-section of the pillar is already known, and in this case is found to be

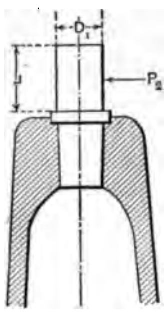

Fig. 18. Pintle of Pillar Crane

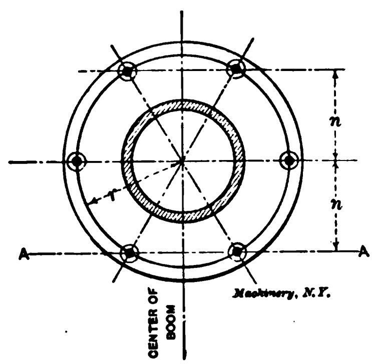

Fig. 19. Lay-out of Arrangement of Flange Bolts

$Q_4 = 5{,}000$ pounds. The moments of the overturning loads with respect to axis *A-A* are:

$$M_4 = Q\ (A - n) \qquad (31)$$

or

$$M_4 = 10{,}000\ (13 - 1\tfrac{1}{2}) = 115{,}000 \text{ foot-pounds,}$$

$$M_5 = Q_1\ (a - n) \qquad (32)$$

$$M_5 = 5{,}000\ (3\tfrac{1}{4} - 1\tfrac{1}{2}) = 8{,}750 \text{ foot-pounds,}$$

in which $n =$ distance of the foundation bolts from the center of the crane. In this case n is found by scaling to equal $1\frac{1}{2}$ foot.

The sum of the overturning moments =

$$M_4 + M_5 = 123{,}750 \text{ foot-pounds.}$$

The resisting moments of the foundation bolts are:

$$M_6 = 2P_3n + (2P_3 \times 2n) \qquad (33)$$

in which P_3 equals the stress in one foundation bolt. The resisting moment of the weight Q_4 of the pillar is:

$$M_7 = Q_4n = 7{,}500 \text{ foot-pounds.} \qquad (34)$$

The sum of the resisting moments is therefore equal to

$$M_4 + M_5 = 2P_3n + (2P_3 \times 2n) + Q_4n \qquad (35)$$

and transposing

$$P_s = \frac{(M_4 + M_5) - M_7}{6n} \tag{36}$$

Substituting the value s, the stress on one foundation bolt

$$P_s = \frac{123{,}750 - 7{,}500}{6 \times 1\frac{1}{2}} = 12{,}910 \text{ pounds.}$$

Using a safe unit stress of 12,000 pounds, the area of one bolt is $\frac{12{,}910}{12{,}000} = 1.08$ square inch with a corresponding diameter of $1\frac{1}{4}$ inch.*

Foundation

Referring to Fig. 16, we find the moments which tend to overturn the crane with its foundation about an axis passing through e to be: Sum of overturning moments =

$$Q\left(A - \frac{D}{2}\right) + Q_1\left(a - \frac{D}{2}\right) \tag{37}$$

This sum of the overturning moments is resisted by the moment of the combined weights Q_5 of the foundation and the pillar:

$$\text{Sum of resisting moments} = Q_5\frac{D}{2} \tag{38}$$

The equation of moments therefore reads:

$$Q\left(A - \frac{D}{2}\right) + Q_1\left(a - \frac{D}{2}\right) = Q_5\frac{D}{2} \tag{39}$$

and transposing

$$Q_5 = \frac{Q\left(A - \frac{D}{2}\right) + Q_1\left(a - \frac{D}{2}\right)}{\frac{D}{2}} \tag{40}$$

Assuming the diameter D of the foundation to be 9 feet and substituting the values:

$$Q_5 = \frac{10{,}000\,(13 - 4\frac{1}{2}) + 5{,}000\,(3\frac{1}{4} - 4\frac{1}{2})}{4\frac{1}{2}} = 17{,}500 \text{ pounds.}$$

Deducting from this combined weight of foundation and pillar the amount for the latter, we get the theoretical weight of the foundation:

$$17{,}500 - 5{,}000 = 12{,}500 \text{ pounds.}$$

* The calculation of the foundation bolts as here given is correct only on the assumption that the base flange of the crane and the bolts are made of inelastic materials. For a more fundamental treatment of the subject of foundation bolts, see MACHINERY, December, 1906, engineering edition: Flange Bolts, or MACHINERY'S Reference Series No. 22, Chapter III. For an article on the Working Strength of Bolts, which should also be considered in this connection, see MACHINERY, November, 1906, engineering edition, or MACHINERY'S Reference Series No. 22, Chapter II.

Using a factor of safety of 3, the weight of the actual foundation must be:

$$12{,}500 \times 3 = 37{,}500 \text{ pounds.}$$

Having calculated the different parts of the crane as described it is good practice to test the pillar for its rigidity, as the amount of deflection must not be too great. The load on the unsupported end of the pillar was found by formula (25) to be $P_2 = 18{,}280$ pounds. The deflection N in inches is:

$$N = \frac{P_2 h^3}{3\,EI} \qquad (41)$$

in which

h = the height of the pillar in inches = 96 inches,
E = the modulus of elasticity = 12,000,000 for cast iron,
I = the moment of inertia $= \frac{\pi}{64}(D^4 - d^4) = 7{,}257$.

Substituting these values we find the deflection

$$N = \frac{18{,}280 \times 96^3}{3 \times 12{,}000{,}000 \times 7{,}257} = 0.062 \text{ inch,}$$

or about 1/16 of an inch, which is not excessive.

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CHAPTER I

CHARTS IN DESIGNING*

Charts which are used for recording data or for assisting the designer in proportioning his work or in making calculations, may be classified, according to the purpose for which they are intended, under two general heads, *i. e.*, Record Charts, and Calculating Charts.

To the first class belong cards from all recording devices, and diagrams of any kind laid out from known values, such as graphical representations of results of tests. Such charts may constitute a record pure and simple; they may be for the purpose of showing better the existing relations of the quantities involved, for ascertaining mean, intermediate, or proportional values, or they may be developed with a view to discovering losses, irregularities, errors, etc.

Record Charts

The following development of one of a series of tests on the power required to drive rivets is a fair illustration of the applicability of record charts:

Experiment on ¾-inch rivets holes punched 13/16 inch at top, about ⅞ inch at bottom, 1.4 inch grip, about 1½ inch projection of blank before driving. The rivets were driven on an Allen riveter, the frame having been previously calibrated as a spring balance by measuring the deflection in thousandths of an inch with a micrometer, for known loads suspended from the die. Chart Fig. 1 shows the calibration of the frame.

In Chart Fig. 2 the full line represents graphically the results of the experiments as given in Table I. The dotted curve represents allowances for efficiency and inequalities in length of the rivets, plotted with a view of determining the energy required for a power-driven machine. In this test the mean average pressure during the period of driving was 40,925 pounds. The distance passed over under this mean pressure was 29/64 inch.

$$\frac{40{,}925 \times 29}{64 \times 12} = 1{,}545 \text{ foot-pounds of energy.}$$

With allowances as per dotted line $= \dfrac{38{,}000 \times 1.25}{12} = 3{,}960$ foot-pounds.

If exerted every 3 seconds the H. P. to drive $= \dfrac{3960 \times 60}{33{,}000 \times 3} = 2.4$. Of course allowances would have to be made for other losses and for extra stroke.

The uses which have developed for charts have become as numerous as the different branches of industry with which they are associated.

* MACHINERY, September, 1904.

Their construction, when taken up in detail, is even more varied than their uses. Cold figures never convey to the mind as clear a conception of relative values as a graphical representation, and in the case of experiments the tabulated data often bring to view, when charted, points which would not otherwise be noted.

Frequently a record chart is used to interpolate between known values, obtaining unknown ones, as was the calibration of the riveter frame in the foregoing example.

Similar to the last-mentioned class of charts, but closely allied to Calculating Charts, belong stress diagrams, and all kindred matter in the art of graphostatics, so much used in the determination of stresses in framed structures, bending moments in beams, etc.

Classes of Calculating Charts

Calculating Charts may be defined as those which express mathematical relations of quantities. They are of two varieties: Those

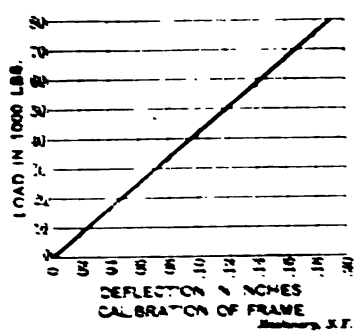


Fig. 1

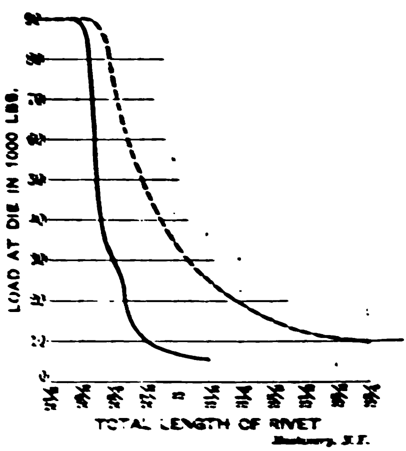


Fig. 2

upon which certain mathematical operations may be performed according to any sequence, the quantities being either concrete or abstract and of any denomination and measure desired; and those designated as dealing only with concrete numbers of definite measure and denomination, the quantities involved having a fixed mathematical relation and sequence.

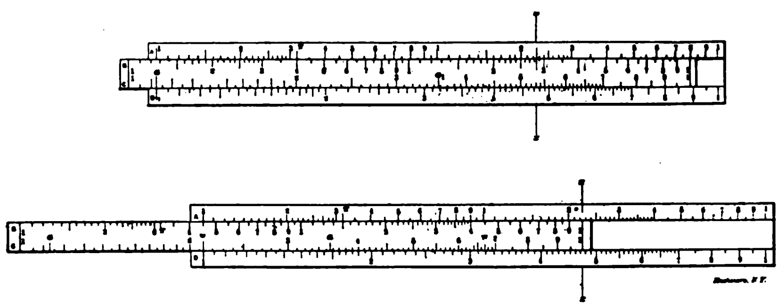
Slide Rules

Under the first subdivision, if this liberty of classification be permissible, come all forms of calculating instruments, such as the slide rule, Sexton's omnimeter, Thatcher's calculating machine, etc. The principles upon which these operate are quite generally known, leaving little to be said in this connection. There is, however, a spirit of trust far too prevalent regarding the accuracy of calculations performed on a slide rule. Because one reads from the rule 975 pounds as the weight of a casting and the actual multiplications give 974.235 pounds, is that any reason why the rule should not be used for estimating weights? In all probability one would set the weight down as 980 pounds for the sake of even figures, and if a casting, it might come out of the sand weighing over 1,000 pounds.

TABLE I

No. of Rivets.	Deflection of Frame.	Pressure at Die, pounds.	Total Length of Rivet Head to Head	Height of Head being Formed.	Remarks.
1	.016	6000	3 [illegible]	[illegible]	End stove up.
2	.022	9000	2 [illegible]	[illegible]	End stove up ¼" more.
3	.036	15000	2 [illegible]	[illegible]	Head commencing to form in die.
4	.063	26000	2 [illegible]	[illegible]	Head formed in part only.
5	.095	39000	2 [illegible]	[illegible]	Head not completely formed.
6	.150	62000	2 [illegible]	[illegible]	Fairly well formed head.
7	.208	82600	2 [illegible]	[illegible]	Better head, hole well filled.
8	.214	87800	2 [illegible]	[illegible]	Some hotter than last rivet; good head.

In calculating stresses, for instance, who cares for a hundred pounds more or less? If the assumed fiber stress be 10,000 pounds per square inch, an error of even 50 pounds is only 0.5 per cent. The rule has a wide sphere of usefulness, and those persons who cry out so against it as an unreliable instrument are only lacking in judgment as to where and where not extreme accuracy is required.



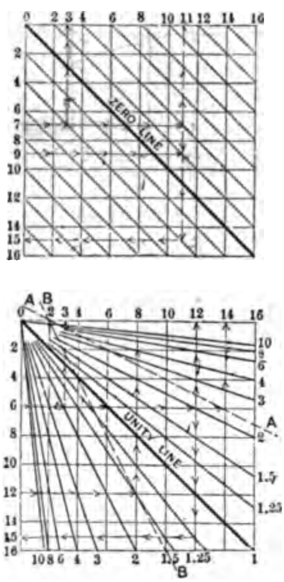
Figs. 3 and 4

The settings, Figs. 3 and 4, illustrate the location of two often used constants which may be applied to any slide rule, and after used a short time their utility will become apparent. In Fig. 3, set 4 on the *B* scale under π on the *A* scale; at left index make mark a on the *C* scale. Set second 4 under π and mark a_1. Now, setting a or a_1 to any diameter on the *D* scale gives the area on the *A* scale opposite either index; then, setting the runner to a given length of cylinder on *B* gives the volume on *A*.

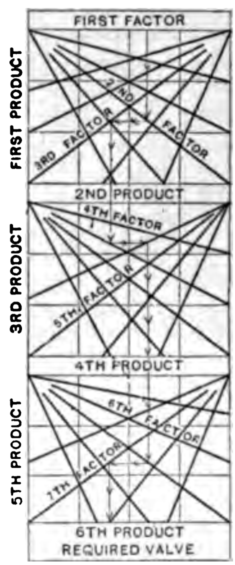
Set the rule as shown in Fig. 3, with the runner at 283 on the *B* scale as indicated at *x*; set the right index of the slide at this line *xx* and at the left index mark *W*, Fig. 4, on the *C* scale. Mark second *W* in line with the middle 1 of the *A* scale. Setting either *W* to any diameter on the *D* scale gives the weight per inch of round steel bars on the *A* scale; setting the runner to the length of the bar on *B* gives the weight of the bar. As an aid to the memory for the uses of these points, *a* stands for areas and *W* for weights.

Calculating Charts

Taking up Calculating Charts in detail, simple mathematics may be divided into three elementary operations, each class consisting of a



Figs. 5 and 6



Machinery, N.Y.

Fig. 7

couplet in which the one member is the inverse of the other, *viz.*: addition and subtraction; multiplication and division; raising to a power or extracting a root.

Chart Fig. 5 illustrates how the operations of addition and subtraction may be performed graphically. Following the fine line in the direction indicated by the arrow heads represents operations as follows: $7 - 4 = 3$, $7 + 4 = 11$, $7 + 4 + 4 = 15$, $9 + 2 = 11$, $9 + 2 + 4 = 15$. Following these lines in the opposite direction would perform the inverse operation. It will be noticed that when the zero line is crossed to find the next value, addition is performed, but when the zero line is not so crossed subtraction results, *i. e.*, the quantities represented by the diagonal lines on the same side of the zero line as the quantity it is desired to combine with, are negative.

Chart Fig. 6 illustrates in a similar manner the operations of multiplication and division. The arrows indicate the following operations: $6 \div 2 = 3$, $6 \times 2 = 12$, $6 \times 2 \times 1.25 = 15$. The line *A-A* represents multiplication by 2 combined with addition of a constant whose value is also 2, thus: $6 \times 2 + 2 = 14$. This line is drawn parallel to the multiplier (in this case parallel to 0-2) and has its point of origin at the constant. Line *B-B* represents division by 2 with addition of a constant, thus: $12 \div 2 + 2 = 8$. It will be noticed that in addition and subtraction the parallel diagonal lines have their origin at the value they represent, and their terminus at a similar value on the side adjacent, if the scale on that side were reversed; while in multiplication and division the lines are divergent, and have their origin at the common zero point of the two scales.

In Figs. 5 and 6 the operations may be carried on continuously within the range of their readings, thus: $[(9 + 2 = 11) + 4 = 15] - 2 = 13$, and similarly, $[(6 \times 2 = 12) \div 1.5 = 8] \times 1.25 = 10$. This is dealing with abstract numbers, however. A single line chart handling concrete values will combine two quantities and give the result of whatever the operation may be. A chart having two sets of sloping lines will combine three values giving the result. If more than three values are to be combined, a multiple chart, as illustrated in Fig. 7, may be used. The following is a concrete example of the last-mentioned chart, illustrating plain multiplication and division. (See Fig. 8.)

Tank Chart

Required: the thickness of tanks of various diameters and heights for containing different fluids. The formula for thin cylinders is as follows: $t = \dfrac{pd}{2sy}$, where t = thickness in inches, p = pressure in pounds per square inch, d = diameter in inches, s = allowable working stress in pounds per square inch, and y = the efficiency of the riveted joint.

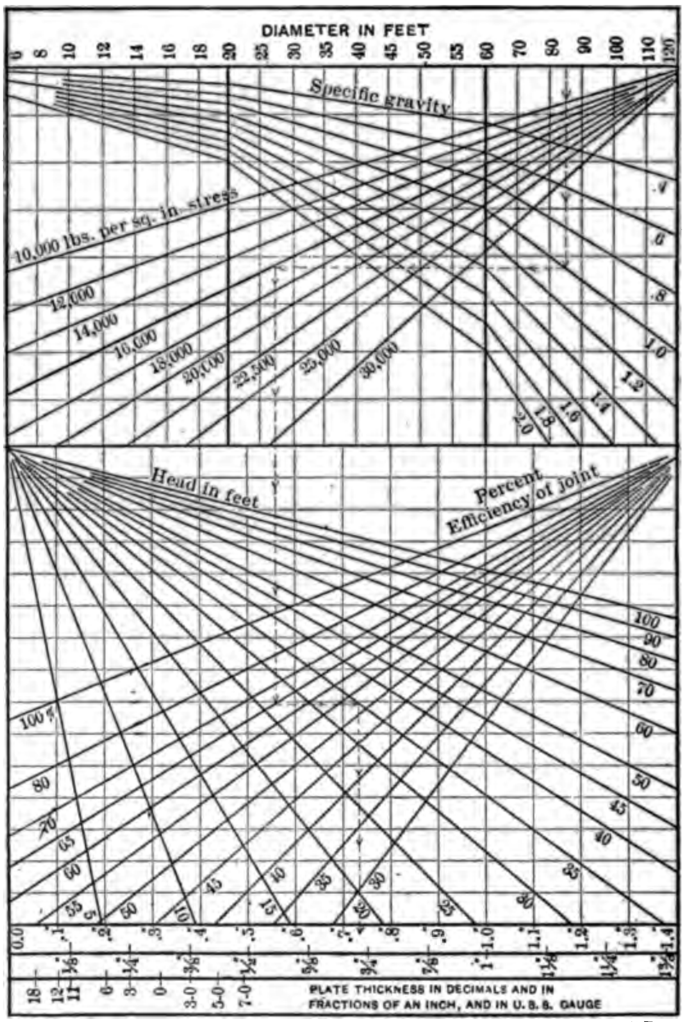
Let D = diameter in feet, H = head in feet, and G = specific gravity of fluid; then

$$\frac{0.433\, HG\, 12\, D}{2\, sy} = 2.598 \frac{HGD}{sy}$$

In operations of multiplication and division the order in which the factors are taken does not affect the value of the result; it is therefore advisable to choose those quantities which are most constant for any given problem as the first factors of the chart. In this case the order has been chosen as follows: diameter, specific gravity, stress, efficiency of joint, and head.

Next in order of importance come range of readings and scales. It is always desirable to make the range of a chart as great as is consistent with reasonable accuracy. In the case under consideration the range of diameters has been chosen between 6 and 120 feet, but in order to secure this range three different scales have been used; 2 feet

per space from 6 feet up to and including 20 feet; 5 feet per space from 20 feet up to and including 60 feet; and 10 feet per space from 60 feet up to and including 120 feet. (See Fig. 8.) This change of scales necessitates three different points of origin for the lines representing



Machinery, N.Y.

Fig. 8. Chart for Tank Calculations

the second factor, but the error of interpolation is thus kept within reasonable limits without making a chart of abnormal proportions.

To aid one in deciding upon appropriate scales and limits of readings for the other factors and products, it is well to tabulate these factors and their resulting products, first choosing assumed limits of reading for the intermediate quantities which will produce the minimum final value, then *vice versa*, and lastly, what may be considered a practical problem of maximum sizes. (See Table II.) From these values the range of the different products has been chosen, as indicated in the last two lines of the table. Their scale will be determined by the size of the sheet it is desired to use. Since it is not necessary

TABLE II. TABULATED FACTORS AND PRODUCTS FOR USE IN SELECTING LIMITS OF READINGS AND SCALES.

Assumed Values.	First Factor, Diameter in Feet = D	Second Factor, Specific Gravity = G	First Product = $D\,G$	Third Factor, 1 ÷ stress = $\frac{1}{S}$	Second Product = $\frac{D\,G}{S}$	Fourth Factor, 1 ÷ Efficiency of Joint = $\frac{1}{y}$	Third Product = $\frac{D\,G}{Sy}$	Fifth Factor, Head in Feet × Constant = $H \times 2.598$	Fourth Product, Thickness in Inches = $2.598 \frac{H\,G\,D}{Sy}$
Assumed values giving minimum final reading	6	0.65	3.9	$\frac{1}{25000}$	0.00015	$\frac{1}{0.80}$	0.00019	10 × 2.598	0.005
Assumed values giving maximum final reading	120	2.0	240	$\frac{1}{8000}$	0.03	$\frac{1}{0.50}$	0.06	100 × 2.598	15.6
Practical problem of maximum size	120	1.1	132	$\frac{1}{16000}$	0.00825	$\frac{1}{0.60}$	0.01375	40 × 2.598	1.43
Fixed upon as minimum reading......	6	0.4	0	$\frac{1}{30000}$	0.0002	$\frac{1}{1.0}$	0.0	5 × 2.598	0.0
Fixed upon as maximum reading......	120	2.0	160	$\frac{1}{10000}$	0.0086	$\frac{1}{0.30}$	0.015	100 × 2.598	1.4

to have the points of origin lie within the border lines, it is desirable to keep the minimum readings greater than zero when this can be done without impairing the utility of the chart.

The problem illustrated by the arrows in Fig. 8 is a tank 85 feet in diameter, specific gravity = 1.0, 16,000 pounds fiber stress, 65 per cent efficiency of joint, and 35 feet head. The thickness is read to the nearest even fraction on the side of safety as ¾ inch.

Cone Pulleys

The chart Fig. 10, for cone pulleys, illustrates the variable scale principle combined with subtraction and division. For crossed belts the belt will be of uniform tension when $A + A_1 = B + B_1 = C + C_1$, etc., Fig. 9, but for open belts the middle steps must be larger in diameter than for crossed belts.

The formula used for this case is as follows: (See Fig. 9.)

$$x = \frac{(A-A_1)^2-(B-B_1)^2}{2\pi L};\quad x_1 = \frac{(A-A_1)^2-(C-C_1)^2}{2\pi L};$$

$$x_2 = \frac{(A-A_1)^2-(D-D_1)^2}{2\pi L}, \text{ etc.}$$

where x = total correction to be added to the sum of $B + B_1$, x_1 = total correction to be added to the sum of $C + C_1$, etc., to preserve a uniform belt tension, the diameters used in the formula being to the center of the belt.*

Proceeding to construct a chart which will give the values of x, x_1, x_2, etc., for any case within its range, we first inspect the for-

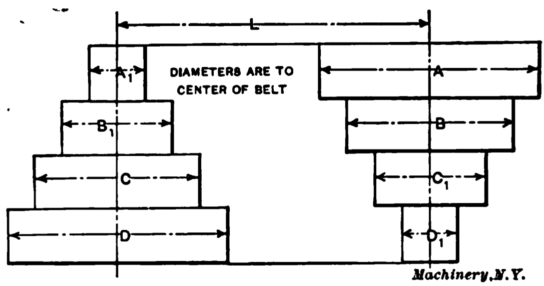


Fig. 9

mula. We have three main values involved: $(A - A_1)^2$, $(B - B_1)^2$ and $2\pi L$, the difference between the first two being divided by the last. As 2π is a constant it does not enter as one of the variable factors. The first two factors are to be squared. This is best accomplished in the following manner: Lay out at the top of the chart in lead pencil a uniform scale (in this case the range has been chosen from 0 to 1,000); this represents the values of $(A - A_1)^2$; now spot off underneath this scale by the aid of a table, the square roots of such numbers as will make a fairly uniform looking scale, inserting their values in ink, and erasing the former scale. In this case we wish to subtract from the first quantity $(A - A_1)^2$ a similar quantity $(B - B_1)^2$. Since the range of $(B - B_1)^2$ should be nearly the same as for $(A - A_1)^2$, lay off on one side of the chart the same uniform scale from 0 to 1,000 starting with 0 at the right-hand corner on the top scale. Now, parallel lines drawn from one value on the top scale to an equal value on the side scale will represent the values $(B - B_1)^2$. The lines have the same values as their point of origin on the top scale and need not be numbered. The squaring operation has already been performed by the variable scale. The values on the side now represent the result of

* See MACHINERY'S Reference Series No. 14, Details of Machine Tool Design, page 3: Elementary Principles of Cone Pull ys and Belts.

the first operation and are equal to $(A - A_1)^2 - (B - B_1)^2$, which value is to be divided by $2\pi\ L = 6.283L$.

Now lay off a uniform scale on the bottom for values of x, x_1, x_2, etc., which is the required answer. The range of this scale has been chosen from 0 to 2 inches, as this is probably the limit to which one could trust to the accuracy of the chart, since 2 inches increase in the sum

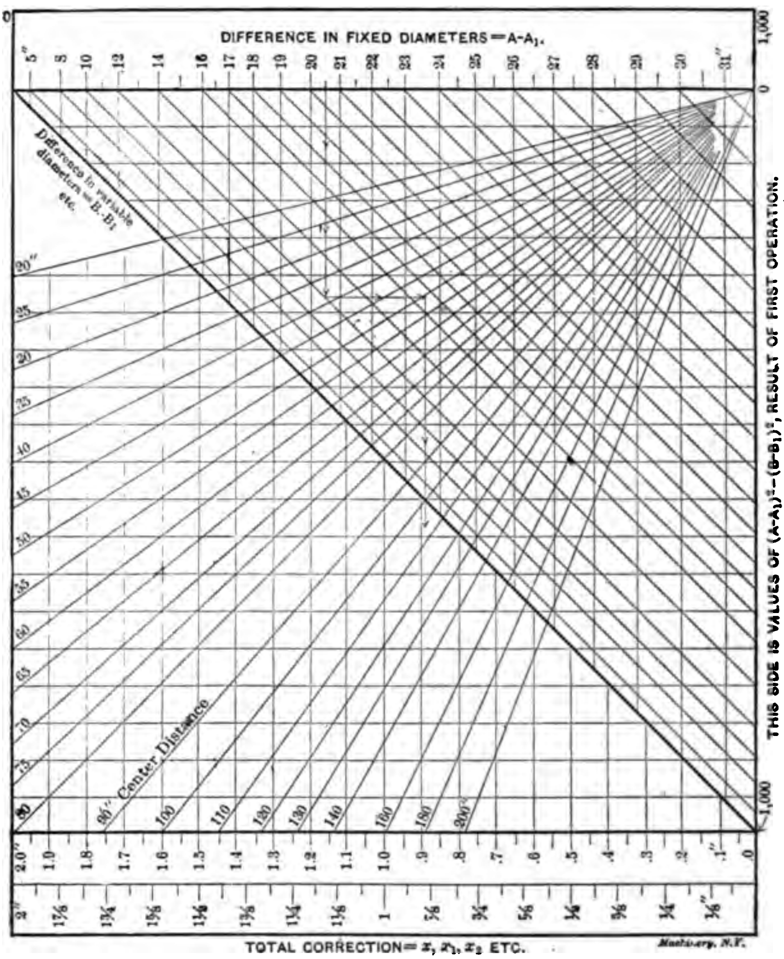


Fig. 10. Chart for Calculating Cone Pulleys

of the diameters would make approximately 6 inches increase in belt length. Choosing the intersection of the graduated upper scale with the zero point of the side scale as a point of origin, draw diverging lines to perform the last operation which is one of division. While these lines have a value of $6.283L$, they are designated by the values of L, the constant affecting the final value only.

Example in a Pair of Cones, Fig. 11

$A = 26.25$ inches,
$B = 22$ inches,
$A_1 = 5.75$ inches,
$B_1 = 10$ inches,

as calculated for crossed belts according to the formula $A + A_1 = B + B_1$, the ratio of speed reduction being fixed by other conditions. $L = 50$ inches. $A - A_1 = 26.25 - 5.75 = 20.5$ inches, $B - B_1 = 22 - 10 = 12$ inches; correction $x = \dfrac{20.5^2 - 12^2}{6.283 \times 50} = 0.879$.

Solving this same problem by Chart Fig. 10, follow as indicated by the arrows; the result is read off at the bottom as 0.88 inch, which is near enough for all purposes. If the pair of cones are both to be cast

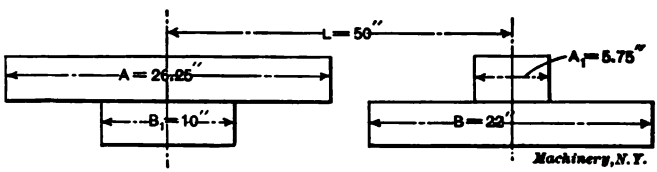


Fig. 11

from the same pattern, the corrections need only be carried out up to and including the middle step, as the remaining steps are obviously duplicates of the already corrected ones.

Charts for Belt Transmissions*

The charts given in Figs. 12 and 13 will prove of value in calculations of transmissions. Fig. 12 is for the approximate determination of the width of belt to transmit a given horse-power at any stipulated speed. The formula upon which the chart is based is:

$$W = \frac{33{,}000 \times \text{H.P.}}{\dfrac{T}{2} \times V}$$

where W = width in inches,
T = working tension in belt,
V = velocity in feet per minute.

The three variable factors are H.P., $\dfrac{1}{T}$, and $\dfrac{1}{V}$. The above formula is based upon a coefficient of friction of 0.3 and an arc of contact of 135 degrees approximately, and takes no account of centrifugal action, which considerably reduces the effective belt tension at high velocities. We are only concerned with the constants in fixing the limits of the chart.

In plotting this chart, maximum and minimum values are taken as given in the accompanying table:

* Machinery, April, 1905.

	H. P.	T	V
Maximum	100	180	5,000
Minimum	10	60	500

Horse-power values are plotted to any suitable scale on the left of the diagram (in this figure carried to 120), values of $\dfrac{33{,}000 \times \text{H. P.}}{V}$ along the top line, and from the zero of the horse-power scale sloping velocity lines are drawn through the correct points upward. On the right hand of the chart, downward, a scale of widths is plotted,

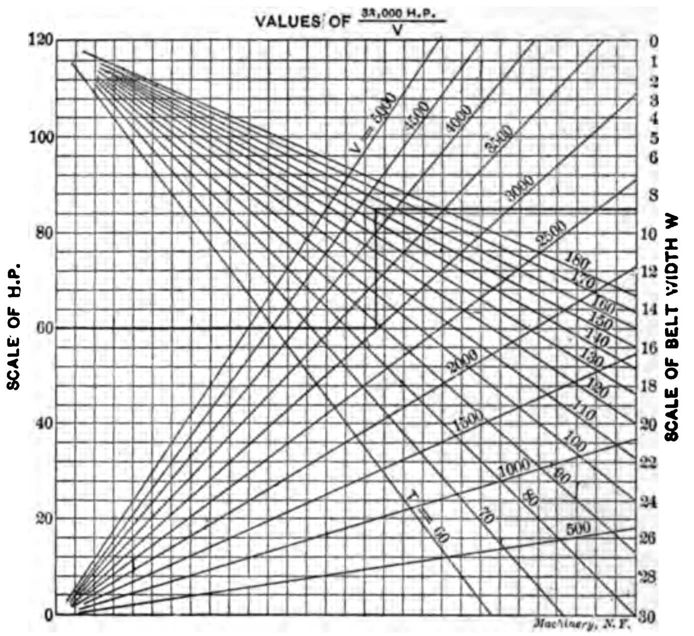


Fig. 12. Chart of Horse-power Transmitted by Belts

reading to 30 inches. From the zero of the top scale sloping lines are drawn downward through the correct points for the several belt tensions.

As an example of the use of this chart, let us find the width of belt necessary to transmit 60 horse-power at a belt speed of 3,000 feet per minute, with an effective tension of belt of 150 pounds per inch of width. Enter the chart at 60 on the horse-power scale, and move along the heavy line in the figure to the intersection of the 3,000 velocity line, thence vertically to the intersection of the 150 tension line, and on the scale of widths find the required width of 9 inches. Obviously the process may be reversed, and, given width, speed, and stress per inch width, we may find the horse-power transmitted.

As the width varies directly as the horse-power, it is obvious that the chart is applicable to any values within a reasonable limit, as by

multiplying or dividing any horse-power on the chart by a multiple of ten, the corresponding width will be the width on the chart with the decimal point correspondingly shifted, and *vice versa*.

Fig. 13 is a chart expressing the relation between revolutions per minute, diameter of pulley in inches, and velocity of periphery in feet

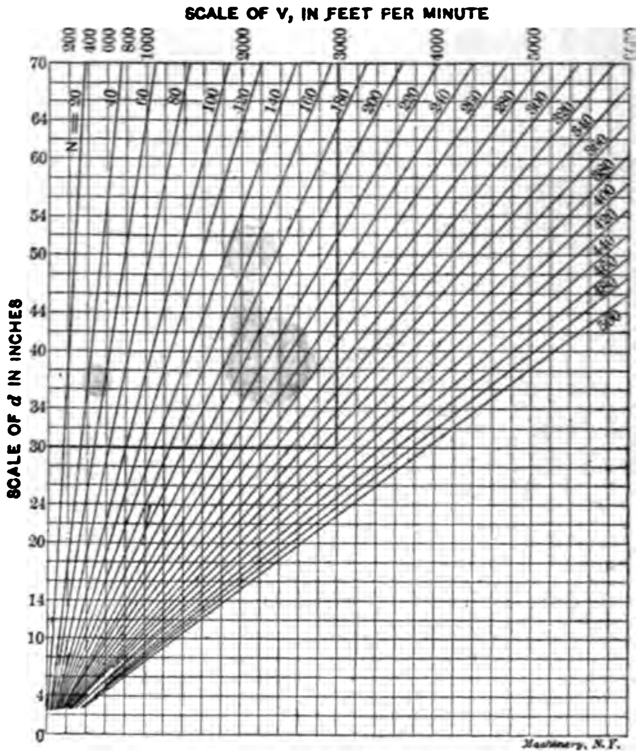


Fig. 13. Chart Giving Belt Speed

per minute, or the relation $V = \frac{\pi d N}{12}$, where N represents revolutions. This chart is drawn to read to a maximum of 6,000 feet per minute. Following the heavy line, we find that a pulley 30 inches in diameter, running at 380 revolutions per minute, has a peripheral speed of 3,000 feet per minute. This chart may well be used in connection with Fig. 12.

CHAPTER II

THE DESIGNING OF MACHINE FRAMES*

For men of limited experience, a machine frame of the common C-type such as is used in punching and shearing machines, in some kinds of riveters, and, in a modified form, in slotters, drill presses, steam hammers, and many other tools, is always more or less difficult to design. The stresses are rather complex, and the means for securing the best distribution of metal for bearing these stresses is not easy. Much has been written on the subject, and the author of this chapter does not claim any originality in the ideas involved in the discussion. It is believed, however, that the method of presenting the solution of the problem has some points that may merit consideration. A common punching and shearing machine frame will be used for illustration, because the form and proportions of these frames are so

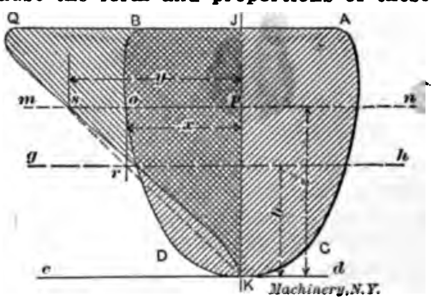


Fig. 14. Graphical Method for Finding the Center of Gravity

well known. The method employed, however, is perfectly general and may be used for many other machine frames.

In the discussion which will follow it will be necessary to use the moment of inertia and center of gravity of an irregular section. The method of finding these quantities which is outlined below is not original with the author, but it is added here in the hope that its use may become more general.

Center of Gravity and Moment of Inertia of Irregular Figures

The following graphical method for finding the center of gravity and moment of inertia of an irregular figure will be found to offer an advantage over the ordinary method by calculation, in the matter of time saved.

Let *ABDC*, Fig. 14, represent any irregular section. In the figure a section symmetrical about a vertical axis is shown, and the constructions carried out for half of the figure. For an unsymmetrical section both sides would have to be considered, as well as a construction for the gravity axis parallel to two base lines. A case of this sort will rarely arise in practice.

* Machinery, August, 1908.

Lay off the line *gh* at any distance *b* from the edge of the section. The line *gh* is perpendicular to the center line *JK*. Then lay off another line *mn*, parallel to *gh* at any distance *z* from *cd*. Let the distance *op* be called *x*. From *o* drop a perpendicular on *gh* cutting at *r*. Join *K* and *r* and extend the line until it cuts *mn* at *s*, and let $sp = y$. After determining several points *s* in the same way connect them by a smooth curve, giving the figure *KQJ*.

Then by our construction we have,

$$\frac{x}{y} = \frac{b}{z};\quad y = \frac{xz}{b}$$

The area of *KQJ* is

$$G = \int y dz = \frac{1}{b}\int xzdz$$

But $\int xzdz$ is the "static moment" of the original figure *KDBJ*, and therefore bG is the static moment of *KDBJ* about *cd*. If *A* is the area of *KDBJ*

$$\frac{bG}{A} = \frac{\text{static moment of } KDBJ}{\text{area of } KDBJ}$$

= distance from *cd* to center of gravity $= e$.

If the areas *A* and *G* and the distance *b* are measured, the distance to the center of gravity is readily found from this last equation. A planimeter offers the best means of measuring the areas, but if this is not possible, sufficient accuracy may be attained by dividing the areas to be measured into rectangles and triangles, computing their separate areas and taking the sum of the areas of all the subdivisions as the area of the original figure.

There is a similar construction for finding the moment of inertia, which should be carried out on the same figure as was used for finding the center of gravity. The two constructions are separated here in order to avoid confusion in explaining them.

In Fig. 15 the lines *gh* and *mn* are laid off as before so that *pq* is *x*, and *pw* is *y*. Then *we* is drawn from *w* perpendicular to *gh*; join *K* and *e* and extend *Ke* to *f*, calling $pf = y_1$. Several points *f* will give the figure *KSJ*, and we have, as before,

$$\frac{x}{y} = \frac{b}{z};\quad y = \frac{xz}{b}$$

and also,

$$\frac{y_1}{y} = \frac{z}{b};\quad y_1 = \frac{yz}{b} = \frac{xz^2}{b^2}$$

Then the area of *KSJ* is

$$J = \int y_1 dz = \frac{1}{b^2}\int xz^2 dz$$

but the moment of inertia

$$I = \int xz^2 dz$$

from which $$J = \frac{1}{b^2} I, \text{ and } I = b^2 J.$$

In this equation we know b and can measure J as before.

This construction gives the moment of inertia about the line cd, whereas it is generally required about the gravity axis. It might have been so found if the base line had been the gravity axis instead of the line cd. However, the mathematical computation for the transformation of axes is easily made. The following formula should be used:

$$I_o = I_{cd} - e^2 A,$$

where I_o = moment of inertia about gravity axis,
I_{cd} = moment of inertia about cd,
e = distance from cd to center of gravity,
A = area of $KDBJ$.

It is to be clearly understood that the values of e and I found above

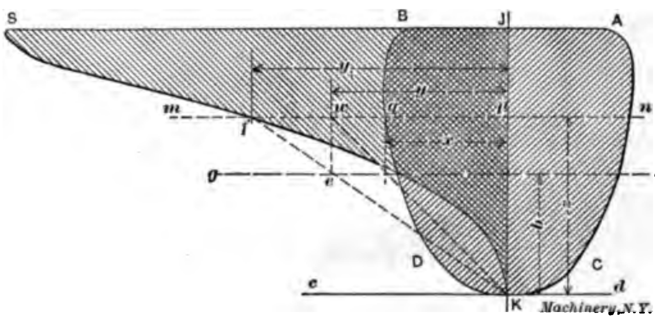


Fig. 15. Graphical Method for Finding the Moment of Inertia

refer only to the figure actually drawn on the drawing board and not to the full-sized section. For the actual values we have

$$I_{actual} = X^4 I_{drawing}$$
$$e_{actual} = X e_{drawing}$$

where X is the scale of the drawing.

Fig. 21 shows the construction applied to a punch frame section.

The Stresses Involved in a Punch Frame

Let Fig. 16 represent the side view of a punch frame. It will be sufficient to investigate the stresses at four different sections, testing at other points later if it appears necessary. The preliminary calculations will be made on sections indicated at AB, CD, EF, and GH or JK.

For determining the stresses on the section AB, consider the upper portion of the punch a "free body" as shown in Fig. 17. It is subjected to an external force, P, acting upwards, arising from the resistance of the material being punched. In the actual punch this force tends to separate the jaws. To balance P there is an internal stress acting over the section AB. The following discussion will show that this internal stress is a compound stress.

Conditions of equilibrium require that the algebraic sum of all es, external and internal, acting on a body must be zero; and also the sum of all couples, or forces tending to produce rotation, must zero. It is then evident that to balance P there must be downward es equal to P. Such downward forces exist in what may be conved as a uniform tensile stress of t_1 pounds per square inch over section AB. The resultant of all these small downward forces ting up the uniform tensile stress can be considered as a single e, P_1, equal to P, acting at the center of gravity of the section AB. algebraic sum of all the forces on the free body is now zero, but nd P_1 together form a couple of value $Pl_1 = P_1l_1$, tending to rotate portion of the punch under consideration in a clockwise direction, that an equal and opposite internal couple is necessary to balance

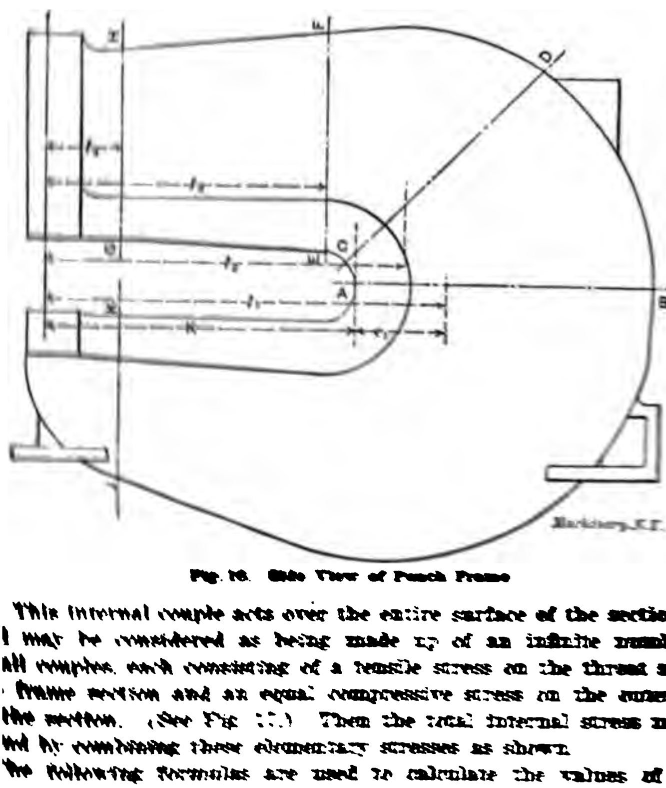


Fig. 18. Side View of Punch Frame

This internal couple acts over the entire surface of the section AB. t may be considered as being made up of an infinite number of all couples, each consisting of a tensile stress on the throat side of frame section and an equal compressive stress on the outer part the section. (See Fig. [illegible]) Then the total internal stress may be nd by combining these elementary stresses as shown.

The following formulas are used to calculate the values of these ernal stresses. Their derivation is comparatively simple, and may found in any treatise on the mechanics of materials.

$P = dts.$ (1)

re d = greatest diameter of hole to be punched on the machine considered

t = thickness of plate punched

s = shearing strength of material punched

$$t_1 = \frac{P_1}{A_1} = \frac{P}{A_1}, \tag{2}$$

where t_1 = uniformly distributed tensile stress over section AB, in pounds per square inch,

A_1 = area of section AB, in square inches.

$$f_1 = \frac{P_1 l_1}{N_1} = \frac{M_1 e_1}{I_1} \tag{3}$$

where f_1 = tensile stress at point A, due to couple $P_1 l_1$, in pounds per square inch,

l_1 = arm of couple in inches $= K + e_1$ (Fig. 17),

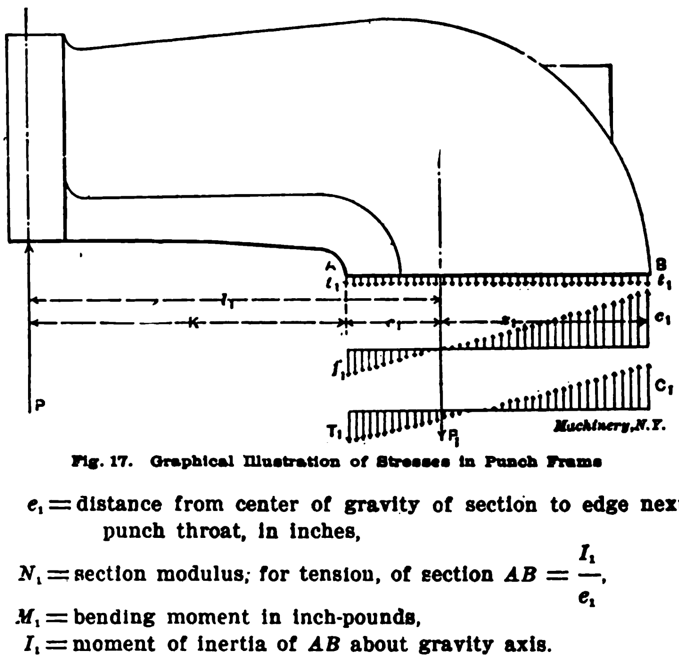


Fig. 17. Graphical Illustration of Stresses in Punch Frame

e_1 = distance from center of gravity of section to edge next to punch throat, in inches,

N_1 = section modulus, for tension, of section $AB = \dfrac{I_1}{e_1}$,

M_1 = bending moment in inch-pounds,

I_1 = moment of inertia of AB about gravity axis.

$$c_1 = \frac{P_1 l_1}{Z_1} = \frac{M_1 z_1}{I_1} \tag{4}$$

where c_1 = compressive stress, in pounds per square inch, at point B, due to couple $P_1 l_1$.

Z_1 = section modulus of section AB, for compression, $= \dfrac{I_1}{z_1}$

where z_1 is the distance from the center of gravity to the outer edge of the section.

The following equations may be written for the maximum stresses on the section AB.

Maximum unit tension

$$= T_1 = t_1 + f_1 = \frac{M_1 e_1}{I_1} + \frac{P_1}{A_1} = \frac{P_1 l_1 e_1}{I_1} + \frac{P_1}{A_1} \qquad (5)$$

Maximum unit compression

$$= C_1 = c_1 - t_1 = \frac{M_1 z_1}{I_1} - \frac{P_1}{A_1} = \frac{P_1 l_1 z_1}{I_1} - \frac{P_1}{A_1} \qquad (6)$$

In equations (5) and (6) everything is known from the conditions that would be given in the problem, as it is received by the designer, except the quantities $\frac{c_1}{I_1}$, $\frac{z_1}{I_1}$ and A_1. It will be seen that these are all factors depending on the size and shape of the section AB under consideration, and these are for the designer to determine.

Method of Designing Frame

After clearly understanding the stresses involved in the design, the next step will be to proportion the frame to withstand these stresses. This may be accomplished in the following manner:

1. Assume the shape of the section at AB, Fig. 16. In making this assumption the designer will have to be guided by experience, either his own or that of others as shown in machines already built, catalogs, textbooks, drawings, photographs, or any other available means of gaining information that is available. The section shown in Fig. 21 will give a fairly good idea of the commonest form used for punches. In making the first drawing it is not necessary to make any assumption as to the *size* of the finished section, the *shape* is all that is necessary.

2. Lay out this section on a drawing board. Use any scale, and make the drawing rather large, say eight inches as the longest dimension. The *size* is of no particular importance, only be careful to get the figure drawn of the same proportions as desired in the final section.

3. Find the gravity axis and the moment of inertia of the figure drawn, by the graphical method explained above.

4. It is evident that the figure drawn will not be the correct *full sized* section of the punch, but it will be a drawing of that section to some scale. Let that unknown scale be such that one inch on the drawing represents X inches on the actual section. If, then, quantities relating to the figure on the drawing board are represented by letters with the subscript d, and quantities relating to the actual section by letters without additional subscripts, the following equations will hold true:

$$e_1 = X(e_1)_d \qquad (7)$$

$$z_1 = X(z_1)_d \qquad (8)$$

$$A_1 = X^2(A_1)_d \qquad (9)$$

$$I_1 = X^4(I_1)_d \qquad (10)$$

We can measure $(e_1)_d$, $(z_1)_d$, $(A_1)_d$, and $(I_1)_d$ directly from the drawing, and insert the values for e_1, z_1, A_1, and I_1 in equations (5) and (6). We then have

$$T_1 = \frac{P_1 l_1 X(e_1)_d}{X^4(I_1)_d} + \frac{P_1}{X^2(A_1)_d} = \frac{P_1[K + X(e_1)_d]X(e_1)_d}{X^4(I_1)_d} + \frac{P_1}{X^2(A_1)_d} \quad (11)$$

$$C_1 = \frac{P_1 l_1 X(z_1)_d}{X^4(I_1)_d} - \frac{P_1}{X^2(A_1)_d} = \frac{P_1[K + X(e_1)_d]X(z_1)_d}{X^4(I_1)_d} - \frac{P_1}{X^2(A_1)_d} \quad (12)$$

Equation (11) can now be used to solve for X, if the allowable value for T_1 is known. A machine of the punch or shear type is subject to severe shock at the moment the tool strikes the metal. For this reason a very low allowable safe stress should be taken. It is suggested that the maximum allowable tensile or shearing stress be not greater than 2,000 to 2,200 pounds per square inch for cast iron. Equation (11) is a 4th power equation, and the quickest and easiest way to solve it is by trial. After finding the value of X from equation (11), it can be substituted in equation (12) to find the value of C_1. This should not exceed 8,000 to 10,000 pounds per square inch. It will usually be found much less than this, because mechanical difficulties are encountered in the manufacture of the punch if the outer part of the section is made thin enough to raise the compressive stress to the highest allowable value. These difficulties are mainly in the shape of extremely high cooling strains in the casting, so that the risk is too great to compensate for the comparatively small saving in iron. Take the value of X found and apply it to the original drawing to find the actual size of section. All linear dimensions of the drawing will be multiplied by X to get the actual size. This will fully determine section AB.

Having section AB, there are two methods of determining other sections. They may be found in a manner exactly similar to that used for section AB, by assuming a *shape* of section and solving for the *size*, or the size and shape may both be assumed and then tested for safety. The latter method will undoubtedly give the most satisfactory results, because it would probably require a great many trials by the first method for the reason that while each section would be strong enough to bear the stresses upon it, there would be considerable difficulty in combining these independently found sections into a pleasing and harmonious whole.

After finding section AB, use it as a basis, together with all the other known data about the punch, from which to assume an outline of the side view. In this part of the work, as in the first assumption of the shape of section AB, reference must be made to the results of past experience, and here again books, drawings, and illustrations will be found helpful. Fig. 16 is a side view of the frame of a punch made by one of the largest manufacturers of machine tools in the United States; if nothing better is at hand, it may be used as a guide. The thickness of the outer part of the frame is kept the same as was found for the section AB, and the frame is a trifle narrower from side to side at the tool end than at the back end.

Now, consider a section along some plane EF, parallel to the line of action of the tool. The "free body" is shown in Fig. 18. The forces acting on the free body are found by a method very similar to that used in discussing the stresses in the section AB. Let all quantities relating to this section be denoted by the subscript 3, and have the same meaning as the same letters had when used with subscript 1 in the discussion of section AB. The following equations may then be used to determine the stresses on the section.

P = the punching pressure.

P_3 = a force equal and opposite to P, causing shearing stress along EF.

P and P_3 with arm l_3 form a couple. This tends to produce bending

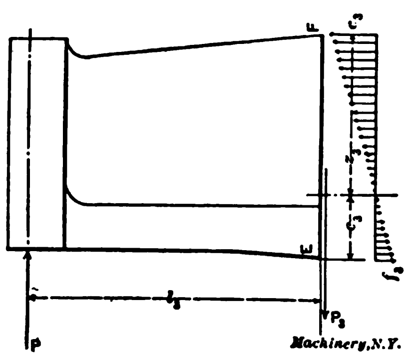


Fig. 18

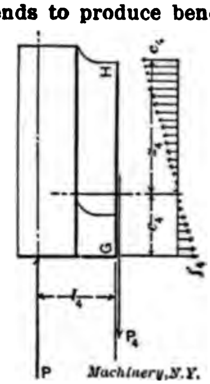


Fig. 19

Graphical Illustrations of Stresses in Punch Frame

in the frame and must be resisted by an internal moment of equal magnitude, M_3.

$$M_3 = P_3 l_3 = P l_3$$

As before, we have the tensile stress in section EF at E due to the couple as

$$f_3 = \frac{M_3 e_3}{I_3} \qquad (14)$$

and the compressive stress in the section at F, due to the couple as

$$c_3 = \frac{M_3 z_3}{I_3} \qquad (15)$$

The shearing stress is distributed uniformly over the entire section and its value is

$$s_3 = \frac{P_3}{A_3} = \frac{P}{A_3} \qquad (16)$$

The maximum tension, compression, and shearing effects are found combining f_3, c_3, and s_3 according to the rules found in various · the mechanics of materials. "Kent's Mechanical Engineer's ". (7th edition, p. 283) gives the following rules, taken from 's "Strength of Materials."

$$T_3 = \tfrac{1}{2}f_3 + \sqrt{s_3^2 + \tfrac{1}{4}f_3^2} \qquad (17)$$

$$C_3 = \tfrac{1}{2}c_3 + \sqrt{s_3^2 + \tfrac{1}{4}c_3^2} \qquad (18)$$

$$S_3 = \pm\sqrt{s_3^2 + \tfrac{1}{4}f_3^2} \quad \text{or} \quad \pm\sqrt{s_3^2 + \tfrac{1}{4}c_3^2} \qquad (19)$$

Since the value of s_3 is uniform over the entire section, while the stresses due to the couple are variable, it is evident that the maximum shearing effect will occur at that part of the section where the stresses due to the couple are a maximum, *i. e.*, probably at *F*.

The section must be designed strong enough to resist both tension and shear. In general, it is not possible to make it equally strong to resist both, when such a material as cast iron is used. At a section near the end, the shearing stress may be the one to fix proportions,

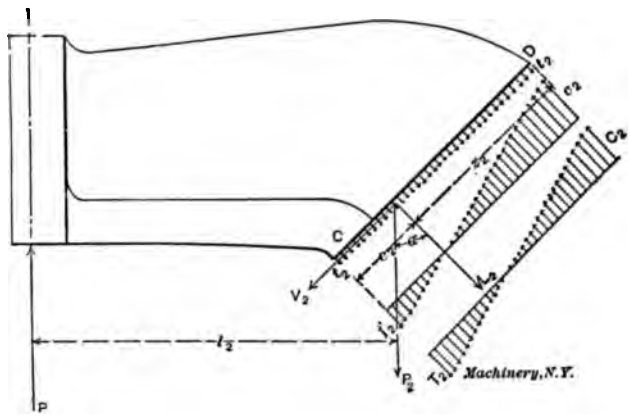


Fig. 20. Graphical Illustration of Stresses in Punch Frame

but when taken near the back of the throat, the tensile stresses are more liable to require consideration.

Equations (17), (18), and (19) should be solved for T_3, C_3, and S_3, and the results should show values within the maxima stated above. If the section is well designed, the values of T_3 and S_3 will not be very different from those mentioned, and as before, the section will probably be excessively strong in compression.

A section taken at *GH* or *JK* would be investigated in a manner exactly similar to that used for the section *EF*. (See Fig. 19.) It may be expected that the section at *GH* will appear much stronger than necessary, but this is because it is necessary to provide a space for the main shaft to pass through, as well as for the sake of appearance. The head will need to be large to accommodate the moving parts within it, and the part of the frame immediately behind the head must not appear too small to support the head.

It is next necessary to consider a section taken at an angle, for example along the line *CD*. The free body is shown in Fig. 20. The forces will be denoted by the same letters as heretofore, but with the subscript 2.

P = the punching pressure.

P_2 = a force equal and opposite to P. P_2 is resolved into two components, L_2 and V_2.

$L_2 = P_2 \cos \alpha$ (produces uniform tension),

$V_2 = P_2 \sin \alpha$ (produces shearing stress).

P and P_2 together form a couple which has a value $P_2l_2 = Pl_2$, which must be resisted by an internal moment, M_2.

$$M_2 = P_2l_2 = Pl_2. \tag{20}$$

As before,

$$f_2 = \frac{M_2e_2}{I_2} \tag{21}$$

$$c_2 = \frac{M_2z_2}{I_2} \tag{22}$$

$$t_2 = \frac{L_2}{A_2} \tag{23}$$

$$s_2 = \frac{V_2}{A_2} \tag{24}$$

The maximum stresses per square inch will then be

$$T_2 = \frac{f_2 - t_2}{2} + \sqrt{s_2^2 + \left(\frac{f_2 - t_2}{2}\right)^2} \tag{25}$$

$$C_2 = \frac{c_2 - t_2}{2} + \sqrt{s_2^2 + \left(\frac{c_2 - t_2}{2}\right)^2} \tag{26}$$

$$S_2 = \pm\sqrt{s_2^2 + \left(\frac{f_2 - t_2}{2}\right)^2} \text{ or } \pm\sqrt{s_2^2 + \left(\frac{c_2 - t_2}{2}\right)^2} \tag{27}$$

Take the greater of the two expressions in equation (27) for the subsequent work. This section will have dimensions between those of AB and EF.

Example of Calculations

In order to illustrate the method of procedure outlined above, the following example is given. The machine in question is designed to punch a 1¼-inch hole in a 1-inch plate. The throat depth is 60 inches.

The punching force necessary may be found from equation (1).

$P = \pi dts = \pi \times 1.25 \times 1 \times 50{,}000 = 196{,}350$ pounds.

The punch frame is to be of the same general form as that shown in Fig. 16, except that the section at JK is to be of the same size as that at GH. By reference to various drawings and photographs the shape of the section at AB is assumed to be that shown in Fig. 21. This is drawn, in this case, on a drawing board, making the longest dimension 5.61 inches (for general practice it would be well to make the drawing larger than this, say 8 inches high), and the figures denoted by G_1 and J_1 constructed for finding the gravity axis and the moment of inertia. In this case only half of the construction is carried out, because the section is symmetrical about the vertical center line, and it will be understood that the areas measured have to be doubled before they are used in the computations, *i. e.*, the values of the areas A_1, G_1, and J_1 used in the following calculations are not the areas actually measured from the drawing as shown in Fig. 21, but twice these values, to account for the construction for the right hand side of the section which is not carried out on the drawing board. With a value of b_1 as 1.98 inch, the areas of A_1, G_1, and J_1 when measured with a planimeter are as follows:

$A_1 = 3.84$ square inches,
$G_1 = 3.02$ square inches,
$J_1 = 4.72$ square inches.

Then the values of the mcment of inertia and the distance from the

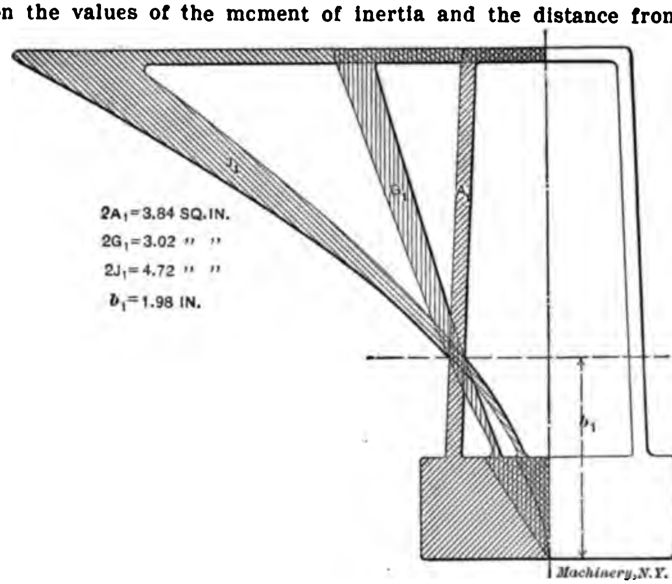


Fig. 21. Section of Punch Frame, showing Method of Determining the Center of Gravity and the Moment of Inertia

bottom of the figure to the gravity axis may be found through the use of equations given in the first part of this chapter, as follows:

$$(e_1)_d = \frac{bG_1}{A_1} = \frac{1.98 \times 3.02}{3.84} = 1.56$$

$$(z_1)_d = 5.01 - (e_1)_d = 5.01 - 1.56 = 3.45$$

$$(I_1)_d = b^2 J_1 = (1.98)^2 \times 4.72 = 18.5$$

$$(I_1)_d = (I_1)'_d - (e_1)^2_d A_1 = 18.5 - (1.56)^2 \times 3.84 = 9.2$$

These values apply only to the figure which was drawn on the drawing board and not to the actual section. It is next necessary to find the size of the actual section. This is accomplished by assuming T_1 as 2.000 pounds per square inch and solving for X in equation (11).

$$T_1 = \frac{P_1[K + X(e_1)_d]X(e_1)_d}{X^4(I_1)_d} + \frac{P_1}{X^2(A_1)_d} \qquad (11)$$

$$2000 = \frac{196{,}850\,[60 + 1.56\,X]\,1.56\,X}{X^4 \times 9.2} + \frac{196{,}850}{X^2 \times 3.84}$$

from which the value $X = 11.45$ is found by trial.

Therefore, to get the actual size of the section at AB, each dimension of the drawing is multiplied by 11.45. This gives the depth of the section as 57.36 inches, and also

$$e_1 = 1.56 \times 11.45 = 17.86 \text{ inches},$$
$$z_1 = 3.45 \times 11.45 = 39.5 \text{ inches},$$

and all other dimensions in the same proportions.

The value of C_1 is next calculated by equation (12) and found to be 3,420 pounds per square inch. It would seem that this is very low

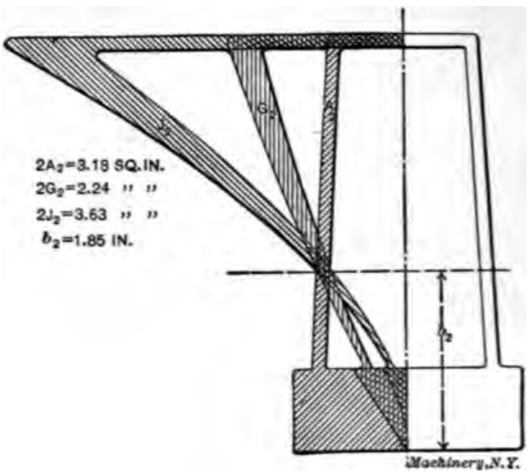


Fig. 22. Section in CD, Fig. 16

for a compressive stress, but the thickness of the outer part of the frame is only about 1⅝ inch, and it does not seem best to make it thinner.

After having determined the size and shape of the section AB, the outline of the punch is sketched in, the general shape being about as shown in Fig. 16, and care is taken to secure a form of pleasing appearance and conforming to general practice. Then drawings are made of sections at CD, EF, GH, and JK, as shown in Figs. 22, 23, and 24, using a scale $X = 12$. The constructions for finding e and z are completed as shown, and the results of the calculations for the sections are tabulated below:

Section	A	G	J	b	e	z	I'	I
CD	3.18	2.24	3.63	1.85	1.304	2.976	12.44	7.03
EF	2.74	1.88	2.58	1.56	1.04	2.51	6.47	3.51
GH, JK	2.26	1.74	2.47	1.27	0.978	2.02	3.98	1.82

The next step in the design is to investigate the stresses in the sections at *CD*, *EF*, *GH*, and *JK*. The calculations are shown below. All calculations are conveniently made with a slide rule.

Section *CD*

This section is taken at an angle $\alpha = 45°$. Then, from equations developed in the discussion of the section *CD* in the earlier part of this chapter, we have:

$$P_2 = P = 196{,}850 \text{ pounds.}$$
$$L_2 = P_2 \cos \alpha = 196{,}850 \times 0.707 = 138{,}700 \text{ pounds.}$$
$$V_2 = P_2 \sin \alpha = 196{,}850 \times 0.707 = 138{,}700 \text{ pounds.}$$
$$l_2 = \text{(by measurement from drawing) } 69.25 \text{ inches.}$$
$$M_2 = P_2 l_2 = 196{,}850 \times 69.25 = 13{,}600{,}000 \text{ inch-pounds.}$$
$$f_2 = \frac{M_2 e_2}{I_2} = \frac{13{,}600{,}000 \times 12 \times 1.304}{12^4 \times 7.08} = 1460 \text{ pounds.}$$

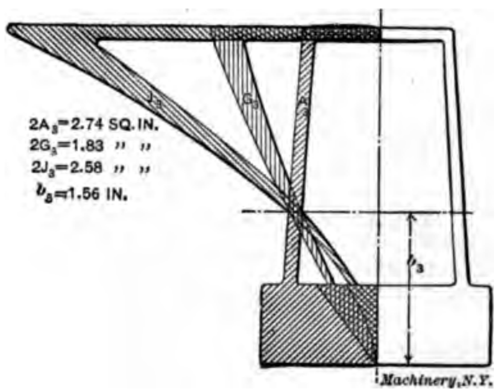


Fig. 23. Section in *EF*, Fig. 16

$$c_2 = \frac{M_2 z_2}{I_2} = \frac{13{,}600{,}000 \times 12 \times 2.976}{12^4 \times 7.08} = 3326 \text{ pounds.}$$
$$t_2 = \frac{L_2}{A_2} = \frac{138{,}700}{12^2 \times 3.18} = 303 \text{ pounds.}$$
$$s_2 = \frac{V_2}{A_2} = \frac{138{,}700}{12^2 \times 3.18} = 303 \text{ pounds.}$$
$$T_2 = \frac{1460 + 303}{2} + \sqrt{303^2 + \left(\frac{1460 + 303}{2}\right)^2} = 1811 \text{ pounds.}$$
$$C_2 = \frac{3326 - 303}{2} + \sqrt{303^2 + \left(\frac{3326 - 303}{2}\right)^2} = 3056 \text{ pounds.}$$
$$S_2 = \sqrt{303^2 + \left(\frac{3326 - 303}{2}\right)^2} = 1544 \text{ pounds.}$$

It will be noted that the tensile stress is somewhat smaller than the maximum value of 2,000 pounds per square inch which was to be allowed. However, the strength of the casting is likely to be least at this point for two reasons; first, because the section is taken at the center of a rather sharp bend for so massive a casting, and second, because of the necessity of introducing a bearing on the upper side near this section, thus requiring the frame to support the reaction from the bearing. For these reasons a lower value for the tension is not out of place.

Section *EF*

If the tabulated values of the properties of the section *EF* are substituted in the equations developed for that section, the following values of the stresses result:

$P_3 = P = 196{,}850$ pounds.

$l_3 = 54$ inches (measured from drawing).

$M_3 = P_3 l_3 = 196{,}850 \times 54 = 10{,}600{,}000$ inch-pounds.

$$f_3 = \frac{M_3 e_3}{I_3} = \frac{10{,}600{,}000 \times 12 \times 1.04}{12^4 \times 3.51} = 1815 \text{ pounds.}$$

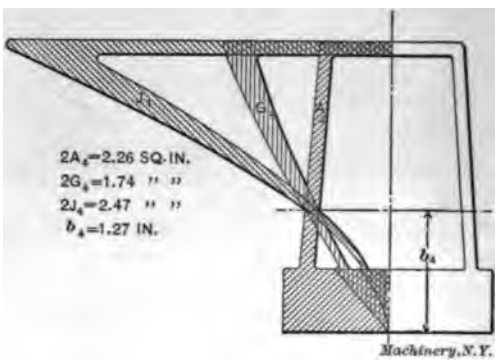


Fig. 24. Section in *GH* and *JK*, Fig. 16

$$c_3 = \frac{M_3 z_3}{I_3} = \frac{10{,}600{,}000 \times 12 \times 2.51}{12^4 \times 3.51} = 4380 \text{ pounds.}$$

$$s_3 = \frac{P_3}{A_3} = \frac{196{,}850}{12^2 \times 2.74} = 498 \text{ pounds.}$$

$$T_3 = \frac{1815}{2} + \sqrt{498^2 + \left(\frac{1815}{2}\right)^2} = 1944 \text{ pounds.}$$

$$C_3 = \frac{4380}{2} + \sqrt{498^2 + \left(\frac{4380}{2}\right)^2} = 4438 \text{ pounds.}$$

$$S_3 = \sqrt{498^2 + \left(\frac{4380}{2}\right)^2} = 2248 \text{ pounds.}$$

These results are all fairly near the requirements, and the section may be said to be satisfactory. The value of S_3 is a trifle higher than was originally specified; this could be reduced by increasing the area of the section slightly. The excess is, however, hardly enough to warrant this.

Sections *GH* and *JK*

Sections *GH* and *JK* are practically alike in the punch under discussion, although they are not so drawn in Fig. 16. The section as shown in Fig. 16 at *GH* is nearer the size to be investigated than that at *JK*. The stresses will result as follows:

$$P_4 = P = 196{,}850 \text{ pounds}$$

$$l_4 = 14.5 \text{ inches (measured from drawing).}$$

$$M_4 = P_4 l_4 = 196{,}850 \times 14.5 = 2{,}850{,}000 \text{ inch-pounds.}$$

$$f_4 = \frac{M_4 e_4}{I_4} = \frac{2{,}850{,}000 \times 12 \times 0.978}{12^4 \times 1.82} = 885 \text{ pounds.}$$

$$c_4 = \frac{M_4 z_4}{I_4} = \frac{2{,}850{,}000 \times 12 \times 2.02}{12^4 \times 1.82} = 1880 \text{ pounds.}$$

$$s_4 = \frac{P_4}{A_4} = \frac{196{,}850}{12^2 \times 2.26} = 608 \text{ pounds.}$$

$$T_4 = \frac{885}{2} + \sqrt{608^2 + \left(\frac{885}{2}\right)^2} = 1189 \text{ pounds.}$$

$$C_4 = \frac{1880}{2} + \sqrt{608^2 + \left(\frac{1880}{2}\right)^2} = 2011 \text{ pounds.}$$

$$S_4 = \sqrt{608^2 + \left(\frac{1880}{2}\right)^2} = 1096 \text{ pounds.}$$

It is to be noted that in the case of these sections all the stresses are decidedly small. This is caused by the large section required at *GH* to provide room for the passage of the main shaft, and also because a smaller section at this point would present a weak appearance as a support for the massive head. In the case of the section at *JK*, however, these reasons do not hold, and the frame in question would have been better proportioned had the section at *JK* been smaller, thus making the stresses more nearly equal over the entire frame. The punch frame illustrated in Fig. 16 is better proportioned in this respect.

The system of design here presented may be applied to frames of many other forms of machines besides punching and shearing machines. It offers the following advantages:

1. It is mathematically correct in principle.
2. The sources of error are comparatively small and are of a kind easily detected.
3. The graphical constructions involved are not laborious, and are sufficiently exact for any ordinary work. The degree of accuracy can be increased at will by employing larger scales.
4 Considerable latitude is allowed for the judgment of the designer.

Physical Characteristics of Cast Iron*

In connection with the previous discussion regarding punch frames, it may be appropriate to call attention to some characteristics of cast iron, as stated by Mr. James Christie in a paper read before the Engineers' Club of Philadelphia. Cast iron is probably the most complex, variable and uncertain form in which iron is used. Not only is the amount of extraneous metals and metalloids variable, but the condition in which the associated carbon exists, and the character of this association, are determined largely by the influence of silicon and possibly other metalloids. Again, the physical properties of the metal are influenced by casting temperature, rate of cooling, etc., so that altogether we can only assume the probable strength and stiffness of a

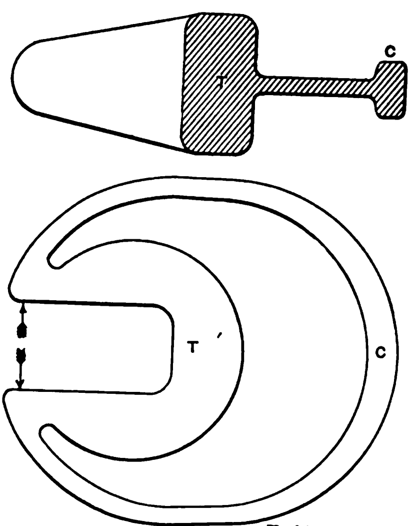


Fig. 25. Illustration of Common Design Intended to Equalize Stresses in Cast Iron

casting in the most general way, and forecast results which will suit an average, from which individual castings may vary widely in extremes.

For heavy machinery, where cast iron is used in heavy masses through which working stresses are imperfectly distributed, the metal probably is much softer and weaker in the middle of the mass, where it has cooled slowly, than at the outer surface, where the metal has more rapidly cooled. Furthermore, castings are usually under considerable internal strain, due to unequal contraction, and although this internal strain gradually disappears, it may have some disturbing influence after the casting has been put in service. It is good practice to assume an ultimate tensile strength of 16,000 pounds per square inch for ordinary iron castings, and to limit working stresses from

* MACHINERY, November, 1907.

2,000 to 4,000 pounds per square inch, according to the conditions and character of the service.

Cast iron offers a high resistance to compressive stress, and although this resistance varies within wide limits, it may be assumed as a working basis to be about six times that of the tensile strength, or, say, 95,000 pounds per square inch of section.

In the middle of the past century, as cast iron became extensively applied to structural purposes, its physical properties were studied with great care, and the experiments of Hodgkinson and Fairbairn in England, and of their contemporaries, yielded a valuable fund of information on the subject. Seeking a section of beam which should exhibit the highest ultimate strength in proportion to area of cross-section, or of the weight of metal employed, Hodgkinson advocated a section in which the tension flange exceeded the compression flange about six to one in sectional area, the web usually tapering in thickness from the tension flange, diminishing toward the other flange. This form of beam was largely adopted, and took precedence as long as cast iron was used for beams in structures. We find that the same method of reasoning influenced the machine designer in disposing cast iron to seeming advantage in the construction of machines, massing the metal to resist tension, and permitting high unit stress on metal in compression; and especially is this observed in machines of the open jaw or gap type, such as presses, punching and shearing machines, etc. It is believed, however, by many good machine designers that the unit stresses should be little, if any, higher in compression than in tension, for the following reasons: In machinery, rigidity or stiffness is usually the chief consideration; many machines do not fulfil the intended purpose properly, not by failure through fracture, but by want of sufficient stiffness. Deflection has to be limited, and when that is done, breaking from excessive tension is sufficiently guarded against. As cast iron yields to compression as much as with the same unit stresses it yields to tension, it follows that the compressive stress should not exceed the tensile strength per unit of section if it is desired to dispose a given mass of metal with least deflection. It is believed that rupture sometimes occurs in a machine apparently through tension, where the origin of the weakness could be traced to a want of material to sufficiently resist compression, the improperly supported tension side severing by cross-bending or transverse stress.

Taking for illustration an open gap machine with frame as shown in Fig. 25, with tension at T and compression at C, if the section is so shaped that the compressive unit stress is six times that of the tensile unit stress, then, elastic moduli being equal, the frame will yield at C six times as much by compression as it does by tension at T. This permits an oscillation of the mass at T around its center. If this oscillation becomes dangerous, by extent or frequency, the frame will break by cross-bending at the mass T, giving the impression that more material is needed to resist tension, whereas the fact may be that more material should be placed at C to prevent excessive yield by compression.

When rapidly alternating stresses occur, it is acknowledged that provision must be made for something more than the greatest stress in one direction alone. There are still differences of opinion and practice on this subject among bridge designers. Some maintain that when the alterations are of slow recurrence, so as to permit actual rest between reversals, no special increase of section is required. Others specify that the sum of the sections required for the stress in opposite direction should be used to suit the conditions. There can be little doubt that the latter estimate is not too great for machinery when the oscillation of the forces occurs with great rapidity, and especially when the metal under consideration is cast iron, with a modulus of elasticity about one-half that of steel or wrought iron. It is a safe general rule for ordinary cast iron in machine construction to limit tensile stress to 4,000 pounds per square inch of section, under the most favorable circumstances; to 3,000 pounds when loads are suddenly applied; and to 2,000 pounds when the force alternates in direction; these unit loads are to be further limited to suit the ratio of length to section, as required for columns or any members in alternate extension or compression, or for beams or members subjected to alternating transverse stresses.

CHAPTER III

BENDING STRESSES IN WIRE ROPE*

The study of different engineering problems brings out the proposition of two kinds of stresses, one seen and the other unseen. Those stresses which are caused by direct thrust or pull are easily seen and provided for, but the unknown forces, or the ones which fail to make an impression on our senses, are more difficult to grasp. Engineers have provided for such forces on some kinds of work by using a larger factor of safety. This method at best, however, only approximates, and should never be used except as a last resort, after all possible means for the determination of the stresses have been exhausted.

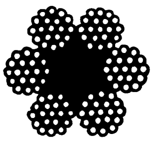
TABLE III. TOTAL AREA IN SQUARE INCHES OF WIRES OF DIFFERENT ROPES

Diameter of Rope.	6 x 7 Construction.	6 x 19 Construction.	6 x 37 Construction.	8 x 19 Construction
2¾	......	2.6892	2.6704	......
2½	......	2.2224	2.2068	......
2¼	......	1.8000	1.7876	......
2	......	1.4216	1.4124	......
1¾	......	1.0888	1.0812	......
1⅝	..	0.9390	0.9324	......
1½	0.8884	0.7997	0.7929	0.6714
1⅜	0.7007	0.6723	0.6676	0.5643
1¼	0.5791	0.5556	0.5517	0.4664
1⅛	0.4691	0.4500	0.4469	0.3778
1	0.3706	0.3554	0.3581	0.2985
⅞	0.2840	0.2722	0.2703	0.2285
¾	0.2184	0.1999	0.1982	0.1678
⅝	0.1448	0.1389	0.1379	0.1166
9/16	0.1178	0.1125	0.1117	0.0945
½	0.0926	0.0889	0.0883	0.0746
7/16	0.0710	0.0681	0.0676	0.0571
⅜	0.0584	0.0500	0.0495	0.0419
5/16	0.0362	0.0347	0.0345	0.0291
¼	0.0231	0.0222	0.0221	0.0186

Wire rope, on account of its numerous applications to various enginering problems, such as derricks, traveling cranes, elevators, ore- and coal-handling machinery, etc., is becoming a vital adjunct to the solution of the problem of the economical handling of many different materials. The draftsman or designer of any apparatus requiring the use of wire rope has to decide upon the size of sheaves and drums which he will employ, and in a great many cases has only a vague idea of what size these should be made. To be sure, the catalogues of wire rope manufacturers give in a general way the smallest diameter

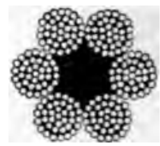
* MACHINERY, June, 1907.

of sheave or drum that should be used for any given size of rope, but it was never intended that sizes of sheaves given as a minimum should be the only sizes to employ. Nevertheless, judged by this standard, a good many pieces of apparatus, of excellent design in other respects, show an almost total disregard for such recommendations, and the sheaves and drums used are of extremely small diameter. This is caused by a desire to make a compact apparatus, and to bring the first cost as low as possible, both of which are desirable features from a



Machinery, N.Y.

TABLE IV. STRENGTH OF 6 x 19 AND 6 x 37 ROPES



Machinery, N.Y.

Diameter in Inches.	Approximate Circumference in Inches.	Weight per Foot in Pounds.	Approximate Breaking Stress in Tons of 2000 Pounds. Iron.	Approximate Breaking Stress in Tons of 2000 Pounds. Crucible Steel.	Approximate Breaking Stress in Tons of 2000 Pounds. Plow Steel.
2 3/4	8 5/8	11.95	114	228	305
2 1/2	7 7/8	9.85	95	190	254
2 1/4	7 1/8	8.00	78	156	208
2	6 1/4	6.30	62	124	165
1 3/4	5 1/2	4.85	48	96	128
1 5/8	5	4.15	42	84	111
1 1/2	4 3/4	3.55	36	72	96
1 3/8	4 1/4	3.00	31	62	82
1 1/4	4	2.45	25	50	67
1 1/8	3 1/2	2.00	21	42	56
1	3	1.58	17	34	44
7/8	2 3/4	1.20	13	26	34
3/4	2 1/4	0.89	9.7	19.4	25
5/8	2	0.62	6.8	13.6	18
9/16	1 3/4	0.50	5.5	11.0	14.5
1/2	1 1/2	0.39	4.4	8.8	11.4
7/16	1 1/4	0.30	3 4	6.8	8.85
3/8	1 1/8	0.22	2.5	5.0	6.55
5/16	1	0.15	1.7	3.4	4.50
1/4	3/4	0.10	1.2	2.4	3.00

business standpoint. The purchaser of such an apparatus, however, is also interested in the cost of maintenance as well as in the first cost. He finds after a short period that his ropes have all gone to pieces, and procures another rope only to get the same result. In the design of his machine too small sheaves were used, and he finds it necessary to replace, at considerable expense, the small sheaves by larger ones, or to continue to have a high cost of maintenance. An example of this will perhaps make this point clear. A 1¼-inch crucible steel rope 6 × 19 runs over a 3-foot sheave and drum on a machine designed to lift 10 tons. According to the tables furnished by rope manufacturers, this rope has a strength of 50 tons, which would give a factor of safety of 5. We will assume that the factor 5 is as small as it is advisable to use for this particular service. What we actually have, however, is this:

Stress on rope due to load to be lifted .... = 10.00 tons
Stress on rope due to bending over 3-foot sheave .... = 7.25 tons

Total stress .... = 17.25 tons

This gives a factor of safety of less than 3, which is altogether too small, showing how necessary it is to take into account the bending

TABLE V. STRENGTH OF 8 x 19 ROPES

Machinery, N. Y.

Diameter in Inches.	Approximate Circumference in Inches.	Weight per Foot in Pounds.	Approximate Breaking Stress in Tons of 2000 Pounds. Crucible Steel.	Approximate Breaking Stress in Tons of 2000 Pounds. Plow Steel.
1 1/2	4 3/4	3.48	65	86
1 3/8	4 1/4	2.51	56	74
1 1/4	4	2.18	45	60
1 1/8	3 1/2	1.82	38	50
1	3	1.32	27	35
7/8	2 3/4	1.05	21	27
3/4	2 1/4	0.89	16.5	21
5/8	2	0.58	11.6	15
9/16	1 3/4	0.43	9.4	12.3
1/2	1 1/2	0.31	6.6	8.55
7/16	1 1/4	0.27	6.1	7.95
3/8	1 1/8	0.18	3.8	4.25
5/16	1	0.12	2.2	2.93
1/4	3/4	0.066	1.6	1.95

stress due to winding around a pulley or drum. It would be necessary to use in this case either a larger rope and larger sheaves and drums, or a stronger rope and sheaves and drums enough larger to reduce the bending stress to a smaller amount. A 1¼-inch plow steel rope, 6 × 19, has a strength of 67 tons, and with a factor of safety of 5 gives a working stress of 13.4 tons. The problem could then be solved as follows:

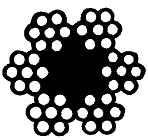
Stress on rope due to load to be lifted .... = 10.00 tons
Stress on rope due to bending over 6-foot 6-inch sheave ... = 3.36 tons

Total stress .... = 13.36 tons

Note that the sheave had to be increased to more than double its original size to reduce the stress sufficiently. The bending stresses given in the two cases just outlined are the actual stresses that any rope of the construction noted would have. This comparison shows how vital it is that the bending stresses be carefully considered.

For the purpose of determining what the stress is, curves have been plotted for various kinds of commercial ropes of standard makes and various constructions. These curves, given in Figs. 26 to 33, show graphically the effect of the different sizes of sheaves on different ropes, so that it is posible for anyone to look at the curves and tell at a glance exactly what bending stress is put on the rope. For example, take the curves for 6 × 19 ropes, Figs. 28 and 29. There are

TABLE VI. STRENGTH OF 6x7 ROPES



Machinery, N.Y.

Diameter in Inches.	Approximate Circumference in Inches.	Weight per Foot in Pounds.	Approximate Breaking Stress in Tons of 2000 Pounds. Iron.	Approximate Breaking Stress in Tons of 2000 Pounds. Crucible Steel.	Approximate Breaking Stress in Tons of 2000 Pounds. Plow Steel.
$1\frac{1}{2}$	$4\frac{3}{4}$	3.55	34	68	91
$1\frac{3}{8}$	$4\frac{1}{4}$	3.00	29	58	78
$1\frac{1}{4}$	4	2.45	24	48	64
$1\frac{1}{8}$	$3\frac{1}{2}$	2.00	20	40	53
1	3	1.58	16	32	42
$\frac{7}{8}$	$2\frac{3}{4}$	1.20	12	24	32
$\frac{3}{4}$	$2\frac{1}{4}$	0.89	9.3	18.6	24
$\frac{11}{16}$	$2\frac{1}{8}$	0.75	7.9	15.8	21
$\frac{5}{8}$	2	0.62	6.6	13.2	17
$\frac{9}{16}$	$1\frac{3}{4}$	0.50	5.3	10.6	14
$\frac{1}{2}$	$1\frac{1}{2}$	0.39	4.2	8.4	11
$\frac{7}{16}$	$1\frac{1}{4}$	0.30	3.3	6.6	8.55
$\frac{3}{8}$	$1\frac{1}{8}$	0 22	2.4	4.8	6.35
$\frac{5}{16}$	1	0 15	1.7	3.4	4.35
$\frac{9}{32}$	$\frac{7}{8}$	0.125	1.4	2.8	3.65

two sets of curves for this kind of rope, one plotted to show the relation between the diameter of rope and the bending stress (Fig. 29), and the other between the diameter of rope and the size of the sheave (Fig. 28). With any two factors known, the third can easily be obtained. Suppose that we desire to use a 1-inch diameter, 6 × 19 rope, of a strength of 34 tons to hoist a load of 4 tons. What is the minimum size sheave that can be used with a factor of safety of 5?

34 tons divided by 5............= 6.8 tons total permissible stress
Direct load.......................= 4.0 tons

Difference................= 2.8 tons permissible bending stress

Using the diagram, Fig. 29, we find that the line representing 2.8

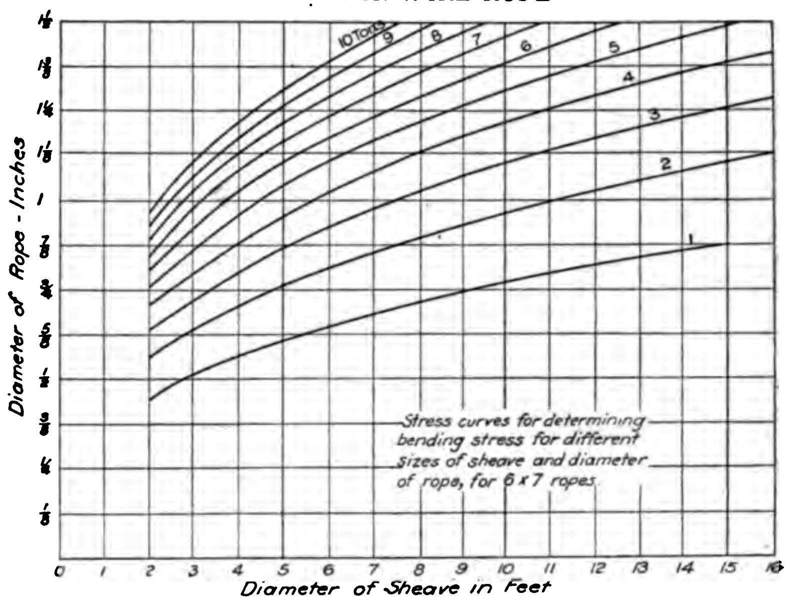


Fig. 26

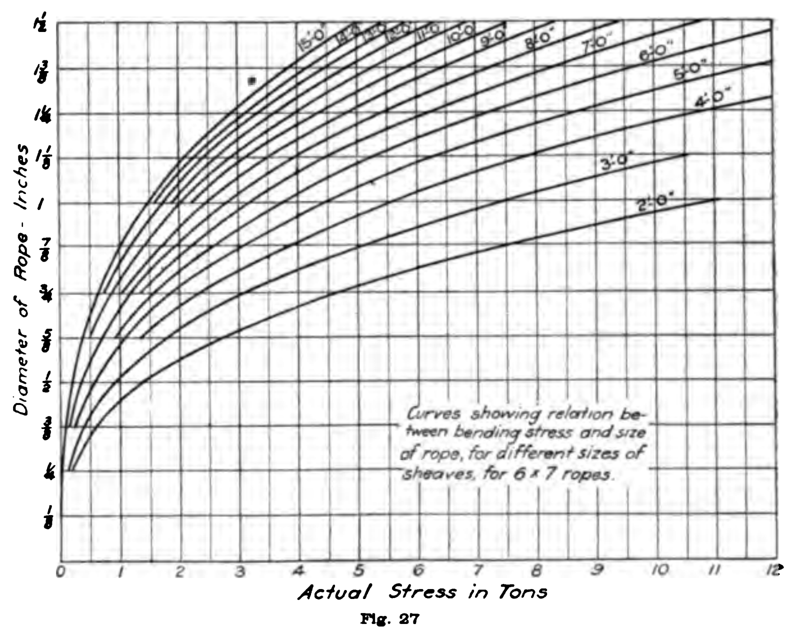


Fig. 27

tons stress intersects the line for 1-inch rope on the curve marked 4 feet. Consequently we will require a sheave 4 feet in diameter.

We can reverse the question. A 6 × 19 rope, 1 inch in diameter, is

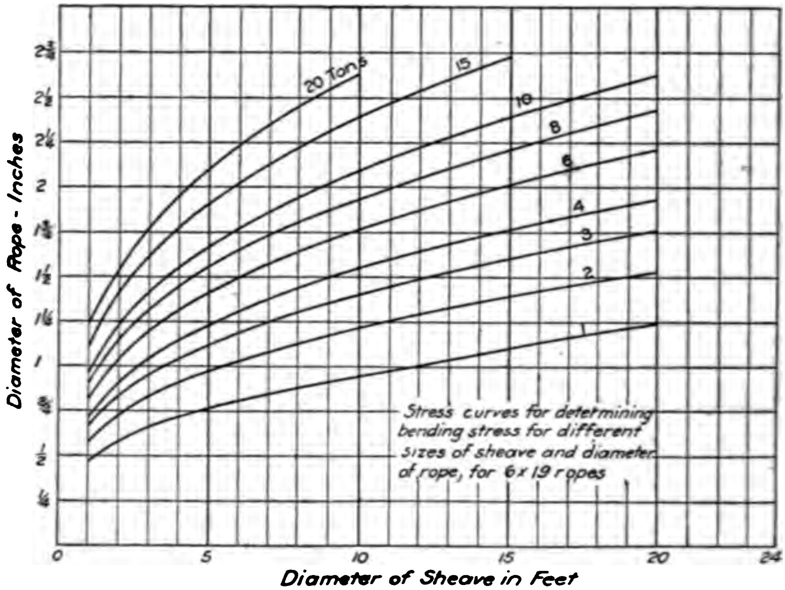


Fig. 28

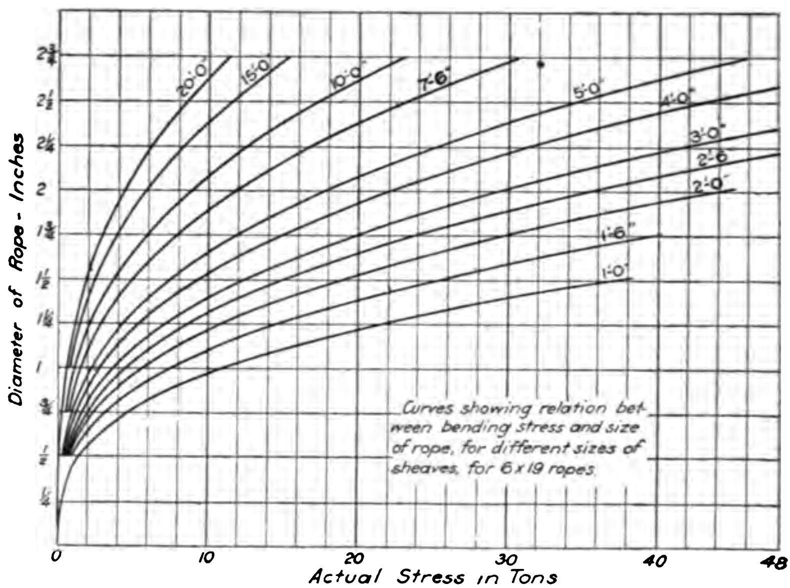


Fig. 29

to be run over a 3-foot sheave; what load will it lift, assuming a factor of safety of 5 and a strength of rope given as 34 tons? Referring to the curves in Fig. 28, we find by following the line for 1-inch

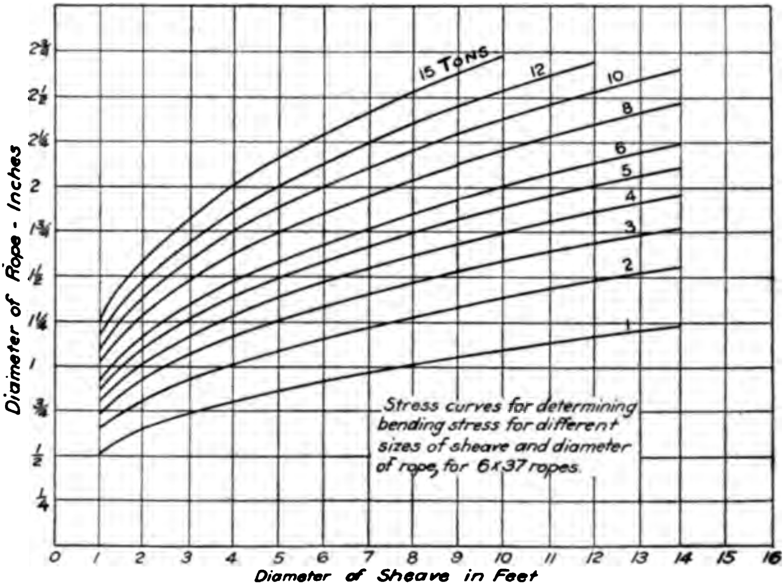


Fig. 30

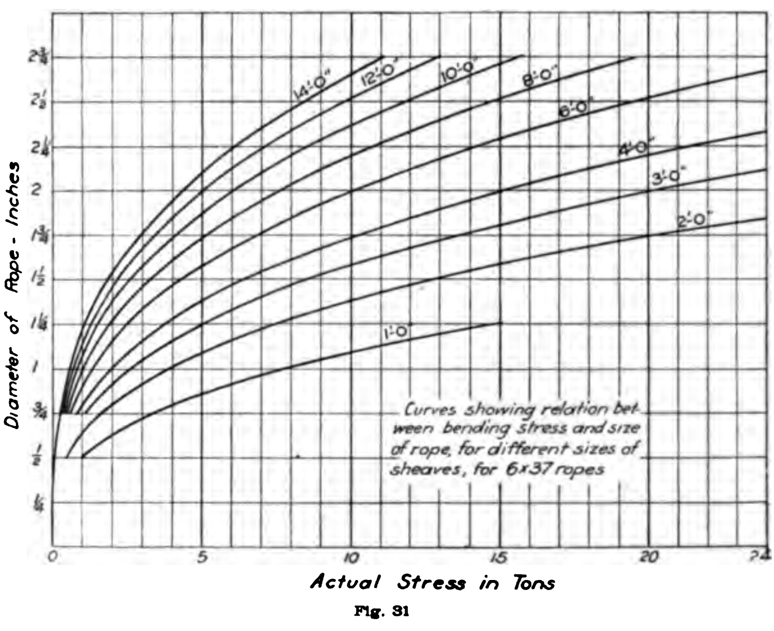


Fig. 31

rope until it intersects the line for a 3-foot sheave that the bending stress lies between the curves for 3 tons and 4 tons. By proportioning we find that it is about 3.7 tons.

We then have:

34 tons divided by 5............= 6.8 tons total permissible stress..
Deducting...................... 3.7 tons bending stress .

Difference...............= 3.1 tons allowable working load.

Calculations to Obtain Curves

The method of calculating these curves is as follows:

Let S = stress per square inch,
E = Young's modulus of elasticity of steel = 29,000,000,
E_r = modulus of elasticity of the rope as a whole,

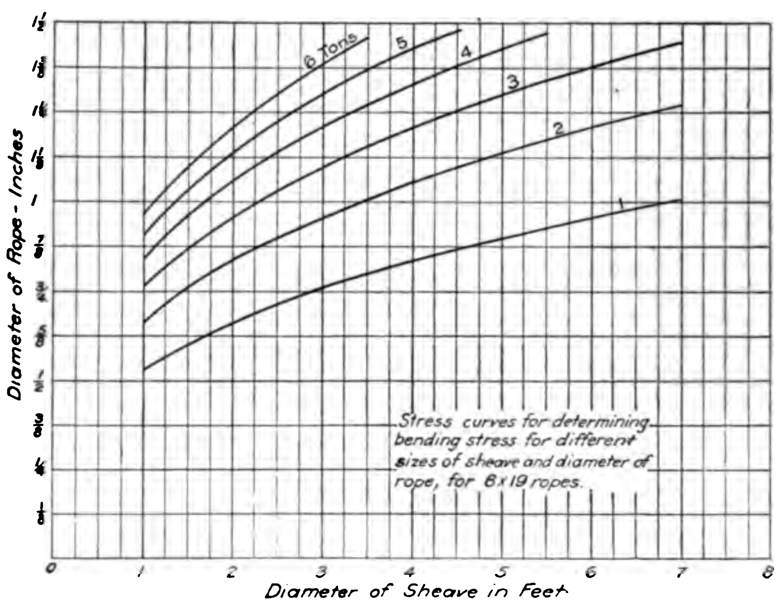


Fig. 32

d = diameter of wire of rope in inches,
d_r = diameter of rope in inches,
D = diameter of sheave or drum in inches.

E_r is a function of E, that is, it is dependent upon its value. E_r is less than E on account of the structure of a rope, and varies with the flexibility of same. A low modulus is indicative of a flexible rope and *vice versa*. In considering this problem, we will take into account the modulus of elasticity E_r of the rope as a whole. This conception is different from the ordinary conception of the modulus of elasticity, but is perfectly feasible, as the modulus of elasticity is, strictly speaking, 'he ratio between the force applied to any material per square inch id the amount of elongation expressed as a fraction of the original ngth.

The stress produced in a round bar of the diameter d_r, when bent 'und a sheave of diameter D, is given by the well-known formula:

$$S = E \frac{d_1}{D} \tag{28}$$

If for d_1 we substitute d, the diameter of the wire in rope, and for E the modulus of elasticity of the rope E_r, we have:

$$S = E_r \frac{d}{D} \tag{29}$$

From this formula we can get the required stress per square inch,

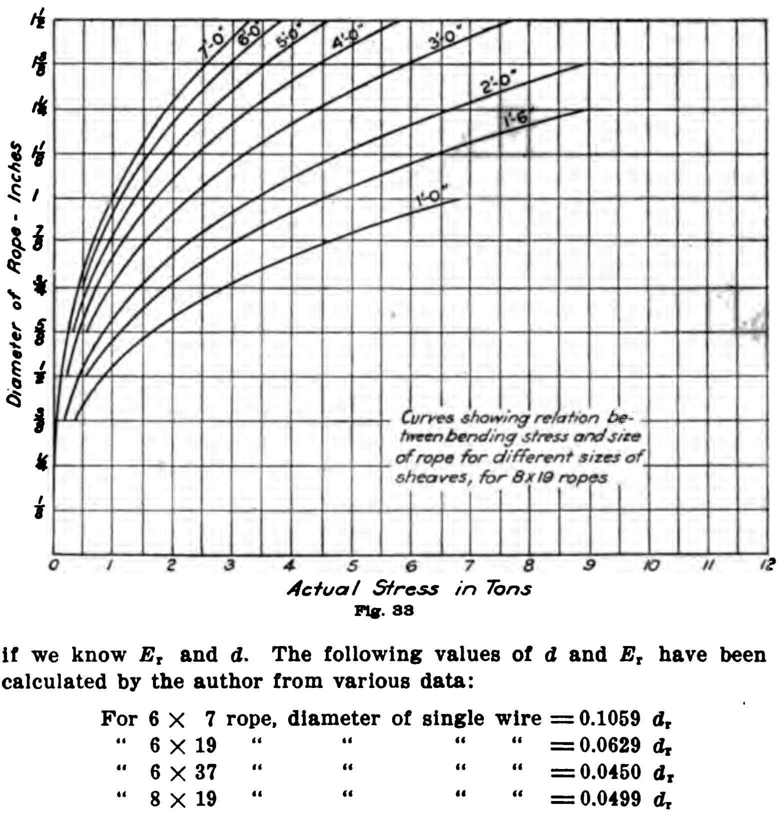


Fig. 33

if we know E_r and d. The following values of d and E_r have been calculated by the author from various data:

For 6 × 7 rope, diameter of single wire = 0.1059 d_r
" 6 × 19 " " " " = 0.0629 d_r
" 6 × 37 " " " " = 0.0450 d_r
" 8 × 19 " " " " = 0.0499 d_r

Allowance has been madé in these values for the angle of twist of the strands and wires, which decreases the sizes of the wire somewhat in any given rope.

The values of E_r which the author has obtained by theoretical calculation are:

For 6 × 7 rope E_r = 13,700,000
" 6 × 19 " E_r = 12,000,000
" 6 × 37 " E_r = 11,300,000
" 8 × 19 " E_r = 11,000,000

CHAPTER IV

DESIGN OF BILLET AND BAR PASSES*

In practice, engineers and designers usually make use of methods for arriving at conclusions which are particularly adapted to their line of work. These methods may be original with themselves, or they may be the methods of others modified to suit the requirements of their own particular practice. In the original design of work it is frequently very desirable that approximate results first be obtained in order that an idea may be formed of the best way in which to produce the desired results. In different lines, the work and plans of the engineer, however broad and thorough he may be, are very frequently subject to changes by the men who are familiar with the practical features and peculiarities, and who are directly concerned in the repair and operation of the plant in process of development. During the discussions or conferences between the different men interested, short methods are especially desirable for quickly arriving at approximate results, and in the hope that it will prove of value to those who are interested in the design of billet and bar rolls, the author will give a description of a method by means of which a series of such passes may be quickly proportioned.

Determining the Number of Passes

In the designing of roll passes in general, it is necessary that the size and shape of the finished product be known. The product being decided upon, the number and proportions of the passes, and the dimensions of the bloom or slab to be used must be fixed according to circumstances. The first feature to demand attention will quite likely be the number of passes the material will have to make through the rolls before being brought to the dimensions required. If the rolls are to be designed for a mill already existing, this will depend upon the strength of the mill, the number of passes available in that mill, etc. On these limiting circumstances, and also on the available supply, will depend, in a large measure, the size of bloom or billet which can be used. When designing rolls for a new layout the conditions mentioned above do not apply, as the arrangement of the new mill will depend very largely on the product desired.

It is desirable first that the number of reductions or passes be at least approximated, depending, of course, on the shape of the section to be rolled, etc. By reduction is meant the difference between the area of cross-section of the billet before passing through the rolls and after passing through, divided by the original area. For example, if a 4×4 billet becomes $2\frac{5}{8} \times 4\frac{1}{4}$ after passing through the rolls, the reduction would be found thus:

* MACHINERY, May, 1906, and May, 1907.

$$(4 \times 4) - (2\tfrac{5}{8} \times 4\tfrac{1}{4}) = 16 - 11.15625 = 4.84375.$$

$$\frac{4.84375}{16} = 0.3027, \text{ or } 30.27 \text{ per cent.}$$

Reductions vary from 60 per cent downward, depending on strength of mill, engines, etc. A convenient formula for quickly obtaining the average reduction or number of passes is

$$P = 100\left(1 - \sqrt[n]{\frac{a}{A}}\right)$$

where P = average per cent reduction,
n = number of passes,
a = finished area of billet,
A = original area of billet.

For example, we will assume that a 4 × 4 billet is to be reduced to a $4\frac{1}{4} \times \frac{1}{4}$ strip in five passes. The formula will then appear as follows:

$$P = 100\left(1 - \sqrt[5]{\frac{1.0625}{16}}\right) = 100\ (1 - \sqrt[5]{0.0664})$$

$$= 100\ (1 - 0.5813) = 100 \times 0.4187 = 41.87 \text{ per cent.}$$

Of course, it is understood that the reductions would not probably be made exactly to the figures given, for in practice they would have to be changed somewhat, but the result given is the average reduction, and from this one is enabled to form a good idea of the work to be accomplished.

Drawing the Reduction Diagram

Generally speaking, the first pass is a "shaping" pass or "leader," and usually does not have much draft, depending on the shape to be produced. Having determined, as previously described, the area of this first pass, and the total number of passes, or the average reduction, we may proceed to draw the diagram shown in Fig. 34, from which the series of passes may be proportioned. Draw two lines meeting at A and making an angle of 95.5 degrees with each other; bisect this angle with line AX. To locate line I, for a square bar, use the following formula:

$$\sqrt{\frac{A}{2.2}} = z$$

in which A = area of pass,
2.2 = constant for the angle of 95.5 degrees only,
z = vertical dimension of pass on line AX (Fig. 34).

If the angle of 92 degrees is used, take 2.071 as the constant.

Assuming, for example, the area of the first pass to be 26.95 square inches, and substituting this value in the formula, we have:

$$\sqrt{\frac{26.95}{2.2}} = z = 3.5 \text{ inches.}$$

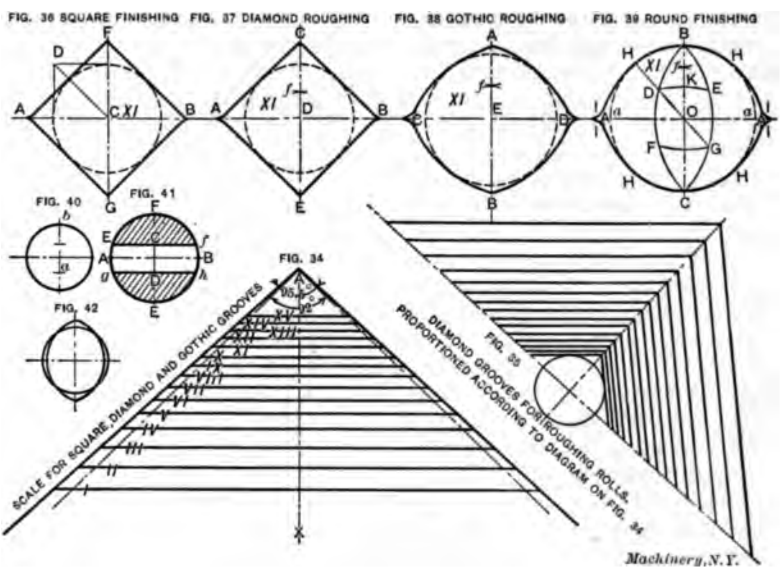
Measuring off this distance from point A on the line AX (Fig. 34), locates a point through which draw a horizontal line (line I in this case). The line thus obtained will be the horizontal dimension of the first pass.

Having found out by the preceding methods the horizontal line for the first pass, the corresponding line for the second pass may be found by the following formula, which is based on the principle that the areas of similar figures vary as the squares of their corresponding dimensions:

$$z^2\ (1 - a) = y^2$$

in which z = vertical dimension on line AX (Fig. 34),
y = vertical dimension of succeeding pass,
a = required reduction.

For the purpose of illustration, we will assume a reduction of 20.28



Figs. 34 to 42. Design of Billet and Bar Passes

per cent, and taking the dimensions of pass No. 1, as found above, the formula would appear thus:

$$\sqrt{3.5^2\ (1 - 0.2028)} = 3.125, \text{ or } 3\tfrac{1}{8} \text{ inches.}$$

This distance laid off on line AX (Fig. 34) will locate a point through which line II will pass. In a similar way line III may be laid off from line II, and so on, until the required number of passes have been completed.

Directions for Laying out Billet Grooves

On the scale (Fig. 34) for 95.5 degrees, the horizontal lines numbered I to XV represent graphically the drafts of a corresponding number of grooves for Gothic and Diamond roughing rolls laid out as previously described, and relate to an angle of 95.5 degrees only. Dotted lines for 92 degrees are drawn in, the use of which will be described later.

Diamond Grooves: Through line AB, Fig. 37, draw perpendicular line CDE. Refer to scale (Fig. 34, angle of 95.5 degrees) and take the vertical distance from horizontal line (XI in this case) to A, along line AX, and lay off the same from D to C and E respectively. Lay off the distance DA and DB, each equal to one-half line XI on scale (Fig. 34). Draw lines AC, CB, BE, and AE and the Diamond is formed. The tailoring curves at A and B complete the figure. These are described with a radius Df equal to ¼ the diameter of the inscribed circle.

Square Finishing Grooves: Through horizontal line AB, Fig. 36, draw perpendicular line FG. From point of intersection at C describe a circle with a diameter equal to side of square required. Draw two lines tangent to circle parallel to AB and FG, respectively, and intersecting at D, and from C mark off CF and CG a distance equal to CD. From point A on scale (Fig. 34, angle of 92 degrees) mark off distance CD on perpendicular line AX, and draw through the point thus located a horizontal line intersecting the sides of the 92-degree angle. (In this case the line so located corresponds to line XI.) The length of this line is the width of the groove on line AB. The tailoring curves at corners A and B are drawn with a radius of 1/10 the length of the side of the square.

Gothic Grooves: The depth AB and width CD, Fig. 38, are laid off by referring to the scale (Fig. 34, angle of 95.5 degrees, line XI in this case). From the intersection E of the horizontal line CD with the perpendicular AB, lay off the distances EC and ED each equal to one half line XI on the horizontal line, and on the perpendicular line lay off the distance EA and EB each equal to the distance from A to line XI. With a radius equal to AB draw curves AD, DB, BC, and AC. By describing the tailoring curves at C and D with a radius equal to ¼ the diameter of the inscribed circle, the figure is completed.

Round Grooves: Draw horizontal line AA, Fig. 39, and perpendicular to it draw BC. From the point of intersection describe circle of required size. From centers aa draw arcs BDC and BGC; with same radius from points B and C draw intersecting arcs DE and FG. From the points of intersection, with radius GH, describe the opening curves HI. The tailoring curves at II are drawn with a radius Bf equal to one half the distance BK.

Oval Grooves: Fig. 40 is the round for which the oval is intended. Draw the round and divide the diameter into three equal parts, and with a radius ab equal to two of these parts draw a circle, Fig. 41. The vertical diameter is divided into three equal parts. Through C and D draw lines parallel to the horizontal diameter AB. The segments $CEFf$ and $gDhE$ when put together form the oval required, as shown in Fig. 42.

A few general remarks regarding some of the conditions which must be taken into account when designing bar rolls, passes, guides, etc., may here be in place. Special reference is made to continuous mills.

In rolling mills where the piece is passing through from two to ten or more stands of rolls at one and the same time, difficulties are encountered that are not present where the piece is in only one stand at a time. In the latter case the piece is free to elongate, and the rolls can be driven at varying speeds without reference to any of the succeeding passes, while, in the first case, the diameters and speeds of the rolls and the reductions must be correctly proportioned and adjusted to each other.

As the increase in length of a billet is directly in proportion to its decrease in area, this elongation must be taken care of in various ways. If the piece passes from one stand of rolls into a second stand whose peripheral speed is too great, there must be a slip of the bar between the rolls, or there is a likelihood of its parting if the section be small. Whereas, if the peripheral speed of the second stand be too slow in comparison with the speed of the first, it will cause a bow or loop of ever-increasing length, between the stands. This would cause trouble and could not be allowed. The different stands of rolls on continuous mills, especially on the roughing stands, where the speeds are the slowest, and the reductions the greatest, and consequently the heaviest strains are present, are usually driven by means of gearing from a main driving shaft, while on the finishing stands, where the speeds are considerably higher, the section rolled being small, and the strains comparatively light, belts are frequently used. After ascertaining the sections to be rolled down and the dimensions of the finished product, the number of passes may be fixed and the same roughly designed. These will then be the basis upon which the proportions of the driving gearing can be designed.

If the exact speeds required cannot be readily obtained by means of the gearing alone, the small discrepancies can be overcome by increasing or diminishing the diameters of the rolls, by altering the reductions or proportions of the passes, or by a combination of both methods. It is apparent that in a mill where the piece is passing through a number of passes at one time, care must be taken in the proportioning of the different parts, so that each pass, diameter of roll, reduction in gearing, etc., will bear the proper relation to the whole.

"Fins" are frequently the cause of considerable trouble and annoyance, and not infrequently cause the rejection of the finished product. Where a "fin" is formed, it is rolled back into the bar at the next pass. If this takes place on a continuous mill, unless the fin is too pronounced, it is not likely to cause much trouble, as the distance between the stands is comparatively short, and the bar so well covered by the guides that it has no chance to cool. It is, therefore, rolled back into the bar without detriment to the finished product. However, on a mill where the bar is rolled backwards and forwards, the last end of the bar out of the rolls being the first into the succeeding pass, the fin has an opportunity to cool sufficiently to prevent welding into the bar properly, and consequently produces a fine crack in the finished bar and causes its rejection.

The bar in traversing the short distance between the stands on continuous mills, passes through a guide. This guide is usually of cast iron, flared or chamfered at the entering end, so as to more surely catch the outcoming bar. The opposite, or "delivery" end, closely approximates the shape and dimensions of the bar, and is set as closely as necessary to the rolls and directly in line with the next pass. As before stated, the distance between stands being short, the bar is forced into each succeeding pass, should it for any reason not "bite" at once. This is especially true in the case of the heavier sections. Again, it would not be practicable to produce bars by rolling down only one way; *i. e.*, the bar must be turned frequently through an angle of 90 degrees. In mills where the bar is entered into the next pass by hand, no particular attention need be paid to the turning of the section, but in mills where the turning is done mechanically, it is effected by guides. Generally speaking, the turning is accomplished without the necessity of moving parts, by means of what is known as "quarter turn" or "twist" guides. These are used whenever the bar must be turned before entering the next pass, and differ from the ones previously described, in that they are given a twist, so the bar on being forced through will be delivered in a turned position.

In order to facilitate the removal of "cobbles," as bars are called that have become twisted or entangled in the rolls, guides, etc., these parts are made so as to permit of ready removal. The guides are made in halves, held together and in place by means of key-bolts. The roll-housings are also designed with the same object in view, so that in case of a "mess," the guides can almost instantly be taken out and the rolls raised or taken out without much loss of time. The points mentioned are of prime importance and must not be overlooked by the designer.

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NUMBER 25

DEEP HOLE DRILLING

SECOND EDITION

CONTENTS

INTRODUCTION

The difficulties to be overcome in producing deep drilled holes can be classified in three groups. In the first place the drill has a great tendency to "run out," thus producing a hole that is neither straight, nor uniform in diameter; in the second place great difficulties are encountered in trying to remove the chips in a satisfactory manner, and in the third place the heating of the cutting tool is difficult to prevent.

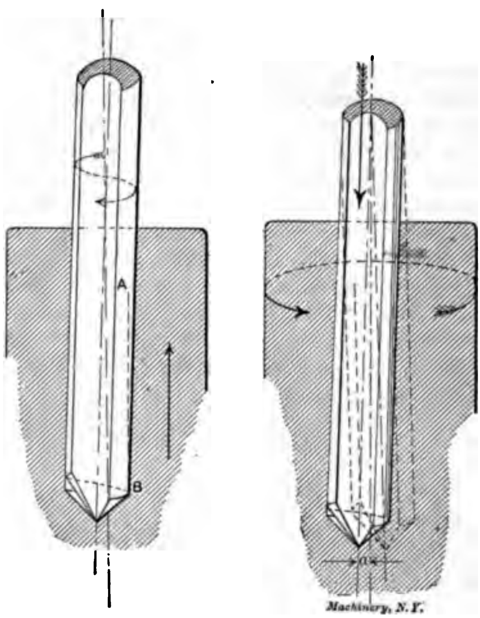

Fig. 1. Comparison between Action of Cutting Tool when Drill and when Work revolves

The principle involved in common drill presses where the drill is given a rotary motion simultaneously with the forward motion for feeding is the one least adapted to produce a straight and true hole. Better results are obtained by giving only a rotary motion to the drill, and feeding the work toward it. It has been found, however, that for drilling deep holes the reversal of this, that is, imparting a rotary motion to the work, and the feed motion to the drill will answer the purpose still better. It seems as if there could be no material difference between the two latter methods. An analysis of the conditions involved will show, however, that there is a decided difference in the action of the drill. If the drill rotates, and the work is fed forward as shown to the left in Fig. 1, the drill, when deviating from its true course, will be caused to increase its deviation still more, by the wedge action of the part *B*, which tends to move in the direction *BA* when the work is fed forward. In the case of the work rotating and the drill being fed forward, as shown to the right in Fig. 1, the point of the drill when not running true will be carried around by the work in a circle with the radius *a*, thus tending to bend the drill in various directions. The drill is by this action forced back into the course of "least resistance," as it is evident that the bending action, being exerted on the drill in all directions, will tend to carry the point back to the axis of the work where no bending action will appear. The chips, as is well known, are carried off by forcing a fluid into the hole, which upon its return carries with it the chips. This fluid being oil will serve the double purpose of carrying away the chips and lubricating the cutting tool, keeping it at a normal temperature.

In the following chapters we shall deal with the practice of deep hole drilling as met with in a number of prominent American shops, presenting at the same time a collection of useful data covering different classes of work. The relation between ordinary drilling and deep hole drilling, dealing with first and fundamental principles, is treated in Chapter I, followed by a detailed account of the practice of deep hole drilling at the Pratt & Whitney Works, Hartford, Conn. In Chapter II the boring of large guns, according to the practice employed at the Watervliet Arsenal, is described. Chapter III is devoted to illustrating and describing various constructions of deep hole drills of merit, together with hints regarding their making, thereby completing the treatise.

CHAPTER I

PRINCIPLES OF DEEP HOLE DRILLING*

The process of drilling deep holes in metal is a familiar one in many shops, particularly where firearms are manufactured, or heavy ordnance is constructed. Since the adoption of hollow spindles for lathes and other machine tools, the methods for machining the bores of guns have been employed in machine tool shops for drilling these spindles; and through this and other means the principles of the operation have become better understood. It is not an easy matter, however, even with the best appliances, to drill or bore a deep hole smooth and round, of exactly the required diameter from end to end, and perfectly straight. While many mechanics are familiar in a general way with the methods and tools for doing this work, specific information upon the subject will be appreciated by those who have not had actual experience in deep hole drilling.

* MACHINERY, December, 1901.

It is well known that a long, or deep, hole—that is, one long in proportion to its diameter—is best roughed out and finished by using a tool on the end of a long bar which enters the work from one end. This is true, whether drilling into solid metal, or boring and reaming a hole that has already been drilled or bored out. A boring bar which extends through the piece, and on which is either a stationary or a traveling head, is not satisfactory for very long work, owing to the spring and deflection of the bar, which is made worse by the fact that the bar must be enough smaller than the bore to allow room for the cutter head. While a long hole may sometimes be finished satisfactorily by means of such a boring bar, by packing the cutter head with wooden blocks which just fill the part of the bore that has been machined, and so support the bar, the method is fundamentally incorrect for long work.

The best methods for machining deep holes are nothing more nor less than an adaptation of what has been found successful in ordinary

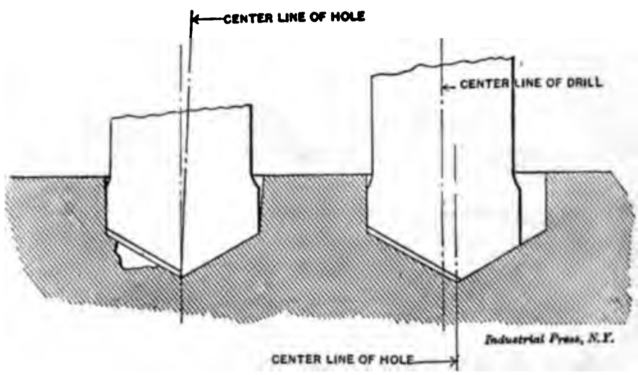

Fig. 2 Fig. 3

drilling and boring in the engine lathe or chucking machine. We will therefore first discuss certain types of chucking tools and drills, and show their relationship to tools that may be used for deep hole drilling.

The Flat Drill

To start with first principles, consider the ordinary flat drill. It is useful for rough work or in drilling hard metals, because it can be easily made and tempered; but it has too much of an inclination for drilling holes that are neither round nor straight, and whose diameter seems to bear no relation to the diameter of the drill. When a flat drill runs into a blow hole or strikes a hard spot, it is deflected, as in Fig. 2, the only resistance to this deflection being the narrow edges of the drill. Under such conditions the hole will be out of round, and crooked. Add to this natural tendency of a flat drill to run out the fact that such drills are often carelessly made, and one understands why they have a reputation for poor work. Thus, if the point is not in line with the axis of the drill, and if the lips are of

The general principle, however, of first starting with a true hole, and then having the drill body designed to follow in its path and so guide the cutting edges, is the fundamental principle of deep hole drilling.

Fig. 6 shows how a flat drill may be adapted for deep hole drilling. The drill from which the illustration was made was employed for drilling a four-inch hole through steel rolls seven feet long. Instead of depending upon the narrow edges of the drill proper to guide and support the cutting edges, the cutting edges are formed on a blade inserted in a cylindrical cast-iron head, the outside diameter of which is turned to a sliding fit in the hole that is being bored. The cutting edges are grooved to break up the chips, enabling the latter to pass out through the passage *E*, on each side of the head. The grooves

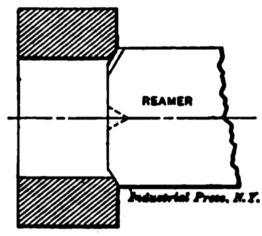

Fig. 5. Flat Reamer

are laid out so that those in one blade come opposite to the lands in the other blade. In the illustration, *A* is the cutter, *B* one of the screws holding the cutter to the head *C*, and the head is attached to the bar by the shank *D*.

The Twist Drill

The modern twist drill accomplishes all that is attained by the arrangement in Fig. 6, and in addition can be ground without seriously affecting the rake, and will free itself from chips more readily, owing to its spiral flutes. The lands of a twist drill present a large cylindrical surface to bear against the sides of the hole and take the side thrust. If the drill is also guided by a hardened bushing, at the point where it enters the metal, as in the case of jig work, the drill will have very little chance to deflect, and the hole will be accurately located and will be quite true and straight.

The twist drill in a modified form is also employed for deep hole drilling. The hollow drill introduced by the Morse Twist Drill Co., New Bedford, Mass., is adapted for this purpose, and in Fig. 7 is shown the arrangement recommended by this company for feeding this drill into the work. The drill has a hole lengthwise through the shank, connecting with the grooves in the drill, as indicated. The shank can be threaded and fitted to a metal tube which acts as a boring bar and through which the chips and oil may pass from the point of the drill. Oil is conveyed to the point on the outside of the tube, as shown in Fig. 7.

In using the hollow drill, the hole is first started by means of a short drill of the size of the hole desired, and drilled to a depth equal to the length of the hollow drill to be employed. The body of the hollow drill acts as a stuffing, compelling the oil to follow the oil grooves provided, and the chips to flow out through the flutes and the hollow shank. The methods of supporting and driving the work, and of feeding the drill, are clearly shown in Fig. 7. Drills of this type are regularly manufactured in sizes up to three inches in diameter, and it is stated that the best results are obtained, when drilling tool steel, by revolving the drill at a cutting speed of 20 feet per minute, with a feed of 0.0025 inch per revolution, while machine steel will admit of a cutting speed of 40 feet per minute and a feed of 0.0035 inch per revolution.

Number of Cutting Edges Desirable

When drilling a hole out of solid stock, some type of drill having two lips or cutting edges is usually the most feasible, and probably nothing will be devised that on the whole surpasses the twist drill

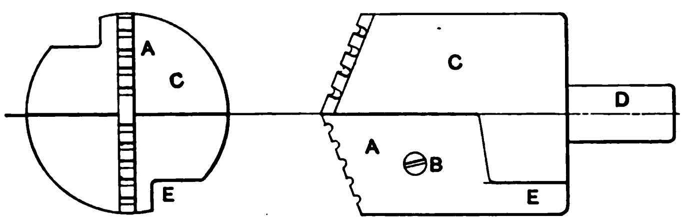

Fig. 6. Flat Drill with Inserted Blade for Deep Drilling

for such work. As is well known, the ordinary twist drill is always provided with two flutes, but twist drills having three or more flutes have been devised, made and tried. The advantages gained by adding to the number of cutting edges have, however, not been great enough to justify the increased cost of manufacture. When added to this comes the weakness caused by the increased number of grooves, and the complicated operation of correctly grinding such drills, it is clear why drills having two flutes only have been and should be adopted.

An end mill, like that in Fig. 11, can be used for drilling, if it has a "center cut," and it will presently be explained how a tool with a single cutting edge may be advantageously employed, particularly for deep hole drilling. The familiar D-drill is of this type, and also its modification as used by the Pratt & Whitney Co. in drilling gun barrels.

When it comes to truing up or enlarging a hole previously drilled or bored, the two-lip drill is not suitable in any of its forms. For boring a true hole nothing can surpass a single-pointed boring tool, the ideal condition for finishing a hole being when the cutting point is a real diamond, or a rotating wheel of abrasive material.

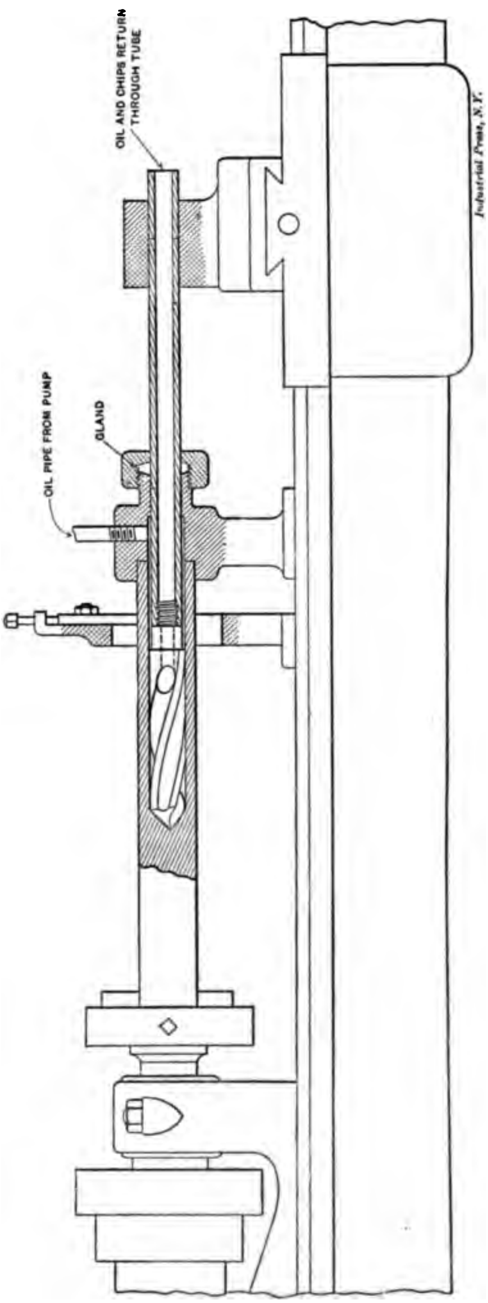

Fig. 7. Arrangement of Machine when Drilling Deep Holes with Hollow Twist Drill

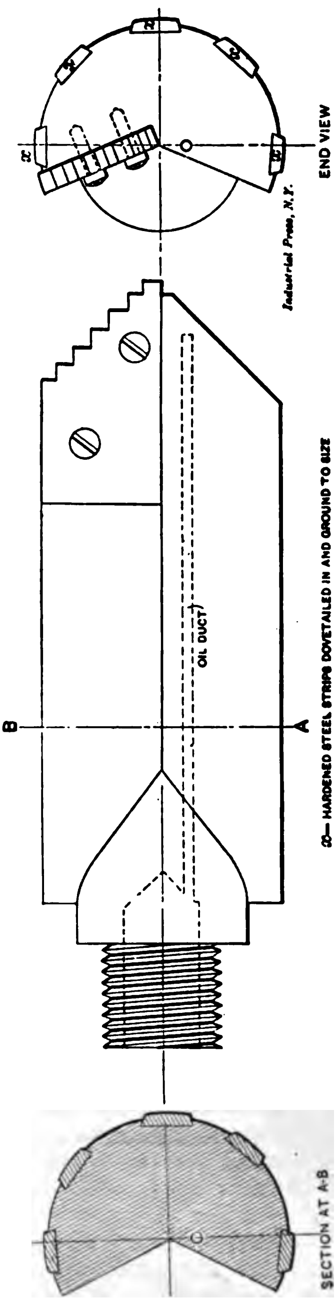

Fig. 8. D-drill with Inserted Blade, used for Deep Hole Drilling

It is obvious that when a hard or soft spot is encountered in boring with a tool having a single cutting edge, only that particular place is affected by the spring of the tool; while with a double cutter, as shown in Fig. 9, first sketch, any deflection due to irregularities, such as at *a* or *b*, will cause the tool to spring and the cutting edge on the opposite side to introduce similar irregularities in the opposite side of the hole. This is one objection to the two-lip drill for accurate work.

With three points the tool is somewhat better supported when a high place is encountered, as shown in the second sketch, Fig. 9, and when a cutting point strikes a low place the other two edges are not moved away from their position so much as if they were opposite

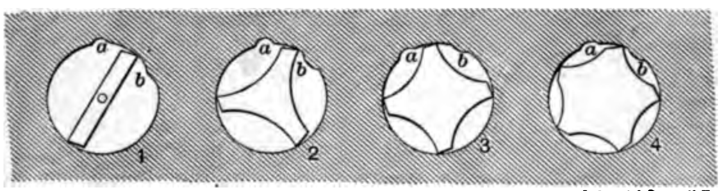

Fig. 9. Effect of the Number of Cutting Edges

the first edge. Hence a tool with three edges should prove better than one with two, and one with four, being better supported, would seem better on this account than one with three, but has the disadvantage of opposite cutters. Five edges ought to give still better results.

In Fig. 10 is shown a four-grooved chucking drill which is suitable for truing up either a drilled hole or a cored hole, but obviously it cannot be used for drilling out solid stock. It has less tendency to

Fig. 10. Four-lip Drill for Cored Holes

"run out" than a two-lip drill, and the edges are less likely to catch under the scale or in breaking through, since each has only half the depth of cut to take.

In general, it may be said that in boring the best results are obtained when the tool has a single cutting edge, but if it is desirable to have more cutting edges, a tool with several will be more satisfactory than one with only two. Any machinist who has tried to true up the taper hole in a lathe spindle, first by boring and then by reaming, will appreciate the superiority of the boring tool over the multi-blade reamer. A reamer sometimes refuses to produce a perfectly round hole, and will do this whether the number of teeth is odd or even. The writer has seen the photograph of the bore of a 12-inch gun that had been reamed out with an eleven-sided reamer, and the bore had eleven distinct sides, clearly visible in the photograph. The trouble was overcome by spacing the reamer teeth unequally.

Advantages of the End Cut

One trouble with reamers, however, is that the teeth necessarily cut on their side edges instead of on their ends, and the whole effect of any unevenness in the hole is to crowd the reamer to one side. The condition exists to a less extent with a flat or twist drill, where the cutting edges are at an angle with the center line, and the resultant of any unusual pressure is felt partly as a side thrust and partly as an end thrust. Now, by making a drill to cut squarely on its end and but very little, or not at all, on its sides, the side thrust is mostly done away with. The end mill shown in Fig. 11 is a good illustration of such a tool, and it is known to be capable of boring very accurate holes when used for that purpose in the milling machine.

In Fig. 12 is a boring tool with a single cutting edge, which cuts on its end and is capable of drilling a true hole in solid metal. It consists of a round tool steel bar, with one end flattened and ground to form a cutting edge, as shown. It is designed to be held in the tool-post of the lathe, in a position perpendicular to the face plate.

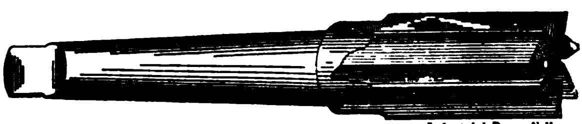
Industrial Press, N.Y.

Fig. 11. End Mill with Center Cut, suitable for Accurate Drilling

The inner edge or corner of the cutting edge should be slightly rounded to help support the cutter and prevent chattering, and the width *A* of the cutting edge should be from 1/32 to 1/16 inch less than the radius of the hole to be drilled. The objection to this tool is that it cannot be supported stiffly enough by the tool-post for a hole of great depth, and for this purpose the D-drill shown in Fig. 13, and which works on the same principle, is well adapted. In its simplest form it consists of a round bar of the diameter of the hole to be drilled, one-quarter to one-half of which is milled out to give a passage for the chips. The end of the bar is shaped with a cutting edge on one side, extending almost or quite to the center, and with the other side relieved to give clearance for the cutter. Such a drill should be supported by a bearing close to the hole to be bored, in case it is to start the hole itself, and it is better yet to start the hole with a twist drill and true it up with a single-pointed boring tool, and then let the drill be guided by this hole. This is the surest way of getting a hole concentric with the axis of the lathe. As the bar is of the same diameter as the hole, the cutter will be supported by one-half the surface of the hole, and if it is once started right in an accurate hole, it will continue in the right direction.

It is desirable to have the cutter blade separated from the bar or head, as the case may be, so that it may be renewed or removed for grinding, particularly in drilling large holes. In Fig. 8 is an illustration of such a cutter head as used by a large ship- and engine-building concern, in drilling propeller shafts. The body of the cutter is made from soft steel with tool steel strips *x* dove-tailed in and ground in place to size. The cutters are made by jigs and are interchangeable, and their shape is such as to break up the chips, which are washed

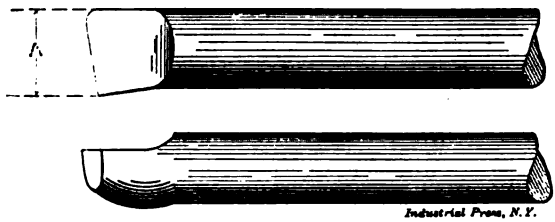

Fig. 12. Boring Tool with the Cutting Edge on the End

out by the force of oil supplied by a pump, through the hollow boring bar, to which this cutter head is fastened. One of these cutters, four inches in diameter, has bored 12 inches in a piece of nickel steel in one hour, cutting a fair and smooth hole, and no trouble has ever occurred, even when holes have been drilled to a depth of 32 feet.

Rotating Work vs. Rotating Drill

In deep hole drilling it is customary to have the drill fixed so that it cannot turn and rotate the work, following the usual method in this respect for boring accurate holes in the lathe. Since the outer end of the work can be supported in the center rest, it is always possible to make the work run true, while it is not so easy to make a drill run true and coincident with the axis of the work. The difference in principle involved has already been explained in the introduction and shown graphically in Fig. 1.

Deep Hole Drilling Attachment for the Lathe

An attachment is shown in Fig. 14 for performing deep drilling rapidly and economically on the engine lathe. This attachment is built by the Lodge & Shipley Machine Tool Co., Cincinnati, Ohio. It

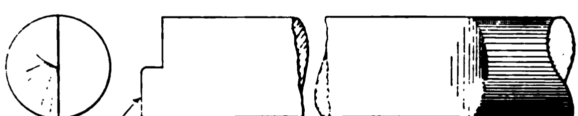

Fig. 13. D-drill used for Deep Drilling

consists essentially of a drill spindle, mounted on the cross-slide in place of the usual tool-post, in combination with an electric motor and suitable gearing for rotating the spindle. A support is provided for holding the outer end of the work, the other end of which is clamped in the chuck or face-plate of the lathe. Provision is also made for forcing a copious supply of lubricant to the point of the drill used. The purpose of the attachment is to make it possible to drill a hole true with the center line by the usual method of rotating the work, and at the same time to give the high cutting speed of which high-speed tools are capable, without necessitating a high rate of revolution for the heavy spindle and gearing of the lathe.

The drill spindle bearing, with the bracket on which the motor is mounted, is cast as one piece with the bed-plate. This plate is bolted to the wings or arms of the carriage. The 3-horsepower 2 to 1 variable speed motor shown, is connected to the drill spindle through an intermediate rawhide gear. The spindle is bored to supply lubrication to the drill; it has a large bearing, and is ring oiled. The drill

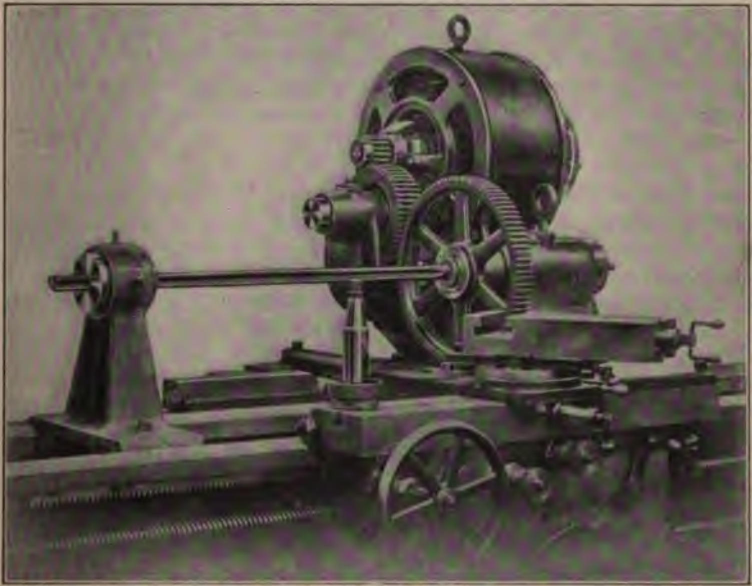

Fig. 14. Lodge & Shipley Motor-driven Attachment for Deep Drilling in the Lathe

shank is fitted to the hole in the spindle by reducing bushings. The outer end of the drill is carried in a free bushing, revolving in a support bolted to the lathe bed. The drill used is of the special construction known as the "Chard" deep drill. A flat blade of high-speed steel is held in position at the end of a steel shank by a tapered pin; it is so ground as to break up the chips and thus facilitate their removal. Lubrication under pressure sufficient to clear the chips and cool the cutting edge is supplied by a pump attached to the lathe at the rear of the head-stock, and driven from the lathe countershaft. Flexible tubing connects this pump with the hollow spindle through a nipple at the rear. Two copper tubes, flush with the surface of the drill, carry the lubricant to the cutting edge. This type of drill has been in use for some time on lathe spindles, back gear sleeves and pulley sleeves. Under favorable conditions a 2-inch drill has bee advanced at the rate of 2¼ inches per minute. The drill is illustrat and described in Chapter III.

This whole attachment may be easily removed from the carria by the use of an over-head crane, suitable I-bolts being provided f this purpose. Only a few minutes times is required to change t machine over for engine lathe work. The particular device illustrat is used regularly for drilling holes in locomotive driving axles, t holes being 1 inch in diameter and 44 inches deep.

Drilling Deep Holes by the Pratt & Whitney Method

A highly satisfactory drill for use in drilling deep holes is o brought out by the Pratt & Whitney Co., principally for use in conne tion with their gun-barrel drilling machines. The tool in question a development of the old D- or hog-nose drill, already described, whic has one cutting lip only. It is carefully ground on the outside, and supplied with an oil duct through which oil at high pressure may brought directly to the cutting edge.

Referring to Fig. 15, *A* is the cutting edge, *B* the oil duct, and the chip groove. In milling the latter groove, the cutter is broug directly to the center line, so that in this respect the drill is ve free-cutting as compared with the ordinary two-lip twist drill whic has a central web. In the end view, the shape of the chip groove clearly indicated. The cutting edge *A* is radial. In sharpening t drill, the high point or part first entering the work is not groun in the center as is usually the case in drills, but to one side as show in Fig. 15, in which *D* is a cross-section of the work being drille and *E* the high point of the drill. Grinding the drill in this mann makes possible its running true or straight, the teat *F* on the wor acting as a support to the drill, which, owing to its periphery bein partly relieved, would have a tendency to travel in a curve aw from its cutting side. The piece being drilled is run at very hig speed, the periphery speed at the outer diameter of the hole being high as 130 feet per minute on machine steel. The feed, however, quite fine, on a 0.3-inch drill averaging 0.0004 inch per revolutio while on a 3-inch drill it is about 0.0008 inch. These figures, of cours are dependent to a great extent on the material being drilled. Th drills are made of high-grade steel and left very hard, so that th fine feed has little tendency to glaze the cutting edge.

The piece being drilled is held and revolved at one end by a sui able chuck on the live spindle of the machine, while the other en which should be turned perfectly true, runs in a stationary bushin having at its outer end a hole the diameter of the drill. The dr enters the work through the bushing, and is thus started perfectl true. The arrangement is indicated in the upper view in Fig. 16, i which *A* represents the chuck, *B* the work, *C* the bushing, *D* the su port for holding the bushing, and *E* the drill. Through the oil du of the drill, oil is forced at a pressure varying from 150 to 200 pound per square inch. After passing the cutting edge, the oil returns t the reservoir by the way of the chip groove, forcing the chips along in its travel. In drills of large diameter, especially when working on tough, stringy material, the cutting edge is usually ground so as to produce a number of shavings, instead of one the full width of the cutting lip, so that no trouble is experienced in getting chips out of the way. The oil, of course, is used over and over again, and with a large reservoir will be kept quite cool.

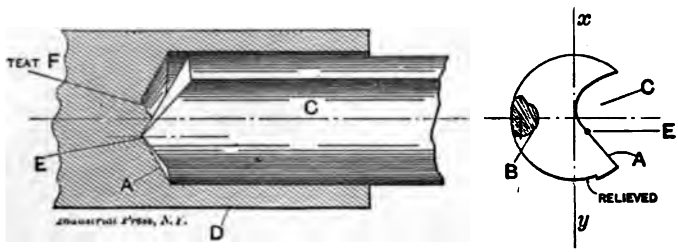

Fig. 15. Pratt & Whitney Co.'s Deep Hole Drill

The drill is made up of the drill tip and shank, the tip varying in length from 4 inches to 8 inches, while the length of shank is determined by the depth of hole that is to be drilled. The lower view in Fig. 16 will clearly illustrate the construction of a small, complete drill, *A* being the tip, *C* the shank, and *D* the oil duct. The shanks on small drills are made from steel tubing, rolled as shown in the cross-section at the right-hand end. The tip is carefully fitted and

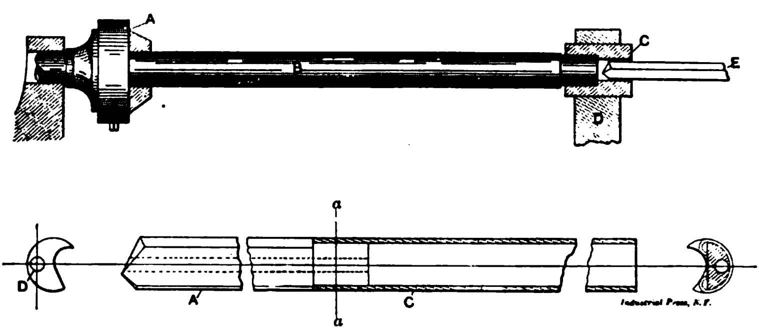

Fig. 16. Arrangement for Starting the Deep Hole Drill, and Method of Fastening Shank to Drill

soldered to the shank, which, it should be noted, is a little smaller in diameter than the tip.

The relief or clearance of the cutting edge of the drill, the amount the "high point" of the drill should be off center, and the number of rings on the end of the drill, when provided with notches for breaking up the chips, depend entirely upon the material that is to be drilled. For instance, on very soft stock, the supporting teat should be more substantial than on hard spindle or gun steel, so that it is evident that on soft stock the high point should be more off center, or nearer to the outer diameter, than on hard stock. Figs. 17 and 18 are reproductions from actual photographs of a 3-inch drill, and the reader will obtain a very clear idea from the engravings of the appearance of the tool described. These figures illustrate a drill ground on the end so as to produce several shavings.

The present practice in relieving the large drills is shown in Fig. 21. The straight, or radial, edge is the cutting edge of the drill and the

Figs. 17 and 18. Side and End View of Deep Hole Drill

distance *B* is about $\frac{1}{8}$ inch on a one-inch drill. The surface *A* is left of the full radius of the drill, and makes a good back rest. When the drill is ground on its periphery, it is made very slightly tapering toward the shank to free itself. As previously stated, in milling the chip groove the cutter is brought exactly to the center of the drill. When hardening and grinding, however, the location fre-

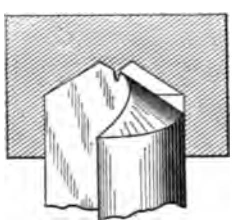

Fig. 19

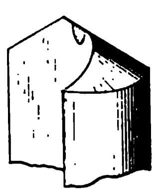

Fig. 20

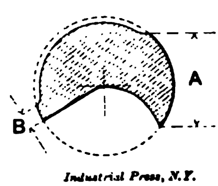

Fig. 21

quently changes slightly so that the groove does not come to the center of the drill. In such cases it is necessary to grind out the lip at the point as shown in the illustrations made from photographs in Figs. 17 and 18. Generally the operator grinds this a little beyond the center, but no trouble results, as the small teat produced thereby is broken off when the bottom of the ground out place comes in contact with the end of the teat, as indicated in Figs. 19 and 20.

Fig. 22. Pratt & Whitney Co.'s Tube and Gun-barrel Drilling Machine

subjected. After having ascertained the amount of existing wa
having investigated the concentricity of the inner and outer c
ferences, the tube is centered and spotted in a special lathe
nated for the purpose, preparatory to the first rough boring
tool used in rough boring, commonly known as the hog-nos
and represented in Fig. 23, consists of a semi-circular cast-iro

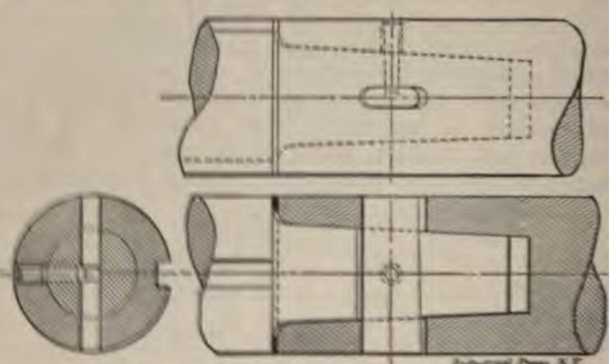

Fig. 24. Front End of Boring Bar showing Method of Attaching Boring Head

provided with a shank on one end secured to the front end
boring bar (which has a taper and flat keyhole to receive it, Fig
and a cutter clamped on the forward end, the material of wh
high-grade tool steel, very carefully tempered. The cutter is
with the edge cutting square on the forward end, and with t
turned slightly upward, in order to roll the chips backward, a
cated in Fig. 25.

Fig. 25. Detail of the Boring Tool

Provision is made to throw a jet of a lubricating mixture in
of the tools during the operations of boring, reaming, and fin
the lubricant serving to diminish the friction on the tools a
as to preserve their temper. This is accomplished by a rota
pump which conveys a continuous stream of oil by means
wrought-iron pipes connected with an intermediate flexible m
hose which travels with the boring bar and is attached to a
a which passes through the interior of the boring bar, termi

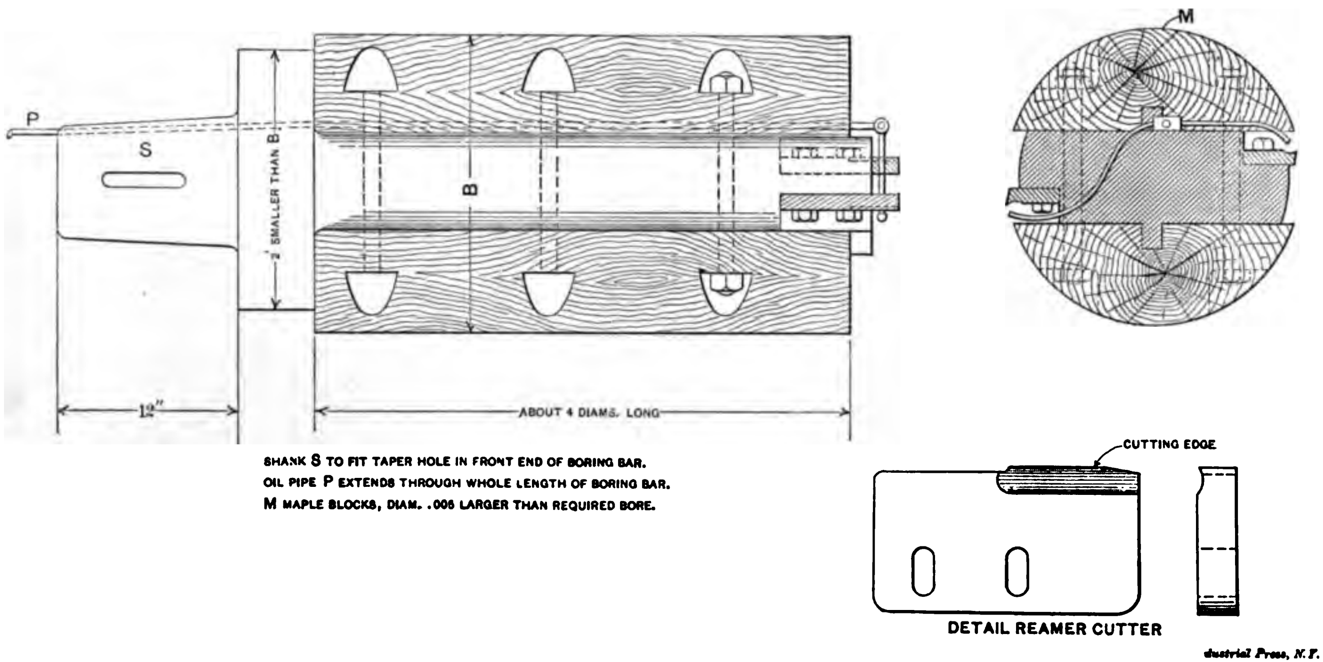

Fig. 26. Tool for Finishing and Reaming Holes in Large Guns

at the end of the cutting tools. The pump is provided with
changes of speed.

Although this tool, because of its design, is capable of re
considerable quantity of metal, it has proved most practic
allow it (though this is seldom required) to remove more t
3/4 to 1/2 inch of metal on the side, leaving approximately
to be removed by the reaming and finishing tool in the s
operations.

The rough boring being completed, the hog-nosed tool is
and the tool shown in Fig. 26, for finishing and reaming, i
in its place. This is made of a cast-iron body about four
in length, in which are bolted two semi-circular blocks of
selected and well seasoned oak or maple that have been previ

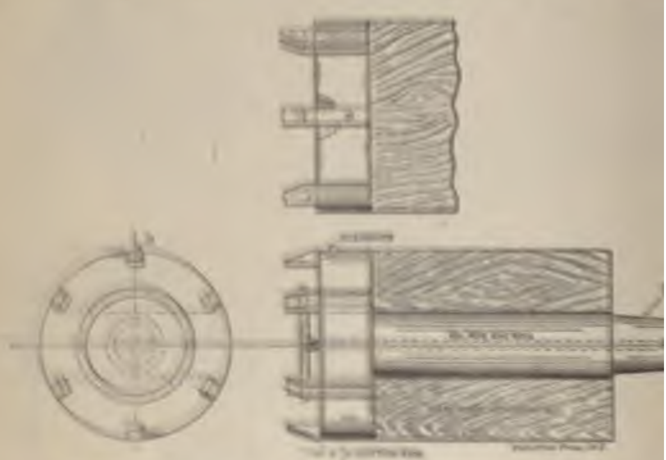

Fig. 27. Boring Tool for Holes of from Three to Six Inches in Diam

merged for a long period in a bath of lard oil and that
guides for the cutters bolted to the front end of the head,
in detail in Fig. 26. These blocks, while in position, are t
to a diameter about 0.005 inch larger than the desired bore,
being to force them in and thus give them a smooth and
bearing surface. The cutters used are made from a very h
steel, ground to exact size, and oilstoned to conform to the
diameters.

To insure a perfect alignment for the tool before comme
actual reaming, it is good practice to counterbore the tubes
4 inches deep. The method of supplying oil while reami
same as employed in the operation of rough boring. It m
interest to note that, with the same cutting speed, the fee
from four to five times that of the hog-nosed feed, by giving
a taper in front of about 1 to 10, with a straight cutting ed
long on the side. (See detail of reamer cutter in Fig. 26.)

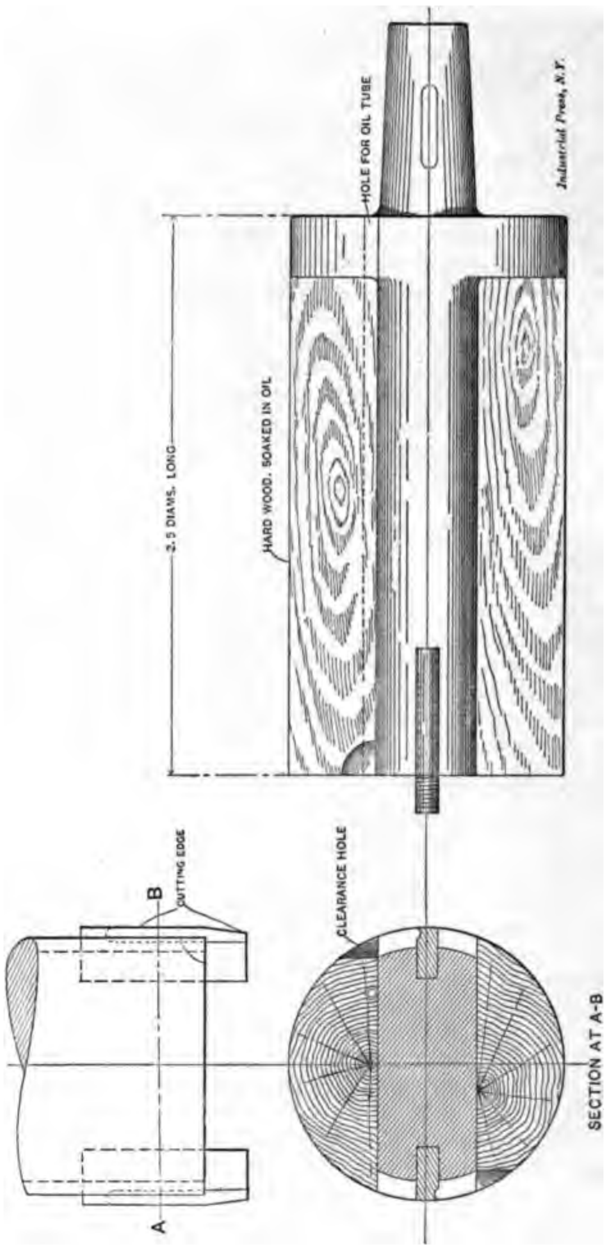

Fig. 28. Finishing Tool for Reaming Holes of from Three to Six Inches in Diameter

struction also serves to maintain the size in case the forward part of the cutter becomes slightly worn.

Multiple cutter wood reamers, illustrated in Fig. 27, have been successfully used for bores of from 3- to 6-inch caliper, for lengths varying from 5 to 15 feet. Greater feed can be employed with tools thus constructed, and consequently considerable time is saved by their use. It must be borne in mind, however, that in tubes of great length, in which there is an unequal distribution of stock on the interior, these tools have a tendency to run out of their true axis, thus leaving an unsymmetrical bore. This difficulty is obviated and a perfect and uniform bore obtained, by the use of the previously described hog-nosed tool and wood-lined reamer, although some time is sacrificed owing to the reduced feeds at which they have to be run compared with the feeds obtainable with a multiple cutter wood reamer.

In boring guns, lathes are employed which are specially designed for the purpose. Nevertheless, if such a machine is not available, for short lengths and small bores, it is advisable to use the old and well-known method of attaching to the cross-slide of a common lathe of suitable length of bed a fixed support or bracket having a bore to receive a number of bushes, the bores of which correspond to the varying diameters of boring bars. The hub of this support or bracket has one side split to enable it to be tightened by a bolt, in order to retain and grip the bushing which is likewise split for the purpose of gripping the boring bar. The boring and reaming tools shown in the engravings, Figs. 27 and 28, are particularly applicable in this case. After having determined the required feed and set the lathe in motion, the carriage will serve to feed the bar in the usual manner. The other functions are performed in the ordinary way and are too obvious for further explanation.

Machines Employed in Boring Large Guns

While, as just mentioned, any lathe may be employed for deep hole drilling, when large guns are bored, use is made of machines especially designed for the purpose. In the following is shown and described the general design of the boring and turning lathe employed at the Government Army Gun Factory, at Watervliet Arsenal, N. Y., for performing the necessary operations in connection with a 16-inch breech-loading rifle. Of course, only the most characteristic features of this huge lathe can be dwelt upon.

Before entering upon a description of the lathe, however, a few explanatory words with reference to the gun constructed may be in place. The semi-longitudinal section in the upper part of Fig. 29 will serve to explain the method of assembling the built-up or hooped gun, which is composed of several parts united to form a whole. The parts are properly arranged to support the stresses upon them, and the gun has therefore been termed "built-up or hooped." These parts are assembled according to modern practice adopted by the ordnance department, for which purpose the army gun factory is equipped with cially-designed oil and high-pressure steam shrinkage furnaces.

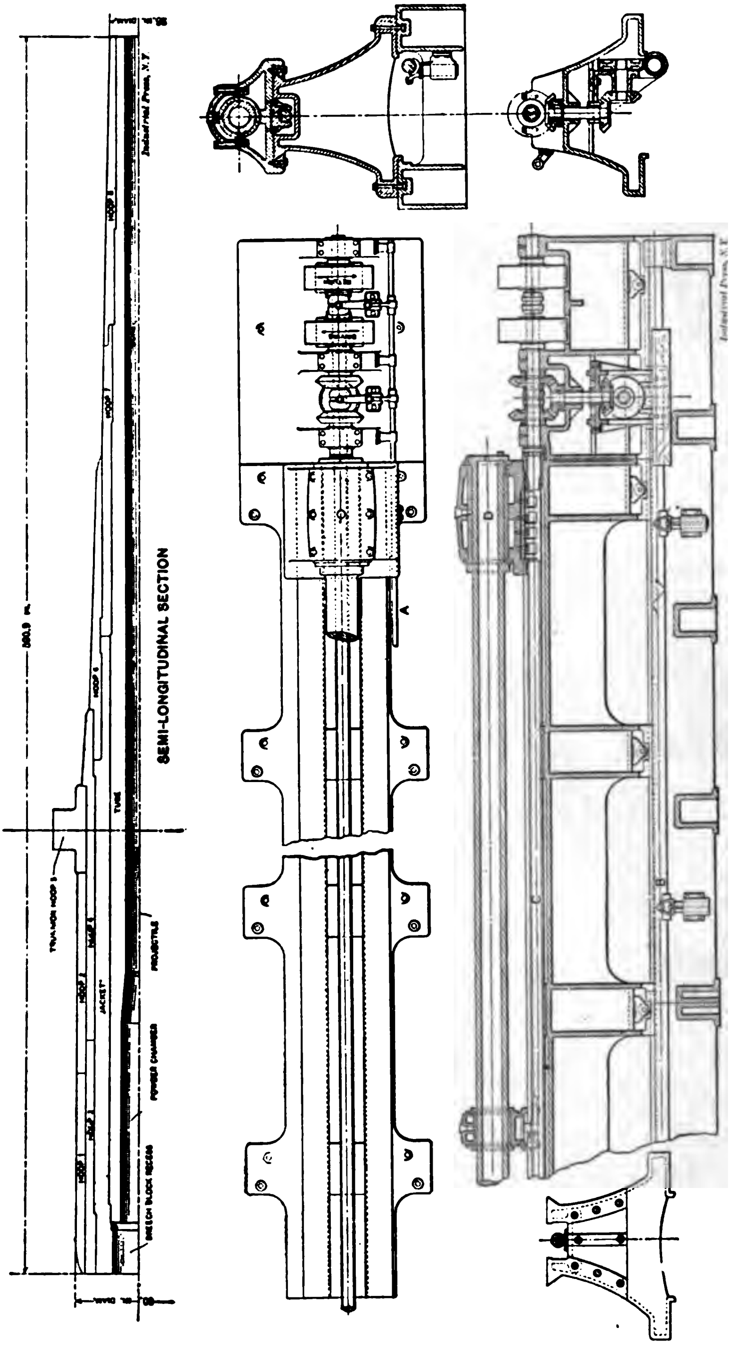

Fig. 29. Detail Drawing, showing Hoops on 16-inch Gun, and Plan and Sectional Elevation of Boring and Reaming Bar for 16-inch Gun Lathe

Preparatory to shrinkage, all parts constituting the gun have been subjected to various accurate and particular operations performed by special machines and have been measured with exact instruments and gages designed and made at the army gun factory, for ascertaining the required dimensions. The central portion of the gun is called the tube, upon which are shrunk the series of hoops. The jacket is shrunk on the rear portion of the tube, overhanging the tube a certain distance to form the breech recess. The tube and jacket are made of nickel steel, the remaining hoops being of fluid compressed steel containing no nickel. After the ingots for the tube, jacket and hoops have been forged on mandrels under hydraulic pressure, they are tempered and annealed, in order to give them the desired physical qualities.

The lathe for boring the tubes for this gun consists of the bed, headstock, two tool carriages, muzzle, intermediate, and breech rests, and a boring bar with carriage and four supports, as shown in Fig. 29. The main bed is 44 feet long by 9 feet 10 inches wide, made in two sections. The boring bar carriage bed is 67 feet long and 6 feet wide, and is made in three sections. All sections are firmly joined by taper bolts. The weight of the whole lathe is 280 tons. There are two similar tool carriages provided with a hand lever by which the lead-screw nut is engaged and disengaged. The lower lateral slide traverses the length of the main bed by means of a feed-screw, or by the use of a ratchet wrench. On this slide is the lower cross-slide, and on the intermediate cross-slide the two top cross-slides are located, forming a support for the cutting tool. The lateral and cross-feed motions are derived from the splined feed-shaft. Longitudinal movement of the carriages upon the bed can also be accomplished by means of a rack and pinion. In addition to the taper attachment, the cross slide is fitted with a circular base, 4 feet 4 inches in diameter, graduated in degrees. Each of these carriages weighs 18 tons.

Fig. [illegible] shows the interior of the sea-coast gun shop of the arsenal, with guns of different caliber under construction. The total length of this shop is 1,000 feet, the south wing is 150 feet wide, and the north wing 150 feet wide by 600 feet long. In the foreground of this illustration a gun is shown being bored and turned in the lathe described.

The steady rests have forged steel jaws, the inner ends of which are provided with brass concave caps and openings for taking in various sizes, which is accomplished by fitting into the steady rests internal rings containing an extra set of long chuck jaws. The rests consist of a housing in three parts, i. e., base, top, and chuck ring, and are thus arranged to facilitate the removal of the work. They are bolted to T-slots in the bed by large bolts, and are easily moved in a longitudinal direction by hand by applying a ratchet wrench at the square end of a horizontal transverse shaft, shown in engraving Fig. [illegible], journaled in the base of the rest. This shaft terminates in a miter gear engaging a similar gear on the upper end of a vertical shaft, which in turn carries a pinion engaging a rack secured to the

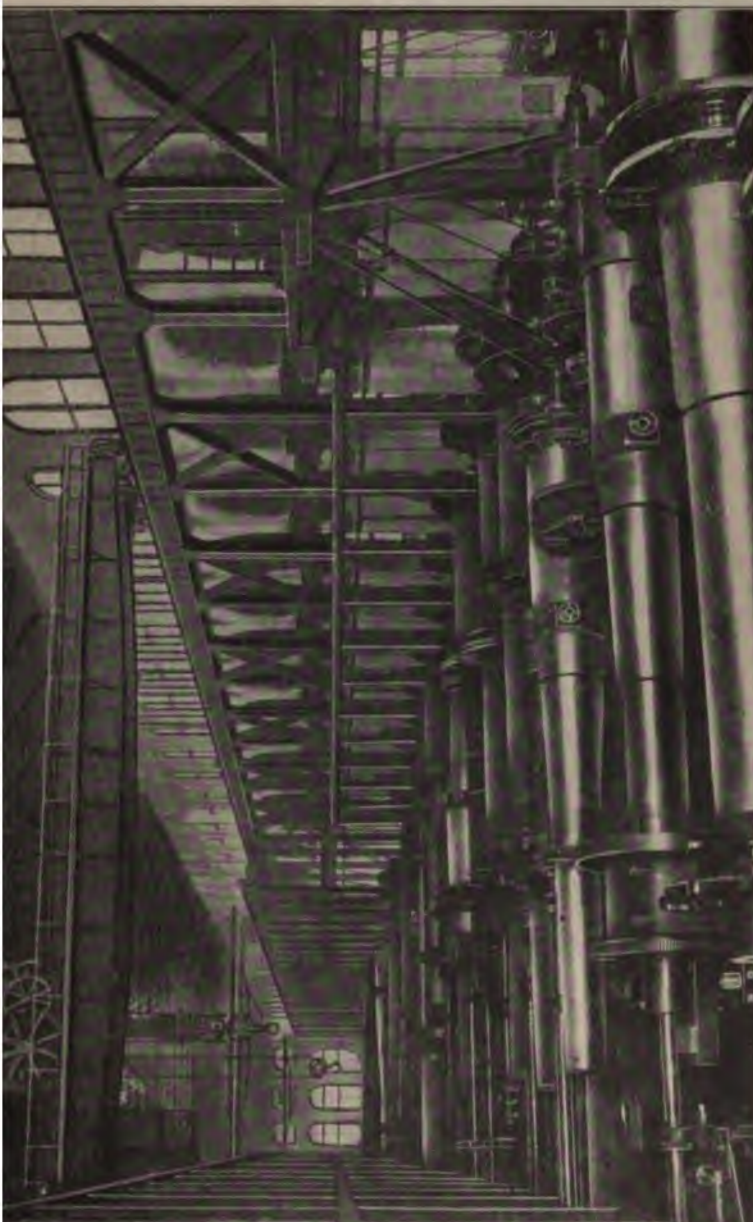

Fig. 30. View of the Interior of the Watervliet Arsenal Gun Shop

which is in communication with the friction clutches and pulleys located in the rear of the lathe. The pulleys are driven by a separate line shafting. The boring bar is of forged gun steel, 11 inches in diameter, 61 feet 3½ inches long, with a hole 4 inches diameter throughout the whole length; its weight is equal to 6 tons. The head-stock is 23 feet 4 inches long by 9 feet 10 inches wide, and is made in one casting. The cone has five steps for a 10-inch belt and is strongly back-geared, giving a wide range of changes of speed by means of triple gearing to the face-plate.

The main spindle is made of close-grained cast iron, 24 inches diameter at the main journal, and has a bearing in a cast-iron shell 31¾ inches long. The rear bearing is similar in construction, but is only 16 inches diameter by 25½ inches in length. The spindle is provided with a flange to which the face-plate is bolted. It is also provided with a thrust bearing of ample surface, with anti-friction washers at the front bearing. The main spindle bearings are made in halves and capped. The face-plate contains slots for clamping the work and has several adjustable steel chuck jaws, shown in Fig. 33, which have V-shaped faces. It also has a cast steel ring gear securely bolted to it. This gear is 10½ inches face, 8 feet 9½ inches outside diameter, by 3.59 inches circular pitch, and serves to drive the face-plate. The face-plate weighs 13 tons.

All boring, reaming, turning and facing operations are performed in the lathe described, the mechanism permitting of boring and turning simultaneously, in order to facilitate rapid progress in construction. The principal remaining operations for the completion of the gun consist mainly in rifling the bore, and threading and slotting the breech. The 16-inch breech-loading rifle boring and turning lathe was designed by the ordnance department and manufactured by the Pond Machine Tool Co., Plainfield, N. J.

CHAPTER III

CONSTRUCTION OF DEEP HOLE DRILLS*

It has always been an expensive operation to drill out the interior of a hollow spindle. Indeed, it has not only been an expensive operation, but, on account of the spindle being of high-carbon steel, it has also been a difficult operation. Thus we find on the market hollow-drawn tubing, such as the Shelby tubing, which is intended to be furnished at a less price than one can bore out the interior of a solid bar. Of course, for some purposes this tubing answers very well. But there are many cases, as in certain kinds of spindles, where such tubing cannot be used.

Nearly all lathes nowadays must be built with a hollow spindle. This is also true of spindles of boring machines, drill presses, etc., and thus we have a large variety of work on machine tools which involves deep drilling. The cost of deep drilling has been greatly reduced recently, by several manufacturers, by means of what may be termed a hollow drill. A strong flow of oil is forced through this drill against the cut, and on its return carries the chips with it, thus performing the double function of keeping the drill cool and clearing out the chips.

A perspective view of one of these drills fitted up complete for work, is indicated in Fig. 34. This drill was developed in the shops of the Lodge & Shipley Machine Tool Co., Cincinnati, Ohio, by Mr. N. D. Chard. The body of the drill *B* is made of machine steel. The point *P* is made of tool steel, and is held in position by the taper pin *T*. A hole *H* is drilled in the shank, and from this hole the oil is led to slots *S*, which are milled along the outside. These slots run the full length of the drill, and then shoot down at the ends, as indicated. *F* is a flat spot for holding the drill.

A longitudinal sectional view of the drill is shown in Fig. 35, from which the construction of the passageway for the oil is better seen. *H*, again, is the oil inlet hole already referred to. Two small holes *J* are drilled into it in the manner indicated in this figure. The holes *K* are then drilled, and a piece of brass tube is bent in the arc of a circle and the ends are entered into the holes *J* and *K*. It is then hammered down into place, and the joint is flushed with solder. The slot *P* is milled out so as to have a semi-circular bottom. Into this slot the cutter or point of the drill is neatly fitted. These points are best made of Novo steel, as they then hold up better under high speed. They are made as shown in Fig. 37. The hole *H* in the cutter is reamed through the drill, while the cutter is clamped firmly back against its seat at the end of the slot. The angle *A* is made about 20 degrees. The cutting edge is nicked at several places, as at *N*, in

* MACHINERY, January, 1904, February, 1905, and October, 1905.

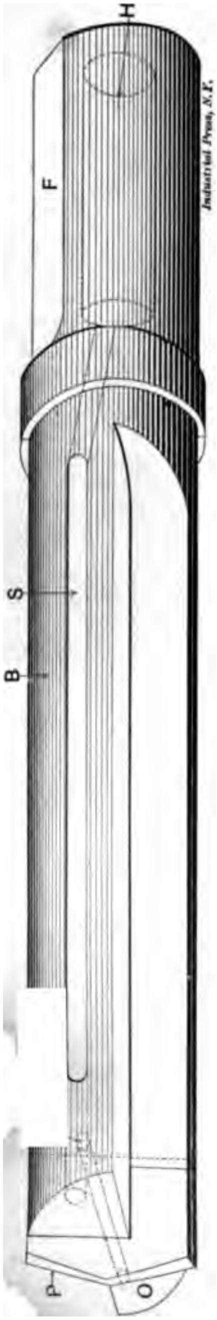

Fig. 34. Lodge & Shipley Oil-tube Drill for Deep Drilling

Fig. 35. Section of Lodge & Shipley Deep Hole Drill

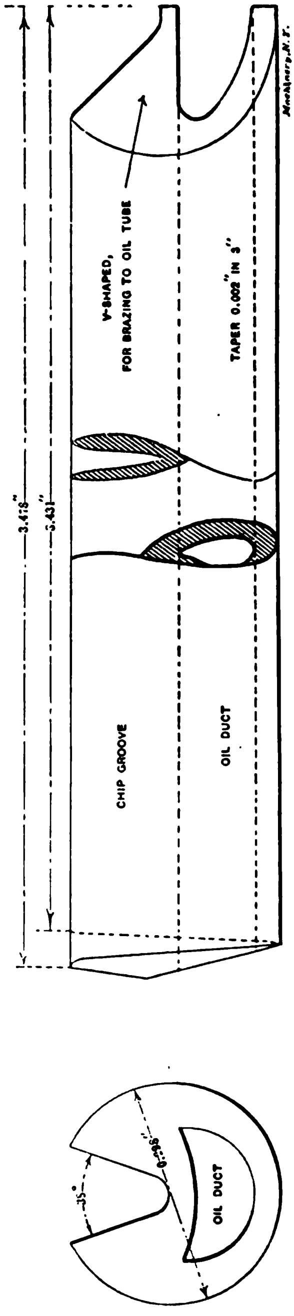

Fig. 36. Detail of Deep Hole Drill, 0.308 inch Diameter, Enlarged about Four times Natural Size

order to break up the chips, this being done on the corner of an emery wheel. After the drill is put into place, it is ground up accurately to the diameter *D*.

The arrangement of a turret machine for boring out lathe spindles with this drill is shown in the illustration in Fig. 38. Instead of the long slide and hexagonal turret, a special slide *A* is provided, which receives the drills. *B* is a steady-rest, and *S* is the spindle. A lathe carrier is screwed onto the end of the spindle, as shown, and a plate *P* holds the spindle up against the center by means of a few bolts. At *C* is shown a Brown & Sharpe No. 2 pump, which is belted up to run 350 revolutions per minute. This gives an ample supply of oil for drilling a three-inch hole. A flexible tube leads from the pump to the fitting *F*, and from thence the oil enters the spindle. The trays *T* catch the oil, and from these trays a pipe leads to the suction side

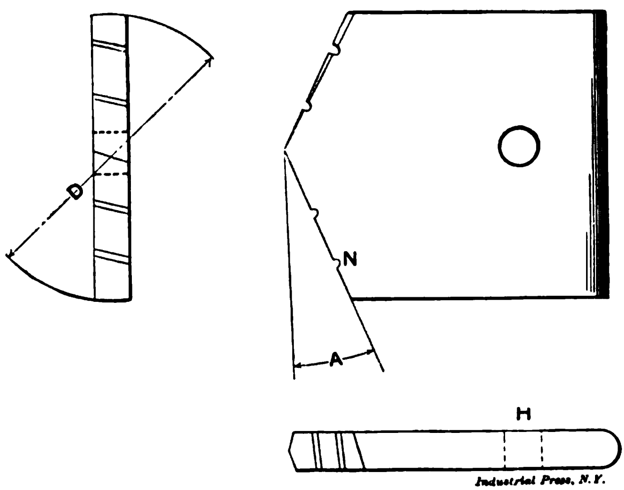

Fig. 37. Detail of Blade Inserted in Drill in Figs. 34 and 35

of the pump. The drill is made long enough so as to run a short distance beyond the middle of the spindle to be drilled, the spindle being then turned around and drilled from the other end. With a drill made up in the manner explained, it is possible to drill a 1 5-16 diameter hole in a 0.40 carbon steel spindle at the rate of 1⅜ inch per minute. The cutting speed would be 60 feet per minute at the largest diameter of the drill.

To Measure the Radii of Deep-hole Drills

Deep-hole drills of the type shown in Fig. 39 should be made so that the cutting edge or lip *a* of the drill is a radial line, or in other words, so that it goes exactly through the center of the drill. It is clear that if a drill is made having the cutting edge under the center, a small core of the stock in the center will not be cut off, and will instead tend to remain and form a slender column or thread remaining in the hole. On the other hand, if the cutting edge lies over the

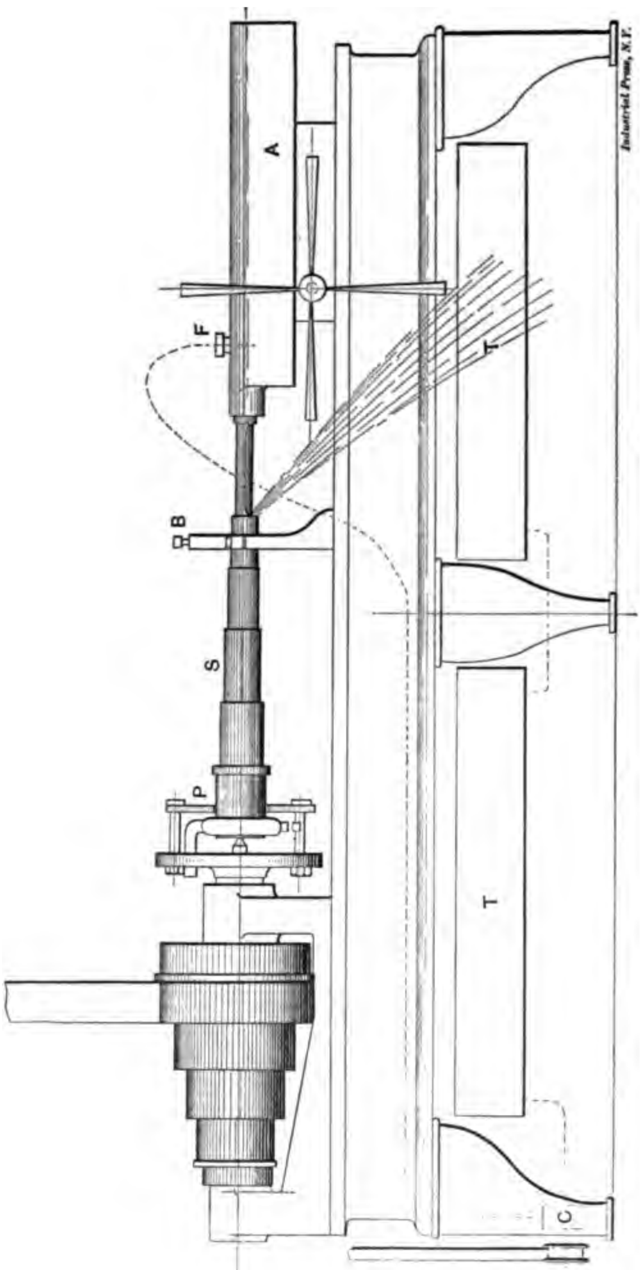

Fig. 38. Turret Lathe Arranged for Deep Hole Drilling

center, the center portion cannot be cut out, but will be crushed together, the same as occurs with an ordinary twist drill. In deep hole drilling this is not permissible, as it will very soon cause inaccuracy of the hole or breaking of the drill. Hence it follows that the

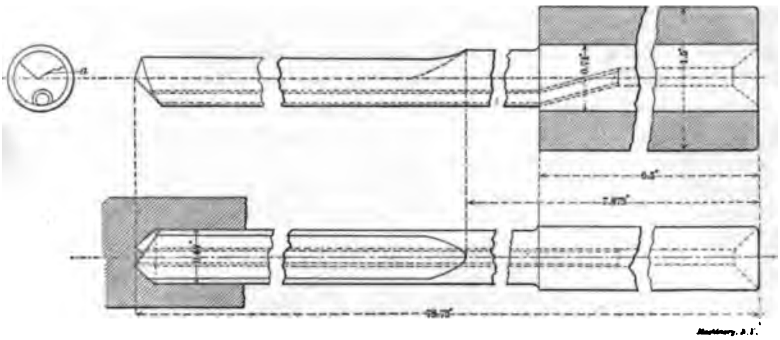

Fig. 39. Type of Deep Hole Drill Gaged by Method shown in Figs. 40, 41 and 42

location of the center exactly in the cutting edge is very important. A glance at a cross-section of the drill, Fig. 39, convinces us that a direct measuring of the drill center cannot be made with ordinary instruments. Hence Ludwig Loewe & Co., of Berlin, Germany, employ for this purpose a special micrometer with disks and a V-shaped anvil clock, the principle and use of which is described in the following.

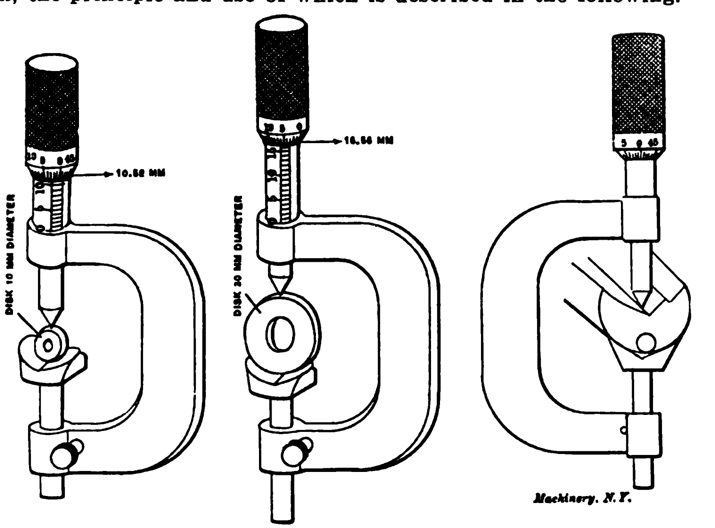

Fig. 40 Fig. 41 Fig. 42

Fig. 40 shows the special micrometer, which measures the radii of deep hole drills up to 60 mm. (2.362 inches). Its construction is the same as that of an ordinary micrometer with a V-point for measuring screw threads, but in addition it has an adjustable V-block which is eter reading of 8.2 mm., it is known that the apex of the angle of the groove lies exactly in the center of the drill.

To adjust the micrometer for drills from 30 to 60 mm. diameter, the radius of the drill is multiplied by the constant 1.1045, and 15 mm. is subtracted from the product, or $r \times 1.1045 - 15$ mm. $=$ micrometer setting. The diagram, Fig. 48, however, saves this calculation and is

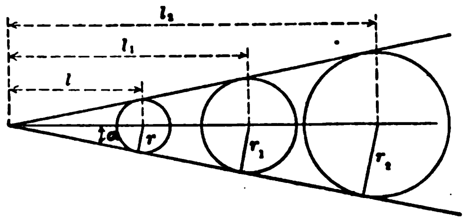

Fig. 45

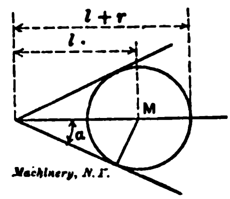

Fig. 46

used in the same manner as Fig. 47. For example, to find the micrometer setting for a drill having a radius of 27.35 mm.: In the upper horizontal scale we find the point corresponding to 27.35, and follow the vertical line downward until it intersects the diagonal, thence to the right or left to the vertical where we read 15.2. With this datum, any excess or deficiency of thickness of the drill over the

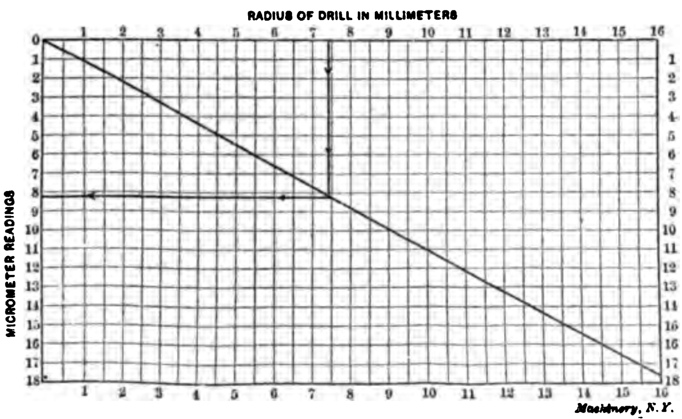

Fig. 47

center can be read off directly from the scale. Figs. 43 and 44 are made from photographs showing substantially the same as Figs. 40, 41, and 42.

Principle of the Gage

The principle of the gage depends upon the proposition that the homologous sides of similar triangles are in direct proportion. Hence in Fig. 45 we have $\frac{r}{l} = \frac{r_1}{l_1} = \frac{r_2}{l_2}$, etc. Therefore the values of l_1 and l_2 are easy to determine, since $l = \frac{r}{\sin a}$ in which r corresponds to the radius of the drill, and $a = \frac{1}{2}$ the constant angle of the V-block of the micrometer, whence we derive the constant 1.1045.

It will be observed that the product of the radius by the constant is used directly when setting for drills 32 mm. or less diameter, but that when measuring drills of 30 to 60 mm. diameter the product is diminished by 15 mm. This is because in setting to the 30 mm. disk, the micrometer screw is adjusted to 16.56 mm., instead of 31.56 mm., in order to make the scope of the micrometer include the larger drills without the need of making a larger frame. The principle is illustrated in Fig. 46. For the disk M we reckon $l + r = \frac{r}{\sin a} + r$. Hence if one sets the screw of the micrometer to this measure, and pushes the V-block with the disk M resting in it, until the disk is found to be in perceptible contact with the measuring point of the

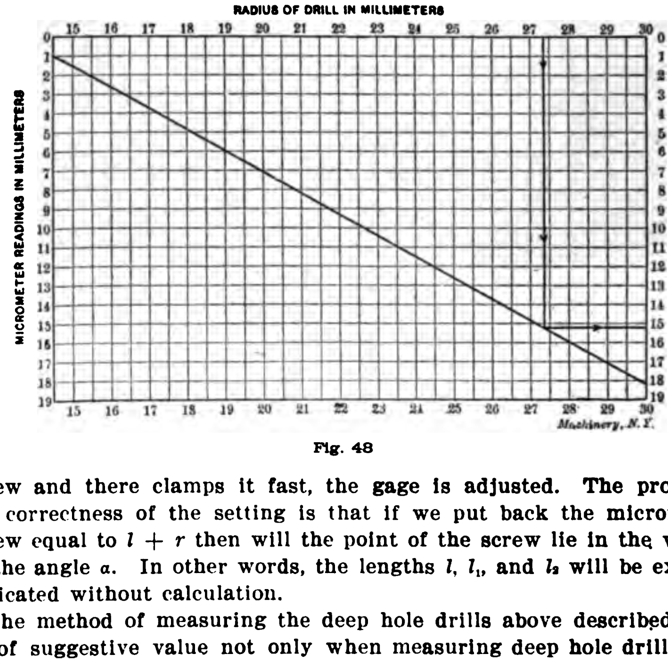

Fig. 48

screw and there clamps it fast, the gage is adjusted. The proof of the correctness of the setting is that if we put back the micrometer screw equal to $l + r$ then will the point of the screw lie in the vertex of the angle a. In other words, the lengths l, l_1, and l_2 will be exactly indicated without calculation.

The method of measuring the deep hole drills above described may be of suggestive value not only when measuring deep hole drills, but for many other occasions where it is not possible to take direct measurements with ordinary micrometers.

Another Type of Deep Hole Drill

The type of drill shown and described in the following differs in some essentials from the types already described. In the factory where it is used, a plain, everyday micrometer is used for gaging it, the end of the measuring point on the micrometer spindle being pointed a little less than the included angle of groove in drill, which is about 38 degrees, as shown in Fig. 36.

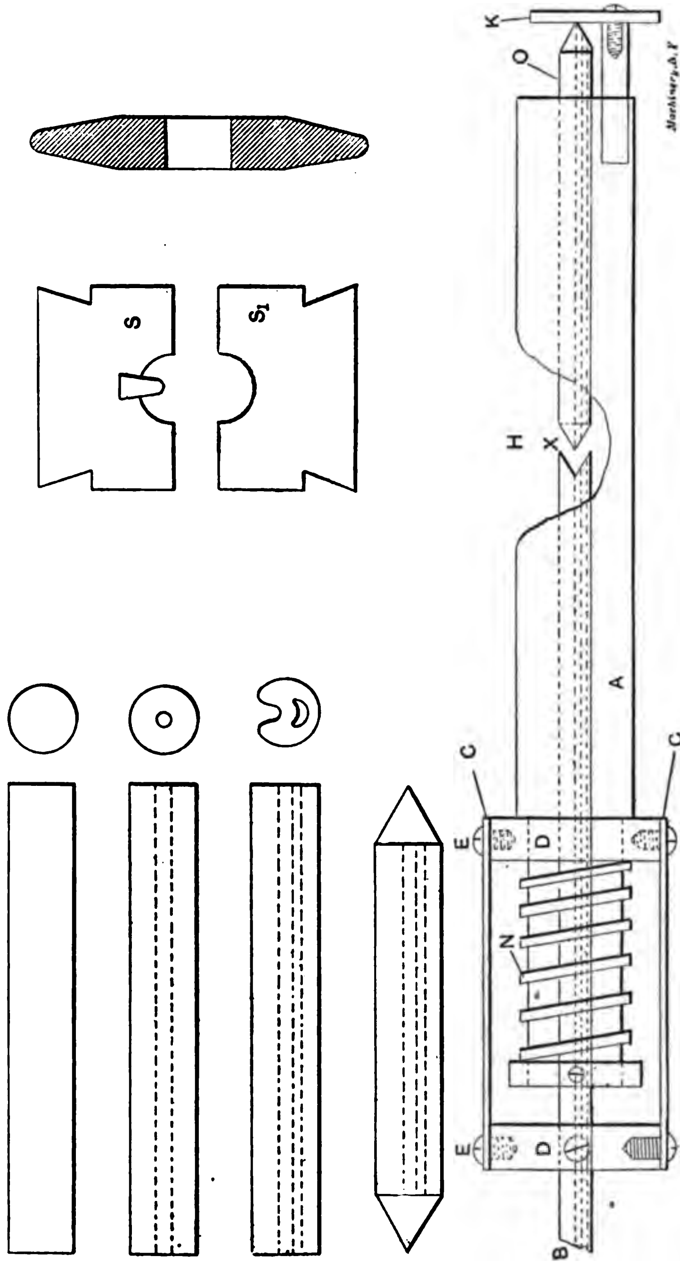

Fig. 49. Successive Steps in Making a Deep Hole Drill

In making these drills, half-inch Novo stock is cut into lengths of about 4 inches, Fig. 49; then a hole is drilled through the center; next the piece is heated and struck up in dies S S_1, which operation forms a groove and forces the center hole below the center into a crescent shape, as shown. It is then "pack annealed"; then both ends are pointed to 60 degrees, and turned in a lathe on female centers to within 0.010 inch of finished size, this being allowed for grinding. After being pointed, the work is held in a milling machine vise and the groove is finished with the cutter shown in the upper right-hand corner in Fig. 49. The groove is milled to a depth equal to one-half the finished diameter, plus 0.005 inch, minus the amount to be ground out of the groove, which varies according to the size of the drill and the grade of steel used. In short, the bottom of the groove must be exactly central when the drill is finished.

The drill is hardened two-thirds of its length, and the outside and groove are ground. It is now placed in a brazing fixture, as shown in the bottom view in Fig. 49. *A* is a body of suitable length with a hole drilled through it to accommodate all sizes of drills under ½-

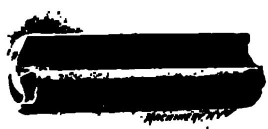

Fig. 50. Two Views of the Drill

inch. *B* is a steel tube a little under the size of the drill; a groove is rolled the whole length of *B* to conform with the shape of the groove in the drill. A V-groove is milled in the end of the tube to fit the back end of the drill. Collars *DD* are bored out, one a loose fit on *A*, the other on *B*.

The drill *O* is placed in the end of *A*, then *B* is inserted into the other end of *A*. The two are brought together at the opening *H*, which is milled out to allow point *X* to come in contact with the flame. A small wooden plug is fitted into *O* and *B* to prevent the brazing stopping up the oil-hole. A small piece of silver solder is placed between the two ends; then the swinging stop *K* is moved into place against the end of drill *O*. Collars *DD* are held together by strips *C* and screws *E*. Now *B* and *O* are brought together at *X* and are held firmly in place by the stop *K* and tension of spring *N*, *B* being held by *D* with the set-screw shown. With this rig, the brazing of the tube and drill is satisfactorily accomplished, the device insuring correct alignment.

This type of deep hole drill is far superior to the old drill with round oil hole, it being cheaper to make, at the same time as the crescent-shaped hole allows a greater flow of oil at the cutting edge.

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CONTENTS

CHAPTER I

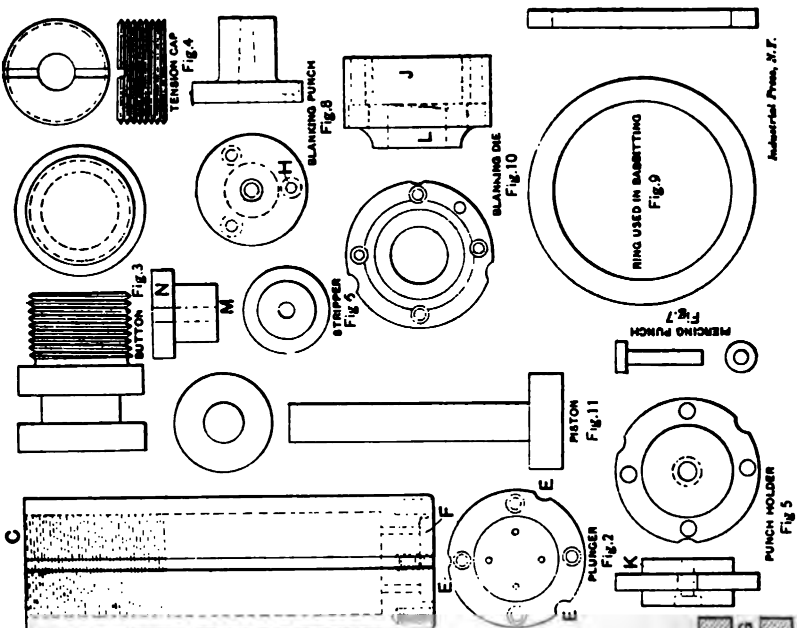
PRINCIPLES OF SUB-PRESS DIE CONSTRUCTION*

If we attempt to define the sub-press die, we find that we cannot define it as a special class of die, but merely as a principle on which all different classes of dies, cutting as well as shaping dies, may be constructed and worked. The sub-press principle is simply that the upper and lower portions of the die, the punch and die, are combined into one unit either by guide rods fastened into the lower part of the die, and extending through holes in the upper part, or by some other provision for guiding. This construction permits of a high degree of accuracy, eliminates the necessity of lining up the punch and die each time they are set upon the press, and thus saves a great deal of time and cost.

Owing to the large number of parts of which a sub-press die is composed its first cost is, of necessity, much higher than that of an ordinary die. When, however, we consider that a sub-die, when properly made, will run ten hours per day, for weeks at a time, without grinding, the first cost sinks to a minimum. In using an ordinary double die it is almost impossible to obtain two blanks that are exactly alike, one reason being that the stock to be punched is more or less wrinkled and does not lie flat on the face of the die. The consequence is, therefore, that after the piercing punches have perforated the wrinkled stock, and it is then flattened out, there is a greater distance between the holes than there is between the punches. Also, the pilot pins that are depended upon to locate the stock cannot do so exactly, since they are made a trifle smaller than the piercing punches in order to prevent them from pulling the blank up out of the die. On a certain class of work the double die answers all purposes, but when accuracy is required a sub-die is the only one that will give satisfaction.

In order to avoid a complicated drawing and to set forth the principles of the die in such a way that they may be readily understood by those not familiar with sub-dies, the die used for punching an ordinary washer has been selected for an illustration. The general principles of sub-dies are, of course, the same whether one or one hundred punches are employed. Having selected a frame with its proper cap, as shown in Fig. 1, of size suitable to the work, it is placed in a chuck, being held by the upper end, and, having faced off the bottom, the recess *A A* is bored to fit snugly the corresponding step on the base of the press. This base is finished on both top and bottom, and has a step, above referred to, turned to fit the bottom of the frame. A slot at *G* is cut in each end to receive the finger straps by means of which the frame is fastened to the face-plate of a lathe.

* MACHINERY, July, 1903.



Figs. 2 to 11. Details of Sub-press Die.

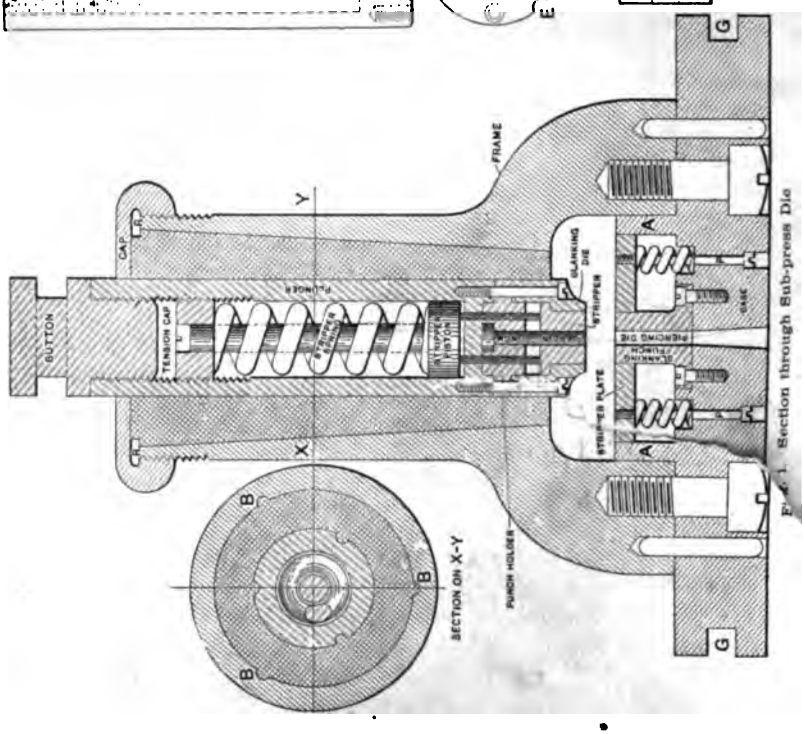


Fig. 1. Section through Sub-press Die

The center is recessed to receive the stripper plate and blanking punch, and a hole is drilled completely through to allow scrap punchings to fall to the floor. The base and frame are then fastened together by means of bolts and dowel pins as shown. Together they are clamped to the face-plate of the lathe, being centrally located by means of a plug center which fits the taper of the lathe spindle, and passes through the hole in the center of the base. In this position the frame is bored out to a taper of about one-half inch per foot. After boring, a splining tool is substituted for the boring tool, and with the lathe locked by means of the back gears, three or four grooves *B* are cut the entire length of the bore by sliding the carriage back and forth. At the same setting the upper end of the frame is faced off and threaded to receive the cap which is screwed on the frame. After the cap is in place, the hole for the plunger in this cap is bored out to the required size. This insures the hole in the cap being central with the inside of the frame.

The plunger, shown in detail in Fig. 2, is the next piece to receive consideration. After being centered and rough-turned, it is put in the center rest, and the hole *C* bored and threaded and fitted with the button shown in Fig 3. The internal thread in the plunger is carried down to a considerable depth in order to allow of the insertion of a tension cap, Fig. 4, by means of which a sufficient tension is placed upon the stripper spring to force the punching back into the stock upon the return stroke of the press. A dog is fastened to the button and the plunger turned to fit the hole in the cap, great care being exercised to keep the sides perfectly parallel. After turning, the lathe is blocked by the back gear, and three grooves *E* are splined, about 1/16 inch deep, for the entire length. It is essential that these grooves be parallel with the axis of the plunger. Before the plunger is completed, a ring, three-quarter inch wide, is made of machine steel and forced onto the lower end of it. The outside of this ring is trued up, using the plunger as an arbor, after which this end of the plunger is placed in the center rest, where the ring prevents it from being scored or injured by the center rest jaws. In this position the recess seat *F* is bored out to receive the punch holder shown in Fig. 5.

The punch holder is made, as are also the die stripper and punch, Figs. 6 and 7, by turning from a bar held in the chuck and finishing complete before cutting off. The recess which receives the head of the piercing punch should be bored at the same time to insure its being central with the rest of the die. The stripper, Fig. 6, should be made of tool steel and left large to allow for grinding after hardening, while the hole is bored sufficiently small to allow for lapping to exact size. The blanking punch, Fig. 8, which also contains the piercing die, is made of tool steel in the same manner, being finished complete before it is cut off, and it is left with sufficient stock to grind after it has been hardened. The holes *H* are drilled and counterbored for screws to hold the punch to the base.

After the parts are hardened, the blanking die is the first to be ground. It is gripped in a chuck, upper end outward, and the large hole *J* is ground out to fit the step *K* on the punch holder. Then the hole *L* is ground perfectly straight and of the same diameter as the master templet. The top face is also ground off, thus completing the die. In the stripper, the hole *M* is lapped to the same dimension as that in the templet. A round piece of cold-rolled steel is gripped in a lathe chuck and turned to fit nicely this hole in the stripper. Without disturbing the chuck, wring the stripper onto this arbor and grind the flange or shoulder *N* to fit nicely the larger bore, and the smaller diameter to fit the smaller bore, of the die. The blanking punch is finished in exactly the same manner as the stripper, being ground to fit the recessed seat in the base. The minor parts, such as the stripping plate, stripper piston, pins and springs, are then made, and the press is ready for assembling.

In assembling, first force the punch holder, Fig. 5, into the seat *F* of the plunger, and then force the die onto the holder; transfer the holes in the die through the holder and into the plunger, and after they are drilled and tapped, fasten the parts together as shown in the sectional view, Fig. 1. Remove the die and drill four holes in the punch holder and plunger for the stripper pins *O*. Place the stripper piston in the plunger, above this the spring, and lastly screw the tension cap into place. The stripper pins *O*, which are hardened for their entire length, are placed in their holes in the punch holder, and the stripper placed in the die, which is then secured in its place on the punch holder.

The blanking punch is placed in its seat in the base and securely fastened by cap screws, after which the springs shown are placed in position and the stripper plate drawn down by means of the screws *P*, until it is a trifle below the top of the blanking punch. The frame is now ready to be babbitted. Screw the button onto the plunger, and with a piece of oily cloth wipe the plunger all over, then sprinkle flake graphite onto it. The oil on the plunger will cause the graphite to adhere, and after the surplus has been blown away a thin coating will be left over the entire surface. The plunger is lowered inside of the frame until the blanking punch enters the die. In the cap insert the ring shown in Fig. 9, to prevent the babbitt from flowing into the recess *R*, and screw the cap onto the frame. As the cap is an exact fit for the plunger, it therefore aligns it with the frame and with the blanking punch. The grooves on the plunger must be plugged with putty where they pass through the cap in order to prevent the escape of the babbitt while pouring. A pair of parallels, of a height equal to the projection of the button beyond the top of the cap, are now placed on the bench, and the die inverted upon them. Great care should be taken to avoid any vibration during pouring, as very little will affect the alignment of the plunger. Before pouring, heat the frame with a torch or jet of gas, and when the babbitt has attained the proper heat, which is a very dark red, pour it in from both sides of the die simultaneously. Allow it to remain until thoroughly cool, then remove the plunger, strap the frame to the face-plate of a lathe, and cut a spiral oil groove the entire length of the babbitt.

As the blanking punch has already been ground, the next step is to grind the faces of the blanking die, piercing punch, and stripper, while all are in their proper positions in the plunger. They should be ground so that the face of the stripper, die, and punch are all flush with each otner. After grinding, the parts should be taken from the plunger and thoroughly cleaned so that no emery can possibly remain in the working parts. Oil all of the running parts in a thorough manner, then put them together in their proper positions, and replace the plunger in the frame. In setting up a sub-press die, care should be taken to have the punch come to the face of the die only, and not enter it.

CHAPTER II

CONSTRUCTION AND USE OF SUB-PRESS DIES*

The sub-press die is an old device dating back at least one, and possibly two generations, and having its origin in watch and clock factories where its ability to perform blanking operations of the most delicate nature was early recognized and fully appreciated. That this tool, though familiar in the field just mentioned, has yet capabilities in other directions which have not hitherto been fully recognized, is the impression that must be strongly borne upon an appreciative mechanic who is acquainted with the work being done in the shops of the Sloan & Chace Manufacturing Co., of Newark, N. J. This firm has for many years built precision machinery for watch makers, fine tool makers and others, whose work requires great accuracy. The tools described in the following were constructed by this firm.

A section of a typical blanking sub-press of the common cylindrical type is shown in Fig. 13. It is doubtless familiar to most toolmakers, so will need but a few words of description. To base *B* is screwed and dowelled the cylinder *A* lined with babbitt, as shown at *C*, this lining being provided with ribs which engage corresponding grooves in plunger *D* which works up and down within the babbitt lining under the action of the ram of the press in which it is used. Nut *U* furnishes an adjustment for tightening the babbitt lining to take up all slack due to wear, as fast as it is developed. The die is usually the upper member, while the punch is placed in the base. *K* is the die, screwed and dowelled to plunger *D*; accurately fitting the opening in this die is the shedder *H*, which is normally forced downward with its face flush with the face of the die by the action of spring *M*, which acts through the piston *N* and pins *O*. A similar construction is used in the bottom member. *J* is the punch, screwed and dowelled to the base. *L* is the stripper, surrounding the punch and accurately fitting it, and held firmly at the upper extremity of its movement by the pressure

* MACHINERY, December, 1906.

of the springs *Q*; it is restrained with its face flush with that of the punch by the heads on stripper screws *R*. Thus it will be seen that the faces of the punch with its stripper and the die with its shedder may be ground off smooth and flush with each other, presenting to the eye the appearance of two solid plates of metal, the division between the fixed and spring-supported members not being visible if the fitting has been well done.

With this construction in mind, the details of the punch and die shown in Fig. 14 will be readily understood. Similar letters in each

Fig. 12. Examples of Sub-press Work

case refer to similar parts, but only the members of the device actually working on the metal are here shown. The outline of the punching which is to be made will be understood from the outline of the punch and its stripper, as shown in the plan view. There are two small holes, *c c*, and one larger hole, *b*, in the blank. For punching these small holes, in addition to the simple arrangement shown in Fig. 13, openings are necessary in the punch, and small piercing punches have to be placed within the aperture of the die, passing through holes in the shedder; the holes in the punch are continued through the base of the sub-press so that the waste material drops through beneath the machine. The piercing punches in the upper member are held to die pad *G* by holding screws *g* which draw these parts up into their tapered seats against the shoulders formed on them for the purpose. The fitting at all the cutting edges is done with great accuracy. The punch *J* fits die *K* very closely; the shredder *H* is fitted to the die very closely; the stripper *L* is fitted to the punch, and small punches *f* are accurately aligned and closely sized to their corresponding openings in the face of main punch *J*. Disappearing pins are shown at *h h*; they are used to guide the strip of stock, and are pressed down by the

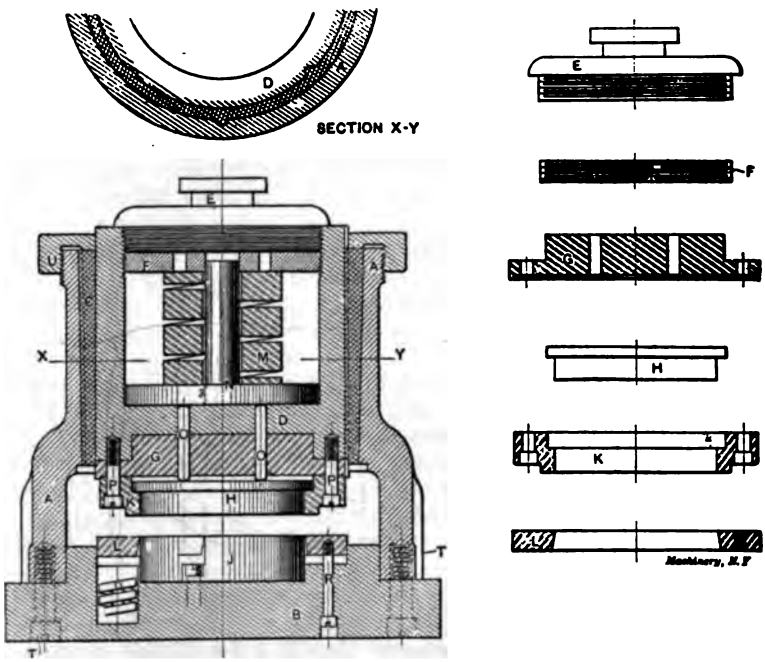


Fig. 18. Construction of Typical Sub-press

descent of die *K*, returning under the action of their springs as the ram ascends.

It will be understod, of course, that the sub-press is a complete unit, with punch and die and ram guiding surface always in place, so that no setting is necessary. The operator only needs to place the sub-press on the bed of the punch-press, insert the button on cap *E* in the holder provided for it in the face of the ram of the machine, and strap the base of the tool to the bed of the machine. He is then ready to commence work at once without any need of wasting time in matching up his dies, it only being necessary to adjust the length of the stroke to the proper amount. This is one of the advantages of the sub-press. Another one will be immediately recognized upon considering the action of the parts on the strip metal from which the blank is punched. With the work in place, die K, and with it small punches f descend, the latter passing through the stock until they almost meet the corresponding cutting edges in the lower member. As soon as shedder H strikes the stock its motion is arrested, and it remains behind until the blank is separated, being meanwhile powerfully pressed upon the work by spring M. As the stock, while being sheared, is pressed down around the blank, it carries with it stripper L which also, by the influence of springs Q, exerts a heavy pressure on the stock. The whole area of metal being thus firmly held between plane surfaces, there is no danger of buckling or distortion of the stock as would otherwise be likely. As the ram moves upward again the blank is still firmly held on the stationary top of punch J by the shedder H. The stock, however, is carried upward with die K by stripper L, forcing the stock back over the punching again until the movement of the stripper is arrested by the heads of screws R, at the time when the face of the stock is flush with the top of the punching. The work is thus pushed back into the stock in the same position that it occupied before it was severed from it and in many materials when the work has been nicely done, it is difficult at a careless glance to believe that anything has been done at all, both sides presenting a flush smooth surface where the parting occurred.

This condition is taken advantage of oftentimes in clock manufacturing. Gear blanks, for instance, are punched out from strips of metal and inserted back in their places again, minus, of course, the stock which has been punched out to form the arms and the hole for the "staff" or little shaft on which it is mounted. These strips, thus prepared, are then taken to machines where the staffs are inserted and fastened, it being much easier to handle the little wheels in this way than if they were severed and handled in bulk. A strip of stock thus treated is shown in the photograph reproduced in Fig. 12, the second one from the right at the bottom of the group; five of the pieces are shown in place in the stock while three have been pushed out. Besides the advantages of permanent setting of the punch and 'ie and the holding of the stock to prevent distortion, which allows ery narrow bridges of material to be left between wide openings, he suitability of the device for delicate work such as the piercing of mall holes in thick stock will be appreciated by reference to Fig. 14. t will be noted that, no matter how small punches e and f may be, o portion of their projecting ends is at any time left unsupported aterally by shedder H or by the work. The shedder, pressing down firmly on the work, supports the end of the punch at the point where the pressure is applied. It is thus possible to use a very much more slender punch for a given thickness of stock than can be used in any —y. In Fig. 12, where a number of samples of sub-press work the topmost piece with the rack teeth in it, which is about ick, has at its left-hand end four 0.025-inch holes pierced will be seen that the thickness of the stock in this diameter of the hole punched. Such a ratio has ken with ordinary punches and dies, although the writer does not remember ever having seen the ratio of 1.5 to 1 exceeded; and in that case the hole in the die was considerably larger than the hole in the punch with the result that the pierced hole was very much tapered, the scrap coming out in the form of a conical plug. In the die under discussion, however, no allowance of this kind is made, the hole in the die being a very close fit to the punch, with the result that the hole pierced in the blank is as nearly a perfect

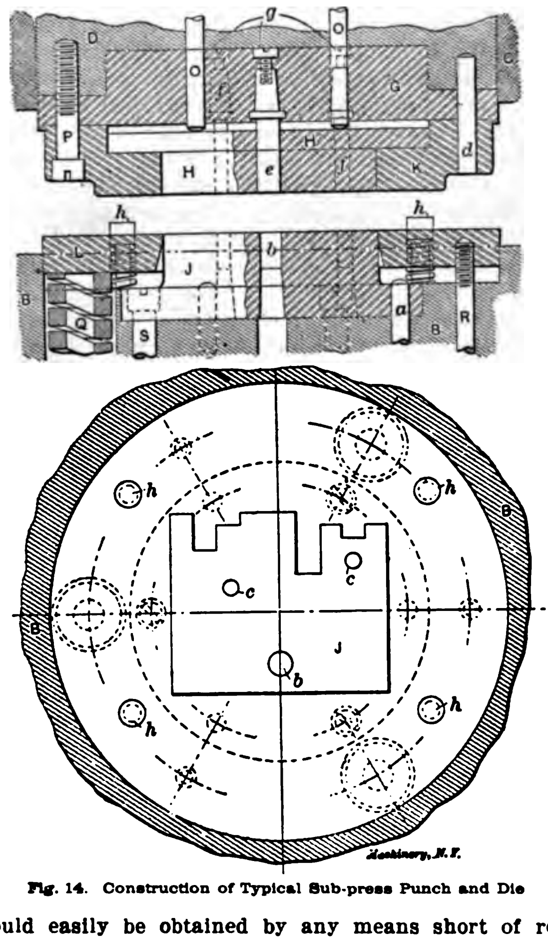


Fig. 14. Construction of Typical Sub-press Punch and Die

one as could easily be obtained by any means short of reaming or grinding.

Another advantage of the sub-press, dependent in part on the accuracy of alignment provided, and the corresponding accuracy in fitting which can be given to the cutting edges, is that the work is remarkably free from fins and burrs. A consideration of the action of the press will show that there is practically no chance for burrs to form in a piece even where they would in an ordinary blanking die.

It is, of course, necessary for the die to descend until the punch has all but entered it, if clean work is to be produced. There appears to be a slight difference in the practice of different operators in this respect, although this difference in practice would be expressed in the dimensions of only 0.002 or 0.003 inch, perhaps. Some of them adjust the stroke so that the die does not quite meet the punch. Others prefer to have them meet and even enter by an infinitesimal amount.

Attention has already been called to two of the samples of work shown in Fig. 12. The small parts there illustrated are within the ordinary range of the sub-press as ordinarily used, but it is safe to say that there are many die-makers who consider themselves familiar with this tool who have yet to see dies built on this principle large enough to blank out such a piece as the largest one shown, which is quite 14 inches square. Nor is this the limit possible. The writer saw here dies of this type being made for heavy armature work, blanking out armature segments measuring possibly as high as 26 or 28 inches across extreme dimensions. The same advantages that obtain in the smaller presses, result from the use of the larger ones. There is a saving of time in setting up the tools; there is a possibility of punching small holes in thick stock or of leaving narrow bridges of metal between openings of considerable area; the dies, owing to their accurate and permanent alignment, may be fitted to each other much more closely, produce work that requires less finishing and comes more nearly to dimensions than can be done in any other way. At the same time, the construction effects a great increase in the life of the die, making it unnecessary to grind it anywhere nearly so often as would otherwise be the case. The only disadvantage that can be set off against these advantages is the increased cost, and it appears to be conceded that even with this consideration the balance is strongly in favor of the sub-press die.

Of course the larger sizes of these tools are not made in the familiar circular form illustrated in Fig. 13. Fig. 15 shows three different styles. The one at the rear has the sliding head guided by four vertical posts carefully ground and lapped to fit cast-iron bushings. This is the construction used on heavy work. At the left is shown one in which the plunger is rectangular in shape. This works in a bearing lined with babbitt the same as the cylindrical form shown at the right of the cut and outlined in Fig. 13, although the bearing is not adjustable. The cylindrical form is used for the smaller sizes.

The making of a sub-press die requires all the skill of a first-class toolmaker. The method pursued by at least some of the men who are engaged in this work at the factory mentioned is about as follows: Taking the dies in Figs. 13 and 14 as examples, the base *B* and cylinder *A* are machined and fitted together according to methods that would naturally be pursued by any good mechanic. The inner surface of the cylinder is grooved so that the babbitt may be securely locked in place. Plunger *D* is then machined, and the outer surface ground and fluted with semi-circular grooves. Especial pains are taken to have these grooves parallel with the axis of the plunger in both planes; if this is not done the die may be given a slight twisting movement instead of the perfectly straight forward one that is required, since upon these grooves depends the angular location of the punch and die with relation to each other. The plunger is now inserted within the cylinder and, with proper precaution, the space between them is filled with babbitt which flows into the grooves in the cylinder and those in the plunger as well, locking with one and guiding the other. After being cooled, the plunger is pumped up and down to insure a perfect bearing, and the nut *U* is screwed down until all slack is taken up. Die *K* is now made to accurately fit the templet or model furnished the tool-maker as a sample. After it has been completed, it is hardened and fastened in place. Then the model is

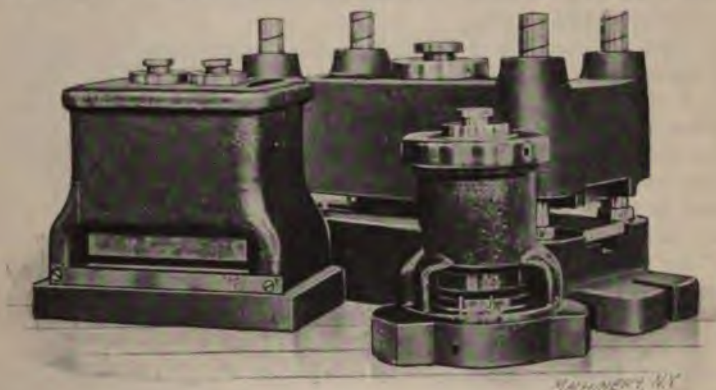


Fig. 15. Three Forms of Sub-presses

inserted within it, and such holes as may be called for in the blank are transferred to die pad *G*. This is done by punches with outside diameters ground to fit the holes in the templet, and provided with sharp points concentric with the outside. The pad after being thus prick-punched, is put on the face-plate, the slight punch marks are carefully indicated, and holes are carefully bored to a taper to fit the punches which are to be inserted in them. The punches are finished by grinding on centers after they are hardened. They are supported at the shank by a male center, while the opposite end is temporarily ground to a point which revolves in a female center in the other end of the grinder. The punch may thus be ground all over with the assurance that the pointed end is true with the exterior—a necessary provision as will appear later.

It might be noted here that no draft is given to any of the cutting edges of these tools, since they do not enter each other, at least not to any appreciable extent, and since the stock in entering and leaving the cutting edges is positively moved, no clearance is necessary, and the die cuts practically the same kind of a blank at the end of its life that it did at its birth. Shedder *H* is fitted to die *K* and the holes for the punches are transferred to it in the same way as for the die pad, by means of carefully machined prick punches which fit the holes in the models, these prick punch marks being afterward indicated to run true on the face-plate. The punch is now worked out a very slight amount larger in all its outlines than the die. The model is laid upon it, the holes transferred to it as in the case of the other parts, these holes being then indicated and bored out, but not ground in this case, being left three or four thousandths inch smaller in diameter than finished size. The punch is fastened in place in the base, lining up as nearly as possible with the die. The ram is forced downward in a screw press until the punch enters the die very slightly, cutting a thin chip from its sides to bring them to the shape required. The punch is then worked down to this point all around and again entered in the die a short distance further, the operation being repeated until the two parts fit perfectly.

In finishing the holes in the punch, after the hardening process, plugs are driven into each as shown in Fig. 16. The punches *f*, Fig. 14, still with their ends pointed concentric with their outside surfaces, are fastened in position in the upper member, and the ram is brought down until these punches mark slight centers in the top of the brass plugs, when the ram is again raised and the punch *J* removed. The punch is then strapped to the face-plate and each of the small plugs is in turn indicated from the prick punch marks, when it is removed and the hole is ground to size with a steel lap charged with diamond dust in an internal grinding fixture. The stripper is fitted to the punch in the usual manner. With the parts thus made and fitted great accuracy is obtainable.

A die of the four-posted type is detailed in Figs. 20 and 21, Fig. 21 showing the lower member or punch, while Fig. 20 shows the upper member or die. This sub-press is used in making the piece with rack teeth shown in the upper right-hand corner of Fig. 12. A slightly different method of procedure is followed in this case than with the sub-press just described. The punch and die are finished before the upper and lower members are lined up with each other. When the time comes for doing this the punch is entered in the die, the two parts being parallel with each other as to their faces, when bushings *A* are slipped over the posts until they rest in the bottom of the cast counterbores in die holder *D*, Fig. 20. This counterbored space has had pockets gouged out in the sides for the babbitt to flow into and lock with. The grooves shown in the posts in Fig. 15 are not yet cut in Fig. 21, they being still smooth and true as the grinding left them. The space *C* being poured full of babbitt and allowed to cool, the punch and die are permanently aligned with each other without possibility of shifting. The posts are then removed and the spiral grooves for oil distribution are cut in them.

…e of the noticeable points about this die, as shown in Fig. 20, …h the work is so closely fitted in the tool itself that the eye …able to distinguish the construction, is the fact that the cutting edge which shears out the rack teeth is built

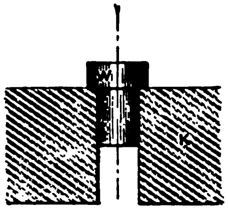


Fig. 16. Plug for Centering Holes for Grinding

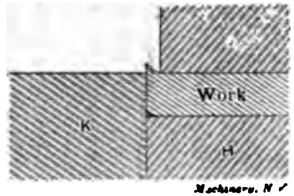


Fig. 17. Action of Badly fitting Punch and Die

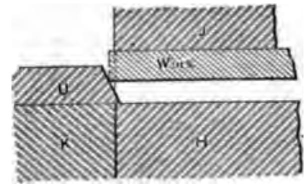


Fig. 18. "Nest" with work in Place

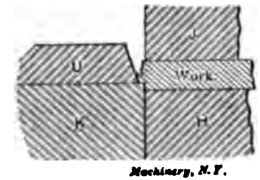


Fig. 19. Work being Trimmed in Shaving Die

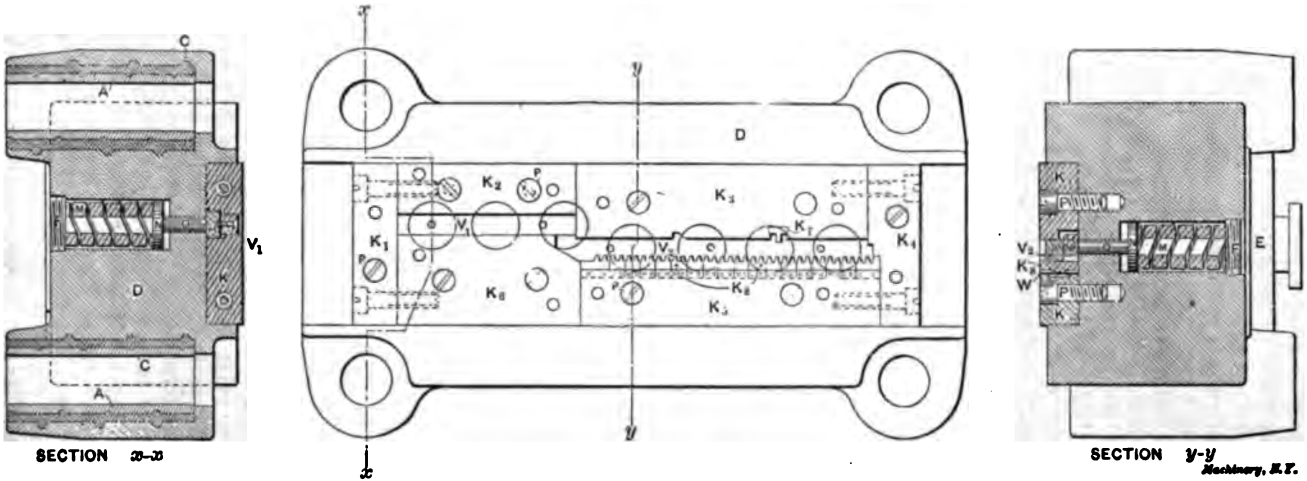


Fig. 20. Plan View and Sections of Upper Member, or Die, of Sub-press shown in Fig. 22

up of small segments, each containing two teeth only, these segments being dovetailed into the larger piece, K_5. Each of these small pieces, K_8, is secured by two dowels which pass through from side to side of K_5, locking the parts firmly together. This costly and difficult construction was necessitated by the demand for accuracy in the spacing of the teeth. With the sectional construction shown the parts are not affected sensibly in the hardening. That piece K_5 may not be warped out of shape, it is ground to size in all its surfaces, top, bottom, sides and even in the dovetail, so that when completed its plane surfaces are straight and parallel. The dovetail of the die sections K_8 are next machined to fit this and inserted, being then spaced the proper distance apart. The holes in K_5 are then continued to pieces K_8, which are taken out and hardened, and returned to be doweled in place. It will be seen that this die is constructed on the sectional plan throughout. This makes it possible to finish on the surface grinder most of the cutting edges. Troubles due to distortion in hardening are thus entirely avoided. The proper end measurements between vital points in the model are also preserved by leaving a slight amount of stock where two sections of the die come together, the parts being ground away at this point until the proper dimensions are obtained.

In the few cases where the grinding wheel will not finish the cutting surface, extended use is made of diamond laps, these being in the form of steel sections of proper contour to fit the part of the die they are working in, these steel pieces being charged with diamond dust and reciprocated vertically in filing machines, of which a large number are used in this shop. The little dovetail in which part K_7 is inserted, for instance, was finished in this way. The back of the dovetail is perpendicular but the two sides slope somewhat from the vertical, forming a wedge-shaped opening enlarged toward the rear. Section K_7 is then driven in from the rear, finished off, and ground with its front face flush with the rest of the die. In Fig. 22, which shows this sub-press, this little section has not yet been finished off, so that it is seen to project above the remaining part of the die.

This is the first operation, the tools used being the blanking punch and die. The pieces produced are afterward subjected to the action of a shaving die, the original blanks being left with 0.002 or 0.003 inch stock for the purpose, which is trimmed off in the last operation. The punch for this first or blanking die has the rack section subdivided into four parts only, which are matched up carefully with the sectional die just described. In the shaving die, however, this punch is built in sectional form as described above for the blanking die, so that great refinement in measurements is secured.

The sub-press just described is that shown at the back of Fig. 15, and opened up in Fig. 22. Its action is exactly identical with the smaller one just described; it has all its advantages and presents the same deceptive appearance of perfectly homogeneous surfaces in the punch and die when completed. In the illustration, Fig. 22, the

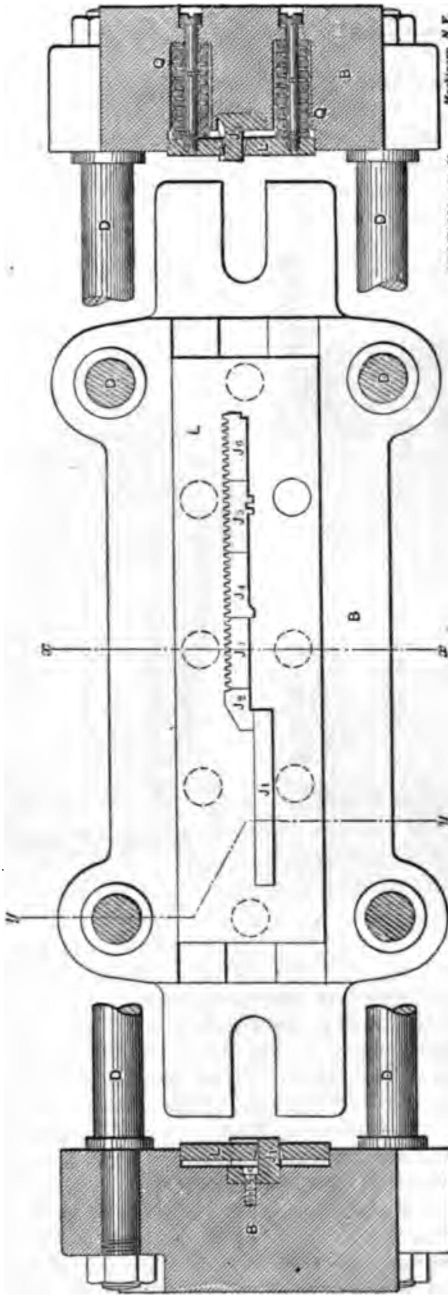


Fig. 21. Plan View and Sections of Lower Member, or Punch, of Sub-press Die shown in Fig. 22

shedder and stripper springs have been slacked up in order to show the outlines of the cutting edges, but this is not the normal condition.

A feature of the shaving die system, to which reference has been made, is the use of a "nest" to locate the work. In this trimming operation the punch is in the upper member and the die in the lower one. On the surface of the die, of which an example is shown in Fig. 23, are placed steel guiding plates, U_1 and U_2, which form the nest referred to. They have their edges shaped to the outline of the

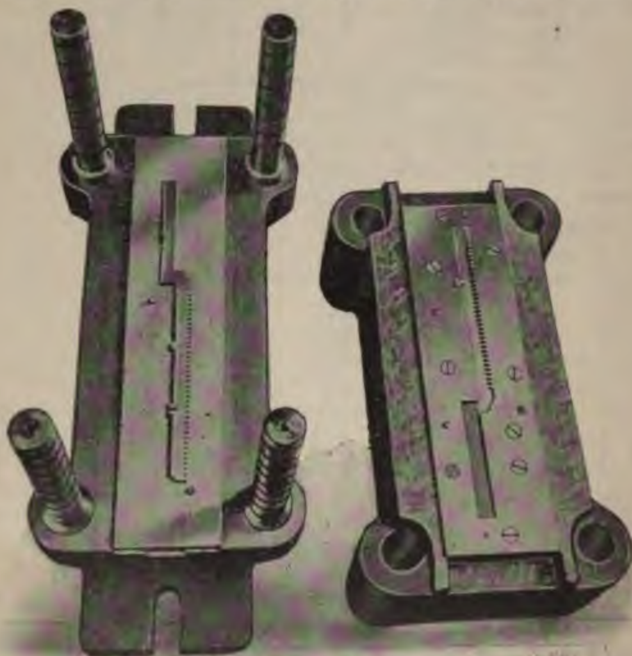


Fig. 22. An Instructive Example of Sub-press Construction

piece to be operated upon and they are pressed inward by flat springs *W* at the outer edge, being allowed a slight lateral movement although retained from sidewise displacement by shoulder screws *V*. The holes through which these screws pass are slotted to permit this; the end of the slot limits the inward movement of the plate. As shown in the enlarged views, Figs. 18 and 19, the inner edges of these plates are beveled backward so as to form a recess in which the work may be located. The descent of the punch forces the plates out, which, as they are displaced, still guide the work so that it is properly centered over the die. These beveled edges of the plates have the further advantage of curling the chip out of the way where it does not clog the tool and may be easily cleaned off. The shedder coming up from below and removing the work, closes the lower opening effectively so that the whole device is chip tight.

Even greater accuracy is advisable in the fitting of the punch and die in this shaving sub-press than is necessary in that used for

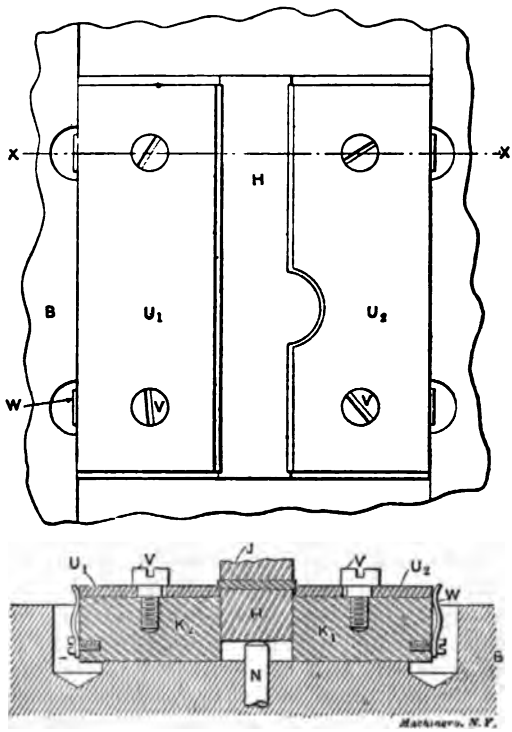


Fig. 23. Shaving Die with "Nest" for Locating the Work

blanking only, if it is desired to produce clean work free from burrs. The necessity for this will be appreciated upon examining Fig. 17, which shows in magnified form the action of the cutting edges. If the punch does not match up closely with the edge of die *K*, the stock is bent upward, leaving a sharp burr, while the punch impresses the outline of its cutting edge on the top surface of the blank.

soft and is located a few thousandths inch more than the thickness of the punching below the top of the die. When the die is sharpened, the stripper is ground off the same amount. No springs are used with the stripper, it being actuated by two 1-inch studs fastened with screws on the stripper. These studs pass through the die and holder, and are actuated by a bar fastened to the gate of the press, thereby forcing out the punchings from the die. The six punches *N*, Fig. 26,

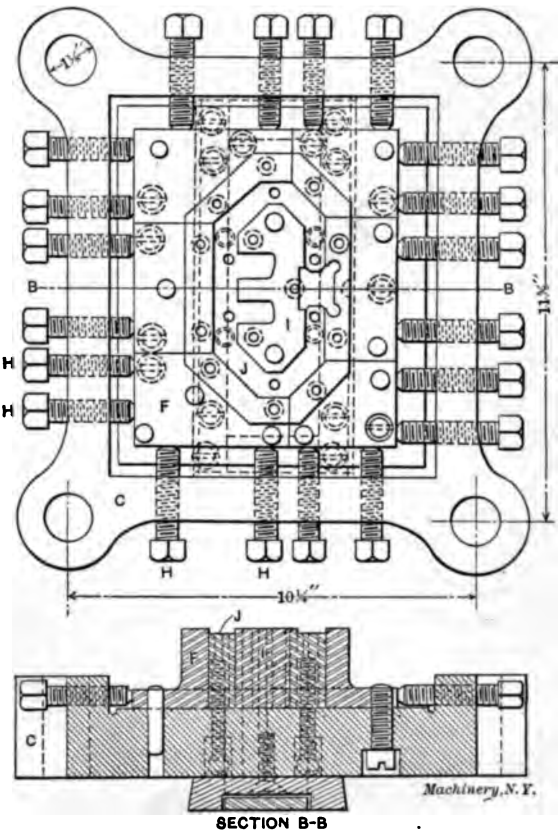


Fig. 24. Sub-press Die for Irregular Blank

are upset, as shown, at the end where they are inserted in the holder, while the other end is hardened, straightened, and lapped to size. The holes for the punches are located after the die is finished and assembled.

The cast-iron punch holder *K*, shown in Fig. 25, is planed on top and bottom and across the four bosses. The four sub-press pins *D* are of tool steel, hardened as far as the head, ground to a light driving fit on the head end, and ground to a sliding fit in the die holder on the other end. The holes for these pins were located so as to

Punch and Die for Armature Disks*

A compound punch and die for the armature disks for the cores for electric motors, having many interesting features, are shown in plan views and also in cross sections in Figs. 27 and 28. The die-holder *A*, Fig. 21, is of cast iron, and is first planed on the bottom. It is then strapped to the face-plate of a lathe, and faced and bored to re-

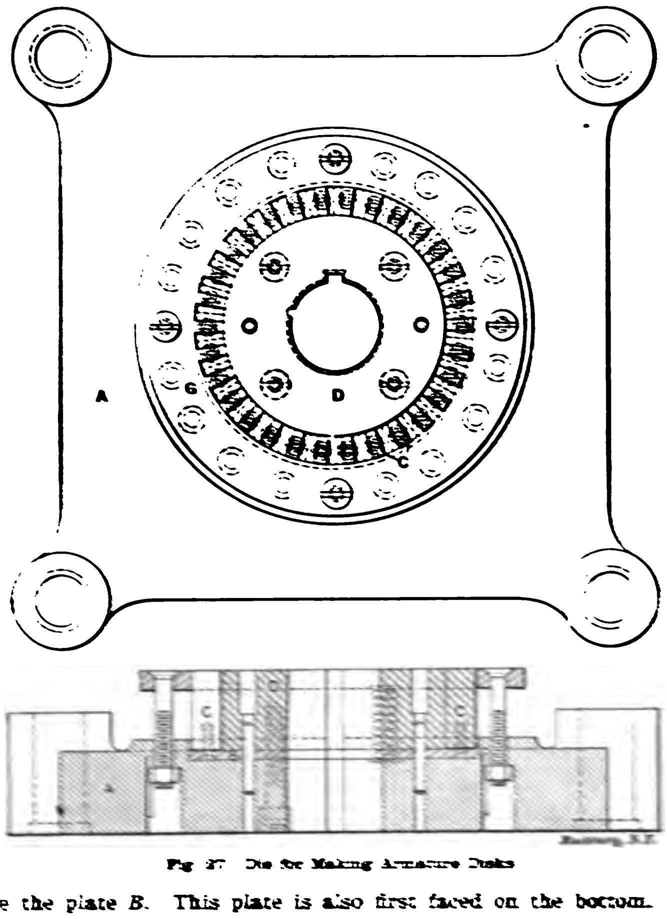


Fig. 27. Die for Making Armature Disks

ceive the plate *B*. This plate is also first faced on the bottom. It is then turned over and bored to the outside diameter of the disk to be punched; the depth of the bored hole is about $\frac{1}{4}$ inch. The die sections *C* are all milled in a fixture; they are then drilled and tapped for $\frac{1}{4}$-inch flat-headed screws. After this, the sections are hardened and tempered. The plate *B* and the sections *C* are then assembled, and after being assembled the sections are ground. The guide ring *D* is now machined. The keyway and marking notch are tapered one-half degree for clearance; the large hole for the shaft in the armature tapers also one-half degree. The ring is drilled and tapped for four ½-inch screws, and drilled and reamed for two dowel pins. After this the ring is hardened, tempered, and ground to a very close fit in the circle formed by the sections *C*. The center hole is also ground to the required dimensions. The stripper is now made,

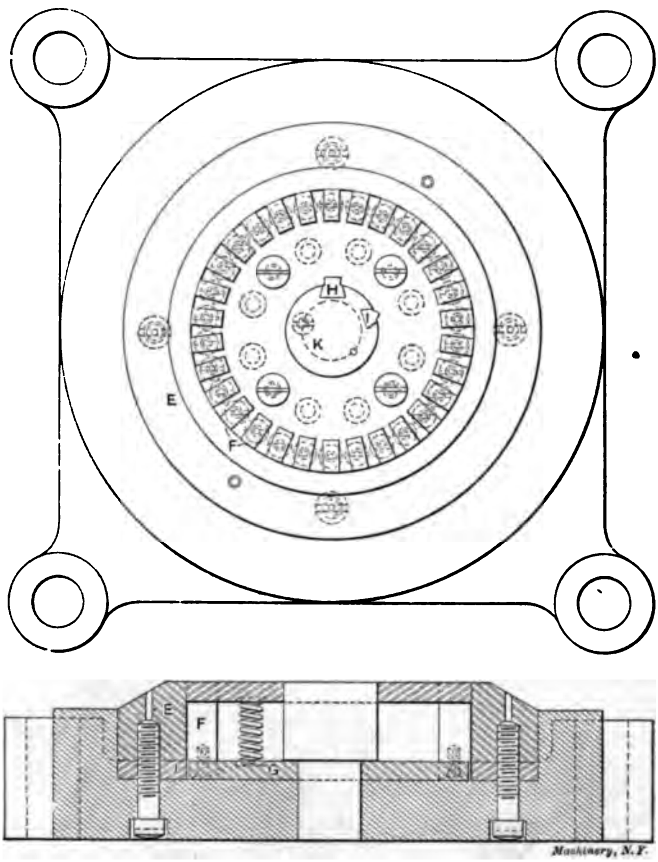


Fig. 28. Punch for Armature Disks

the working of which is plainly seen from the engraving and the whole is assembled, and the die is ready.

We are now ready to proceed with the punch. In this, *E*, Fig. 28, represents a tool steel ring, which, after it is machined, drilled and tapped, is hardened and ground. Punch sections *F* are located in the plate *G* which is milled with the proper number of slots. The punch sections should be left a little softer than the die, because the punch and die will wear much longer and give much better results if this is the case. The sections *F* are held in place by a ring *J* which is shrunk on the outside at the bottom. At *H* and *I* are shown the punches for the keyway and the marking notch. These are fitted into the center punch, being dovetailed into this. They taper from the bottom up, when the punch is in working position, and are driven in so that when punch *K* is assembled they cannot work out. The holes for the sub-press pins are drilled and reamed with the punch and die together, and the holes in the die counterbored to a depth of

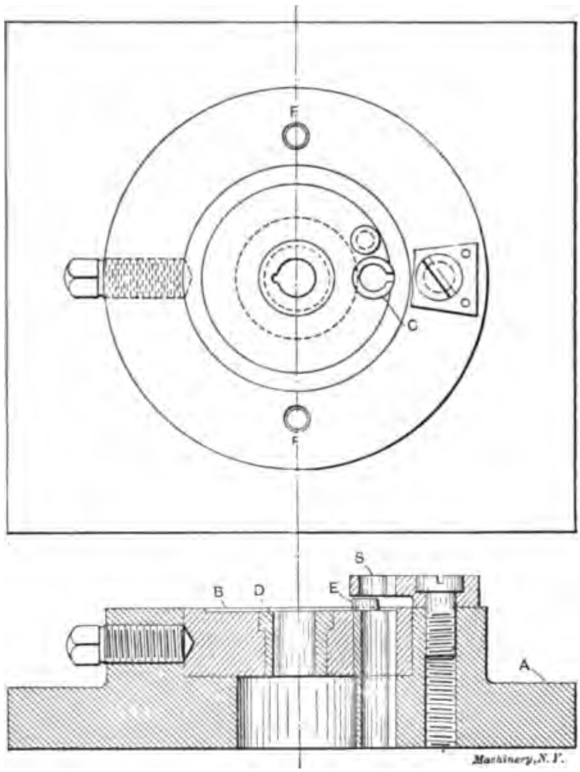


Fig. 29. Die for Armature Laminations used during Experimental Stage

about 3/8 inch. The pins should be a driving fit in the die and a working fit in the punch.

Tools for Making Armature Laminations*

The engravings, Figs. 29, 30, 31, and 32, illustrate the method of producing the armature lamination, Fig. 33, of a motor, during the experimental, and, later on, the manufacturing stage. In the first case the cost of tools is considered, and in the second, the manufacturing cost. In Fig. 29, *A* is a die holder for holding round dies.

* MACHINERY, August, 1907.

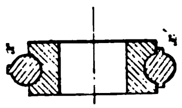


Fig. 32

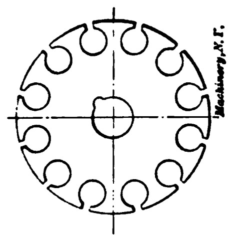


Fig 33

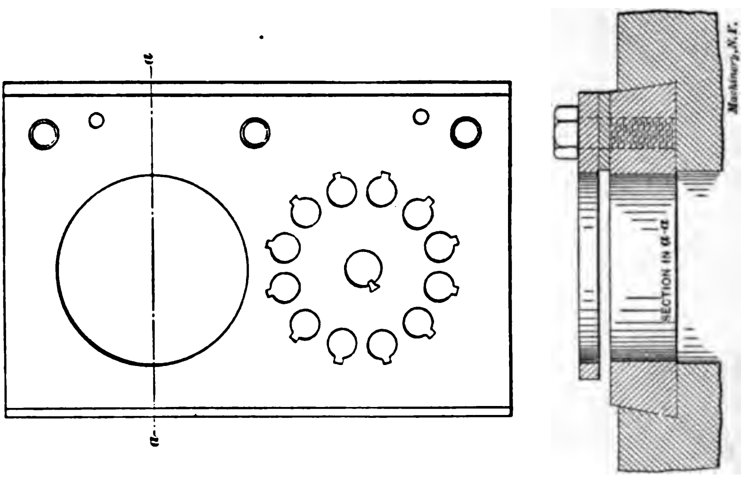


Fig. 31. Die used with Punch in Fig. 30

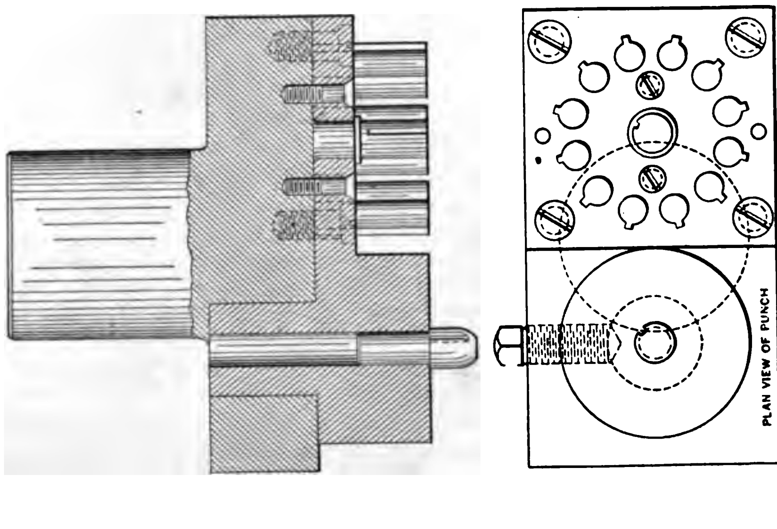


Fig. 30. Punch for Manufacturing Laminations

These holders were made for holding ordinary blanking dies, and instead of fastening the stripper to the die, it is fastened to the bolster or holder. The first operation is the punching of the blank, the second the punching of the slots. This is done with the die, Fig. 29. The pilot or index pin *E* is removed, and one slot is punched in the blank. After this operation is completed, the pin is replaced and the rest of the slots are punched, the pilot or index pin being located so as to index correctly. The die holder *B* is made from machine steel and

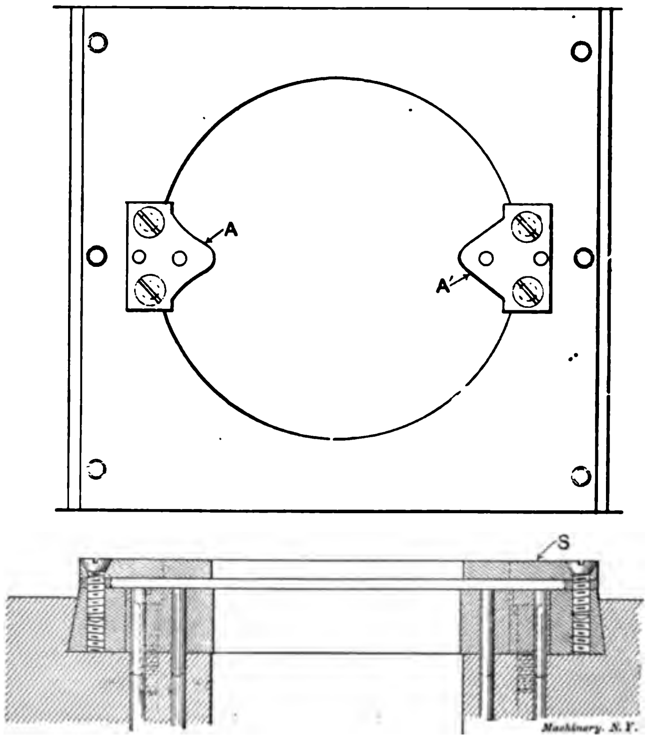


Fig. 34. Die for Outside of Blank shown in Fig. 33

recessed to allow the blank to fit properly; the die proper, *C*, is sweated fast in its place so as to avoid any chance of shifting its position. The die *D* is used for the last operation, the punching of the center hole for the shaft. The stripper *S* is removed when punching this hole, and another is fastened at *F F*. This latter is, of course, removed when punching the slots. The pilot pin *E* is also used in the last operation for locating the keyway properly in the blank. The cost of these punches and die was small, but the manufacturing cost would come high if used to produce large quantities.

As enough laminations were wanted to warrant a more expensive punch and die, and the manufacturing had to be cheapened, the design shown in Figs. 30 and 31 was adopted. These illustrations need no further explanation as to the operation of the tool. It is readily seen that a complete lamination is obtained at each stroke of the press. A special milling cutter was made to mill the punches. Fig. 32 illustrates the method of milling the punches as well as the broach for sizing the holes in the die. First both sides are milled, as shown at x', leaving a key at both sides of the punch or broach. Then one of the keys is milled off as shown at x. A small section is

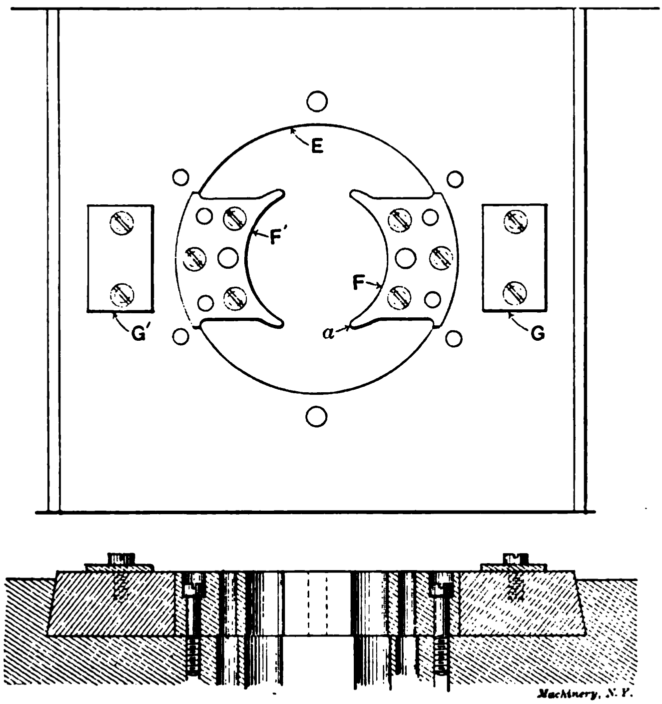


Fig. 35. Die for Inside Shape and Holes of Blank in Fig. 38

inserted at the center hole of the die, leaving a solid key in each blank instead of the keyway in the experimental lamination shown in Fig. 33.

Sectional Punches and Dies*

The punches and dies in Figs. 34, 35, 36, and 37 were made for producing the punching Fig. 38, in two operations, and illustrate to some extent sectional die making. As a perfect punching was required in regard to the inside and outside diameters, the design shown was adopted, which proved to be all that could be desired as to accuracy and cost of making, particularly when compared to previous methods and results. In making the punch and die for the first operation,

* Machinery, May, 1907.

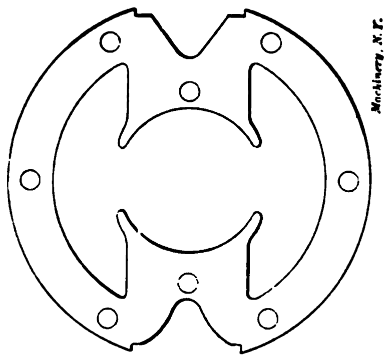


Fig. 38. Finished Punching

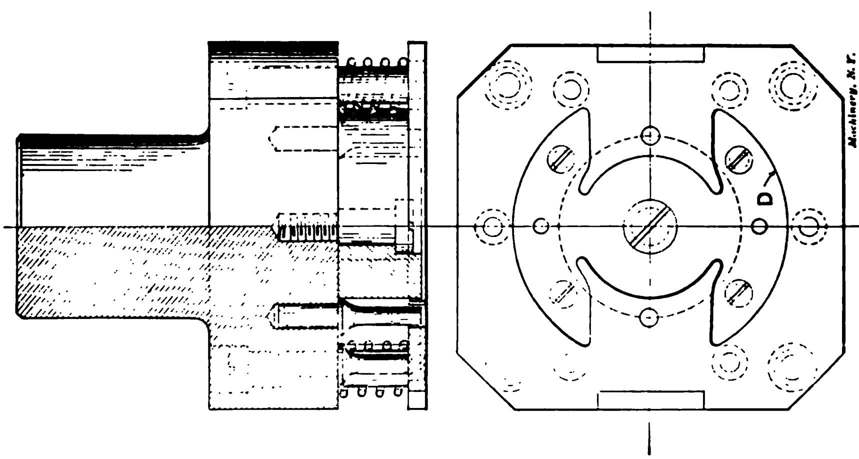


Fig. 37. Punch for Inside Shape and Holes

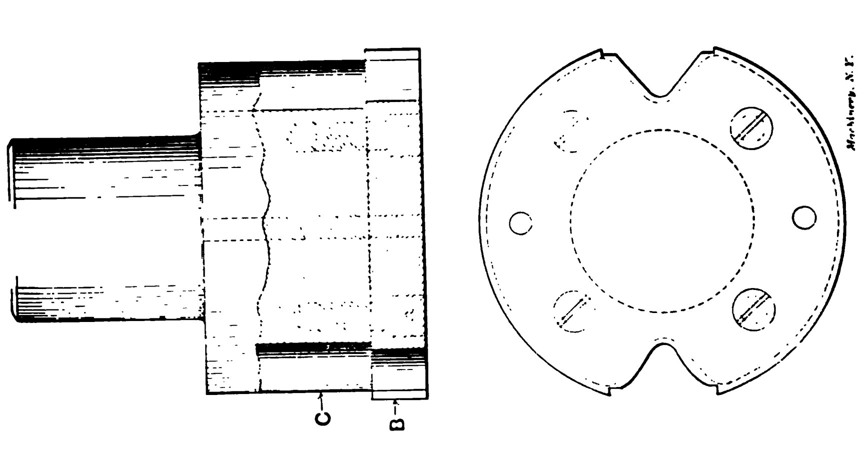


Fig. 36. Punch for Punching Outside of Blank

Figs. 34 and 36, the punch was made first. *B* is the punch proper, and *C* is the holder which is made of cast iron. The punch was hardened and screwed and doweled to the holder before grinding the outside diameter to the correct size. Then the die was machined, and after hardening ground to fit the diameter of the punch. The sections *A* and *A'* were then fitted to the die and fastened with the screws and dowel pins as shown, and sheared by the punch. As the sections *A* and *A'* were small, they did not alter any in hardening.

For the second operation, the punch and die, Figs. 35 and 37, were made in the same way, that is, the punch was hardened and ground on the diameter *D*, and the die ground at *E* to fit diameter *D*. The sections *F* and *F'* were then machined in the proper way and sheared by the punch. In hardening the sections *F* and *F'*, one of them altered so much at *a* that it had to be discarded and another made. This could have happened had the die been made solid, which would have condemned the whole die, and a new die would have had to be made. The rest of the design is readily understood by referring to engravings Fig. 34, where *S* is the stripper, and Fig. 37, where the stripper is on the punch, the blank being placed on the die guided by strips *G G'*.

Hardening Small Blanking Dies*

It is manifestly an unprofitable investment to equip a manufacturing establishment with all the latest and most approved facilities for hardening, if there are only a few pieces to be treated occasionally. In cases of this kind the diemaker must make the best of the apparatus on hand.

If a blacksmith's forge is used, let the die be placed in the fire with the cutting face upward. During the period of heating, keep the fresh coal away from the die by surrounding it on all sides and on top with red-hot cinder coal. When turning on the blast, be careful not to give it too much air. The more sparingly the blower is used, the better are the chances of the steel becoming evenly heated throughout. Turn it into the fire for about a minute, then shut it off and let the heat *soak* into the die instead of *blowing* it in. This is probably the most important point. The block of steel must be evenly saturated with heat and kept from contact with cold air until it reaches the proper hardening temperature. Remember to blow a little and then stop the air while the steel absorbs the heat. While the die is being heated, prepare a pail of clean water, taking the chill from it, that is, heating it until lukewarm. The die, held in one hand with tongs, is then plunged into the water and kept moving all the time; when the die is cool enough, take hold of it with the other hand and stir the water with it until both water and die arrive at the same degree of heat. Now instead of taking the die out of the water and reheating it over the fire or letting it cool in the air, just let the water and steel cool off together.

* H. J. Bachmann, MACHINERY, March, 1910.

CHAPTER IV

DRAWING AND FORMING DIES

Only those who have made a specialty of drawing sheet metal know just how to proceed to lay out a die so that the desired result will be a certainty the first time it is tried. First, we are confronted with finding the diameter of the blank. There are several methods by which we can determine the size of the blank. One way is to cut out a blank of the same thickness as the stock of which the model shell is made, and then keep reducing the diameter until the blank balances the model. Another way is to multiply the circumference

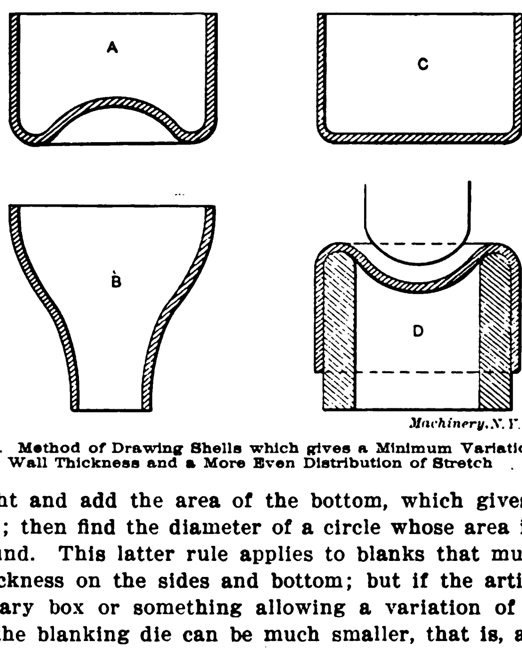
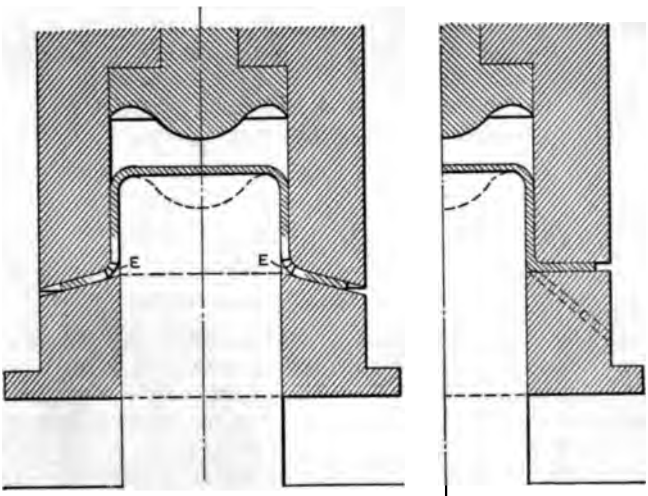


Fig. 39. Method of Drawing Shells which gives a Minimum Variation in Wall Thickness and a More Even Distribution of Stretch

by the height and add the area of the bottom, which gives the area of the blank; then find the diameter of a circle whose area is equal to the area found. This latter rule applies to blanks that must be of a uniform thickness on the sides and bottom; but if the article should be an ordinary box or something allowing a variation of thickness, the size of the blanking die can be much smaller, that is, a shell $1\frac{1}{2}$ inch long can be drawn from a blank 3 inches in diameter, or from a 2-inch blank. The sides and bottom will, of course, be thinner in the latter case. A good rule to follow is to make the height of the shell at first draw, one-third the diameter of the blank, that is, a 3-inch blank, for best results, should produce a shell 1 inch long at first draw. The shells should be annealed if drawn to any length.

Fig. 39 shows a novel method when the blank is to be drawn into a ·ell of great length or into a shape as shown at *B*. The first operation ves the shell as shown at *A*, and this will save at least one drawing operation. It will be noted that the indented end in the shell bottom presents surplus stock, so to speak, whereas, if it is drawn at the first operation in the ordinary way as shown at *C*, the succeeding operations must be gentle to prevent a greater reduction in the thickness of the walls on the angular sides of the shell. Another advantage in shaping the first shell as shown at *A* is that when drawing it to the angular shape, the stretch or draw of metal is more evenly distributed over the entire surface and the strain on the metal is not as great as if the draw commenced at the corner. Another novel feature is to shape the blank as at *A* (if the shell is to be a long one) and then use a die for the second operation, as illustrated at *D*. The blank, when forced through, is actually turned inside out. This op-



Machinery, N.Y.

Figs. 40 and 41. Diagrammatical View illustrating the Stretch of the Metal and the Effect of Angular Punch and Stripper Faces

eration presents two good points. First, we are enabled to get a suitable amount of stock in the shell so distributed as to draw to the best advantage. Second, if the shell is to be a long one, the diameter of the blank must of necessity be large, therefore the stock is distorted considerably in changing from a flat blank to a shell of much smaller diameter, and if we could see the grain of the stock as it passes over the die, it would present an appearance similar to that shown by the lines *E* in Fig. 40. That is, the inside of the stock stretches and the outside compresses. Therefore, by turning the shell inside out, the stock is apparently restored somewhat to its original texture of close grain, which will allow further stretching.

A very important point which must not be overlooked is the angle on the face of the stripper and blanking punch. In making long, small diameter shells, if the angle on the stripper and blanking punch is, say 8 degrees, the shells will break out at the corner; but by changing the angle to 10 or 12 degrees, they will come out without breaking. This point can be better understood by referring to the half section, Fig. 41. It would be almost impossible to draw the shell when the edges of the blank were at right angles to the shell, but by making the angle of the punch and stripper as shown exaggerated by the dotted lines, the shell is practically drawn by the blanking punch, and leaves very little work for the drawing punch. The above suggestions were contributed to the August, 1908, issue of MACHINERY by Mr. Frank E. Shailor.

Dies for Making Tin Nozzles*

In the following pages is described and shown a set of dies for the production of nozzles for tin cans of large sizes used to ship liquids. The dies are of the combination type used in single action presses, and perform from one to three operations at one stroke of the press. From 12,000 to 15,000 pieces of finished work can be turned out per day from these dies according to the speed of the operator.

The first die, Fig. 43, is composed of eight principal parts: *A* is a gray iron bolster plate made to separate at the line *a-b* so the die can

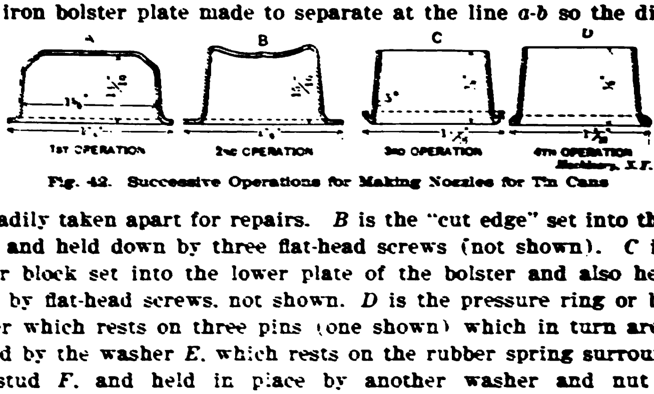


Fig. 42. Successive Operations for Making Nozzles for Tin Cans

be readily taken apart for repairs. *B* is the "cut edge" set into the top plate and held down by three flat-head screws (not shown). *C* is the center block set into the lower plate of the bolster and also held in place by flat-head screws, not shown. *D* is the pressure ring or blank-holder which rests on three pins (one shown) which in turn are supported by the washer *E*, which rests on the rubber spring surrounding the stud *F*, and held in place by another washer and nut (not shown) with which to regulate the pressure while drawing the shell. *G* is the punch and drawing die combined, the outside diameter of which is fitted to the cut-edge *B*. The inside diameter equals the center block *C* plus twice the thickness of metal. *H* is a forming pad made to fit the top of the center block *C*. It forms the top of the shell at the end of the stroke and also serves as a knock-out for the shells.

In operation the tools are set into an inclined press. The punch coming in contact with the cut-edge *B* cuts the blank, which is held by the pressure ring *D* against the end of the punch *G*, but as punch *G* continues down, the blank is drawn over the center block *C*; and, as the punch ascends, the stem *I* in the top of the punch shank comes into contact with a bar in the press pushing the pad *H* down, and the shell represented at *A*, Fig. 42, slides off back of the press.

Fig. 45 shows the second operation or redrawing tools. *A* is the bolster plate. *B* is the drawing ring, supported by pins and a rubber

* MACHINERY, May, 1905.

spring, the same as in Fig. 43. The center block in this die is tapered and the punch *F* is also bored out tapering to fit it. The pad in punch *F* is of peculiar shape, as will be noticed, and will be explained later. The shell is placed on the drawing ring, and the punch, as it descends, draws it down and compresses it to the shape of the center block *C*. The shell is knocked out on the up-stroke of the stem *H*, the same as in the first operation, and the drawn piece looks like *B*, Fig. 42.

Fig. 44 shows the tools for the third operation, which really consists of three operations. *A* is the bolster plate of the die; *B*, the trim-

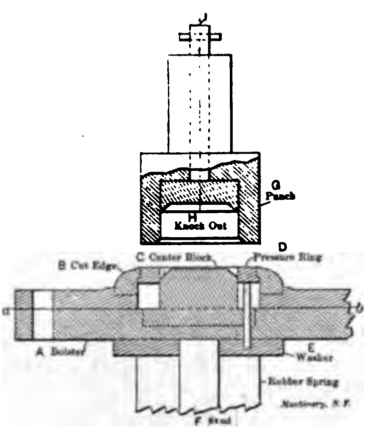


Fig. 43. First Die for Tin Can Nozzles

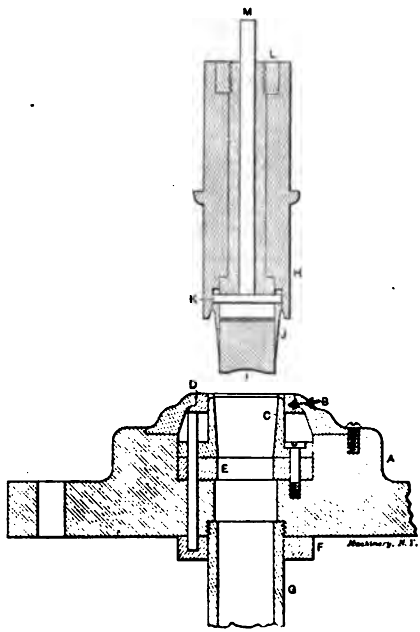


Fig. 44. Tool for Third Operation

ming die; *C*, the center block; *D*, the drawing ring; *E*, the lower die; *F*, washer; *G*, tube through which the bottom of the nozzle passes after being punched out. These bottoms are used for roofing shells for fastening tar paper in place on roofs, etc., so that in the process really two articles are made at once. These tools are used in an inclined press. As the punch comes down, punch *I* cuts out the bottom, and at the same time punch *H* trims the lap edge; as it continues downward it presses the shell over the edge of the center block *C*. As the punch ascends the knock-out bar comes in contact with the pin *M*, carrying the stripper *J* down by the cross-pin *K* and ejecting the nozzle in the shape of *C*, Fig. 42.

Fig. 46 represents the tools for the fourth and finishing operation. They consist of a simple punch and die, yet much depends on these tools, for the nozzles all have to be of an exact size on the finished edge to receive a sealing cap, and this cap when closed on must be watertight. The die consists of a bolster-plate *A* and a die-block *B*, made of tool steel, hardened and tempered. The punch is also hardened and tempered and ground out to gage. The tools are set in the press, and the nozzle is slipped on the die-block. The punch in coming down passes over the work until the edge turned up on *C*, Fig. 42, comes in contact with the shoulder *F* on the inside of the punch. As the punch continues downward, this edge is curled over and pressed down to the

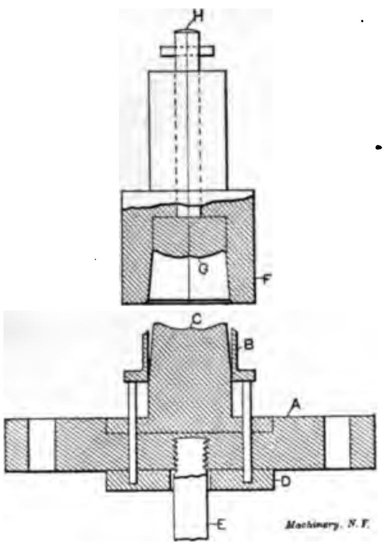


Fig. 45. Redrawing Tool for Second Operation

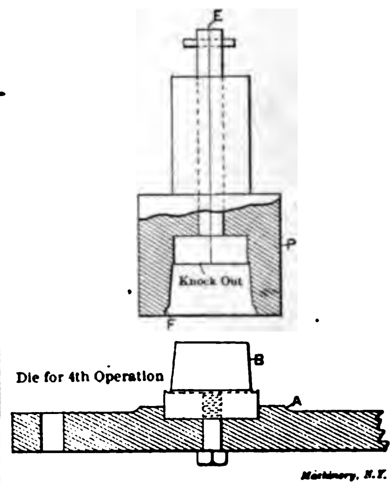


Fig. 46. Tools for the Finishing Operation

shape of *D*, Fig. 42. As the punch rises the shell is knocked out by the knock-out stem the same as in all the other dies.

Punch and Die for Blanking and Forming Copper Cups*

The die in Fig. 47 is designed to blank and form up a copper cup or capsule used in the manufacture of balance wheels for watches. The copper strip is fed into the press, which then blanks out and draws the metal into the shape shown at *R*, at the same time punching the center hole. Referring to the illustration, *A* is the base of the sub-press, *B* the body, *C* the cap, and *D* the plunger, all these being of cast iron machined to size. The body and base are held together by two screws *E* after the usual well-known manner. *F* is the buffer plug which receives the thrust of the press piston, and *G* is the babbitt lining of the body *B*. *H* is the outside diameter die, held in place by four screws and two dowel pins. *H'* is the outside diameter punch, also held in place by four screws and two dowels. *I* is the die for cutting out the center hole, and *J* is the punch for this hole. *H'* and *I*

* MACHINERY, April, 1908.

also serve as forming dies in bringing the metal to the proper shape. *K* and *L* are shedders, supported by four push-pins, those of the former resting upon springs whose tension is controlled by short threaded plugs, as shown, and those for the latter abutting against the piston *M*, which is in turn pressed down by the large spring *N*, the tension of which is controlled by the plug *O*. The block *P* is used merely to hold the punch *J* firmly in place.

The operation of the die is as follows: The press ram being at the top stroke, the copper strip is fed in across the top of *H*, and as the

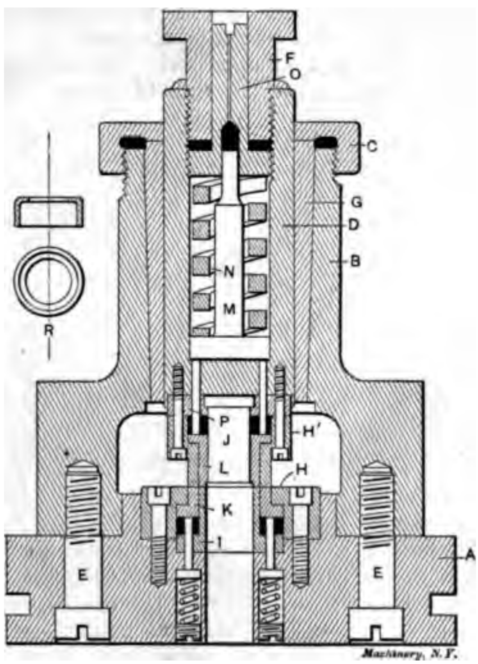


Fig. 47. Punch and Die for Blanking, Piercing, and Forming the Copper Capsule shown at R

ram descends, the blank is cut from the strip by the punch *H'* and drawn to a cup shape between the inside edge of *H'* and the outside edge of *I*. Simultaneously, the center hole is punched by *J* and *I*. As will be sen by referring to the illustration, *J* is made a trifle short so that the drawing operation will have begun before this hole is punched. This prevents any distortion of the piece by the punch *J*. Some little trouble has been experienced with this tool at first, on account of the air in the hollow plunger *D* forming a cushion when it was compressed by the rising of the piston *M*, thus preventing the proper working of the die. This was finally obviated by making a small groove at the side of the piston where it worked in the plug *O*, and drilling a vent hole through *O* as shown. This allowed free communication to the atmosphere, and from then on the die gave complete satisfaction. The variation in size among the cups, or capsules, as they are called, is never more than 0.001 of an inch either in diameter or in length.

Punches and Dies for Drawing an Odd-shaped Brass Cup*

The set of punches and dies described in the following paragraphs, while not exceptionally out of the ordinary, may have one or two points that will be of interest to any one engaged in this class of work. Figs. 48 to 51 show the progressive operations from start to finish by which the piece shown in Fig. 51 is produced. This is a corrugated, conical cup, drawn from a round blank of soft brass 3/32 inch thick and 2 5/16 inches in diameter. The corrugations project externally, and internally form six equally spaced, square grooves

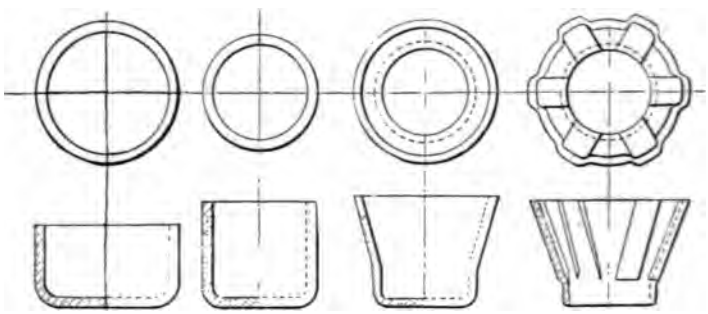


Fig. 48 Fig 49 Fig. 50 Fig. 51

which converge radially and disappear within about 3/16 inch of the bottom. The specifications in this case require that there shall be developed on the outside a distinct shoulder at the base of the conical part and that this shoulder be formed on the corrugations only. On the inside no shoulder is to be visible, but the formed grooves are made to disappear uninterruptedly near the bottom. It will be seen that the pressure in the last drawing has to be much greater than ordinary, in order to accomplish these results, for the stock at the point where the straight and conical portions met was pressed out quite thin in order to develop the shoulder.

The dies and the shanks of all of the punches used are made of a uniform size so that two holders, one for the punches and one for the dies, are all that are required. The punches are secured by a set-screw, and the dies are seated in the holder and held fast by four set-screws, equally spaced around the side, with their points set into the circumference of the dies. To make the change from one operation to the other it is only necessary to loosen the screws, remove the tools in use, and substitute the ones next in the set. Each drawing operation is followed by a careful annealing in order to insure the equal flow of the metal, and to minimize any possibility of cracking.

* C. H. Rowe, MACHINERY, August, 1903.

The main problem is to make the finishing punch as cheaply as possible, and have it stand up under the severe work required of it. It was first made in sections by turning it to shape, and then dovetailed to receive the elevated pieces, which were made separately and forced into the dovetails up against a shoulder. Then the end was drilled and tapped, and the straight tip screwed on. The parts, having been carefully fitted, were marked, removed and hardened, and then replaced as before. The punch made in this way had been used but for a short time when the elevated pieces began to chip and crack where the strain was the greatest, and often one hundred or more cups would be run through before the defect was noticed. For this

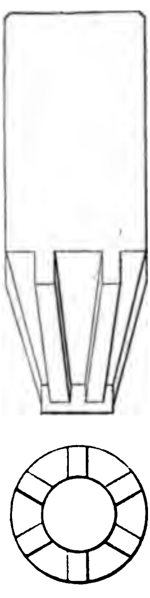


Fig. 52

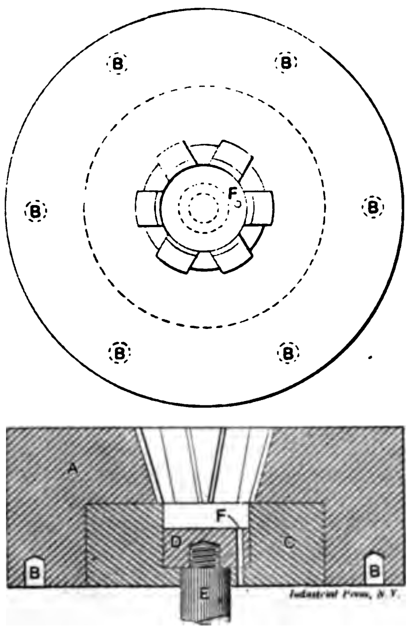


Fig. 53

reason the hardened sections were replaced by soft ones, but after a short run they would flatten down and make continual repairing necessary. To overcome the trouble the punch was finally made as shown in Fig. 52, from a solid piece of tool steel. This was milled, chipped, and filed to a finish, and after hardening was drawn to a light straw color. After this no more trouble was experienced.

The first and second drawing dies were made from tool steel disks, 5 inches in diameter and 2 inches thick, with holes to draw the cups as shown in Figs. 48 and 49. These dies were of the plain push-through type with a sharp edge on the under side to strip the work from the punch, which was made slightly tapering and of a diameter equal to the inside of the cup. In this case the punch was made less than the hole in the die an amount equal to two thicknesses of the

CHAPTER V

DIE-BEDS OR BOLSTERS FOR PRESSES*

The subject of die-beds or bolsters is one of considerable importance, and is deserving of greater attention than it often receives in the shop or designing room. It has been the experience of the writer that many of the troubles encountered in the use of press tools are due to these parts being badly designed or poorly constructed. Many a fine die has been ruined because it has not been properly secured in the die-bed and consequently has shifted while in operation; or because the holes in the die-bed through which the blanks or punchings are supposed to pass have not been made large enough to allow them

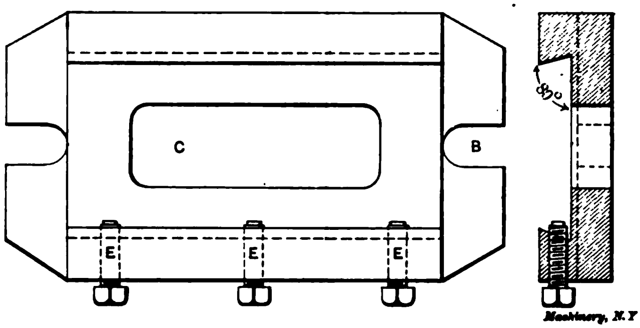


Fig. 54. Die-bed of the Style commonly used in Jobbing Shops

to pass through freely. As a consequence the blanks get jammed in the die-bed and pile up into the die itself and are compressed by the pounding of the punch, until the punch or die breaks from the strain. The principal functions of a die-bed are: first, that of supplying an adequate support for the die, and a holder to hold the die in its proper position to be engaged by the punch; and, second, to furnish a means of attachment to the press. Two of these principal points to be considered therefore in the design and construction of a die-bed are first, the method of securing the die, and second, the method of securing the die-bed to the press. Due consideration, of course, should also be given to proportion and strength.

In Fig. 54 we have an illustration of a die-bed of the type generally found in the jobbing shop. The dovetail method or holding the die, with set-screws *E* to lock it in proper position, is employed. It is fitted with a flange on each end with slots *B* to receive the clamping bolts which pass through them into the press bolster. In the center is a rectangular cored hole to let the punchings pass through. This style of die-bed is cheap and convenient for use where several dies are

* MACHINERY, January, 1910.

to be used in one die-bed. The dies can be easily slid into place and fastened by means of the set-screws, and are easily removed when another die is to be used. This bed has the following disadvantages: first, that of being held by set-screws which have always a tendency to jar loose in punch press work, and second, the cored holes *C*, being necessarily made large to accommodate various shapes of blanks, weaken the bed and lessen the support to each of the dies. It is always better, if possible, to have a separate die-bed for each die.

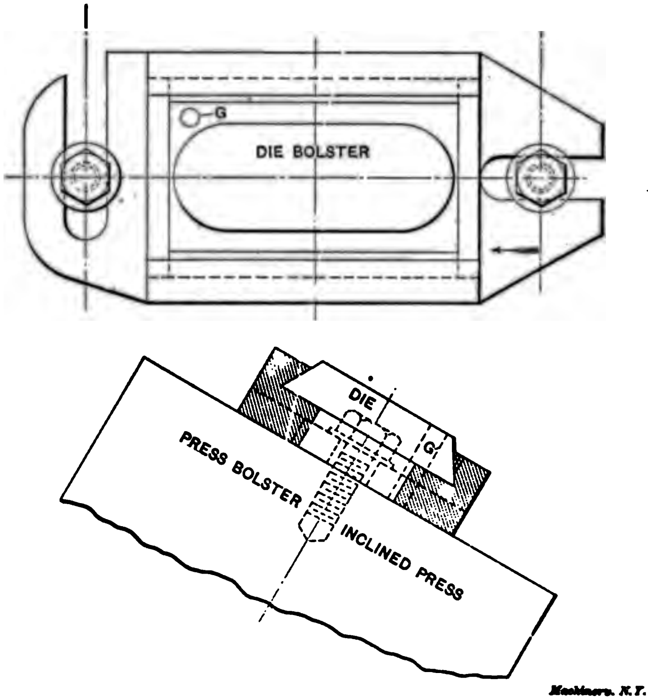


Fig. 55. Die-bed adapted to Inclined Presses

In Fig. 55 we have a bed for use on an inclined press. In this bolster the dovetail method of holding the die is used, but without the use of set-screws. The dovetailed opening to receive the die is slightly tapered and the die is driven into place with a copper mallet, and is then made doubly secure by the insertion of a dowel which is driven through the die into the die-bed. The dowel is shown at *G*. The method of clamping this bed to the press bolster is different from that shown in Fig. 54 in that the bolt slot in one flange runs at right angles to that in the opposite flange. By having the slots in this position the die-bed may be attached or removed without the necessity of taking out the bolts, thus not only saving a great deal of time and trouble in setting the tools, but also preventing the bolt holes from getting filled with scrap or dirt and the bolts from getting lost. This is an excellent die-bed for blanking and piercing work.

An improved type of die-bed for general utility is shown in Fig. 56. In this bed the dovetail method of holding the die is used. In the illustration it will be noticed that there are four parallel pieces or gibs *E* placed along the sides of the die. The object of this is to provide for dies of various sizes. When a larger die is to be used one or more of these gibs may be taken out. This bolster, in addition to four bolt slots, has a flange *B* all around it so that it may be clamped in any position. The set-screws *H* which hold the die in place should be provided with a lock-nut as shown at *I* to lessen the chances of

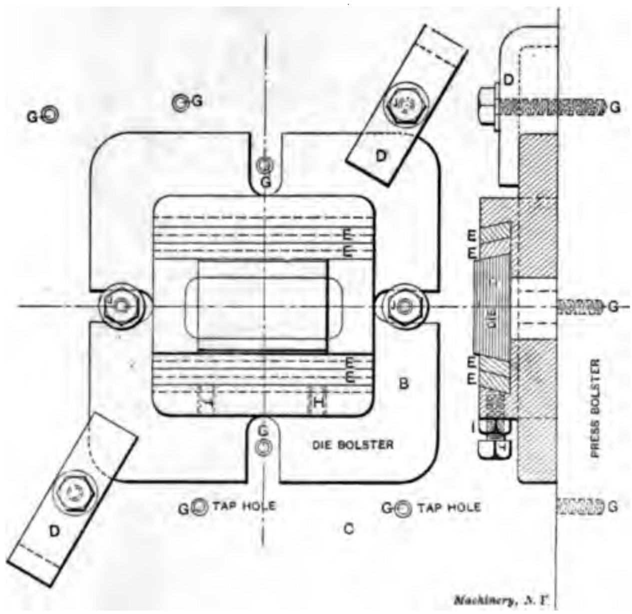


Fig. 56. Improved Form of Die-bed for General Use

jarring loose. The great advantage of having a flange all around the bolster will be apparent when it becomes necessary to swing the die-bed around enough to bring the bolt slots out of line with the tap holes in the press bolster. In a case of this kind a die-bed with a flange all around it may be clamped by means of clamps as shown at *D*, using the tap holes *G* located at different places in the press bolster *C*.

In Fig. 57 we have another die-bed of the dovetail and side set-screw variety, but with the additional feature of end-thrust set-screws. This end-thrust arrangement is an original and novel feature. In order to obtain this additional means of holding the die securely, two square grooves *B* are cut in each end of the die-bed at right angles to the opening for the die. Into these grooves a plate *C* is fitted in which hold the dies in place. The square forming die shown is made in four sections *B* which are held tightly against each other by means of the set-screws *C*, and are held from working up by screws through the bottom of the die-bed—one in each section of the die. The square recess

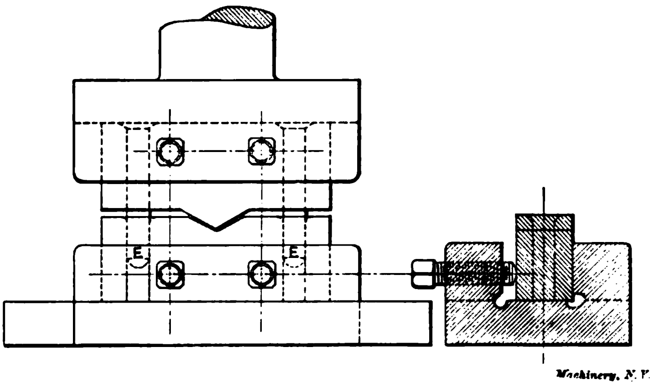


Fig. 59. Simple Form of Die Holder adapted to Bending Dies

is cast in the bed so that in preparing the bed for use it is only necessary to plane off the bottom and top of the flanges and mill the bottom of the recess, and drill and tap for the set-screws. The sides of the recess need not be machined as the dies have no bearing on them.

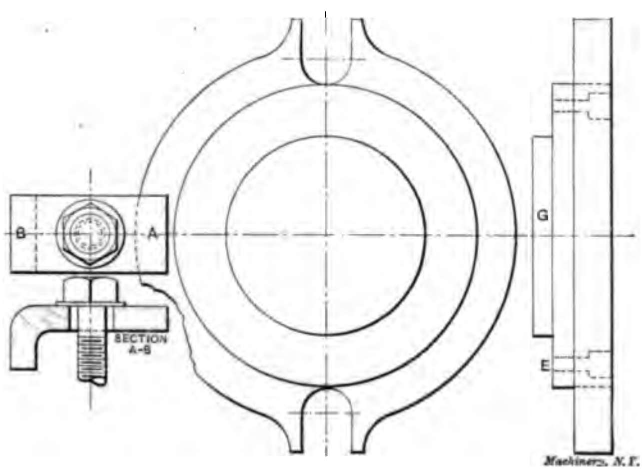


Fig. 60. Die-bed or Bolster for Round Drawn Work

A very simple type of die-bed for bending and forming dies is shown in Fig. 59. It is simply a vise similar in some respects to a milling vise, but having two set-screws to take the place of the movable jaw. The die is simply set in the bed and clamped against the solid jaw by

means of the set-screws. This type of bolster is intended for use only on dies that do not require a "push up," and when the bending or forming operations are done on a solid surface. In order to obtain the best results from this die-bed, the complete outfit of punch holder, punch and die of the type shown in the sketch should be used. The punch holder and punch are made just the same as the die and die-bed. They are kept in alignment when in operation by the two guide pins *E* which are secured in the punch and which enter the die at every stroke of the press, making it practically impossible for the tools to shift while in operation. If it be desired to change the tools it is not necessary to disturb the punch holder or die-bed. They may be left in the press, and by simply loosening the set-screws in the die-bed and punch holder, the punch and die held together by the guide pins may be taken out and set aside and another set slipped into their places.

Fig. 60 represents a bolster for combination dies for round drawing work. This bolster requires but little explanation. It is circular in shape with two steps or extensions, two bolt slots and a flange all around it to allow it to be clamped at any convenient place. When the combination dies are turned in the lathe the bottom die is counter-bored to be a driving fit on the extension *G*, and is held down by screws that pass through the bed at *E* into the die.

CHAPTER VI

FEED STOPS FOR PRESSES*

The simple feed-stops here illustrated are not new or novel in their construction; experienced toolmakers will recognize them at once as "old acquaintances," but there are certain points concerning them of which an explanation will be of benefit to those who are not experienced in punch and die work.

Fig. 61 shows a fixed stop-pin *C*, which is the most common of all feed-stops. It is the particular form of this simple stop to which attention is called. The common way to make a fixed stop-pin is to bend over a piece of steel rod and drive it into the die. This appears simple enough, but it is not so simple as it looks. The difficulties and disadvantages connected with making a bent stop-pin are as follows: First, the difficulty of bending the pin at right angles without breaking it or bending the part to be driven into the die; second, after the pin has been made and hardened it is apt to break in driving it home to its place in the die because of its uneven shape; third, in driving the pin into the die it is apt to swing around out of its proper position making necessary to knock it around again and thus increasing the chances

* MACHINERY, September, 1909.

of breaking it. Every time the die is ground, this difficulty is experienced and the result is frequent breakage and consequent loss of time in waiting for new stops. All these difficulties are overcome by making the style of pin shown in Fig. 61. This is simply a shoulder pin turned to a nice snug fit in the die. The shoulder, which acts as the stop for the stock, may be made larger or smaller in diameter according to the width of scrap desired between blanks. This stop is quickly and easily made, is easily taken out and put back again after grinding the die, and it will last as long as the die itself. It is a good idea to cut a hole through the stripper *A* directly over the stop-pin as shown at *G* so that the operator can see the pin when the press is in operation.

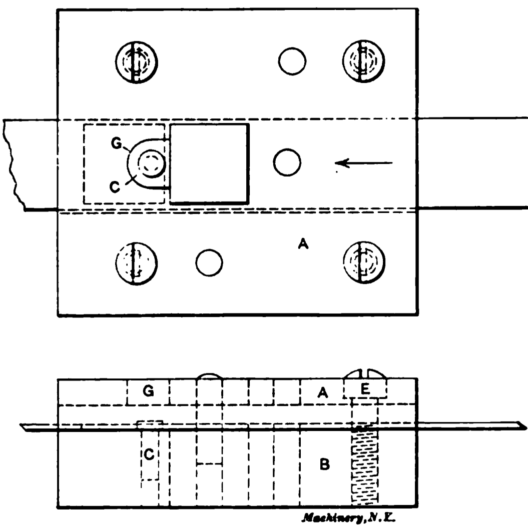


Fig. 61. Latch Stop-pin of Simple Design

The stop shown in Fig. 62 is excellent because of its simplicity, and also because of the great variety of work to which it may be applied. This stop is of the latch variety, but it differs from most stops of this type in that it requires no mechanism to lift it. It is not operated by the action of the press nor by the punch, as is generally the case with latches. Its construction is simple. A hole is drilled through the stripper *A* to receive the pin *K* which passes through a hole in the stop *C*. The stop swings upon this pin. A light flat spring *D* is fastened to the top of the stripper so that the end of the spring rests on top of the stop. In securing this spring to the stripper, it is only necessary to place one end under the head of the screw *E* with a piece of the same material under the opposite side of the screw as shown in the plan view. By this method the spring can be quickly and easily attached or removed, and a straight piece of spring material can be used. The stripper, of course, should have dowel pins in it to insure its coming back into the same position every time the die is ground. The dowels

and screws are shown at *H* and *G*, respectively. The stripp
also be cut off at the stop end as shown at *L* so that the st
outside of the stripper and in full view of the operator. The
this stop is as follows: The stock *F* is fed to the left, a
punched strip passes the stop, the point of the stop *M* drops
springs into the hole made by the blanking punch. The oper
pulls the strip back against the straight outer edge or the
holds it there until the next blank is punched. This proc

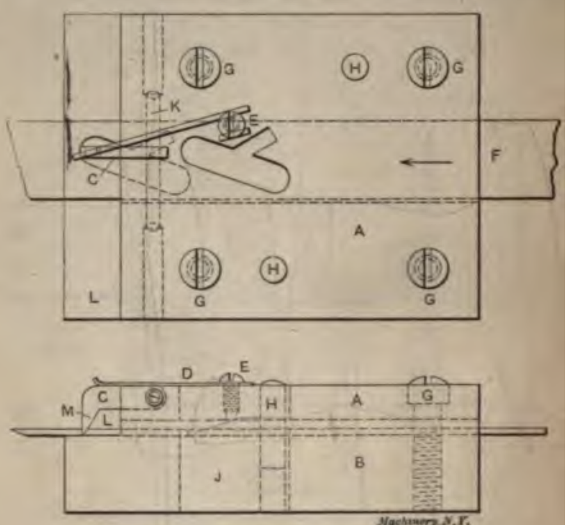


Fig. 62. Improved Form of Fixed Stop-pin

peated at each stroke of the press, the scrap between the bla
pushed past the stop each time and then pulled back against
inner beveled edge of the point *M* causes the stop to lift as
between the blanks is pushed against it, while the outer edge,
at right angles with the die, prevents the stop from lifting
edge of the scrap is pulled back against it.

By this simple stop the operator can feed the stock at wil
waiting for the operation of a mechanically-lifted stop, to say
of the time that is saved by not having to adjust an autom
An operator can make about 40,000 blanks per day with d
with this form of stop on a press making about 100 strokes pe
These stops are used only on hand-fed work.

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By GEO. L. FOWLER and CARL J. MELLIN

Part 1

BOILER AND CYLINDERS

SECOND EDITION

CONTENTS

he has set himself. It is, therefore, in view of these limitations impossible to formulate any set of hard and fast rules that can be made to serve and which will be accepted as absolutely correct in all quarters.

In order, then, that what follows may serve somewhat as a guide to show what has been done in certain concrete cases, a number of assumptions will be made as a basis of work, and two locomotives will be worked out and developed that will meet the requirements of this supposititious road and the assumptions that will be made in connection with its operation. It will be understood that any variation from the assumed conditions may cause modifications of design that will be more or less extensive according to the character of the variations.

In order that a start may be made, we will suppose that two locomotives designed respectively for freight and passenger service are to be designed for a division one hundred and fifty miles in length, that is laid with rails weighing 75 pounds to the yard, whose bridges are of such strength that they are up to the full capacity of the rail,

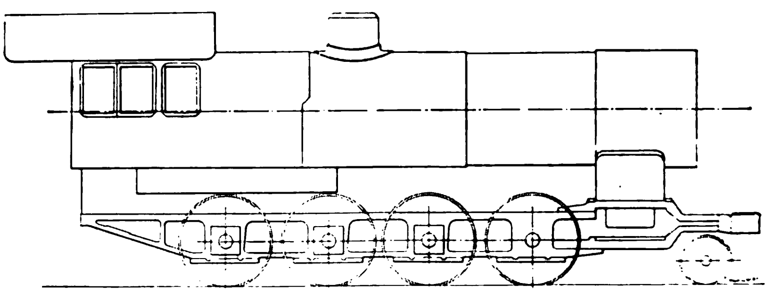


Fig. 1. Outline of a Consolidation Freight Locomotive

with clearances that permit the usual widths and heights of design, and finally with a ruling grade of one per cent 10 miles in length. These figures, scanty as they are, will suffice as a guide to indicate to the locomotive designer the conditions that he must meet. It is, of course, beyond the province of this work to enter into a discussion of the construction of the track, so that it will be merely taken for granted that this is of the most approved character and is kept in first-class condition.

With this data at hand, the requirements usually sent to the locomotive designer are that he shall supply an engine that will haul a given tonnage over a ruling grade at a minimum speed. In the drawing up of the specifications in this form, judgment, backed by experience, must be exercised that the requirements do not call for a heavier engine than the rails are able to carry.

The first steps, then, in the determination of the size and character of the engine that can be used is to ascertain the weight per wheel that can be safely carried upon the rail. This depends upon the metal of which it is formed, the shape, and size or weight. If we take e fiber stress to be put on the rail under a static stress as 12,500 pounds, we find that a wheel load of 22,000 pounds will meet the requirements for a 75-pound rail, and this experience has proved to be good current practice.

For heavy freight work on roads of ordinary curvature, it has been found that four driving wheels coupled are about as many as can be satisfactorily worked. Engines with larger numbers have been built, but even those roads using them have reverted to the consolidation type, having four pairs of wheels coupled and a pony truck in front.

In the case of our supposititious division then, the type that current practice would suggest for adoption would be a consolidation engine, and the weight upon each driving wheel according to the assumed conditions would be about 19,400 pounds, with 21,000 pounds on the truck, or about 176,000 pounds for the total weight of the whole machine.

Experience has shown that the tractive power of an engine can be made from 22 to 26 per cent of the weight on the drivers. In the

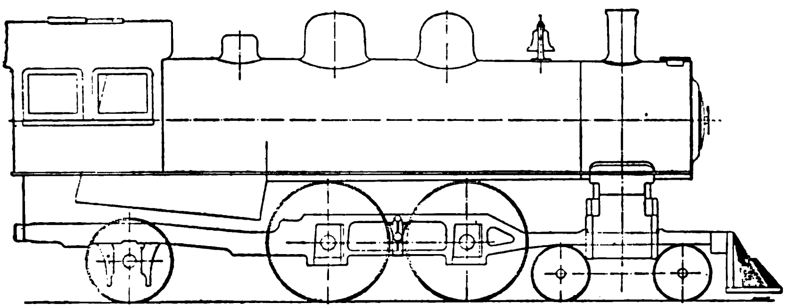


Fig. 2. Outline of an Atlantic Type Passenger Locomotive

present case we will use the former as suitable to the designs to be presented, which would make a total tractive force of 34,200 pounds from which 10 per cent should be deducted for the internal friction and resistance of the engine, leaving 30,800 pounds or about 31,000 pounds available tractive power. When an attempt is made to refer this drawbar pull to the weight of train that can be hauled, we find at once a mass of variable resistances that again render accurate calculations an impossibility. Train resistances will vary with the number of cars, the conditions of the journal lubrication, the direction and force of the wind, and many other minor details of the construction. The best that can be done then is to take an approximate formula for train resistance and make due allowance for the excess resistances resulting from emergencies.

There are some variations in the formulas given by different authorities. The work of foreign specialists is of little or no value to American designers on account of the difference in the rolling stock experimented with, and of all the work done in this country the formula known as that of the *Engineering News* is the most widely accepted for approximate reliability. This formula is

$$R = \tfrac{1}{4} v + 2 \qquad (1)$$

in which R is the resistance in pounds per ton of 2000 pounds in the weight of the train, and v the speed of the train in miles per hour. By substituting a value for v of 10 miles per hour, in the example under consideration, we find the rolling resistance to be

$$R = \frac{10}{4} + 2 = 4.5 \text{ pounds per ton of 2000 pounds.}$$

The resistance due to grade is found by the formula

$$R' = l \times \frac{p}{100}, \text{ in which} \qquad (2)$$

R' = resistance in pounds,
l = load in pounds to be carried up the grade,
p = percentage of gradient,

From this it appears that the resistance per ton of load on a 1 per cent grade is

$$R' = 2000 \times \frac{1}{100} = 20 \text{ pounds.}$$

To these resistances should be added the resistance due to curves, which is found to be about 0.5 pound per degree of curvature per ton of load where the curves are not compensated for in the reduction of the grades. On modern roads, however, the curves are generally compensated for in the gradient and it will therefore be omitted in the example and the rolling and grade resistances only used. Hence $R + R' = 4.5 + 20 = 24.5$ pounds to the ton, leaving the minor uncontrollable resistances to be taken care of by a certain margin provided for such purposes and other emergencies.

We have seen that the available tractive power of an engine to do the required work will have to be 30,800 pounds under the driving wheels, but it would not be advisable to load the engine to these figures under all conditions without allowing a reasonable margin for the uncontrollable resistances, previously referred to, in the variation of car resistances and weather conditions of about eleven per cent, which leaves 27,400 pounds for moving the train, inclusive of the weight of engine and tender.

By dividing this amount by the resistances obtained from Formulas (1) and (2) we get the total weight of the train to be hauled up the one per cent grade at the rate of ten miles per hour, *viz.*:

$$\frac{27{,}400}{24.5} = 1118 \text{ tons of 2000 pounds.}$$

Before it is possible to determine the load behind the tender, it is necessary to work out the dimensions of the engine and thus ascertain the weight of the engine and tender. With the requirements before us, experience has taught that the following dimensions of heels and their related parts will most satisfactorily meet the condi-'s, namely:

Diameter of driving wheels, 57 inches.
Boiler pressure, 200 pounds.
Strokes of pistons, 26 inches.

With these dimensions given, we get the cylinder diameter from the formula:

$$d = \sqrt{\frac{T \times D}{P \times 0.85 \times S}}, \text{ in which} \qquad (3)$$

d = diameter of cylinder,
T = the required tractive power,
D = diameter of driving wheels,
P = boiler pressure,
S = stroke of pistons.

As 85 per cent is the generally adopted coefficient of the boiler pressure for average cylinder pressure at low speed, that is taken as a constant factor. By substituting the values decided upon in tne formula we obtain the cylinder diameter

$$d = \sqrt{\frac{34{,}200 \times 57}{200 \times 0.85 \times 26}} = 21 \text{ inches.}$$

The weight allowed on the driving wheels is 155,000 pounds and by ordinary proportions in the design of an engine of this type about 21,000 pounds will come on the truck, making a total weight of the engine alone of 176,000 pounds, and the weight of the tender filled with coal and water will be about 110,000 pounds, which added together, makes 286,000 pounds or 143 tons.

This amount will now be deducted from the previously obtained total weight of the train to find the load behind the tender, namely: $1118 - 143 = 975$ tons.

A rational specification for an engine to work on such a grade, then, would be one capable of hauling a train of 975 tons at a speed of ten miles an hour, leaving a reasonable margin to be utilized under favorable conditions. The result will be a consolidation locomotive of the general outline shown in Fig. 1.

Turning now to the matter of the passenger locomotive, there are three types in common use in this country. They are the eight-wheeled American or 4-4-0 type, the Atlantic or 4-4-2 type, and the ten-wheelers or 4-6-0 type. The latter is heavier, and has a greater tractive power than the other two, and is intended for what might be called special services.

To these may be added a fourth, the Pacific type (4-6-2). The last, having the same number of drivers as the ten-wheeler, has greater boiler power in proportion to its adhesive weight, and is used in what might be called exceptionally heavy service. The first two classes bear, in a general way, the same relation to each other as the two last, namely, that of having the same number of driving wheels. The Atlantic type has the greater boiler power and is capable of maintaining a higher speed than its predecessor, the eight-wheeled engine.

The Atlantic type, first introduced in 1893, possesses so many advantages for heavy and fast passenger service, that it has been rapidly introduced for that purpose, supplanting the first type in many places where it was originally used.

In order, then, to simplify matters, it will be decided at the outset that the design will be made for the Atlantic type; and, as the work progresses the advantages possessed by the same will be set forth.

With the same weight of rail and the same conditions of track as those set forth in the determination of outlines of the freight engine, we would naturally have the same weight upon the driving wheels. But as there are but two pairs instead of four, the available tractive force drops to 18,250 pounds.

In the case of the passenger engine, there are some complications introduced into the calculation that do not enter into that of the freight engine. The most important one is that of speed. It is evident that ten miles an hour would not at all answer the requirements of passenger service even on the grade mentioned, and the work should be based on a speed of at least thirty-five miles an hour. Owing to the shorter cut-off that will be involved by such a speed, the full adhesive weight of the locomotive cannot be used so that a very liberal reduction will have to be made. The designer knows from analysis that only about 60 per cent of the total adhesive weight is available at such speeds, and as this should be cut down still more to allow a suitable margin for wind and other uncontrollable resistances there is left but little more than 10,000 pounds tractive power that can be used at the speed decided upon.

Referring back to the formula of the *Engineering News* for the train resistance at 35 miles per hour, we find it to be 10.75 pounds per ton, to which should be added the 20 pounds due to grade, making a total of 30.75 pounds per ton. If we divide the available tractive power of 10,000 pounds by 30.75, we obtain 325 tons as the weight of the train, inclusive of the engine, that can be hauled. If the speed were to be dropped to 25 miles per hour, this weight would be raised to something more than 350 tons. It would, therefore, be a reasonable specification on the part of the railroad officers to call for a locomotive capable of hauling 350 tons up a 1 per cent grade at a speed of 25 miles an hour, and, if such a specification were to be made, an engine like that shown in Fig. 2, having about 20,000 pounds on each of the driving wheels, and weighing in all 168,000 pounds, would be offered for the service.

CHAPTER II

THE BOILER*

In the preliminary considerations regarding the designing of the locomotive two main points have been approximately settled: the weight of the engine that can be made to produce the tractive power needed to perform the work that is required and the size of the cylinders that will be needed in order to utilize the adhesive weight of the engine, with the steam pressure of 200 pounds per square inch that it has been assumed the boiler is to carry.

It may be remarked here, that in the designing of a modern locomotive all things are made subordinate to the boiler and cylinders, and of these two the boiler is the more important. On it depends the whole action of the engine. If it fails to supply the requisite amount of steam the engine either cannot haul its train without losing time upon the schedule on which it is supposed to run; or, if it supplies the steam, its grate area and heating surfaces may be too small to do the work properly, and the result will be that the engine is extravagant in the use of fuel. For these reasons, then, it is the end and aim of every designer to use as large a boiler as possible in order to obtain an ample supply of steam, and at the same time secure that supply on a minimum fuel consumption. At the same time he is limited by the allowable total weights which must include not only the boiler itself but the cylinders, wheels, axles, machinery and other parts.

In this, as in the work that has already been done, it is impossible to lay down any hard and fast rules, and the designer will frequently find himself thrown back on his own judgment and experience in default of formulated data bearing upon the subject that he has in hand. With this understanding of the matter attention may now be turned to the determination of certain points connected with the boiler of the consolidation freight locomotive that we have in hand, and of which a preliminary outline has been laid down in Figs. 3 and 5.

The two important elements in the boiler are the heating surface and the grate area. The former takes the precedence and is usually based upon some assumed service that the engine is to render. In the case in hand this has been arbitrarily placed at the hauling of 975 tons as a speed of 10 miles an hour up a 1 per cent grade, or of moving 1118 tons including the weight of the engine and tender.

For some time the empirical rule for the determination of the amount of heating surface was to make it, in square feet, 400 times the cubic contents of a single cylinder in cubic feet. This rule is,

* MACHINERY, Railway Edition, November and December, 1904, and January, 1905.

however, only approximately followed and is regarded merely as a rough guide as to what should be aimed at as a minimum. For, as already stated, it is the desire of the designer to make the heating surface as large as possible, so that this ratio is exceeded wherever it is possible to do so and still keep within the limitations of weights. This is especially true of work in connection with passenger engines, where the demand for steam is apt to be excessive.

Then, too, after the general dimensions of the boiler have been decided upon it is quite possible to vary the heating surface through comparatively wide limits by a variation in the spacing of the tubes. It is in this particular especially that good judgment must be exercised. There is the constant temptation, backed by desire to run the heating surface up, to use a large number of tubes, but it must be borne in mind that it is sometimes advisable to space the tubes more widely apart and put in a smaller number than it is to crowd them; because it is necessary that the steam formed in contact with the lower rows should be free to rise to the surface of the water, otherwise poor evaporation, damp steam or even priming may be the result. It is, therefore, usually better to sacrifice some of the heating surface that it might be possible to obtain, as this will give an actual increase in the evaporative efficiency.

Turning now to the determination of the amount of heating surface and regarding the rule given merely as an approximate guide, it is considered that a more correct basis for the estimate will be to ascertain the weight of steam that will be required to maintain a given speed and tractive power, and from that calculate the amount of heating surface that will be needed to produce it. In short, it is necessary to determine the amount of water that is to be evaporated per minute or per hour, and this, in turn, swings back to the cylinder, where the point of cut-off and the pressure will have to be assumed. This assumption should be based on the records of performances of other engines, and should be critically scrutinized in order to determine the influences that necessary variations in valves, valve motion and steam passages may have upon the result. This means a thorough investigation and the securing of reliable data if a close degree of accuracy is to be obtained.

For the solution of the specific problem that we have before us, the data available have made possible the development of the following formula for the determination of the amount of water to be evaporated per minute by an engine in heavy freight service:

$$W = 2uvnpc \times 1.25 \qquad (4)$$

in which

W = pounds of water to be evaporated per minute,
u = the volume in cubic feet of the two cylinders,
v = the percentage of the stroke at which cut-off takes place,
n = the number of revolutions per minute of the driving wheels,
p = the weight of 1 cubic foot of steam at the cut-off pressure,
c = the factor of evaporation from and at a temperature of 212 degrees F.

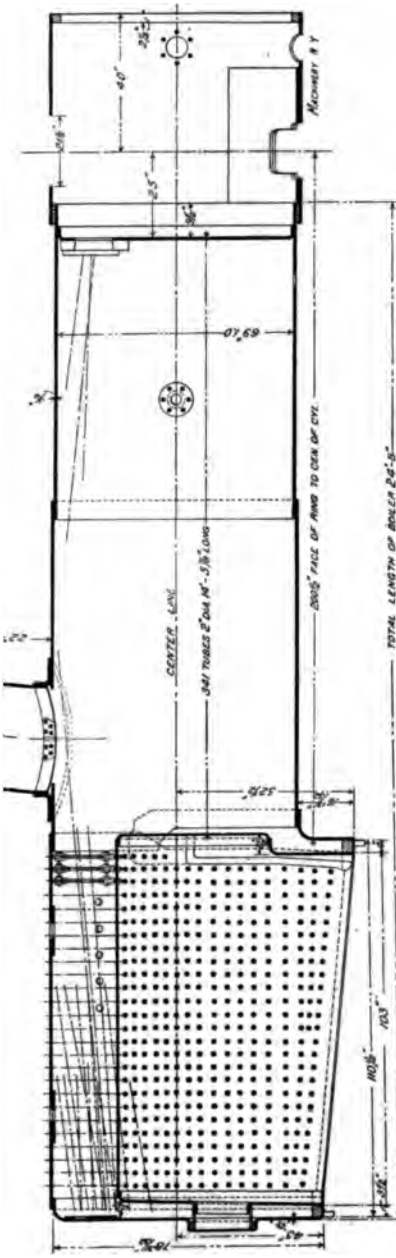


Fig. 3. Section of Boiler of Consolidation Freight Locomotive to haul Train of 975 tons up Grade of one per cent at Ten Miles per Hour

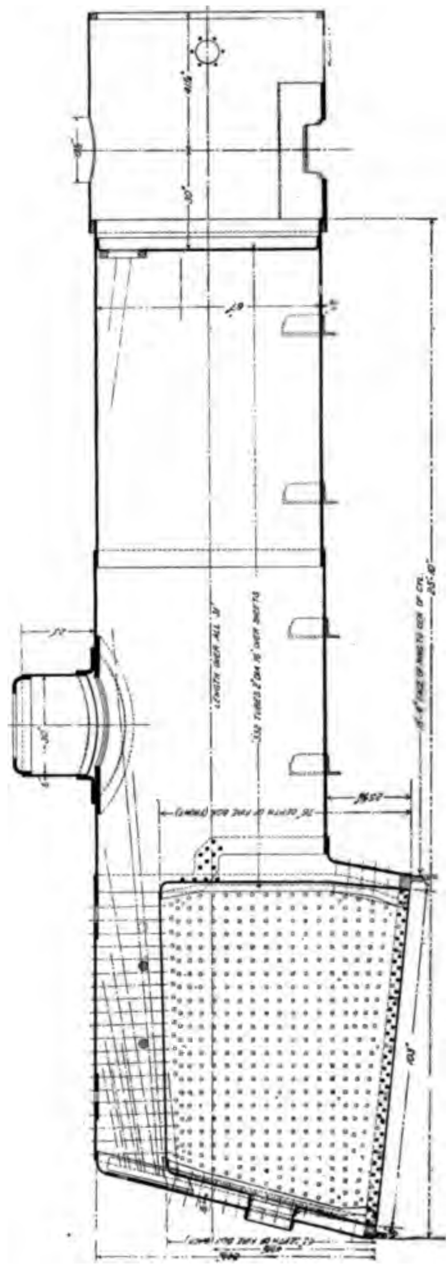


Fig. 4. Section of Boiler of Atlantic Type Passenger Locom[illegible]nd Train of 550 tons up Grade of one per cent Twenty-five Miles per Hour

It will be seen that this formula involves a few points that nave not yet been settled in the course of our work, and which must be decided upon before further progress can be made.

Taking up Formula (4) seriatim, the coefficient 2 is used because the cylinders must be filled with steam twice at each revolution. The factor v is readily determined, and for the case of the cylinders for the consolidation locomotive that are 21 inches in diameter and of 26 inches stroke it is 10.4 cubic feet. For an engine working at the speed and conditions that have been assumed, practical experience shows that the valves may be made to cut off at about 70 per cent of the stroke.

The speed of the engine and the diameter of the drivers are needed in order to determine n. We have already assumed that the speed

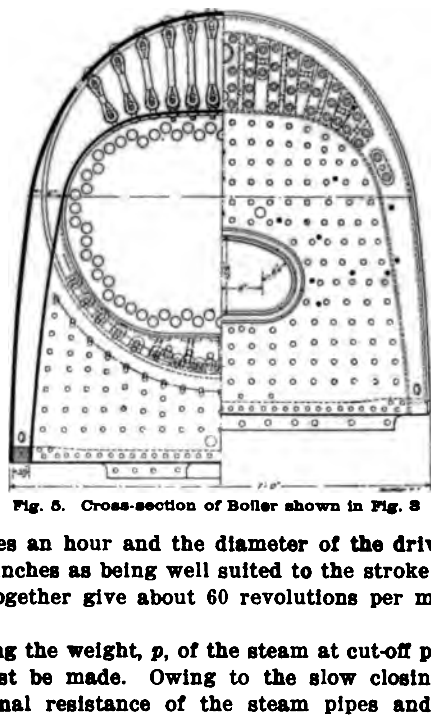


Fig. 5. Cross-section of Boiler shown in Fig. 3

shall be 10 miles an hour and the diameter of the driving wheels has been put at 57 inches as being well suited to the stroke adopted. These factors taken together give about 60 revolutions per minute, or, more exactly, 59.

In ascertaining the weight, p, of the steam at cut-off pressure another assumption must be made. Owing to the slow closing of the valve, and the frictional resistance of the steam pipes and passages, this pressure will be somewhat below that of the boiler, and for the 70 per cent of cut-off at which the engine will be worked at this speed, it has been found to be about 80 per cent of full boiler pressure, or in the case of our assumed pressure of 200 pounds per square inch it is 160 pounds. Referring to the steam tables as ordinarily published we will find that, at this pressure, the weight will be 0.3873 pounds.

The factor of evaporation is obtained by dividing the difference between the heat of the steam at the observed pressure (in this case 200 pounds), and the total heat of the feed water (which is taken at 32 degrees) by the factor of 965.7 which gives in the case in hand about 1.24. The 1.25 at the end of the term is an allowance made for clearances, leakages and the like. With these quantities obtained they may be substituted in Formula (4) with the following result:

$$W = 2 \times 10.4 \times 0.70 \times 60 \times 0.39 \times 1.24 \times 1.25 = 528 \text{ pounds.}$$

That is, the requirements of the engine are such that it must be supplied with the equivalent of 528 pounds of steam per minute from and at 212 degrees F., or 31,680 pounds per hour. The actual evaporation requirement is the amount obtained without the factor 1.25, or 25,350 pounds hourly.

Having determined the amount of work that the boiler will be required to do, the next step is to ascertain the details of dimensions

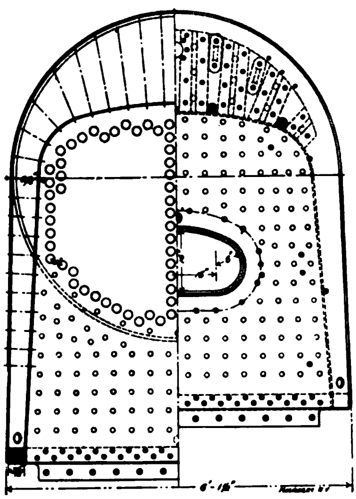


Fig. 6. Cross-section of Boiler shown in Fig. 4

of that boiler to meet the requirements. The first movement in this matter is one that varies with every road and every locality for which an engine can be designed. It involves a determination of the heating capacity of the coal that is to be used. For this reason, when a builder is called upon to design a locomotive whose performance is to be guaranteed he requires a sample of the coal that is to be used in order that its value may be ascertained and the proper proportions of grate area and heating surface be arranged. In the case in hand, it will be necessary to make an assumption and this will be done on the basis of a good quality of bituminous coal, that can be depended upon to evaporate 7½ pounds of water per pound of fuel in a properly proportioned firebox with a suitable ratio between heating surface and grate area.

Experimental examination has shown that, for this grade of coal, one square foot of heating surface is capable of absorbing the heat developed by 1⅛ pounds of coal burned per hour. We can now emboiler and engine may balance well on the wheels and not tend to put an excessive load on the rear drivers or front truck, the overhang of the firebox back of the rear drivers should be made with due consideration to the position of the wheels and distribution of the weight. On this basis the approximate length of the boiler from the back head to the front tube sheet would be about 24 feet 6 inches or 104 inches more than the wheelbase. Now comes the proper distribution of this distance into firebox and tube lengths.

The requirements of the service will demand that the firebox extend out over the wheels, so that constructional limitations will decide the total width. With a firebox tapering in at the top, a total width of about 7 feet at the mud-ring can be used. Allowing for the thickness of metal and a width of ring on each side of 3½ inches will leave about 75¼ inches, or 6¼ feet, for the width of the grate. In order to obtain 57 square feet of grate area the length should be a little more than 9 feet, and deducting 4 inches at the rear for the water leg, would leave a length of about 15 feet for the tubes.

As already stated, these dimensions are not fixed, so that in the designing of the boiler some modifications are possible, to suit the requirements and conveniences of construction. If the work were to be undertaken from the start a great many trials would have to be made on the drawing board in order to secure the proper adjustments. Without reviewing these steps one by one, it will be permissible to state that if a firebox of the width given be used, and it be made 6 feet high above the mud-ring, it will have a heating surface of about 15½ square feet per foot of length in addition to about 54 square feet for the ends, if no allowance is made for tube and door openings. Suppose, then, this firebox be made 8 feet 6 inches long inside; the heating surface will be 186 square feet, leaving 2630 to be made up in the tubes. With 0.5 square feet per lineal foot as the approximate heating surface of a 2-inch tube, this would require a total tube length of 5260 feet. As the available length is 15 feet, this would require 351 tubes.

With this as a guide, it will be found, in laying out the best form of firebox, and allowing the space needed above it for water and steam, and by keeping within the limits imposed, that 341 tubes 2 inches in diameter will be the most convenient number and that best adapted for service. They will have a length of 14 feet 6 inches over the tube-sheets and the inside diameter of the smallest ring of the shell will be 69 inches, all of which is shown in the general outline of Fig. 3. On taking this boiler and calculating the actual heating surface it will be found that there are 2570 square feet in the tubes to which the 186 square feet in the firebox is to be added, making 2756 square feet in all.

Without entering into the details of the proportioning of the parts in order to secure the proper strength, which is really the next step, but which is outside the province of this work, attention is to be turned to making an estimate of the weight of such a boiler as that outlined. It will be found that the weight of the materials in the boiler itself with its firebox and bracing complete, but without the tubes will be about 30,000 pounds to which must be added the weight of the latter or 11,000 pounds, thus bringing the total weight up to 41,000 pounds.

With the weight and dimensions thus established, it will be possible to go on and work out the other parts of the mechanism.

Turning now to the passenger locomotive, the boiler must be worked out somewhat differently. Here we are not confronted with the hauling of a very heavy load at a low speed, but with the work to be done at a comparatively higher speed with a lighter load. This will involve the working of the engine at a shorter cut-off and here again experience in the working of an engine will be called into play.

In the preliminary considerations of this matter the requirements were laid down that the engine should be capable of hauling a passenger train weighing 350 tons up a grade of 1 per cent at a speed of 25 miles an hour. The engine selected for this purpose was of the Atlantic type, as being that best adapted to the work. As in the case of the consolidation locomotive the diameter of the driving wheels must be arbitrarily determined. In this the designer is to be guided by the necessity of obtaining a reasonably good speed on lighter grades and level; and for this a diameter of 77 inches will be found to be well suited.

From Formulas (1) and (2) it will be found that the resistance of the train will be $R = \frac{25}{4} + 2 = 8.25$ pounds, and $R' = 2000 \times \frac{1}{100} = 20$ pounds, or that the total will be 28.25 pounds per ton, and $350 \times 28.25 = 9888$ pounds, or that, in round numbers, a drawbar pull of 10,000 pounds will be required to do the work.

It is evident that the varying resistances of the train do not become any the less when there is an increase of speed, but rather increase. On the other hand, the power of the engine diminishes with the increase of speed. So instead of the 11 per cent margin allowed for uncontrollable resistance, in the case of the freight locomotive, an allowance of at least 15 per cent must be made for the passenger locomotive. By allowing a 15 per cent increase of resistance and a similar loss of power in connection with the needed drawbar pull we have

$$\frac{10{,}000 \times 0.15}{1 - 0.15} + 10{,}000 = 1765 + 10{,}000 = 11{,}765 \text{ pounds.}$$

Further, the internal friction of the engine is not reduced in proportion to the fall in the mean effective pressure in the cylinders. So instead of taking this at 10 per cent, as in the case of the consolidation, it has been found by tests and experience that an allowance of 18 per cent of the theoretical tractive power must be made. Hence the minimum tractive power needed would be $11{,}765 \times 1.18 = 13{,}882.7$ pounds, or in round numbers, 13,900 pounds.

For the determination of the piston speed, to be used in calculating the proper diameter, the following formula may be used:

$$S = \frac{12 \times 5280\, v \times 2\, p}{3.1416\, D \times 60 \times 12} \qquad (7)$$

in which

S = the piston speed in feet per minute,

v = speed of the train in miles per hour,

D = diameter of the driving wheels in inches,

p = the stroke of the piston in inches.

In order to fill in the required factors in this case, it is necessary to assume a piston stroke that will be well adapted for use with the wheel diameters that have been given and the service to be performed. This will be placed at 26 inches. Then by substitution Formula (7) becomes:

$$S = \frac{12 \times 5280 \times 25 \times 2 \times 26}{3.1416 \times 77 \times 60 \times 12} = 473 \text{ feet per minute.}$$

Indicator diagrams have shown that at a piston speed of 500 feet per minute the mean effective pressure in the cylinder will be about 62 per cent of the boiler pressure.

By substituting the values which have now been obtained in Formula (3) it is possible to obtain the diameter of the cylinder. This is then:

$$d = \sqrt{\frac{13{,}900 \times 77}{200 \times 0.62 \times 26}} = 18.5 \text{ inches, nearly.}$$

In order that there may be an ample margin of power the cylinder diameter, in this case, will be increased to 19½ inches, as the weight allowed will make it possible to use a boiler of sufficient capacity to supply such a cylinder.

The boiler dimensions may now be determined in the same way as for the freight engine by the substitution of the several values in Formula (4), for the calculation of the equivalent amount of water to be evaporated per minute. This then becomes:

$W = 2 \times 9 \times 0.55 \times 110 \times 0.30 \times 1.24 \times 1.25 = 506.4$, or 500 pounds per minute, or 30,000 pounds per hour.

Taking the same quality of coal as before, and allowing 1½ pound of coal per square foot of heating surface per hour, the latter, according to Formula (5) should be:

$$\frac{30{,}000}{7.5 \times 1.5} = 2666 \text{ square feet,}$$

and from Formula (6), with a rate of combustion of 85 pounds per square foot per hour of grate area, the latter will be:

$$G = \frac{30{,}000}{7.5 \times 85} = 47 \text{ square feet.}$$

Proceeding as before, it will be found that the most convenient dimensions for tubes and surfaces will give a boiler with 2767 square feet of heating surface in the tubes and 162 square feet in the firebox, making a total of 2929 square feet and a grate area of 46.3 square feet.

This is somewhat in excess of that called for by the formula in the way of heating surface, but it will be found to be of great advantage in a passenger engine and can only be obtained in this type of locomotive, though there is some objection to the extra weight on the forward truck and trailing wheels.

The length of the firebox will be 102 inches by 65½ inches wide. There will be 332 tubes of 2 inches diameter, 16 feet long, and the diameter of the smallest ring of the shell will be 67 inches. Finally, the weight of the boiler will be 39,300 pounds.

In the matter of the evaporative qualities of the various grades of coal and the consumption per hour, the following table is presented as deduced from a report made to the American Railway Master Mechanics' Association in 1897:

	Water Evaporated per pound of Coal	Coal to be Burned per sq. ft. of Grate Area per hour
Large Pennsylvania anthracite	8 lbs.	60 lbs.
Fine Pennsylvania anthracite	6½ lbs.	35 lbs.
Virginia semi-bituminous	9 lbs.	65 lbs.
Illinois bituminous	7 lbs.	90 lbs.

Finally the following ratios were suggested:

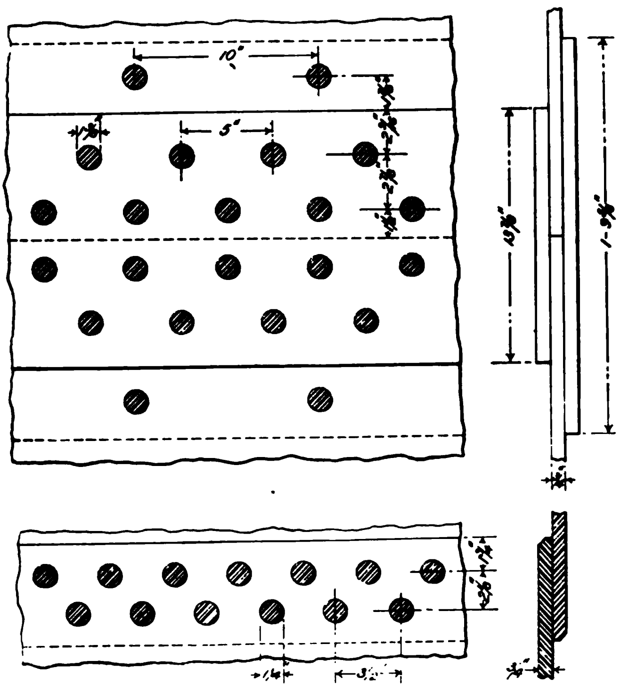
	Cyl. Volume in cubic feet to grate Area in square feet	Cyl. Volume in cubic feet to Heating Surface in square feet	Heating Surface to Grate Area
Large anthracite	1:4	1:180	40:1
Small anthracite	1:9	1:200	20:1
Bituminous	1:3	1:200	60:1

In the work that has preceded, the various steps leading up to the determination of the general dimensions and proportions of the boilers have been indicated. With this accomplished there still remains a great deal of work to be done in laying out the details of the several parts of the boiler itself. While it will be impossible to enter into a discussion of boiler construction in detail the importance of this part of the locomotive is such that some attention should be paid to it here.

It has already been remarked that the boiler is the life of the locomotive, and that upon it depends the efficiency of the whole machine. Hence it is of the utmost importance that the greatest care should be exercised in its design and construction to make sure that it is possessed of the requisite strength and power of endurance.

In this, attention is first turned to the shell whose strength depends upon the thickness of the plate of which it is formed and the type of longitudinal seam used to connect the edges of the same. In regard to the latter, that one should be selected that will give the highest percentages of strength, as compared with that of the solid plate, consistent with practicability of construction and maintenance in service.

Without entering into a discussion of the relative merits of the various types of joints it may be stated that the sextuple riveted, double butt strap joint with welts inside and outside the main plates as shown in Fig. 7, is best adapted for this work and will, therefore, be used. In this joint there are three rows of rivets on each side of the joint, all of which pass through the inner strap, and but two through the outer. For an analysis of the strength of such a joint, the reader is referred to the various handbooks on boiler construction, where it will be found that the calculated strength of such a seam is a little more than 86 per cent of that of the solid plate and that 85



Figs. 7 and 8. Sextuple Riveted Butt Strap Seam, and Double Riveted Lap Seam

per cent can be counted upon in regular working practice and construction.

With this as a preliminary basis, the strength and thickness of the shell can be calculated from the following formula:

$$L = \frac{S\,C}{f} \tag{8}$$

in which

L = the working stress that can be put upon the plate per square inch of section,

S = the ultimate tensile strength of the steel in pounds per square inch,

C = the percentage of strength of the seams of the plate,
f = the factor of safety that it is necessary to allow.

The specifications for boiler steel of the American Railway Master Mechanics' Association state that "the desired tensile strength is 60,000 pounds per square inch, with minimum and maximum limits of 55,000 and 65,000 pounds." As, in the design it will be necessary to work to the minimum, 55,000 pounds per square inch will be taken as the tensile strength. Practice has further shown that a factor of safety of 5 is well adapted to working conditions. Hence by the substitution of the values obtained in the second term of the equation we have

$$L = \frac{55{,}000}{5} = 11{,}000 \text{ pounds}$$

per square inch when referred to the solid portion of the plate, although when referred to the efficiency area of the seam this is

$$L = \frac{55{,}000 \times 0.85}{5} = 9350 \text{ pounds.}$$

The actual thickness of the shell plate is calculated by the formula

$$T = \frac{D' P}{2 L} \tag{9}$$

in which

T = the thickness of the plate in inches,
P = the working pressure,
D' = the diameter of the boiler in inches.

The coefficient 2 is in the denominator to allow for tne stress carried by two sheets, at opposite ends of the diameter. Referring to the boiler intended for the consolidation engine, Formula (9) becomes, by substitution,

$$T = \frac{69 \times 200}{2 \times 9350} = 0.738.$$

We therefore take ¾ inch as the proper thickness of the smallest ring of the boiler shell.

In the case of the boiler for the Atlantic type engine, the inside diameter is 67 inches and Formula (9) becomes

$$T = \frac{67 \times 200}{2 \times 9350} = 0.716.$$

Owing to the liberal factor of safety that has been adopted, it will be allowable to take the sheet in nearest sixteenths which will give 11/16 inch as the thickness of the smallest ring.

In these calculations the rings of the sheet have been considered as though they were integral and unbroken. This is true of the front sheet but in the second, the opening for the dome weakens it to such an extent that it is generally made about 1/16 inch thicker than that calculated. Hence, in the case of the consolidation locomotive this sheet should be 13/16 inch thick, while for the Atlantic ¾ inch may be used, although this falls a little short of the extra 1/16 inch called for.

The dome opening should be made as small as possible, besides being thoroughly braced by a heavy dome base and an inside liner around it, as clearly shown in Fig. 4 of the passenger engine boiler. This opening should be limited to that actually needed for the entrance and adjustment of the standpipe, and for its use as a manhole—a rule that applies equally well to the diameter of the dome. The idea that the dome can serve any useful purpose as a storage reservoir for steam has long since been discarded as it is merely a means of elevating the throttle above the water and thus securing dry steam for the cylinders.

Although the neutral part of the shell within the lines of rivets holding the dome in place should not be taken into consideration in calculating the strength of the former, it nevertheless does serve a very useful purpose, and materially adds to the strength of this part, in that the flexible edge around the opening transfers the stress from the edge to a line inside the inner row of rivets of the dome base, and thus to the solid portion of the shell, and thereby lessens the tearing effect that would exist if the opening were cut out so as to leave only the usual margin inside the rivets.

In the designing of the circumferential seams, the steam pressure may be disregarded, and the work done with consideration only to tightness and structural strength. A brief consideration of the stresses to which the boiler is subjected will show that, when the tubes are in position they, together with the bracing of the front and back heads, so relieve the longitudinal seams from the stresses due to seam pressure alone that the latter becomes a negligible quantity. For that reason a double riveted lap seam is used, which, while it has perhaps less than 70 per cent the strength of the solid plate, is ample for its work. Such a seam is shown in Fig. 8.

Turning now to the firebox, the inside sheet should be made thin so as to offer the minimum resistance to the transmission of the heat of the fire to the water beyond. The proper spacing of the staybolts will, of course, make it possible to use almost any thickness of sheet; hence the choice must be made as the result of experience rather than from any mathematical calculations. This experience has shown that when the steam pressure ranges from 180 pounds to 200 pounds per square inch, the best results can be obtained with side sheets ⅜ inch thick. This does not hold for the tube sheets, where, on account of the necessity of expanding and fixing the tubes, the thickness should never be less than ½ inch and the use of ⅝ inch will frequently be found advisable. So, basing the choice on this general principle, the thickness of the metal in the side and tube sheets of the two boilers will be taken at ⅜ inch and ½ inch respectively, the crown-sheets being given the same thickness as the sides.

The general practice at the present time for supporting the crown-sheets is by means of radial stays, while the flat surfaces are held the ordinary staybolts. In some cases the latter are replaced at the upper forward portions of the firebox, where there is the greatest difference between the expansion of the inner and outer sheets, by flexible staybolts of various designs. Owing to this difference in the expansion of the two sheets, and the bending of the staybolts resulting therefrom, it is desirable that the latter should be made of a comparatively small diameter. In practice, on boilers carrying a high steam pressure, the diameters range from 7/8 inch to 1 1/8 inch as a maximum, and they are usually spaced so that the stress put upon them does not exceed 5500 pounds per square inch of section. The spacing runs from 3 3/4 to 4 1/2 inches from center to center of the bolts. If, then, 4 inches is adopted in the boilers under consideration, the total stress on the 16 square inches held by each bolt will be 3200 pounds, which will call for a staybolt 7/8 inch diameter and load it to about 5330 pounds per square inch of section.

To provide for the injurious stresses imparted on the flue-sheet and shell by the expansion of the former in advance of the latter in raising steam in the boiler, sling stays of a telescopic nature are applied over the forward part of the firebox so that the latter is allowed to rise slightly until the shell of the boiler is heated by the water and steam, which then expands and gradually takes up the slack thus formed and brings the stays under tension as the temperature and pressure increases.

The width and shape of the water legs has a most important bearing on the efficiency and durability of the boiler. These two points should be so related to each other that there is room for the inflowing current of water and the escaping steam that is generated in contact with the inner sheet. Carefully conducted experiments have shown that, in some cases, where the sheet is vertical there is hardly any water in contact with the upper portions, but that they are covered with a layer of steam bubbles ascending to the surface. Such a condition has the double disadvantage of lowering the rate of evaporation and leading to an overheating of the sheets. In order to avoid this, the sheet should be so sloped in that the steam, in rising vertically, tends to leave it and give free access to the water. It will be noticed from the cross-sections in Figs. 5 and 6 that this has been done in both boilers.

Again, owing to the fact of there being more steam at the upper portion of the leg than the lower, it should be widened at the top, and this has also been done in both cases. At the mud-ring the width used is 3 1/2 inches while at the top 6 inches is required. Such a space will be found to work well in practice, and is founded on sound reasoning rather than on mathematical calculations. There are many other points of more or less importance that could be referred to, but available space requires that the discussion should be limited, in the main, to general principles. Finally a few recommendations should always be borne in mind, and the work designed in accordance therewith. Among these, one of the first importance is that flat surfaces should be avoided as far as possible, and where they cannot be done away with entirely, they should be reduced to the lowest dimensions, so as to cut down staybolt stresses to a minimum. Wherever flanging or bending of the sheets is required, it should be done with a liberal radius of curvature, so as to secure flexibility and avoid undue internal stress of the material. In the application of the bracing great care should be exercised that it gives strength without adding too much to the rigidity. A boiler is subjected to such varying temperatures in its different parts that it must expand and contract differently; hence it is of the utmost importance that it should be capable of this internal movement of its parts relatively to each other,

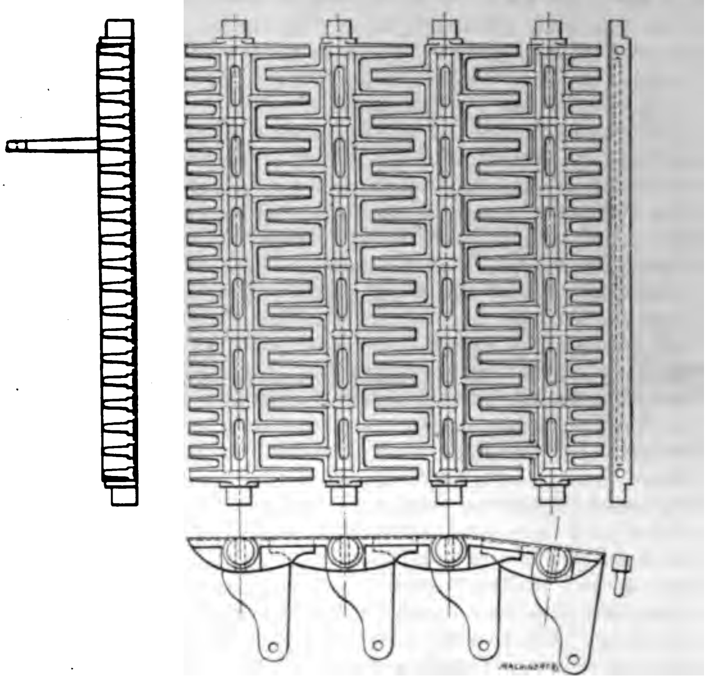


Fig. 9. Grate for Bituminous Coal

without putting an undue stress upon the metal of which they are composed.

A few words may be added regarding the actual work of construct- the boiler, for this, while not strictly belonging to the work of designer, should nevertheless, be borne in mind by him, and may form a part of his specifications. Without taking up the matter l its detail, a few points will be touched upon.

Next to the formation of a tight joint at the sheets, it is of the utmost importance that the tubes should be well and efficiently set. To do this it is advisable to make the holes in the front tube-sheet 1/16 inch larger than the diameter of the tubes, while those at the box end should be of the exact size. Both edges of the holes be chamfered to a radius of 1/16 inch. Further, a copper ferrule should be used at the firebox end, and this should have the same outside diameter as the tubes, a thickness of about 1/16 inch and a length ¼ inch greater than the thickness of the sheet. The ferrules should be lightly rolled and expanded in place before the tubes are inserted, while these should be swaged down so that they will just enter the ferrules. At the front end, on the other hand, they should be expanded and rolled out to fill the holes, care being taken that the ends are annealed before they are put in place. As to length, the tubes should be cut so that they will project from ⅛ to 3/16 inch beyond the sheets at each end. A satisfactory system of tube-setting is to turn back about 10 per cent of the tubes at each end to be beaded over in order to act as stay tubes, and then to round up the edges of all the tubes with a mandrel. Of course it goes without saying that the rolling of the tubes should be carefully done, and if necessary, should be repeated at the firebox end after the boiler has been fired and tested.

Turning back to summarize the stresses that have been given as allowable for staybolts and other related and similar parts, experience has shown that for those in the side, front and back water legs the load should not exceed 5500 pounds per square inch of section. For radial stays they are longer and are not subjected to such excessive bending stresses, due to the variation in the expansion of the two plates that they connect, the load may be raised to 8000 pounds per square inch, while on the other bracing on which there is little or no bending stress, a working load of 10,000 pounds per square inch may be imposed.

In the firebox the common American practice on all engines burning bituminous coal is to use the finger-grate type of bar having openings between each bar of from ¾ inch to ⅞ inch in width. Such a grate is shown in Fig. 9. As for the brick arch it may be considered to be practically out of use on all wide firebox engines like those which we are now considering in detail. For the narrow firebox type of boiler, however, the brick arch offers a very important advantage. The reason for this variation of practice in the two types is due to the fact that the movement of gases in the large box is slower and combustion more perfect than it would be in the narrow construction, if this latter be without the assistance of the arch for mingling and maintaining a high temperature of the gases.

CHAPTER III

THE CYLINDERS*

Closely allied with the boiler in importance are the cylinders. As the boiler is the important element in converting the potential energy of the fuel into that of the steam, so the cylinders serve as the means of converting this potential into dynamic energy and thus produce the useful work for which the machine, as a whole, is designed. The matter of the size of the cylinders has already been considered, in the determination of the general dimensions of the engine, where it was found that a diameter of 21 inches, and a piston stroke of 26 inches would be suited to the work that it is intended that the freight or consolidation engine should perform. At the same time the diameter of the cylinder and stroke of the piston of the passenger engine were calculated to be 19½ inches and 26 inches respectively. The cylinders are invariably made of cast iron, and it is of great importance that the metal should be of a character suitable for the work that it has to perform. It should be of a fine grain and as hard as can be worked, the latter quality being needed in order that it may withstand the hard wear of the pistons and valves. A common form of specification is to require that the "cylinders shall be made of a hard, compact, tough iron, of not less than 25,000 pounds tensile strength per square inch, and so cored as to produce uniform shrinkage in cooling." A metal that will meet these requirements can be made from 50 per cent pig, 25 per cent of old carwheels, and 25 per cent of high grade machinery scrap.

The practice of the several builders varied somewhat in the past in the matter of the form of the cylinder and saddle. At one time the saddle was made a separate casting, with the cylinders bolted on outside the frames. In this case the steam and exhaust connections to the smokebox were made either direct from the cylinder casting or from the side of the steam chest. This practice was succeeded by the almost universal adoption of the cylinder and half saddle cast in one piece as shown in Figs. 10 and 11. It will be observed in the two cylinders here illustrated, that Fig. 10 is adapted to the use of a flat slide valve, while Fig. 11 is fitted with a bore for a piston valve, the former being intended for use on a consolidation freight locomotive and the latter on the Atlantic passenger engine. The reasons for this variation in practice will be explained later. The length of the cylinder is dependent upon four factors: the stroke, the thickness of the piston, the clearances at the end, and the amount of counterbore allowed for the inset of the cylinder heads. The length of the working barrel of the cylinder is usually made equal to the stroke of the piston plus the width over the piston packing rings less ¼ inch. This last

* MACHINERY, Railway Edition, January. 1905.

subtraction is made so that the piston rings will travel beyond the edge of the counterbore at each end by ⅛ inch and thus prevent the formation of shoulders at the end of the stroke due to the wear of the cylinder barrel. This, of course, involves the determination of the widths of the packing rings, their number and the spaces between them, but this will be referred to later, it being sufficient to state

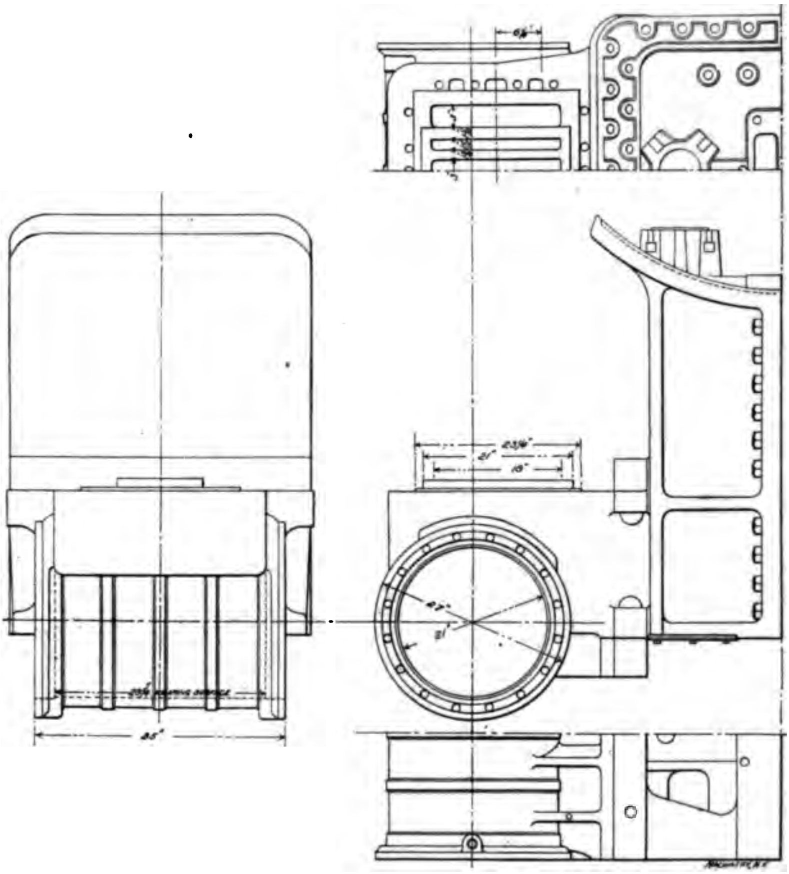


Fig. 10. Cylinder and Half Saddle for Consolidation Locomotive

here that the width of the pistons over the packing rings, in the case of those used in the two cylinders here shown is 3½ inches, which, with the stroke of 26 inches and the subtraction of ¼ inch, will make the bore of the cylinder 29¼ inches long in each case. The inner edge of the ports should come nearly opposite the outer edge of the packing rings at the end of the stroke, which places them 29¾ inches apart. The ends of the cylinders outside the working barrel are usually bored from ½ inch to 1 inch larger in diameter than the barrel itself to allow for wear and reboring without interfering with the fit of the cylinder heads. Sometimes, as in the case of Fig. 11, the cylinder is bored through from end to end to the full diameter of the counterbore and a bushing is inserted which can be renewed when it is worn, without disturbing or changing the dimensions of the pistons and packings.

As already indicated, the steam ports and cylinder heads enter the counterbore at the ends which usually extend from 4 to 5 inches beyond the actual stroke of the piston. In Fig. 10, this extension is 4½ inches, making the total length of the cylinder 35 inches, while in Fig. 11 it is 5 inches, making that cylinder 36 inches long. Subtracting from this distance the total thickness of the pistons (5¼ inches for the freight and 5½ inches for the passenger locomotive), we will have 3¾ inches and 4½ inches respectively for the clearances and the counterbore for the heads. It is desirable that the clearances should be as small as possible, though it is necessary that they be large enough to avoid all possibility of the pistons striking the heads. This will frequently be sufficient to take care of the port and steam requirements when the possible variations of motion due to faulty workmanship, wear, and changes effected by the keying of the rods, are guarded against. At times, however, this is not the case, since it has frequently been found to be desirable to give a high-speed engine a somewhat greater clearance than a slow one because of the longer period during which the steam is worked expansively, followed by an earlier compression. The cylinder heads should be so designed and proportioned that they will extend into the counterbore of the cylinder sufficiently for the clearance left between their inner faces and the piston, when the latter is at the extreme end of the stroke to be not less than ¼ inch nor more than ⅜ inch. This clearance should be so divided that when the new engine is turned out of the shop there will be from 1/16 inch to ⅛ inch more clearance allowed at that end of the cylinder toward which the wear of the rod brasses has a tendency to draw the piston, than at the opposite end. As to whether this will be the front or the back will depend on the details of the ends of the main rods.

The design of the cylinders is so closely allied to that of the valve motion, and so much depends upon the proper distribution and flow of the steam, that the size and arrangement of the ports will be considered in connection with the valves. In the actual construction of the cylinder, however, the ports as they enter the barrel are usually about 2 inches shorter than the diameter of the bore, increasing somewhat in length at the valve in flat slide valve engines, so that they are a trifle longer or about the same length as the nominal diameter of the cylinder. In the case of the piston valve the length of the port as it enters the cylinders is of about the same proportion as before, but is narrowed as it approaches the steam chest on account of the diameter of the latter being less than that of the cylinder. Here the steam ports extend entirely around the steam chest, being interrupted at a few points for the insertion of bridges to carry the ing rings of the valve over the openings. In laying out the

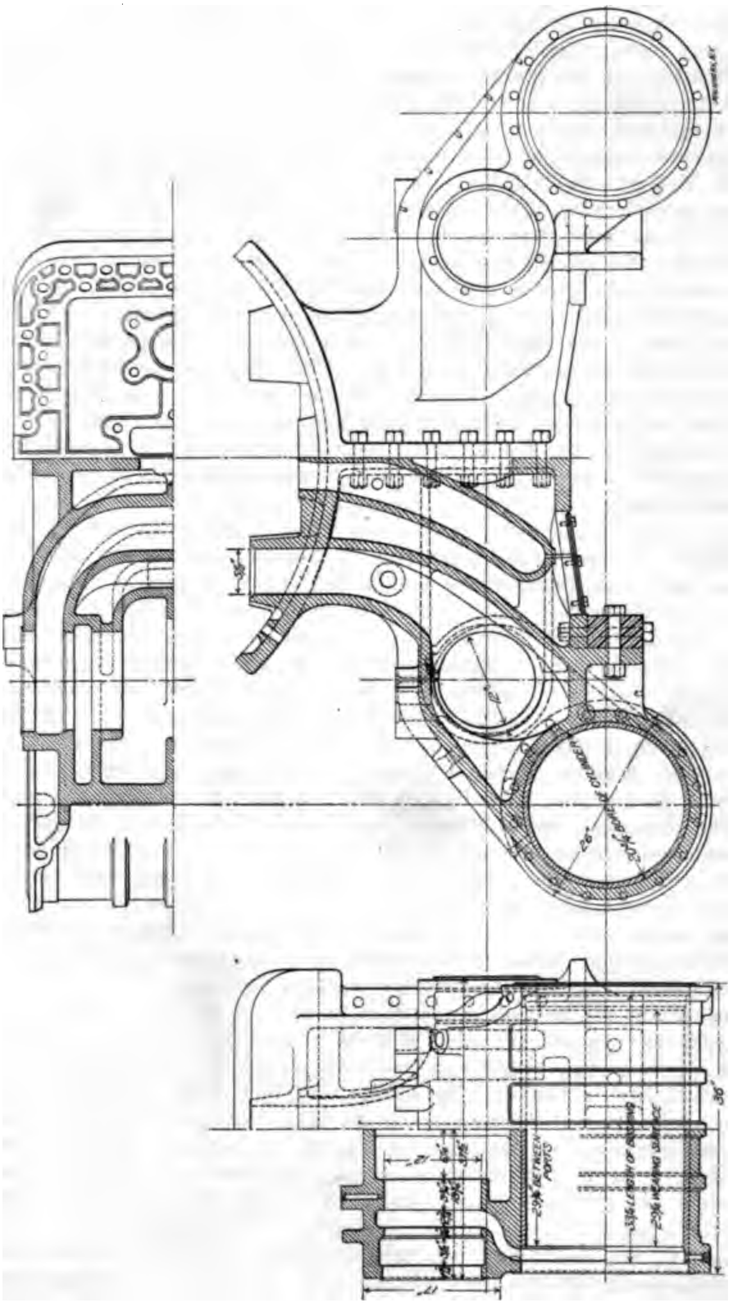
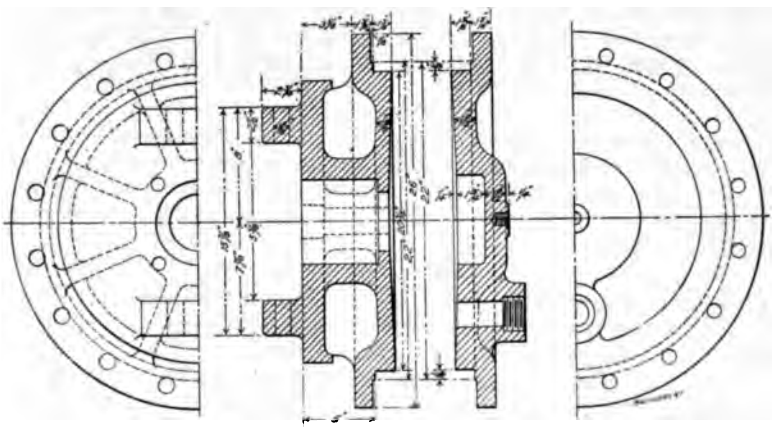


Fig. 11. Cylinders and Saddle of Atlantic Type Locomotive

course of the ports care should be taken that they are of a uniform section, as short and direct as possible, well rounded on all curves and turns, and with the turns of as long a radius as possible. These are matters of the utmost importance, since it is essential, for an efficient action of the machine, that the flow of the steam from the valve to the cylinder should be free and unimpeded, and devoid of eddies and cross currents which will tend to reduce the speed of such flow. This is especially important in the case of high-speed engines where any checking of movement of the steam will have a very marked effect on the work than can be done. Where the ports widen out in the saddle to meet the steam and exhaust pipes, they should be clean and roomy, and this is especially true of the exhaust ports. It has been found to be fully as difficult, if not more so, to get rid of the steam after it has done its work in a high-speed locomotive as it is to get it into the cylinder in the first place; hence the exhaust passages should be designed with ample spaces and easy curves that there may be the minimum amount of resistance set up to the escape of the low-pressure steam found in the cylinder at the end of the stroke. In short, every effort should be made to reduce back pressure to the lowest point.

The steam passages should be protected in every possible way from radiation. Under no circumstances should they be allowed to pass along the outer walls of the saddle, but should always be protected by an air space insulating them from the cold outer air. It is well, too, to add to this the further protection of some non-conducting material. Not only should these air spaces separate the steam passages from the external walls, but they should be so arranged as to isolate them, whenever practicable, from the exhaust passages wherein the temperature is much below that of the steam at boiler pressure. In fact, the end and aim should be to deliver as many heat units as possible to the cylinder, for it is upon this that the efficiency of the engine largely depends. In addition to this care by insulation, the steam passages should be laid out in an elastic curve, so that when their walls are heated and cooled by the admission and withdrawal of the steam, the expansion of the metal may not be such as to put any additional stress upon the other parts of the casting. This is especially necessary at that portion of the saddle near the frame fits, which must be ribbed in order to withstand the working stresses.

As for the thicknesses of metal to be used in the saddle, there are reliable formulas available whereby a mathematical calculation of can be made. Experience has shown, however, that too great a kness is detrimental to the securing of the greatest strength. This articularly true where there is an abrupt change in thickness, as e casting is apt to crack at such a point. All outside ribs should be avoided, since they have the double disadvantage of being liable to fracture and serving as a good radiating medium for such heat as the saddle necessarily takes up from the steam. Furthermore, the corners should be well rounded so that both the external and internal stresses ay be evenly distributed over the large surfaces. The flanges for bolting the cylinders and half-saddles together and to the boiler must be made heavy and be ribbed so as to withstand the pressure and driving of the bolts. Here, too, the metal should be arranged so as to join the walls with curves of long radius and large fillets. The walls should be well braced internally with ribs so as not to yield at the root of the flanges. No formula is available for calculating the thickness of these ribs since no one knows exactly, or even approximately, the stresses to which they are subjected. They are ordinarily made from 2½ inches to 3 inches thick, which, in the cases before us, is taken at 2¾ inches.



Figs. 12 and 13. Back and Front Cylinder Heads for Atlantic Type Locomotive

About the only point in the design of the cylinder that is subjected to a mathematical analysis is the thickness of the shell of the barrel, which may be calculated from the formula

$$t = \frac{dp}{2S} \qquad (10)$$

in which

t = thickness of the shell in inches,

d = diameter of the bore in inches,

p = steam pressure in pounds per square inch,

S = safe stress in metal in pounds per square inch.

Allowing 2500 pounds per square inch as a safe working stress for the metal, and substituting the values that have been already found for the two engines, the formula becomes

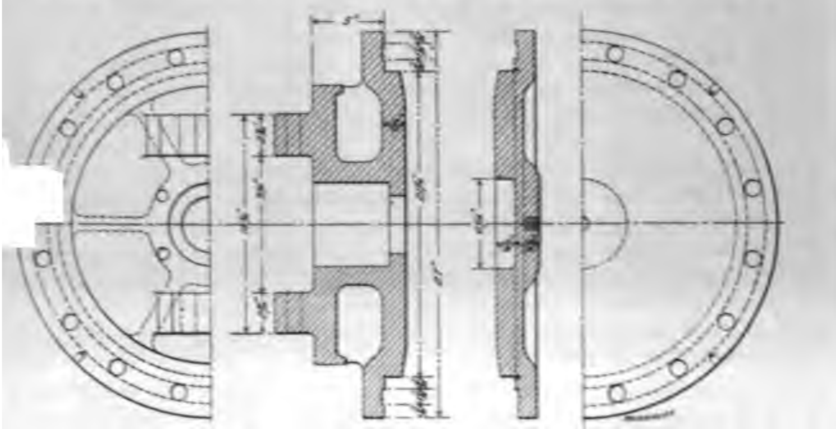
$$t = \frac{21 \times 200}{2 \times 2500} = 0.84 \text{ inch}$$

for the consolidation engine, and

$$t = \frac{19.5 \times 200}{2 \times 2500} = 0.78 \text{ inch}$$

for the Atlantic type. Unfortunately the resistance to the pressure of the steam is not all that is required of a cylinder. It is subjected to wear, and provision must be made for reboring, besides which it must sustain many of the shocks to which the locomotive is subjected while in motion. Hence, while the calculated thicknesses are a trifle less than ¾ inch it will be found advisable to increase this by 50 per cent or more and make the shells from 1¼ to 1½ inch thick.

The cylinder heads are usually simple in construction and are bolted in the flanges cast at the ends of the cylinder by studs spaced about 5 inches apart from center to center. In the case of the front head these studs have no work to perform other than to withstand the pressure of the steam against the head. At the back they have not only this, but the indeterminate stresses due to the support of



Figs. 14 and 15. Back and Front Cylinder Heads for Consolidation Freight Locomotive

the front end of the guides and the varying loads that they transmit to the heads. These studs are usually made 1 inch in diameter for the cylinder sizes that are here considered, so that with sixteen studs to withstand the pressure in a counterbore 21¼ inches in diameter, the load on each stud would amount to less than 5700 pounds per square inch of section of the metal, hence there is an ample margin beyond for safety. The thickness of metal in the heads must be taken independently of any stress that may be put on them by the steam pressures. The guides must be supported, and there must be provision on the back head for the packing box of the piston rod. Besides this the heads are exposed to external shocks and blows, so that while far in excess of the requirements for resisting the steam pressures, the heads can well be made from 1¼ to 1½ inch thick. As the heads are especially exposed to the action of the wind, they must be well protected against radiation of heat. At the front, a plain disk head is generally used which is protected by a casing, enclosing an air space in front of the flat surface. Sometimes this space is filled with a non-conducting material that serves to considerably lessen the radiation. At the back, by the use of a two-bar guide and the flange of the packing box, the construction lends itself very readily to the formation of an air space and pockets for insulating material in the head itself. This is clearly shown in Figs. 12 and 14 of the back heads for the two engines.

Let us briefly review the work to be done on the cylinder and heads. The seat for the smokebox is cast with chipping strips that are chipped to the proper radius, while the two parts (smokebox and saddle) are held together by bolts. The connection between the steam passage and the steam pipe in the smokebox is made by means of a ground joint. The abutting surfaces of the two half-saddles are accurately planed and are held together by bolts that are usually about 1¼ inches in diameter. In common practice these bolts are tapered about 1/16 inch to the foot and they are driven home. The surfaces for the bearing of the frames are planed and the frames and cylinders bolted together with tapered bolts turned to a driving fit. The details of these fastenings will be considered under the subject of the frames. The bore and valve faces of the cylinder are finished with a smooth-cutting tool and are left as they come from the machine. As the joint between the heads and the cylinder ends must be steam-tight, the two surfaces in contact are ground so that no packing is required. Of course in all this work the utmost care must be exercised that a true alignment of every surface is obtained, for without that not only would the working of the moving parts be defective, but the stresses set up would be excessive and the liability of the fixed parts to work loose be greatly increased.

CHAPTER IV

THROTTLE VALVE—DRY AND STEAM PIPES*

With the four principal items in the construction of the engine decided, namely: the weight, wheel arrangement, boiler, and cylinders, the following steps will be that of the working out of the various details upon which the successful operation of the locomotive depends. Closely connected with the work done in the cylinders and boilers are the means by which the steam generated in the one can be conducted to the other so as to perform its proper functions. In this it is of the utmost importance that the flow of the steam from the boiler to the cylinder should be free and unimpeded and that there should be the minimum drop in pressure when it reaches the steam chest. There must necessarily be some drop, else there would be no flow, but this will decrease as the size of the passages is increased.

The proper sizes of the throttle and the dry pipe are determined, like so many other things in locomotive practice, not so much by a theoretical calculation as by an empirical formula deduced from practice, that has been shown to give satisfactory results. As already stated, the flow of the steam must be free enough to maintain a high pressure in the steam chest, and yet the limits of space in the boiler and smokebox cut down the available dimensions of the pipes to a low figure. In this, too, the designing of a locomotive differs from that of a stationary engine in that the latter is built to run at a constant speed, and with a comparatively uniform degree of admission in the cylinder, so that the steam pipe can be proportioned accordingly and be made to deliver a constant quantity of steam at a uniform rate of flow. In the locomotive, on the other hand, the steam consumption varies between wide limits and is greatest at high speeds when the earliest cut-off is used.

It has been found, then, that the sectional area of the throttle and dry pipe should not be less than one-fifteenth (1/15) that of both cylinders, while the steam pipes in the smokebox should not be less than one-twelfth (1/12) the sectional area of one cylinder. The reason for this apparent discrepancy is that when the engine is working slowly, an area of one-fifteenth can deliver all of the steam required; while, when it is running at a high speed, the point of cut-off is moved back until it never occurs later than at half stroke, so that only one cylinder is taking steam at a time. This makes the dry pipe proportionately larger or raises it to two-fifteenths of the area that it is obliged to supply.

Referring these proportions to the two engines that we have in hand, it will be seen that in a consolidation freight locomotive having wheels 57 inches diameter, and a piston stroke of 26 inches, the maxi-

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* *Machinery*, Railway Edition, March, 1905.

mum velocity of the piston with the engine running at a speed of ten miles an hour will be about 6 2/3 feet per second, so that at times the flow of steam through the dry pipe would be fifteen times this, or about 100 feet per second. Were the engine to be running at the rate of thirty-five miles an hour the velocity of flow would only be increased to 175 feet per second, because under these circumstances only one cylinder would be taking steam at a time. In the case of the Atlantic passenger locomotive, with 77-inch drivers and a 26-inch

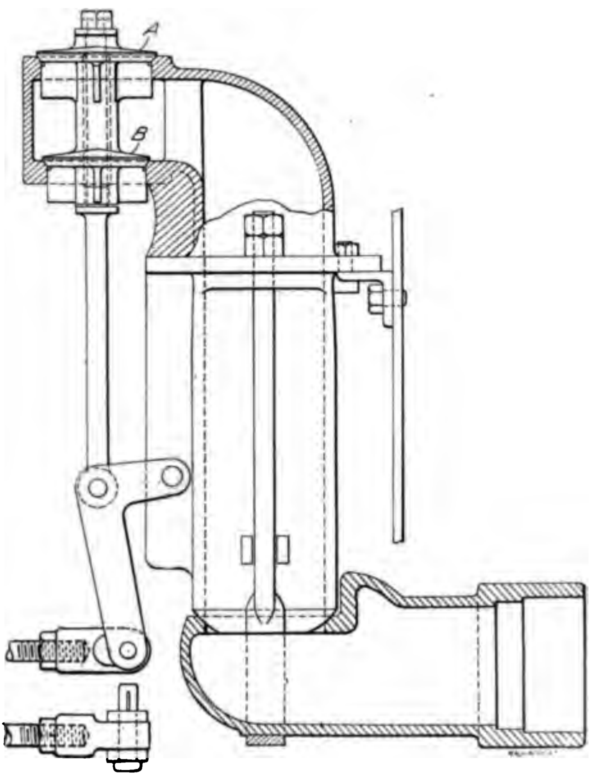

Fig. 16. Section of Ordinary Type of Throttle Valve

stroke of piston, the velocity of flow through the dry pipe would be, at speeds of fifteen and sixty miles an hour, 111 feet and 223 feet per second respectively.

Such a velocity does not obtain in practice, however, even though the proportions given are maintained, since the steam contained in the steam chest and pipes expands somewhat during admission and is later replenished during expansion and compression; the result is a practical uniformity of velocity of flow through the throttle valve and pipes. Consequently the actual speed is much below that given by the figures above and does not increase in the direct proportion of the piston speed, as might be expected.

As for the types of throttle valve, dry pipe and steam pipes that are to be used, there is little variation in current practice. The double-seated balanced type of poppet valve is universally used with some slight variations in the details of its construction.

The ordinary throttle valve is shown in section in Fig. 16. In this the two valves, *A* and *B*, close from the top and when raised admit steam to the throttle casting at the top and bottom. In order to assemble this valve it is necessary that the upper should be the larger

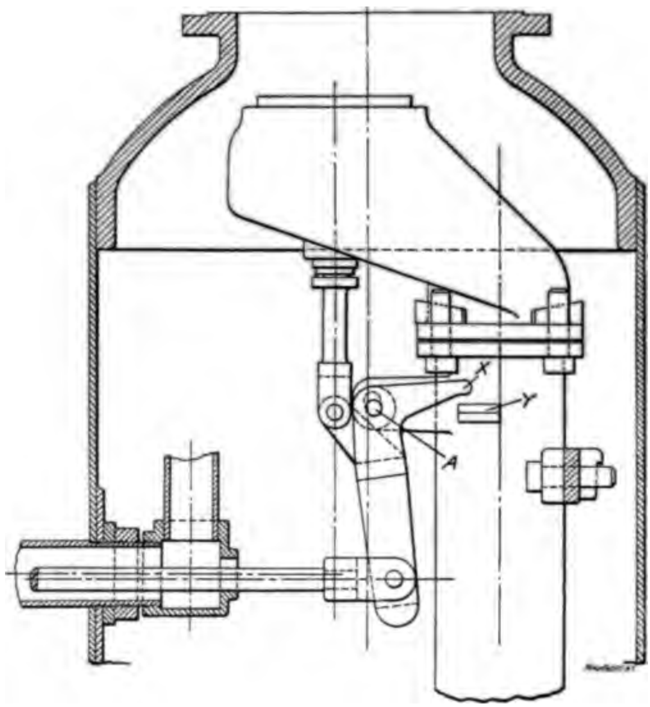

Fig. 17. Section of Throttle Valve with Variable Fulcrum

of the two, so that the lower may be put in from the top. This destroys the perfect balance, as the steam, acting upon the greater area of the upper disk, tends to close the valve and will overcome that against the bottom face of the lower disk. This prevents the valve from opening accidentally. The casting may be supported from a bracket bolted to the inside of the dome. As for the inside diameter of this casting, the proportions given above make it a little more than 7½ inches for the consolidation locomotive with 21-inch cylinders, and a little less than 7⅛ inches for the Atlantic type with 19½-inch cylinders. Hence a casting with 7¼ inches diameter of opening can well be adopted for the two. As a matter of fact these dimensions will be found to vary slightly in practice to meet the exigencies of other requirements that may come up in the working out of the details.

In this connection attention may be called to the fact that the departure from an exact balance of the throttle valve due to the variation in the diameters of the two parts, may cause a considerable preponderance of closing load, especially when high steam pressures are used. This frequently makes it somewhat difficult to open the throttle. In order to avoid this, a system of levers has been designed like that shown in Fig. 17. In this the bell-crank has a double fulcrum. When closed and about to be opened, the bearing is on a pin, *A*. This gives a long leverage and a powerful purchase to assist in the opening of the valve. As the bell-crank turns, the arm *X* comes in contact with the

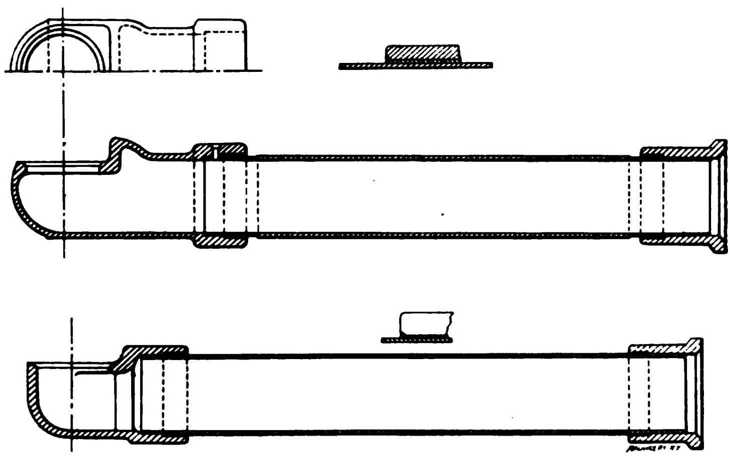

Fig. 18. Dry Pipes for Locomotives

lug, *Y*, after the valve has started and then serves, by the shorter leverage given, to cause the valve to move more rapidly for the same amount of movement of the throttle stem, the pin *A* rising from its bearing in the slot in the main casting.

The dry pipe is usually made of wrought-iron pipe with cast-iron ends riveted on. The dimensions called for will be the same as those of the throttle casting. But as wrought-iron pipe of these approximate dimensions rises by inches in diameter there is nothing available for the purpose between the nominal diameters of 7 inches and 8 inches. It will, therefore, be necessary to use the former as being nearest to the estimated size, though slightly smaller. A common form of dry pipe is shown in Fig. 18.

It is evident that such a pipe as this cannot be put in through the dome, so it must be run into the hole in the front tube sheet which is, therefore, made large enough to pass the castings riveted to the ends. For a fastening and joint, the arrangement shown in Fig. 19 is used. The dry pipe, *A*, with its castings, is put through the hole in the tube sheet, a heavy reinforcing ring, *E*, having first been fastened against the inside of the sheet in order to increase its stiffness. The hole in the sheet itself is beveled to match the bevel on the casting. The latter is cut with a spherical ground joint on the outside to take the brass ring, *B*, which is also ground with a flat outer face to bear against the tee-head, *C*. The latter has flanges cast upon either side to which corresponding flanges of the steam pipes, *D*, are bolted. All of these joints are ground and fitted as no soft packing would be able to withstand the intense heat of the smokebox to which they are subjected.

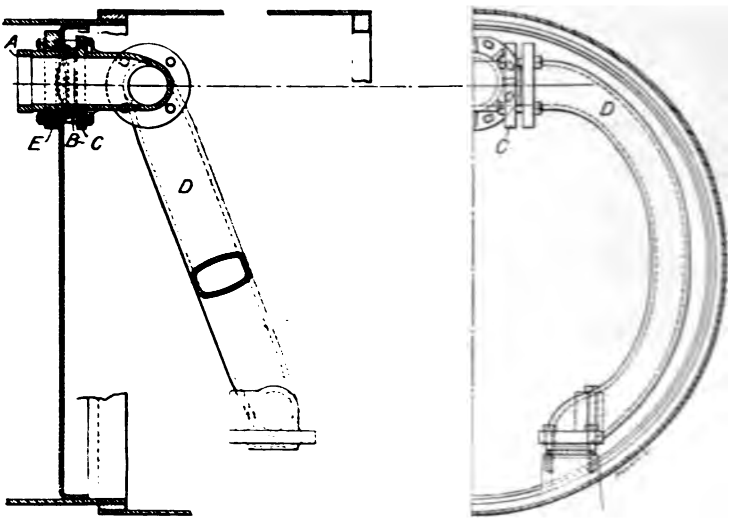

Fig. 19. Tee-head and Steam Pipes

It has already been stated that the area of the steam pipes is put at about one-twelfth that of one cylinder. Hence, for a cylinder 21 inches diameter, the corresponding area of steam pipes would be a little less than 29 square inches and for a 19½-inch cylinder it would be a little less than 25 square inches. Owing to the desirability of encroaching to as slight an extent as possible upon the diametral dimensions of the smokebox, these steam pipes are not made round except at the ends, but are flattened, as shown, so that they lie close to the shell with the longer dimension corresponding with the longitudinal dimensions of the smokebox.

The connection between the foot of the pipe and the cylinder casting is also made by means of a ground joint. The rest of the passage to the steam chest and cylinder has already been treated.

CHAPTER V

PISTON AND PISTON ROD*

Piston design for locomotive work has been subjected to many changes and there are several forms in use, all of which may be included in two classes: the built-up and the solid. The built-up form of spider bull ring with a follower permits of the removal of the packing without the necessity of removing the piston from the cylinder. Where, however, reduction of weight is of the first importance, a solid piston of a double Z-section, and made of either cast iron or steel casting is used, in which case the packing rings are usually made in one piece and sprung into place. In pistons of this type, when made

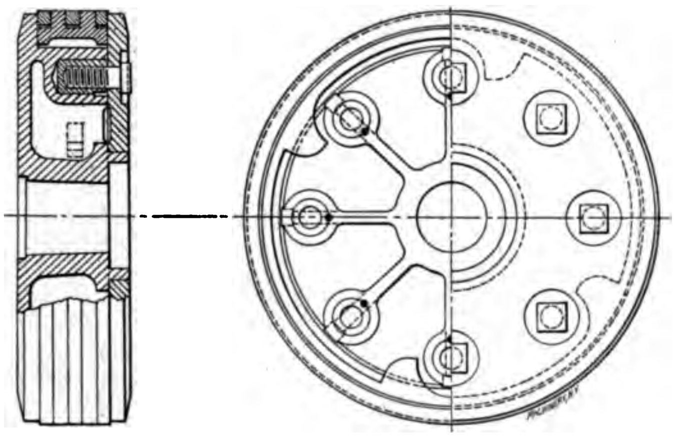

Fig. 20. Piston with Follower and Bull Ring

of cast steel, the rim is usually surrounded by a cast iron or brass ring, the former being fused or bolted to the body, so as to secure a better wearing surface against the cylinder than the steel would afford. The fused rim is, however, much to be preferred as bolting weakens and adds to the weight of the piston. The first type with the follower and bull ring is shown in Fig. 20, and the second in Fig. 21.

The stresses to which a piston is subjected are of two kinds: one is that of the punching or shearing of the disk about the boss, due to the steam pressure exerted upon it between the rim and the hub; and the other the breaking stress exerted across the diameter. The first should, therefore, be treated as a combination of bend and shear but must be estimated separately. The direct shear alone may be easily provided for by laying out several concentric circles with radii of

* Machinery, Railway Edition, March, 1905.

r, r′, r″, etc., and subtracting the area multiplied by the pressure inside the circle under consideration from the total pressure on the piston and dividing the remainder by the allowed strength of material per square inch of section. This will give the total area of metal to be used; which, in turn, being divided by the circumference of the circle will leave a quotient equal to the thickness of the metal to be used at the point in question, or the minimum thickness of metal to be used in that particular circle. The first part of the problem may be expressed by the formula:

$$A = \frac{\pi R^2 P - \pi r^2 P}{S} \tag{11}$$

in which

A = the required area of metal on the given circle,
R = radius of cylinder,
r = radius of section sought,
P = boiler pressure,
S = allowable stress on the material.

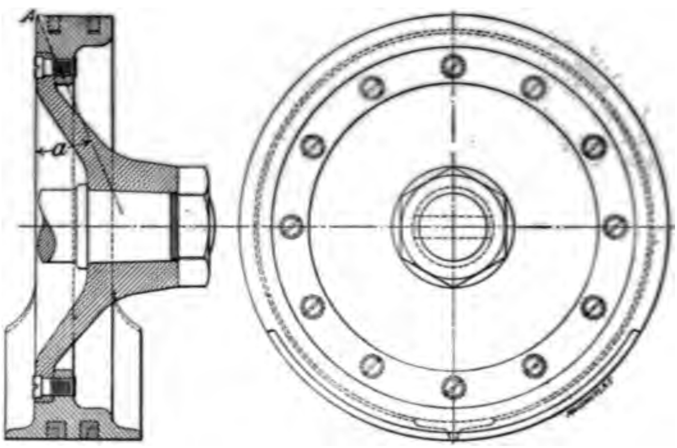

Fig. 21. Cast Steel Z-shaped Piston

The radial bending stress should be provided for by ample internal ribbing so as to maintain the requisite stiffness when double-bottomed pistons are used. The disk or single-walled pistons upon which no ribs can be used, should be so designed that the bending stresses are transformed into tension and compression as the loading alternates, and thus transfer them to those sections that would otherwise be subjected to a low stress, such as the rim. This may be accomplished by making the disk conical or of a double Z-section in which a smaller amount of material can be used than in any other form of piston in proportion to the strength developed. It is best, therefore, that the rim should be cast solid with the main body and be made of brass or cast iron so as to secure the full advantage of its ·ing qualities.

In providing for the diametral breaking stress across the disk, which by the way, usually provides for the other stresses, as in the one illustrated, the piston may be considered as a beam whose section is that of the piston through its center, with one-half of the total load concentrated on each half at the center of gravity of the same. The distance from the center of gravity to the center of the piston is 0.42 R. Then

$$S = \frac{0.42\, R \times R^2 \times P}{2M} \qquad (12)$$

in which

S = working stress of the metal,
R = radius of cylinder,
M = modulus of section.

M must be calculated for each section of piston to which it is desired to apply the formula.

For disk pistons there are a number of empirical formulas in use. One is to take the thickness near the boss as:

$$t = c D \sqrt{p} \qquad (13)$$

D = diameter of piston in inches,
p = pressure per square inch,
t = thickness of metal, and
c = 0.0046 for cast steel, and
c = 0.008 for cast iron.

The thickness of the plate near the rim may be taken as 0.6 the thickness at the boss. Calculation now gives for a piston 21 inches diameter a thickness at the boss of about 1⅜ inch and 13/16 inch at the rim for steel; and 2⅜ inches and 1⅜ inch at the boss and rim respectively for cast iron. For the calculation of the thickness of the conical portion of a Z-shaped piston, Unwin gives the following formula for cast steel pistons:

$$t = 0.003\, d \sqrt{p} \qquad (14)$$

in which

t = thickness of metal,
d = diameter of the cylinder,
p = pressure in pounds per square inch.

For a 21-inch piston working under 200 pounds per square inch of steam pressure the thickness would be 0.88 inch.

After having passed through the various forms of rings expanded by steam and springs, practice has returned to that of a simple split ring held out by its own elasticity. The formulas that are given for the width and thickness of these vary between wide limits, ranging on a 20-inch piston ring from ⅜ inch to 3 inches. In practice it has been found that on pistons of 18 to 20 inches diameter the width of the rings may be about ½ inch, and for pistons of from 21 inches to 24 inches it may be about ⅝ inch with a thickness ⅛ inch greater than the width; or the width may be expressed approximately as about ⅛ inch less than one-thirtieth the diameter of the cylinder. In a general way it may be stated that the piston packing in ordinary use is that of a snap ring type of from ½ to ⅞ inch square section to suit the size of the cylinder and of a reasonable tightness. Such rings possess the advantage of simplicity of manufacture and ease of maintenance, and are therefore to be preferred. The only other packing in use is the Dunbar, which still remains a favorite on some roads. It is made in small segments of one L-shaped and one rectangular section, fitted into each other so as to form a ring of square section with the several parts overlapping at the joints. The main objection raised to it is its cost. Great width of packing rings is not required, it being simply necessary to get a good contact around the whole surface, and this can be obtained if the elasticity of the ring is sufficient to give an outward pressure when sprung into position in the cylinder. The size, therefore, is somewhat dependent upon the character of the metal of which the ring is made. It is, of course, desirable to avoid any excess of pressure on account of the wear that would result. The number of rings used varies from two to three with different designers.

The total width of the piston may be placed at approximately one-quarter the diameter of the cylinder, though there are variations from this, the thickness usually being less where but two packing rings are used, when it may be one-fifth.

Piston Rod

Closely allied with the calculations for the piston are those for the rod, the section of which is, to some extent, governed by the areas through the cotter hole or the bottom of the thread at the smallest part of the rod, where it is attached to the crosshead. As this section of the rod must withstand the full boiler pressure on the piston, it should be of such section that the stress will not exceed 10,000 pounds per square inch of metal. As the piston is usually tapered in its fit in the crosshead the main body will be large enough for truing or turning up for wear in the stuffing-box. Still the dimensions should be carefully checked for safety, and the determination of the size of the cotter or key is of the first importance. These cotters are usually subjected to a stress in one direction only, as the taper or shoulder on the rod takes that in the other. It is, however, well to calculate this stress as equal to the steam pressure on the whole surface of the piston. The proportions of the cotter must, then, be such as to sustain this full load both in shear and compression. For shear a steel cotter may be safely subjected to 10,000 pounds per square inch of section and to 24,000 pounds in compression. Taken in single shear, the stress will be one-half the total load and the section resisting this will be equal to the thickness multiplied by the width. We thus have the formula:

$$W = 2btS_1 \qquad (15)$$

in which

W = load on the cotter,

= width of cotter,

ness of cotter,

stress on metal.

In the case of a 21-inch cylinder working under 200 pounds pressure this formula becomes:

$$69{,}272 = 20{,}000bt,$$

or

$bt = 3.46$ square inches.

If then t is assumed to be ¾ inch, b becomes 4.61 or 4⅝ inches.

The next and most important step in the matter is to determine the diameter of the body of the piston rod. The stresses to which this member is subjected are both in tension and compression; and each must be taken into consideration in calculating the proper diameter. For tensile stress we have

$$d = 2\sqrt{\frac{W}{\pi S}} \qquad (16)$$

in which

d = diameter of the rod,

W = total load on the piston,

S = allowable working stress on the material = 10,000 pounds per square inch.

By substitution, we have, for a 21-inch piston

$$d = 2\sqrt{\frac{69{,}272}{31{,}416}} = 3 \text{ inches nearly.}$$

The determination of the compressive strength is a much more complicated matter owing to the fact that the piston rod cannot be considered as a strut pure and simple, because of its motion and because of the bending stresses that are put upon it by the looseness of the crosshead in the guides and the piston in the cylinder. A number of empirical formulas have been proposed for this work, but none of them has as yet been accepted as adapted to all conditions on account of the variation in the results that have been obtained in experiments conducted upon a large scale. It is generally considered, however, that when the length of a column is less than twelve times its diameter, the compression can be safely placed at the same figure as the tension, and this is the case with nearly all locomotive piston rods. Cases, however, may arise when the compression would be excessive and, in such, it is well to refer to Rankine's formula for columns with round, free ends, which is

$$\frac{P}{A} = \frac{S}{1 + \dfrac{q\,l^2}{r^2}} \qquad (17)$$

in which

P = the total load,

A = area of the section of the rod,

S = ultimate compressive strength of the material = 150,000 pounds per square inch,

l = length of rod in inches,

r = radius of gyration $= \dfrac{d}{4}$,

$$q = \frac{4}{25,000} \text{ for steel,}$$

$$q = \frac{4}{36,000} \text{ for wrought iron.}$$

This formula can be readily changed to terms indicating the diameter, by substitution, and a proper allowance of a factor of safety for S.

A similar formula of Merriman is

$$C = \frac{B}{1 - \frac{n B}{10 E} \times \frac{l^2}{r^2}} \qquad (18)$$

in which

C = the maximum compression stress per square inch of area,
B = the load per square inch of section of the rod,
E = modulus of elasticity = 30,000,000 for steel; 25,000,000 for wrought iron,
n = 1 for round end bearings and ¼ for square ends.

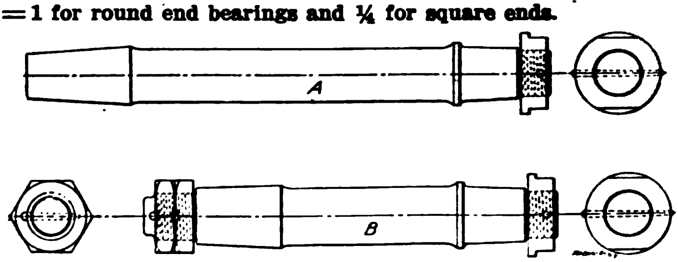

Fig. 22. Piston Rods for Locomotives

For an approximate estimate, we may use the formula:

$$W = \frac{1125 \pi d^4 S}{4500 d^2 + 4 l^2} \qquad (19)$$

in which

l = the length of the rod in inches,
S = allowable stress per square inch of section.

For the determination of l it will be necessary to lay out the rod on the drawing board so as to allow for the necessary clearances, and the length, in the case of the consolidation cylinder with 26 inches stroke, will be found to be about 38 inches from the back face of the piston to the boss on the crosshead.

Then assuming the maximum fiber stress to be 9000 pounds per square inch Formula (19) becomes

$$69,272 = \frac{1125 \times 3.1416 \times d^4 \times 9000}{4500 d^2 + 4 \times 1444}, \text{ from which } d = 3.3.$$

As some allowance must be made for the truing up of the rod on account of wear the addition to this amount will depend upon the personal opinion of the designer, modified, to an extent, by the character

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NUMBER 28

LOCOMOTIVE DESIGN

By GEO. L. FOWLER and CARL J. MELLIN

Part II

VALVE MOTION

SECOND EDITION

CONTENTS

49-55 Lafayette Street, New York City

CHAPTER I

THEORY OF VALVE MOTION*†

Next in importance to the boiler in determining the efficiency of the locomotive as a whole, is the valve motion, and too much stress cannot be put upon the value of a proper design for this element in the machine. The Stephenson link motion, which may be said to be the one universally used upon American locomotives, possesses the peculiarity of being exceedingly sensitive to a close adjustment of all of its parts in order that a correct action and proper distribution of the steam may be obtained; while with the roughest kind of haphazard design or no design at all, it will do its work after a fashion and make the wheels go round. It is evident, however, that in order that the steam distribution in the cylinder may be as efficient as possible for all speeds and all points of cut-off, the utmost care must be exercised in the designing of the valves and machinery by which they are driven.

Up to within a few years the flat side valve was in universal use. At first it was the common unbalanced D-valve, which was followed, as the steam pressure was increased, by the balanced valve, while the current practice, where high pressures are used, lies between the flat balanced valve and the piston type. The piston valve possesses some advantages over the flat valve in that it is fully balanced, though the slide valve can be quite satisfactorily designed in this respect. In short, it is a matter of choice and convenience of construction and maintenance as to which shall be used, though the tendency of modern practice is toward an increasing use of the piston valve.

The main difference to be considered in the designing of the two types of valves is that while the flat slide valve is invariably arranged for an outside admission of steam to the cylinder, the piston valve should be arranged for an inside admission. This is not absolutely necessary, but the advantages are that the steam passages can thus be better protected from the cold and radiation, and the steam chest heads and packings relieved from all pressure except that of the exhaust steam, which is but a few pounds above that of the atmosphere and so puts really very little stress upon these parts. If the piston valve is so designed that the admission is on the inside, it should be made hollow with as large a passage through the center as possible.

* The present number of MACHINERY's Reference Series is the second part of a treatise on complete Locomotive Design, covered by Nos. 27, 28, 29, and 30 of the Series, and originally published in RAILWAY MACHINERY (the railway edition of MACHINERY). Each of the four parts of the complete work treats separately on one, or more, special features of locomotive design; and while the four parts make one homogeneous treatise on the whole subject, each part is complete by itself. In order to give concrete form to the examples and theoretical considerations, it is assumed that a consolidation freight locomotive and an Atlantic type passenger engine are being designed. It is further assumed that these locomotives are designed for a division 150 miles long, laid with rails weighing 75 pounds per yard, and with a ruling grade of one per cent ten miles in length.

† MACHINERY, Railway Edition, April, 1905.

The object of this is to secure a large area for the movement of the exhaust, so that, at the instant of release, the pressure may be reduced to a minimum. The use of the hollow valve facilitates this by permitting an escape through the exhaust passages at each end; and these latter should be carried to the base of the exhaust pipe and made to meet at a very acute angle so that the two currents will combine without forming obstructive eddies and thus aid in the fanning of the fire. Such a combination will be found to permit of the use of a larger exhaust nozzle than is possible with a solid valve.

As for the mechanisms by which the valve is moved, there are four, or as they are sometimes classified, five different systems that are in use upon locomotives, and which are named after their designers, *viz.*: the Stephenson, Gooch, Allan, Walschaerts, and Joy.

The first is known as the shifting link motion, and is that mostly in use in the United States. The second is directly opposite in its

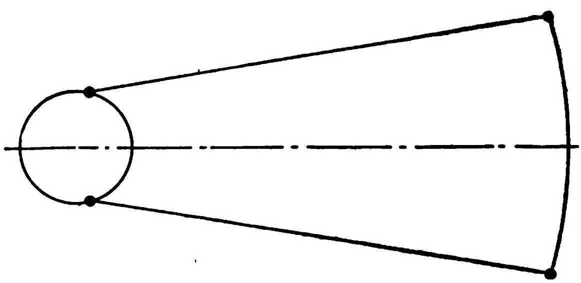


Fig. 1. Open Eccentric-rods

action, in that the link is stationary and the link-block, attached to the valve-rod, is moved up and down. The Allan is a combination of the first two in that both the link itself and the valve-rod are shifted, by which it becomes possible to make the link straight. This, of course, greatly simplifies the construction and maintenance of the link, and the motion is extensively used in Europe. All of these motions are operated with two eccentrics, one for the forward and the other for the backward motion.

In the Stephenson and Allan motion when the eccentric-rods are open, as in Fig. 1, the lead is increased as the link is hooked up and the point of cut-off made earlier. If, however, the rods are crossed as in Fig. 2, the hooking-up of the link reduces the lead, though this reduction is much less than the increase in the former case. Again, with either open or crossed rods, the corresponding increase or reduction of the lead is much less with the Allan than with the Stephenson valve motion. With the Gooch, Walschaerts, and Joy motions the lead is constant for all points of cut-off.

It will be seen, by reference to the diagrams Figs. 1 and 2, that the eccentric-rods are said to be "open" when, with the eccentric centers upon the same side of the axle as the link, the rods are not crossed; and "crossed" when the rods do cross in that position. This holds true whether the motion be transmitted direct to the valve stem or indirectly through a rocker arm. This term of direct or indirect application refers to the relative movement of the valve and the link-block. When they move in the same direction the motion is said to be "direct"; when in the opposite directions, "indirect."

The Walschaerts gear is driven by a combination of an eccentric or short-stroke return crank from the main crank-pin and a connection to the crosshead. The Joy gear is driven from a connection to the connecting-rod. The former has been extensively applied on the continent of Europe and the latter to some extent in England. The Walschaerts gear has recently been applied to several of the largest engines built in the United States with such pronounced success that the use is being extended, and some engineers have expressed the opinion that the prospects are that it will eventually take the lead over the Stephenson gear, as it has abroad, on account of the many advantages which it possesses and which have not been properly demonstrated in American practice until lately.

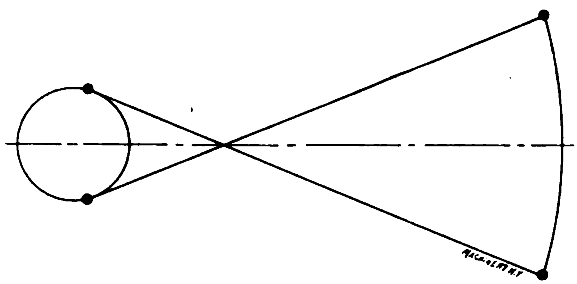


Fig. 2. Crossed Eccentric-rods

As the Stephenson motion is the one mostly used in this country it will be first considered. It is probably the most flexible of any in use, and can be most readily adapted to irregularities in the running and operation of the machine. At the same time it will get out of adjustment very easily and requires the utmost care in its designing in order that it may work properly. With this as an introduction we may now enter upon an examination of the motion and its requirements. To work out the valve motion theoretically and mathematically is a long and tedious operation. As its application to, and development for use upon, a locomotive involves a number of irregularities that are necessarily neglected, when treating the subject from a purely mathematical standpoint, this method will be simplified by use of diagrams by which the steps needed for its development for actual use will be shown.

In the first place, the introduction of a rocker-arm between the eccentrics and the valve serves as a means of increasing or reducing the travel of the latter as compared with the throw of the eccentrics as well as transferring the line of motion to suit convenience in locating valves and valve spindles. The angularity of the connecting- and eccentric-rods introduces irregularities that can, to a great extent, be compensated by the location of the link-saddle pin. The longer these rods can be made the better for the action of the motion, and it is well not to make them too short, when it would become impossible to compensate for the irregularities that would be introduced.

Experience and calculations have shown that, to secure a satisfactory action of the valve motion, the connecting-rod should not be less than six times the radius of the crank, that is, three times the piston stroke, and that the eccentric-rods should not be less than eight times the throw of the eccentric. As a matter of fact, the eccentric-rods are usually of a greater length than this. In the case of the link the radius should be equal to the distance from the center of the link-block, when in its central position, to the center of the axle, and the distance between the eccentric-rod pins should not be less than two and a half times the throw of the eccentric. If it is less than this the angle assumed by the link relative to the block will be such that the slip of the latter will not only be excessive, but it will be apt to stick and put undue stresses on the entire mechanism of the motion.

As already intimated, the irregularities introduced by the angularity of the rods may be compensated for by adjustment of the location of the saddle-pin, which will always be inside the center line of the arc. If, however, it is carried too far, it will give an objectionably long slip to the link, especially if the rod lengths are near their minimum. So, as this offset of the saddle-pin, as it is called, is less with an outside admission valve and an indirect motion, or, what is the same thing, an inside admission and direct motion, than where the contrary conditions exist, it is always best to use one or the other of these two combinations when no other advantage is to be gained by a reversal of the conditions. The adjustment of the valve also requires that particular attention should be paid to the lead that is obtained at full travel, as well as the increase resulting from a linking-up of the engine, so that, when running with an early cut-off the lead and pre-admission may not be excessive.

This can best be studied by a consideration of the effect of the movement of the link on the travel of the valve. With the link in full gear the block is usually so related to the pin of the corresponding eccentric-rod that its motion to and fro is equal to the diameter of the path of the controlling eccentric. Then, if the two rocker-arms are of the same length, the travel of the valve will also be the same. In other words one eccentric controls the valve and the other merely causes the link to oscillate about the block, as far as relative motions are concerned. But, when the link is raised, both eccentrics have an effect on the motion of the valve, and the resultant of this is as though another and controlling eccentric of a shorter throw were to be introduced between the two. The throw of this resultant eccentric decreases as the link is raised until mid-gear is reached, when the throw is at the minimum. Meanwhile its position has shifted from the center of the forward eccentric to a point in line with the crank and midway between the two actual eccentrics. At this point the radius is equal to the sum of the lap and lead in mid-gear. As the link is raised still further, the radius gradually increases and shifts its center until, in full gear back, it coincides with the center of the backing eccentric.

The center of this imaginary or resultant eccentric has then traversed a path that is a parabolic curve connecting the centers of the two eccentrics and passing through a point in line with the crank and distant from the center of the axle by an amount equal to the sum of the lap and lead in mid-gear. The height of this curve or its distance from a straight line connecting the two eccentric centers is equal to the increase of lead between full and mid-gear. For a given maximum cut-off and valve travel, the required angular advance and lap can be calculated, when no lead is allowed in full gear or an excessive pre-admission is to be avoided, by the following formula, based on a radius equal to 1:

$$\sin d = \sqrt{1-p} \qquad (1)$$

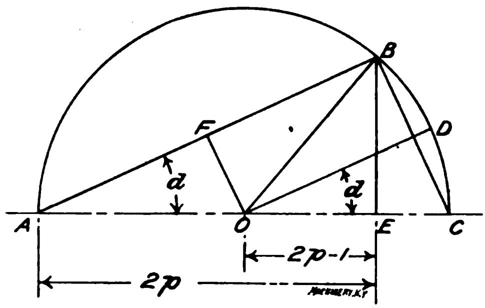


Fig. 3. Diagram for Finding Angular Advance of Eccentric

where $\sin d =$ lap, or lap and lead,
$p =$ cut-off in hundredths of the stroke, and
$d =$ angular advance of eccentrics.

If r equals radius of the throw of the eccentrics, then multiplying the second term of the equation by r the formula gives the value of $\sin d$ as follows:

$$\sin d = r\sqrt{1-p} \qquad (2)$$

An analysis of this formula is readily made by reference to diagram Fig. 3. Draw the semicircle ABC to represent the path of the center of the eccentric, with a center at O and a radius equal to 1. Lay off the distance $AE = 2p$ and draw EB at right angles to the diameter AOC. Connect the points A and B and B and C, and from O draw OF at right angles to and bisecting the line AB, and also draw OD parallel to AB. With this construction the arc AB will be equal to that swept through by the crank and the eccentric from the beginning of the stroke to the point of cut-off, and the arc DC which is one-half of BC will be equal to the angular advance of the eccentric. Then the angle DOC equals d.

From the diagram

$$AC : BC = BO : EC.$$

By multiplying and substituting the value $AC = 2$ and $EC = 2 - 2p$ we have

$$BC = \sqrt{2\,(2 - 2p)} = 2\sqrt{1 - p}$$

and

$$\frac{BC}{2} = \sin d = \sqrt{1 - p}$$

For example, suppose a maximum cut-off of 0.85 of the stroke is desired with no lead in full gear and a 6-inch throw of the eccentric and travel of valve. The advance of the eccentric and the lap of the valve will be obtained by substituting these values in the formula:

$$\sin d = 3\sqrt{1 - p} = 1.16 \text{ inch,}$$

or nearly 1 3/16 inch lap without lead. The radius being 3 inches the value obtained based on the radius as a unit must therefore be mul-

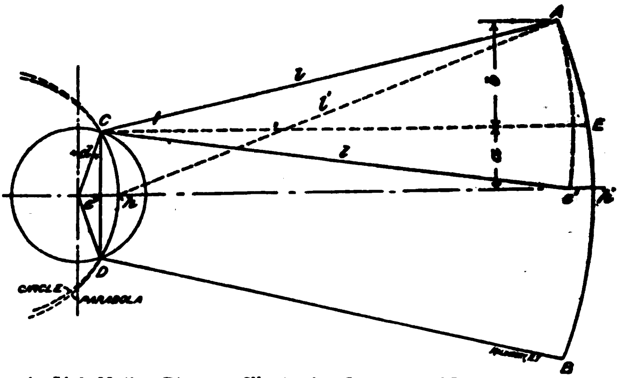


Fig. 4. Link Motion Diagram Illustrating Increase of Lead with Open Rods

tiplied by this amount, as above, in order to get the result expressed in inches as provided for in formula (2) and in the example given.

Having thus determined the angular advance and lap of the eccentric it remains to obtain the positions of the virtual eccentric as the engine is hooked up towards mid-gear. It is now in order to examine the curve that the center of the virtual or resultant eccentric traverses in passing from the point of full gear forward to full gear back.

Let the diagram, Fig. 4, represent the eccentrics and the link, in which the circle CD is the path followed by the actual centers about the axle. C will then be the point occupied by the forward eccentric when the crank is on the center, and D that of the backing eccentric. ’e first step in the problem is to determine the height eh or distance he resultant eccentric center from the chord CD, when the link n mid-gear. This height eh is equal to the displacement of the ve caused by moving the link from full to mid-gear. In Fig. 4

a = the angular advance of the eccentric,

$$a = \frac{CD}{2},$$

b = one-half the distance between the pivot points of the link, minus a,

$2\,(a + b)$ = distance between pivot points of the link,

l = length of eccentric rods,

l' = radius of the link.

When the link is in full gear forward the eccentric-rod occupies the position Ce', and when the link is in mid-gear the rod occupies the position CA. It will be seen that, in moving from e' to A, the pivot point of the link moves away from the vertical center line, passing through the center of the axle until it reaches the line CE and after that approaches the same until it reaches A. The amount of separation may be expressed as equal to

$$l - \sqrt{l^2 - a^2}$$

and the approach as

$$l - \sqrt{l^2 - b^2}$$

which would make the total approach

$$[l - \sqrt{l^2 - b^2}] - [l - \sqrt{l^2 - a^2}] = \sqrt{l^2 - a^2} - \sqrt{l^2 - b^2}$$

At the same time the center line of the link that is occupied by the link block has moved to the point h', which is that of the center when the pivot points are at A and B. This distance, $e'\,h'$, through which the block is moved, is equal to the versed sine of the angle Ahh', less the distance by which the pivot point A actually approached the vertical passing through the center of the axle. If the radius of the link be taken as equal to the length of the eccentric-rods, the center of the link in mid-gear will be at the point h.

The versed sine of the angle Ahh' will then be

$$l' - \sqrt{l'^2 - (a + b)^2}$$

Hence

$$e'\,h' = l' - \sqrt{l'^2 - (a + b)^2} - (\sqrt{l^2 - a^2} - \sqrt{l^2 - b^2})$$

Then

$$hE = hh' = Ce' = l = l'$$
$$\text{and } eh = e'h'$$

In this case the radius of the link l' is assumed to be equal to the working length of the eccentric-rod l. An inequality in these dimensions makes no difference in the accuracy of the formula.

According to Prof. Zeuner in his analysis of the Stephenson valve motion, the curve passing through the points ChD is a parabola, but between the limits C and D in which it is used, it coincides so closely with a circle that it may be regarded as one whose radius is expressed by the formula:

$$r = \frac{B^2 + a^2}{2\,B} \qquad (3)$$

in which

$$B = eh.$$

Before taking up the working out of the details of the valve motion mechanism, a few diagrams will be presented illustrative of the various conditions that will be encountered in practice.

Fig. 5 is a diagram showing the action in full gear conditions. To construct it, draw the circle *CDGH* with the diameter equal to the travel of the valve. From the same center and with *N* equal to the lap of the valve, draw the lap circle. At the extremity of the horizontal diameter with the lead in full gear as a radius, draw the small lead circle *C*. Draw the line *EF* tangent to the lead and lap-circles; then parallel to it the dotted line *CD* which defines the point *D*; and also the line *GH*, defining these two points as well. By drawing *FP* at right angles to *AB*, the point of cut-off *P* is located and can be measured from *C*, in the percentage of the diameter of the valve travel circle to which it will bear the same relation as the piston position, at that instant, bears to the stroke of the engine.

If the point of maximum cut-off has been decided upon, the process may be reversed by laying off the desired cut-off point *P* on the line *AB* and drawing line *PF* to intersect the travel circle. Draw the lead circle as before, about *C*, and then lay down the line *EF* tangent to the same. The lap is the perpendicular distance from the line *EF* to the center *O* and its circle may be drawn tangent to *EF*. The line *GH* is drawn as before parallel to *EF*.

These few lines practically define all of the points that will be needed for a study of the valve action at full gear when the exhaust lap is zero or line and line. The line *OE* indicates the crank position when the valve opens and *OF* when it closes at the cut-off. The exhaust opens at *G* and compression begins at *H*, and by projecting these points to the diagram below, it is possible to obtain the prominent points of an indicator diagram for full gear action.

When the center of the axle lies in the axis of the cylinder and valve motion the crank will coincide with the line *KL* at the instant that the centers of the two eccentrics are at *C* and *D* respectively. The curved line connecting *C* and *D* indicates the locus of the virtual or resultant eccentric, that operates the valve under the combined influence of the two eccentrics, and its distance from *EF* at any point represents the lead that the engine is working with, when hooked up so that the center of the virtual eccentric coincides with the given point.

Fig. 6 is a diagram showing the effect of raising the link and thus shortening the period of steam admission. In a general way it closely resembles that given in Fig. 5. A change has, however, been made by the introduction of a small exhaust lap, in order that the effect of delaying the exhaust opening at *G* and advancing the compression at *H* may be seen. By raising the link so that the virtual center of the eccentric is at *T*, the valve travel becomes equal to the diameter of a circle drawn through *T* with *O* as a center. The lead at this point, being greater than that used for the describing of the circles at *C* and *D*, is taken as the radius for drawing a circle at *C'* where the circle passing through *T* intersects the horizontal diameter. The line *E'F'* is then drawn tangent to this new lead circle as well as to the lap circle, intersecting the new valve travel circle at *F'*. The radius *OF'* is then drawn through this point, *F'*, which is, in turn,

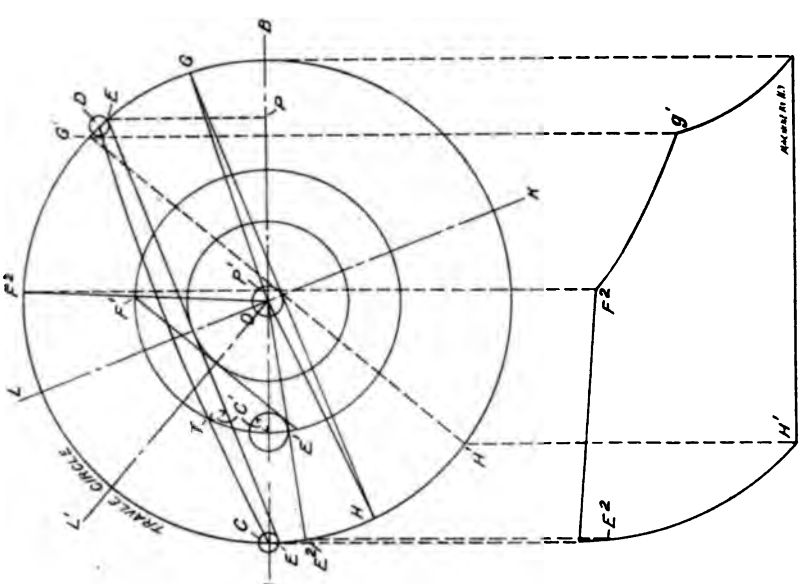


Fig. 6. Diagram showing Effect of Raising the Link, thus shortening the Period of Steam Admission

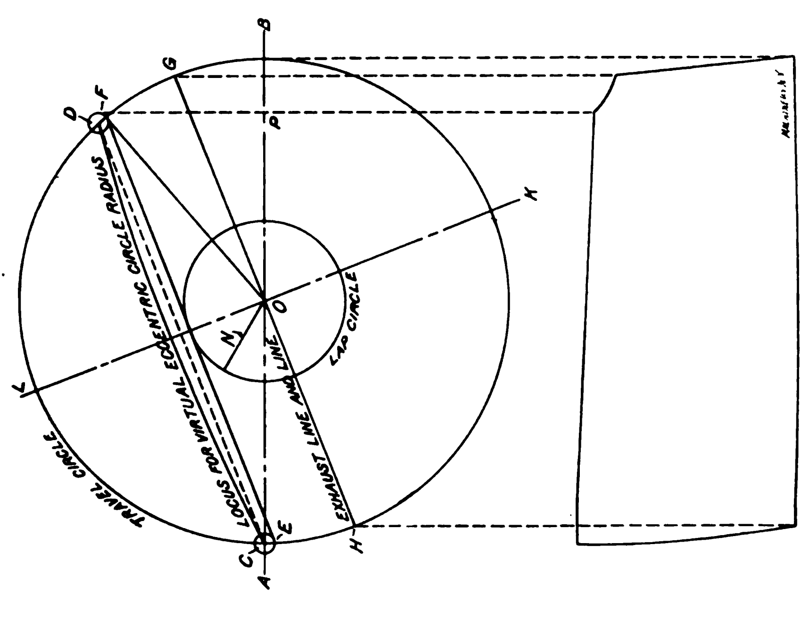


Fig 5. Diagram showing Conditions when in Full Gear

projected upon the line *AB* which it intersects at *P'*. This latter point indicates the point of cut-off or valve closure which, in this instance, is at about half-stroke.

With the change in the angle between the center line *AB* and the cut-off line *E' F'*, there is a corresponding change in the exhaust and compression line, the two remaining parallel, so that the point of exhaust opening *G* is advanced to *G'* and the commencement of compression *H* is advanced to *H'*, while pre-admission is moved ahead to E^2. It thus appears that, with the advancement of the point of cut-off, the same thing is done with the opening and closing of the exhaust and the lead opening. If these points are then projected upon the line *AB*, their location with reference to the stroke of the piston

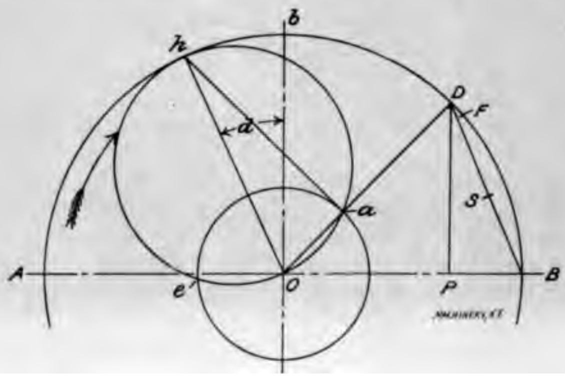


Fig. 7. Zeuner's Diagram

is at once fixed; while by extending them below, the outline of an indicator card is obtained.

Fig. 7 is the well-known Zeuner diagram. It is a method of graphically determining the valve events and was devised by Prof. Zeuner, of the Polytechnic Institute of Zürich, Switzerland, and is, therefore, known by his name. It offers a simple and reliable means of ascertaining not only the main valve events, but the amount of port opening both for steam admission and exhaust, for any piston or crank position throughout the entire stroke, and for any desired link position of the gear.

The angle of advance is obtained by Formula (2) when the lap in full gear is equal to sin *d*, or, in other words, when the valve is set with no lead, or by this diagram, where the lead can be introduced without much complication. It will be noted, however, that the angle of advance *d* must be laid off towards the crank, or 90 degrees — *d* in advance of the same, in order to obtain a correct reading.

In the construction of the diagram, let *AOB* be the diameter of a circle upon which the semi-circle *ADB* is described. The desired point of cut-off is laid down at *P*, the period of admission extending from *A* to *P*. From *P* a line *PD* is drawn at right angles to *AOB* to *D*, which is then connected with *B* by the line *DB*. On this line *DB* lay off DF from D equal to the amount of lead that is desired; and divide BF into two equal parts at S. Then $BS = FS$ will be the required lap.

With BS as a radius draw a circle about O, as a lap circle, and from the point a where it intersects the line OD draw ah at right angles to OD. Draw the radius Oh and with this as a diameter draw the valve circle $Oeha$, and the angle of advance will be found to be equal to hOb. The application of this diagram will be set forth in Figs. 8 and 9.

In Fig. 8, which is a combination diagram, the lap and lead circles as well as the positions C and D of the eccentrics are the same as in

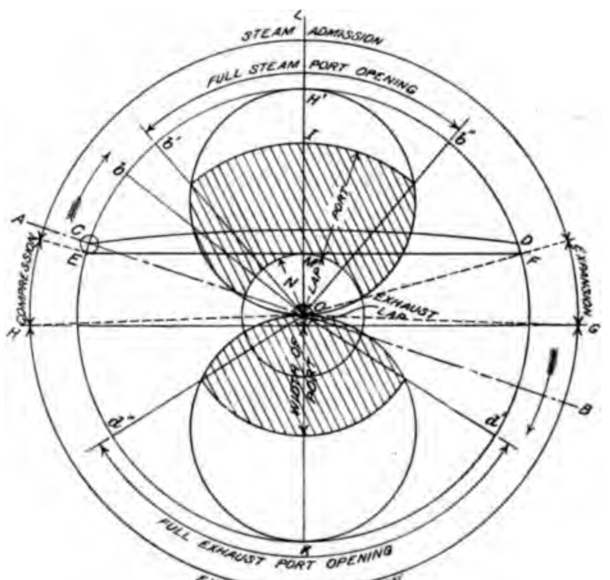


Fig. 8. Combination Zeuner's Diagram

Fig. 5, when the crank is at K for a direct valve motion, at L for an indirect. A modification has, however, been introduced in the shape of a small exhaust lap for the purpose of illustrating its appearance on the diagram. On this diagram, as thus constructed, the Zeuner diagram is imposed, by means of which the port opening at any point of the stroke, or rather at any crank angle, can be determined under both the admission and exhaust periods.

By locating this diagram relatively to the crank, the center line of the diagram will make an angle equal to that of the angle of advance, but on the opposite side of the 90 degrees to that of the actual location of the eccentric, or at a point where the eccentrics would be when the crank had reached B, and was turning in the opposite direction. With this location of the diagram it will be found that its centers will fall on the line OL of the original diagram, Fig. 5, which is one at right angles to EF. As the diameter of the valve circle is equal to the radius of the eccentric circle, it will be seen that the former intersects the lap circle at the same points as OE and OF by which the opening and closing of the valve is indicated respectively.

On the line OL lay off the distance IM outside the lap circle equal to the width of the port and draw the port circle through I with O as a center. By cross hatching that portion of the area of the valve circle lying between the lap and port circles, a graphical representation of the port opening is obtained. It is now possible to find the actual port opening for any position of the crank throughout the admission period, wherever a radius is drawn. For example, when the crank is on the line Ob, the opening will be about three-quarters the full width of the port, whereas at any point between b' and b'' there is a full width of port opening. At the latter point the valve commences to close, an act that is finally accomplished when the crank reaches OF.

The exhaust valve circle is located on the opposite side of the center and is of the same diameter as the steam circle. With the small exhaust lap as a radius, a circle is drawn about O as a center, from which the width of the steam port is laid out along OK and the port circle drawn just as on the steam side. The lines OG and OH indicate the exhaust opening and commencement of compression respectively, while the lines Od' and Od'' indicate the limits of the crank positions where there is a full port opening. A study of this combination diagram will show that all of the valve events coincide exactly in the two that have been thus superimposed, throughout the entire revolution of the crank.

Fig. 9 is a Zeuner diagram laid out to show its adaptability to the determination of the several valve events for the different points of cut-off. From the diagrams already discussed the full gear location of the eccentric was found, which is here indicated as r^5. It also appeared in the discussion of the diagram given in Fig. 6, that the line OL representing the center line of motion moves more and more towards the center line of the crank as the link is raised and a shortened cut-off obtained, until, in mid-gear, the two coincide; and, further, that the intersection of these lines and the circles forms the radius of the virtual eccentric and indicates the positions of the latter when the crank is at A, or at the beginning of the stroke.

At the various points of cut-off indicated by r', r^2, r^3, etc., the center of the virtual or resultant eccentric at R', R^2, R^3, etc., falls on the line CD, which represents the locus of these radii, and the diameters of the circles passing through these points are equal to the travel of the valve at the corresponding link positions. For the sake of simplicity no exhaust lap or clearance is shown in the diagram, Fig. 9, but the valve edge is considered to be line and line. Under these conditions the exhaust and compression points fall at right angles to the radial line of these several positions. Thus, the exhaust opening of the position corresponding to R^3 is at e^3, while its closing and the commencement of compression occurs at f^3. The same, of course, holds true of the other positions. If R^3 is located, then r^4, or the point in the revolution of the crank where the valve closes is found by means shown at the lower part of the figure, thus facilitating the study of the action of the valve by the clearer detail so obtained.

Thus far attention has been directed solely to the forward motion. In the consideration of the back gear, it will, of course, be found that exactly the same conditions will obtain, but they will appear in a reversed position on the diagram. That is to say, the valve circles should be drawn below the line *AB* while the exhaust can be obtained by projecting each eccentric position line to the opposite side of *O*.

With the principles as set forth in these diagrams well in mind, it becomes possible to make an analysis of the consecutive events of the motion of the valve at various rates of admission and to enter intelligently upon the task of working out the details of the mechanism by which the valve is to be operated, and the theoretical studies can thus be put into a tangible and practical form.

CHAPTER II

CALCULATION OF VALVE DETAILS*

With the study of the movement of the valve and the eccentric thoroughly in mind, the next step is to so proportion the various parts of the gear that an efficient distribution of the steam will be obtained. It must be borne in mind, however, in this work, that many of the ratios and proportions that will be given are outlines only, and that variations from the figures are allowable and even required in order to meet the exigencies of design and construction.

As an example of this, in starting at the cylinder, the ports in the valve face should be made with an area of from 1/10 to 1/12 that of the cross-sectional area of the cylinder. This is, in itself, quite a range, but is necessary on account of the great differences in the amount of steam that has to be admitted to the cylinder by the different classes of engines, such as fast express, road freight, and switching. With 1/10 or 1/12 the cylinder area as the port area, the next step is the proportioning of the length of the latter to the width. In this the width should be, or rather may well be, made about 1/12 the length, subject to necessary variations. And finally the average travel of the valve may be put at about three and one-half times the port opening. These are the rough starting figures, and the lap of the valve may be determined either from Formula (2) or fixed arbitrarily at from one-fifth to one-sixth the travel of the valve in full gear. The use of the formula is to be preferred as the latter method leaves much uncertainty as to what the point of maximum cut-off will be and may involve irregularities in the equalization of the same on the two strokes.

In the days when 16 inches represented the standard or maximum diameter of cylinder, the valves were small and light, and the pressure of steam upon their backs did not cause enough frictional resistance to necessitate balancing. But with the increase in cylinder diameters and steam pressures, the larger valves that are used call for a balancing so as to relieve the rods of the tremendous stress that would otherwise be imposed upon them in the work that they are called upon to do. This balancing may be accomplished by any one of the accepted methods.

It should remove the steam pressure from the back of the valve over an area amounting to the sum of the areas of one steam port, the exhaust port and the two bridges, plus 8 per cent of this sum for plain valves and plus 5 per cent for Allan-ported valves. The reason for the use of a smaller amount with the Allan-ported valve is that steam cuts under and through the port and balances its own area when one of the openings is covered by the seat.

* MACHINERY, Railway Edition, May, 1905.

... balancing is introduced in order to lessen the resistance of valve u... rdinary working conditions; but, in calculating the ...ensions ... ms and rods, it is necessary to consider the work that would hav... to be performed in case the balancing strips were broken and t... ull pressure were to be put upon the back of the valve. The fr... nal resistance of the valve on its face may be taken as 20 per cent ... he total load or 20 per cent of the valve area multiplied by the ... pressure. Of course such a high coefficient of friction would ...y obtain if the valve face were very dry, but this is exactly what has to be provided for in case a breakdown under these adverse conditions is to be avoided. For this reason the valve stem and other parts of the motion must be made heavy enough to sustain the stress thus imposed in case of ... ident to the balancing strips and lubricating apparatus. This m... xpressed by the formula:

$$R = 0.2 \ldots \qquad (4)$$

in which

R = the frictional resistance of the val... e,
P = boiler pressure in pounds per sq... inch,
l = length of valve in inches,
w = width of valve in inches.

The area through the keyway of the ... alve stem should, therefore, not be less than $\frac{0.2\,Plw}{10{,}000}$, which would p... t a stress of 10,000 pounds per square inch of section on the metal when in tension. In the matter of diameters the valve stem and eccentric rods must be made strong enough to carry this load in compression and for that purpose should be calculated accordingly. It must be borne in mind that this stress has to be carried back through each of the working and sustaining parts of the eccentrics, increasing somewhat as it advances on account of the added resistance of the motion of the several pieces.

Leaving the valve stem, the stress is next sustained by the rocker and its shaft, two sections of the same part that must be considered independently. The size of the arms may be calculated from the formula:

$$P = \frac{Sbh^2}{6l} \qquad (5)$$

in which

P = the maximum load or resistance to be overcome,
S = the maximum fiber stress permissible in the metal used,
b = thickness of the rocker arm,
h = width of the rocker arm at any desired point,
l = length of the rocker arm from the valve stem or link block connection to any desired point (see Fig. 10).

It will be noted that Formula (5) is that used for calculating the stresses imposed on a beam that is fixed at one end and loaded at the other, the expression $\frac{bh^2}{6}$ being that of the moment of resistance.

This involves the final assumption of one of the two dimensions b or h.

If, for example, b is taken to be 1½ inches and the resistance is put at 10,000 pounds, and the size of the arm is desired at 8 inches from the outer center, with a fiber stress of 10,000 pounds per square inch, Formula (5) becomes

$$h = \sqrt{\frac{6 \times 10{,}000 \times 8}{10{,}000 \times 1.5}} = 5\tfrac{5}{8} \text{ inches.}$$

As the rocker shaft is usually supported by the rocker box for its entire length, it is subjected to torsional stresses only, and these are covered by the general formula:

$$P = \frac{S \pi D^3}{16 R} \tag{6}$$

in which

P = maximum load on, or resistance to the motion of, the rocker arm,

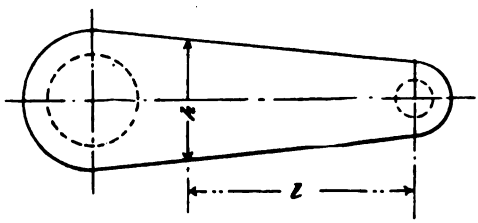


Fig. 10. Rocker Arm

S = allowable fiber stress in the metal,
D = diameter of rocker shaft,
R = length of rocker arm,

the torsional moment of resistance being expressed by $\frac{\pi D^3}{16}$ for a round solid section.

As it is the diameter of the shaft that is to be found, the formula should be transformed into

$$D = \sqrt[3]{\frac{16 P R}{\pi S}}$$

and then substituting $P = 10{,}000$; $S = 10{,}000$; and $R = 10$ inches, we have

$$D = \sqrt[3]{\frac{1{,}600{,}000}{31{,}416}} = \sqrt[3]{51} = 3\tfrac{3}{4} \text{ inches, approximately.}$$

The link should be strong enough to sustain the thrust of the link block when unsupported by the saddle. The formula used is a modification of the general formula for a beam fixed at the ends and loaded in the middle and is

$$P = \frac{4Sbh^2}{6L} \tag{7}$$

in which

P = the stress imposed by the valve,

b = width of link across the face,

h = thickness of metal in link,

L = length of slot.

Let $S = 10{,}000$; $L = 15$ inches; and fix the width at $3\frac{1}{2}$ inches; then the formula for the thickness h becomes

$$h = \sqrt{\frac{6PL}{4Sb}} = \sqrt{\frac{900{,}000}{140{,}000}} = 2\tfrac{1}{2} \text{ inches.}$$

The eccentric rods have already been referred to and they can be calculated as indicated with the understanding that, owing to their position and the liability to cramping and the imposition of excessive stresses due to looseness of the parts, it is well to give them a strength capable of resisting a stress 25 per cent in excess of the calculated resistance of the valve.

Finally as to the eccentrics, the only point to be covered is the width of the face. The diameter is usually fixed by the diameter of the axle, the throw, and the constructional requirements of the type of eccentric that is to be used. The width of the face, therefore, is determined solely by the amount of pressure that it is decided to put upon it per square inch of area. As in the case of all other bearings, an ample surface is a good investment and will repay in immunity from hot and seizing straps in the future operation of the machine. It is well, therefore, to limit the pressure to 250 pounds per square inch measured by a multiplication of the diameter by the width of the face.

This, then, closes the outline of the work to be done in the calculation of the dimensions of the several parts of the valve motion and it remains to examine the methods of application and the modifications that will have to be made in order to adapt the formulas to the two locomotives under consideration.

CHAPTER III

DESIGNING THE VALVE MOTION*

The principles upon which the several parts of the valve motion are designed having now been set forth, the next step will be the study of the application of these principles to the two engines that we have under consideration. In this work the start is made at the engine cylinders where the point affecting the whole of the valve motion is to be found in the width of the steam ports. It has been stated that the area of these should be from 1/10 to 1/12 that of the cross section of the cylinder. By referring to the cylinder drawings it may be assumed that we find that for the consolidation engine, with a diameter of 21 inches, the port measures 18 inches by 1½ inch, giving it a ratio of 1 to 12.8 to the cylinder section. As the piston speeds on this

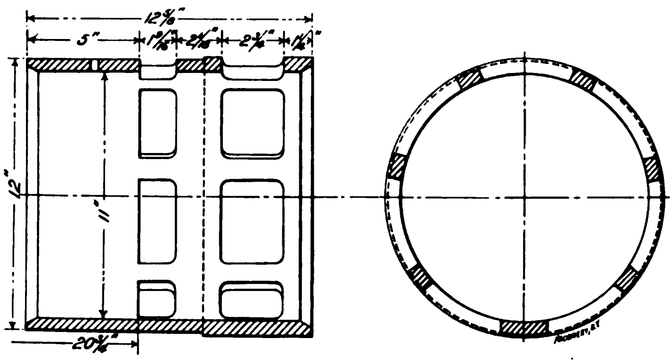


Fig. 11. Valve Chest Bushing for Atlantic Type Locomotive

engine are to be comparatively low, and as it is always desirable for constructional reasons to keep even dimensions at all times, this variation is allowable.

In the case of the Atlantic type engine, we assume that the drawing shows that the port opening in the cylinder casting is 2¼ inches wide. This is evidently too wide and is designed for the use of a bushing to be pressed into the interior. The use of a bushing for piston valves has the three-fold advantage of making the castings of the cylinder ports easier, of facilitating the cutting of the admission ports to the exact size, and of making it possible to renew the steam chest without reboring it. Accordingly bushings like that shown in Fig. 11 are pressed into the space for the valve. In this the width allowed for the port is 1 9/16 inch, and it extends entirely around the bushing except for six bridges 1 inch wide and one 2 inches wide at the bottom. There is also a space surrounding the whole port opening in the cylinder

* MACHINERY, Railway Edition, July and August, 1905.

casting by which steam can flow around the bushing into the cylinder. The diameter of this space is about 16 inches and it is 2¼ inches wide. The outside diameter of the bushing is 12 inches. The available opening for the flow of steam with this bushing in position is, at a maximum (1 9/16 × 12) + (16 — 12) × 2¼ = 27¾ square inches.

In the case of this engine a bushing is used in the cylinder as well as the valve case, and the inside diameter of this bushing is 19½ inches. This makes the ratio of port opening to cylinder area very nearly as 1 to 10.75, from which it will appear that due allowance has been made for the difference in the maximum piston speed of the two engines. The ratio of length of port to width is approximately 12 to 1.

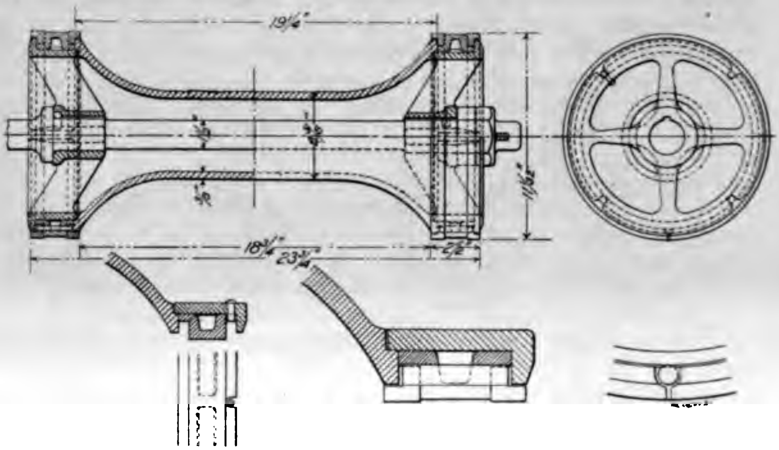


Fig. 12. Piston Valve for Express Passenger Locomotive, Atlantic Type

In the case of both of these engines the width of the ports is about 1½ inch, so that the travel of the valve should be 5¼ inches. Taking this figure for the Atlantic type engine, the valve diagram, Fig. 13, can be constructed according to the directions already given.

Before starting on this it may be stated that, in the case of high-speed engines, it is frequently more troublesome to get the exhaust steam out of the cylinder than it is to get the live steam into it. It is, therefore, common and approved practice to give the valve a negative exhaust lap. That is to say, the valve, when in its central position, leaves both ports uncovered to a small extent on the exhaust side. This makes the exhaust opening a little earlier than would otherwise occur and so hastens the outflow of steam and cuts down the back pressure. For this reason, the valve on this Atlantic type engine is given 1/16 inch negative exhaust lap.

As certain points in the designing of the valve motion must be decided arbitrarily, we will assume a negative exhaust lap of 1/16 inch, a lead in full gear of 1/16 inch and a maximum point of cut-off of 0.83 of the stroke. In Fig. 13 draw the circle *ABC* with a diameter of 5¼ inches, the assumed travel of the valve, and at A describe the lead circle 1/16 inch in radius. Draw the diameter AC and lay off D, making AD 0.83 of AC. From D erect the perpendicular DE and from E draw a line tangent to the lead circle. From the center O draw a circle tangent to the line AE, and its radius will be equal to the lap of the valve which will be found to be 1 inch in the present instance. By drawing OF at right angles to AE and the arc of the port opening IH, the angle of full port opening is obtained. In like manner the other elements of the motion of the valve may be studied. We have determined from this that the steam lap of the valve should be 1 inch.

In designing the valve, which is intended for inside admission, we take the distance between the ports on the bushings (Fig. 11) which is 20¾ inches, allowing 1 inch for lap on each side, making 18¾

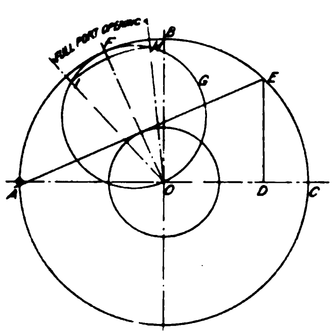


Fig. 13. Valve Diagram for Atlantic Type Locomotive

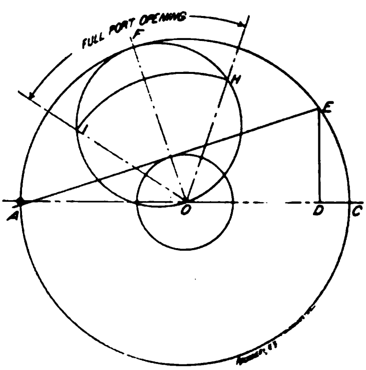


Fig. 14. Valve Diagram for Consolidation Locomotive

inches as the distance between the lips of the valve. The distance between the outside edges of the ports is 23⅞ inches, and allowing 1/16 inch for negative lap at each end makes the over-all length of the valve 23¾ inches.

With these dimensions it is possible to construct the valve shown in Fig. 12. In this the body is of cast iron with spring packing rings, the former being hollow so as to secure a perfect balance and permit the exhaust steam to escape through the center and flow out of the passages at each end of the steam chest. The packing can be of any desired type, that here shown being sprung in and turned 1/32 inch larger than the bore of the steam chest.

In the case of the flat valve of the consolidation freight locomotive, similar assumptions must be made, but they must be based upon different conditions of service. In the first place the engine is to be worked more slowly, and it must be able to exert its maximum tractive effort. For these reasons it is desirable that the maximum point of cut-off should be later and the period of full port opening longer. Hence it will be found to be advisable to increase the travel of the valve which may well be brought up to 6 inches, and by putting the maximum point of cut-off at 9/10 the stroke, the lap of the valve, with

1/16 inch lead in full gear, becomes ⅞ inch. The diagram, Fig. 14, which corresponds to Fig. 13 for the express engine, shows the features desired. It is given in order to illustrate the difference that will be found in the diagrams of high- and low-speed engines. From this the lap will be found to be ⅞ inch, and this with the dimensions we assume as taken from the cylinder drawing makes the outside dimensions of the flat valve 21 inches. On engines of this character negative exhaust or inside lap is unnecessary, and the valve is made line and line. It may be balanced in any desired manner in accordance with the general proportions already laid down. A valve proportioned to meet these requirements is shown in Fig. 15.

It may be noted here that the slide valves should be made of hard cast iron and of a size suited to meet the conditions of the steam

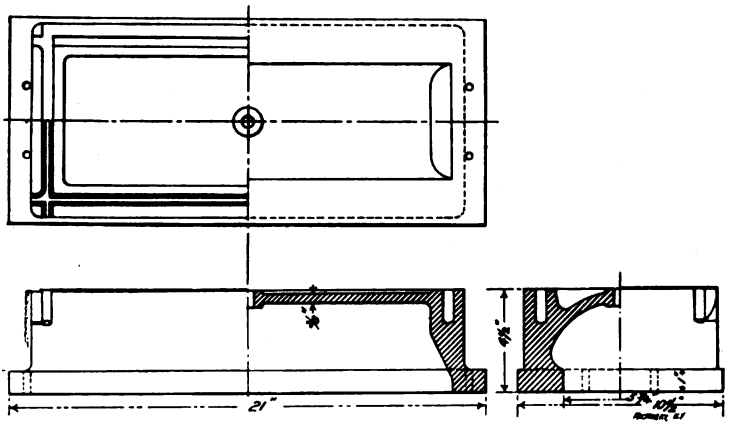


Fig. 15. Balanced D-valve for Consolidation Locomotive

ports. It will be found that on ordinary engine service the outside or steam lap will vary from ⅞ inch to 1¼ inch, dependent upon the service required and the capacity of the cylinder. The lead allowed does not ordinarily exceed 1/16 inch in full gear, and is often made line and line especially when the construction is such that the eccentric rods are less than 4 feet long. The negative exhaust lap varies from line and line to ⅛ inch for high-speed engines.

The balancing of the slide valves is of special importance and the parts should be well and accurately made. Two general methods are in use, of which the oldest is known as the Richardson. It consists of ½-inch by 1½-inch strips set in suitable grooves and resting on springs, thus forming a rectangular enclosure on the top of the valve and bearing against a smooth balance plate above the valve.

A later form is the American balance which consists of a conical ring cut through at one point, and fitted to a taper bearing on top of the valve. A cover piece similar in section to the Dunbar L-section ring is employed to cover the joint. This packing requires no springs since its reaction on the taper bearing due to its elasticity and the steam pressure tend to lift it; hence the upper part of the ring bears against the slides upon the balance plate in the same way as the Richardson valve.

The piston valve is usually made with a hollow cast iron body with followers and packing rings of an L-section at each end, and is worked inside a bushing through which the steam ports are cut as previously described. The L-shaped rings possess an advantage over square rings in that they give a positive opening and closing edge, whereas this is not necessarily the case with the latter, because the projection of the follower which holds them in position interferes somewhat with the free passage of the steam, if not taken into account as a part of the lap; while it is liable to leak, causing wire-drawing and excessive pre-admission if it is so considered. On account of the many bridges

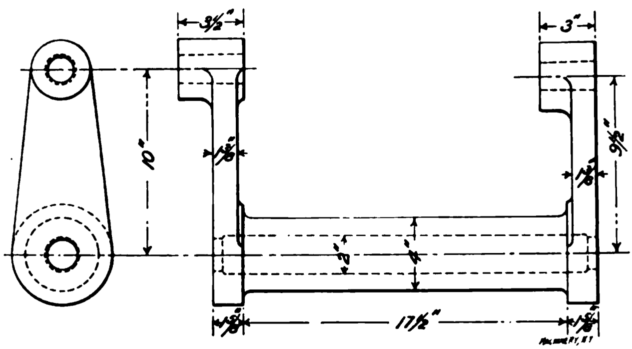


Fig. 16. Direct Motion Rocker Arm for Atlantic Type Locomotive

on the ports of the bushing and the more or less checked flow of steam past the top of the valve towards the cylinder, the diameter of the piston valve should be such that the circumferential ports should be at least 50 per cent longer than those for a flat slide valve. Further, the valve should invariably be made for inside admission, so that the stuffing-boxes are not obliged to sustain boiler pressure, and the valve events can be made the same as with the ordinary slide valve.

With the proportions of the valve thus worked out, the designing of the other ports of the motion becomes a comparatively simple matter. Before this can be done, however, it is usually necessary to locate some of the other parts, and work out the frames and driving mechanism so as to insure proper clearances when the engine is in motion. Such work usually decides the location of the link, rocker-boxes and connections. As would be clearly shown in a side elevation of the engines, the arrangement of the working parts of the two are quite different. In the case of the Atlantic type engine, the valve has a direct motion from the eccentrics obtained by the use of a rocker like that shown in Fig. 16. From this it will be seen that the valve arm is longer than the eccentric arm, so that the throw of the latter is less than the travel of the valve. According to the proportions given, the throw of the eccentric should be 4.98 inches. This can be made 5 inches, which will be done.

In the case of the flat valve of the consolidation engine, an indirect motion rocker like that shown in Fig. 17 is used. Here, too, it has been found to be convenient to make the eccentric arm shorter than the one driving the valve stem. With the proportions chosen, the throw of the eccentric becomes 5.08 inches, and it is made 5 inches as before, thus modifying to a very small extent the several valve events as found from the diagram, but not enough to materially affect the action of the engine.

With the valve and the eccentric proportioned, it becomes possible to work out the details of the whole motion. Starting with the eccen-

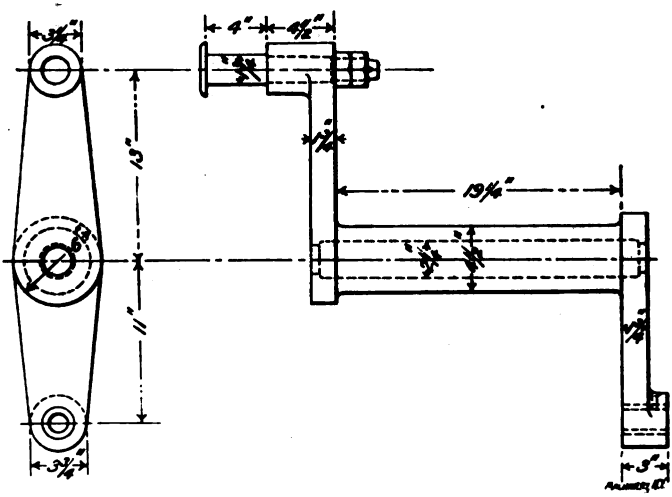


Fig. 17. Indirect Rocker Arm for Consolidation Locomotive

tric, its diameter is dependent, to an extent, upon the diameter of the axle. The form used depends upon the taste of the designer. On many roads solid eccentrics are in use that must be put in place before the wheels are pressed upon the axle. In some cases the two eccentrics are made in one piece, but the general practice is to make the eccentric in halves and of cast iron, securely bolted together and keyed to the axle. For this detail there are a number of designs, one of which is shown in Fig. 18. It is made as small as possible, consistent with ength. If the axle is assumed to be 9 inches in diameter, at least 1 of metal should be allowed on the thin side, which would make 'side diameter 16 inches; the allowance for re-turning makes ches. The width of the eccentric must be sufficient to permit ing without tilting, and bolting the two parts together without ger of splitting or cracking.

We have seen that the eccentric must have ample strength to drive valve under the most adverse conditions when the packing strips are broken and the surface dry; it must also have ample bearing area to prevent heating. Substituting the values of the valve in Formula (4) we have a boiler pressure (P) of 200 pounds; a length (l) of 21 inches, and a width (w) of 10½ inches. Hence the resistance

$$R = 0.2 \times 200 \times 21 \times 10.5 = 8820 \text{ pounds.}$$

As the pressure on the bearing surface should not exceed 250 pounds to the square inch, the area of this surface should not be less than

$$\frac{8820}{250} = 35.3 \text{ square inches.}$$

As the diameter is 16¼ inches, the width should not be less than 2 3/16 inches. As an increase over this is desirable, and as there is

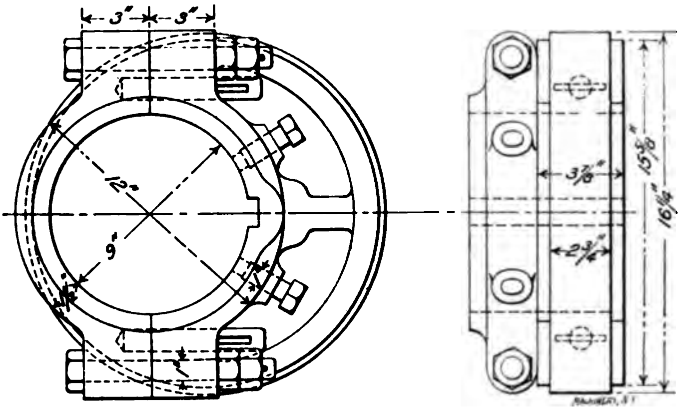


Fig. 18. Eccentric for Consolidation and Atlantic Type Locomotives

plenty of room, it is made 2¾ inches, which cuts the pressure down to about 200 pounds, thus giving good working conditions so far as bearing surface is concerned even under exceptional resistances.

The eccentric is keyed to the axle, as adjustment of this part and fastening by setscrews are things of the past and are entirely obsolete methods of construction. The details of the form shown here are sufficiently distinct in the engraving to require no explanation. Attention is merely called to the fact that every precaution in the way of check nuts and cotters is used to prevent the parts from becoming loose or lost.

Closely allied with the eccentrics are the eccentric straps. They may be of cast iron or bronze and must have a strength sufficient to move the valve under adverse conditions without an appreciable amount of yield. This is necessary in order that they may preserve their full bearing surface in contact with the eccentric at all times and not pinch the latter because of some distortion or yielding. Formula (7), which is the general one for a beam fixed at the ends and loaded in the middle, may be used for the calculation of the body of the strap. It may be modified, however, to

$$P = \frac{4\,Q\,S}{l} \qquad (8)$$

in which

P = the stress imposed by the valve,
S = allowable fiber stress in the metal,
l = distance between fastening bolts in inches,
Q = the section modulus.

The latter must be worked out for all sections other than a rectangle which is $\frac{b\,h^2}{6}$ as already given. In the case of the strap shown in Fig. 19, the width is 3¾ inches and the depth 2⅞ inches. The section is flat on one side and semi-circular on the other, and if the computa-

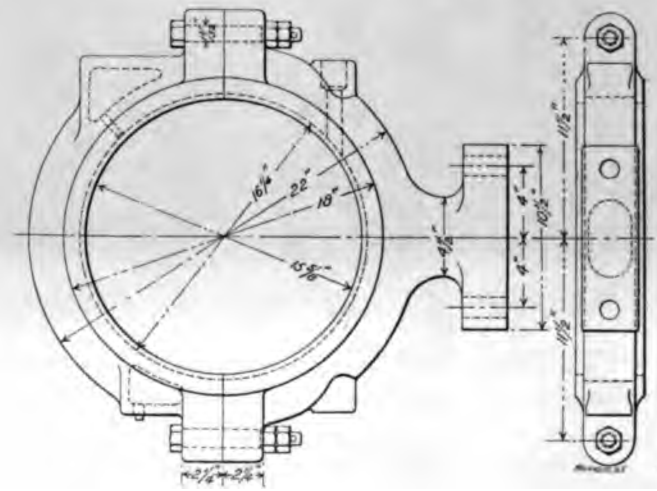


Fig. 19. Eccentric Strap for Consolidation and Atlantic Type Locomotives

tions are made as to its strength it will be found that the maximum fiber stress put on the metal will fall below 2500 pounds per square inch of section. This is low, but it is one of those places where plenty of metal is a good investment as it insures against hot straps and annoying delays upon the road when the engine is in service. The neck of the strap is also made of ample strength and the foot is faced to receive the direct thrust of the rods. The bolts used for the fastenings should be of ample size to hold the parts firmly together, and when this is done their strength will be sufficient to carry the load that is put upon them. The fastenings of the eccentric rods to the straps is made much more secure than it was when engines and valves were lighter. At that time there was a possibility of adjusting the length of the rods, with the result that they frequently slipped in service. Current practice does away with this adjustability with a resultant simplification of the parts.

Fig. 20 shows a substantial form of eccentric rod that is used on the engines under consideration. It will be seen that it is made in "rights" and "lefts" so that the jaws at the forward end line up together to take the link. It is of the utmost importance that these rods should be exceedingly stiff and rigid in order to prevent springing when they are working under compression. If that occurs the action of the valve is not what it should be, and an extra and unnecessary stress is put upon the bolts, pins, links and eccentric straps, all of which tends to a more rapid wear and an increase of the cost of maintenance, to say nothing of the danger of causing delays and breakdowns on the road.

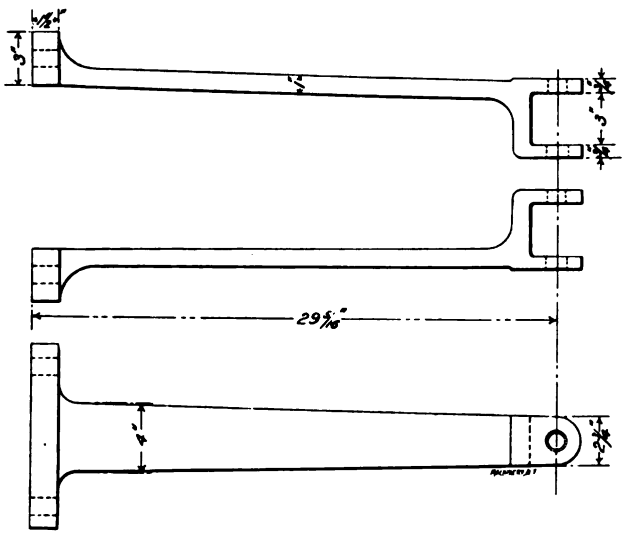


Fig. 20. Eccentric-rod for Consolidation Type Locomotive

The distance from the center of the axle to the rocker arm, or the radius of the link, is 49 inches in the case of the consolidation engine and 60 inches for the Atlantic type. This causes a variation in the length of the eccentric rods of from 29 5/16 inches to 40 5/16 inches. As these rods are flat they must be calculated accordingly. We have seen that the probable maximum load imposed by the valve will be 8820 pounds. In order that there may be an ample margin of strength, it will be well to take this load at 10,000 pounds and proportion the parts accordingly. By using the following formula for this purpose, and assuming the length of the rod to be 30 inches and the thickness 1 inch, we have

$$\frac{P}{A} = \frac{S}{1 + \frac{q\,l^2}{r^2}}$$

in which

P = resistance of the valve = 10,000 pounds, By substitution and transposition, the formula becomes

$$\frac{4P}{\pi d^2}=\frac{10{,}000}{1+\dfrac{(0.00016\times 7225)\times 16}{d^2}}, \text{ and finally}$$

$$d=\sqrt{\frac{2+\sqrt{74\pi+4}}{\pi}}=2.35,$$

so that the valve rod may be made about 2⅜ inches in diameter.

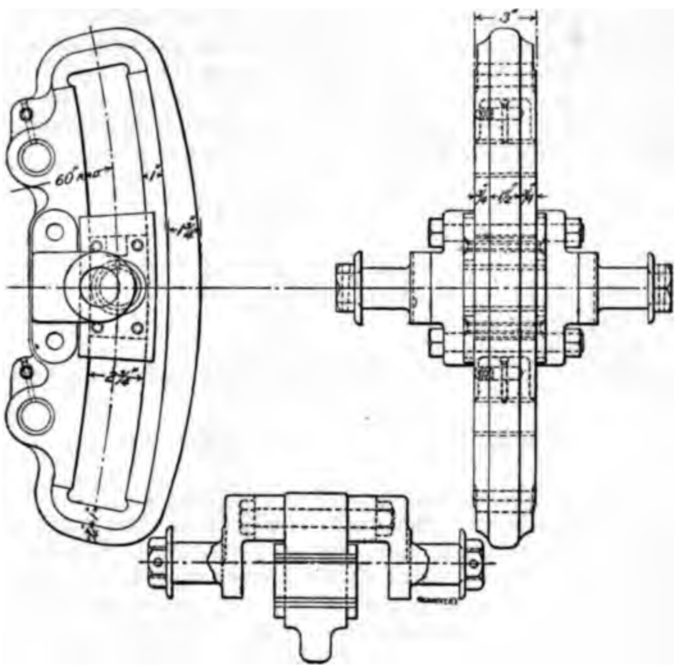


Fig. 22. Cast Steel Link for Atlantic Type Locomotive

The link is also a part that needs most careful attention not only in the designing and proportioning, but in the manufacture. For many years the links on American engines were made of wrought iron, case-hardened and carefully ground to truth. Of late cast steel has been introduced with good results as far as operation is concerned and with a considerable saving in first cost. This metal also possesses the advantage of permitting a better distribution to carry the loads, and can, therefore, be made lighter.

The common form of skeleton link is shown in Fig. 21. It is adapted for use on the consolidation locomotive and owing to limits of space it has been made a trifle shorter than the general proportions given. With an eccentric throw of 5 inches the distance between the eccentric rod pins should be $12\frac{1}{2}$ inches, but in this case it is made 12 inches. The link is built up of four parts, the front *A*, the back *B*, and the two fillers at the ends *CC*. The holes for the eccentric rod pins are protected by case-hardened bushings, and the block is given an ample bearing surface 5 inches long. As in the case of the eccentric rods the link must be of ample strength to do the work without springing. For this we may use Formula (7) though it is common practice to depend upon the supporting of the saddle to prevent springing. Then, again, as the load of the unlubricated, unbalanced valve is excessive, and as it is desirable to make the link as light as possible so that it may be easily handled in reversing, and on account of the support of the link saddle, the stress allowed for may be dropped to 5000 pounds instead of making it 10,000, and the allowable fiber stress may be raised to 12,000 pounds. This leaves the link with ample strength for the ordinary working while in the case of an accident it will not be overstrained on account of the length of the bearing of the link block. The formula as thus modified becomes:

$$P = \frac{4\,S\,b\,h^2}{6\,L}, \text{ in which}$$

P = load imposed by the valve = 5000 pounds,
b = width of the link across the face = 3 inches,
h = thickness of metal in the link,
L = half the length of the slot = 9.75 inches,
S = allowable fiber stress = 12,000 pounds per square inch section.

$$h = \sqrt{\frac{5000 \times 6 \times 9.75}{4 \times 12{,}000 \times 3}} = \sqrt{2.03} = 1.42$$

from which the thickness may be made 1 7/16 inch.

In case of the cast steel link shown in Fig. 22 for the passenger locomotive, a similar course of reasoning can be followed, except that the fiber stress put upon the metal should be kept down to 10,000 pounds, or even less. It will also be well to work out the section modulus as this has an important bearing on the rigidity of the link.

The method of calculating the size of the rocker has been provided for in Formulas (5) and (6).

Taking the rocker for the consolidation engine as shown in Fig. 17, if the valve resistance is placed at 10,000 pounds, the length of the arm at 13 inches and the thickness at $1\frac{3}{4}$ inch, then using Formula (5).

$$P = \frac{S\,b\,h^2}{6\,l}, \text{ in which}$$

P = valve resistance = 10,000 pounds,
S = allowable stress of metal = 12,000 pounds,
b = thickness of arm = 1.75 inch,
h = width of arm,
l = length of arm = 13 inches.

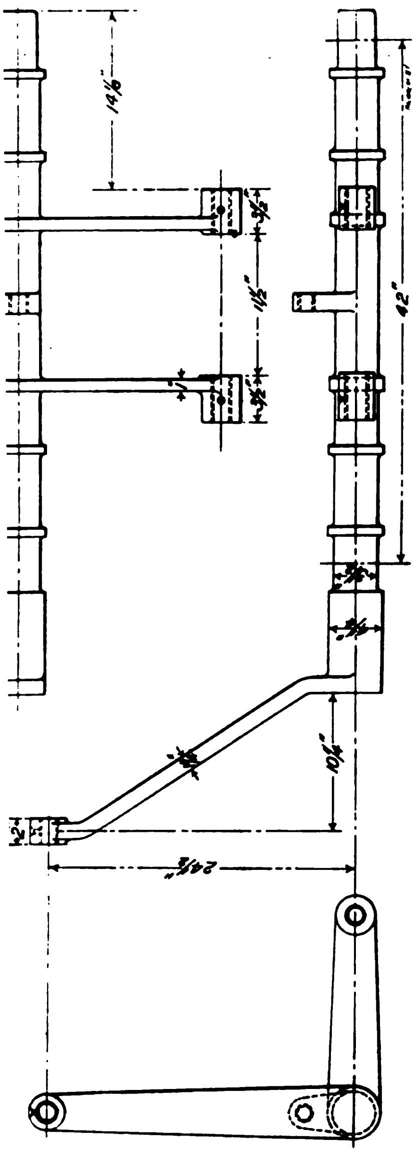


Fig. 23. Lifting Shaft for Consolidation Locomotive

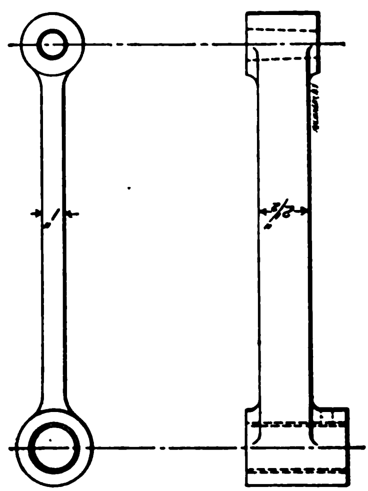


Fig. 24. Link Hanger for Consolidation Locomotive

on and substitution this becomes

$$= \sqrt{\frac{6 \times 13 \times 10,000}{1.75 \times 12,000}} = \sqrt{37} = 6 \text{ inches, approx.}$$

s well to have an ample bearing surface for the rocker,
; is a desirable quality in the moving parts, cast steel
erial and the bearing is made hollow. By assuming a
es and deducting the value of the metal thus removed
th of the shaft, this core would be able to carry a load

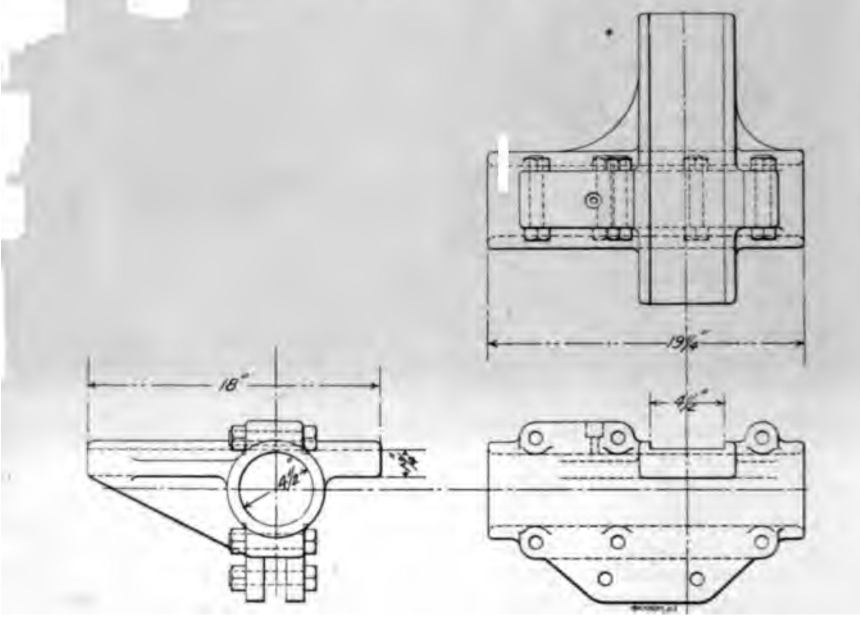


Fig. 25. Rocker-box for Consolidation Locomotive

of about 1750 pounds if calculated by Formula (6). This load is then added to the actual valve pressure, making $P = 11,750$ pounds.

$$D = \sqrt[3]{\frac{16 P R}{\pi S}}, \text{ in which}$$

D = diameter of rocker shaft,
P = valve resistance = 10,000 + 1750 pounds,
R = length of rocker arm = 13 inches,
S = allowable fiber stress = 10,000 pounds.
By substitution the formula then becomes

$$D = \sqrt[3]{\frac{16 \times 11,750 \times 13}{31,416}} = \sqrt[3]{77.8} = 4.27 \text{ inches.}$$

In this case the diameter is made 4.5 inches. These are the outlines of the methods to be followed in the proportioning of the parts of the valve motion.

The lifting-shaft (Fig. 23) and the link-hanger (Fig. 24) are important parts that must be carefully designed and located. In the matter of strength, if they are made stiff enough to do the work, that is all that is required, and their only load is to carry the weight of the link and resist the forces due to the angularity of the link and the binding action of the block in its slip. They are, however, closely associated with the proper location of the saddle-pin on the link, for on the proper combination of dimensions and positions of the saddle-pin, hanger, lifting-shaft and box depends the smooth and correct action of the valve. All parts of the valve motion may be carefully and accurately laid out and if the bearing of the lifting-shaft is improperly located, the action of the valve will be defective.

An outline of the method to pursue is to take a templet of the link (full size preferred) and locate it in the extreme positions of mid-gear. On transverse lines through the centers of the two positions of the templets locate points equally distant from the center which coincide with the arc swept through by the lower end of the hanger when the lifting shaft is in mid-position. These points will indicate the proper position of the saddle pin. The lifting-shaft box should be so adjusted that the lower end of the hanger will sweep through the corresponding position of the saddle-pin when the link is raised and lowered to full backing and forward gears respectively. The designer should make himself familiar with all of the vagaries and peculiarities of the Stephenson valve motion, and this involves a careful study both of what has been published and of the working out of the problems on the drawing board.

It only remains now to call attention to the form of rocker-box that is used, that intended for the consolidation locomotive being shown in Fig. 25. These boxes are usually bolted to the guide yoke, are made of cast iron and afford a support for the rocker bearing throughout its whole length. All pin holes of the working parts of rockers, link and rods are protected by case-hardened bushings.

With this outline of the course to be followed in working out the Stephenson link motion the reader is cautioned against trusting to any haphazard methods of design and recommended to master its intricacies in every particular before attempting to make a practical application of a design to a locomotive.

CHAPTER IV

THE WALSCHAERTS VALVE MOTION*

Until very recently, whenever an American has considered the designing of a locomotive, the work has invariably been associated with the use of the Stephenson link motion for the operation of the valves. This is true with the exception of a very few instances where interested parties have had some special design of gear to exploit. That the Stephenson gear has held its own for so many years speaks well for its efficiency, and indeed it has been found that in the matter of steam consumption, in special cases it holds this figure down to within a very small percentage of the best that can be obtained with the Corliss gear.

On the continent of Europe, on the other hand, the Walschaerts gear or a modification of it is almost exclusively used; and it is claimed to possess many advantages over the Stephenson motion, one of the more prominent of which is the maintenance of a constant lead for all points of cut-off, an advantage that is not universally acknowledged, however. During the last five or six years the Walschaerts gear has also received a more and more extensive application in this country, hence it is necessary to discuss and analyze it in considering the designing of a locomotive.

The reason for the change of attitude regarding the Walschaerts gear is due to a number of difficulties that have been experienced with the Stephenson valve motion on the large and heavy locomotives of modern construction. Among these are the excessive wear of the heavy eccentrics and straps, and the large amount of space between the frames that is occupied by the eccentrics, rods, links, and hangers, making it exceedingly troublesome or quite impossible to properly brace the frames; the Walschaerts gear has, therefore, been very successfully applied to several thousand engines of recent design.

It is more accessible than the Stephenson motion in that it is applied outside the wheels and requires only a single eccentric return crank and a connection to the crosshead for its operation. The eccentric crank may be a comparatively small pin attached to the main crank, and it does the work of the two heavy eccentrics of the ordinary gear. This arrangement leaves the entire space between the frames clear for the bracing of the same.

Further, it produces a more uniform steam distribution with a lower percentage of pre-admission, to which is added a constant and moderate amount of lead for early cut-off, though on the resultant economy in steam consumption there can be but slight difference when both the gears are in first-class condition. Finally, it is not so likely to get out of

* MACHINERY, Railway Edition, September, 1905.

order as when the driving is done by eccentrics; since, when the parts have been properly fitted, they are not liable to get out of place except when damaged by collision or other accident to which they are more exposed.

The Walschaerts gear is, in reality, much simpler to understand than the Stephenson. Of course, since the results of the two are nearly identical, each must have parts whose functions correspond to those of the other. In the case of the Walschaerts gear the valve receives its motion from two sources, the crosshead and an eccentric crank whose center is located 90 degrees from the center line on the main crank, when the center lines of the cylinder and gear motion coincide and pass through the center of the axle.

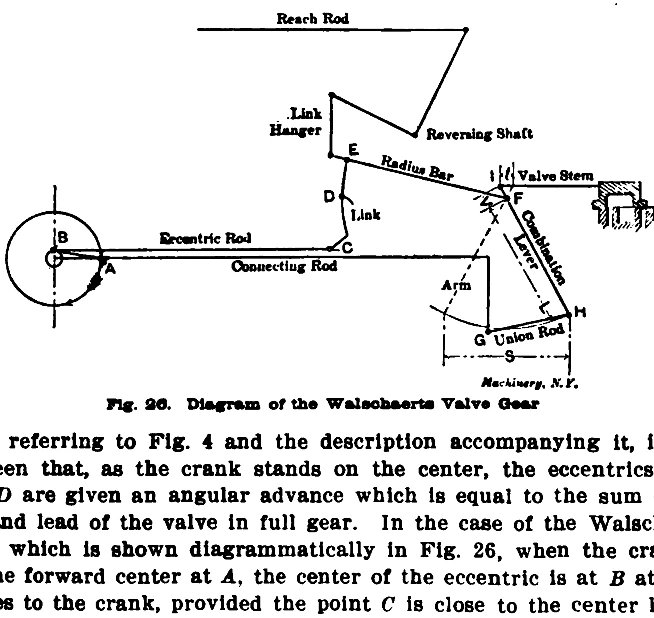


Fig. 26. Diagram of the Walschaerts Valve Gear

By referring to Fig. 4 and the description accompanying it, it will be seen that, as the crank stands on the center, the eccentrics at *C* and *D* are given an angular advance which is equal to the sum of the lap and lead of the valve in full gear. In the case of the Walschaerts gear, which is shown diagrammatically in Fig. 26, when the crank is on the forward center at *A*, the center of the eccentric is at *B* at right angles to the crank, provided the point *C* is close to the center line of the cylinder. From *B* an eccentric rod runs to *C*, the lower end of a fixed line *CDE* which is pivoted at *D*. This link has a groove in which the link block attached to the radius rod slides up and down. The length of this rod from *E* to *F* is equal to the radius of the link itself. If now the valve stem were to be connected directly to the end of the radius bar, it is evident that when the crank is on the center, the valve would be in its central position either for forward or backward motion and for any position of the link block of the radius bar in the link. Under these circumstances, there could be no lap or lead to the valve.

The lap and lead is arranged for by dropping a rigid arm down on the crosshead to *G* and from this a union rod is led out to the lower end of the combination lever at *H*, this lever being pivoted at the end of the radius bar at F and extending up to the valve stem connecting at I. It is evident that the inclination of the combination lever will be the same at the end of the stroke regardless of the position of the radius bar; and that, therefore, the horizontal displacement of the point I and the valve stem will be the same on either side of a vertical line through F. This horizontal displacement is equal to twice the sum of the lap and the lead, hence the latter is constant for all points of cut-off. These same statements hold true for the opposite end of the stroke, when A is on the back center and B at the bottom, barring a slight variation due to the angularity of the rods.

In the case shown, where the eccentric B follows the crank the engine runs ahead when the radius block is at the top of the link and backwards when it is at the bottom. These conditions would be reversed by having the eccentric lead the crank. The crosshead, then, imparts a motion to the valve equal to the lap and lead when the crank is on either center, just as the angular advance of the eccentrics does in the case of the Stephenson gear.

It will be noticed that the eccentric and the crosshead tend to move the valve in opposite directions during the first half of each stroke and in the same direction during the last half; or, in other words, they work in opposite directions during the first and third quarters of a revolution of the crank starting from either dead point, and together during the second and fourth quarters. The motion derived from the crosshead is constant and is not subjected to reversal in the reversing of the motion of the engine, which is done entirely by a change in the motion imparted by the eccentric, which also controls the variation of the points of cut-off.

In order to accomplish this the motion of the eccentric is transmitted through an oscillating link pivoted at its center and so slotted that a link block attached to the back end of the radius bar can be moved through its whole length, and by placing this above or below the center, a reversal of the engine will be obtained. This motion, either direct or indirect, is taken up by the radius bar and carried out to the combination lever, where it is combined with that obtained from the crosshead and the resultant imparted to the valve. The motion is therefore the same as though it were derived from an eccentric the center of which could be moved on the line of a chord across the axle from one extremity to the other of the throw.

It is evident from this that the several connecting points along the combination lever bear a most important relationship to each other, which must be maintained in order that a proper movement of the valve may be obtained.

The distances between these several points may be found by the formula

$$S : t = L : V, \text{ or } V = \frac{L\,t}{S} \qquad (9)$$

in which

S = stroke of piston.

t = twice the sum of the lap and lead,

L = distance between the crosshead connection H and that of the radius bar F, Fig. 26,

V = distance between connection of the radius bar F and that of the valve stem I, Fig. 26.

For an outside admission of steam, as in the case of the ordinary slide valve, the connection F, Fig. 26, falls below that of the valve stem so that the crosshead increment of the motion is the same as though it were derived from an eccentric or crank opposite the main crank; while the increment controlling the direction of engine rotation is derived from what amounts to an eccentric leading the crank for forward motion, just as in the case of direct-acting eccentrics in the case of the Stephenson gear.

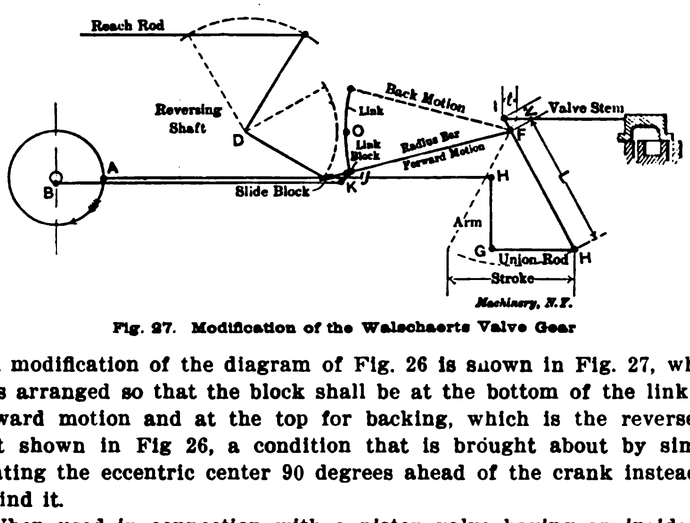


Fig. 27. Modification of the Walschaerts Valve Gear

A modification of the diagram of Fig. 26 is shown in Fig. 27, where it is arranged so that the block shall be at the bottom of the link for forward motion and at the top for backing, which is the reverse of that shown in Fig 26, a condition that is brought about by simply locating the eccentric center 90 degrees ahead of the crank instead of behind it.

When used in connection with a piston valve having an inside admission, the valve stem is usually attached below the connection F of the radius bar on the combination lever, and the eccentric follows the main crank when going ahead with the block at the lower end of the link, which results in the crosshead motion having the same effect as an eccentric on the same side of the axle as the crank.

The reversal of the motion is accompanied by means of a reversing shaft with lever and reach-rod attachments as in common use, and acting upon the radius bar either by a sliding block pivoted directly to the reverse arm, as shown in Fig. 27 or by a hanger from which the bar is suspended. In the first case the bar slides to and fro in the block, and the slip of the link block is equal to the versed sine of one-half of the arc through which the link moves. When the hanger is used it must be of sufficient length to avoid an excessive slip of the block. It is for this reason that the reversing shaft is frequently placed below the radius bar so that the swing of the hanger will tend

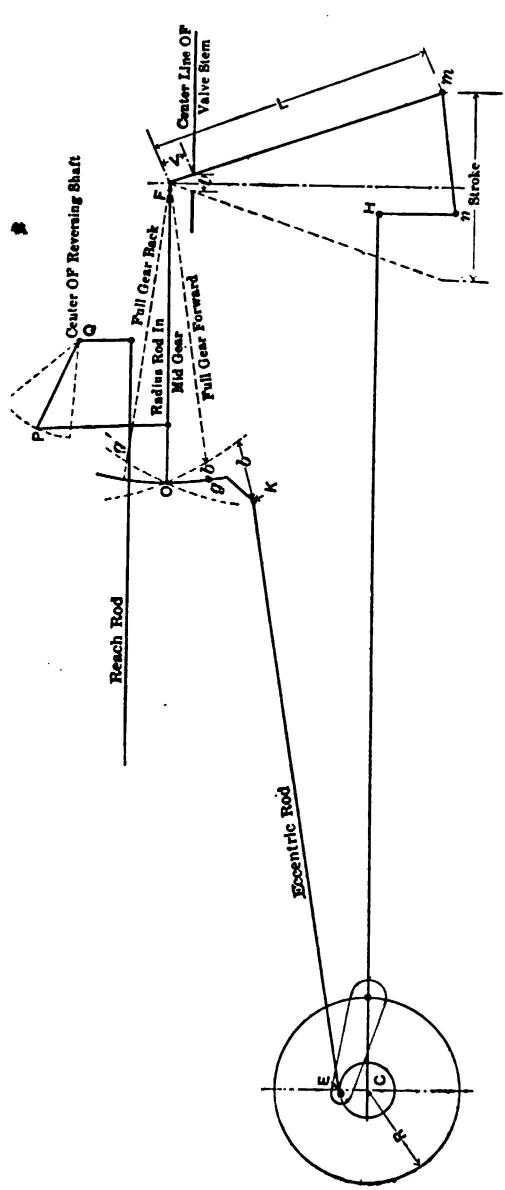


Fig. 28. Method of Reversing the Walschaerts Valve Gear

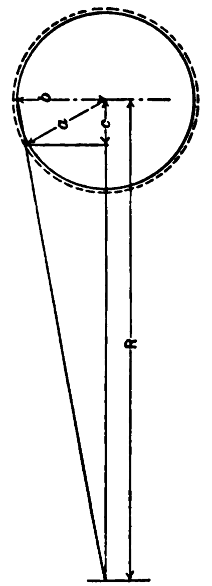


Fig. 30

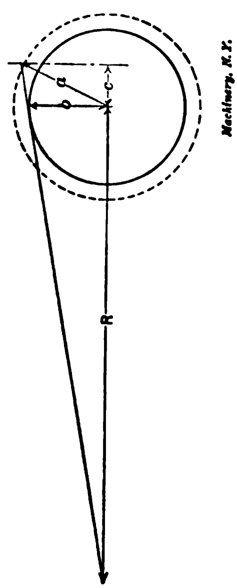


Fig. 29

nside admission valve the sign is changed and we have

$$R : (R - c) = L : (L - V),$$

the valve stem falls below the point F, which must, a longer travel than with outside admission in order to me travel of the valve.

of the virtual eccentric at right angles to the center s equal to $\sqrt{a^2 - c^2}$, from which we may obtain the half point $F = b$ by the following proportion:

$$\sqrt{a^2 - c^2} : (R + c) = b : R$$

$$b = \frac{R\sqrt{a^2 - c^2}}{R + c}$$

for outside admission and

$$\sqrt{a^2 - c^2} : (R - c) = b : R$$

or

$$b = \frac{R\sqrt{a^2 - c^2}}{R - c}$$

for inside admission according to Formulas (10) and (11).

In this case b may be considered as equal to the radius of the eccentric crank, as it would be were it possible to attach the front end of the eccentric rod to the link block. This is laid out graphically in Figs. 29 and 30. If a is the same in both cases, b will be greater in Fig. 30 than in Fig. 29.

With these formulas as a basis it is possible to proceed with the determination of the actual radius of the eccentric crank. Starting with Fig. 27 as a basis, and having settled the travel of the valve and the throw of the point F, the next thing to determine is the distance Og which the link block will have to be moved from its central position to full gear. In this there are two antagonistic requirements to be reconciled. On the side of the angularity of the radius bar, it is desirable that this movement should be as small as possible, while with the angularity of the link in view it should be as large as it can be made. Hence it is necessary to compromise between the two.

Practical experience has shown that it is not well to swing the link through an angle of more than 45 degrees, so that assuming this as that to be used, we have:

$$Og = \frac{b}{\tan 22\frac{1}{2}\text{ deg.}} \qquad (12)$$

in which the angle given is that of half the travel on each side of the center.

The connecting point K between the eccentric rod and the link should be as near the center line of the engine as practicable. With inside admission piston valves it frequently happens, however, that the link fulcrum O is rather high, so that a large crank radius would be required in order to secure the requisite amount of motion. Consequently good judgment must be used in order to secure practical results. If, however, it is found necessary to locate this point K at any appreciable distance above the horizontal center line of the engine, the center of the eccentric crank should be set back with the same angularity so that a line drawn through it and the center of the axle will be at right angles to one drawn from the center of the axle to the point K.

The fore and aft position of K is a matter of importance and it should be such that it will swing through the same angle on each side of its central position at the same time compensating for the angularity of the rod.

No absolute formula can be given for the location of the point K, as it must be worked out for each individual case. It will always be back of the tangent to the link drawn through O in the central position, and the distance will depend upon the inverse ratio of the relation existing between the throw of the eccentric and the length of the eccentric rod. It is further influenced by the variation in the angularity between the center line of motion and the tangent to the link as well as its distance from the fulcrum O. Under ordinary conditions the point K will fall from 2 inches to 5 inches in the rear of a tangent to O.

With very short eccentric rods, there may be some difficulty in securing this equal angularity of swing, in which case the distance of K from the tangent to the link can be reduced somewhat and not materially affect the opening and closing points of the valve. The maximum port opening will be affected, it is true, but as this is usually more than that actually required, and as the irregularity gradually disappears as the cut-off is made earlier, it will have very little influence on the working of the engine.

Having obtained the horizontal motion of the link-block at the point g from Formulas (10) and (11) as well as the angular swing of the link, it is evident that the point K must move through the same angle; we thus have by making $k = b'$,

$$Og : OK = b : b'$$

or

$$b' = \frac{OK \times b}{Og} \tag{13}$$

This will also be the radius of the eccentric crank, but owing to the angle made by OK with the tangent to the link, the radius will decrease as this angle increases and must be laid out in each case to meet the conditions involved.

The location of the center Q of the reversing shaft (Fig. 28) must be such that the end of the arm QP at full gear forward will be in such a position that the lower end of the hanger will swing through an arc tangent to the radius bar at its point of attachment. An exact equality in this respect is impossible to attain throughout the whole range of cut-off, so that especial attention should be directed toward securing it over the range in which the engine is to work, which should be between a 30 and 60 per cent cut-off in forward gear for road engines, and half and full gear, front and back, for switching engines.

It will be found, then, that owing to the unavoidable irregularities inherent in the transformation of circular into rectilinear motion (inherent in all types of valve gears) the greater portion of the arc through which the point *P* passes is in one of the upper quadrants of the circle.

The crosshead connecting point *n*, Fig. 28, in relation to the point *m* at the lower end of the combination lever will also be found to have more or less effect upon the regularity of the motion, of which advantage can be taken in the laying out of the gear. Ordinarily the best position will be found by locating the point *n* on the same horizontal line with *m* when the combination lever is in a vertical position.

With the slide bearing pivoted directly on the lifting arm as indicated in Fig. 27, the movement of the end of the radius bar is prac-

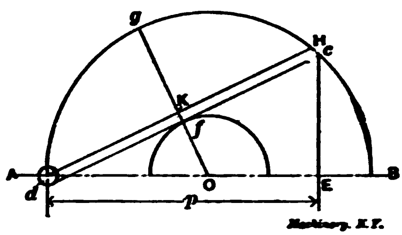


Fig. 31

tically in a straight line throughout the whole range of cut-off in both directions, a fact which greatly simplifies the construction of the gear, and is, for that reason, to be preferred to the use of a swinging hanger.

The valve diagram may be laid out in the same general way as with the Stephenson gear, but as there is no increase of lead for the earlier points of cut-off, a definite amount of lead must be given from the start, by which a complication is added to the formula for finding the lap for a given lead, valve travel and maximum cut-off. It can, however, be done as follows:

In Fig. 31 let *AB* be the travel of the valve and *E* the point of ·ximum cut-off. Draw *Ec* at right angles to *AB* and describe the circle *AcB*, on *AB* as a diameter. With the lead as a radius and a center, draw the circle *d*, and draw *cd* tangent to circle *d*. *Og* at right angles to *cd* and draw the lap circle tangent to *cd*, radius of which, *Of*, will be the required lap.

Further, draw *AH* parallel to *cd* and it will be the locus of the virtual eccentric center and

$$AO = a; \; OK = c$$

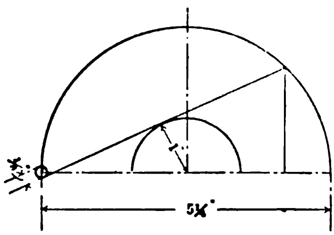
in Formulas (10) and (11).

Zeuner's diagram is as applicable here as in the case of the Ste-n gear with the difference that the locus of the virtual eccentric : line instead of a curve, and the points R^1, R^2, R^3, R^4 of

Fig. 9 determine the radii for different travel circles of the cut-off as well as the corresponding diameters of their respective valve circles.

As in the case of the designing of the Stephenson valve motion, it is necessary to make some arbitrary assumptions in order to determine the dimensions of the several parts by the use of the formulas that have been developed. This is done in connection with the diagram of Fig. 28. In this we have certain dimensions already given. The first is the stroke of the piston which is 26 inches. By taking the travel of the valve at 5¼ inches and its maximum point of cut-off at 0.83 of the stroke as in the case of the Stephenson gear and fixing the lead at ⅛ inch, the valve diagram of Fig. 32 can be constructed from which the lap of 1 inch will be obtained. The sum of the lap and lead will then be 1⅛ inch. As the half stroke of the crosshead is 13 inches the ratio of the lap and lead to this motion will be as 1 to 11.55.

By first laying off an outline of the main crank, the center line of the valve stem, the connecting rod and crosshead and a vertical line



Machinery, N.Y.

Fig. 32

passing through the center of the connection between the valve stem and the combination lever when it is in the central position, the proportions of the latter are first obtained. The length of the long arm should not be less than 2¼ times the stroke, if too great angularity of motion is to be avoided. In this case the attachment of the valve stem will be below the point *F*. If we assume the distance between the two to be 3½ inches then the total length of the rod becomes 40.415 inches or a little more than 40 13/32 inches. Laying off the lap and lead on the center line at *t*, the position of the combination lever at a forward end of the stroke is obtained. The radius of the link should not be less than eight times the travel, but must necessarily be adapted to the engine and varied according to the requirements of construction, assuming it in this case to be 42 inches. The length of the link is determined by the travel of the point *F*, which is ascertained from Formula (11) which by the substitution of values becomes

$$b = \frac{13\sqrt{2.625^2 - 1.125^2}}{13 - 1.125} = 2.6 \text{ inches.}$$

Then the half length of the link, *Og*, is obtained by the substitution of values in Formula (12), which then becomes

$$Og = \frac{2.6}{0.41421} = 6.3 \text{ inches.}$$

With the proper allowance for the length of the link block the link should be at least 8½ inches long on each side of O or 17 inches in all. If the point K of the attachment of the eccentric rod is taken at 11½ inches from the link fulcrum, the radius of the eccentric crank, in order to give the link a throw of 45 degrees will be

$$11.5 \times \tan 22\tfrac{1}{2} \text{ deg.} = 11.5 \times 0.4142 = 4.75,$$

or by substitution of the values in Formula (13)

$$b' = \frac{11.5 \times 2.6}{6.3} = 4.75$$

so that it can be made 4¾ inches.

This is, however, subject to slight modifications depending on the angle between the motion center and the radius OK, but in ordinary cases this is so insignificant that it may be left out of consideration.

22. CALCULATION OF ELEMENTS OF MACHINE DESIGN.—Factor of Safety; of Bolts; Riveted Joints; Keys and Keyways; Toggle-joints.
23. THEORY OF CRANE DESIGN.—Jib Cranes; Calculation of Shaft, Gears, rings; Force Required to Move Crane Trolleys, etc.
24. EXAMPLES OF CALCULATING DESIGNS.—Charts in Designing; Punch eter Frames; Shear Frames; Billet and Bar Passes; etc.
25. DEEP HOLE DRILLING.—Methods of Drilling; Construction of Drills.
26. MODERN PUNCH AND DIE CONSTRUCTION.—Construction and Use of s Dies; Modern Blanking Die Construction; Drawing and Forming Dies.
27. LOCOMOTIVE DESIGN, Part I.—Boilers, Cylinders, Pipes and Pistons.
28. LOCOMOTIVE DESIGN, Part II.—Stephenson Valve Motion; Theory, ion and Design of Valve Motion; The Walschaerts Valve Motion.
29. LOCOMOTIVE DESIGN, Part III.—Smokebox; Exhaust Pipe; Frames; ads; Guide Bars; Connecting-rods; Crank-pins; Axles; Driving-wheels.
30. LOCOMOTIVE DESIGN, Part IV.—Springs, Trucks, Cab and Tender.
31. SCREW THREAD TOOLS AND GAGES.
32. SCREW THREAD CUTTING.—Change Gears; Thread Tools; Kinks.
33. SYSTEMS AND PRACTICE OF THE DRAFTING-ROOM.
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CHAPTER I

SMOKEBOX AND EXHAUST PIPE ARRANGEMENT*†

The arrangement of the smokebox and the draft appliances contained therein ranks among the more important of the details in the designing of a locomotive, for upon it largely depends the efficiency of the whole machine. Perhaps it is on account of this very importance, coupled with the wide range of variables that must be taken into consideration, that there is as yet no standard construction that is acknowledged to be the one that will produce the highest efficiency under all conditions.

The problem to be solved, with some modifications, is to secure the largest possible exhaust nozzle, and uniform action on the fire over the whole surface of the bed.

The action of the exhaust is, for the most part, that of a jet of steam drawing air with it by the friction of its sides; though, at very low speeds, the plunger action is also influential since there is a perceptible interval of time between the exhausts. When the plunger action predominates, the form and diameter of the stack as well as the heights of the top and bottom of the same above the exhaust nozzles are matters of importance. The jet action is, however, the one that for the most part controls in all the working of the engine, but the smokebox details must be arranged to suit both. In considering the jet action, the length of the stack, and, consequently, the height of its top above the nozzle, is of minor importance. It is evident that the shorter the stack the less will be the frictional resistance of the gases, so that it is useless to extend it beyond the point where they have obtained their maximum velocity. On the other hand, were the steam allowed to escape into the atmosphere through too short a stack, the interval between exhausts would be sufficient to permit the air to rush back into the smokebox and firebox and, by destroying the partial vacuum that had been created, add very materially to the work that would have to be done. For this reason it is necessary to sharpen the exhaust by contracting the nozzle, thus prolonging the time of its action and increasing the velocity of the steam. On the other hand, if this contraction is made too great, there will be an excessive action

* The present number of MACHINERY'S Reference Series is the third part of a treatise on complete Locomotive Design, covered by Nos. 27, 28, 29, and 30 of the Series, and originally published in RAILWAY MACHINERY (the railway edition of MACHINERY). Each of the four parts of the complete work treats separately on one, or more, special features of locomotive design; and while the four parts make one homogeneous treatise on the whole subject, each part is complete by itself. In order to give concrete form to the examples and theoretical considerations, it is assumed that a consolidation freight locomotive and an Atlantic type passenger engine are being designed. It is further assumed that these locomotives are designed for a division 150 miles long, laid with rails weighing 75 pounds per yard, and with a ruling grade of one per cent ten miles in length.

† MACHINERY, Railway Edition, October, 1905.

and a breaking up of the bed of the fire, coupled with an undue back pressure in the cylinders.

The stack should be of such length that at moderate speeds one exhaust is entering at the bottom before the last of the preceding one has escaped at the top. At the same time it should be of such shape that resistances are reduced to a minimum. That this may be done it should increase in area in proportion to the loss of speed of the gases, by which means the inertia of those leading will be utilized to reduce the resistance of the succeeding ones; which, in turn, serves to increase the efficiency.

The exact amount of retardation of the gases in their passage from the nozzle to the air is only obtained by experiment, but it is evident that the stack should be flared, with the enlarged portion at the top. For practical purposes, however, the straight taper will answer every requirement with no noticeable difference from that of a form theoretically correct.

The most exhaustive experiments along this line that have thus far been made are probably those of Von Borries and Troske that were carried out in Germany in the early nineties and subsequently published in this country, from which it appears that the proper taper for a stack is approximately one in twelve and that its length should be three or four times the diameter at its smallest point or choke, a proposition that was confirmed by the committee of the American Railway Master Mechanics Association on Exhaust Pipes and Steam Passages in 1896.

The results of the Von Borries and Troske experiments may be approximately expressed by the formulas:

$$d = 0.156\sqrt{\frac{S \times R}{S + 0.3\,R}} \qquad (1)$$

in which

d = diameter of the exhaust nozzle,

R = area through tubes,

S = grate area,

all expressed in inches.

$$h = 14\,d \qquad (2)$$

in which

h = height of the top of the stack above the top of the exhaust nozzle of a straight pipe.

$$D = 3.8\,d \qquad (3)$$

in which

D = diameter of the stack at the top.

$$D' = 0.65\,D \qquad (4)$$

in which

D' = an imaginary diameter which the bottom of the stack would have were it to be drawn down to the level of the top of the exhaust nozzle.

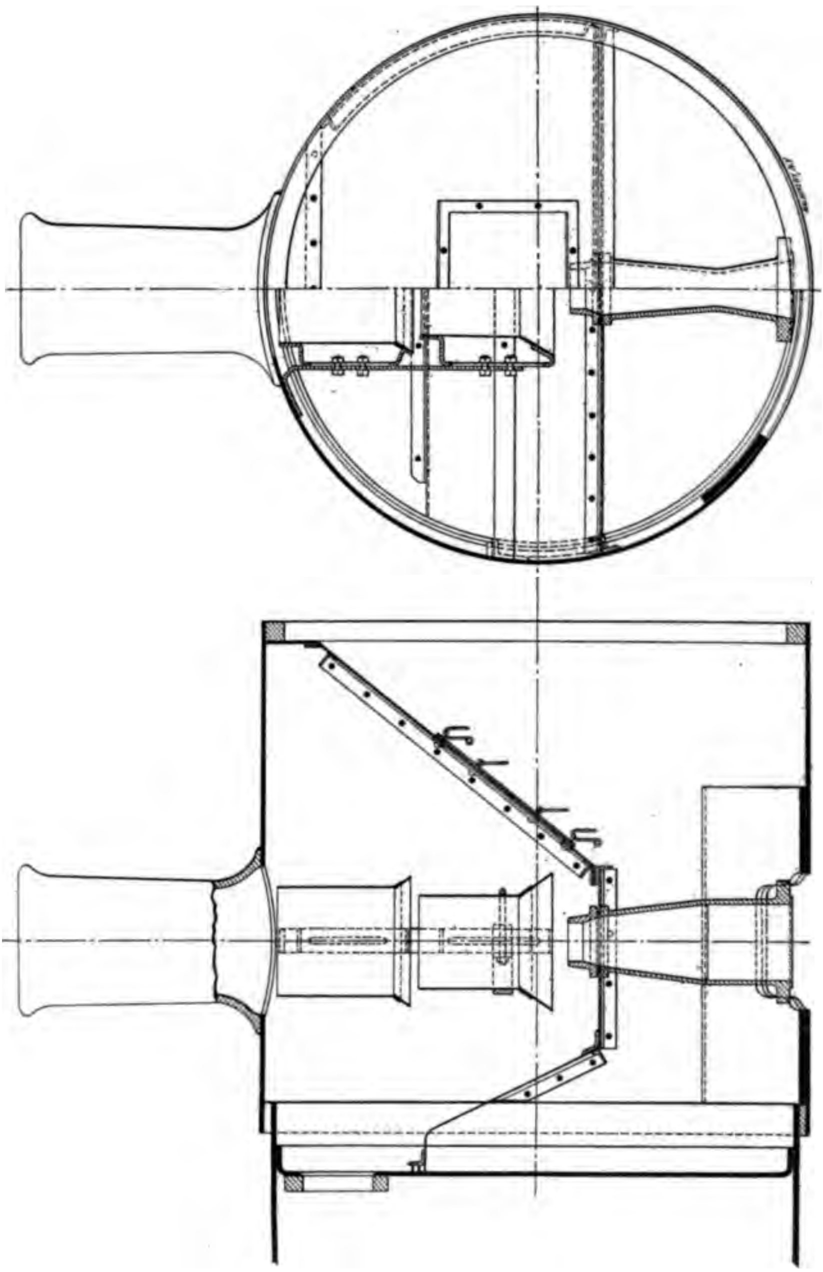

Fig. 1. Longitudinal and Cross-section of Smokebox of Express Passenger Locomotive

The height C of the choke above the top of the nozzle should not be more than 0.4 h or

$$C = 0.4\,h \qquad (5)$$

When circumstances do not permit the heights required by these formulas to be used, it is good practice to use at least the diameters of stack obtained by following the rules that have been laid down.

As for the effectiveness of the exhaust, it has been found that, under certain conditions, one pound of steam, discharged from the nozzle, will displace about 2.5 pounds of smoke-box gases, according to the following formula, that has been deduced from earlier experiments.

$$A = D\sqrt{\frac{2\left(\frac{T}{S}\right)^2\left(1-\frac{V}{S}\right)}{\frac{V}{S}\left[l+2\left(\frac{T}{S}\right)^2\right]}}$$

in which

A = weight of gases displaced in pounds,
D = weight of steam discharged through the nozzle in pounds,
V = area of exhaust nozzle in sq. inches,
S = area of stack in sq. inches,
T = area through tubes in sq. inches,
l = a coefficient (approximately 4).

The weight of air displaced is in direct proportion to the weight of the steam discharged through the exhaust nozzle, when all adjustments have been properly made.

According to the experiments referred to above, the area of the stack, if straight, should be about twelve times the area of the exhaust nozzle, and in one that is well proportioned and tapered, the effect of the nozzle can be increased 12 per cent above that of a straight stack.

The base of the straight stack should be from 30 inches to 36 inches above the top of the nozzle, and the length about three times its own diameter. The choke or smallest diameter of a taper stack should be about 24 inches above the nozzle and its area should be about nine times the nozzle area. The length of the stack should be from three and a half to four times the diameter of the choke, with an area at the top not more than twice that of the choke.

These rules run remarkably close to each other, though the latter is considerably older than the former. They have, however, not yet secured the recognition in this country that they have abroad, and we find a variation of smokebox arrangements in use. This is due, in all probability, to the great variety of fuel burned.

Taking up the smokebox details in general, it will be found to be advantageous to locate the nozzle as low as the area across the box will allow. The baffle plate or diaphragm should be carried down, from above the top row of tubes, to the top of the nozzle, from which point it should be continued horizontally forward well to the front of the

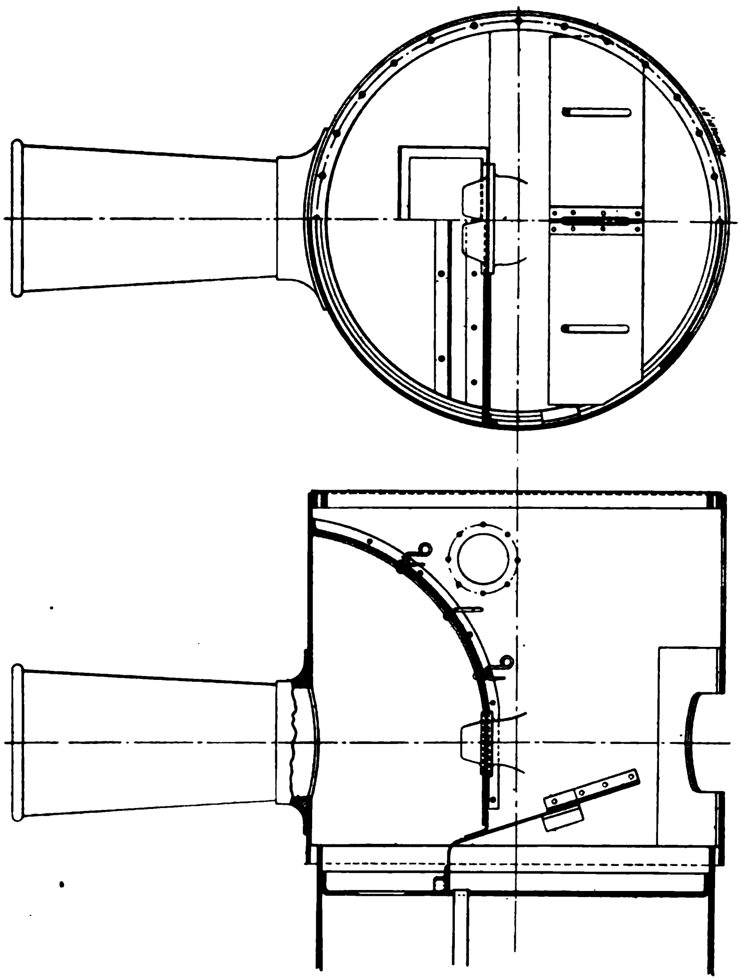

Fig. 2. Longitudinal and Cross-section of Smokebox of Consolidation Freight Locomotive

exhaust pipe, from which the inclined spark-arresting netting starts and extends to the top of the smokebox as shown in Figs. 1 and 2. This general arrangement has come into common use, and, in many cases, constitutes the so-called self-cleaning front end.

In its entirety, it consists of an adjustable diaphragm plate so located as to allow but a limited passage for the gases and sparks between it and the bottom of the smokebox. The diaphragm is carried well forward to limit the area of the passage so that the cinders and gases are kept in a state of constant circulation by the draft, with the result that the former are ground so small that they pass through the netting and out at the stack at a temperature below the igniting point, to which they have been cooled by the process.

The size of the smokebox is not a matter of much importance, except that a large one modifies the effect on the fire caused by the pulsations of the exhaust, on the same principle that an air chamber on a pump will cause a more uniform flow of water than would otherwise be obtained, whereas with a small box the fire will be more disturbed. On American locomotives the size of the smokebox has been reduced to a practical uniformity in the matter of length, ranging, as it does, from 6 feet to 7 feet, which is considered to be large enough to effect the desired uniformity in draft and affording sufficient room for the location of the diaphragms, netting and deflectors. Any length less than 4 feet would be considered small on a standard engine.

With the use of the low exhaust pipe, it is frequently found to be advantageous to introduce one or more draft or petticoat pipes between the exhaust nozzle and the bottom of the stack, as shown in Fig. 1, whereby, what amounts to several jets milder than that of the original issuing from the nozzle, are obtained. This holds especially for simple engines. For compounds it is advisable to extend the stack down into the smokebox to within 20 inches or 24 inches of the top of the nozzle and to secure the increased length of stack by which a more uniform action on the fire is obtained. In this arrangement of the smokebox no specific rules or proportions can be given for the working out of the details that will be of any value. The specific dimensions of the several parts will depend not only upon the size of the engine and the work that it is intended to do, but most particularly upon the fuel that is to be burned. For this a special adjustment is needed, and one that will give perfect satisfaction with a certain grade of coal will not be found suitable for another of a different character.

In Figs. 1 and 2 are given the longitudinal and cross-sections of the smokeboxes for the two engines that are being developed, from which the general proportions and arrangement of the parts according to modern approved practice may be determined.

CHAPTER II

THE FRAMES*

Although the frames are the foundation upon which the locomotive is constructed, they are not the first thing to be taken into consideration in the designing of the machine. As a support they can be varied in form to suit the requirements of the boiler, machinery and other parts and so only take on their final shape when these other parts have been arranged, though their attachments and presence must be borne in mind throughout the whole of the preparatory work. That they must receive careful consideration goes without saying, for upon them depends much in the matter of strength and rigidity, and perhaps even more in the way of maintenance, as broken frames are usually expensive to repair. In short, weak frames are a source of constant trouble in the way of failures and repairs, since they are frequently the cause of abnormal wear of the moving parts of the machine as well as a never ending, though sometimes obscure, agency in producing hot boxes.

As the width of the frame is necessarily limited, and, as the lateral stresses to which it is subjected are very great, it follows that it must not only be of ample size, but must be thoroughly braced so as to be able to withstand the shocks to which it is exposed when in service. It is impossible, however, to present any absolute formula by means of which the dimensions of a frame can be calculated. It is, of course, comparatively easy to calculate the stresses to which it will be subjected under the direct action of the steam in the cylinder, but this falls far short of being sufficient to provide for the diagonal and lateral stresses that are set up when the machine is worked heavily. These are of a very serious nature, and cannot be estimated when the speed is high or the track rough.

Experience shows that the lack of proper bracing is more often the cause of frame failures than the actual size of the sections used. For, if the frames are held rigidly in position both horizontally and vertically, these high running stresses become merely those of compression and tension. It is, therefore, apparent that the bracing or construction of the transverse frame work is a matter equal in importance to that of providing longitudinal strength. Hence, it cannot be too strongly enforced upon the attention of the designer that it is impossible to estimate the components of the forces that are set up in every direction, as in the case of the derailment of one or more pair of drivers. Nor can all of the varying conditions be ascertained as they exist at high speeds or on a rough track, where stresses of a momentary character are set up, that differ widely from second to

* MACHINERY, Railway Edition, November, 1905.

second, and where the momentum of the engine is an important factor in determining the intensity of the side blows that are delivered.

As already stated the stresses imparted by the working of the engine are comparatively easy to estimate and this can be done by the following formula:

$$F = \frac{P \times \pi d^2 c}{4 e} \qquad (7)$$

in which F = stress produced,

P = boiler pressure,

$\frac{\pi d^2}{4}$ = piston area,

c = distance from cylinder center to frame center on the opposite side of the engine in inches,

e = distance between frame centers in inches.

Practice has shown, however, that no matter what may have been the attempt to hold the frames in line, it is impossible to eliminate the bending moments, and that an allowance must be made for them as well as for the other incidental stresses that have been referred to. The most practical way of accomplishing this is to assume a suitable fiber stress for that portion of the frame section which is above the pedestals and this may be done by the formula:

$$A = \frac{F}{S} = \frac{P \pi d^2 c}{4 e S}, \qquad (8)$$

in which A = the section area of the frame,

F = total stress as obtained by Formula (7),

S = fiber stress.

For the point in question the fiber stress should be placed at 4,000. The same Formula (8) holds good for the upper frame section between the pedestals as well as for the lower rail. The value of S, the fiber stress, should be changed to 5000 for the former and 7500 for the latter. These values give the lowest practical limit that it is advisable to use for wrought iron frames. For cast steel frames, these values of S had better be made 3000, 4000 and 5500 respectively. In all cases the nearest larger even dimensions should be used, so that the sectional area may not fall below the requirements of these formulas.

For practical reasons the width of the frame section should not vary more than ½ inch for engines having cylinders more than 18 inches in diameter, and the ratio of depth to width should be kept as near as possible to 5 : 4, the section above the pedestal being taken as a base. The greatest width of the pedestal legs should not be less than ½ inch. This, naturally, causes a deviation from the figures obtained by the formulas, and it is here that the necessity for good judgment comes in; for the effect of the reduction of sectional area by bolt holes must be considered, and this is particularly true in the case of small engines.

When larger holes than those used for the ordinary frame bolts are put in to take such parts as the equalizer or brake hanger bolts, an

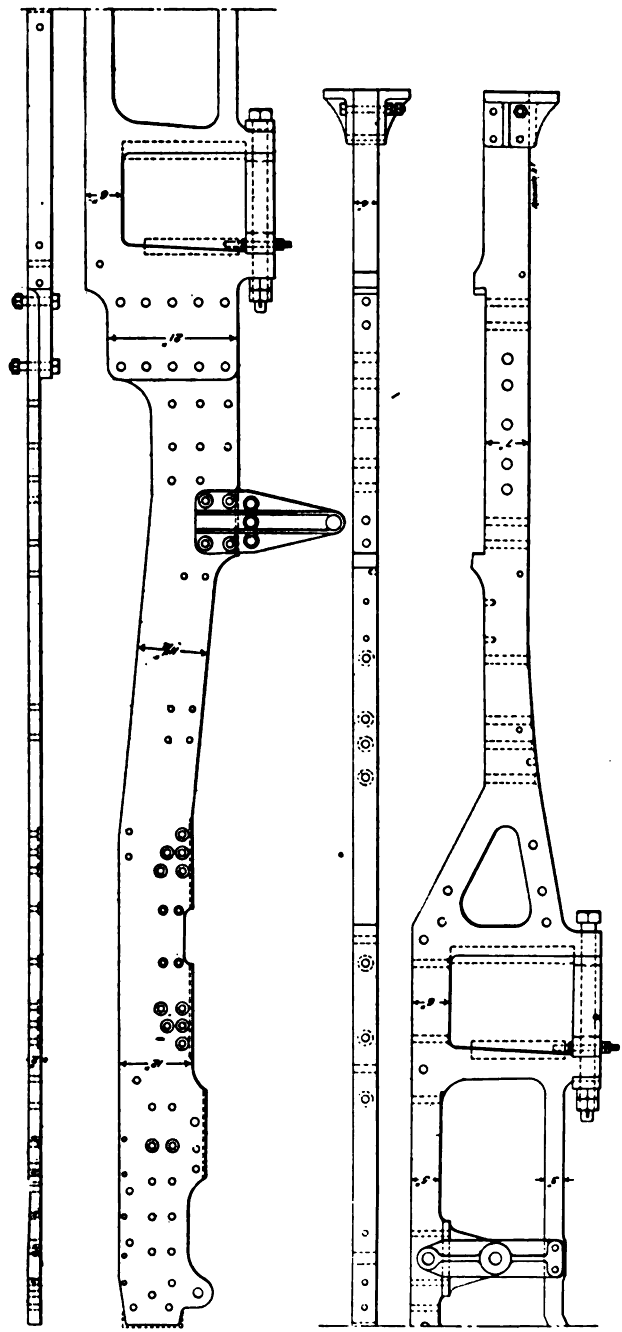

Fig. 3. Frame for Atlantic Type Passenger Locomotive

additional depth should be used at such places, to compensate for the material drilled out; and, in every instance, large fillets should be used at all unions between the horizontal and vertical members of the frame.

The front extension that passes beneath the cylinders when a single bar frame is used, should be of an area equal to the sum of the areas of the top and bottom rails of the main frame. When top and bottom rails are used at the cylinders, the former should have the same sectional area as the top rail of the main frame and that of the latter should not be less than that above the pedestals. Finally the splices or connections between the front extension and the main frame should be made as long as possible, and be well bolted and keyed together.

At all points where there is any fitting at the corners of the frames for cylinders or braces and especially for the pedestal shoes and wedges, these corners should be well rounded not only on the frame, but on the piece to be fitted to the same. As a rule, the wedges are fitted to the back leg of the pedestal so that the greater pressure exerted on the shoe and the vertical front leg, in running ahead, may bear perpendicularly against the face of the same. The slope of the rear leg where the wedge has a bearing usually ranges from ¾ to 1 inch in 12, which gives ample opportunity to adjust for wear.

Reverting now to the bracing for the frame, it may be again reiterated that it can hardly be made too substantial. The cylinder castings, which usually bear the greater portion of the burden of keeping the frames in line, should be relieved as far as possible by broad, well-ribbed cast steel cross-ties, having both horizontal and vertical extensions so as to form a rigid diagonal and transverse bracing between the upper and the lower rails, as well as by the horizontal and diagonal bracing that is needed between the frames themselves; and this bracing should be placed as close to the pedestal as possible.

No specific rule can be given either for the extent or the exact location of such bracing, because the requirements of the Stephenson link motion that is placed between the frames practically prevent the application of such bracing at points where it is most needed and where it would do the most good. The frequent results of this prohibition are to be found in a constant increase of frictional resistance and an ultimate breaking of the frames and cylinders. At the back end of the frames the footplate should extend along the frame for as great a distance as possible, as no adequate bracing can be put beneath the ashpan on the ordinary type of engine.

The pedestal binders on heavy engines should be of wrought iron with deep notches to receive the lower ends of the legs. For small and medium-sized engines, however, a bolt and thimble makes a satisfactory binder. For many years wrought iron was the only material used in the frames of American locomotives, and the preceding discussion has been based on this practice, although the cast steel frame has also been considered. The latter possesses some decided advantages over the forged frame and it is now being extensively intro-

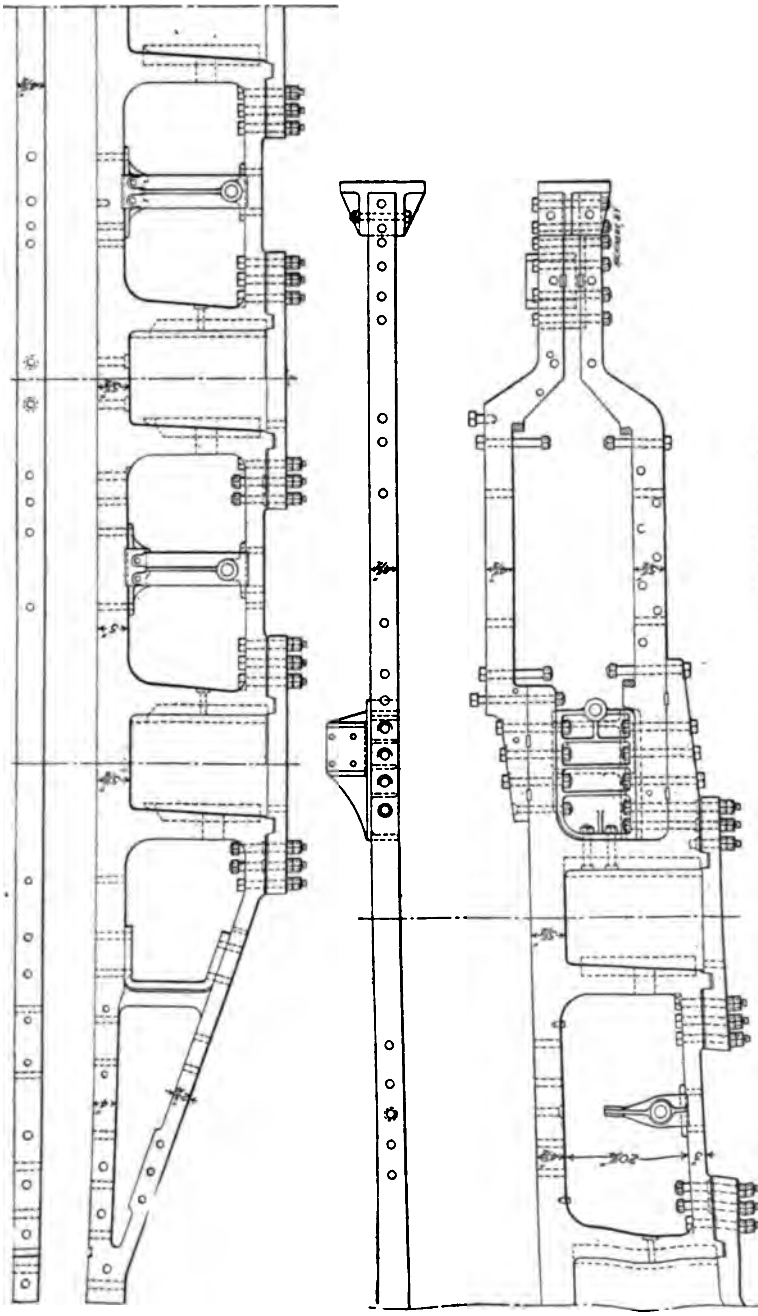

Fig. 4. Frame for Consolidation Freight Locomotive

duced. The chief of these advantages is the ability to use I-sections and otherwise to so distribute the material that it is disposed to the best advantage to resist the stresses to which it will be subjected. This makes a stronger and a lighter structure, but has the resultant disadvantage in the difficulty which this lighter section offers to welding, if it is broken. This disadvantage more than offsets the primary advantage of the best disposition of the metal, and current practice is now leaning strongly toward the use of the rectangular section.

Another advantage in the use of the cast steel frame is to be found in the possibility of casting on all brackets and of making special arrangements for the attachment of auxiliary parts such as brake cylinders, braces, hangers, and the like; so that it is probable that cast steel frames of a section corresponding to that of those of wrought iron will be increasingly used.

Turning now to the frames of the two engines that we have under consideration, let us see how their designs conform to the principles that have been thus far set forth. Fig. 3 shows the frame of the Atlantic type express locomotive, and Fig. 4 that of the consolidation locomotive.

In the case of the former an application of Formula (7) becomes

$$F = \frac{200 \times 298.65 \times 64.5}{43} = 89{,}579 \text{ pounds,}$$

in which

$$\frac{\pi d^2}{4} = 298.65.$$

Then from Formula (8) we have

$$A = \frac{89{,}579}{4{,}000} = 22.375 \text{ approximately.}$$

In this case 22.375 square inches is the lowest limit that would be allowable for the sectional area of the frame above the pedestal. In order to have even figures for the frame dimensions, the latter are given a depth of 6 inches and a width of 4 inches, producing a sectional area of 24 square inches. Likewise the upper frame section should, according to the formula and a fiber stress of 5,000 pounds, have an area of 18 square inches, which it has in the depth of 4.5 inches and width of 4 inches. The lower rail, with the fiber stress of 7,500 pounds should have an area of 11.9 square inches. It is made 12 by a depth of 3 inches and a width of 4 inches. In this case the single bar at the cylinder connection is made 7 inches deep instead of the 6 inches that is found over the pedestal, so as to compensate for the holes that are drilled there. In the consolidation locomotives the areas are less than those called for by the formula because of the distribution of the stresses through four pairs of driving wheels instead of two. Large fillets are used at the union between vertical and horizontal members, and in the case of the frame of the consolidation engine the junction of the front and main sections is made very secure by means of bolts and wedges.

CHAPTER III

CROSSHEAD AND GUIDE BARS*

The piston rods and the pistons are at the origin of the motion of the machinery, which must now be considered. Starting with the outer connection of the piston rod, the crosshead will be found to exist in various forms, from which a selection can be made to meet the requirements of any particular case. It should be borne in mind, however, that for heavy or high-speed engines the double-bar guide, or what is ordinarily known as the alligator type of guide and crosshead, will give the most satisfactory results, and is, consequently, the one that has been most extensively adopted. In fact, where conditions will admit of its being used, this type of crosshead will probably give the best satisfaction for any kind of an engine, even though it does necessitate the use of more metal by which the weight is greater than in a single-bar structure. In the designing of these parts it is also desirable to have the guides as close together as possible so as to reduce the distance to the piston rod to a minimum, and this holds for either a single- or double-bar construction.

As for the crosshead pin, it might be very small so far as its power to resist the working stresses is concerned, since these are applied in shear only, but it must be made large enough so as to provide an ample bearing surface for the brasses in the main rod. In short, it should be of such diameter and length that the load does not exceed 5000 pounds per square inch of projected area. The formula for these dimensions may be expressed as:

$$dl = \frac{P}{S} \qquad (9)$$

in which

d = diameter of crosshead pin,

l = length of crosshead pin,

P = total pressure upon the pin,

S = allowable pressure (5000 pounds) per square inch of pin.

In this it is necessary to assume either d or l, and the pressure P may be calculated by multiplying the area of the piston by the boiler pressure. The length is the dimension most commonly assumed and this is taken to suit the width of the crosshead.

In connection with the designing of the pin it should be kept in mind that, because of the action of the engine on curves and the side play in the driving boxes, the bearing on the crosshead pin is liable to wear faster at the ends than in the center. Under such conditions there will be a bending moment in addition to the shear, so that it is advisable to calculate the stress to which it will be subjected under

* MACHINERY, Railway Edition, December, 1905.

full boiler pressure from the formula:

$$S = \frac{P\,l}{4\,m} \qquad (10)$$

in which

S = the fiber stress to which the metal will be subjected,
P = total piston pressure,
l = length of crosshead pin,
m = moment of resistance of the circular section of the pin.

The moment of resistance of circular sections may be found from the formula

$$\frac{\pi\,d^3}{32},$$

and for the ordinary diameters of crosshead pins ranging from 3 to 4½ inches the moments of resistance are as follows:

3 inches,	2.66	4 inches,	6.28
3¼ "	3.38	4¼ "	7.53
3½ "	4.21	4½ "	8.95
3¾ "	5.18		

An examination of these figures will show that they vary as the cubes of the diameters.

Owing to the great amount of motion between the guide and the crosshead and the exposure of the surfaces to external influence, the pressure at this point should not be allowed to exceed 70 pounds per square inch, while it is desirable to drop well below these figures if possible, even cutting it down to one-half that amount, which is done in many instances.

In this, as in other cases, it is well to remember that large bearing surfaces are a good investment. The pressure against the guide can be calculated from the formula:

$$P' = \frac{P\,r}{L} \qquad (11)$$

in which

P' = the total maximum pressure against the guide,
P = maximum pressure on the piston,
r = radius of the crank in inches,
L = length of connecting-rod in inches.

This is an approximate formula that gives results sufficiently accurate for steam engine practice where the ratio between the connecting-rod length and crank radius is not less than 6 : 1. The formula for thrust which gives the theoretical reaction on the guide is $P' = P \tan\theta$ in which P' = thrust against guide; P = maximum pressure on piston; θ = greatest angle made by connecting-rod with axis of cylinder.

The area of the sliding surface of the crosshead will then be determined by

$$A = \frac{P}{t} \qquad (12)$$

where A = the required area of the sliding surface,
t = the allowable pressure per square inch.

According to the statement already made t should not be more than 70.

The strength of the guide should be such as to reduce the deflection to a minimum, which under no condition should be allowed to exceed 1/32 inch and should be kept down to 0.01 inch or less when circumstances will allow. In cases where the cylinder center is raised above, but is still parallel with, a line drawn through the centers of the driving wheels, this distance should be added to the radius of the crank r in Formula (11) as well as in the determination of the thickness of the guides, an allowance that will somewhat increase the value of the pressure against the guide, P', above what it would be were the crank radius alone used.

When the guides are forged the section is usually rectangular, in which case the moment of resistance is calculated from the well-known formula:

$$m = \frac{bh^2}{6} \tag{13}$$

in which
m = moment of resistance,
b = width of guide,
h = thickness of guide.

Having given the width, length and allowable fiber stress S, it is possible to calculate the thickness. The width is usually determined by the requirements of the construction, and the bars are fastened at both ends. This places the calculation on the basis,

$$\frac{P'l'}{4} = \frac{Sbh^2}{6}, \text{ or } h^2 = \frac{6\,P'l'}{4Sb}, \text{ and } h = \sqrt{\frac{3\,P'l'}{2Sb}} \tag{14}$$

in which l' = length of guide.

$P' = \dfrac{Pr}{L}$ of Formula (11) with the necessary correction for the height of the center of the cylinders above the driving wheel centers.

It is always desirable and will usually be found necessary to check off this determination of the value of the thickness of the guides h, in order that the deflection may not exceed the amount given above. This deflection may be determined by the formula:

$$f = \frac{P'l'^3}{4Ebh^3} \tag{15}$$

in which
f = the deflection in inches,
$P' = \dfrac{Pr}{L}$, as before,
E = the modulus of elasticity, which for steel may be placed at 30,000,000, and b, h and l' have the same values as in Formulas (13) and (14).

The thickness of the guides having thus been determined, it is always well to add from 1/8 to 1/4 inch to the amount so as to compensate for wear.

When the guide-yoke is set ahead of the rear ends of the guides the length of the latter should be considered as the distance between the yoke bolts and those in the lugs of the cylinder head. The guide-yoke to which the back ends of the guides are fastened and by which they are supported must be strong enough to sustain the guide pressure P'. The special form that should be given to the yoke is largely dependent

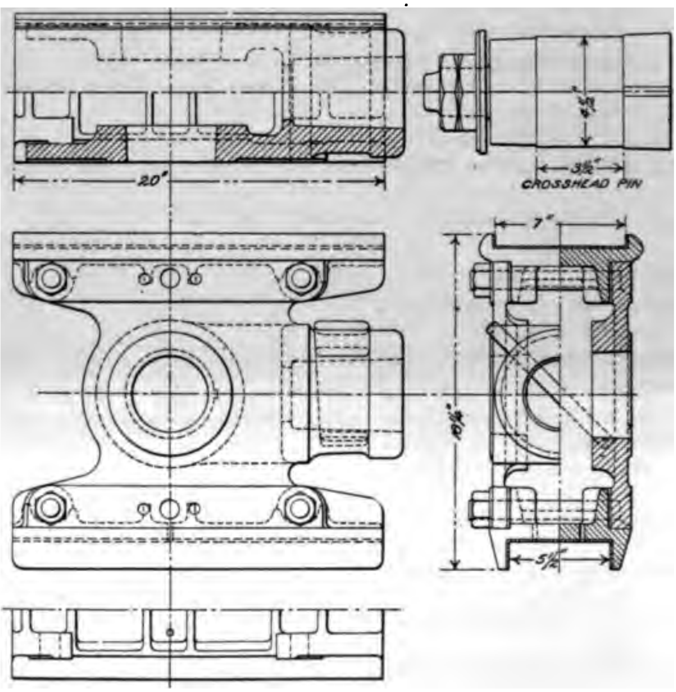

Fig. 5. Crosshead for Passenger Locomotive

upon the other conditions of the design of the engine as a whole, and no definite shape can be recommended that would not call for wide variations. The points to be borne in mind in this connection are that the guide-yoke should not only be strong enough to carry the guides and hold them rigidly in position but should also be made to serve as one of the most valuable and efficient braces for the frame. Its position is one where a substantial support for the frame is needed, and it is customary to take advantage of the opportunity thus afforded and utilize it to the utmost.

With the principles thus enunciated it is possible to make a direct application to the engines that have been kept under consideration, always bearing in mind that exigencies of construction may require a deviation, more or less pronounced, from the mathematical deductions, and that such a deviation is invariably made for the purpose of avoiding unusual or awkward dimensions.

Starting with the crosshead pin, by the substitution, in Formula (9), of the values for the cylinder dimensions and steam pressures assumed to have been decided upon, and by taking a safe margin from the maximum allowable stress of 5000 pounds, using 4000 for the passenger engine, we have:

$$dl = \frac{59{,}730}{4000} = 14.93,$$

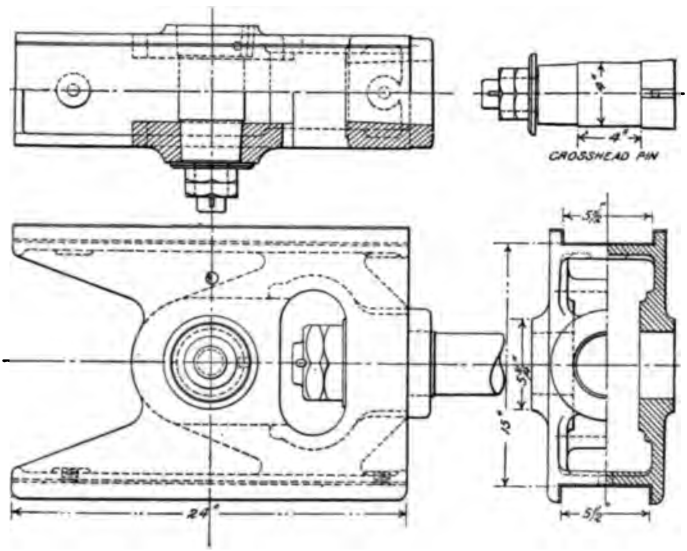

Fig. 6. Crosshead for Consolidation Freight Locomotive

while for the freight engine, where the speed is less, the 5000 pounds may be retained, giving

$$dl = \frac{69{,}276}{5000} = 13.85.$$

Figs. 5 and 6 illustrate the crossheads that have been designed for the passenger and freight engines respectively. In the case of the former, a pin 3½ inches long and 4.3 inches in diameter would satisfy the requirements; but in order to allow for some wear it has been made 4½ inches.

If these figures are checked by Formula (10) we have

$$S = \frac{59{,}730 \times 3.5}{4 \times 8.95} = 5840 \text{ pounds.}$$

This stress is much below what is required for strength in the pin, so that the size in this and other similar cases is governed by the area of the bearing surface rather than by strength. Pursuing a similar course for the freight engine to whose crosshead a length of 4 inches is suited to the other exigencies of the design, it will be found that with a fiber stress allowance of 11,000 pounds a diameter of 4 inches will be required.

Passing to the area of wearing surface on the crosshead, and substituting the values that we have already obtained in Formula (11) together with the length of the connecting-rod and noting that, in the case of the passenger locomotive, the crosshead center is 3 inches above

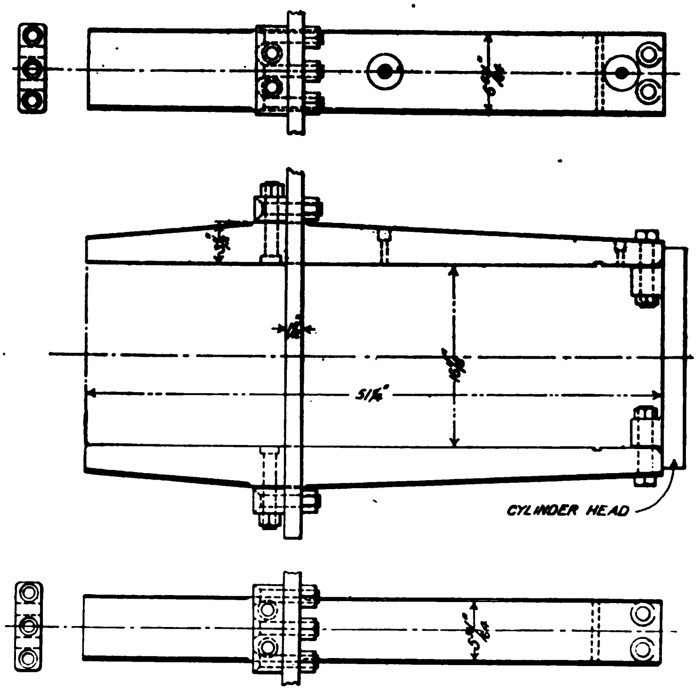

Fig. 7. Guide Bars for Passenger Locomotive

the center of the driving wheels, while in the freight engine it is 2 inches above, we have:

$$P' = \frac{59{,}730 \times 16}{125} = 7645 \text{ pounds for the passenger engine,}$$

and

$$P' = \frac{69{,}276 \times 15}{133.5} = 7784 \text{ pounds for the consolidation engine.}$$

As there is an opportunity in these cases to use ample wearing surfaces, the crossheads are made 20 inches long and 7 inches wide the passenger, and 24 inches long and 5.5 inches wide for the freight engine, dimensions which reduce the pressure per square inch to about 55 pounds and 67 pounds respectively.

The guides for the consolidation locomotive, illustrated in Fig. 8, have their width fixed by that of the bearing surface of the crosshead, while their length is determined by the length of the crosshead, the stroke and their own fastenings. With Formula (14) a thickness would be obtained very much less than that used. The dimensions adopted

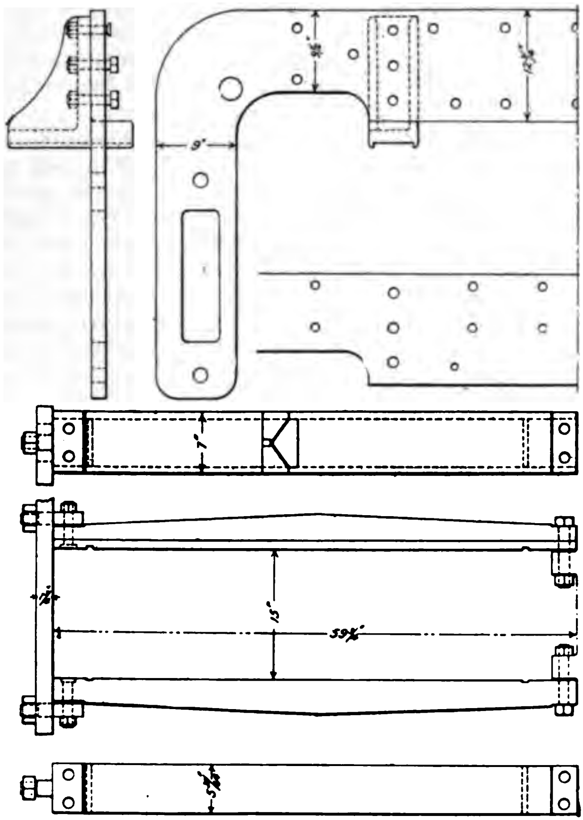

Fig. 8. Guides and Guide-yoke for Freight Locomotive

are rendered necessary by the desirability of securing rigidity in accordance with Formula (15), which then becomes

$$0.022 = \frac{7784 \times (59\tfrac{3}{4})^3}{120{,}000{,}000 \times 7 \times h^3}; \quad h = 4.5 \text{ inches approx.}$$

In examining the engraving Fig. 8, it will be observed that the upper bar is widened above the lip of the crosshead in order to increase the strength, and at the same time reduce the thickness to a minimum, as well as for a protection against grit, by preventing dust and cinders working in between the crosshead and the top guide.

Two designs of guide yokes are shown for the two engines in Figs. 7 and 8. The one for the passenger engine, Fig. 7, holds the guide near the center while the other holds it at the ends. Both are much heavier than the calculations for the mere static load would call for, to allow for incidental stresses due to various conditions such as derailments and other accidents that will put excessive stress on the yoke. In fact, the guide-yoke must be designed in accordance with good judgment and past experience of what is suitable and not as the result of mathematical work. It should be always borne in mind that these formulas are to be used as outlines and approximations rather than for the final and unchangeable determination of the dimensions that are to be used.

Before leaving this subject attention may be called to a few details of current practice in crosshead construction. Steel castings have replaced cast iron and forgings, and when made from a steel casting the crosshead is usually made in one piece. This is the case on the consolidation locomotive, Fig. 6. For the passenger engine, however, it is built up with plates bolted in position. A decided advantage gained by the use of steel castings for this work is that it makes it possible to very materially lessen the weight of the crosshead. This is an important item that the designer should always remember, that the weight of the reciprocating parts snould be kept down to the minimum, without a sacrifice of strength. In this, skill and judgment must be used and no invariable rule can be given for the disposition of the material where the stresses of operation must be cared for, while every effort must be made to avoid the internal stresses that may be set up in the process of casting and cooling.

The ideal location for the crosshead pin is at the center of the length of the crosshead, a condition that is realized in the case of that of the consolidation engine. But where this is impossible, as in the case of the passenger engine, it should be placed as near the center as possible. Especial attention should also be paid to the boss around the seat of the piston rod to see that it is of ample strength and not liable to be cracked by keying.

These precautions must be borne in mind in the designing, else the result will be a broken crosshead or piston rod which never ends with a simple break of the first part yielding, but invariably involves other portions of the engine in a break-down that is crippling and costly, including, as it usually does, at least a cylinder head, if not the cylinder itself.

CHAPTER IV

CONNECTING AND SIDE RODS*

The main or connecting-rod is the part of the mechanism by which the reciprocating motion of the crosshead is converted into the rotating motion of the crank-pin. It is subjected not only to the tensile and compression stresses of the piston-rod but also to other stresses due to the vertical motion as well as to buckling loads imposed by the compression thrust on long columns. In short it is subjected to the stresses of compression, tension, horizontal deflection or bending due to compression, and vertical deflection due to compression, centrifugal force and inertia at high speeds. It is, therefore, of the greatest importance that the section should be as light as possible and yet of ample strength to carry the loads imposed. That is to say, it should be of such form and dimensions as to make the most economical utilization of the material, a consideration that is being more and more severely imposed with the increasing powers and speeds of locomotives. Both experience and mathematical calculations show that these requirements are best fulfilled by the I-section. In the matter of the material, it should be the best obtainable.

As to the area of the section, it should be large enough to withstand the crushing and pulling stresses to which it will be subjected; but, because of the importance of securing lightness of construction a comparatively high fiber stress can be allowed.

As the vertical bending moment is far greater than the horizontal, the section should be deep with broad top and bottom webs, in order that the material may be placed in the most favorable position for resisting these stresses. The vertical web must be of sufficient thickness not to buckle under the compressive load.

An analysis of the condition under which the rod works will show that, when the compression stresses have been cared for, there will be ample material to withstand the tensile loads. A further analysis will show that in compression the horizontal bending moment is simply that of a column with square ends while the vertical is that of one with round ends, and that the centrifugal inertia due to the motion acts as a load practically at right angles to the axis. The first consideration is for the crushing stress which is obtained from the formula:

$$S = \frac{P}{A}$$

in which

P = pressure on the piston,

A = area of the rod,

* MACHINERY, Railway Edition, February and April, 1906.

S = allowable pressure per square inch,
or where S is given the formula becomes

$$A = \frac{P}{S} \qquad (16)$$

The same formula also holds for tension, and S may be taken to be the same, since the resistance of wrought iron and steel is about the same for both compression and tension.

The vertical bending moment is usually calculated on the basis of the piston pressure working at an estimated maximum speed in miles per hour equal to the number of inches in the diameter of the driving wheels. At such a high speed the piston pressure at mid-stroke will not be more than one-third of the maximum, so that the bending moment due to compression will be small, though that due to inertia will be maximum. The horizontal bending moment is based upon the full piston pressure, P, and it is at its maximum at slow speeds.

The calculation of the various stresses to which the connecting-rod is subjected is a complicated matter, but must be carefully worked out in order to avoid not only weakness, but excess of weight; for on the one side, a weak rod is exceedingly dangerous and is apt to cause a disaster if it breaks at high speed when it is under the greatest stress due to the centrifugal inertia; while, on the other hand, too much material will set up other disturbing elements that affect the smooth running of the engine.

After calculating the direct tension and compressive stresses in accordance with the formula given, the next step is to determine the lateral stresses on the rod when considered as a column with square end supports for which the following formula can be employed:

$$F = \frac{B}{1 - \frac{N B l^2}{10 E R^2}} = \frac{10 B E R^2}{10 E R^2 - N B l^2} \qquad (17)$$

in which

F = maximum compressive stress per square inch of concave side of column,

B = load per square inch of section of the rod $= \frac{P}{A}$,

E = modulus of elasticity = 30,000,000,

N = constant = ¼ for square bearing,

l = length of rod in inches,

R = radius of gyration of the section of the rod under consideration.

The same formula may be used for the vertical bending moment, except that as the rod has here become a column with rounded ends the value of N = 1 and the radius of gyration for the section must be taken about the horizontal axis. In both cases B is based on the full piston pressure.

The radius of gyration R is obtained by extracting the square root of the quotient of the moment of inertia I divided by the area of the section.

$$R = \sqrt{\frac{I}{A}} \tag{18}$$

in which

$$I = \frac{BH^3 - bh^3}{12}$$

where the several symbols of the second term of the equation are equal to the dimensions called for by the corresponding letters in Fig. 9.

When the speed is at the maximum it becomes necessary to combine the two bending forces due to compression and the inertia of centrifugal action respectively.

The general formula for centrifugal force is

$$C = \frac{Gv^2}{gr} \tag{19}$$

in which

C = centrifugal force,
G = weight of rod in pounds,
v = velocity in feet per second,
$g = 32.2$ = velocity acquired by gravity at the end of one second,
r = radius of motion in feet.

If the rate of revolution per minute is n then

$$v = \frac{2\pi r n}{60} = 0.1047\, r\, n$$

and

$$v^2 = 0.0109\, r^2 n^2$$

whence

$$C = \frac{0.0109\, G r^2 n^2}{32.2\, r} = 0.00034\, Grn^2 \tag{20}$$

Then taking the assumed maximum speed in miles per hour, V, as equal to the diameter, D, of the driving wheels in inches we have

$$n = \frac{V \times 5280 \times 12}{D \times \pi \times 60} = \frac{336\, V}{D} = 336$$

whence $n^2 = 112{,}896$.

Formula (20) then becomes

$$C = 38.38\, Gr \text{ or } 38.4\, Gr.$$

For convenience this may be converted into terms of the stroke, s, of the piston in inches.

$$s = 2r \times 12 = 24r$$

whence

$$C = \frac{38.4\, Gs}{24} = 1.6\, Gs. \tag{21}$$

This is the simplest form in which the centrifugal force can be expressed and is as applicable to the side as to the main rods, though the effects will differ with the difference in the motion of the two rods.

Taking the main rod first, the centrifugal force is at zero at the crosshead and at the maximum at the crank-pin. If the centrifugal motion were to be considered as though the whole rod were in circular motion, as in the case of the side rod, with the length of the rod $= l$ and the centrifugal force $= C$, then the load can be represented by a rectangle as indicated by the dotted lines in Fig. 10, in which case the rod would be supposed to be loaded uniformly throughout its whole length with a burden equal to the centrifugal force. But as there is no load at the crosshead end, the rectangle may be divided diagonally, forming a right-angled triangle whose apex is above the crank-pin and which may be taken to represent the centrifugal force as applied. The center of gravity of this triangle is at a distance of $l/3$ from the crank-pin, and it is at this point that the whole of the centrifugal force

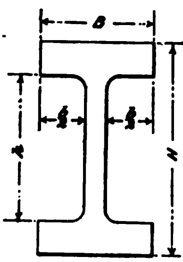

Fig. 9

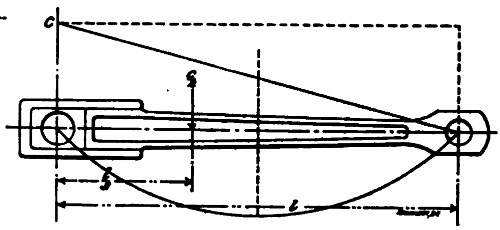

Fig. 10

($C/2$), as represented by the triangle, may be considered to be applied to the rod.

As, however, the main rod is heavier at the crank-pin than at the crosshead end, this is not strictly true because the load would fall somewhat nearer the crank-pin than $l/3$, but the maximum stress on the rod will fall between 1/3 and 1/2 the length of the rod from the crank-pin. The exact determination of this point would involve complicated calculations of no real value, in that a safety margin must be provided in any event, and the movement of the point of load application nearer the center merely decreases the margin.

It is, therefore, safer and better to assume the inertia load of $C/2$ to be at the middle of the rod or on the same section as the maximum vertical bending moment due to compression, working as indicated by the curved line in Fig. 10. As the rod is supported at both ends, the maximum moment (M) due to centrifugal action will then obtain when

$$M = \frac{C\,l}{2 \times 8} = \frac{1.6\,G\,s\,l}{16} = 0.1\,G\,s\,l \qquad (22)$$

The fiber stress (T) of the rod at this point due to inertia will then be

$$T = \frac{0.1\,G\,s\,l}{W} \qquad (23)$$

where $W =$ modulus of section.

The combined vertical bending stress will then be equal to the sum of that obtained by Formulas (17) and (23)

$$K = \frac{10\,B\,E\,R^2}{10\,E\,R^2 - N\,B\,l^2} + \frac{0.1\,G\,s\,l}{W} \qquad (24)$$

When the speed has reached the indicated maximum the value of B falls considerably below the full boiler pressure, because at mid-stroke the steam in the cylinder is, as already stated, rarely more than one-third the initial pressure. But as a further precaution in the case of temporary spurts of speed, as in the slipping of the wheels, B may be taken as equal to $\frac{P}{2A}$.

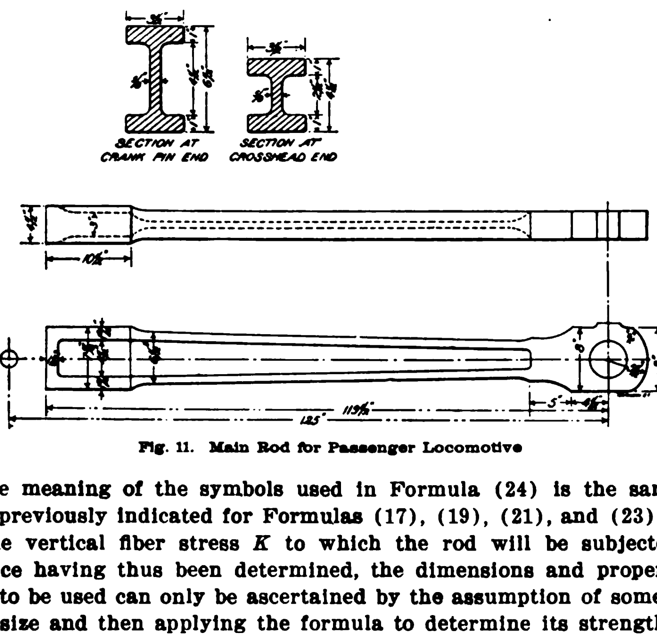

Fig. 11. Main Rod for Passenger Locomotive

The meaning of the symbols used in Formula (24) is the same as that previously indicated for Formulas (17), (19), (21), and (23).

The vertical fiber stress K to which the rod will be subjected in service having thus been determined, the dimensions and proper section to be used can only be ascertained by the assumption of some definite size and then applying the formula to determine its strength. If the section is found to be too light or too heavy, it must be increased or decreased accordingly until the proper figures are found that will meet the requirements of the case in hand.

The stresses at the several points of the connecting-rod cap be shown graphically by laying out a momentum curve on the lower side of the rod and an inertia triangle above it as shown in Fig. 10.

In the case of the side rod, the inertia effect due to centrifugal action is represented correctly by the rectangle of Fig. 10, which is equivalent to a load uniformly distributed over its whole length. As for the compression stresses due to the thrust of the piston, these are probably greater at low than at high speeds. The strain on the side rod must always be that of overcoming the slip of the coupled wheels to which it is connected, due to the slight inequalities of circumference that always exist, and the resistance of its own inertia stresses.

Turning now to the application of the principles that have been laid down for the rods of the engines whose parts have been used as a basis of comparison, the main rod of the passenger engine is shown in Fig. 11, the side rod for the same in Fig. 12. The main and side rods of the consolidation freight locomotive will be shown later.

In the case of the passenger locomotive the diameter of the cylinder is 19½ inches, and the boiler pressure 200 pounds per square inch. This makes the pressure per square inch of area $B = \dfrac{59{,}730}{A}$ pounds, and for the consolidation locomotive with 21-inch cylinders $\dfrac{69{,}276}{A}$ pounds.

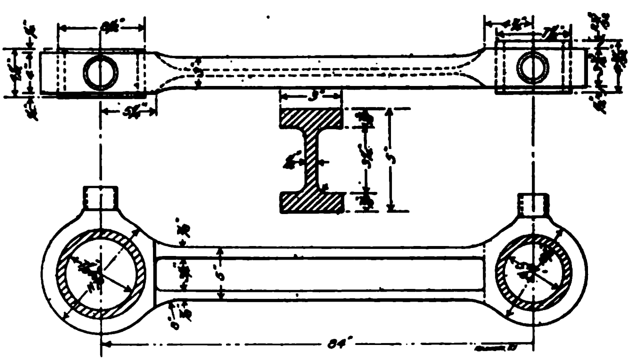

Fig. 12. Side Rod for Passenger Locomotive

The diameters of the driving wheels (D) of the two engines are 72 inches and 57 inches respectively so that the maximum speeds for which the rod stresses should be calculated are 72 miles per hour for the Atlantic and 57 miles per hour for the consolidation engine.

The first thing to be done is to determine the minimum area of the section of the rod according to Formula (16) in which it is necessary to assume the value of S which may be placed at 7000 as an approximation to meet requirements for the other stresses. For the passenger locomotive the formula then becomes:

$$A = \frac{59{,}730}{7000} = 8.53 \text{ square inches.}$$

As already stated, the method of determining the values of the various factors in Formula (24) is to assume a given section and weight of rod. When the radius of gyration and section modulus of this section have been substituted in the formula, the value of K, as so determined, must not exceed the allowable stresses that are settled empirically.

If, then, we take the main rod of the Atlantic locomotive as shown in Fig. 11, the weight will be found to be 404 pounds.

From Formula (18) the radius of gyration R will be found to be equal to 2.03 and the section modulus

$$W = \frac{I}{q} = \frac{38.2539}{2.75} = 13.9$$

in which q = the distance from the neutral axis to the outer fibers.

By the substitution of these values Formula (24) becomes

$$K = \frac{10 \times 3500 \times 30{,}000{,}000 \times 4}{10 \times 30{,}000{,}000 \times 4 - 1 \times 3500 \times 15{,}625} + \frac{1 \times 400 \times 24 \times 125}{14}$$
$$= 12{,}239.$$

$K = F + T = 3667 + 8572 = 12{,}239$ in the above demonstration, or

$$F = \frac{10 \times 3500 \times 30{,}000{,}000 \times 2^2}{10 \times 30{,}000{,}000 \times 2^2 - 1 \times 3500 \times 125^2} = 3667 \text{ pounds per square}$$

inch for vertical compressing bending stress under high speed, where

$$B = \frac{P}{2A} = \frac{60{,}000}{2 \times 8.5} = 3500 \text{ pounds.}$$

$$T = \frac{0.1 \times 400 \times 24 \times 125}{14} = 8572.$$

This falls within the stresses allowable for steel in this position which should not exceed 14,000 pounds fiber stress, and should be held as much below this as possible. In this calculation the area and section at the center of the rod are used as the basis of the work. The formulas can be applied in the same way to the side rods as well as to all the rods of the consolidation locomotive. The details of the construction of these rods will be discussed later.

Side Rods

It will be seen by a reference to Fig. 12, which represents the side rod of a passenger locomotive having but two pairs of driving wheels, that the construction is exceedingly simple and that all of the keys and gibs that formerly constituted so unimportant a part of the rods of engines have been entirely dispensed with. The rod in the present construction consists merely of a fluted bar with solid ends into which brass bushings are pressed. These are made to fit over the crank pins with an easy play and afford no means of adjustment to take up the wear. Such rods are the present universal practice on American locomotives. They are used until they become so worn that the pound on the pins is objectionable, when the bushings are renewed.

The side rods of a consolidation locomotive as well as those used on engines having more than two pairs of driving wheels, such as the ten-wheel and mogul classes, require a special arrangement. It is evident that the moving of a locomotive over a rough or uneven track involves a variation in the height of the driving wheels so that a rigid rod extending the full length of the wheel base would be impossible to operate safely. This necessitates the use of a horizontal joint at each

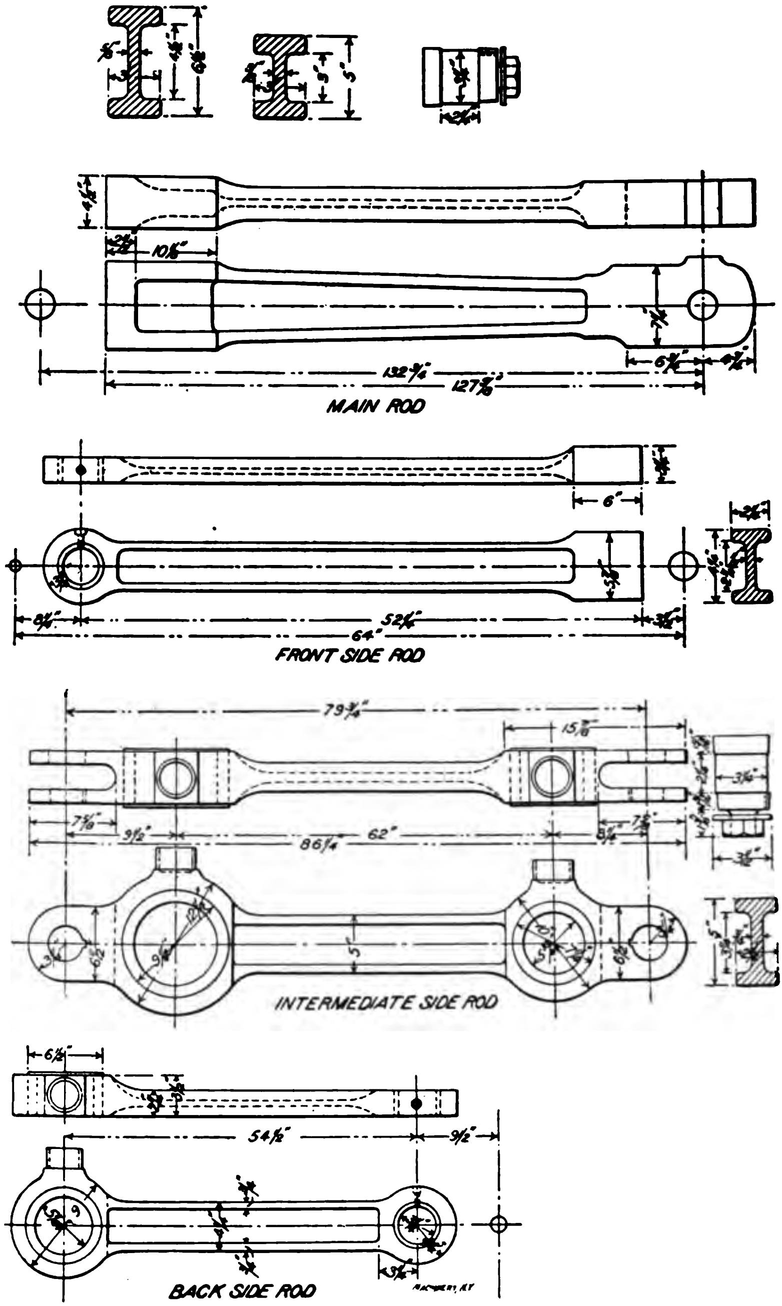

Fig. 13. Main and Side Rods for Consolidation Locomotive

wheel so that there may be a free vertical movement between the wheels without causing any cramping of the rod.

The general form of these rods is the same as that shown for the passenger locomotive with four driving wheels. That is to say, the

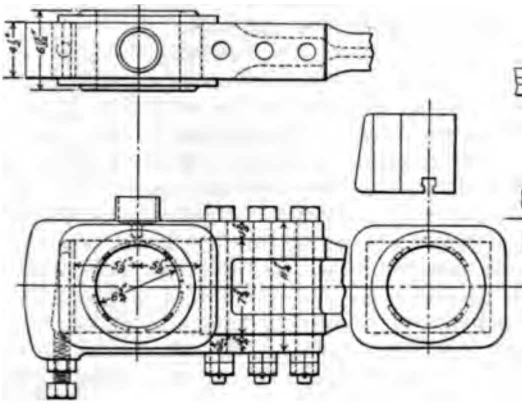

Fig. 14. Main Crank-pin Connection for Connecting-rod

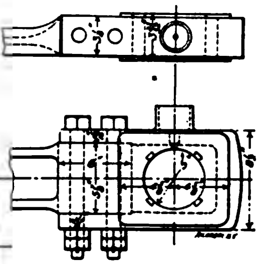

Fig. 15. Front End of Front Section of Side Rod

body of the rod is fluted and the bearings are solid brasses that admit of no adjustment; but in the case of the consolidation locomotive that we have in view, the side rod consists of three sections. The principal rod reaching from the main crank-pin to the pin in the third pair of

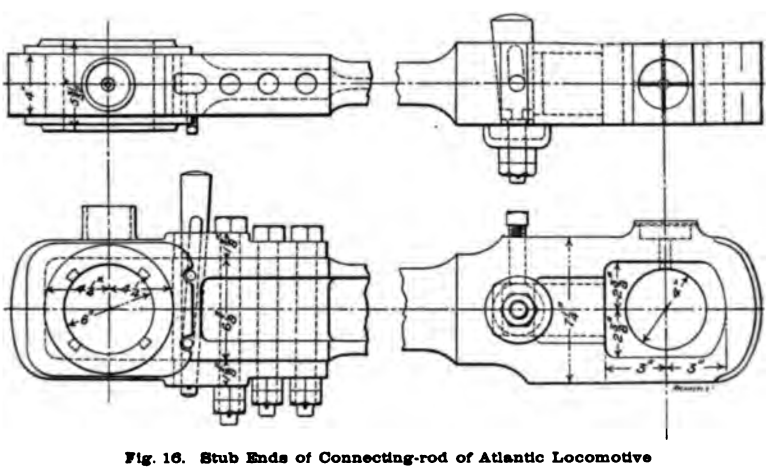

Fig. 16. Stub Ends of Connecting-rod of Atlantic Locomotive

wheels is solid like the passenger engine rod; and, as far as its action in connection with these two pairs of wheels is concerned, is the same. Each end of this rod carries a prolongation forming a knuckle joint with the stud ends attached to the front and back sections of the rod. The pin for these places has a tapered head at the back to fit into a counterbore in the back lip in order to clear the face of the wheel. The rear section of the rod is also solid as far as the connection between the two rear pairs of drivers is concerned and is pivoted in the fork of the intermediate section. Thus, as far as adjustment is concerned, there is none possible longitudinally between the three rear pairs of wheels of this consolidation locomotive.

This is as far as it has been found to be advisable to carry this scheme of non-adjustment. Hence the front section of the side rod is provided with a stub and strap at its front end, by which the distance between the centers of the front and back crank-pin brasses can be adjusted. Even here, however, the adjustment is made with liners and not with keys and wedges. Fig. 13 shows the general construction of the main and side rods on the consolidation locomotive. Fig. 15 shows the arrangement of the details of the stub end at the front end of the front section of the side rod. The brass is shown with babbitt pockets, and is in a strap to which the oil cup is forged solid and held by two bolts of 1⅛ inch diameter. Any adjustments as to length of this rod necessitates the removal of the strap, the filing of the brass and the insertion of liners. Fig. 14 shows the arrangements of the details of the stub end that is used for the main crank-pin connection of the connecting-rod. In this the strap is held to the end of the rod by three bolts of 1⅝ inch diameter each, and an adjustment is provided at the outer end by a wedge acting against the brass, which can be moved up and down by a screw bolt. Of course no adjustment can be made here without taking the brass out and filing it. There are many forms of stub ends that vary in detail from those shown, and which are usually based on a matter of choice in construction rather than any inherent merits of one over the other, the general principles of all being about as stated here.

CHAPTER V

CRANK-PINS AND AXLES*

Turning to the subject of the crank-pin the two considerations that are most prominent are the bending stresses to which it is subjected and the necessity for the provision of a proper bearing surface. The latter is usually the one that determines the size of the pin; but whether it be the limitatiion of the fiber stress due to bending, or the necessity of providing an ample bearing surface, the one calling for the larger diameter of pin is the one that should be selected to control. As regards the distribution of the work to the several pins, it is a problem which has not yet been solved, as the loads vary constantly, so that it is impossible to lay down any hard and fast rule in regard to it. If the bearings were perfect and without any play, the pins might have a truly proportional load to carry; but, as this is never the case, the best that can be done is to agree upon an arbitrary division. There are moments, for instance, when the main crank-pin will have to sustain the whole load, as when it is on the dead center and the side-rod bearings have considerable play. In other positions of the crank, the side rod is counteracting the stress on the main pin, so that if this assistance from the side rod is considered, a comparatively high fiber stress can be allowed. This pin should have a wheel fit somewhat larger than the bearing. The length of the leverage of the main rod on the pin is usually taken from the wheel hub to the center of the bearing, and the diameter at the former point, that is, just outside the wheel hub, as shown in Fig. 18, may be found by the formula

$$D = \sqrt[3]{\frac{P l \times 32}{\pi S}} = \sqrt[3]{\frac{10.2 P l}{S}} \qquad (25)$$

In which

D = diameter of pin just outside the wheel fit,
P = the total pressure exerted on the pin by the piston,
S = the allowable fiber stress for the material,
l = the length of the pin as shown in Fig. 18.

The fiber stress should be limited to 16,000 pounds per square inch for steel and 14,000 pounds for wrought iron.

There should be only a small reduction of diameter of the pin for the bearing so that the requisite surface can be obtained without making the pin too long. This usually calls for so large a diameter that calculation of strength at the shoulders is unnecessary. The required projected area for the bearings must be such that the pressure per square inch does not exceed 1600 pounds per square inch, or

* MACHINERY, Railway Edition, April, 1906.

$$\frac{P}{D\,l'} = \text{or} < 1600 \qquad (26)$$

in which the letters have the same meaning as in the case of Formula (25), except that l' = length of bearing on the pin.

The side-rod crank-pins will never have any greater pressure brought upon them than that needed to slip the pair of wheels to which they belong so that the general principle is established that the maximum stress to which a side-rod pin can be subjected is that required to slip

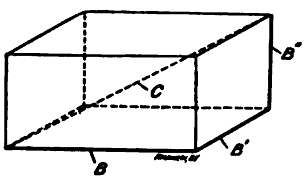

Fig. 17. Crank-pin Stress Diagram

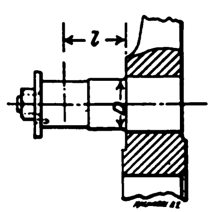

Fig. 18. Dimensions for Formula (25)

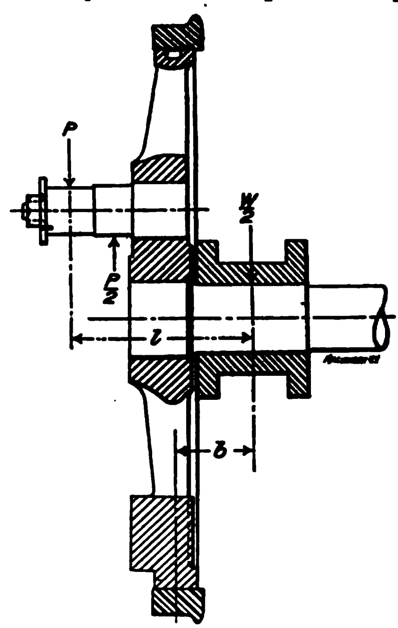

Fig. 19. Dimensions for Formulas (28) to (32)

the wheels to which it belongs. This may be calculated by the formula

$$P' = \frac{0.3\,W'\,D}{g} \qquad (27)$$

in which

P' = the frictional resistance of the driving wheels against the rails, transferred to the crank-pin,

W' = weight of the wheel with its load on the rails,

D = diameter of the driving wheels, in inches,

g = stroke of the piston, in inches.

In order to find the diameter of the pin, the value of $P' \times l'$ is substituted in Formula (25) for $P \times l$. In this the fiber stress should be placed at 12,000 pounds for a side-rod pin instead of the 16,000 pounds used with the main pin, inasmuch as certain conditions, for which it was allowable there, do not exist here in the case of the side-rod pins. Having found the diameter required to meet the required fiber stress, the bearing surface may be proportioned according to Formula (26) substituting, however, the value of P' as given in Formula (27). It will be observed that the values of P and P' as given in these three formulas indicate the highest stress that can be applied to the pins, so that the working pressure of 1600 pounds per square inch can only be exerted at very low speeds when the full boiler pressure is put upon the piston.

It is quite apparent, without argument, that the driving axles are among the most important of the details of a locomotive. Like the main rod, they are subjected to many and varying forces, such as the horizontal and vertical bending moments and torsion. The vertical bending moment, when considered as static, is the smallest to which the axle is subjected and is readily calculated. On the other hand, the shocks to which the part is exposed at high speeds are almost impossible to determine. At the same time the torsional stress under those same conditions, is comparatively slight, so that the increase of one may be considered to compensate for the decrease in the other. The horizontal bending force is due directly to the pressure exerted by the piston. The same is true of the torsional stress, whereas the vertical bending moment is due to the weight on the axle.

The torsional stress is never in excess of P' in Formula (27) since anything above this is transferred through the side rods to the other drivers, as one crank is in a favorable position to slip the wheels, when the other is on the dead center, thus taking up all of the slack in the rods, and relieving the axle from the full bending force of the piston. It is customary, therefore, to consider this bending moment as equal to one-half that exerted by the piston or $\frac{P\,l}{2}$, and the formula for the horizontal bending moment alone would be

$$B = \sqrt[3]{\frac{32\,P\,l}{2\,\pi\,S}} = \sqrt[3]{\frac{5.1\,P\,l}{S}} \qquad (28)$$

in which

B = the required diameter of an axle that would resist the horizontal bending moment,

P = the piston pressure,

l = distance between center of main rod and center of journal box,

S = allowable fiber stress.

The weight on the axle is a constant load and must, therefore, always be taken into consideration in connection with the horizontal bending moment, whereby the resulting bending moment of these two forces is equal to the hypotenuse of a right-angled triangle of which they are, themselves, the two sides. Consider, then, the total load on the axle as equal to W and the horizontal distance between the center of the rail and the center of the driving box as b, as indicated in Fig. 19. The vertical bending moment for $\frac{W}{2}$ or that for each box will be obtained when

$$B' = \sqrt[3]{\frac{32\,W\,b}{2\,\pi\,S}} = \sqrt[3]{\frac{5.1\,W\,b}{S}} \qquad (29)$$

in which

B' = the vertical bending moment.

When the crank stands on the quarter the torsional stress is to be added and may be expressed in the same way as the bending forces. If this stress of torsion is indicated as B'', we then have

$$B'' = \sqrt[3]{\frac{16\,t\,r}{2\,\pi\,S}} = \sqrt[3]{\frac{5.1\,t\,r}{2\,S}} \qquad (30)$$

in which

t = the resisting force at the crank-pin to the turning or slipping of the wheels,

r = radius of crank, in inches.

This value t is to be found in the same manner as that of P' in Formula (27).

Thus

$$t = \frac{0.3\,W'\,D}{g} \qquad (31)$$

in which

W' = total load of one pair of wheels on the rail,

D = diameter of drivers, in inches,

g = stroke of piston, in inches,

0.3 = coefficient of friction.

It will be noted, of course, that all of the power required to turn or slip a pair of wheels does not go through the axle. One-half is absorbed directly by the wheel to which the crank is applied, and the other half, or only enough to turn the wheel at the opposite end of the axle, goes through the latter, thus reducing the torsion by one-half, a condition that is provided for in Formula (30) by the constant 2 in the demoninator of the general formula for torsion.

By combining Formulas (28), (29), and (30), it is possible to determine the required diameter D of the axle, thus:

$$D = \sqrt[3]{\sqrt{\left(\frac{5.1\,P\,l}{S}\right)^2 + \left(\frac{5.1\,W\,b}{S}\right)^2 + \left(\frac{5.1\,t\,r}{2\,S}\right)^2}}$$

reduces to the form

$$D = \sqrt[3]{\frac{5.1}{S}\sqrt{(P\,l)^2 + (W\,b)^2 + \left(\frac{t\,r}{2}\right)^2}} \qquad (32)$$

h is preferable as decreasing the size of the number to be squared. le problem can be indicated graphically by measuring the a parallelopiped, where the value of B obtained in length; that of B' in Formula (29) the width, and (30) the height, as shown in Fig. 17.

In applying Formula (32) to the determination of the diameter of the axle for the consolidation locomotive, the following values for the several symbols will be obtained:

$P = 69{,}270$, or in round numbers 70,000 pounds,

$l = 22$ inches,

$W = 40{,}000$ pounds less the weight of the wheels and axles or 32,000 pounds,

$b = 10$ inches = horizontal distance from center of rail to center of box,

$$t = \frac{0.3\,W'\,d}{g} = \frac{0.3 \times 40{,}000 \times 57}{26} = 26{,}000 \text{ pounds},$$

r = radius of crank = 13 inches,

S = allowable fiber stress = 16,000 pounds.

With these values the diameter of the axle (D) becomes

$$D = \sqrt[3]{\frac{5.1}{16{,}000}\sqrt{(70{,}000 \times 22)^2 + (32{,}000 \times 10)^2 + \left(\frac{26{,}000 \times 13}{2}\right)^2}}$$

This may be simplified by cancellation if written

$$D = \sqrt[3]{\sqrt{\left(\frac{5.1 \times 70{,}000 \times 22}{16{,}000}\right)^2 + \left(\frac{5.1 \times 32{,}000 \times 10}{16{,}000}\right)^2 + \left(\frac{5.1 \times 26{,}000 \times 13}{2 \times 16{,}000}\right)^2}}$$

and

$$D = \sqrt[3]{\sqrt{\left(\frac{5.1 \times 70 \times 11}{8}\right)^2 + (5.1 \times 2 \times 10)^2 + \left(\frac{5.1 \times 13 \times 13}{16}\right)^2}}$$

$$D = \sqrt[3]{\sqrt{(491)^2 + (102)^2 + (53.8)^2}} = \sqrt[3]{\sqrt{241{,}081 + 10{,}404 + 2894}}$$

$$D = \sqrt[3]{504} = 7.96 \text{ inches.}$$

By making a suitable allowance for wear and truing, the axle is made 9 inches in diameter at the journal.

The front and rear axles or all except the main driving axle may, of course, be made smaller, but it is customary to make them all of the same diameter, though exceptions to this rule are frequent. The same formulas may be applied to their determinations by the substitution of the proper values for the several symbols. Attention should be called to the fact that the horizontal bending moment, which is the greatest in the case of the main axle, is reduced to less than one-third on the other axles of a consolidation locomotive.

Turning now to the crank-pin and applying Formulas (25), (26), and (27) to the consolidation locomotive, the bending forces of the main pin at the wheel seat will give a diameter

$$D = \sqrt[3]{\frac{10.2\,P\,l}{S}} = \sqrt[3]{\frac{10.2 \times 70{,}000 \times 8.375}{16{,}000}} = \sqrt[3]{373} = 7.19 \text{ inches.} \quad (25)$$

The diameter of this crank-pin is made 7¼ inches, which is quite sufficient when the liberal allowance made in the formula for the pressure on the piston is considered, as this is rarely reached, and the statement is emphasized by the fact that no allowance has been made for the reaction of the side rod when the wheels are slipped into bearing by the opposite crank, which is on the quarter at the time the maximum stress comes on the one at the dead center where the calculation is based.

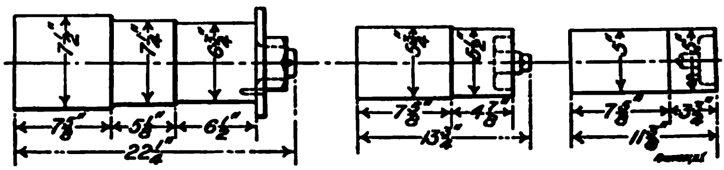

Fig. 20. Crank-pins used on Consolidation Locomotive

This diameter will be that of the side-rod bearing which, because of its large diameter, can be made very short. The pressure will seldom be more than three-quarters that of the main pin, and usually will be less. However, it will be found that, in the case of this consolidation locomotive the bearings are made longer than the 1600 pounds pressure per square inch required, for reasons of construction. Thus

$$l = \frac{3 \times 70,000}{4 \times 7.25 \times 1600} = 4.5 \text{ inches approx.}$$

But for the reasons just given it is made 5⅛ inches long.

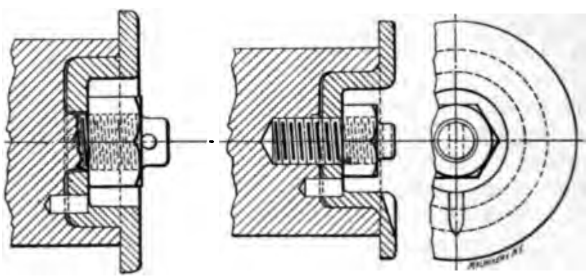

Fig. 21. Method of Attaching Crank-pin Collar

The pins are usually reduced in diameter for the main bearing, in order that the connections may be as light as possible. In the present case the projected area called for amounts to

$$D\,l' = \frac{P}{1600} = \frac{70,000}{1600} = 43.75 \text{ or a bearing about } 6.25 \times 7 \text{ inches. (26)}$$

Taking Formula (27) for the side-rod pins we have

$$P' = \frac{0.3\,W'\,D}{g} = \frac{0.3 \times 40,000 \times 57}{26} = \text{about } 26,000 \text{ pounds}$$

and

$$\frac{26{,}000}{1600} = 16.2 \text{ square inches of bearing area}$$

and

$$D = \sqrt[3]{\frac{10.2\,Pl}{S}} = \sqrt[3]{\frac{10.2 \times 40{,}000 \times 2}{16{,}000}} = \sqrt[3]{51} = 3.7 \text{ inches, or a}$$

4×4 inch pin. (25)

Here we have a variation in diameter in practice, demanded by the clearances at the front limiting the length of the pin which is accordingly made 5 inches diameter and 3¾ inches long.

As for the forms of the crank-pins used on such a locomotive as this consolidation engine, they are simple and are shown in Fig. 20.

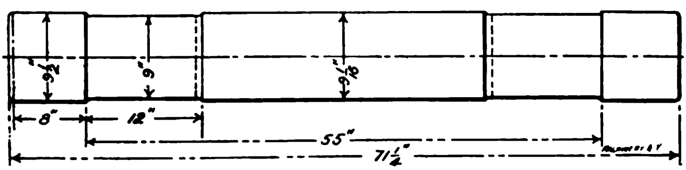

Fig. 22. Driving Axle for Consolidation Locomotive

In this the three forms shown are for the main, intermediate and front and back pins respectively. Fig. 21 shows the method of attaching collars at the ends so as to occupy a minimum of space. As all of the rods are practically fitted with solid bearings, it is necessary that the pins should have washers at the outer ends to hold them in place.

Fig. 22 illustrates the common form of driving axles used at present. There is an enlarged wheel seat at the end. The journals are inside this and are slightly smaller in diameter than the wheel seat, while the body of the axle at the center is slightly larger. Abrupt changes of diameter are avoided and square shoulders are not allowed. The same process in the determination of dimensions may be followed in the case of the passenger locomotive.

CHAPTER VI

DRIVING WHEELS AND COUNTERBALANCING*

The driving wheels, on which the engine is carried and by which it is propelled, are subjected to stresses far beyond those due to the mere requirements of carrying the load put upon them or transmitting the thrust, exerted by the steam, into rotation and pull upon the rails. Among the first in importance of these secondary stresses are those due to the shrinkage of the tires upon the center. In order that the wheels may withstand these stresses, the rim should be heavy between the spokes, and the latter stiff enough to carry the load without bending. In the shrinking on of the tire under ordinary conditions an allowance of 0.010 inch to the foot is made. That is to say, the tire is bored out 0.010 inch smaller than the diameter of the center upon which it is to be placed for each foot of diameter of the latter. This allowance is modified somewhat according to the proportions of carbon entering into the composition of both the tire and the center.

When the center is made of cast steel in accordance with prevailing practice, it should be well annealed so that the greatest possible degree of toughness may be obtained. In effecting this annealing the utmost care should be taken to do the work uniformly throughout the whole extent of the material, and the same degree of hardness obtained as far as it is possible. It is essential that this should be done in order that the application of the tire may be successfully made, for the allowance for shrinkage must be met by the compression of the center and the elongation of the tire in such proportions that the latter will be securely held in place without overstraining either.

When soft material is used either in the tire or the center, a greater allowance must be made for shrinkage than where one or both are hard. These terms are, however, indefinite as to their limits, and there is, as yet, no standard of reference by which the work has been scientifically laid down. The result is that while the figures given for the proper shrinkage are those recommended and extensively used, experience and good judgment must enter into the work, and no hard and fast rule can be laid down that will meet all of the exigencies ot varying conditions. In the design of the center, the spokes should be placed as close together as practicable and should be of such a section that they can withstand, without bending, the stress required to elongate the tire, in accordance with the amount of shrinkage allowed; this stress to be divided among all of the spokes.

If we place the shrinkage at 0.010 inch per foot of the diameter of the wheel, and assume that the center compresses as much as the tire elongates, there will be a total elongation of the tire of about 1/64 inch for each foot of circumferential length, and a corresponding com-

* MACHINERY, Railway Edition, May, 1906.

pression of the rim of the center. It is evident, then, that in order that such a compression may take place in the rim, the spokes must be able to be compressed a corresponding amount without changing form. This requirement does not determine the form of the section of the spoke but does influence its thickness. The width of the spokes laterally should be as great as other conditions will permit, as there is no means of determining the stress in this direction that may be brought to bear upon the wheel by a rough track, a derailment or the movement at high speeds around curves and over frogs and switches. Furthermore, the wheel should be dished as little as possible, because this, with the stresses induced by the shrinkage of the tire, tends to increase the transverse stress on the spokes.

If the strength alone were to be considered, the best form of spoke section would be that of an I-beam; but as this is impracticable, because cracking would occur in the cooling of the casting, the next best, or rectangular form, is used, rounded at the corners to an elliptic section. Again, an allowance must be made in the spoke lengths for rough usage and indeterminate stresses, at the same time using the utmost discretion in the work lest the weights run up to excessive amounts, as may very easily occur.

The hub should be of ample strength to resist the stresses to which it is subjected by the pressing in of the axle and crank-pin. It is essential, not only that there should be sufficient thickness to resist these unknown stresses but that this thickness should be as uniform as possible so as to avoid shrinkage cracks and unequal stresses due to the cooling of the metal after casting.

In boring for the axle and pin the work should be done with the utmost care and the dimensions so proportioned that the pressure required to force the wheel and axle together will be from 12 to 14 tons per inch of diameter of the axle. Where steel or wrought iron wheel centers are used, the fit should have a slight taper which will result in a better fit and less crushing of the surfaces of the material than when the fit is straight, when this may be so great as to exceed the elastic limit of the material. As a general rule the taper should be not more than 3/64 inch or less than 1/64 inch to the foot. Of the two, the larger taper is to be preferred where comparatively soft steel is used in the wheel centers; while, when the steel is hard, the smaller taper should be used. The taper for the crank-pin fit may be made 1/32 inch to the foot and the pressure regulated to the same amount per inch of diameter as in the case of the axle.

There is another detail in the designing of the wheels that is of great importance and which has received the closest examination for many years, and that is the counterbalancing. To do this all parts connected to each crank-pin should be carefully calculated, or better still, weighed, if it is possible to do it before the counterbalance weights are determined.

In doing this, the side rods, the rear end of the main rod, together with all pins and straps, are to be considered as revolving parts and the weight of each assigned to the wheel and pin with which they are connected. An approved method of weighing is to couple the side rods together and, supporting each bearing on a knife-edge with the rod horizontal, take the weights on these knife-edges in succession. The weight of the end of the main rod is obtained in the same way. In addition to these revolving weights those of the reciprocating parts, including the front end of the main rod, the crosshead, piston and piston-rod must be taken.

Of these the counterbalance in the wheel must be the equivalent of all of the revolving parts and a portion of the reciprocating. Regarding the latter the rules and practice have not yet taken such shape that a fixed proportion of the reciprocating parts to be counterbalanced is established. In this the designer is between two evils. The greater the proportion of reciprocating parts that are counterbalanced the smaller will be the longitudinal motion of the engine, but the greater will be the vertical disturbance. This puts an excessive pressure on the rail when the counterweight passes the lower center and tends to lift the wheel when it passes the upper; while, on the other hand, if too little of the reciprocating parts are counterbalanced, there will be an excessive longitudinal motion or nosing of the machine. As a matter of fact, an attempt is made to strike a happy medium between these extremes by which the minimum vertical effect on the rail is produced with the maximum longitudinal disturbance that can be tolerated.

To accomplish this, it is well to take the weight of the whole engine into consideration when calculating that of the counterbalance; for the relation between the two has a marked influence on the perceptible horizontal disturbance. Hence it comes about that the question has to be reversed and decided along the lines of what proportion of the reciprocating weights, that of the engine will permit to be left unbalanced.

The rule adopted by the American Railway Master Mechanics Association is to allow 1/400 part of the weight of the engine to remain unbalanced in the reciprocating parts. This, while entirely empirical, works well in practice and is quite generally used throughout the United States. It is, however, modified in some instances by limiting the counterweights to from 55 per cent as a minimum to 65 per cent as a maximum of the weights of the reciprocating parts for road engines, though the practice in this respect is not universal. This weight is equally divided among all of the driving wheels of the locomotive. In case the wheel centers are so small that there is not room to put the whole of the counterbalance that should be apportioned to the main driver on the side opposite the crank-pin, and get it within a reasonable area that does not exceed the area of half of the half-circle, the remaining wheels should not be balanced to compensate therefor, unless the extra weight to be balanced is less than 65 per cent of the weight of the reciprocating parts divided by the numbers of wheels.

This may be expressed by the formula

$$B = \frac{0.65\,W}{n} \qquad (33)$$

in which

B = the maximum allowance balance weight which is added above that needed to counterbalance the revolving parts,

W = the total weight of the reciprocating parts,

n = the number of coupled wheels on each side of the engine.

If this does not make up for the deficiency of balance of the main wheel, the total balance should be left that much short, and the maximum speed limit of the engine should be correspondingly reduced.

In designing the wheel, it is best to cast the counterweight in solid with the center. Should the weight be too great for this, pockets may be cast in symmetrically on the side opposite the crank to be filled with lead in the final adjustment, since by the use of lead, a greater weight can be obtained in a smaller compass than with iron. In other cases the major portion of the weight may be cast in, and the deficiency made up by the addition of lead in the final adjustment.

This is usually done after the axle and crank-pin have been pressed in. The journals of the axle are placed upon horizontal straightedges, and the wheel turned so that the crank-pin is on a horizontal line with the center of the axle. A weight is then hung upon the pin by a ring somewhat larger in diameter than the pin so as to secure a central bearing. This weight should be equal to the sum of the weights of the ends of the side rods that will be attached to the pin, and the proportion of the reciprocating weights that is to be apportioned to the pin. In the case of the main pin the weight of the rear end of the connecting-rod is to be added.

The reciprocating weight to be balanced on each wheel may be found by the formula:

$$r = \frac{R - \dfrac{W}{400}}{n} \tag{34}$$

in which

r = the reciprocating weight to be balanced at each wheel,

R = the total weight of the reciprocating parts,

W = the total weight of the engine in pounds,

n = the number of coupled wheels on each side.

Still it must be borne in mind that this weight is subject to the conditions imposed previously, that the total weight counterbalanced or $R - \dfrac{W}{400}$ should lie between 0.55 R and 0.65 R and r will be the same for each wheel.

As for the driving boxes, their dimensions, like so many other parts of the locomotive, are not determined by any special rule except that in a general way they are proportioned according to the diameter of the journals, compounded with those indispensable requisites for good designing in all branches of mechanics—experience and good judgment. The prime requisites are that they shall have sufficient strength to sustain the load that they have to carry and sustain the pressures to which they may be subjected in the working of the engine. Nearly all of these forces partake of the nature of compression stresses, which, of themselves, would not seriously strain the material, but they are coupled with shocks and blows both lateral and longitudinal, especially when the wedges become worn or loose in service.

Here again the importance of reducing all weights to a minimum manifests itself and so cast steel has come to be the usual material used for driving boxes. The crown brass is usually made with a thickness at the top equal to one-quarter the diameter of the axle and is forced into place by a hydraulic pressure of from 20 to 25 tons. The length of the bearing is governed by the diameter of the journal and

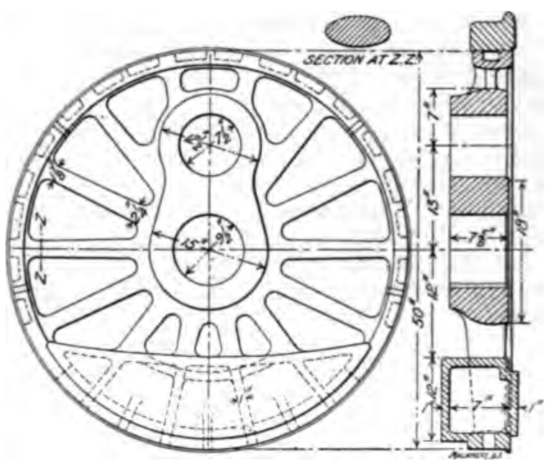

Fig. 23. Cast Steel Driving Wheel Center for Consolidation Locomotive

the weight that it has to carry and is so proportioned that on fast passenger engines the load carried does not exceed 170 pounds per square inch of projected area, and 180 pounds for freight engines, while 190 pounds per square inch can be allowed on switching engines.

In order to meet these conditions it frequently becomes advisable to increase the diameter of the axle by a fraction of an inch, above the calculated requirements, so that the bearing may be shortened and the center of the box be brought centrally beneath the springs. There should be sufficient side play between the flanges of the boxes and the wedges to permit one box to rise until it strikes the frame while its mate at the other end of the axle remains down in its normal position. The reason for this is mainly that the flanges may not be broken in case a spring breaks on one side.

The oil cellar is usually a simple cast-iron box fixed to the lower shanks of the driving box. It is held in position by horizontal pins that may be easily removed so that the cellar may be lowered for repacking or replacing the pads, which should always be high enough so to press against the axle journal and thus supply lubrication for a should the feed from the top of the box be cut off.

Turning now to the practical application of these principles, Fig. 23 illustrates a cast steel driving wheel center for the consolidation locomotive in which pockets are cast in the counterbalance for the use of lead filling, and Fig. 24 shows a wheel of the same material, but of larger diameter intended for the Atlantic engine with the counterbalance cast solid with the rim.

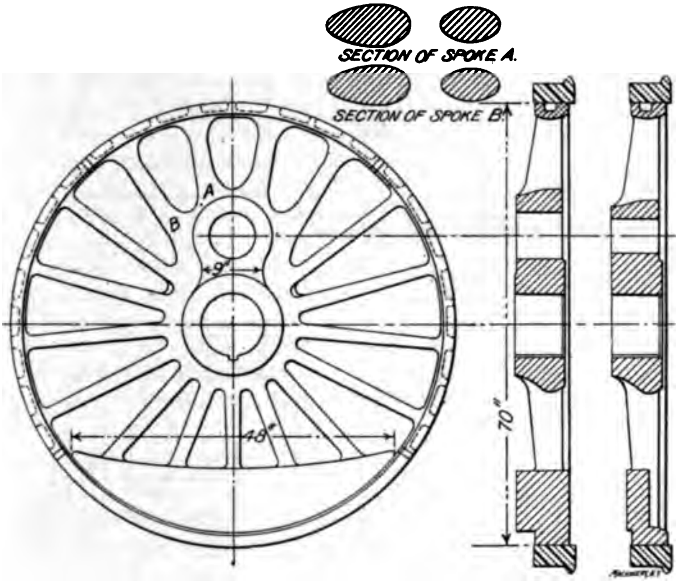

Fig. 24. Cast Steel Driving Wheel Center for Atlantic Type Locomotive

In the former case the weight to be counterbalanced on the main driver is, according to the rule of the Master Mechanics Association:

$$966 - \frac{176{,}000}{400} = 526 \text{ pounds,}$$

and for the Atlantic engine

$$1057 - \frac{168{,}000}{400} = 637 \text{ pounds,}$$

in both of which cases the weights are from 55 to 65 per cent of the weights of the reciprocating parts which are 966 pounds and 1057 pounds respectively.

Again, if it should have so happened that there had not been room on the main wheel for this amount of counterbalance, the distribution of the weights among the other wheels would have been, according to Formula (34), for the consolidation engine

$$r = \frac{966 - \dfrac{176{,}000}{400}}{4} = 131.5 \text{ pounds,}$$

and for the Atlantic engine

$$r = \frac{1057 - \dfrac{168,000}{400}}{2} = 318.5 \text{ pounds.}$$

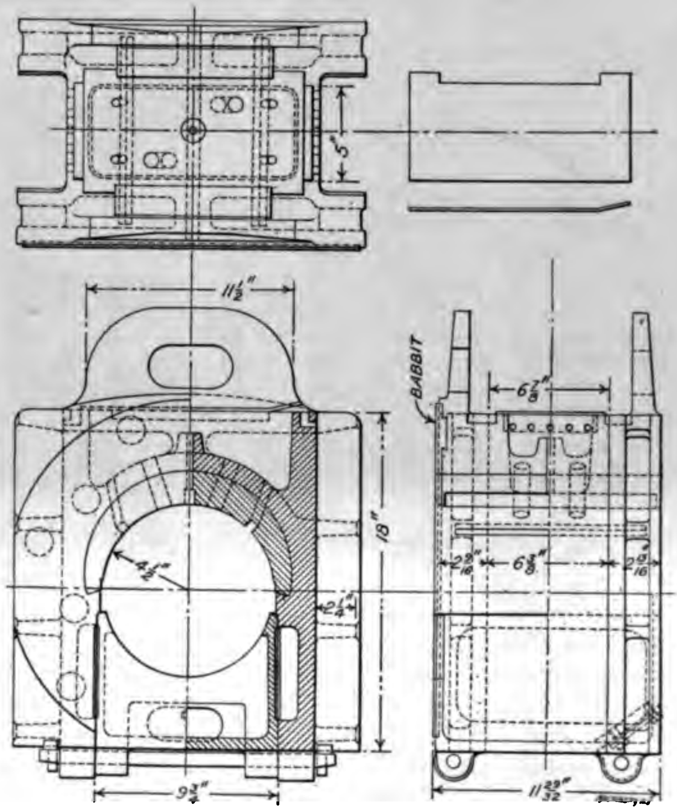

Fig. 25. Driving Box for Consolidation Locomotive

To these must be added the weights of the revolving parts, which, aside from the crank hubs, are on the consolidation engine for

the forward pin	100 pounds
the main pin	545 pounds
the third pin	215 pounds
the rear pin	100 pounds

in the case of the Atlantic engine for

the forward pin	140 pounds
the main pin	490 pounds

All of these weights must be reduced in proportion to their distance from the center of the axle as compared with that of the crank-pin.

For example, if the center of gravity of the counterbalance of the main driver of the Atlantic engine is two and one-half times the distance of the crank-pin from the center of the axle, the actual weight to be used becomes

$$\frac{490 + 637}{2.5} = 450.8 \text{ pounds.}$$

Fig. 25 shows the form of box that is used for the consolidation locomotive and the same general features obtain in that for the Atlantic. Particular attention is called to the form of the flanges, where they bear against the wedges. They are tapered from the center to the ends, the distance between them widening, so that they can have a rocking motion without binding, and thus fulfill the conditions imposed that the box shall be able to rise and strike the frames and not bind while its mate is in the normal position. It will be seen that the flanges are quite heavy and that the depth is sufficient to give a good bearing on the wedge faces. The length of the bearing is made 11¾ inches long, and as the diameter of the axles is 9 inches, this makes 105¾ square inches of projected area of bearing. As the weight on the drivers is 19,375 pounds, from which must be deducted the weight of the wheel and one-half the axle, which will amount to at least 3000 pounds, the load on the journal falls within the limit of 180 pounds per square inch, or will be about 155 pounds per square inch, a margin that leans to the side of safety and cool running.

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CHAPTER I

SPRING RIGGING AND EQUALIZERS*†

The proper distribution of the weights on the wheels should be considered as soon as the type of engine and the service which it is to perform have been decided upon. The weight that can be placed upon each wheel is limited by the rail, the bridges on the line and the general features of the track construction. In this distribution of the weight the method of spring suspension and equalization plays an important role, and it must be so designed that a safe load, or one that will hold the truck wheels down on the rails under all conditions of speed and track curvature, is put on them, while that on the driving wheels must be sufficient to give them the proper adhesion for the development of the required power.

It may be stated that in a general way the number and size of the driving wheels are chosen in connection with the allowable weights on the same according to the service performed, while the size of the cylinders is determined by the weights and dimensions chosen. The cylinder dimensions and boiler pressures are so proportioned that the power exerted at the rail, which is expressed by the following formula, shall fall between 22 and 23 per cent of the total load on the driving wheels:

$$T = \frac{d^2 \times 0.85\, p\, S}{D} \tag{1}$$

in which

T = the tractive power,
d = diameter of cylinders,
p = the boiler pressure,
S = stroke of piston in inches,
D = diameter of driving wheels in inches.

When these figures have been decided upon, a skeleton drawing of the engine is prepared by which the weight and its distribution is outlined, and here the locomotive designer's judgment and experience are called into play. The layout is afterward carefully checked by a calculation of the weights of the various parts and their moments with reference to the center of the truck when a four-wheeled bogie is used; otherwise they are taken from the center of the cylinder.

* The present number of MACHINERY'S Reference Series is the fourth part of a treatise on complete Locomotive Design, covered by Nos. 27, 28, 29, and 30 of the Series, and originally published in RAILWAY MACHINERY (the railway edition of MACHINERY). Each of the four parts of the complete work treats separately on one, or more, special features of locomotive design; and while the four parts make one homogeneous treatise on the whole subject, each part is complete by itself. In order to give concrete form to the examples and theoretical considerations, it is assumed that a consolidation freight locomotive and an Atlantic type passenger engine are being designed. It is further assumed that these locomotives are designed for a division 150 miles long, laid with rails weighing 75 pounds per yard, and with a ruling grade of one per cent ten miles in length.

† MACHINERY, Railway Edition, August, 1906.

When the desired weight of engine has been obtained, the wheels should be so arranged that they form two groups or systems of supports. The one at the front should be cross equalized so that the center of support falls in the center line of the engine. This is very readily accomplished where a four-wheeled truck is used at the front and takes care of the front system. All of the remaining wheels are then included in the system for the rear. When these can be equalized on each side so as to form three points of suspension, ideal conditions have been obtained. This can be readily accomplished in the case of eight- or ten-wheeled engines.

When the rear system is entirely composed of driving wheels, as in the case of the engines just mentioned, the momentum of the load falls along the line of the center of gravity of the axle support. But, when trailing wheels, carrying a lighter burden, as in the case of the Atlantic engine, are used, it falls along a resultant line of these several centers. It is then necessary to so adjust the wheel base that the weight is carried on a line coinciding with the resultant axle centers of gravity.

In the case of mogul or consolidation locomotives where a two-wheeled truck is used at the front, one or two pairs of the forward drivers are equalized with it. The common supporting point of the system is brought to the center of the engine by a transverse equalizer, usually placed on a line with the front end of the forward driver springs, and this coincides in its longitudinal center, as it crosses the engine, with the center of gravity of that portion of the machine and load that it is proposed to carry on the forward system of suspension. The center of the load of the rear system falls, as in the previous case, in the center of its wheel base if the wheel spreads are equal; otherwise in the center of gravity of its respective axle supports.

With these points known, as well as the distance between them, together with the weight of the engine above the axles, the design should be modified by the shifting of the boiler and other parts, so as to bring the load on the proper centers in a way to secure the desired distribution on the axles. The weights of the several parts should be calculated with all possible accuracy, especially where this data regarding the parts of an earlier engine are not available for comparison; and for this work no short-cut rule can be given.

The weight on the axles having been determined, the next point is to ascertain the size of the springs to carry the load, which is equal to that on the axles, less the weight of the driving boxes, spring saddles, and the springs themselves. This is found by means of the following formula:

$$P = \frac{S\,b\,h^2\,n}{6\,l} \qquad (2)$$

in which

P = load on one end of the spring,
S = allowable fiber stress in the steel, usually put at 80,000 pounds,
b = width of spring leaves,
h = thickness of spring leaves,

n = number of spring leaves,

l = length of spring from edge of spring band to point of spring hanger bearing, as indicated by Fig. 1.

In this P is, of course, equal to one-half the total load on the spring.

For ease of riding, the leaves should be made as broad as possible, and experience shows that a length of from 36 inches to 42 inches from one point of suspension to the other is that best adapted to locomotive work, though this is, of course, to be governed by circumstances and the conditions surrounding the design, which may involve the use of either a longer or shorter spring. As for the thickness of the leaves, that varies from 3/8 inch to 7/16 inch.

The spring when first made must have a certain amount of free height or set, so that, when the load has been applied, it will deflect to a point best adapted to the carrying of the load. This deflection may be found by the formula:

$$F = \frac{6\,P\,l^3}{E\,b\,h^3\,n} \qquad (3)$$

in which

F = the deflection,

E = modulus of elasticity of the steel, which may be put at 31,500,000.

The other symbols have the same significance as in the case of Formula (2).

For the helical springs that are frequently used at the extremities of the spring suspension, the total carrying capacity is calculated by the formula:

$$P = \frac{S\,\pi\,d^3}{8\,D} \qquad (4)$$

in which

d = diameter of steel of which the spring is made,

D = diameter of coil to center of steel.

The other symbols are the same as in Formula (2).

The diameter of the steel to be used thus becomes:

$$d = \sqrt[3]{\frac{8\,P\,D}{\pi\,S}}$$

The deflection of the helical springs will be

$$F = \frac{8\,P\,D^3}{\pi\,G\,d^4} \qquad (5)$$

in which G is the modulus of elasticity for torsion, and may be put at 12,000,000, or

$$F = \frac{D\,S\,l}{d\,G} \qquad (6)$$

Finally the length of the wire between end coils will be found by the formula:

$$L = \frac{F\,G\,\pi\,d^4}{8\,P\,D^3}, \text{ or } \frac{F\,d\,G}{D\,S} \qquad (7)$$

in which the significance of the symbols remains as before.

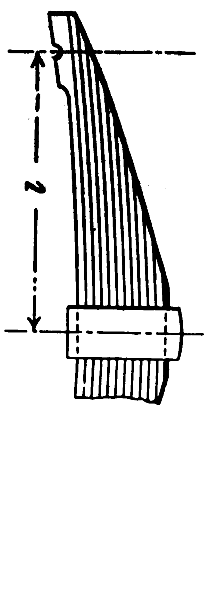


Fig. 1. Diagram for Use with Formula (2)

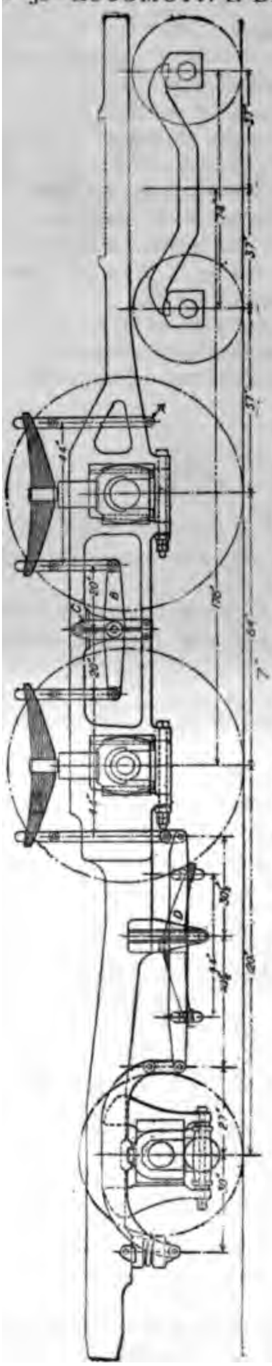


Fig. 2. Spring Rigging for Atlantic Type Passenger Locomotive

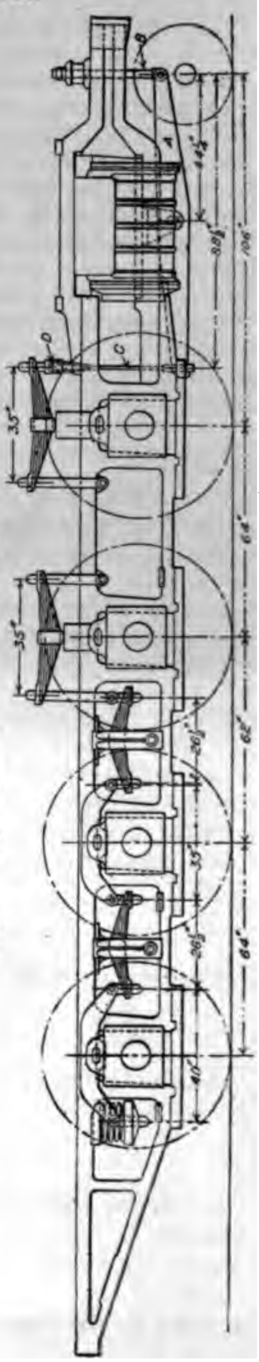


Fig. 3. Spring Rigging for Consolidation Type Freight Locomotive

Figs. 2 and 3 illustrate the manner in which the principles that have thus been laid down have been followed in the case of the two engines, the designing of which we are here following. On the Atlantic locomotive (Fig. 2) there are two systems of equalization: One at the front cared for by the four-wheeled truck, and which is not shown in detail, as the load is carried on the center plate and needs no further explanation. In the rear system, there are four points of support for the frame. Starting at the front, the forward spring hanger carries the frame at the point *A* and transmits the load sustained at that point to the semi-elliptic spring set on the axle box. A hanger dropping down from the rear of this spring takes hold of the equalizer *B*, which is pivoted at its central point on the fulcrum *C* and serves to sustain its portion of the load.

In the same way the stresses are transmitted to and through the spring over the rear driver and down to the equalizer *D*, which forms the connection between the rear driver and the trailing truck wheel. At this point the overhang of the firebox makes it impossible to place the semi-elliptic springs above the frame, so the equalizer is made of two bars, and a spring shorter than the distance between the supporting hangers is used. This is also done in order that the spring may not be of excessive weight and length. The rear axle box is fitted with a yoke, at the back end of which the helical spring for supporting the rear frame is placed. In this suspension every point of support is a pivot, and there are no rigid connections. Hence there can be no variation from the proportioned distribution of the weights.

For example, it is evident that the load carried by the forward spring hanger at *A* must be equal to that on the rear hanger of the same spring, otherwise the spring would be tipped down toward the hanger with the greatest load upon it. This would extend back through the whole system of suspension. If such a tilting should occur toward the front, for example, that action would then tilt the intermediate equalizer *B*, which would put an additional stress upon the forward hanger of the rear driver spring. Such an additional load would be transmitted on to the helical spring at the back, which would at once resist further compression by adding to the weight it was normally lifting and thus check any tendency of the whole system to tilt by sending forward to the point *A* a greater lifting power, and thus, by increasing the stresses on the forward hanger, equalize the system and distribute the proper proportion of load among all of the points of support.

In order to perform the supposititious work of this engine, as stated in the foot-note on page 3, one weighing 167,000 pounds, with 23,000 pounds on each driving wheel, would have to be offered. This leaves 75,000 pounds to be distributed between the trailing wheel and the front truck. In proportioning the rear equalizer, the two arms have been so designed that a load of 18,500 pounds is put upon each of the trailing wheels, leaving 38,000 pounds to be carried by the front truck. These weights include those of the wheels, so that the actual load on the springs will be less by the amount of the weights of the wheels, axles, boxes and springs. As the total of these weights will amount to about 13,000 pounds for the driving wheels, with their axles and boxes, and 9000 pounds for the trailing wheels, with the same connections, the springs with the arrangement shown must provide for sustaining a total load of 107,000 pounds. As part of the load is carried twice, so to speak, due to the equalizing arrangement, the various springs will be loaded as follows:

Driver box springs, each	19,750 pounds
Intermediate driver and trailing truck equalizer spring	17,275 pounds
Trailing truck helical spring	6,600 pounds

In these weights the total load on both ends of the spring is given, so that in the calculation of the same, the figures must be divided by two as already stated.

The suspension for the consolidation locomotive is shown in Fig. 3. Starting with the forward driver, equalization is accomplished by means of the equalizer *A* with a bearing on the bottom of the saddle casting on the center line of the engine. At the front this equalizer is suspended and pulls down upon a bolt *B* whose upper end rests upon a spring cap that is carried by the truck. At the rear it pulls down on the hanger *C*, that is attached to the center of the cross-equalizer *D*, whose outer ends are carried by the hangers coming down from the front ends of the springs. The balance of the equalization of this system is clearly shown by the engraving.

As for the rear system, which includes the three pairs of drivers, it resembles that of the Atlantic engine, except that the springs are of a sufficient length to reach from one driver box yoke to the other, thus taking the place of an intermediate equalizer. In this engine the weight on the drivers may be given as 19,500 pounds each. With the proportions of the forward equalizer shown in Fig. 3, the weight on the two truck wheels would be about 21,000 pounds, so that the total weight of the engine would be 176,000 pounds. By making the same reduction as before for the wheels, axles, and boxes, the weight to be carried by each of the springs becomes 16,300 pounds and they may be calculated accordingly.

The spring strengths are not the only calculations that have to be made in the work of the suspension. This also involves that of the equalizers and hangers. The former can be calculated from the formula for beams supported at the ends and loaded in the middle as follows:

$$b\,d^2 = \frac{18\,P\,L}{S} \qquad (8)$$

in which

b = the thickness of the bar,

d = the depth of the bar,

L = distance between hanger connections in feet,

S = safe fiber stress = 8000 pounds for steel castings and 14,000 pounds for wrought iron.

P = load to be carried.

Fig. 4 gives the general proportions for these parts for the Atlantic engine. It is necesasry, however, in the designing of these parts to make them of ample strength to withstand the shocks to which they will be subjected, hence the low fiber stress that has been specified in the schedule for Formula (8).

The spring hangers must also be designed of ample strength; these are not only subjected to a tensile stress that may be applied with more or less suddenness but one which is also constantly varying when the engine is in motion. In this, too, the fiber stress should not be

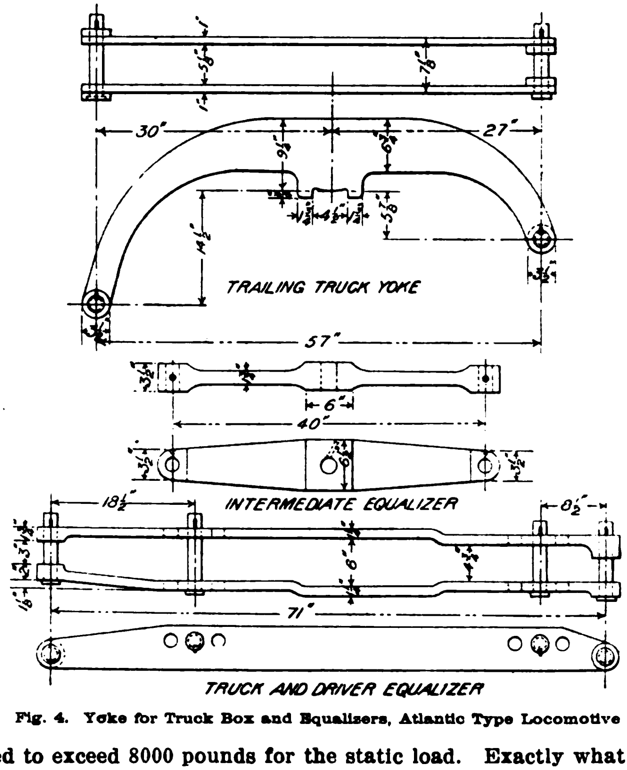


Fig. 4. Yoke for Truck Box and Equalizers, Atlantic Type Locomotive

allowed to exceed 8000 pounds for the static load. Exactly what it will amount to when the engine is in motion is not known, but it is apt to be as much as 50 per cent in excess of the calculated load. Fig. 5 shows a number of types of spring hangers that are used on consolidation and mogul locomotives, including not only those that are to be found upon the engine under consideration, but some others as well.

In the whole of the spring suspension, as in other parts of American cars and locomotives, nothing is fastened down, but dependence is placed upon the weight to hold the parts in position. The hangers, therefore, rest upon the ends of the springs through the intervention of the hanger gibs, some of the forms of which are shown in Fig. 6. For these no calculation is to be made. They are subjected to a shearing stress only

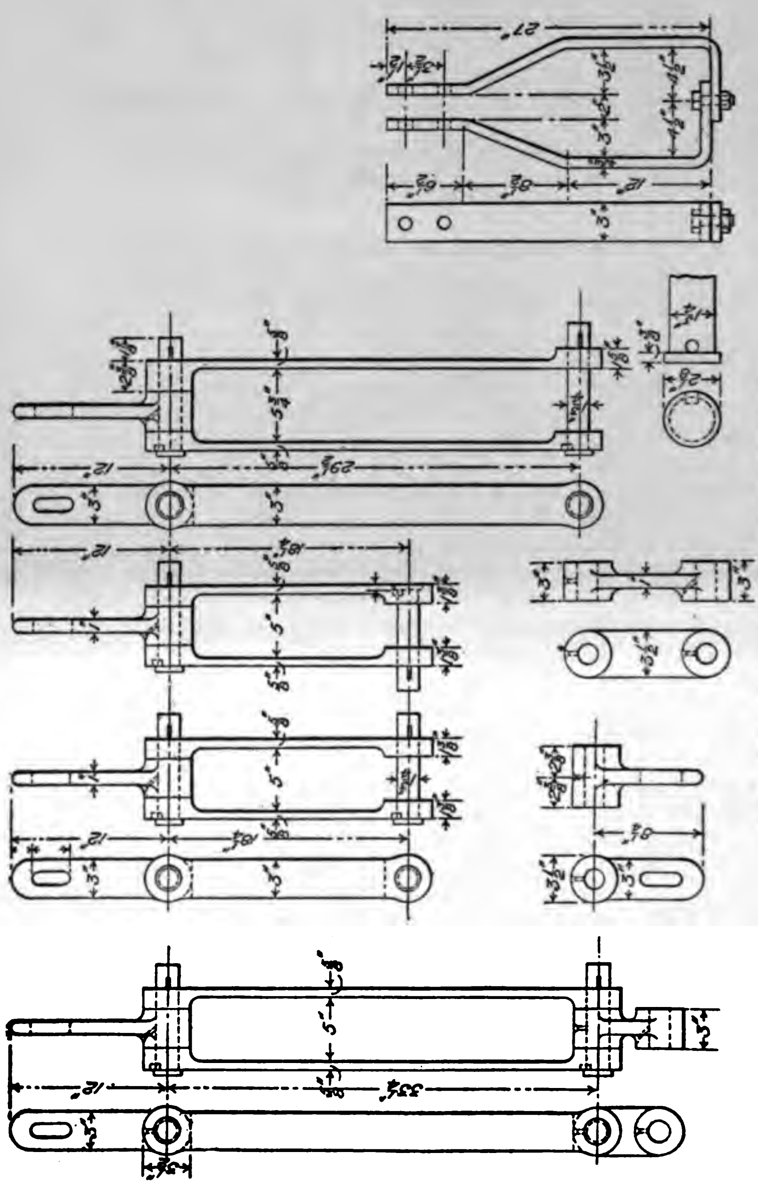


Fig. 5. Spring Hangers for Consolidation Locomotive

and must be made heavy enough to withstand this and allow for a large amount of wear as well. As the latter is the larger item of the two, the actual stress that it put upon the gibs when the engine is new and everything up to its original dimensions, is very small.

It will thus be seen that for a proper adjustment of the spring suspension, great care must be taken so that the center of gravity of the portion that is to be carried by the front or back system of suspension should fall in the proper place. In the case of the Atlantic engine, we have the two driving wheels located 84 inches apart and each carrying 23,000 pounds. The trailing wheel is 120 inches behind them and car-

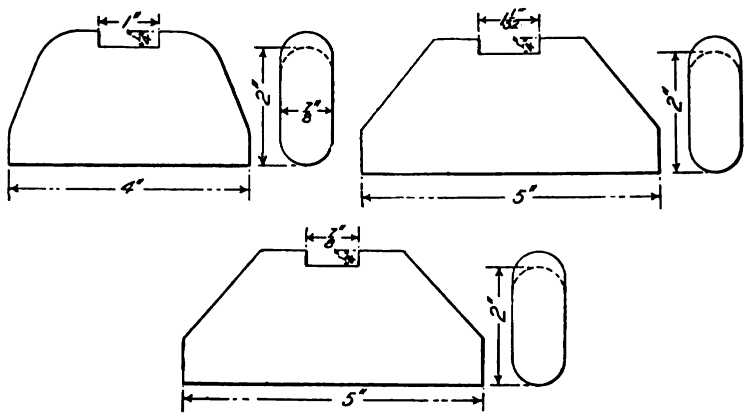


Fig. 6. Spring Hanger Gibs

ries 18,500 pounds. The center of gravity of these three weights falls 0.4 inch back of the rear driver, and that is the point at which the center of gravity of the weight to be carried should be arranged to fall. The same care should be taken in the case of the consolidation locomotive that the center of gravity of the weight carried by the rear system should fall in a line with the next to the rear pair of drivers when these are spaced equal distances apart.

When these precautions are taken, the engine will not only be easy riding, but there will be far less trouble with the parts of the spring suspension in the way of breakages than where these precautions are ignored.

CHAPTER II

THE TRUCKS*

The truck is an essentially American characteristic of the locomotive. For many years European locomotives, especially those used in freight service, were built without any truck, and the guiding of the engine was done by the flanges of the front pair of wheels. In this country, however, the truck has always been used upon road engines and has been considered an essential detail in their safe and satisfactory operation.

Broadly speaking, the engine truck proper, or the one located at the front end, may be divided into two classes, the two- and four-wheeled types. The four-wheeled truck is the one that has been universally used on locomotives intended for passenger service, while the two-wheeled has been applied to freight engines, or those that are used, for the most part, in freight service, with an occasional assignment to passenger work. The exclusion of the two-wheeled truck from engines designed for passenger service has been due to the necessity of using a large boiler and so increasing the total weight of the engine beyond the requirements of adhesion represented by the tractive power that it was desired to develop. Under these circumstances the extra weight, beyond that needed for adhesion, could be carried on four wheels to better advantage than on two. As for safety, there is no difference of opinion that the two-wheeled truck is quite up to all the demands of the most exacting service.

In the designing of the trucks there is little else to be done than to secure ample strength to carry the load imposed and arrange for axles and boxes of sufficient bearing surface to do the work required without heating. At the same time care must be taken that sufficient weight is put upon the truck to hold it down and cause it to keep the rails upon the sharpest curves to be encountered, and thus prevent the flanges of the wheels from climbing the rail and causing a derailment when called upon to guide the direction of motion of the machine from a straight line to a curve and through the latter. In the case of the consolidation engine under consideration, as well as upon those of the mogul class, the two-wheeled pony or Bissel truck is used.

The plan of framing this type of truck is shown in Fig. 7, and the details of the working parts in Fig. 8. From these drawings, as well as from a comparison with Fig. 3, it will be seen that the truck is pivoted at the center between the wheels. As this would not hold them in place on the rails, but would allow them to swing into a position approximately longitudinal to the track, if any obstruction were to check the motion of one, a radius bar, *A*, is added that reaches to a point beneath the engine, where it is pivoted on the center point. This

* MACHINERY, Railway Edition, November, 1906.

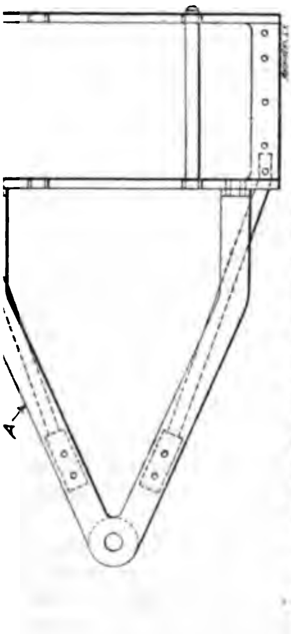


Fig. 7. Outline Plan of Pony Truck Frame for Consolidation Locomotive

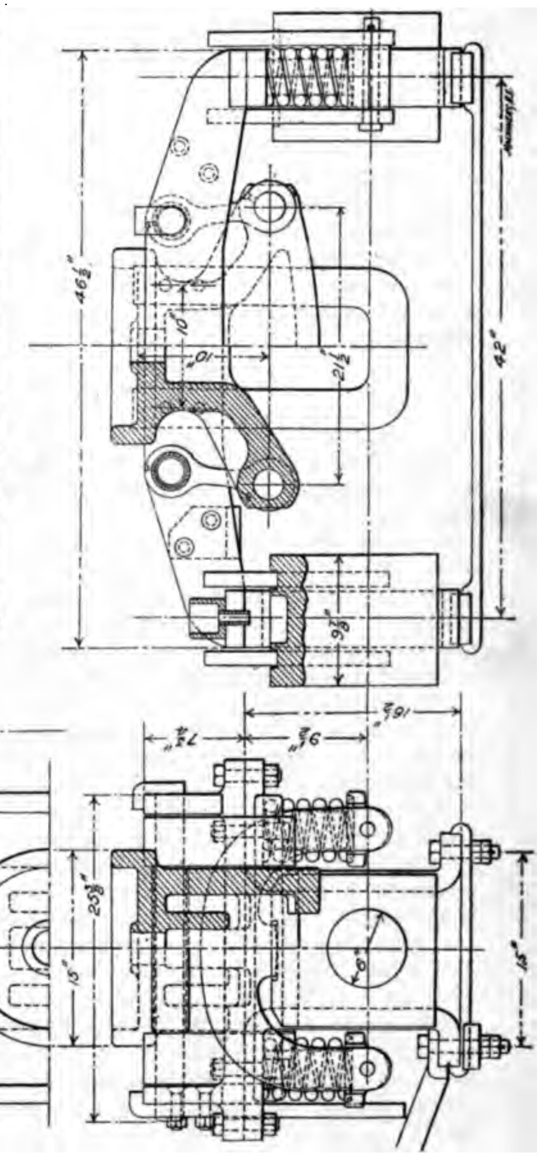
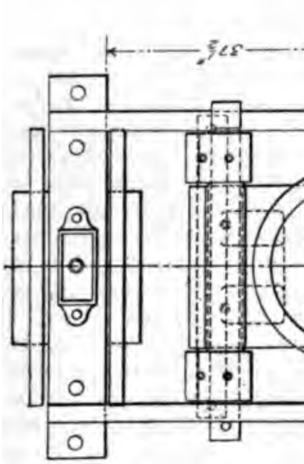



Fig. 8. Pony Truck Frame or Consolidation Locomotive

point is carried back far enough to insure stability of the wheels upon the rails, and at the same time, permit of sufficient side motion to allow the truck to swing out of line with the center of the engine when entering and passing over a curve.

The formula used for calculating the length of the radius bar is as follows:

$$R = \frac{A \times B}{A + B} \times 0.85 \qquad (9)$$

in which

R = the length of the radius bar,
A = the total wheel-base for consolidation and mogul locomotives,
B = the distance from the front driving wheel to the truck wheel.

In the case of the consolidation locomotive under consideration, $A = 314$ inches and $B = 124$ inches. Formula (9) therefore becomes:

$$R = \frac{38{,}936}{438} \times 0.85 = 75.56 \text{ inches.}$$

This, then, may be taken as the distance from the pivotal point of the radius bar to the center of the truck axle, which in this case is made 6 feet 3½ inches.

As the radius bar has no load to sustain and the only stress to which it is subjected is that of holding the wheels on the track, it is usually made of a flat bar of steel about 5 inches by 1¼ inch laid flat and stiffened by round braces rising diagonally from the foot of the pedestals and bolted to the horizontal portion of the bar itself at a convenient distance back of the truck frame.

The design of frame, as shown in Fig. 8, may be taken as typical of that in use upon mogul and consolidation locomotives in the United States, and is of an exceedingly simple construction. The weight of the front end of the engine is carried on the center plate, and this in turn is suspended by the center plate hangers from the transoms that reach across from side to side. These hangers are spread a small amount at the bottom so as to increase the tendency of the truck to return to the central position when coming back to a straight line from a curve. As the center plate and the pivot pin of the radius bar are normally located in the center line of the truck, it would be merely an extension of the rigid wheel base if no lateral flexibility were given to the wheels. It is this lateral flexibility, and the pull on the center plate by the hangers that tends to guide the front of the engine out of a straight line and around a curve. In the case of the truck under consideration, the frame is carried by two helical springs at each side, and these, in turn, rest on seats attached to yokes that set on top of the axle boxes.

We have already noted that the weight on the truck of this engine is to be about 21,000 pounds, or 10,500 pounds on each wheel. In order to carry this load an axle 6 inches in diameter is provided with a journal 9⅞ inches long. This gives a load of about 177 pounds per

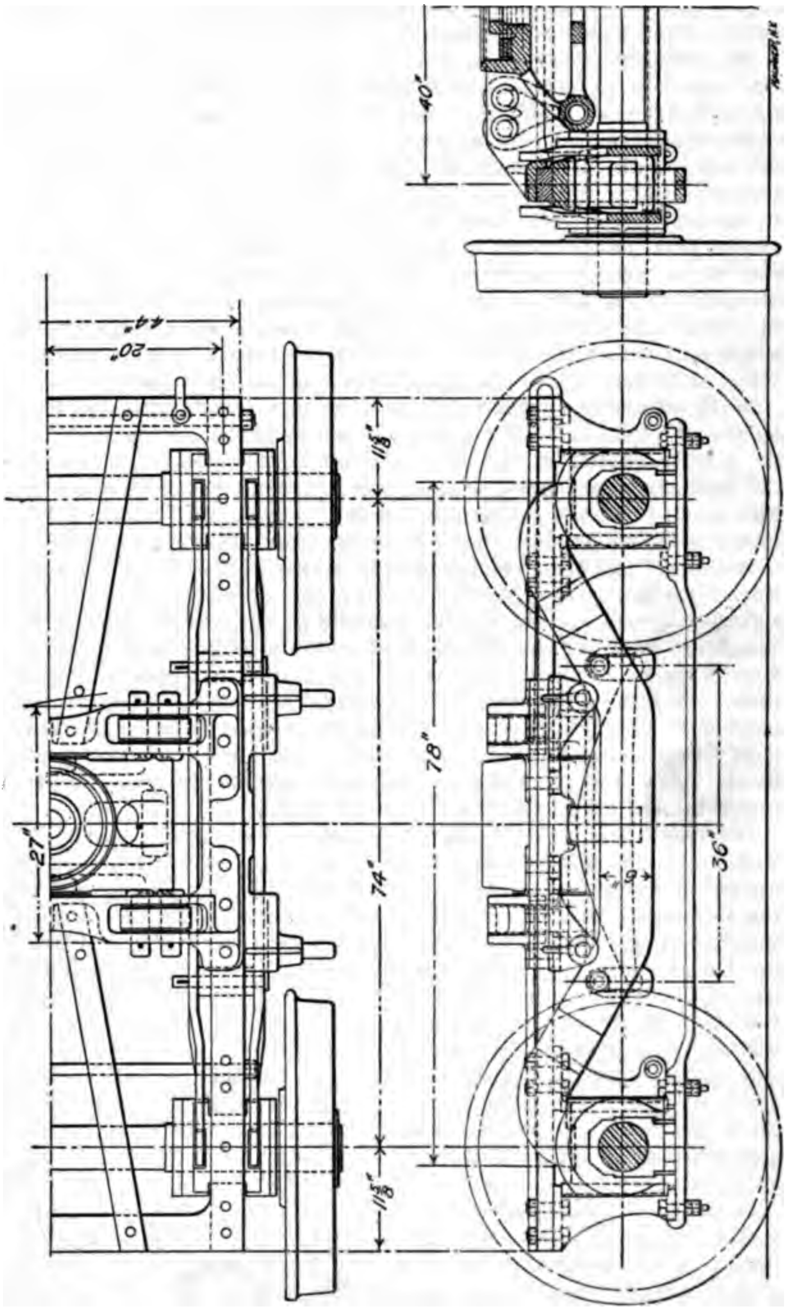


Fig. 9. Four-wheeled Forward Truck for Atlantic Locomotive

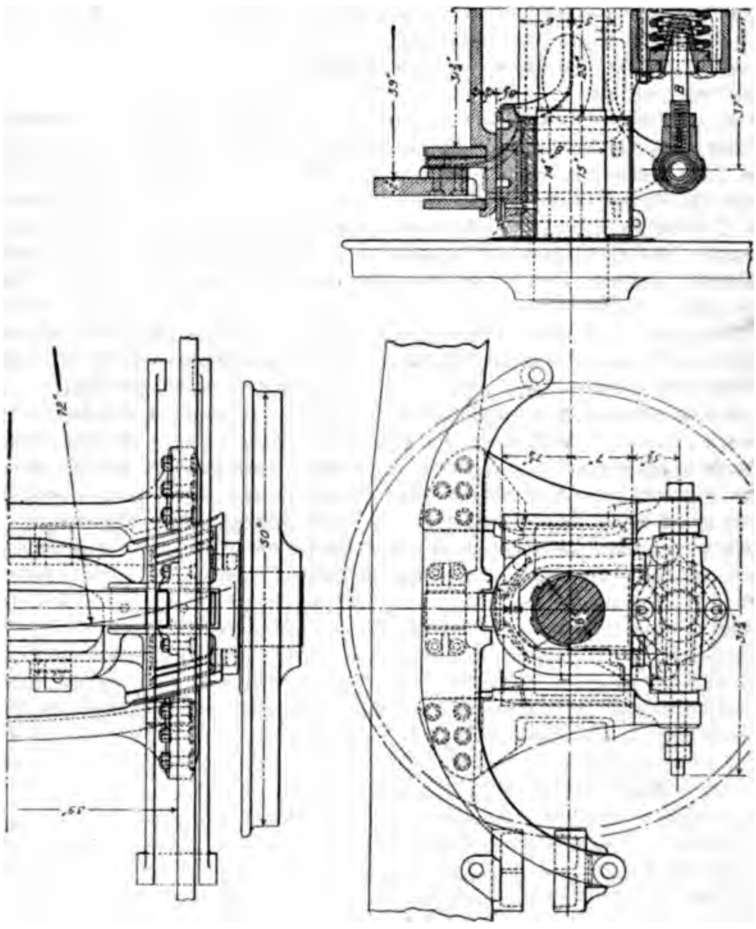


Fig. 10. Radial Trailing Truck for Atlantic Locomotive

The axle is, of course, straight; but the boxes, instead of having their wearing surfaces parallel to its center line, are set in pedestal jaws that are cut at an angle and on a radius of 6 feet. Hence, the transverse movement of the wheels is the same as though they were swinging through the arc of a circle whose center is at the center of curvature of the faces of the pedestals. Owing to this, the wheels are held approximately in a true radial position. The swing of the overhang of the engine beyond the rigid wheel base throws these wheels out of center, and the return of the frame to its central position on a straight track would restore the normal condition of things, but assistance is rendered in overcoming the resistance of the boxes to this movement by a centering spring placed beneath the axle. The pedestals are, of course, rigid transversely with the main frame of the engine and carry the push bars *B*, which bear against followers that are set in castings that move laterally with the boxes. Any movement of the latter, in either direction, compresses the intermediate spring, which thus has a constant tendency to restore the truck to its central position.

It will thus be seen that but very little of a definite guiding nature can be said regarding the designing of the trucks that are to be used under American locomotives. The general design of the pony and the four-wheeled truck has been established by such long usage that it is no longer a subject for discussion. Details are being constantly varied to meet the changing demands of weight, proportions and service, but with no essential change in the main features of the designs.

With the rear truck of the Atlantic, Pacific and Prairie types of locomotives, the case is somewhat different. These locomotives have hardly been in service long enough to have settled down to an established basis of construction in detail and it will probably be a number of years before this will be done. Meanwhile the general principle of the freedom of lateral movement for the wheels, with the approximation to the maintenance of a radial position for the axle has been accepted and acted upon, but beyond this the truck may be said to be in a process of development, though the engines to which it is applicable may be considered as among the standards of American railroad practice.

CHAPTER III

CAB, CAB FITTINGS, AND ACCESSORIES*

As the cab contains all of the various means that are required for the operation of the engine, the greatest care is exercised to so place them that they are all of easy access to the operator. No absolute rule can be laid down for the arrangement of these several fittings, except to say that, in a general way, the planning is to be done for the convenience of operation and the comfort of the men in the cab. It therefore becomes largely a matter of good judgment on the part of the designer or of the choice of the purchaser, and is further dependent, to a great extent, upon the space available and other conditions appertaining to the form and construction of the appliances that are to be used. These include the injectors, lubricators, engineer's brake valve, gages, etc., which are usually bought from manufacturers who make a specialty of such supplies.

Formerly the cab was invariably built of wood, strongly braced, and so constructed that it could be lifted from its place and set aside for painting and repairs while the engine itself was in the repair shop. The advantages claimed for the wooden cab were that it was warmer in winter, cooler in summer, cheaper in first cost, and more easily repaired in case of accidental damage. Whether the claims could be substantiated or not, the fact is that now the steel cab is used generally upon new work. The cab is usually made of plates about ⅛ inch thick, and is strongly braced with angles and tees. In the case of the two engines that have been under consideration, there is but little difference in the size of their cabs. Some of these differences are due to the character of the engine and some to the preferences of the user. In the case of the consolidation locomotive, the cab has a body length of 7 feet, with a rear roof projection of 3 feet 6 inches. These are approximately the standard dimensions of the modern cab, and allow it to set over the boiler for a sufficient distance to house the cab attachments and levers, and still leave ample room at the back for the footplate accommodations for the fireman. This projection of the boiler back into the cab is usually from 48 to 50 inches at the top. In the case of the consolidation locomotive, it is 49 inches. Where the back head of the boiler is sloping, as it is in the Atlantic locomotive, it is sometimes necessary to place the bottom of the firebox farther back in the cab, in order to have room on the top for the necessary attachments. This was the case in this instance, and the cab in consequence is 6 inches longer than that of the consolidation.

The corners, bottom edge, carlines and plates are made of angles riveted to the sheets, and it is common, though not universal, practice

* MACHINERY, Railway Edition, December, 1907.

to countersink the heads on the outside so as to obtain a smooth surface. At the front there is usually a door upon each side giving access to the running board, though sometimes this is replaced, on the right-hand side, by a drop window. The sides are invariably provided with windows running the whole length of the cab and arranged to suit the user. The sashes are made to slide past each other so that the actual point of opening can be varied. The side opening in the consolidation locomotive cab that is here illustrated is divided into two parts. The front space, which is 23½ inches wide in the clear, is filled by a fixed sash, while two sliding sashes are placed in the space at the rear. These move past each other and make it possible to have the double space clear. In the Atlantic engine, the more common arrangement of one fixed and one sliding sash is used. The fixed sash is placed at the front and the sliding sash moves across inside of it. This gives one wide opening for the engineer to lean out of.

Cab Roof

The roof is arched from eave to eave and is carried by carlines made of angles. It is of sheet metal, as in the case of the consolidation, or of wood as in the Atlantic engine. Wood is preferable for the roof on account of its freedom from the annoyance of sweating or moisture in cold weather. The roof is exposed on the inside to the steam that occasionally rises from the boiler and its attachments, and when becoming cold, this steam is apt to be condensed on the lower surface of the roof and falls back in large drops, to the annoyance of the crew.

With the boiler projecting back into the cab for a distance of 4 feet or more, it occupies so much of the cubic contents of the latter that it would heat the air to an uncomfortable extent were the cab roof not provided with a ventilator. This is, therefore, always done at present. It consists of a trap that opens upward on a hinge at its front end, or of a sliding hatch. The trap is to be preferred because its position is such that, when the engine is in motion, it not only deflects such cinders as may strike the roof from the stack, but it has a tendency to assist in drawing the hot air out of the cab. The cab is supported by heavy brackets attached to the frames, and is entirely self-contained.

Throttle and Reverse Lever

Within the cab there are but two attachments that belong to the locomotive itself and are so necessary for its operation that they are included in the basic design for the engine. These are the reverse and throttle levers. The reverse lever is usually pivoted in a bracket bolted to the frame, and works in or beside a quadrant notched to receive its latch. As the point of cut-off is varied by the movement of the reverse lever to and fro, it is desirable that the notches in the quadrant shall be as close together as possible so as to permit of small variations in the point of cut-off. For this reason they are frequently cut to resemble a straight-tooth gear and the lever latch is made with a number of teeth to mesh with them.

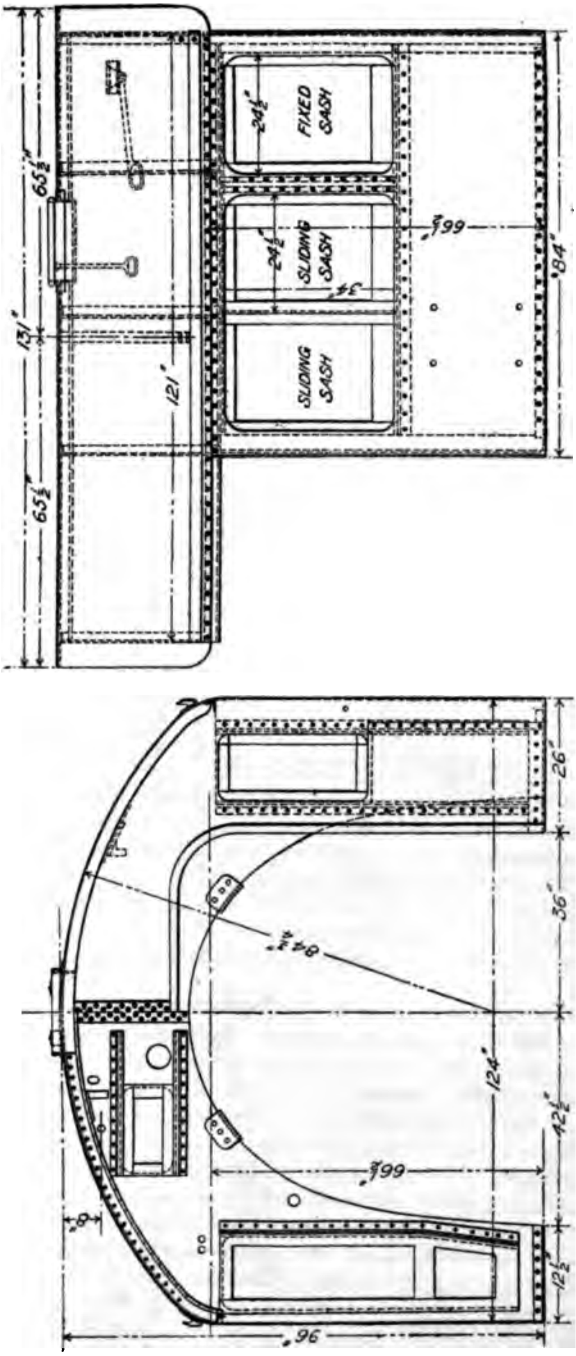


Fig. 11. Steel Cab for Consolidation Locomotive

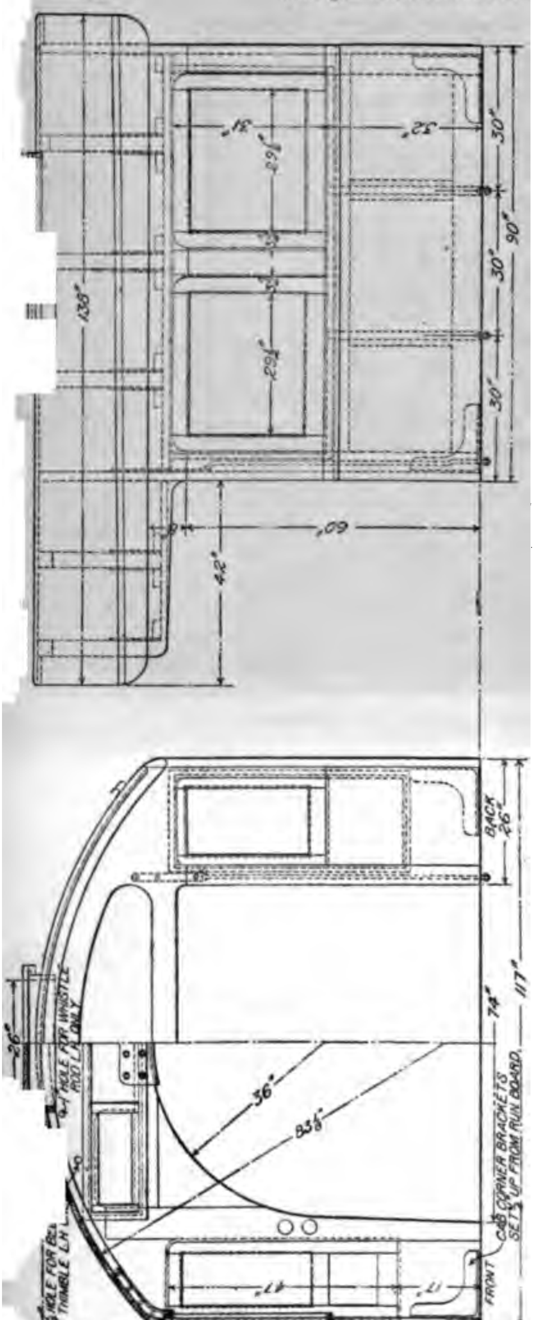
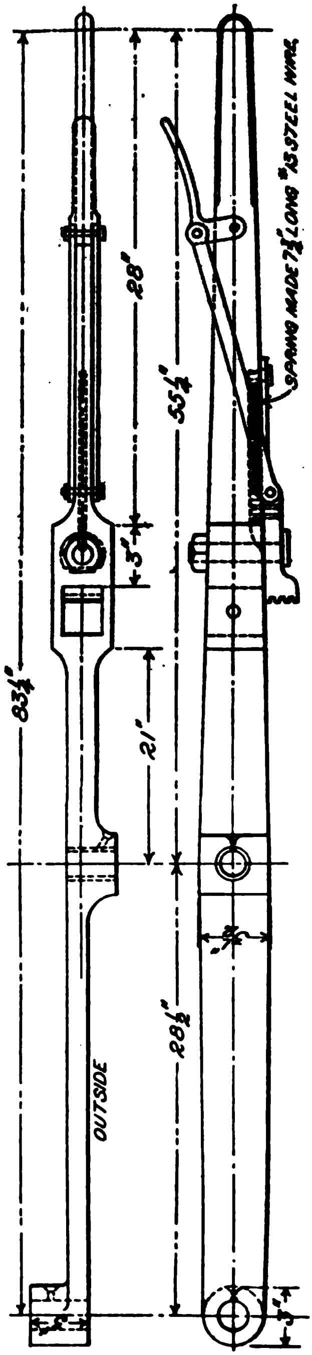


Fig. 12. Steel Cab for Atlantic Type Locomotive



It is essential in designing the reverse lever that it should be made of a strength sufficient to resist the jars and jerks to which it will be subjected, and also have a length sufficient to enable the engineer to move it, and thus shift the point of cut-off when the engine is in motion. The eccentric straps of a locomotive have been likened to a pair of prony brakes having a constant tendency to throw the links down into the position of full forward gear. So, as there is nothing to resist this tendency when the latch is lifted from the quadrant, but the pull of the man on the lever, the leverage should be sufficient to enable him to do it without danger of having it jerked from his grasp, or of being pulled forward himself. A leverage of about three to one, as in the lever illustrated, will usually be found to be sufficient for this purpose.

Especial attention must also be paid to the latch. It must be cut square and fit the quadrant with little or no sloping sides, to prevent it

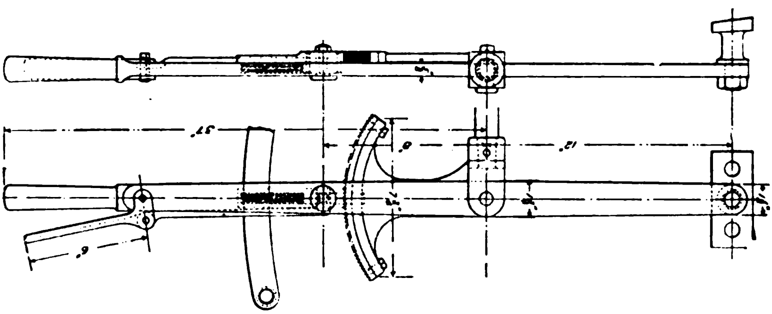


Fig. 14. Throttle Lever for Consolidation Locomotive

jarring out. The spring that holds it down in place must be stiff, and there must be such ample leverage on the latch lifter that it will be easily operated. When the engine is working, there is a constant and severe jarring of the lever as the pull of the links tends to throw it forward, and unless it is rigid in itself, and the latch is held down firmly by the spring, it is likely to work itself free, and in its movement forward may inflict severe injury upon the engineman or the machinery.

The throttle lever should be carefully designed so that it will hold the valve in any desired position. Strength is required in order that it may be able to hold the valve firmly down upon its seat, and not because it is otherwise subjected to any severe stresses. When the valve is open, the load upon the lever sinks to insignificance; and, as the valve is usually balanced, the leverage required for manipulation is not great. It should, however, be provided with a latch that admits of fine adjustment and will hold it secure when the valve is shut. The quadrant had best be made of wrought iron, case-hardened and cut with teeth of fine pitch. Good results can be obtained if there are about six to the inch. It will be noted that, in the design illustrated, which is the one in common use, the quadrant moves with the throttle stem, so that the effect is to make the possible adjustment even less than that corresponding to the pitch of the notches. The length of leverage indicated not only gives the engineer a very delicate and sensitive control of the valve, but is, in part, necessitated by the fact that the throttle stem is in line with a vertical plane passing through the center of the boiler, and it is necessary to bring the handle out on one side to be within easy reach.

Running-boards

The running-boards are closely allied to the cab, and frequently extend back into it, forming the foot-boards. They are of simple design; as shown in the cut, Fig. 16, they are made of steel plate about 3/16 inch thick, and are stiffened at the edges, usually by a 2 × 2-inch

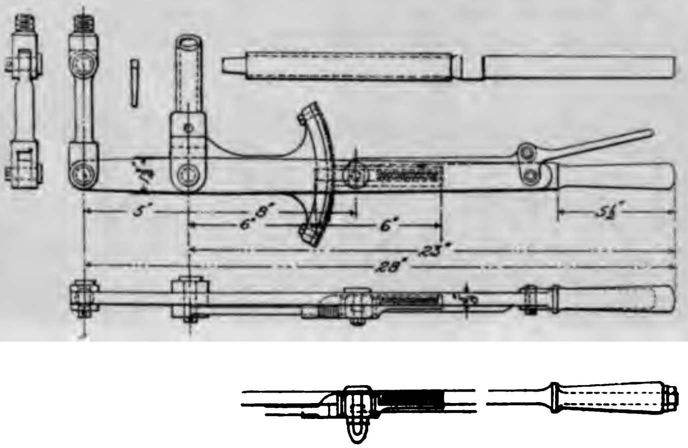


Fig. 15. Throttle Lever for Atlantic Type Locomotive

angle. They must be cut away where necessary to allow for the placing of pipes, air-pump, and other fixtures. As the load on them rarely exceeds the weight of a few men, they are readily carried by simple brackets of flat iron riveted to the boiler. They should reach from the cab to the steam chest or the front end.

Cab Fittings and Accessories

While the designer of a locomotive must be familiar with, and provide for, the various cab fittings and accessories that are to be applied to the machine, he has nothing to do with their designing, and frequently is not even consulted regarding the selection of fittings from those that are upon the market. The manufacturing of these parts is in the hands of specialists, and the purchaser usually selects what he pleases and orders them put upon the engine. The part played by the designer in the matter is that of so locating them that they may be readily manipulated and their indications easily seen by the engineer and fireman.

These parts consist of the injector, engineer's valve, water gages, steam gage, whistle, lubricator, injector cock, cylinder cocks, blow-off

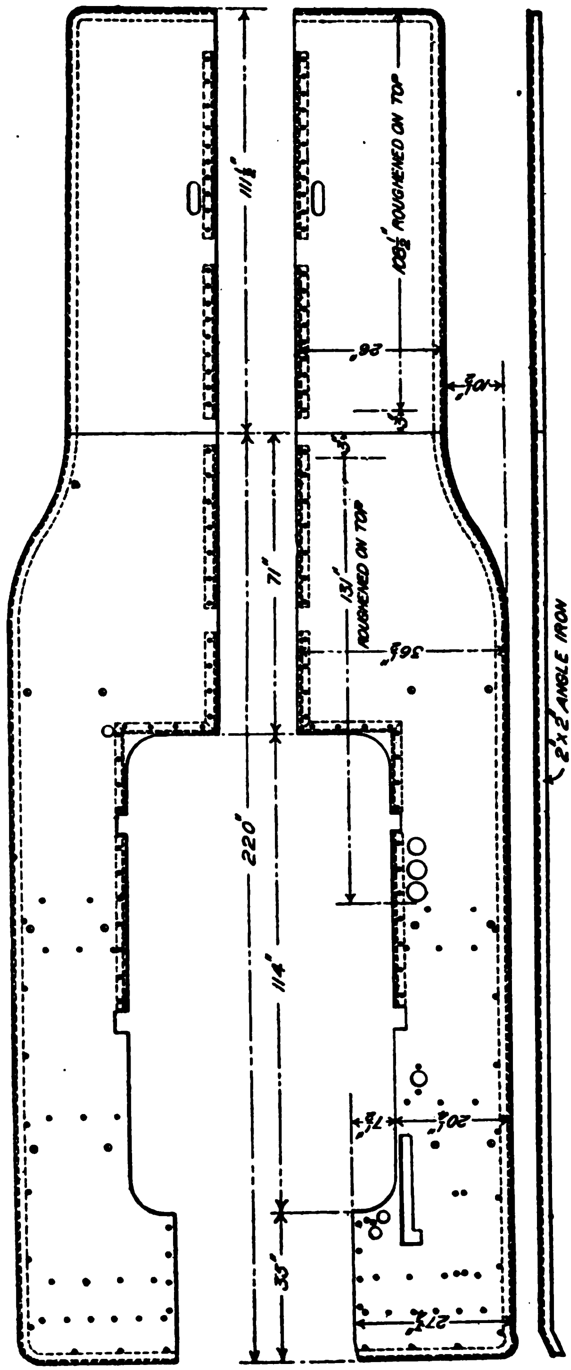


Fig. 16. Design of Running-boards

, headl :, air-pump, and sander. It is essential that some of ... shoul. .. so placed as to afford the maximum of convenience to : engineer in the running of the engine. The reverse and throttle ..ers are the essential features of the engine itself, and near them t be located the handle for the working of the injector and the neer's valve. One is needed in order to control the supply of .r to the boiler, and the other for the manipulation of the air-kes, for very few locomotives are now built that are not equipped with such brakes. With these handles easy of access, the engineer can control his train with safety, and the time expended in making the various movements is reduced to a minimum. Next to the injector and the engineer's valve come the whistle and water gages. The whistle lever is usually so located that it can be worked by reaching up and grasping either the lever itself or a cord attached to it that is near the roof; and the gage cocks are close at hand on the left, often near the engineer's valve. Owing to the introduction of the water glass, many engineers have come to depend upon it entirely for water level indications, with the result that it often happens that the gage cocks are placed out of reach. This should never be done, for they should always be close at hand.

The steam and air gages are usually placed upon the top of the boiler and face directly to the rear. The air gage indicates the amount of air carried in the reservoir and train pipe, and should often be consulted by the engineer, especially after an application of the brakes in order to note whether the pressures have been unduly reduced. So with the steam gage, he watches it when the pressure is low rather than when it is high, but for the fireman its visibility is of the first importance, and it should be so placed that it can be read from all points on the foot-plate and the tender, as well as from the seat upon the left-hand side. As these attachments are in constant use, and as it is upon them that the safe running of the train depends, the greatest pains should be exercised that they be so placed as to afford the utmost of convenience to the men. Of secondary importance are the handles for operating the cylinder cocks and sander. Usually it is necessary for the engineman to reach forward to the front of the cab or down to the floor to grasp them. Of the two, the sand lever should be the more easy of access.

Within the cab there are a number of other attachments. The lubricator for oiling the valves and cylinders is usually placed upon the top of the boiler. As it works continuously after being started, it is never allowed to crowd more important parts out of desirable places. Under present conditions the lubricator should have at least three points of feeding, one for each steam chest and cylinder, and one for the air-pump. Sometimes a lubricator is used with a multiple feed with pipes leading to the several axle journals. The same requirement holds true for the location of the valves controlling the steam heat and operating the blower. They are only manipulated occasionally and so need not be in prominent positions; but, at the same time, care should be taken to put them where they will be readily accessible, and not tuck them away where there is a likelihood of burning the hand when an attempt is made to reach them.

For fixtures that are applied outside the cab, the designer should see that they are properly located. The injector check, for example, should be placed somewhat back from the front tube-sheet, though there is by no means an agreement as to exactly where this should be. Practice varies in this respect from 10 inches to 5 feet back of the sheet and the probability is that 3 feet would be a good safe average location. This refers to the usual practice of placing the check on the side of the boiler and about on a line with its center. When this is done there should be an ample space of not less than 6 inches left between the opening and the first tube opposite it. When the injector check is placed in the back head, there should be a pipe leading from the inside opening forward, beneath the water line to a point within 3 feet or 4 feet of the front tube-sheet.

The headlight must be of such a size and so located that it will not interfere with the draft of the stack. The top of the stack is usually the highest point on the engine, so that ordinarily there is no trouble. Occasionally, however, the top of the ventilator of the headlight has been brought up flush with, or raised slightly above, the top of the stack. In these cases, when steam is shut off and the engine is drifting, an eddy is formed over the top of the headlight that turns down into the stack and drives the smoke and gases out into the cab.

All oil cups that are not forged solid with the parts that they are intended to serve should be of the strongest and most substantial construction; else they will become loosened and lost or broken by the jarring of the engine or the movement of the parts.

As already stated, the designer has little to say regarding the details of the cab fixtures that are to be used. A certain standard has been adopted by the customer or user, and this is specified. If the designing is done by the engineering department of the road that is to use the locomotive, it naturally follows that the regular standards will be applied. This applies not only to those items mentioned above but to other details as well, such as lagging for boiler and cylinders, the material for castings, the finish of the running-board, the type of air-brake and pump, the size of the air reservoir, the arrangement of the air-pump exhaust, the whistle signal, gong, turret or combination fitting on the back end of the boiler, washout plugs, feed hose connections, lazy cocks, safety valves, vacuum valves, gage lamps, signal flags and lanterns, oil cans, jacks and engine tools, the color and method of painting, and frequently the material used. In all this the designer is merely a passive instrument who must place each individual piece in its proper position, see that provision is made for fastening it securely, and that it is readily accessible for manipulation and repairs. This last item is of the utmost importance and it cannot be emphasized too strongly. While a locomotive is intended for service on the road and nothing should be sacrificed to make it as efficient as possible, it must always be borne in mind that the working parts will wear out and will eventually need to be repaired. For that reason all parts should, as far as possible, be so constructed and placed that they can be readily removed and replaced. Neglect of this precaution will result in that minor and general repairs will be far more expensive than they would have been had the parts been more accessible. This accessibility is also of the first importance in the matter of inspection. Where inspection is easy, it is much more apt to be thorough than where it is difficult, and neglect along these lines finds its direct reflection in the cost for repairs. Finally, strength and security should invariably occupy the first place in the mind of the designer, and cheapnes of first cost should not blind him to the requisites of strength and durability. And as he takes particular pains that the frames, boiler and cylinders are rigidly fastened in position, so he should see to it that all of the accessories should be secure, using flanges for all larger boiler connections instead of screw connections, for it will be upon the detailed attention that has been paid to these seemingly minor matters that the successful operation of the locomotive may depend.

CHAPTER IV

THE TENDER*

The locomotive having been designed for the service for which it is intended, it remains to provide it with a suitable tender for carrying the supply of fuel and water. In American practice this tender, until recently, almost invariably consisted of a U-shaped tank carried on a metal framework, though wood is still occasionally used on the smaller engines for this purpose. The tank, in turn, is mounted upon two four-wheeled bogie trucks. The fuel is carried between the legs of the U of the tank.

Tank Capacity

The first point to be settled in the design is the capacity of the tank. This will depend not only upon the size of the engine and the work which it is to do, as governing its steam consumption, but also upon the location of the water tanks and the distances that will have to be run between stops. Where the tender is to be fitted with a water scoop for taking water from track tanks, and these are located at frequent intervals, obviously the tender can be of less capacity than where long runs must be made. Further, a variation in tank capacity will be called for between a mountain and level division with the same distance between water stops, because of the greater steam consumption upon adverse grades when compared with the level. As a matter of fact, however, no actual distinction is made in this respect between engines on the same system. Provision is made for a sufficient water supply for the heaviest work that is to be done on any part of the line; and, then, if the engine is transferred to lighter work, there is simply an excess. The one thing needful is that the tank should hold enough water to do a little more than supply the engine, when working at full capacity over the severest section of the line.

Tank Construction

There are two general types of tank construction that are used in the United States, known as the plain U-tank and the water bottom type. The latter type varies somewhat and generally has no water legs, but in their stead a slope from the back end of the coal space, extending down toward the front and reaching across the tank, and then turning horizontally to form a water space beneath the floor. It is used where the foot-board of the engine is high, so as to raise the tank floor up to the same level for the convenience of the fireman.

Figs. 17 and 18 illustrate the tank designed for use on the consolidation and Atlantic locomotives, respectively. The tank for the consolidation locomotive has a capacity of 6000 gallons and is without a water bottom. It consists of a U-shaped tank, with a sharp incline at the

* MACHINERY, Railway Edition, January, 1908.

contained water due to the varying motions of the tender. Whenever the brakes are applied, the water rushes to the front and banks up in the legs, and then flows back to the rear. In addition to this, there is a constant surge to and fro laterally as the frame rolls on the trucks. When it is taken into consideration that the water in a half-empty tank weighs from 24,000 to 32,000 pounds, it will readily be understood that the effect of the wave action of this weight, as it moves about, may be very severe. There is a two-fold tendency in this action. One is to shift the tank upon the frame, and the other is to cause the sides to bulge. The surging effect is lessened by the interposition of splash plates in the legs and body, by which the space through which the water must flow is greatly contracted, and the effects of a high momentum destroyed. For the protection of the side plates against bulging, they are stayed with internal transverse, vertical and horizontal bracings of plates and angles riveted to the sheets so as to effect the most uniform distribution of the stresses possible, and so prevent any deformation of the tank.

The splash or dash plates are riveted to the vertical angles as shown at *B* in Fig. 17. In spite of these precautions the splashing effect of the water longitudinally is still very severe, so that the grill of internal stiffening angles should be such that the openings should not be more than from 18 inches to 24 inches across. The tank, as a whole, must be so firmly fastened to the frame that there is no possible danger of displacement because of the surging of the water or the shocks to which it will be subjected in service. This fastening is usually effected by means of heavy angle lugs having one leg riveted to the vertical plates and the other bolted to the framing. Although it is the side or vertical portion of the tank that is called upon to withstand the heaviest shocks due to the movement of the water, the top and bottom sheets, by presenting the larger surfaces to coal and the weather, are more exposed to wear and corrosion, and are, therefore, generally made from ¼ inch to 5/16 inch thick. It is especially desirable that the face of the coal space should be as smooth as possible, so as to afford no lodging place for coal or dirt, as such accumulations are apt to promote corrosion.

At the front end of the coal space, vertical angles are riveted to the legs to hold the coal boards, and wherever holes are cut for pipe or other connections the sheet should be stiffened by a plate of ample thickness and size, usually in the form of a flange riveted over the place. It is impossible to give any formula for calculating the stresses to which a tender tank may be subjected, because they are so varied in character and so irregular in application that it is not known what they are. For that reason it would be well to follow the general features that have been set forth in the designing of the tank, both as to method of construction and the dimensions of the material, although with the consideration that a reasonable increase may be made in the case of tanks of enlarged capacity.

In addition to the arrangements that have to be made to enable the tank to carry its load sucessfully, there are other attachments, to the location of which careful attention should be paid. This refers to such items as a shield that is placed across the top of the tank, back of the coal space, to protect the tank opening from flying pieces of coal and debris; it is made of wood or steel plates from 20 inches to 36 inches in height. The tank opening may be either round or oblong, and if oblong, the long diameter usually extends across the tank, as shown in the cuts Figs. 17 and 18. Sometimes, however, it extends lengthwise. The effect is the same in either case—the elongated opening avoiding the necessity for a close spotting at water stops.

There are handholes, or grab irons, to be placed near the steps at the front end, or in accordance with the legal requirements governing safety appliances. The lantern brackets at the rear are usually mere hooks located to suit the requirements of the purchaser. The same thing holds true regarding the tool-boxes. Usually there is a box on top of the forward end of each leg of the tank, provided with locks and keys, and in which the lighter engine supplies are kept, such as oil, waste, and hand tools; while, at the rear, either on top of the tank or behind it on the frame, there is another, extending across the engine, in which heavy supplies, such as chain, jacks, extra coupler knuckles and the like, are stored. These boxes are heavy and are made of wood, and must be securely bolted in position.

The other form of tank, that with a water bottom extending forward beneath the floor space, is shown in the engraving of the tank of the Atlantic locomotive. In this case it is given a capacity of 8000 gallons, and is correspondingly longer and deeper than the one of a smaller capacity used with the consolidation engine. In other respects it resembles in construction the one already described. There are, of course, some variations in detail, as, for example, the ladder at the back, which is required because of the great depth (68 inches), making the top so high that a man cannot readily raise himself up to it, as is possible in the case of the smaller tank. It will be noticed, too, that cupboards with doors are placed in the forward end of each leg. These are points that the designer must be prepared to think of and provide, if they are called for in the specifications.

In these large tanks the apron for holding the coal in place is usually made vertical and flush with the side, instead of being flared, as is the practice on the older and smaller tenders, and is only carried back as far as the manhole shield, the space about the hole being protected by a low railing. The top edge of the apron is always stiffened, usually by a bar of half-round iron riveted on the upper edge. These are minor, but somewhat important, details to which a designer should pay close attention, that the effect may be satisfactory not only to the eye, but in the service which is to be rendered.

Tender Frame Construction

The frame upon which the tank rests was formerly made of wood, but present construction is usually of steel. This frame is usually a ort rectangular structure built of rolled sections firmly bolted and ted together, and so braced diagonally that it cannot be twisted or

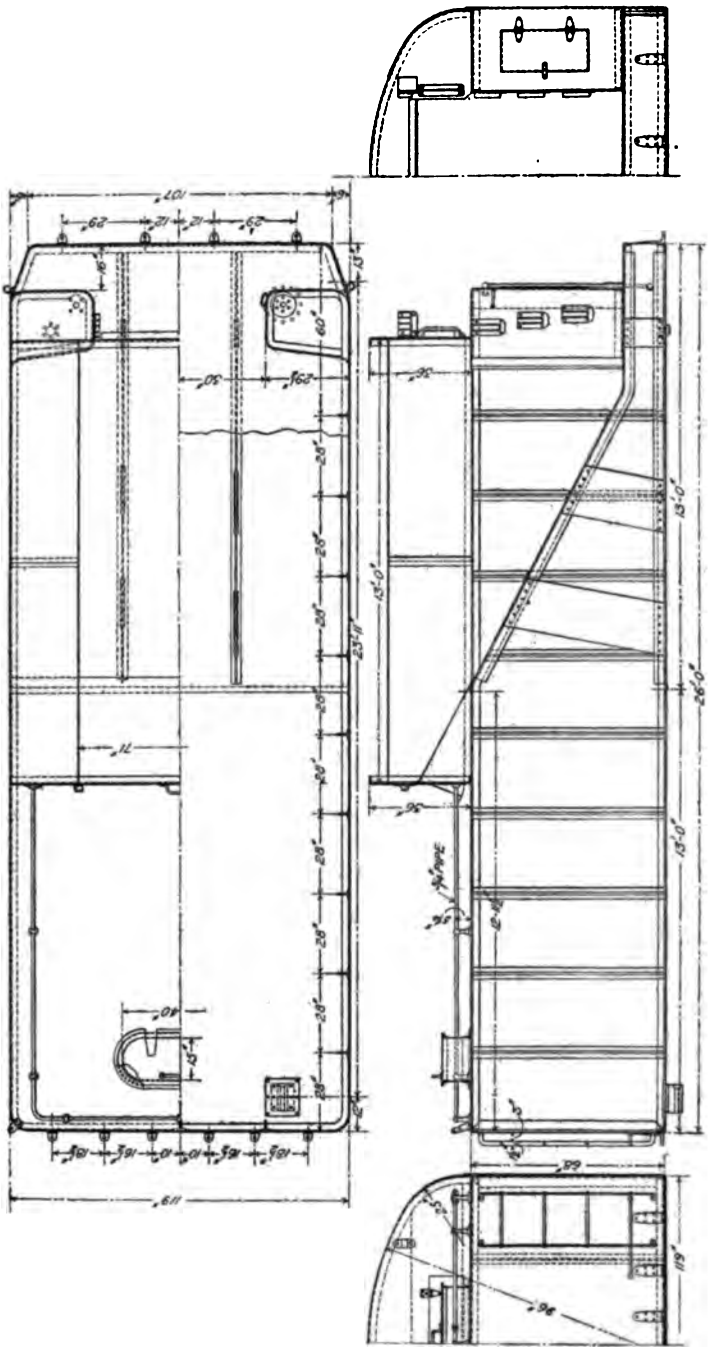


Fig. 18. Tank of 8,000 Gallons Capacity for Atlantic Type Locomotive

distorted by any of the ordinary stresses to which it may be subjected. The load imposed is always uniformly distributed over the whole length. In this connection, the under frame should be of ample strength to carry the whole load, as the water and coal rest fully on the lower frame and are supported to only a slight extent by the vertical walls of the tank. However, the distance between points of support of the frame is short, running from 10 feet to 12 or 13 feet, which is the distance between truck centers.

At the point of support the bolsters should be of ample strength and of great stiffness. Any flexibility or limberness will be sure to manifest itself in the imposition of an excessive weight on the side bearings and a racking of the structure as a whole. Therefore, the body bolsters should be so designed that twice the maximum load can be sustained on the center-plate, with an ample margin to spare in the way of a factor of safety. It is not known exactly what the actual stresses imposed upon tender bolsters may be, other than that they are, at times, at least 50 per cent in excess of the static loads.

The frame illustrated in Fig. 19 is that intended for use under the short 6000-gallon tank for the consolidation locomotive. The coal, water, accessories, and tank will weigh, approximately, 80,000 pounds, to which is added the weight of the frame, which will run the static load on the bolsters up to about 90,000 pounds, or 45,000 pounds on each one. Therefore, each bolster should be of ample strength to sustain at least 68,000 pounds. In this case, four 10-inch channels are used as the sills, and the span between center-plates is 11 feet 8 inches, with an overhang of something less than half the amount. Each of these channels is capable of supporting a uniformly distributed load of about 19,450 pounds for the span given, or 77,800 pounds for the four sills, with a fiber stress of 16,000 pounds per square inch. As this span is but 55 per cent of the total length of the frame, the load actually imposed will be about 49,500 pounds, which, as compared with the 77,800 pounds, cuts down the fiber stress correspondingly; so that, as far as the longitudinal members of the structure are concerned, there is ample strength to carry the load and keep well within allowable fiber stresses.

The perfect uniformity of the loading makes it possible to determine the probable distribution of the same on the bolsters more accurately than is possible on cars, where eccentric and local loading is apt to occur. Where it is intended that the side bearings shall be in contact and carry a due proportion of the load, it is not necessary that the bolster shall be as stiff as where the whole load is to be on the center-plate. In this case, the former condition prevails and the bolster consists essentially of two broad plates 28 inches wide riveted to the top and bottom flanges of the sills, and extending across the full width of the frame. They are ½ inch thick. A diagonal plate of the same width is placed between the center and side sills, and a filling piece is inserted above the side bearings. In this way the plates are protected from bending due to the load on the side sill, and a portion of the load is carried down to the center-plate.

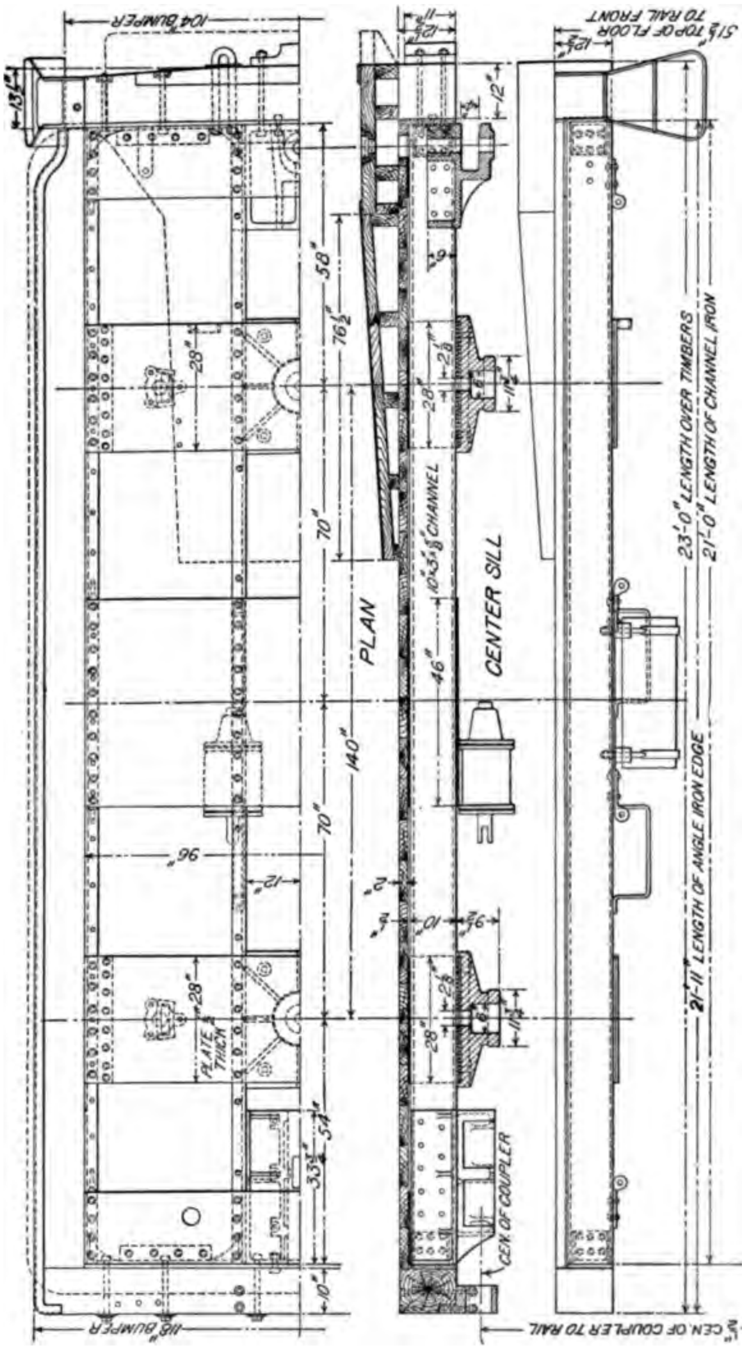


Fig. 19. Tender Frame for Consolidation Type Locomotive

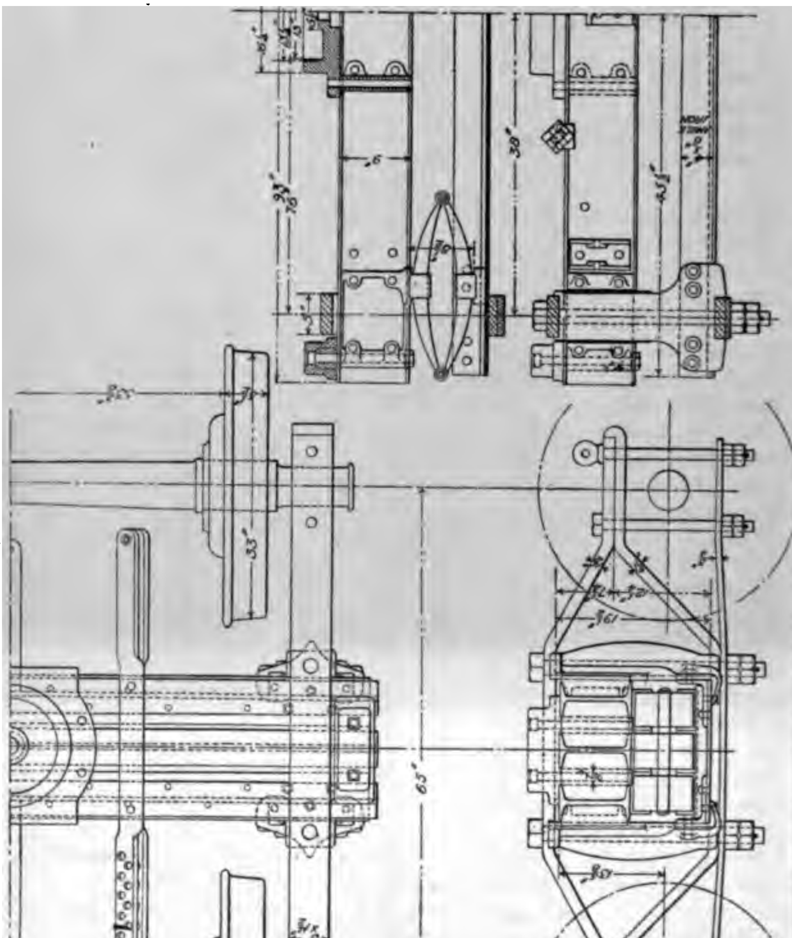


Fig. 20. Atlantic Locomotive Tender Truck for Tank of 8,000 Gallons Capacity

The center-plate should have an ample bearing surface to carry the load, and should have sufficient depth to receive the male portion, so that derailment or excessive shocks should not be able to cause a separation of the parts. As to what constitutes an ample bearing surface, there is no agreement of practice or opinion. The loads in use vary from 300 pounds to 3,000 pounds per square inch of area, in contact. In 1903 the Master Car Builders Association adopted a standard form of center-plate having 100 square inches area. This, for a loaded car, would give a pressure of about 650 pounds per square inch. In the center-plate of the tender under consideration, the area of contact is 80.16 square inches, and this, with a load of 45,000 pounds, has an imposed pressure of something more than 560 pounds per square inch, which is well within the limits of the Master Car Builders recommendations. The form of the center-plate depends upon the adjoining parts, but in any case it should be heavy and strongly ribbed to avoid deflection, and is usually made of cast iron.

Ranking close in importance with the elements forming the frame are the castings at the front and back to which the draft and buffing rigging is attached. These castings are of cast iron or steel, and are of heavy sections, securely riveted between the center sills. There is no rule for the calculation of the stresses to which they may be subjected, as the tractive effort of the engine is often many times exceeded. The draw-pin, at the forward end, should be at least 3 inches in diameter, and the casting must be of such proportions as to resist the full shearing strength of such a pin. The end plates may be 1 inch thick, riveted across the ends of the sills and held in place by suitable angles.

Tender Trucks

The trucks used under tenders are of a great variety of form. Many of these forms are patented or are the objects of special manufacture. When such a truck is to be used, the engine designer has merely to specify the weights that are to be carried, and then provide for connection to the form and dimensions offered. No attempt will be made to describe a variety of trucks, but attention will be confined to the one using an ordinary diamond side frame, like that extensively used under freight cars. As the duties of a tender are similar to those of a freight car carrying heavy bulk freight, it is quite necessary that the trucks used should be similar. Accordingly, the wheel-base is from 5 feet to about 5 feet 6 inches. The side frames, too, are identical in construction with those used under freight cars, the bars being proportioned to the load to be carried, with an allowance for extra heavy service due to the location of the tender immediately behind the engine.

The simplicity of the form of the diamond truck would render the stresses, due to vertical load, as they fall upon the arch bars, easy of analysis, were it not for the blows and horizontal thrusts to which they are subjected. Take the truck for the consolidation locomotive, for example; it is calculated to carry a maximum load of about 49,000 pounds. Of this, one-half, or 24,500 pounds, is upon each side frame. The bolts are called upon to sustain but comparatively little in tensional stress, but must be able to withstand the shear put upon them in giving rigidity to the truss which is formed by the various tension and compression members of which it is composed. By plotting the frame and solving by a simple parallelogram of forces, it will be found that the lower arch bar would have to carry a tension stress of about 28,000 pounds, while the stress on the upper one would be that of compression and be about 27,700 pounds, provided the bars were straight between the points of support. Owing to the requirements of construction, the bars have two bends between the points of support, throwing them out of a direct line, and introducing a bending moment that will be likely to crack the lower bar at the upper face of the lower bend. For this reason the bars must be made considerably stronger than would be required were there no such bending effect. The amount of this will depend upon the distance of the center of the bend from the point of support, and the angle at which the bar stands in the frame. The fiber stress set up by these conditions can be approximately determined by the use of the same methods as those suggested for the determination of the general stresses set up in the frame by the vertical loading. In addition to the vertical loads and the stresses resulting therefrom, there are severe lateral stresses set up by the lurching of the tender from side to side and the effect of centrifugal force on curves. These last are calculable by the regular formula for centrifugal action, but for the other stresses, due to the unevenness of the track, cramping of the wheels, high speed and derailments, there are no data available, and recourse must be had to the experience of the past.

Turning to the two trucks shown, that for the consolidation locomotive, Fig. 21, has been given a lower arch bar of 1⅜ inch thickness and an upper one of 1½ inch, and a tie bar of ¾ inch, all having a width of 4½ inches, based upon the considerations given above. The lower bars act in tension and the upper in compression. The bolster is of cast steel, and the axles are those adopted as the standard by the Master Car Builders Association for cars of 80,000 pounds capacity.

In the truck for the tender of the Atlantic locomotive, Fig. 20, the wheel-base is greater. The heavier loads and harder service which the high speed entails, put greater stresses on the frames, which must, therefore, be made heavier, and with this an increased diameter of column bolts are used, which are made 1¾ inch diameter. This calls for a correspondingly larger hole, so that the width of the bars is made 5 inches, and this is in accordance with the greater width of the journal, which is that of the Master Car Builders standard for cars of 100,000 pounds capacity. The top arch-bar has, in this case, a thickness of 1½ inch, as before, while that of the lower one, owing to its greater width, is made 1¼ inch, while the bottom or tie bar is ¾ inch thick.

The bolster of the Atlantic engine consists of three 9-inch I-beams held by suitable separating pieces to the columns and frames.

With this, the most important points involved in the designing of a locomotive have been treated, although there are numerous details that have been left untouched, since they belong to the class of detail work or of special manufacture. The object of this work is merely to present a guide to the general work, for it will be readily appreciated, that anyone, to undertake the designing of a locomotive, must first have had considerable drawing office experience, to whom the lling out of the parts omitted in this work will be comparatively le. The result of what has been done is shown in Figs. 22 and 23.

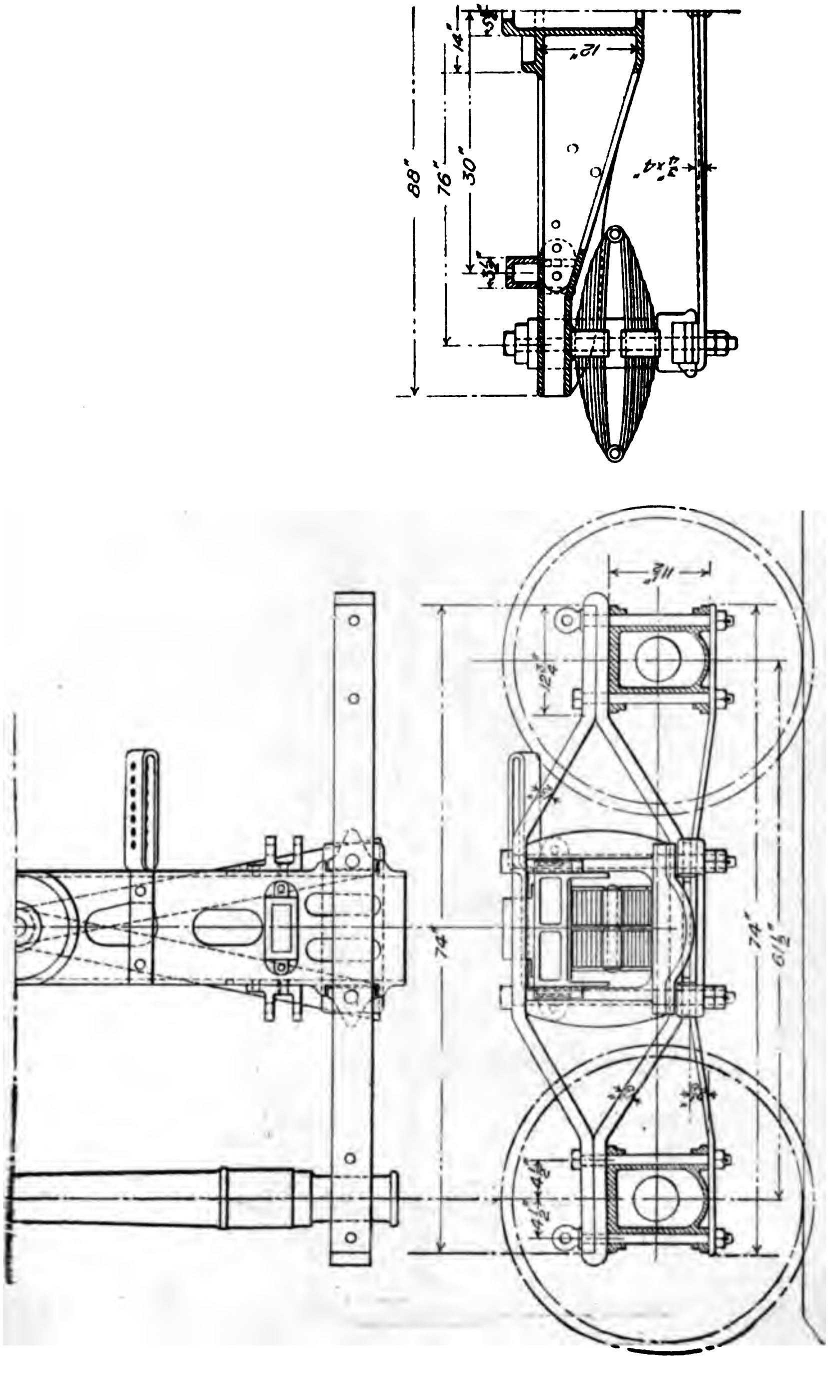


Fig. 21. Consolidation Locomotive Tender Truck for Tank of 6,000 Gallons Capacity

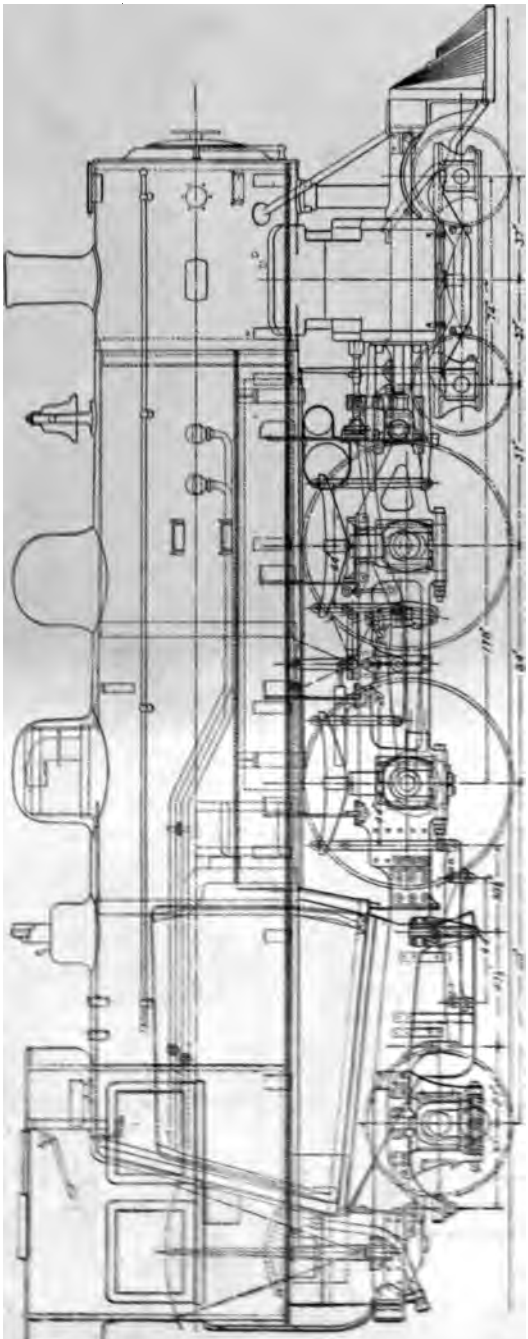


Fig. 22. Locomotive of the Atlantic Type

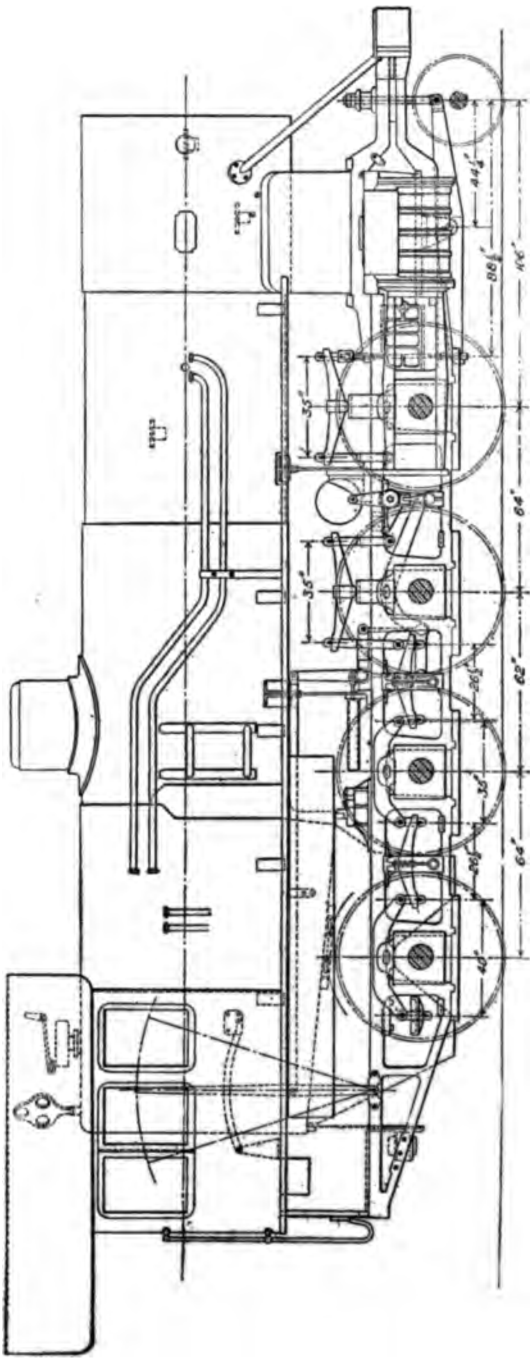


Fig. 23. Locomotive of the Consolidation Type

In conclusion, there are a few suggestions that should be constantly kept in mind during the whole progress of the work, whether it be that of designing a locomotive, a stationary engine or any other piece of machinery, and that is to bring the three elements of utility, simplicity and beauty into one harmonious whole. They are the three important factors entering into the composition, and no one of them should ever be disregarded, for they can always be combined without additional expense, and this combination should be made to enter into every detail of the work.

A successful designer must, therefore, be not only a master capable of solving the technical problems involved in a manner to obtain the highest degree of efficiency, but he must have that practical knowledge gained from experience, from which he will be able to choose the simplest methods of construction, and to this should be added an inborn instinct as to the fitness of things, which should have been cultivated by practice and study, whereby the results, though, perhaps, not falling quite within the recognized realms of art, should still be of such a character that they are pleasing to the eye, and as such claim the attention as artistic creations.

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