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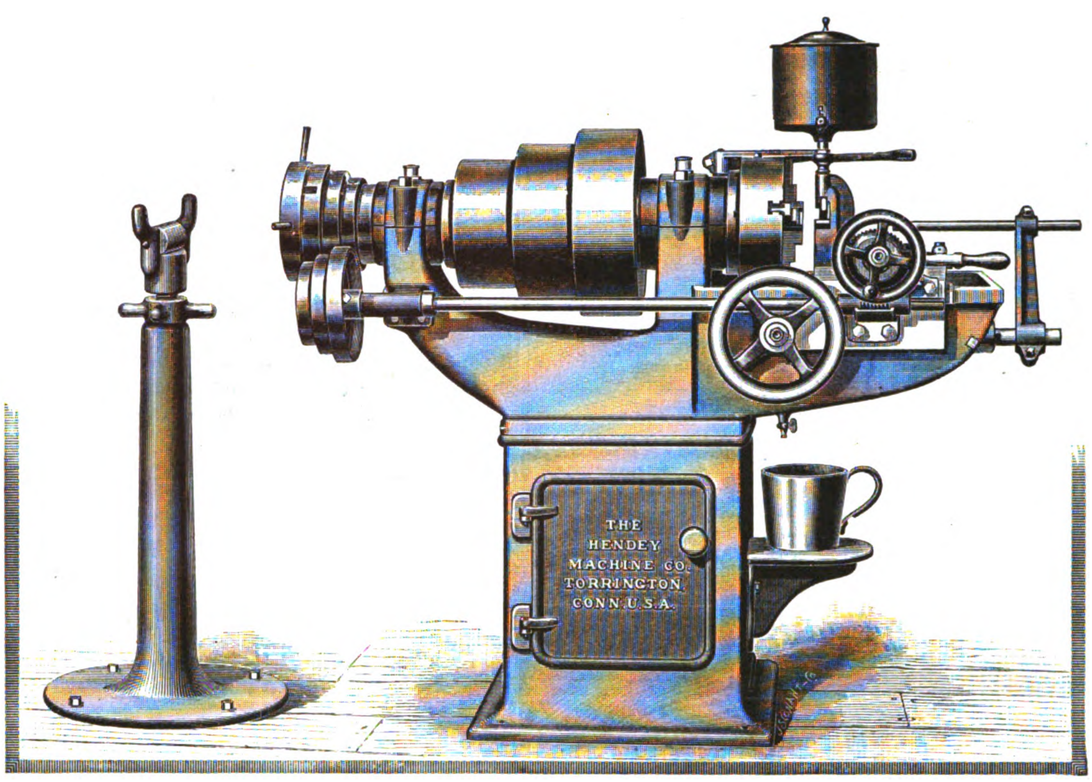
Twenty-one Inch Lathe—Cutting-off Machine.

On this page we present illustrations of two machines which have recently been added to the list of tools built by the Hendey Machine Co., Torrington, Conn.

The first of these is a cutting-off machine for cutting stock up to 2½" diameter, which, besides all the usual features found in such machines, has a tool-post, which is arranged to raise and lower the tool, and a carriage, which can be moved along the bed the same as a regular lathe carriage, so that a number of short pieces can be cut off a bar in succession, without the necessity for moving and resetting it in the chuck. The outer support for the bar is adjustable vertically to suit stock of varying diameters, and also is provided with a roller on which the bar rests, and is thus easily moved along and adjusted. The usual gauges and conveniences for handling oil and chips are provided, as shown in the engraving.



CUTTING-OFF MACHINE.

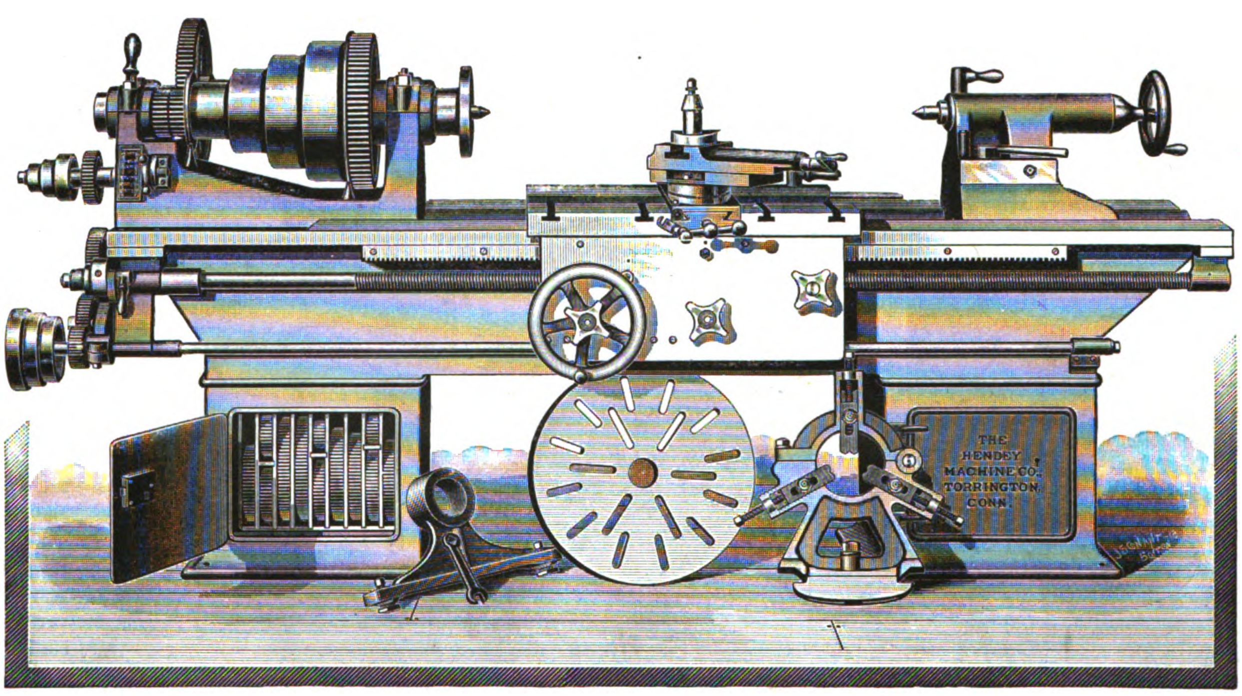
The front bearing is 4⅜"x6". The cone pulley is for a 4" belt, and its largest diameter is 13". Weight of machine, 1,400 pounds.

The lathe shown is of 21" swing, and, in some of its features, differs from the smaller ones of the same general style which we have illustrated heretofore. The pillars upon which it stands are made to serve the purpose of cupboards, one of them being specially prepared for keeping the change gears in it, and both having locks on the doors. The pillars, as will be noticed, are not at the extreme ends of the bed, but are carried in for the purpose of reducing the tendency to flexure as much as possible, and the superfluous metal then removed from the ends, giving, as will be seen, a design of pleasing appearance, while at the same time placing the metal in a position more in accordance with that indicated by the strains the tool is subject to, than is the case with the common practice.

The feed rod may be driven either by belt or gears, as may be desired, and there is an automatic cross-feed and compound rest.

The hole through the spindle is 1¾" diameter; lathe swings over bed, 21⅛"; over carriage, 14⅝". Cone pulley is for a 3¼" belt, and its largest step 13" diameter, with back gear in the ratio of 13 to 1. Spindle bearings are of hard bronze, so made as to be easily renewed, and the front bearing is 3⅜"x5¼".

Foot-stock is 14" long, and its spindle 2¼" diameter. Bed is 19½" wide, 14½" deep, with cross-webs inside 27" apart and 4" wide. Screw is of steel, 1½" diameter, and is geared to cut threads from 2 to 18 per inch. Compound rest is 7½" wide and 18" long. Weight with 8-foot bed 3,500 pounds. The lathe is, as the weight will indicate, intended for solid, hard work, and it is arranged for convenience of operation.



TWENTY-ONE INCH LATHE.

Variations in Taper—Tenons.

BY JARNO.

After measuring the center shanks with a vernier caliper, and writing a list of the ten different rates of taper, then, in order to be sure that there had been no mistake, an angular gauge, Fig. 1 (page 2), was made, by fastening two parallels or straight edges *A A* to the piece *B* with the clamps *C C*. The three centers *F*, *D*, *E*, first tried showed up as in the Fig., the angle of *E* being less, and that of *F* more than *D*. The gauge was changed to fit *E*, and the rest of the centers were tried to see whether any had less angle than *E*. After a few trials the centers were ranged side by side, according to their respective rates of taper. There had been no mistake—the angular gauge agreed with the vernier caliper. A set of Jarno taper reamers and standards will shortly be presented to Waubug.

In a set of tapers it is not usually necessary to have the larger sizes vary, at the small ends, by a difference so little as one-tenth inch, which would be the variation if the numbers were taken in order. Numbers 13, 15, 17, 19, 21 and 23 can be omitted; have every number up to 12 inclusive, and higher than 12 have Nos. 14, 16, 18, 20 and 24. This set answers for every kind of machine tools in common use—lathes, drills, boring mills, milling machines. For some special tools perhaps one or two half number small sizes can be made, each size, of course, being derived directly from the number. For example, No. 2¼ is .25" diameter at the small end, 5-16" at the large, and 1¼" long. One line of tools may need but a few sizes of tapers. For twist drill shanks there is in use a set running by pretty long steps from ⅜" to 2⅛" diameter at the small ends. The Jarno tapers for twist drills are Nos. 3, 4, 6, 8, 10, 15 and 20. With some tool makers there is a tendency to make large tapers not so long in proportion as small tapers. Seldom if ever is there any need of varying the proportion. No. 24, which goes twelve inches deep in the spindle, may appear rather long for a milling machine, but it is not too long for an arbor that goes in a horizontal gear cutting spindle, and carries a seven foot gear. Should any variation be made, let the rate of taper remain the same; let the diameter at the small end correspond to same number of taper, the only variations being in the length, and in the diameter of the large end. Do not call any irregular shank by a number; name it by the diameter of the small end, and by the depth. If we commence to call irregulars by numbers, some of our old troubles will come back. We have had enough.

In addition to the gauge lately shown in the AMERICAN MACHINIST, it is well to have two standard plugs for each number. Keep one plug for reference only, and the other for a working gauge. The number and the year should be stamped upon each plug. One of the tendencies for which we must have a remedy ever ready is that which comes from a strong desire to try a piece by the reference gauge, when it does not fit the working gauge. To avoid any wear of the reference plug, which is sure to come if this desire is gratified, the plug should be out of reach of every one except the referee. As soon as a reference gauge begins to be used upon anything except to test other gauges, so soon do we begin to drift away from the standard. It is sometimes surprising to see how easily and how soon a standard may be lost sight of unless it be marked and guarded.

In establishing the sizes for tenons of twist drills, of milling and of other arbors, we must meet the requirements of two uses of tenons—one to prevent the slipping of the shank in the spindle, the other to back the shank out of the spindle. To prevent slipping, clearly the size of the tenon should be proportioned to the diameter of the shank in order to have the strength proportional. To back the shank out, the tenon for a small size must not be proportionately thin, and for a large size it need not be proportionately thick. It is also desirable that the backing keyhole through the spindle be not large. At best the keyhole is a weakener, taking away the tie between two sides of a spindle at a place where we can ill afford to have them untied. To meet these requirements the rule should add a constant in each size, so that the small shanks should have tenons thicker in proportion than the large. These limitations are not definite, neither are those for the length of the tenon.

Hence the following rules:

Multiply .035" by the number of the taper, and to the product add the constant .035"; the sum is the thickness of tenon.

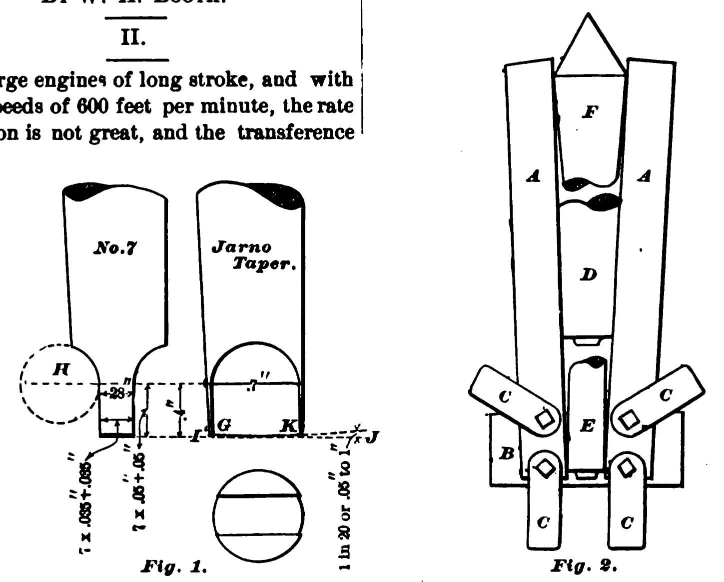
The thickness of the tenon of No. 1 taper is .07". The tenon for No. 7 is seven times .035" plus the constant .035", or .28" as in Fig. 2.

For the length of the tenon: *Multiply the number of the taper by .05", and add to the product the constant .05".* This is the length of the tenon beyond the part of the shank where the diameter of the small end is taken. For No. 7 taper, the length of tenon is seven times .05" plus the constant .05" or .4" as shown.

Tenons are frequently made with an end mill, as indicated by the dotted line *H*, leaving its curvature at the inner end of the tenon, and going up into the body of the shank. The diameter of the mill can be twice the thickness of the tenon. The mill should be carried up until its center is the length of the tenon from the outer end.

The angle for beveling the end of tenon is the same as the rate of taper—one in twenty; that is, the bevels *G K* each make an angle, equal to the taper, with the line *I J* perpendicular to the axis.

The taper of the backing key should be one in twenty.



TAPERS—TENONS.—SEE PAGE 1.

Indicator Rigging for Compound Engines.

BY FRED. W. PARSONS, ELMIRA, N. Y.

(A Paper presented at the New York Meeting of Mechanical Engineers.)

It is sometimes interesting as well as instructive to combine the diagrams taken from the two cylinders of a compound engine.

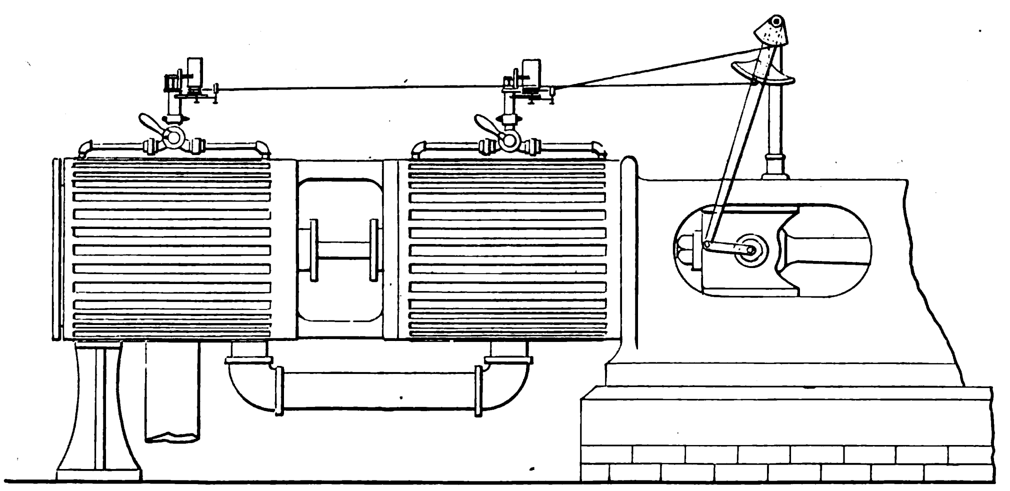
In order that the horizontal scale of measurements shall be the same, it is obviously necessary to reduce the high-pressure diagram to a length bearing the same proportion to the low-pressure diagram as the volume of the high-pressure cylinder is to that of the low-pressure cylinder.

The device described is an ordinary form of indicator rigging with two segments (as shown in the cut). The radius of the large segment is dependent upon the length desired for the low-pressure diagram.

The radius of the small segment is in the same proportion to the radius of the large segment as the volume of the high-pressure cylinder is to the volume of the low-pressure cylinder.

The diagram may be taken with separate indicators upon the high and low-pressure cylinders, as shown in the cut, and afterward traced upon the same paper; or by piping the high and low-pressure cylinders together with a three-way cock between, both diagrams may be taken on the same paper.

It will be readily seen that, by using three and four segments, this device can be used for reducing the diagrams from triple and quadruple expansion engines.



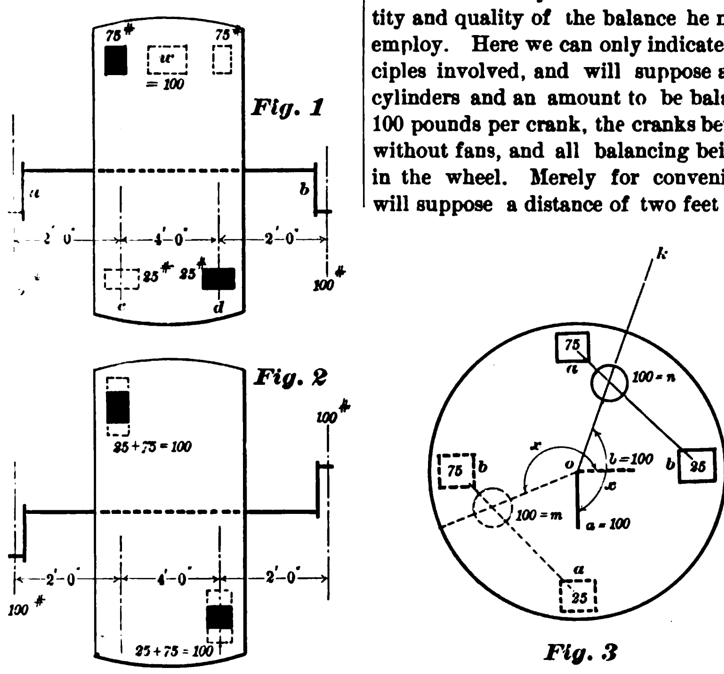
INDICATOR RIGGING.

Balancing Stationary Engines.

BY W. H. BOOTH.

II.

For large engines of long stroke, and with piston speeds of 600 feet per minute, the rate of rotation is not great, and the transference of load or the shifting of the center of gravity of the engine is performed slowly, and does not coincide with the period of vibration of the foundation. In such engines there may be little or no need of any counterbalancing, especially when the rope or belt pulley is very heavy, and probably it is only necessary to consider the conveniences of running, and arrange a balance weight in the wheel in such position and of such weight as to render it easy to stop the engine at a point from which it will again start readily. In very large engines a standing balance may perhaps best be sought for in such engines; it is of no matter where the crank or cranks may be at stopping, for there is always now fixed a barring engine with automatic disengaging motion to help the crank off the dead points. In high-speed engines of short stroke, however, there is very rapid vibration of the center of gravity, and careful balancing is required. Many small engines running at a high speed use steam of great pressure, and consequently develop large powers; at 150 revolutions and a belt speed limited to 4,500 feet per minute, the diameter of belt pulley will only be 10 feet, but its width—for single thickness of belting—may be considerable. In such an engine or pair of engines most designers would aim to reduce all horizontal causes of disturbance to a minimum, and trust to solid foundations and good bearing caps to preserve vertical steadiness. Every designer must be the final authority to himself as to the quantity and quality of the balance he means to employ. Here we can only indicate the principles involved, and will suppose a pair of cylinders and an amount to be balanced of 100 pounds per crank, the cranks being plain without fans, and all balancing being done in the wheel. Merely for convenience we will suppose a distance of two feet from the plane of revolution of the disturbing force to the plane of revolution of the nearest part of the belt pulley convenient to receive a balance weight, and we will suppose four feet more to the same plane on the opposite side of the wheel; our conditions will then be as in Fig. 1.



BALANCING STATIONARY ENGINES.

Taking crank *a* first its disturbing weight has a moment about the plane *c* of 100 pounds x 2'0" = 200 ft. pounds. We will suppose it to be balanced in the plane *d* by a weight whose lever arm about the plane *c* is 4' 0". Hence 200 ft. pounds ÷ 4' 0" = 50 pounds, which will be the weight required to revolve in the plane *d* at the same radius as the crank-pin. We will, however, place instead a weight of 25 pounds at a radius double that of the crank-pin. This weight will eliminate all torsional movements, but there will be an increased tendency to move to and fro in a direct longitudinal direction of the whole engine on its foundation. This is then exactly counterbalanced by adding a weight equal to the sum of the total disturbing weights, or 100 + 50 pounds = 150 pounds at crank-pin radius. Instead, however, we can place 75 pounds at double this radius, as in Fig. 1, to revolve in the plane *c* and opposite to the crank. If the disturbing weight of 100 pounds is supposed as entirely made up of the revolving weights at the crank-pin, and not to include anything of the weight of the reciprocating parts, such as piston, crossheads, etc., we shall have secured a standing balance of the engine, *i. e.*, it will stop equally at any point of the stroke. Many engineers are of opinion that nothing further is necessary to balance an engine, and that all the disturbing force of the piston, etc., ought to be taken up by suitable valve setting and adjustment of the rate of cushioning of the exhaust steam. Leaving this to the designer, there comes now the question of crank positions. First we will suppose both cranks to be on the same side, as in Fig. 1. Then crank *b* will require balancing exactly as crank *a*, but, as it were, cross-cornered on the plan of wheel as shown in Fig. 1, or with the weights of 75 pounds and 25 pounds as shown dotted.

We now see that each 75 pound weight is opposed by a weight of 25 pounds, and we at once remove both the smaller balances and may reduce the larger ones to 50 pounds. Still further we may use a single weight of 100 pounds in place of the two fifties, as shown at *W*. It is of course easy to arrive directly at this result without the foregoing steps, which serve to illustrate the process to be followed in complicated cases, such as Fig. 2, where the cranks are placed, as very frequently they are, opposite.

In this case the 75 pound weight for one crank will coincide with the 25 pound weight of the opposite crank, and similarly the 25 and the other 75 will coincide. In this arrangement two weights of 100 pounds each will be actually required placed in diagonal corners of the wheel in plan, and they cannot be replaced by any single weight. It has often been a matter of surprise to the writer that engine builders, who have objections to placing the cranks of compound engines at right angles with one another on account of the less favorable steam distribution thereby introduced, should so invariably place them opposite, as in Fig. 2, whereby they gain not one iota in uniformity of torsion over the method of Fig. 1, but introduce a complexity in the arrangement of balance weights, as well as doubling their weight, and for real accuracy placing them in a less sightly and dangerous position. If pistons must move stroke for stroke, they should move simultaneously and constantly in the same directions. The remarks immediately foregoing are worth the most consideration to those engineers who, while believing in simultaneously moving pistons, believe also in neglecting other than the *revolving* disturbing elements. When it is desired to balance pistons, etc., by weights disposed in the wheel, then the method of Fig. 2 has the advantage over that of Fig. 1, in that each counterbalance, whilst equilibrating its own engine in the horizontal plane, serves partially to balance the engine vertically also, and entirely balances statically; for when pistons and other *sliding* parts are balanced in the wheel the balance weights are of course heavier than the *revolving* disturbing weights by just the amount of the sliding disturbing weights sought to be balanced. Hence, if the methods of Fig. 1 were used in such a case the weight *W* would always be so much heavier than the cranks, etc., that it might cause an inequality in running, by its gravity. In method 2 the gravity of one weight counteracts that of the other, but there is introduced