INTRODUCTION

The purpose of this book is to enable any engineer, First, To understand the design, construction and use of the Crosby Steam Engine Indicator. Second, To make suitable preparation for applying it to a steam engine, and attaching the mechanism for operating the paper drum. Third, To take diagrams, read them intelligently, and, after some experience, to deduce from them such information concerning the working of an engine as a good instrument skilfully applied and handled is capable of revealing to the studious and observing mind.

CROSBY STEAM GAGE AND VALVE COMPANY

BOSTON, July 1, 1919

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friction. The cylinder is open at the bottom and fitted so that it may be attached to the cylinder of a steam engine and have free communication with its interior, by which arrangement the under side of the piston is subjected to the varying pressure of the steam acting therein. The upward movement of the piston — due to the pressure of the steam — is resisted by a spiral spring of known resilience. A piston rod projects upward through the cylinder cap and moves a lever having at its free end a pencil point, whose vertical movement bears a constant ratio to that of the piston. A drum of cylindrical form and covered with paper is attached to the cylinder in such a manner that the pencil point may be brought in contact with its surface, and thus record any movement of either paper or pencil; the drum is given a horizontal motion coincident with and bearing a constant ratio to the movement of the piston of the engine. It is moved in one direction by means of a cord attached to the crosshead and in the opposite direction by a spring within itself.

When this mechanism is properly adjusted and free communication is opened with the cylinder of a steam engine in motion, it is evident that the pencil will be moved vertically by the varying pressure of steam under the piston, and as the drum is rotated by the reciprocating motion of the engine, if the pencil is held in contact with the moving paper during one revolution of the engine, a figure or diagram will be traced representing the pressure of steam in the cylinder; the upper line showing the pressure urging the engine piston forward, and the lower the pressure retarding its movement on the return stroke.

To enable the engineer to more correctly interpret the nature of the pressures, the line showing the atmospheric pressure is drawn in its relative position, which indicates whether the pressure at any part is greater or less than that of the atmosphere.

From such a diagram may be deduced many particulars which are of supreme importance to engine builders, engineers, and the owners of steam plants.

WHAT IS THE GOOD OF AN INDICATOR?

This question was asked by a young engineer who had come to examine and purchase a Crosby indicator, with a view to rendering his services of greater value to his employer, by a knowledge and use of that instrument. His question was overheard by the proprietor of a large establishment in the city of Worcester, Mass., who took occasion to reply as follows:

"I will tell you what good an indicator did at our works. Our steam engine was not giving sufficient power for our business, and we expected to be obliged to procure a larger one. A neighbor suggested that we have our engine indicated to see if we were getting the best service obtainable from it. This was done with a Crosby indicator, and the result was that when the valves were properly adjusted and other slight changes made we had *ample power*, and the improved condition of the engine made a reduction in our coal bills during the following year of $500."

Another case: An expert engineer was called to indicate several locomotives just completed by one of our prominent locomotive builders, who had in use a large Corliss engine, which had been running only a few months. When the locomotives were indicated, the proprietor proposed that the indicator be applied to the Corliss engine, the engineer of which remarked, "Guess you'll find her all right, as she's running fine."

The first card showed that *nearly all the work was being done at one end of the cylinder.* The valves were changed and a great improvement was apparent in the running of the engine, while the actual consumption of coal was reduced from an average of 3,370 pounds per day, before the change was made, to 2,338 pounds afterwards.

These two instances are valuable in showing "the good of an indicator."

Items of Information to be obtained by the use of the Indicator

The arrangement of the valves for admission, cut-off, release and compression of steam.

The adequacy of the ports and passages for admission and exhaust, and when applied to the steam chest, the adequacy of the steam pipes.

The suitableness of the valve motion in point of rapidity at the right time.

The quantity of power developed in the cylinder, and the quantity lost in various ways: by wire drawing, by back pressure, by premature release, by mal-adjustment of valves, by leakage, etc.

It is useful to the designers of steam engines, as by properly combining the cards the effective pressure on the piston at any point can be determined, and from this the rotative effect or the rotating force acting at right angles to the crank can be calculated.

Taken in combination with measurements of feed water and the condensation and measurement of the exhaust steam, with the amount of fuel used, the indicator furnishes many other items of importance when the economical generation and use of steam are considered.

For every one of these purposes it is important that the diagrams traced by the indicator should truly represent the path of the piston and the pressure exerted on both sides of the piston at every point of that path.

INDICATOR DIAGRAMS

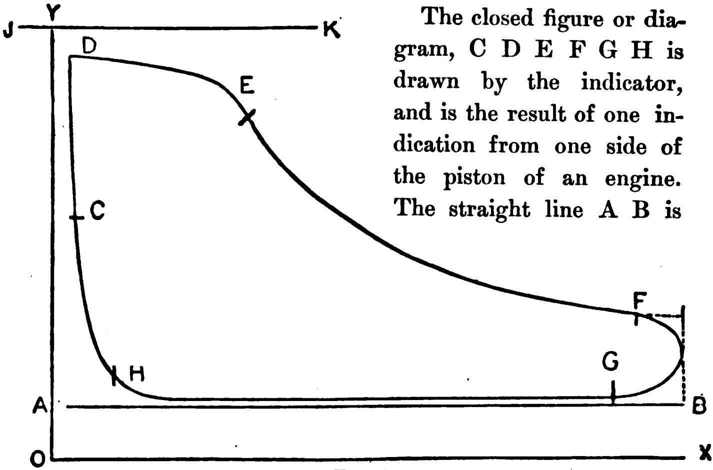
The degree of excellence to which steam engines of the present time have been brought is due more to the use of the indicator than to any other cause, as a careful study of indicator diagrams taken under different conditions of load, pressure, etc., is the only means of becoming familiar with the action of steam in an engine, and of gaining a definite knowledge of the various changes of pressure that take place in the cylinder.

An indicator diagram is the result of two movements, namely: a horizontal movement of the paper in exact correspondence with the movement of the piston, and a vertical movement of the pencil in exact ratio to the pressure exerted in the cylinder of the engine; consequently, it represents by its length the stroke of the engine on a reduced scale, and by its height at any point, the pressure on the piston at the corresponding point in the stroke. The shape of the diagram depends altogether upon the manner in which the steam is admitted to and released from the cylinder of the engine; the variety of shapes given from different engines, and by the same engine under different circumstances, is almost endless, and it is in the intelligent and careful measurement of these that the true value of the indicator is found, and no one at the present day can claim to be a competent engineer who has not become familiar with the use of the indicator, and skilful in turning to practical advantage the varied information which it furnishes.

A diagram shows the pressure acting on one side of the piston only, during both the forward and return stroke, whereon all the changes of pressure may be properly located, studied, and measured. To show the corresponding pressures on the other side of the piston, another diagram must be taken from the other end of the cylinder. When the three-way cock is used, the diagrams from both ends are usually taken on the same paper, as in Fig. 2.

ANALYSIS OF THE DIAGRAM

The names by which the various points and lines of an indicator diagram are known and designated are given on the page following. See Fig. 1.



The closed figure or diagram, C D E F G H is drawn by the indicator, and is the result of one indication from one side of the piston of an engine. The straight line A B is

FIG. 1

also drawn by the indicator, but at a time when steam connection with the engine is closed, and both sides of the indicator piston are subjected to atmospheric pressure only.

The straight lines O X, O Y, and J K, when required, are drawn by hand as explained below, and may be called reference lines.

DIAGRAM LINES EXPLAINED

The admission line C D shows the rise in pressure due to the admission of steam to the cylinder by the opening of the steam valve. If the steam is admitted quickly when the engine is about on the dead-center this line will be nearly vertical.

The steam line D E is drawn when the steam valve is open and steam is being admitted to the cylinder.

The point of cut-off E is the point where the admission of steam is stopped by the closing of the valve. It is sometimes difficult to determine the exact point at which the cut-off takes place. It is usually located where the outline of the diagram changes its curvature from convex to concave.

equal to the boiler pressure shown by the steam gage. The difference in pounds between it and the line of the diagram D E shows the pressure which is lost after the steam has flowed through the contracted passages of the steam pipes and the ports of the engine.

The clearance line O Y is another reference line drawn at right angles to the atmospheric line and at a distance from the end of the diagram equal to the same per cent of its length as the clearance bears to the piston travel or displacement. The distance between the clearance line and the end of the diagram represents the *volume* of the clearance and waste room of the ports and passages at that end of the cylinder.

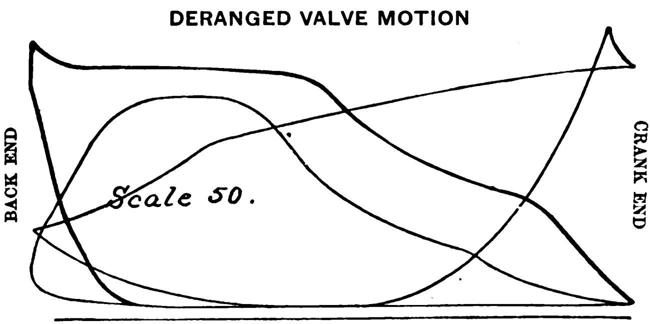


FIG. 2

In Fig. 2 the lighter lines show two diagrams, one from each end of the cylinder of a single valve high pressure engine. This valve admits the steam over its ends and exhausts inside. The derangement is caused by the valve stem being too long; consequently, at the back end the diagram shows that the steam was admitted late, cut off early, exhausted early, and the exhaust valve closed late, so that there is little or no compression. The diagram at the crank end shows the opposite defects, viz., steam is admitted too soon and carried too far on the stroke, the exhaust valve

The unit of heat, or thermal unit, is the quantity of heat required to raise the temperature of one pound of water from 62° to 63° F. This is often called the British Thermal Unit (written B. t. u.), to distinguish it from the German and the French standard, which is larger.

Mechanical equivalent of heat. It has been found by experiment that if one pound of pure water at 62° F. be raised to 63° F., energy is exerted equivalent to lifting seven hundred and seventy-eight (778) pounds one foot high, or one pound seven hundred and seventy-eight (778) feet high. This energy is called the *mechanical equivalent of one thermal unit of heat*, and it is usually designated by the letter J, and its reciprocal or $\frac{1}{778}$ by A.

Saturated steam. When steam is formed in a closed vessel in contact with its own liquid, it is said to be saturated and it will have a certain definite pressure and density corresponding to each different temperature. If, at the same time, the steam contains no liquid in suspension, it is said to be *dry* and *saturated.*

Superheated steam. If, after *all* the liquid has been converted into steam, more heat be added, the temperature will rise and the steam is said to be superheated, because its temperature will be greater than that corresponding to *saturated* steam of the *same pressure.* This difference is the number of degrees of superheat.

Priming. It is possible for saturated steam to hold particles of water in suspension. The steam is then said to be primed or wet. The amount of water so suspended may vary from zero to 5 per cent. Almost all boilers in which the steam is not in contact with the hot gases give steam primed from 0.1 per cent to 3 per cent.

Thermal efficiency of an engine. An engine having a thermal efficiency of 100 per cent would require $\frac{33000}{778} =$ 42.42 B. t. u. per H. P. per minute. An engine which used

250 B. t. u. per H. P. per minute thus has a thermal efficiency of $\frac{42.42}{250} = 0.169$, or 16.9 per cent.

Mechanical efficiency of an engine. This is the ratio of the power delivered at the fly-wheel or shaft to that calculated from the indicator cards. This ratio may be about 90 per cent.

Temperature. There are a number of different thermometric scales. The Fahrenheit and the Centigrade are the two most generally used. In the Fahrenheit system melting ice is taken as 32°, and boiling water, or steam at 14.7 pounds pressure, is taken as 212° F. In the Centigrade system melting ice is taken as 0° and steam at 14.7 pounds pressure as 100° C. A Centigrade reading may be converted into the Fahrenheit scale by multiplying the Centigrade reading by $\frac{9}{5}$ and then adding 32. The zero of each of these two systems is purely arbitrary.

It is possible to construct a thermodynamic scale of temperature based on thermodynamic laws. The zero of this scale, called the *absolute zero*, is 459.5° below the zero of the Fahrenheit scale and 273.1° below the zero of the Centigrade scale. The zero obtained by what is known as the air thermometer agrees very closely with the zero of the thermodynamic scale. The absolute temperature corresponding to 100° F. is $100 + 459.5 = 559.5°$.

Isothermal expansion. If, while a substance expands, an amount of heat sufficient to keep the temperature constant be added, the expansion is said to be isothermal. The pressure may or may not change during the expansion. In the case of a mixture of water and steam the pressure would remain constant as long as any water was present. In the case of a perfect gas the pressure would decrease in the same proportion as the volume increased.

Isothermal compression. This is the reverse of the expansion, and heat has to be abstracted during compression in order to keep the temperature constant. An amount of work has to be done on the vapor or substance in compressing it equal to that which would be obtained as useful work during an isothermal expansion between the same limits.

Adiabatic expansion. A reversible adiabatic expansion is such an expansion as would take place between cut-off and release in a steam engine or compressed air engine, if there was no heat given to or taken from the metal forming the cylinder and the piston. It is an expansion during which no heat is either added or taken away as heat. As work is done on the piston it must be paid for in some way. This work comes from the internal energy the substance had at the beginning of the expansion, and if during the expansion 7,780 foot-pounds of external work or useful work were done on the piston, then the internal energy at the end of this expansion would be 7,780 foot-pounds less than at the beginning.

During an adiabatic expansion both the pressure and the temperature drop as the volume increases. If the substance, when compressed adiabatically, goes back over the same path along which it expanded, the expansion or compression takes place at constant entropy.

Entropy is the name given to a term representing the value of a ratio, or the value of the summation of a number of such ratios. This term appears frequently in the calculation of engineering problems. An example will illustrate. Suppose a gas to expand at a constant temperature of 100° F., and that during this expansion 5 B. t. u. are added. The *increase* in entropy is $\frac{5}{100 + 459.5} = 0.0089$, and is found by dividing the heat added by the constant absolute temperature at which it was added.

Suppose that instead of the *temperature* remaining constant, the *temperature* increased 10° while the 5 B. t. u. were being added. An approximation to the increase in entropy

CHAPTER II

THE CROSBY STEAM ENGINE INDICATOR

The Crosby Steam Engine Indicator is designed and constructed to meet the exacting requirements of modern steam engineering. During the last few years, under the keen search and exhaustive tests of eminent engineers, the practice in this department of science has undergone important changes, tending to establish more correct methods and thereby to reach more accurate results; especially is this true in the use and scope of the indicator, so that the work done with such instruments in former times seems coarse and crude when compared with the more exact attainment of the present.

Educators in the scientific schools of both Europe and America have seen the importance of more exact knowledge and instruction in the technical sciences; and the great achievements of recent years in the construction of buildings, ships, armaments and machines attest the thoroughness with which research in these departments has been prosecuted; in none has there been greater progress made than in mechanical and steam engineering.

A knowledge of these facts has kept us on the alert in the manufacture of all our steam appliances, and especially in that of the steam engine indicator. Within a recent time we have made important improvements, which, as we believe, place it far in advance of any other instrument of its kind. Radical changes in design, more perfect mechanical construction, due to the use of improved and specialized machinery, and careful selection of metals for the different parts, have all contributed to this favorable result.

CROSBY STANDARD STEAM ENGINE INDICATOR

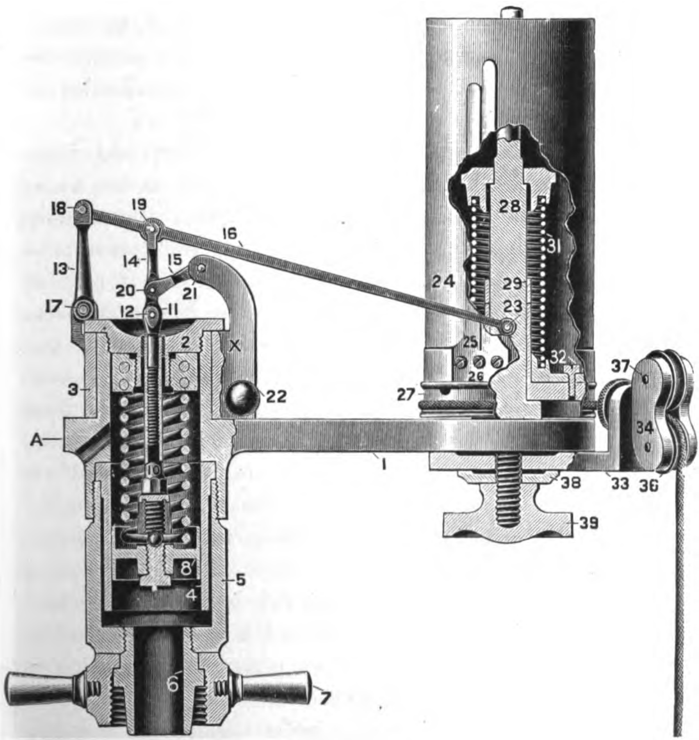
The illustration on page 19 shows the design and arrangement of its parts.

The cylinder, 4, in which the piston moves, is made of a special alloy, exactly suited to the varying temperatures to which it is subjected, and secures to the piston the same freedom of movement with high pressure steam as with low; and as its bottom end is free and out of contact with all other parts, its longitudinal expansion or contraction is unimpeded, and no distortion can possibly take place.

Between the cylinder, 4, and the casing, 5, is an annular chamber, which serves as a steam jacket; and being open at the bottom, can hold no water, but will always be filled with steam of nearly the same temperature as that in the cylinder.

The piston, 8, is formed from a solid piece of the finest tool steel. Its shell is made as thin as possible consistent with proper strength. It is hardened to prevent any reduction of its area by wearing, then ground and lapped to fit (to the twenty-thousandth part of an inch) a cylindrical gage of standard size. Shallow channels in its outer surface provide a steam packing, and the moisture and oil which they retain act as lubricants, and prevent undue leakage by the piston. The transverse web near its center supports a central socket, which projects both upward and downward; the upper part is threaded inside to receive the lower end of the piston-rod. The upper edge of this socket is formed to fit nicely into a circular channel in the under side of the shoulder of the piston-rod, when they are properly connected. It has a longitudinal slot, which permits the straight portion of wire at the bottom of the spring, with its bead, to drop to a concave bearing in the upper end of the piston-screw, 9, which is closely threaded into the lower part of the socket; the head of this screw is hexagonal, and may be turned with the hollow wrench which accompanies the indicator.

The piston-rod, 10, is of steel, and is made hollow for lightness. Its lower end is threaded to screw into the upper socket of the piston. Above the threaded portion is a shoulder having in its under side a circular channel formed to receive the upper edge of the socket, when these



parts are connected together. When making this connection *be sure* that the piston-rod is screwed into the socket as far as it will go, that is, until the upper edge of the socket is brought firmly against the bottom of the channel in the piston-rod, before the piston-screw, 9, is tightened against the bead at the foot of the spring. This is very important, as it insures a correct alignment of the parts and free movement of the piston within the cylinder.

The swivel head, 11, is threaded on its lower half to screw into the piston-rod more or less, according to the required height of the atmospheric line on the diagram. Its head is pivoted to the piston-rod link of the pencil mechanism. This adjustment of the position of the diagram upon the card is a valuable advantage peculiar to the Crosby indicator.

The cap, 2, rests on top of the cylinder, and holds the sleeve and all connected parts in place. It has a central depression in its upper surface, also a central hole, furnished with a hardened steel bushing, which serves as a very durable and sure guide to the piston-rod. It projects downward into the cylinder in two steps, having different lengths and diameters; both these and the hole have a common center. The lower and smaller projection is screw-threaded outside to engage with the like threads in the head of the spring, and hold it firmly in place. The upper and larger projection is screw-threaded on its lower half to engage with the light threads inside the cylinder; the upper half of this larger projection — being the smooth, vertical portion — is accurately fitted into a corresponding recess in the top of the cylinder, and forms thereby a guide by which all the moving parts are adjusted and kept in correct alignment, which is very important but practically impossible to secure by the use of screw threads alone.

The sleeve, 3, surrounds the upper part of the cylinder in a recess formed for that purpose, and supports the pencil mechanism; the arm, X, is an integral part of it. It turns around freely, and is held in place by the cap. The handle for adjusting the pencil point is threaded through the arm, and being in contact with a stop-screw in the plate, 1, may be delicately adjusted to the surface of the paper on the drum. It is made of hard wood with a lock-nut to maintain the adjustment.

The pencil mechanism is designed to afford sufficient strength and steadiness of movement, with the utmost lightness; thereby eliminating as far as possible the effect of momentum, which is especially troublesome in high speed work. Its fundamental kinematic principle is that of the pantograph. The fulcrum of the mechanism as a whole, the point attached to the piston-rod, and the pencil point are always in a straight line. This gives to the pencil point a movement exactly parallel with that of the piston. The mechanism is theoretically correct as well as mechanically accurate; the result is, therefore, mathematical precision in the pencil movement, not merely an approximation. The movement of the spring throughout its range bears a constant ratio to the force applied; and the amount of the movement of the piston is multiplied six times at the pencil point. The pencil lever, links, and pins are all made of hardened steel; the latter — slightly tapering — are ground and lapped to fit accurately, without perceptible friction or lost motion.

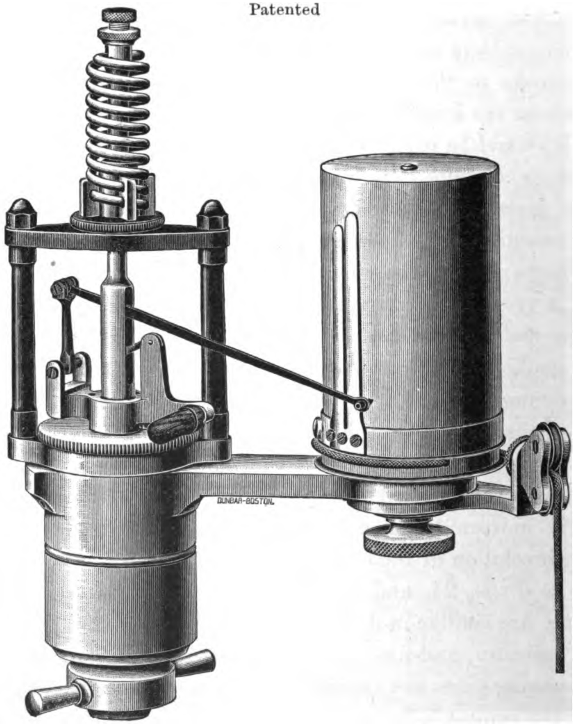
Springs. In order to obtain a correct diagram, the height of the pencil of the indicator must exactly represent in pounds per square inch the pressure on the piston of the steam engine at every point of the stroke; and the velocity of the surface of the drum must bear at every instant a constant ratio to the velocity of the piston. These two essential conditions have been attained to a greater degree of exactness in the Crosby indicator than in any other make, by a very ingenious construction and nice adaptation of both its piston and drum springs.

The piston spring is of unique and ingenious design, being made of a single piece of the finest spring steel wire, wound from the middle into a double coil, the ends of which are screwed into a metal head having four radial wings

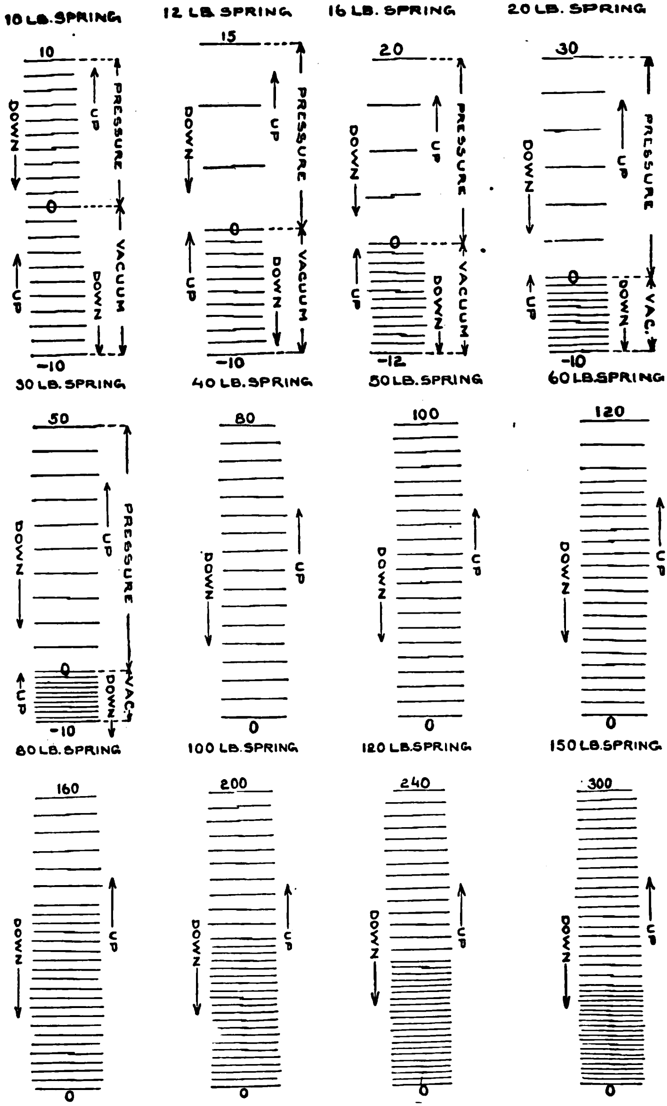
All Crosby indicators (except some of the Standard Steam Engine Indicators numbered below 3737) can be readily changed from right-hand to left-hand instruments as occasion may require.

CROSBY NEW STEAM ENGINE INDICATOR

Patented

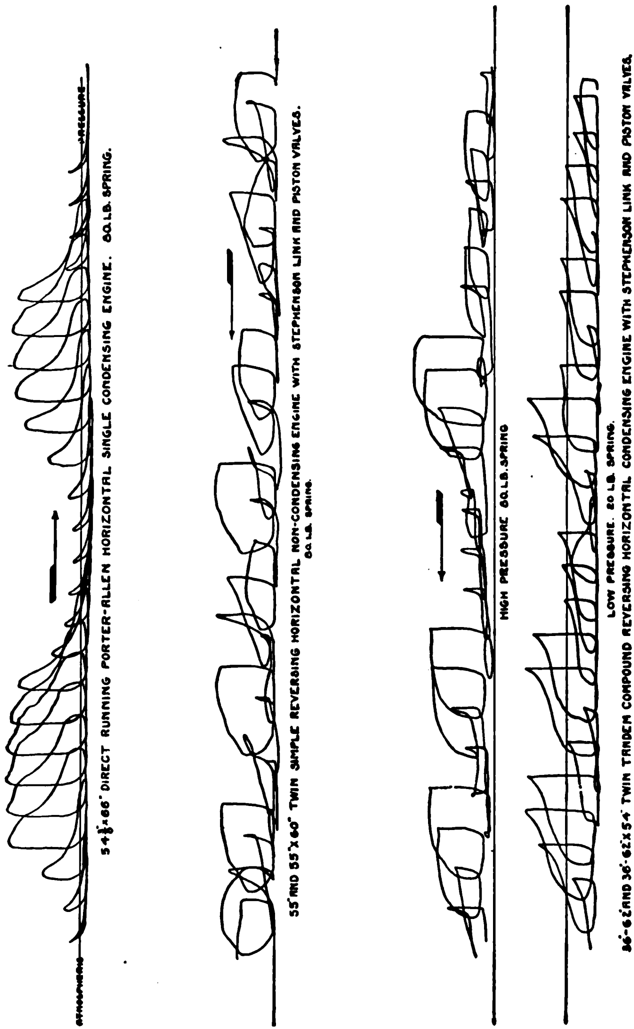


This instrument is a departure from the ordinary steam engine indicator. One difference is in the location of the spring, which is of the same form and construction as the one described and illustrated on page 22. This has been removed from the inside of the cylindrical case near the piston to the outside and affixed above the moving parts,


10 LB. SPRING
12 LB. SPRING
16 LB. SPRING
20 LB. SPRING
10
15
20
30
0
-10
-10
-12
-10
DOWN
UP
PRESSURE
VACUUM
VAC.
30 LB. SPRING
40 LB. SPRING
50 LB. SPRING
60 LB. SPRING
50
80
100
120
0
-10
80 LB. SPRING
100 LB. SPRING
120 LB. SPRING
150 LB. SPRING
160
200
240
300

The cut represents the Crosby New Steam Engine Indicator equipped with a drum for taking continuous diagrams. This drum can be applied to any indicator. It is designed to use a roll of paper 2 inches wide and 12 feet long, upon which the operation of the indicator traces a series of diagrams which will continue until the roll is exhausted, unless interrupted by the operator. The roll of paper is located within an opening in the shell of the drum, thence the paper passes around the outside of the drum and inward to the central cylinder, to which it is attached. The central cylinder is concentric with the drum, and after the paper has been wound upon it, may be withdrawn through the top and the paper easily detached. Upon the top of the drum and coöperating with the central cylinder is a knurled head loosely attached to the drum spindle, which controls the distance between the diagrams so that by adjustment they will vary in number from 6 to 100 per foot of paper. This advantage of taking any number of diagrams on the roll at the will of the operator is of importance, as he will be able to regulate the duration of the test to the speed of the engine in taking a less or greater number per foot of paper. This feature is novel and is an improvement over devices for taking diagrams of this character where the number is fixed for all engine speeds; for it enables the operator by limiting the number according to his own judgment more easily to read and measure the diagrams taken.

The demand for an indicator with a drum for continuous diagrams has been stimulated recently by its use in rolling mills, and in other industries where there is an irregular load on the steam engine, varying rapidly and in such sequence that knowledge of its continuous work could not be obtained except by an unbroken series of diagrams, extending over a definite time of greater or less extent. But its usefulness is not confined to such conditions. Its application to any steam or gas engine will afford abundant exam-



SECTIONS OF CONTINUOUS DIAGRAMS TAKEN WITH THE CROSBY INDICATOR FROM ROLLING MILL ENGINES ples of its action, continuously or intermittently at the will or convenience of the engineer. It furnishes recorded proof of such operation in a form that permits one diagram to be compared with another, and the variations during a cycle of operations to be intelligently observed in the sequence of their occurrence.

The examples on page 31 reproduced from actual diagrams taken on steam engines are given only to illustrate the foregoing statement. Single isolated diagrams can also be taken as with the ordinary drum.

THE LANZA CONTINUOUS DIAGRAM APPLIANCE WITH CROSBY INDICATORS

Patented

This apparatus for taking an uninterrupted record of the pressure changes occurring in any compression chamber or in the cylinder of any steam engine, pump, or internal combustion engine, is a device for accurately feeding a continuous strip of paper in one direction at velocities strictly proportionate to the varying velocity of the piston or pump plunger through as many consecutive strokes of the reciprocating parts as may be desired.

It is the invention of Professor Gaetano Lanza, Professor of Theoretical and Applied Mechanics, Emeritus, of the Massachusetts Institute of Technology.

The paper is drawn forward from a roll and wound afterward in form for convenient removal and study. The successive diagrams or pressure records are not overlapping or deformed in any way, but each pressure cycle is separately developed in its true proportions. The horse power and pressures can be conveniently measured by scale or planimeter and the actual location of the several events determined.

A full description of this instrument and its uses is given on page 196, in the Appendix.

CROSBY HYDRAULIC INDICATOR

Patented

The Crosby Hydraulic Indicator differs from the one shown in the cut in that it has no by-pass and in place of the piston in the upper chamber a guide is substituted.

It is a strong and efficient instrument for indicating under high pressure conditions in all liquids or gases. The piston is $\frac{1}{40}$ of a square inch in area. The cylinder is constructed in such a manner as to afford a uniform area of cross-section below the piston, thus preventing pockets or enlargements. The pencil mechanism is substantial and without appreciable inertia effect.

Fig. 6 represents the Circuit Closer, and is designed to operate the electrically connected indicators, by closing the circuit through them whenever the stylus or marking point is put against the paper on the drum of the indicator to which it is attached. This enables the engineer making the test to control this indicator directly by hand — a feature often desirable — and by its use one Sargent attachment is dispensed with.

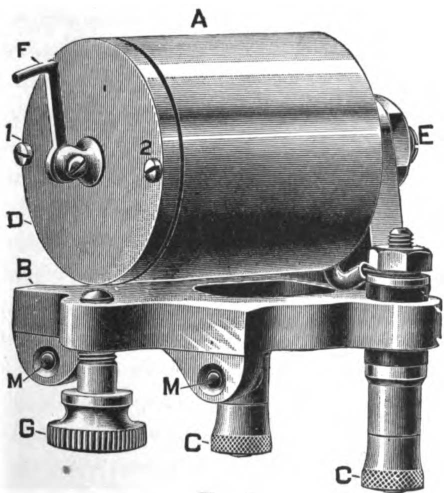


FIG. 5

It consists of a bracket, H, with a tubular projection, I, fastened to it which contains the circuit closing mechanism. It is attached to the indicator plate by the thumb-screw, J, in precisely the same way that the magnets are to the other indicators, and is electrically connected in the same manner through the binding posts, K, K.

To Attach the Sargent Improved Electrical Attachment

To get the position of the hole in the frame of the indicator, take out the screw G (Fig. 5), and place the bracket holding the magnet against the plate of the indicator, so that the hook F (Fig. 5), when placed horizontally, will point to the middle of the arm A (Fig. 3); then scribe through the screw-hole its location upon the plate of the indicator; remove the attachment, and drill a hole where marked that will allow the screw G (Fig. 5) to pass through it. Screw the attachment to the plate and adjust it so that there will be no looseness, turning the screws M M (Figs. 5 and 6), setting them up gently.

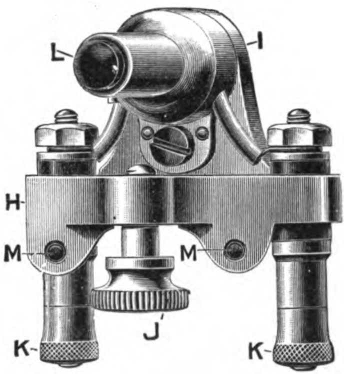


FIG. 6

Drop the hook F (Fig. 5) to a horizontal position, and bring the arm A (Fig. 3) up to its working position, and mark on it the center of the hole to be drilled for a screw-eye. This hole should be so drilled that the latch will stand level with the plate when in use. The size of the hole may be determined from the screw-eye furnished.

To Attach the Circuit Closer

The position of the hole in the indicator plate for attaching the Circuit Closer, Fig. 6, is determined in the same manner as for the electromagnet, taking care that the button L (Fig. 6) impinges the center of the arm A (Fig. 4) when the sleeve is turned into the correct position for use.

The sleeve handle of the indicator is unscrewed far enough to allow the button L (Fig. 6) in the end of the projection I to go in as far as it will, then the marking point must be adjusted until it makes the desired tracing on the paper.

To Operate the Sargent Improved Electrical Attachment and Circuit Closer

For the purpose of illustrating the manner of operating the attachment, assume that it is desirable to procure simultaneous diagrams from a compound engine, taking cards from the ends of each cylinder. Attach the indicators to the engine and arrange the drum motion in the usual manner. On each indicator secure the electrical attachment to its plate by means of screw G, as above described. Make the connections with the battery, having all of the several magnets and the circuit closer in series. Place the paper upon the drum, and bring the pencil arm into such a position as will allow the latch F to drop into the screw-eye before mentioned.

Press the armature firmly against the magnet, and adjust the marking point to the paper in the usual manner. The sleeve handle must be unscrewed enough to allow the full operation of the armature. The circuit should be closed and the armature tension springs adjusted, so that the connected attachments will work simultaneously. Everything should now be in readiness to take diagrams. Connect the drum motions, open the indicator cocks, and as soon as desirable close the circuit, and instantly all of the pencils will be brought against the papers and will remain there as long as the circuit is kept closed.

In order to put on new papers, disengage the drum motions, lift the latch, and swing the pencil arm out of the way.

The Electric Battery

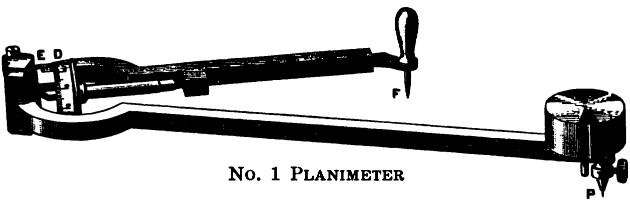
The amount of battery power required will vary with circumstances, and will range from one to two or more cells of a No. 2 Samson battery, or its equivalent.

The battery for operating the attachments is enclosed in a neat hardwood box with a suitable handle for carrying it, and is sealed so as to prevent slopping. It is very compact

The detent device has its pawl attached to the base plate of the indicator and is provided with a suitable handle for operating it. The pawl engages the ratchet located at the base of the drum. See also page 73.

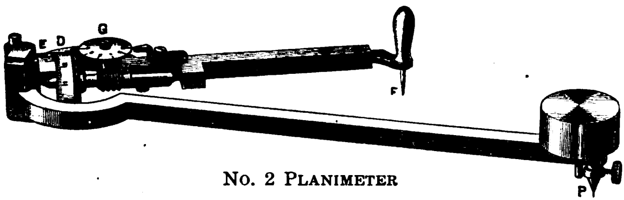
PLANIMETERS

With Directions for Using the Planimeter on Indicator Diagrams



No. 1 Planimeter

This cut represents the No. 1 planimeter. It is the simplest form of the instrument, having but one wheel, and is designed to measure areas in square inches and decimals of a square inch. The figures on the roller wheel D represent *units*, the graduations on the wheel represent *tenths*, and the vernier gives the *hundredths*.



No. 2 Planimeter

This cut represents the No. 2 planimeter, which is the same as the No. 1, with the addition of a counting disc G, the figures on which represent *tens* and mark complete revolutions of the roller wheel. By this means areas greater than ten square inches can be measured with facility. The result is given in square inches and decimals, and the reading from the roller wheel and vernier is the same as with No. 1.

the index line on the post J; from the roller wheel we read 4 (units), for the last figure that has passed zero on the vernier; we also read 7 (tenths), for that number of graduations beyond 4 that have also passed zero on the vernier (shown by the dotted line *a*), then from the vernier we read 3 (hundredths), because the *third* graduation on the vernier *coincides* with a graduation on the roller wheel.

The complete reading will then be 14.73 square inches.

When starting from zero the movement of the counting disc need not be noted when measuring *single* indicator diagrams, as they are of less than ten square inches area.

Directions for Measuring an Indicator Diagram with a No. 1 or No. 2 Planimeter

Care should be taken to have a flat, even, unglazed surface for the roller wheel to travel upon. A sheet of dull finished cardboard serves the purpose very well.

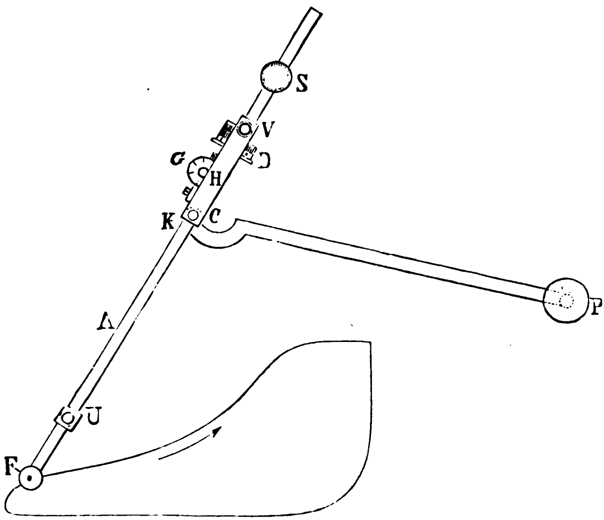


FIG. 8

Set the weight in position on the pivot end of the bar P, and after placing the instrument and the diagram in about the position shown in the cut (Fig. 8), press down the needle point so that it will hold its place; set the tracer point at any given point in the outline of the diagram, as at F, and adjust the roller wheel to zero. Now follow the outline of the diagram carefully with the tracer point, moving it in the direction indicated by the arrow, or that of the hands of a watch, until it returns to the point of beginning. The result may then be read as follows: Suppose we find that the largest figure on the roller wheel D, that has passed by zero on the vernier E, to be 2 (units), and the number of graduations that have also passed zero on the vernier to be 4 (tenths), and the number of the graduation on the vernier which exactly coincides with the graduation on the wheel to be 8 (hundredths), then we have 2.48 square inches as the area of the diagram. Divide this by the length of the diagram, which we will call 3 inches, and we have .8266 inches as the *average height* of the diagram. Multiply this by the scale of the spring used in taking the diagram, which in this case is 40, and we have 33.06 pounds as the *mean effective pressure* per square inch on the piston of the engine.

Directions for Using the No. 3 Planimeter

No. 3 planimeter is somewhat differently manipulated, although the same general principle pertains. The figures on the wheels may represent different quantities and values according to the particular adjustment of the sliding arm A. If it is desired merely to find the area in square inches of an indicator diagram, set the sliding arm so that the 10□ inch mark will exactly coincide with the vertical mark on the inner end of the sleeve H at K, Fig. 7. The sliding arm is released or made fast by means of the set-screw S.

With the wheels at zero and the planimeter and diagram noted down and this quantity subtracted from the reading when the tracing is completed. The difference between the two readings is the area sought.

For instance: Suppose we find that the reading of the wheels, including the counting disc, at the beginning is 47.31, and when the tracing is completed it is 49.43, then $49.43 - 47.31 = 2.12$ square inches, the area measured. Then to measure a second diagram, note down the last reading, viz., 49.43, and when the tracing is completed we read 51.63. Then $51.63 - 49.43 = 2.20$ square inches, the area of the second diagram.

The foregoing directions for using the planimeter are applicable to any single diagram.

The use of Amsler's Polar Planimeter in the measurement of indicator diagrams enables one to measure ten cards with it in the time which would be required to measure one card by any other method, and it insures the utmost accuracy in the work.

The planimeter is a precise and delicate instrument and should be handled and kept with great care, in order that it may be depended upon to give correct results. After using, it should be wiped clean with a piece of soft chamois skin.

THE THROTTLING CALORIMETER

In order that the test of an engine or boiler may be complete a determination should be made of the quality of the steam, i.e., the priming or the amount of moisture carried by the steam. This determination was formerly made by methods which could be made to give satisfactory results in the hands of a physicist or a trained expert, but which were troublesome and unreliable when employed by an inexperienced observer. The quality of steam delivered by a boiler or supplied to an engine can now be determined with ease and certainty by aid of the throttling calorimeter, invented by Prof. C. H. Peabody of the Massachusetts Institute of become superheated if the pressure is reduced by throttling, without loss of heat. The form here shown is simple, substantial, and inexpensive, and has been used by the inventor and others with complete satisfaction. The calorimeter, shown in Fig. 10, is a closed cylindrical metallic chamber K, having an inlet passage at A, controlled by the valve E, an outlet passage at the bottom N, and a thermometer cup at T. The chamber is thickly wrapped with asbestos and hair felt, protected by wood lagging to reduce radiation and loss of heat. The U shaped tubes or siphons for attaching the pressure gages B and C are furnished with the calorimeter; the gages and thermometer are extra, and may be furnished or not, as required.

The nipple A, connecting the inlet valve E with the chamber K, is made of composition, cut with pipe thread and provided with a well rounded orifice for gaging the flow of steam as shown by the full size Fig. 11.

The connection with the main steam pipe from which a sample of steam to be tested is taken, should be as short and direct as possible, and should be well wrapped to reduce radiation. The supply pipe should enter the main steam pipe at least half an inch, when the connection is made on the upper side of the main. If the calorimeter is attached to the bottom half of the main, the entering pipe should extend in beyond the center. The waste pipe N should be at least one inch in diameter for its entire length, and may be larger if longer than twenty feet. The gage C for measuring the pressure in the main steam pipe must be attached directly to that pipe close to the calorimeter.

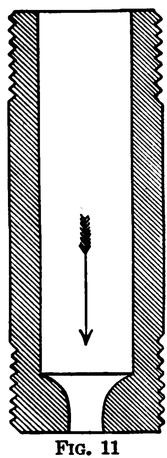


FIG. 11

To use the calorimeter, fill the thermometer cup with oil and insert the thermometer; see that the siphons are filled with cold water and that they do not leak: open the valve E wide, and wait ten or fifteen minutes till the whole apparatus is heated. Read the gage B and add the pressure of the atmosphere* to get the absolute pressure in the calorimeter; find the corresponding temperature from a table of the properties of saturated steam and compare with the temperature in the calorimeter given by the thermometer; the excess of the latter over the former is the superheating of the steam in the calorimeter. The flow of steam through the calorimeter will be sufficient to make the loss by radiation of no consequence and no correction need be applied.

When all is ready, read the pressure of the steam p in the main steam pipe, the temperature t_s in the calorimeter, the pressure $p_{,}$ in the calorimeter, and take the pressure p_a of the atmosphere. From a table of the properties of saturated steam, find the temperature $t_{,}$ corresponding to the absolute pressure $P_{,} = p_{,} + p_a$

From the same tables find the total heat $\lambda_{,}$ corresponding to the pressure $P_{,}$; also the heat of vaporization r and the heat of the liquid q corresponding to the absolute pressure in the steam pipe $P = p + p_a$

The weight or moisture in 1 pound of moist steam drawn from the steam pipe is to be calculated by the equation

$$\text{Priming} = 1 - \frac{\lambda_{,} + 0.48\,(t_s - t_{,}) - q}{r}$$

in which the factor 0.48 is the specific heat of superheated steam at constant pressure.

The calculation will be readily understood from the following example:

Pressure in steam pipe, $p = 69.8$ pounds.

Pressure in the calorimeter, $p_{,} = 12$ pounds.

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* NOTE. The pressure of the atmosphere is commonly assumed to be 14.7 pounds per square inch; it may be taken by aid of a barometer or obtained from published records of the Weather Bureau for the day. Inches of mercury can be reduced to pounds per square inch by multiplying by 0.49.

CHAPTER V

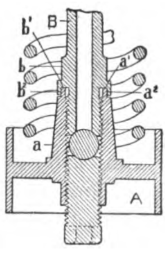
HOW TO HANDLE AND TAKE CARE OF A CROSBY INDICATOR

The indicator is a delicate instrument, and in order to secure good results from its use, it must be handled with care and be kept in good order.

The Standard Steam Engine Indicator

To remove the piston and spring, unscrew the cap; then take hold of the sleeve and lift all the connected parts free from the cylinder. This gives access to all the parts to clean and oil them.

Never remove the pins or screws from the joints of the pencil movement, but keep them well oiled.



Important. In the under side of the shoulder of the piston-rod B, is a circular channel formed to receive the upper edge of the slotted socket of the piston A. In connecting the piston-rod to the piston in the process of putting in a spring, first start back the piston-screw at the bottom of the piston, insert the spring, and then the piston-rod, and BE SURE *to screw the piston-rod into the socket as far as it will go;* that is, until the upper end of the socket, a^1, is brought firmly against the bottom of the channel, b^2, in the piston-rod. This insures a perfectly central alignment of the parts and therefore a perfectly free movement of the piston within the cylinder. Last, screw up the piston-screw lightly against the bead.

To attach a spring. Hold the hollow wrench in an inverted position and insert the piston-rod until its hexagonal part engages the wrench; then, with the spring inverted, insert the combined wrench and piston-rod until the bead of the spring rests in the concaved end of the latter; then invert the piston and pass the transverse wire at the bottom of the spring through the slot until the threads at the bottom of the piston-rod engage those inside the socket of the piston, and with the wrench screw it in as far as it will go; that is, until the upper edge of the socket is in contact with the bottom of the channel in the shoulder of the piston-rod. The piston-screw should be loosened slightly before this last operation and afterwards set up against the bead lightly, to provide against any lost motion, yet not so as to make it rigid. Next, hold the sleeve and cap in an upright position — so that the pencil lever will drop to its lowest point — and engage the threads of the swivel head with those inside the piston-rod and screw it up until the threads on the lower projection of the cap engage those in the spring head, and continue the process until the latter is screwed firmly up against the cap. Then, letting the cap go free and holding only by the sleeve, continue to turn the piston (together with its connections) until the top of the piston-rod is flush with the shoulder on the swivel head.

The piston and its connections may now be inserted in the cylinder and the cap screwed down, which will carry all parts into their proper places.

To detach a spring simply reverse this process.

To change the location of the atmospheric line of the diagram. First, unscrew the cap and lift the sleeve, with its connections, from the cylinder; then — holding the sleeve with the left hand — with the right hand turn the piston and connected parts towards the left, and the pencil point will be raised, or to the right and it will be lowered. One complete revolution of the piston will raise or lower the pencil point $\frac{1}{8}$ inch and this should be the guide for whatever amount of elevation or depression of the atmospheric line is needed.

the oil used at its last cleaning may have become gritty or gummy; it should be wiped off with a soft cloth or tissue paper saturated with naphtha or benzine, and then freshly oiled before it is used. This keeps the instrument in prime condition and insures the best results. An occasional naphtha bath will cleanse every part; but oil should be thoroughly applied afterwards to prevent corrosion.

If any grit or other obstruction gets into the cylinder it may score the cylinder wall and also cause friction and sticking of the piston. This will seriously affect the diagram and lead to bad results. It is not difficult to detect such trouble and it should be remedied at once by taking out the piston, detaching the parts and cleaning them as above described, when the disturbing cause will generally be removed. The inner wall of the cylinder should be frequently lubricated.

It is essential to know whether or not the indicator is in good condition for use; especially, to know that the piston has perfect freedom of motion and is unobstructed by undue friction. To test this successfully, detach the spring and afterwards replace the piston and piston-rod in their usual position, then, holding the indicator in an upright position by the cylinder, in the left hand, raise the pencil arm to its highest point with the right hand and let it drop. It should freely descend to its lowest point. If the opening at the bottom of the indicator cylinder be closed, by placing the thumb over it, the piston should descend slowly from its highest point, if there is no excessive clearance or leakage past the piston. These tests should be made only when all the parts are warm from the steam, in the condition as actually used, and the piston and cylinder should be carefully wiped and lubricated beforehand, at the time when the spring is removed.

The pencil should always have a smooth, fine point; a fine file is the best instrument to use to sharpen it.

in any given case may be found as follows: Divide the boiler pressure, expressed in pounds, by the desired height of the diagram, expressed in inches, and the result will be the number of the spring required. For instance, if the boiler pressure is 70 pounds and the desired height of the diagram is $1\frac{3}{4}$ inches, then $70 \div 1\frac{3}{4} = 40$, the number of the spring required.

In practice, the best diagram for measuring and interpreting is one in which the length is not more than twice the height, for the reason that the points of cut-off, exhaust and compression are better defined than in a longer diagram.

The late John C. Hoadley, an eminent authority in the use of the steam engine indicator, said:

"There are good reasons for keeping the diagram of very moderate length. From $2\frac{1}{2}$ to 3 inches will be found long enough to admit of all useful division, and the movement of the paper cylinder will be slower, and the tracing correspondingly more delicate than if a longer card is made. A similar remark applies equally well to the vertical motion, which can be reduced to any amount desired by using springs of suitable stiffness."

Sir Frederick Bramwell succeeded in obtaining very satisfactory diagrams at extraordinary speeds and high pressures, by limiting the dimensions to one inch in length and width, but this was an extreme case. It is generally more convenient and satisfactory to make the length about $2\frac{1}{2}$ to $3\frac{1}{2}$ inches, and the height from the atmospheric line about $1\frac{1}{2}$ to $1\frac{3}{4}$ inches, by selecting a spring adapted to the pressure.

INDICATOR SCALES

The scales for measuring diagrams are sometimes made of steel, with several different graduations marked on each; they are more often made of boxwood, and these we recommend as being more easily read and less liable to be misapplied, as there is only one graduation marked on each scale of this kind. The scale with which to measure the height of a diagram must correspond with the number of the spring used in the indicator when the diagram is taken. For a diagram taken with a number 40 spring, use a scale graduated in 40 divisions to the inch, or for a diagram taken with a number 60 spring, use a scale graduated in 60 divisions to the inch, and so of all other springs.

Place the scale on the diagram at right angles to the atmospheric line with the zero mark of the scale exactly on that line ; the figures set against these divisions show — at whatever point the line of the diagram crosses the edge of the scale — the pressure per square inch exerted by the steam on the piston of the indicator in tracing it. The divisions below the zero point show vacuum.

The most common scales are those numbered 40, 50, and 60 ; that is, an inch of vertical height on the scale represents, according to the number of the corresponding spring used, 40, 50, or 60 pounds of steam pressure per square inch in the cylinder.

piston when at the end of the stroke, so as not to be obstructed by it, and away from steam passages, to avoid strong currents of steam. By placing the engine on a dead center, it is easy to tell how much clearance there is, and the hole should be drilled into the middle of this space; the same process should be repeated at the other end of the cylinder.

On horizontal engines the most common practice is to drill and tap holes in the *side* of the cylinder at each end and insert short ½-inch pipes with quarter upward bends, into which the indicator cocks may be screwed; on some horizontal engines it may be more convenient to drill and tap into the top of the cylinder at each end and screw the cocks directly into the holes. On vertical engines, for the upper end of the cylinder the cock may be screwed into the upper *head* or *cover*, and for the lower end, into the *side* of the cylinder, after drilling and tapping the necessary hole. It is preferable to drill the holes in the *sides* of a cylinder rather than the heads, because the former gives better results and requires less pipe and fittings.

Before deciding just where to drill the holes it is wise to consider *all* the conditions of the case and devise the *whole* plan for indicating the engine.

Sometimes a drum motion can be erected more advantageously in one place or position in the engine room than another, or one kind may be better adapted for a given place than another. Again, the type of engine and position of the steam chest, the kind of cross-head and the best means for attaching to it, the position of the eccentric, its rods and connections, all should be taken into account when determining the best places to drill the cylinder and locate the indicator, in order to secure a proper connection with the reducing motion, a perfectly free passage for steam to the indicator and the most convenient access to the instrument for taking diagrams.

in such a manner as to permit it to swing edgewise and parallel with the guides of the engine. Near the bottom of the lever is a steel stud, secured by a nut on the outside shown at B. This stud has a T-head projecting inwardly from the lever, and is formed to run freely, but without looseness, in a T-slot, cut in an iron plate, and firmly attached to the center of the cross-head, which, as it moves to and fro, gives to the lever the necessary swinging motion.

Fig. 14 shows the arrangement of the T-headed stud, *a*, in connection with the slotted iron plate, *c*; one of the screws by which it is attached to the cross-head is shown at *d*. The head of the stud should be about one inch in diameter and the shank about one-half inch. The T-slot is milled out of a cast-iron plate of suitable size and shape to give the proper run for the stud.

When the lever is perpendicular, or in the middle of its path, the stud should be near the bottom of the slot, which should be long enough to retain the stud when the cross-head is at the extreme end of the stroke.

By this device the bottom end of the lever is moved the full distance that the cross-head travels in either direction, and for this reason it is more accurate than a lever of the same kind having its lower end slotted to work on a stud inserted in the cross-head, as is sometimes used. *D* is a small pulley placed near and on a level with the pin in the lever, for the indicator cord to pass over.

While this form of reducing lever is commonly made with a *pin* for attaching the indicator cord, greater constancy of motion for the drum would be attained by the use of a sector, such as is shown at S in Fig. 15.

To find the point on the lever at which to attach the indicator cord, proceed as follows: Divide the length of the lever by the length of the piston stroke, and multiply the quotient by the required length of the diagram, all expressed in inches and decimals of an inch, and the product will be the proper distance from the pivot in the top of lever to the point of attachment.

For example: If the lever is 48 inches long and the piston stroke is 30 inches, and we wish to obtain a diagram $3\frac{1}{2}$ inches long, we have $48 \div 30 = 1.6$; $1.6 \times 3.5'' = 5.6''$, the radius required to give a $3\frac{1}{2}$ inch diagram. If we require a diagram $4\frac{1}{2}$ inches long, then: $1.6 \times 4.5'' = 7.2''$, the radius required to give a $4\frac{1}{2}$ inch diagram.

The object of all mechanisms for actuating the drum of the indicator should be such that the relation of piston to drum movement will be constant. Such constancy cannot, however, be fully attained by the use of any form of reducing lever, and so should not be employed when important adjustments or tests are to be made. Their simplicity and the small expenditure of time and money in their construction may entitle them to favorable consideration on the part of beginners in the use of the indicator, and when only ordinary work is to be done.

The forms shown in Fig. 16, Fig. 17, Fig. 18, and Fig. 19 are correct in principle, and when carefully constructed may be relied upon to give correct results.

The Brumbo pulley, shown in Fig. 15, is another form of reducing lever, and one often used by engineers, especially on locomotives. It can be quickly and cheaply made and can be used on almost any kind of engine. The swinging lever E is a strip of straight

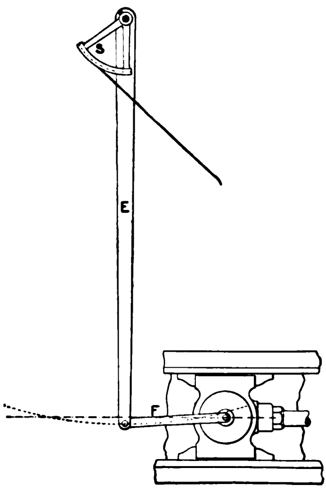


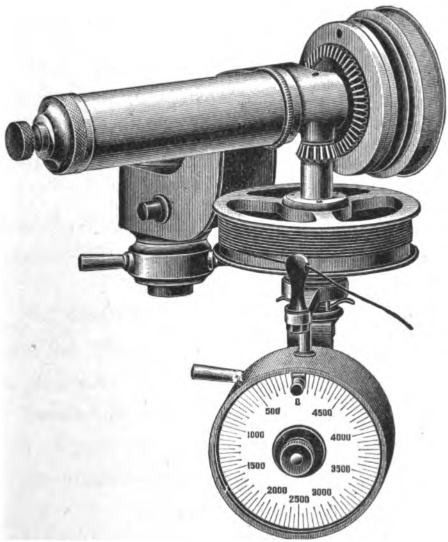
FIG. 15

grained pine, one inch or more in thickness, three to four inches wide, and from one and a half to two times as long as the piston stroke. It is suspended by a bolt or screw, serving as a pivot, from a frame or truss overhead constructed for that purpose, and is connected at its lower end by the wooden link F, to the usual stud or pin fixed in the center or other convenient part of the cross-head; the link should be from one-third to one-half the length of the piston stroke. In the illustration the proportions of lever and link are as 60 to 15; the lever being two times and the link one-half the length of stroke.

The sector S may be constructed of wood or of metal, as here shown; it has a groove in its circular edge for the cord to run in and is screwed to the upper end of the lever or pendulum, so that its center will coincide with the center of the pivot on which it swings. The radius of the sector which is necessary to give the proper motion to the drum to obtain the desired length of diagram can be found as follows: Divide the length of the lever by the length of the piston stroke and multiply the quotient by the length of diagram desired, and the product will be the required radius, all the terms being expressed in inches and decimals of an inch. For example: If the lever is 30 inches long and the piston stroke is 20 inches, and we wish to obtain a diagram 3 inches long, we have $30 \div 20 = 1\frac{1}{2}$; $1\frac{1}{2} \times 3'' = 4\frac{1}{2}''$, the radius required to give a 3 inch diagram.

When the conditions are favorable, the lever should be hung so that it will swing in a vertical plane, parallel with the guides and in line with the indicator, as this arrangement is the most simple, and the use of guide pulleys is avoided. It is not absolutely necessary, however, that the lever shall swing in a vertical plane, but it may swing in a plane at any angle thereto, where the conditions require it. In such cases a man's ingenuity and inventive faculty must aid him. A link made of a thin strip of

The latter is actuated by the moving parts of the reducing wheel and records on a chart every revolution of the engine; so that during the taking of the diagrams by the indicator attached to the reducing wheel the revolutions of the engine are recorded simultaneously.



Its capacity to record 5,000 revolutions permits its use during a considerable period of indicating work; and the average number per minute so determined is more accurate for such purpose than if the revolutions were merely counted intermittently by the ordinary speed instruments. Besides, there is thus preserved by the chart a record of the work done, to be filed with the diagrams taken by the indicator for future consideration.

It has recorded upwards of 4,000 revolutions per minute without a fault. No difficulty will arise in its attachment and use.

After the reducing wheel has been adjusted to the

Bushings may be obtained for attaching other than the Crosby indicators, and elbow nipples are made for attaching the reducing wheel to a vertical engine.

TESTING THE ACCURACY OF REDUCING MECHANISM

Whatever drum motion mechanism is used, its accuracy can be easily tested in the following manner: Lay off on the engine guides points at $\frac{1}{4}$, $\frac{1}{2}$, and $\frac{3}{4}$ of the stroke. Connect the indicator with the drum motion in the same manner as for taking diagrams. When the cross-head is on either dead center, touch the pencil to the paper and make a vertical mark, and in the same way make vertical marks when the cross-head reaches each successive quarter point on the guides. If the marks are exactly at fourths on the card, the motion of the cross-head has been accurately reduced.

CHAPTER VIII

HOW TO TAKE DIAGRAMS

First connect the indicator to the indicator cock.

Adjust the guide wheel under the drum so that the cord leads from this wheel in the right plane.

All reducing motions of the pantograph type, such as the lazy tongs shown by Fig. 16 and the modifications shown by Figs. 17 and 18, require for correct reduction of motion that the string to the indicator should run in a line parallel to the line of motion of the piston-rod.

With reducing motions of the Brumbo pulley type, shown by Fig. 15, it makes no difference at what angle the string leads from the guide wheel to the sector. The string must, of course, be in the plane of the sector.

After the guide wheels under the drum have been adjusted and fastened, change the location of the hook on the drum string so that when the hook is pulled away out as far as it will go, it overlaps the pin or ring or loop or whatever is provided for a connection on the reducing motion, by about $\frac{3}{8}$ of an inch. Then let the drum spring pull the hook away back and note how near the hook comes to the pin or loop on the reducing motion. If the hook is drawn back $\frac{3}{8}$ of an inch beyond the travel of the pin or loop on the reducing motion the cord is the right length. Should the distance at this end be 1 inch, and at the other $\frac{3}{8}$ of an inch, the cord should be lengthened so as to make the distances alike at the two extremes of the travel.

The paper is now put on the drum. The paper must be tight and pushed down to the bottom of the clips.

There are two ways in which the paper may be held by the clips. The ends of the paper may be brought out in the center between the two clips, or the two clips may be used as one piece and the two ends lapped under them.

Next, connect the drum to the reducing motion. If one understands how to do this, it makes no difference whether the engine is making 10 or 600 revolutions per minute. Oftentimes it is amusing to watch one who does not know how, try to catch the loop in the string from a Brumbo pulley reducing motion on a high speed engine.

With one hand pull the hook on the indicator cord out as far as it will go. With the other hand take hold of the string from the Brumbo pulley and let the string be pulled through your fingers till the loop is reached. Hold the loop so that each time the engine reaches the crank end of its stroke you will feel a slight pull on the loop due to the winding of the string on the sector. The hook on the indicator cord will reach $\frac{3}{8}$ of an inch beyond the end of the loop, making it easy to connect. The speed of the engine, it will be seen, does not make any difficulty about connecting on if this method is used.

Steam is now turned on the indicator through the indicator cock, and after a period of ten seconds, in which time the indicator is being heated up by the steam, a card may be taken by pressing the wooden handle moving the pencil mechanism up against its stop. After taking the card, close the cock and draw an atmospheric line. Then disconnect the drum string from the reducing motion.

Should the lines on the card be too faint, the wooden handle may be screwed back a turn or two.

Formerly, graphite was used for the marking point and ordinary paper as drum paper. To-day metallic paper (a paper coated with a salt of lead) is almost universally used and the marking point is a piece of soft brass. This makes it possible to get very fine lines on the card and to take cards with very little pressure on the marking point. Any friction between the pencil and the paper makes an error in the card, and cards should be taken with as faint lines as possible on this account.

Make notes on the card of as many of the following facts as possible. The day and hour of taking the diagram; the kind of engine from which the diagram is taken, and which engine, if one of a pair; which end of the cylinder, the diameter of the cylinder, the length of the stroke, the diameter of the piston-rod, and the number of revolutions per minute; the position of the throttle; the atmospheric pressure; the steam pressure at the boiler and at the engine, by the gages; the vacuum by the gage on the condenser and the temperature of the feed at the boiler; if the engine is compound, the pressure in the receiver; the scale of the spring used in the indicator; the volume of the clearance at each end of the cylinder, and what per cent of the piston displacement each of these volumes is. (Directions for ascertaining the volume of the clearance, and what per cent that volume is of the piston displacement, are given on pages 94 and 96.)

It is often useful to make notes of special circumstances of importance, such as a description of the boiler, the diameter and length of the steam and exhaust pipes, the temperature of the feed water, the quantity of water consumed per hour, etc.

On a locomotive, note the time of passage between mile-posts in minutes and seconds, from which, when the diameter of the drivers is known, the number of revolutions per minute may be calculated. Note also the position of the throttle and the link, the size of the blast orifice, the weight of the train, and the gradient.

On diagrams from marine engines note, in addition to the general facts, the speed of the ship in knots per hour, the direction and force of the wind, the direction and state of the sea, the diameter and pitch of the screw, the kind of coal, the amount consumed, and the ashes made per hour.

CHAPTER IX

HOW TO FIND THE POWER OF AN ENGINE

To find the power actually exerted within the cylinder of a steam engine, it is necessary to ascertain separately *three factors* and the product of their continued multiplication. These factors are: The area of the two sides of the piston; the total travel of the piston in feet per minute; and the mean effective pressure urging the piston forward, designated M. E. P.

The piston area. This, at the back end, is the same as the area of cross-section of the cylinder; at the crank end it is the same, less the area of cross-section of the piston-rod. These areas may be found from their diameters in a table of the areas of circles, or may be computed by multiplying the square of the diameter in inches by the approximate number 0.7854.

The travel of the piston. The total travel of the piston in feet per minute is found by multiplying twice the length of the stroke measured in *feet*, by the number of revolutions of the crank shaft per minute, which should be carefully ascertained by taking the mean of many countings, or the readings of a speed counter during a considerable time. The mean piston speed will be expressed in terms of *feet per minute.*

The mean effective pressure. There are several approximate methods for computing the mean effective pressure, one of which is to divide the diagram into ten equal parts, as shown in Fig. 20. Then through the points of division draw lines, which are called ordinates, at right angles to the atmospheric line. The mean heights or pressures of the small areas thus formed are indicated by the dotted lines midway between the ordinates.

The mean effective pressure of the whole (of each) diagram may now be found, by measuring (on the dotted lines) the mean pressure in each of the small areas with the scale corresponding to the spring used in taking the diagram.

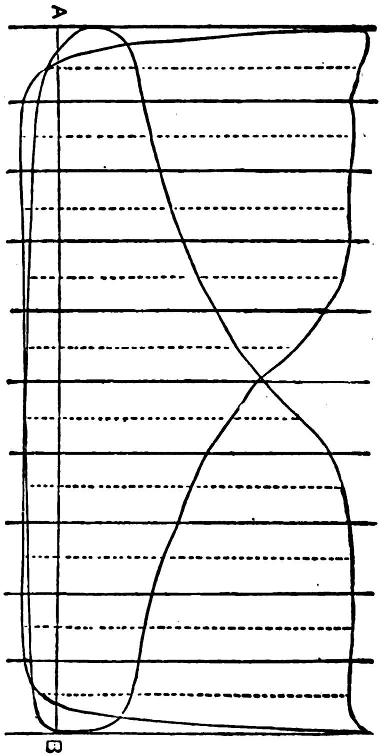


FIG. 20

DIAGRAMS FROM HARTFORD ENGINE. CYLINDER, 16 x 24 INCHES. BOILER PRESSURE, 87 POUNDS. VACUUM BY GAGE, $23\frac{1}{4}$ INCHES. 130 REVOLUTIONS PER MINUTE. SCALE, 50

The sum of these mean pressures, divided by 10, the number of divisions, will give the mean effective pressure sought, in pounds per square inch.

If a diagram has many irregularities of outline, it may be necessary to divide it into twenty equal divisions to insure a correct measurement of the pressures; in such a case we divide the sum of the pressures by 20 instead of 10. In other cases, when irregularities occur only in a part of a diagram, it is only necessary to subdivide one or more of the ten divisions to insure greater accuracy in that part; in such a case we must measure the pressure in each subdivision and divide their sum by 2 to get the mean pressure of that division. (See Fig. 22 for a full illustration of this method.)

If the scale is not at hand, the heights of the divisions may be pricked or marked off on a strip of paper, one after the other continuously until all are measured; then the distance from the end of the strip to the last mark will represent the sum of all the measurements, which can be measured in inches with an ordinary rule. This quantity, divided by the number of divisions in the diagram — or diagrams, if there are two — and multiplied by the scale of the spring used, will give the average or mean effective pressure, the same as by the other method.

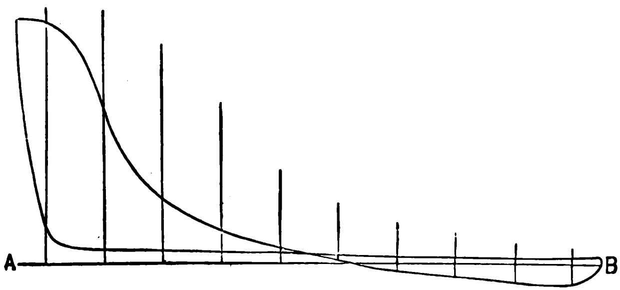


FIG. 21

When there is a loop in the diagram, as in Fig. 21, the area enclosed in the loop should be subtracted from the other part, as it represents loss of efficiency.

The quickest and most accurate method for measuring the diagram and finding the mean effective pressure is by the use of Amsler's Polar Planimeter. With careful manipulation, the planimeter will give the *exact area* of a diagram in

CALCULATING I. H. P.

Let a = the area of the head end of the piston in square inches.

Let r = the area of the piston-rod in square inches.

Let s = the stroke of the engine in feet.

Let n = the number of revolutions per minute.

The I. H. P., or indicated horsepower, of the head end of the cylinder, is

$$\frac{a \times (\text{M. E. P. of head end}) \times s \times n}{33{,}000}$$

The I. H. P. of the crank end is

$$\frac{(a - r) \times (\text{M. E. P. of crank end}) \times s \times n}{33{,}000}$$

The I. H. P. of the cylinder is the sum of these two.

If the M. E. P. and the revolutions per minute are each made equal to 1 in the preceding expressions, then the result obtained from each is the I. H. P. per pound M. E. P. at one revolution per minute. These factors are called the *engine constants*. The *engine constants* for the two ends are respectively

$$\frac{a \times s}{33{,}000} \quad \text{and} \quad \frac{(a - r) \times s}{33{,}000}$$

If the valves of an engine are well adjusted and the M. E. P. for the two ends of the cylinder is nearly the same, an approximate calculation of the indicated horsepower of the cylinder may be made by multiplying the average M. E. P. of the two ends by the number of revolutions per minute and by the sum of the engine constants.

DISCUSSION OF M. E. P.

The indicator card from one end of a cylinder shows the pressures on that end *during a revolution*, or a stroke forward and a stroke back.

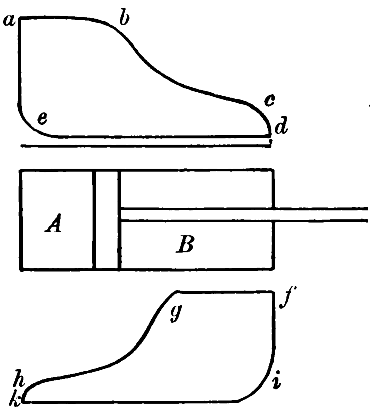
The Mean Effective Pressure, M. E. P. (improperly named perhaps) used in figuring horsepower is calculated from the indicator cards as already explained.

The effective pushing pressure or pulling pressure per square inch of piston area at any point of the stroke is evidently the difference between the total pressures on the two sides of the piston at that point divided by the piston area. Obviously the average pressure, calculated in this way, would not be the same as the mean effective pressure.

(See discussion of stroke cards given later on.)

That the horsepower obtained is correct when the M. E. P. is calculated as previously explained is shown by the illustration which follows.

Let the mean pressure above the atmospheric line corresponding to line $a\ b\ c\ d = P_f$.



Let the mean pressure above atmospheric line corresponding to line $f\ g\ h\ k = P_b$.

Let the mean pressure above the atmospheric line corresponding to $k\ i\ f = E_b$.

Let the mean pressure above the atmospheric line corresponding to $d\ e\ a = E_f$.

Let A = area of head end of piston in square inches.

Let B = area of crank end of piston in square inches.

Let s = stroke in feet.

The total mean pressure on the piston-rod during the forward stroke of the piston is

$$A\ P_f - B\ E_b$$

The total mean pull on the piston-rod during the return stroke is

$$B\ P_b - A\ E_f$$

The total foot-pounds per revolution is

$$A\ s\ P_f - B\ s\ E_b + B\ s\ P_b - A\ s\ E_f$$
$$\text{or } A\ (P_f - E_f)\ s + B\ (P_b - E_b)\ s$$

$P_f - E_f$ is the mean effective pressure of the forward card.

$P_b - E_b$ is the mean effective pressure of the crank card.

CALCULATING HORSE-POWER FROM GAS ENGINE CARDS

There are two types of gas engines — the four-cycle and the two-cycle. Either type may be built single acting or double acting. A four-cycle single acting engine can have but one working stroke in four strokes, or in two revolutions. A two-cycle single acting engine has one working stroke per revolution.

There are two classes of engines of the four-cycle type. In one class the governing is by missing working strokes when the engine is at speed. The proportion of missing strokes to working strokes varies with the load. At full load there is one missing stroke to seven or eight firing strokes.

In the other class there is a working stroke regularly at every fourth stroke, but the power of the working stroke is varied by throttling the charge as it is drawn into the ture all through the exhaust stroke and up to the time the fresh charge enters. This is more apt to be true with slow burning mixtures, which are weak, than with rich mixtures. In some instances the indicator piping has had to be made of smaller size and an indicator with a very small piston used, in order to prevent this firing of the incoming charge.

Two-cycle Engines. An indicator with light spring should be attached to the crank case or compression chamber of the small two-cycle engines in addition to that on the working cylinder. The work of compression in the crank case is to be deducted from that shown by the working cylinder. Large two-cycle engines compress the gas and air each in separate compressors and deliver the same to the engine cylinder where the charge after mixing is put under additional compression. It is of course necessary to have indicators on both of these compressors.

The theoretical efficiency of a four-cycle gas engine is

$$\text{Eff.} = 1 - \left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}}$$

where p_1 is the absolute pressure at the beginning of compression, p_2 is the absolute pressure at the end of compression and k is the ratio of the specific heats of the unburned mixture at constant pressure and at constant volume.

The efficiency may also be expressed as,

$$\text{Eff.} = 1 - \frac{1}{r^{k-1}}$$

r being the ratio of compression which may be found by dividing the total volume of the cylinder by the volume of the clearance space.

The value k varies from 1.27 to 1.40. In general it may be said that the lean gas mixtures have higher values of k than rich gas mixtures.

The Actual Efficiency. Tests made on large gas engines show that of the heat of combustion of the gas supplied, about forty per cent is carried off in the jacket water, about thirty-five per cent goes off in the exhaust, and only about twenty-five per cent is converted into work, as shown by the card.

Example. A small gas engine uses 22 cubic feet of city gas per I. H. P. per hour. The gas has a heating value of 700 B. t. u. per cubic foot. The heat units supplied per minute are $\frac{22 \times 700}{60} = 257$; a horsepower is 33,000 foot-pounds per minute, equivalent to $\frac{33,000}{778} = 42.42$ B. t. u. The thermal efficiency per I. H. P. is $\frac{42.42}{257} = 0.165$, or 16.5 per cent.

The area of the actual card taken from an engine is about .6 that of the theoretical card from which the efficiencies which follow are calculated.

The probable actual efficiency based on brake horsepower becomes,

$$\text{Eff. on brake H. P.} = .85 \times .60 \left(1 - \frac{1}{r^{k-1}}\right)$$

A table of efficiencies for different ratios of compression calculated with $k = 1.35$ shows how the efficiency increases with the compression.

r = compression ratio	$\text{Eff.} = 1 - \frac{1}{r^{.35}}$	actnal eff. on I. H. P.
3	.32	.19
4	.38	.23
5	.43	.25
6	.47	.28
7	.50	.30

Illustration. An engine with compression ratio of 5 and with $k = 1.35$ has 80 B. t. u. supplied to it per minute, what number of ft. lbs. per minute may be taken off at the brake?

Ft. lbs. per min. at brake $= 80 \times 778 \times .85 \times .60 \times .43 = 13{,}630$.

The total suction displacement required for the development of a given power is affected by the heating value of the gas; the amount of air needed to burn the gas and by the proportion of working strokes to total strokes.

Let $H =$ the heating value of one cu. foot of gas at 62° and at atmospheric pressure.

Let $a =$ cu. ft. of air at same pressure and temperature required to burn one cu. ft. of gas.

If 15 per cent excess air be allowed, then the total volume of the air, the excess air, and the gas $= (1.15a + 1)$ per cu. ft. of gas.

The heating value per cu. ft. of the mixture $= \dfrac{H}{(1.15a+1)}$.

This mixture receives heat from the cylinder walls as it gets into the cylinder and at the same time its pressure is reduced due to the resistance offered by the suction valves. Either of these would cause the volume to increase.

Actually the volume is increased from 15 to 25 per cent, due to the above. If 23 per cent be taken as a fair value, then the volume becomes $1.23\,(1.15a + 1)$ and the heating value per cu. ft. of gas mixture in the cylinder becomes $\dfrac{H}{1.41a + 1.23}$.

As has been shown in another part of this book, a horsepower per minute requires that 42.42 heat units be transformed into work in one minute.

The B. t. u. which must be supplied to an engine per minute in order that the engine develop a brake H. P. with a mechanical efficiency of 85 per cent is

$$\text{B. t. u. per min.} = \frac{42.42}{.85 \times \text{actual efficiency}}, \quad \text{(a)}$$

$$\text{Piston displacement} = \frac{57.4 \times 4 \times 100}{\frac{120 \times .30}{1.41 + 1.23}} = 1684$$

$$\frac{1684}{800} = 2.10 \text{ sq. ft. area or about } 19'' \text{ dia.}$$

The length of stroke is generally from one and one-third to twice the diameter, the longer stroke being used with the higher ratio of compressions.

The terminal pressures at compression for different fuels as given by Levin are :—

Natural gas	100 to 150 lbs. gage
Coal gas or city gas	80 to 140 " "
Coke oven gas	100 to 140 " "
Bituminous Producer gas	140 to 160 " "
Anthracite " "	150 to 170 " "
Blast furnace gas	150 to 190 " "
Carbureted gasoline	60 to 90 " "
Kerosene	50 to 80 " "
Alcohol	120 to 180 " "

A calculation similar to that given for the producer gas engine may be made for a gasoline engine.

Gasoline is represented approximately by C_6H_{14}. As the atomic weight of carbon is 12 and that of hydrogen 1, it is evident that in 86 lbs. of gasoline there are 72 lbs. of carbon and 14 lbs. of hydrogen. To burn a pound of carbon 11.5 lbs. of air are needed and similarly 34.5 lbs. are needed to burn one pound of hydrogen.

$$\frac{72 \times 11.5 + 14 \times 34.5}{86} = 15.2 \text{ lbs. air per lb. gasoline.}$$

The volume of one pound of air at 62° is about 13 cu. ft.

13 × 15.2 = 198 cu. ft. air per pound gasoline.

The volume of one pound of gasoline vapor at 62° is 4.2 cu. ft.

$$\text{Piston displacement} = \frac{57.4 \times 4 \times 20}{\frac{4400 \times .24}{63.5}} = 276 \text{ cu. ft.}$$

With a piston speed of 600 ft. per minute this would require an area of piston of $\frac{276}{600} = .46$ sq. ft. or approximately 66 sq. inches. This area corresponds to a diameter of about 9.2″. If the stroke be taken as 15″ the speed becomes $\frac{600}{\frac{2 \times 15}{12}} = 240$ R. P. M.

The Diesel Engine. The Diesel Engine gives higher efficiencies than gas engines.

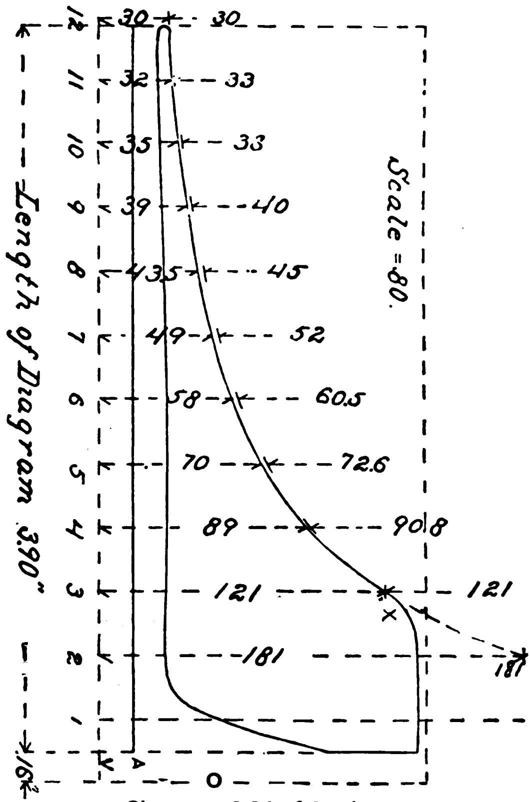
The engine is generally made single acting, having one working stroke in four.

The fuel charge is injected gradually during the first part of the working stroke, being forced into the cylinder by air under 700 or 800 pounds pressure, compressed by a small compressor attached to the engine.

Air drawn in on the suction stroke is compressed to 600 or 700 pounds and its temperature, due to the heat developed by the compression, becomes sufficient to ignite the fuel, which burns gradually as it enters.

The engine is started by means of compressed air.

pounds of water divided by 0.036 or multiplied by 27.8 will give the number of cubic inches. If accurate scales for weighing the water are not at hand, it can be carefully



Clearance 0.04 of Stroke

FIG. 24

measured in a quart or pint measure and the number of cubic inches found directly. A gallon contains 231 cubic inches, a quart 57.75, and a pint 28.875 cubic inches.

The volume of the clearance will rarely be the same at the two ends of the cylinder, therefore the number of cubic inches in the clearance at each end must be divided by the net area of the piston at its own end: that is, the number of cubic inches in the clearance at the end nearest the crank must be divided by the number of square inches in the cross-section of the cylinder, less the number of square inches in the cross-section of the piston-rod; and the number of cubic inches in the clearance at the end farthest from the crank must be divided by the number of square inches in the cross-section of the cylinder. The quotient in each case will be the length of clearance at the respective ends of the cylinder, expressed in inches. In this instance (Fig. 24) it is found to be 0.16 of an inch.

It is convenient to have the length of the clearance expressed as a fraction of the piston displacement or stroke of the piston. To obtain this fraction, divide the number of cubic inches in volume of clearance by the number of cubic inches in the volume swept through by the piston at each end separately, taking care to allow for the volume occupied at one end by the piston-rod, and the quotient will be the decimal fraction that the clearance space is of the volume swept through by the piston.

Fig. 24 illustrates a good method for locating points in the hyperbola through which the curve may be drawn.

First, draw the zero line V, at the proper distance, viz., $14\frac{7}{10}$ pounds by the scale, below and parallel with the atmospheric line; next, draw the clearance line O, as computed, 0.16 of an inch from the end of the diagram; next, locate the point of cut-off X, and draw the perpendicular line numbered 3 through it; next, divide the space between this line and the clearance line into three equal parts; then, taking one of these parts for a measure, point off, on the vacuum line, *equal* spaces toward the left hand until one or more falls beyond the end of the diagram as shown, and erect perpendicular lines from each point. These lines are called ordinates and numbered consecutively 1, 2, 3, 4, etc., beginning with the one next to the clearance line. It is well to bear in mind the fact that vertical distance on a diagram represents *pressure* and horizontal distance *volume.*

In this case we have started the hyperbola from the point of cut-off X, and its course is indicated by the short lines drawn through the ordinates a little above the actual curve, with their calculated pressures written above; the actual pressures of the expansion curve are written below it. The properties of the hyperbola are such, that if the distance of the point X from the clearance line O be multiplied by the height of X from the zero line V, the height of any other point in the curve can be found by dividing this product by its distance from the clearance line. And on this principle we proceed to locate points on the ordinates through which our hyperbola will run.

We find the pressure at the point of cut-off to be 121 pounds with a volume which we call 3, because there are three spaces or volumes between it and the clearance line. Then, $121 \times 3 = 363$, which is our dividend for all the other volumes. Therefore, the height at which the hyperbola will cut ordinate 4 will be determined by dividing 363 by 4, which is 90.8 (it is unnecessary to carry the division beyond one decimal); and at ordinate 5, 72.6; at ordinate 6, 60.5; and so on to the end. At ordinate 12 we find that the hyperbolic and the actual curves practically coincide. In like manner we may extend the curve to the right: $363 \div 2 = 181$ pounds, which would be the pressure if the steam were compressed up to 2 volumes. If desired, the hyperbolic curve can be started just before the point of release and projected in the opposite direction by the same method.

Instead of using figures which stand for pressures or volumes of steam to locate the hyperbola, as in this instance, the distances from the base and perpendicular lines of any point may be expressed in inches and decimal parts, with the same result.

A quick way to draw the hyperbola is to take the whole distance between ordinate 3 and the clearance line, (Fig. 24), as a measure, and set off equal spaces to the left as before directed. Then we would have but four ordinates and would number them as follows: 1 at 3d, 2 at 6th, 3 at 9th,

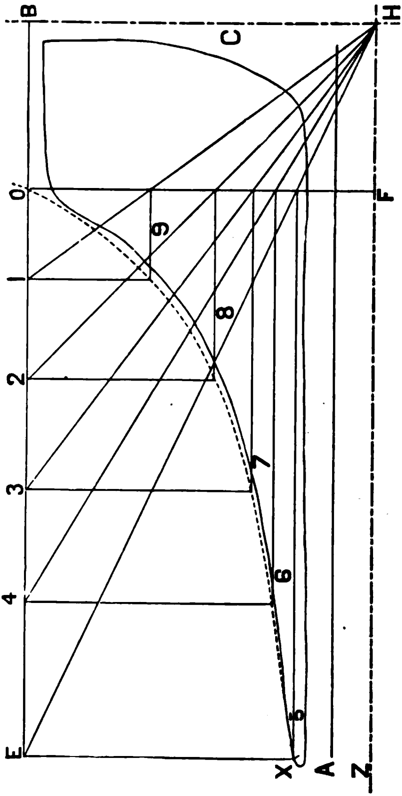


FIG. 25

and 4 at 12th. At 1 we would have a pressure of 121 pounds; at 2, 121 pounds $\div 2 = 60.5$; at 3, 121 pounds $\div 3 = 40.3$; and at 4, 121 pounds $\div 4 = 30.25$.

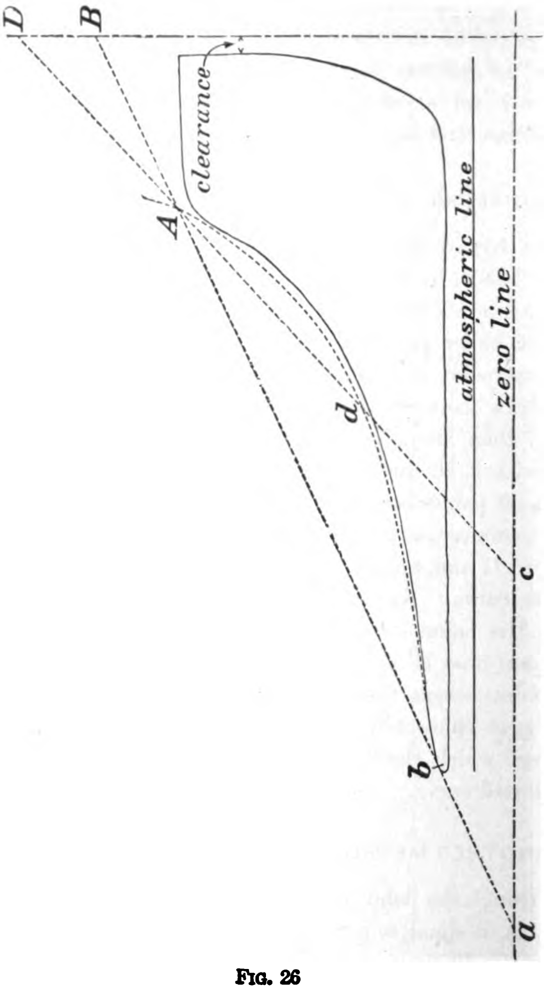


Fig. 26

PART II

increase the steam lap on the head end, delaying admission and hastening cut-off on this end; (2) to decrease the steam lap on the crank end, hastening admission and delaying cut-off; (3) to decrease the exhaust lap on the head end, hastening release and delaying compression; and (4) to increase the exhaust lap on the crank end, delaying release and hastening compression.

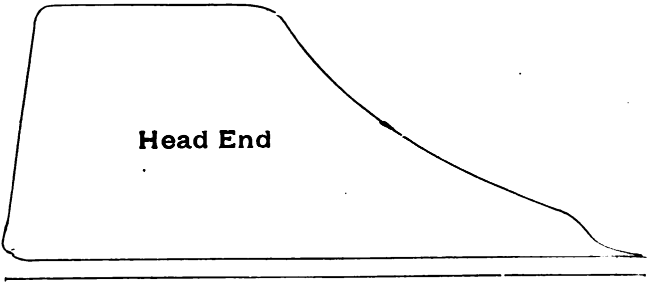


FIG. 31

If the valve spindle is too long and at the same time the eccentric has too little angular advance, or is set too far back, the cards from the engine will be similar to Figs. 33 and 34.

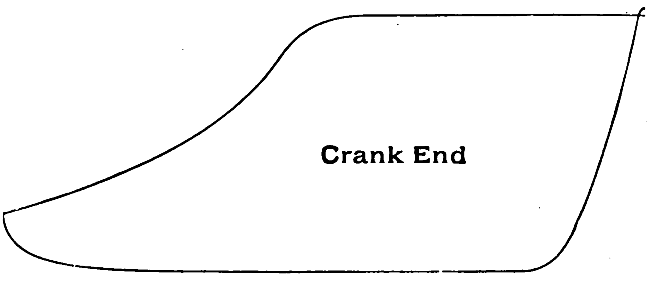


FIG. 32

If the eccentric is set too far ahead and the spindle is too long, the appearance of the cards may be predicted by

On account of this fact the indicator card taken in the ordinary way is of little value in investigating any peculiarities which may be noticed at or near the ends of the stroke.

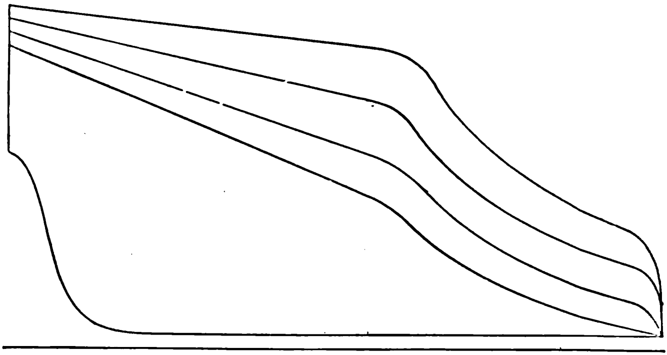


FIG. 36

If, however, the drum motion be taken from the eccentric, which is ordinarily a little over 90° ahead of the crank, the compression and admission lines and the line at release will be spread out at the center of the diagram, while the expansion and exhaust lines will be shortened and appear at the ends of the cards.

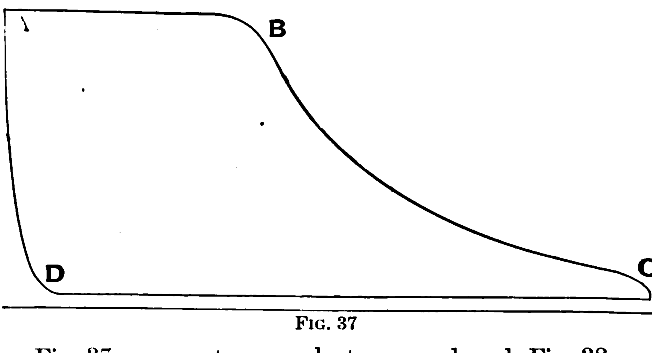


FIG. 37

Fig. 37 represents a good steam card, and Fig. 38 an eccentric card.

the head end be combined with the back pressure line from the card of the other end, we get a diagram known as the Stroke Card, sometimes called the True Card, which, by its distance between lines, gives the effective pressure per square inch on the piston at any point. Such a diagram is shown by Fig. 40 for the Air Brake Card. It will be noted that the greatest effective push on the piston comes at the *right*-hand end, not the left.

If the two ends of the steam piston have not the same area, the pressures from one indicator card may be multiplied by the ratio of the two areas, before plotting, in order to reduce to the same basis for comparison.

5. Air Compressor Cards

Figs. 41 and 42 are respectively from an air compressor and from the air end of a certain type of air pump.

The irregularities in the delivery line of Fig. 41 are due to the vibrations of the delivery valve. The slight drop at

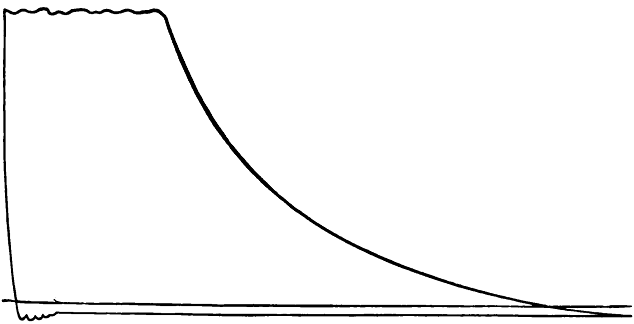


FIG. 41

the beginning of air intake results from resistance to opening offered by the suction valves.

The air pump pumps a mixture of water and air. On account of the large clearance the curves are much less steep than in the case of the air compressors.

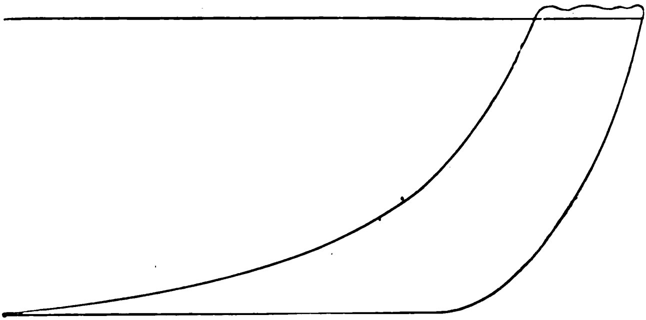


FIG. 42

6. Gas Engine Cards

Figs. 43 and 44 are from an Otto Gas Engine. Fig. 44 has the firing delayed till the end of the stroke, and shows plainly the effect of not having a "lead" on the firing spark.

Engines working on the Otto cycle can have only *one* working stroke in four, and as many of these engines work on the "hit and miss" principle, in figuring the horsepower from the cards it is necessary to note the actual number of explosions per minute.

Starting at exhaust on the right-hand end, the piston

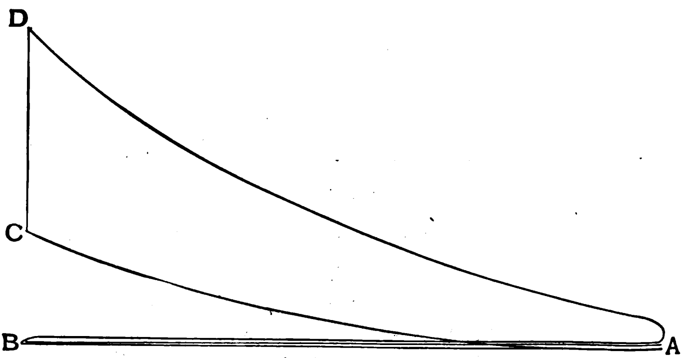


FIG. 43

moves to the left, along AB, and drives out the burned gases through the exhaust valve; next the exhaust valve closes, and as the piston moves to the right, gas and air are drawn in, along BA, the mixture being regulated by the opening of the gas inlet, so as to get the proper ratio of gas to air. On the third stroke this mixture is compressed, along AC, and just before the piston reaches the end of the stroke the mixture is fired; the hot gases expand during the fourth stroke, DA.

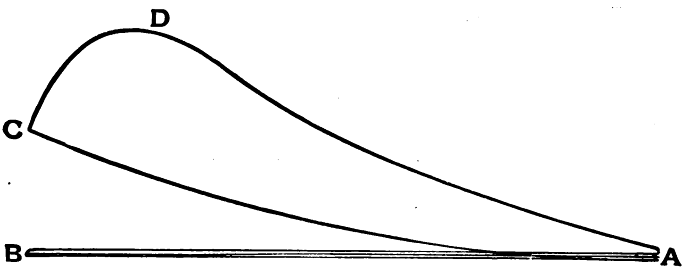


FIG. 44

If the engine is up to speed and gas is not admitted by the governor, the cylinder is filled with air on the filling stroke; this air is compressed on the third stroke and expands back again along nearly the same line on the fourth stroke.

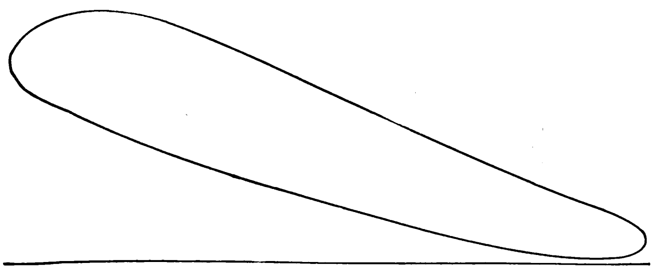


FIG. 45

9. Rotative Effect

The total pressure on the piston-rod at any point may be found by multiplying the pressure measured across the stroke card at that point by the area of the piston. Suppose the total pressure on the piston-rod at the position shown in Fig. 48 is 1,000 pounds. Draw the line P to represent this pressure at some assumed scale, say, for example, one inch representing 500 pounds. Then $P = 2$ inches long.

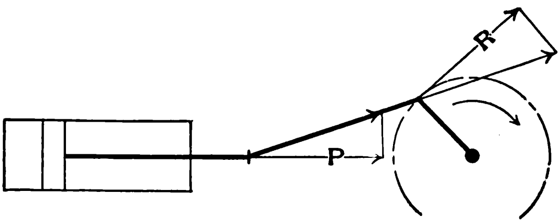


FIG. 48

This force P produces a push along the connecting rod and a downward thrust on the guides. This push along the rod and this thrust on the guides may be found by making a triangle of forces as shown. The length of the different sides of the triangle multiplied by 500 gives these forces in pounds.

The thrust in the connecting rod is seen to be greater than P. This thrust at the cross-head end of the connecting rod is carried to the crank pin where it may be separated into two forces, one force R at right angles to the crank tending to produce rotation, and another force along the crank making a compression in the crank casting.

If, now, values of R are calculated for a sufficient number of points and plotted on a line which represents the development of the crank pin circle, a diagram of rotative effect like Fig. 49 is obtained. The upper half comes from one stroke card and the other half from the other stroke card.

The mean rotative effect is shown by the dot and dash lines.

The amount of energy stored and restored by the fly wheel during a stroke is represented by the area within the curve outside of the dash and dot line.

In this discussion it has been assumed that the entire pressure on the piston was available for producing rotative effect. This is not the case, however. A certain amount of this pressure is used up at the beginning of the stroke in

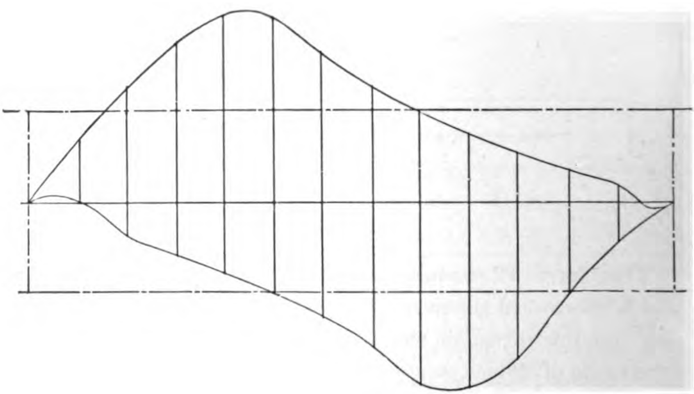


FIG. 49

accelerating the piston, piston-rod, cross-head, and a part of the connecting rod. As these are brought back to rest at the end of the stroke, as much energy is recovered as was lost at the beginning. The dotted line on Fig. 46 shows how the upper line should be changed to make allowance for this, in the case of a low speed Corliss engine. On a high speed engine this is a much larger factor, as may be seen in Fig. 47. In this figure the effective pressure per square inch available for producing rotative effect is shown by the area cross-hatched.

The line AA represents the atmospheric line, the line VV that corresponding to an absolute vacuum, and the line

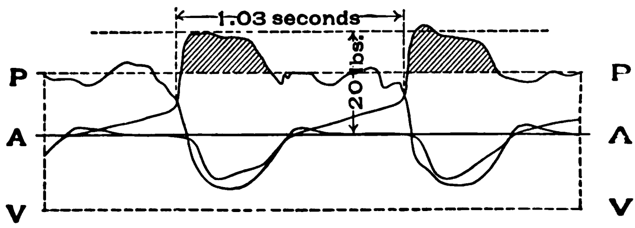


FIG. 57

PP the pressure in pounds due to the hydrostatic head the pump was delivering against. The shaded portions represent deliveries of water.

PART III

To measure pressure below the atmosphere, a vacuum gage or a glass U tube filled with mercury may be used. It is customary to quote the vacuum in inches of mercury instead of in pounds. If the barometer stood 29.9 inches a vacuum of 26 inches would mean that there was an absolute pressure of 3.9 inches, or (3.9×0.491) pounds.

All pressures as given in plots or in tables of the properties of saturated steam are absolute pressures.

Specific pressure and *specific volume.* Specific pressure is absolute pressure on the square foot. Specific volume is the volume of one pound.

The specific volume of water is $\frac{1}{62.4} = 0.016$ cubic feet, 62.4 pounds being the weight of a cubic foot of water at 62°.

The volume of a pound of dry steam, represented by the letter s, varies with the pressure.

The *British thermal unit*, B. t. u., is the amount of heat necessary to raise one pound of water from 62° F. to 63° F.

The *heat of the liquid* is the amount of heat expressed in B. t. u. necessary to raise one pound of water from 32° to the temperature desired.

If the specific heat of water was unity throughout the entire range of temperature, the heat of the liquid would be 32 less than the temperature.

The specific heat is slightly above unity at some temperatures, and slightly below unity at other temperatures.

The heat of the liquid is represented by the letter q.

Relation of pressure and temperature of saturated steam:

Regnault found that the temperature of steam depended upon the pressure; that the temperature of the steam, if it was not superheated, was exactly the same as that of the water in contact with it.

Particles of water may float in steam the same as fog floats in the air. This does not affect the temperature of the steam.

The following example will illustrate the method of calculating the heat equivalent of the external work.

Suppose one pound of water at 32° to be placed in the bottom of a vertical cylinder of one square foot piston area. Let the piston be weighted so that together with the atmospheric pressure there is a load of 100 pounds per square inch on the piston.

If heat is added to the water, vaporization will not begin till a temperature of 327.86° F. is reached. As vaporization takes place the piston rises in the cylinder.

When all the water has been made into steam, the piston will stand 4.432 feet above the bottom of the cylinder.

The pound of water occupied 0.016 of a cubic foot, and as the cylinder is one square foot in sectional area, the piston must have moved up a distance of $4.432 - 0.016 = 4.416$ feet.

The external work done is $100 \times 144 \times 4.416$ foot-pounds. Dividing this by 778, the mechanical equivalent of one heat unit, gives as a result 81.9 B. t. u.

The heat equivalent of the external work is represented by Apu. A is the heat equivalent of one foot-pound and is equal to $\frac{1}{778}$; p is the absolute pressure on the square foot; u equals the change in volume in passing from water to steam.

Subtracting the heat equivalent of the external work from the total latent heat gives the heat equivalent of the internal work. This is represented by the Greek letter ρ (called rho).

In this case, at this pressure, ρ amounts to

$$887.6 - 81.9 = 805.7 \text{ B. t. u.}$$

This heat equivalent of the internal work increases as the volume of a pound of steam increases.

As the volume occupied by a pound of steam at very low

Total heat of a pound of mixture of steam and water above 32°:

Before any water can be made into steam at a given pressure, the whole of the water must first be heated to the temperature corresponding to the pressure. Then if x parts by weight are made into steam, the heat $x\,r$ must be added, making the total heat to be added $q + x\,r$.

Total heat of a pound of a mixture of steam and water above any given temperature.

First find the heat of the pound of mixture above 32° equal to $q + x\,r$, then subtract the heat of the liquid at the given temperature.

Superheated steam is steam of a higher temperature than that corresponding to saturated steam of the same pressure.

The difference of temperature is the number of degrees of superheating.

To tell whether or not steam is superheated, a thermometer, a steam gage, and a table or plot giving the temperatures of saturated steam are needed.

Knoblauch, Linde, and Klebe, from recent experiments made in Munich, have determined the following equation for superheated steam:

$$pv = 85.85\,T - p\,(1 + 0.00000976\,p)\left\{\frac{150{,}300{,}000}{T^3} - 0.08328\right\}$$

A much more simple equation giving results agreeing with the above within 0.8 of one per cent is

$$pv = 85.85\,T - 0.256\,p$$

where p is the absolute pressure in pounds on the square foot, and v is the volume of one pound. T is the absolute temperature of the superheated steam in degrees F.; this is found by adding 459.5 to the temperature of the steam as given by the thermometer.

Having the temperature and pressure as known terms, At the back of the book there are two charts, one giving the different values of t, q, r, λ, $A\,p\,u$, ρ, and s for pressures from 0 to 10 pounds absolute and the other giving values of the same terms from 10 pounds absolute to 250 pounds absolute.

These curves, which are drawn to represent the values given in Peabody's Steam Tables, the tables in general use by engineers, will serve to give values with a moderate degree of accuracy. For accurate work such values should be taken from some reliable steam table which gives these values for each degree difference of temperature or for each pound increase in pressure.

Tables which give values for intervals of 5 pounds, and where values for intermediate points must be obtained by interpolation, are fairly accurate at high pressures, but unreliable at low pressures on account of the error due to interpolation. Even tables reading to one pound are unreliable at low pressures for the same reason.

If either Peabody's Steam Tables or the Steam Tables by Marks and Davis are used, all low pressure values should be taken from the temperature table which gives values for each degree from 32°. The pressure corresponding to each temperature is given also. As there are seventy sets of values for pressures between 0 and one pound absolute, sufficient accuracy can be obtained.

The charts show that the total heat and the heat equivalent of the external work change but little; that the temperature, the specific volume of steam, and the heat equivalent of the internal work (internal latent heat), change rapidly at low pressures and slowly at high pressures.

In order to show the application of the discussion in the preceding pages a few examples will be solved.

Problem (1). How much heat will it take to make 3 pounds of water at 60° F. into wet steam at 150 pounds absolute pressure? The steam is primed 2 per cent.

If the wet steam contains 2 per cent moisture there must be 98 per cent dry steam.

In solving problems in steam where use is made of the values λ, q, r, s, ρ, $A\,p\,u$, etc., it must be remembered that these values are for one pound. It is advisable to work all problems as if the actual weight were one pound and to finally multiply the result by the actual weight.

The heat which must be added to a pound of water at 32° in order to make this into wet steam at this pressure is $q + 0.98r$ where q and r are the values of the heat of the liquid, and the total latent heat at 150 pounds absolute, respectively. The water was originally at 60°. The heat of the liquid of water at 60° must be subtracted from this to give the amount to be added per pound.

$$(q_{\text{150 lbs. abs.}} + 0.98r_{\text{150 lbs. abs.}} - q_{60^\circ\text{ F.}}) \times 3$$

From the chart and the table of heat of the liquid these values are

$$[330. + (0.98 \times 863.0) - 28.1] \times 3 = 3443.$$

Problem (2). What volume will the 3 pounds of wet steam occupy?

From the chart it appears that the volume of one pound of dry steam at 150 pounds absolute pressure is 3.0 cubic feet.

The volume of one pound of mixture or of wet steam is

$$v = 0.98\ (3.0 - 0.016) + 0.016 = 2.940$$

The 3 pounds will occupy a

$$\text{volume} = 3\ v = 8.820 \text{ cubic feet.}$$

Problem (3). An engine is supplied with steam at 144 pounds absolute pressure. The steam contains one per cent of moisture.

The engine uses 2,800 pounds of steam per hour (all through the cylinders, there being no jackets).

The indicated horse-power is 200. The temperature of the exhaust at the condenser is 126° F.

The air pump discharges the condensed steam back to the boilers through a primary heater on the exhaust pipe.

The temperature of the feed-water entering the boiler is 100° F.

Each pound of coal burned under the boilers gives up 14,500 B. t. u., 9,900 of which are taken up by the boiler and utilized in making steam.

What is the number of pounds of coal per horse-power as indicated?

What is the thermal unit consumption of the engine per horse-power per minute?

$$(q_{144 \text{ lbs. abs.}} + 0.99r_{144 \text{ lbs. abs.}} - q_{100^\circ \text{ F.}})\frac{2800}{200}$$

gives the number of thermal units supplied by the boiler per I. H. P.

Substituting the values from the chart or tables

$$[326.7 + (0.99 \times 865.6) - 68.0] \times 14 = 15619.$$

Dividing this by 9900 gives the coal per I. H. P. of the engine alone as

$$\frac{15619}{9900} = 1.58 \text{ pounds.}$$

In calculating the thermal unit consumption of the engine it is customary to assume that the condensed steam could be returned by the air-pump to the boiler at the same temperature as that of the exhaust steam.

The thermal unit consumption per I. H. P. per minute is

$$(q_{144 \text{ lbs. abs.}} + 0.99r_{144 \text{ lbs. abs.}} - q_{126^\circ \text{ F.}})\frac{2800}{200 \times 60} =$$

$$(326.7 + 857.0 - 94.0)\frac{7}{30} = 254.26$$

Problem (4). Suppose that the steam supplied to the engine was of 144 pounds absolute pressure and 400° F. in temperature; that the steam consumption per hour was

2,600 pounds, and that the I. H. P. and other conditions were the same, what would be the B. t. u. per I. H. P. per minute?

$$\left\{ \lambda_{144 \text{ lbs.}} + 0.587\ (400.0 - 355.29) - q_{126^\circ \text{ F.}} \right\} \frac{2600}{200 \times 60} =$$

$$\left\{ 1192.3 + 26.24 - 94.0 \right\} \frac{13}{60} = 243.65$$

The specific heat of superheated steam is taken from the preceding table as 0.587.

Should the engine be provided with steam jackets the weight of jacket steam per H. P. per minute times the B. t. u. given up by the condensation of one pound is to be added to the B. t. u. per H. P. per minute through the cylinders.

Problem (5). What is the thermal efficiency of the engines in (3) and (4) as previously explained?

$$\frac{33000}{778} = 42.42 \text{ B. t. u.}$$

$$\frac{42.42}{254.26} = 0.167 \qquad \frac{42.42}{243.65} = 0.174$$

Carnot engine. It is found in the preceding problem that the thermal efficiency of the engines is low.

One might be led to think that the steam engine was not as economical as it might be made to be. This is not the case, however. Many of our best engines when compared in thermal unit consumption with that of the theoretically perfect engine, working between the same pressures and temperatures, give 70 per cent comparative efficiency.

The theoretically perfect engine, called the Carnot engine, is not necessarily one with 100 per cent thermal efficiency, but one in which there are no losses from friction, conduction, radiation, etc. It is one in which all the heat supplied is accounted for by the sum of the heat withdrawn, and the heat transformed into work.

Evidently an engine to have 100 per cent thermal efficiency must transform all the heat it receives into work, and have none to throw away or be withdrawn. It can be shown that the efficiency of such a theoretically perfect engine is given by dividing the difference of temperature worked through in the cycle, by the absolute temperature at which heat was supplied to the engine.

A Carnot engine working through the same temperature intervals as those given in Problem (3) would have a thermal efficiency:

$$\frac{355.29 - 126.}{355.29 + 459.5} = 0.281$$

Comparing the actual with that of the Carnot:

$$\frac{0.167}{0.281} = 0.59$$

The thermal unit consumption per H. P. per minute for this case is with the Carnot engine:

$$\frac{42.42}{0.281} = 151.$$

The B. t. u. consumed per H. P. per minute by the actual engine and by the theoretical bear the same ratio as that of the thermal efficiencies:

$$\frac{151.}{254.26} = 0.59$$

The only correct way to quote the performance of an engine is by its thermal efficiency or by its B. t. u. consumption per I. H. P. per minute.

The weight of steam per H. P. per hour does not mean anything unless one knows the heat in that steam as supplied to the engine and the temperature and pressure of the exhaust.

One engine may develop a H. P. on 9 pounds of steam, the steam being highly superheated. Another engine with perhaps a higher thermal efficiency than the first may use 12 pounds per H. P.

If two engines work under exactly the same conditions as to boiler pressure, steam, and vacuum, then a comparison may be made of the steam consumptions per H. P. per hour.

In the Carnot engine it is supposed that the same charge of working substance, air, steam, or whatever it may be, is alternately heated and cooled in the cylinder. The actual engine has a new supply of working substance brought into the cylinder on each power stroke.

It would seem better to compare the actual engine with a perfect engine which was similarly supplied. By a perfect engine is meant one in which there is no friction, no radiation, and no absorption or conduction of heat by the cylinder walls; one in which the expansion drops the pressure down to that of the back pressure. Such an engine is called a *non-conducting engine* or an engine working on the *Rankine cycle.*

Non-conducting engine. The amount of heat which must be added to a pound of feed water at the boiler to make it into a pound of steam of the condition as supplied to the engine, assuming that the feed water enters the boiler at the temperature of the engine exhaust, is $q_1 + x_1 r_1 - q_2$; where q_1 is the heat of the liquid, r_1 the total latent heat or heat of vaporization at boiler pressure, and x_1 is the quality of the steam made by the boiler. If there is one per cent priming in the steam then $x_1 = 0.99$. If the quality of the steam after an adiabatic expansion from cut-off down to the back pressure is x_2 the heat to be abstracted during the exhaust is $x_2 r_2$ where r_2 is the latent heat of steam at the pressure corresponding to the exhaust.

The efficiency of any engine is the difference between the heat supplied and the heat exhausted divided by the heat supplied. In this case the efficiency becomes

$$\frac{q_1 + x_1 r_1 - q_2 - x_2 r_2}{q_1 + x_1 r_1 - q_2} = 1 - \frac{x_2 r_2}{q_1 + x_1 r_1 - q_2}$$

The adiabatic line was discussed in Chapter I, page 14. It was shown that for a reversible line the entropy remained constant. This fact is made use of in calculating x_2.

$$x_2 = \left(2.3026 \log. \frac{T_1}{T_2} + \frac{x_1 r_1}{T_1}\right)\frac{T_2}{r_2}$$

where T_1 is the absolute temperature corresponding to the temperature of the steam and T_2 that of the feed water. The derivation of the formula will be found under the discussion of the temperature entropy diagram.

Problem (6). What would be the thermal efficiency of a non-conducting engine working as in Problem (3) with steam primed one per cent at 144 pounds absolute pressure and with exhaust at 126° F.? What would be the number of pounds of steam per H. P. per hour? What would be the B. t. u. consumption per H. P. per minute?

$$x_2 = \left(2.3026 \log. \frac{355.29+459.5}{126.+459.5} + \frac{0.99 \times 865.6}{355.29+459.5}\right) \times \frac{126+459.5}{1021}$$

$$x_2 = 0.79$$

$$\text{The efficiency} = 1 - \frac{0.79 \times 1021.}{327.6 + (0.99 \times 865.8) - 94} = 0.26,$$

or 26 per cent.

The foot-pounds of work done by the engine per pound of wet steam supplied is $778\,(q_1 + x_1 r_1 - q_2 - x_2 r_2)$; this being the difference between the heat supplied and the heat exhausted per pound or the heat per pound transformed into work multiplied by 778. The number of foot-pounds corresponding to a H. P. for one hour is $33{,}000 \times 60$. The steam per H. P. per hour is then

$$\frac{33000 \times 60}{778\,[327.6 + (0.99 \times 865.8) - 94 - (0.79 \times 1021)]} = 8.96$$

The B. t. u. consumption per H. P. per minute is $\frac{33000}{778} \div 0.26 = 163.2$. The engine in Problem (3) showed an actual efficiency of 0.167. This is $\frac{0.167}{0.26} = 0.64$, or 64 per cent of the values so figured are plotted, a line marked liquid line will be obtained. From the plot the entropy of the liquid at any temperature may be read directly :

at 697.5° absolute the entropy of the liquid is 0.35
at 800° " " " " " " " 0.49
at 550° " " " " " " " 0.11

To make steam at a given pressure from water at 32° the water is first heated up to the temperature corresponding to the pressure by the addition of the heat of the liquid q. The entropy increases by an amount which may be read from the liquid line. Next the heat of vaporization r is added and the water gradually passes into steam at the same temperature. The increase in entropy due to the addition of the heat of vaporization is $\frac{r}{T}$. If the value of $\frac{r}{T}$ be figured for each pressure and laid off to the right of the liquid line, the dry steam line is obtained. If instead of vaporizing the entire pound of water, only 80 per cent of it had been vaporized, the heat added at constant temperature would have been 0.80 r and the increase in entropy due to vaporization $\frac{0.80\ r}{T}$ or 80 per cent of the value between the liquid line and the dry steam line. The horizontal distance between the liquid line and the dry steam line has been divided into 10 parts marked $x = 0.10$, $x = 0.20$, etc., and these parts each subdivided into 5 additional parts.

Illustration. The entropy of a pound of dry steam at 800° absolute temperature is read from the chart as 1.58. The entropy of a pound of dry steam at 550° absolute is 2.02. The entropy of a pound of mixture of steam and water which is 80 per cent steam by weight at 550° absolute is 1.64.

Between the liquid line and the dry steam line there are four curves which are used in finding the absolute temperature corresponding to any absolute pressure.

The absolute temperature of steam

at 2 pounds	absolute	pressure	appears	to	be	587.5°
at 4 pounds	"	"	"	"	"	613.°
at 10 pounds	"	"	"	"	"	654.°
at 220 pounds	"	"	"	"	"	850.°

The entropy of a pound of mixture of steam and water at 50 pounds absolute pressure, the mixture being 36 per cent steam by weight, is read on the chart as 0.855.

Beyond the dry steam line are lines marked 250 pounds, 200 pounds, 150 pounds, etc., leading upward from the dry steam line. These lines give the entropy of superheated steam.

Take for illustration 150 pounds absolute. This line starts from the dry steam line at 818°, the absolute temperature of saturated steam at this pressure; as heat is added to the dry steam and the pressure kept constant, the temperature increases and the entropy increases. The temperature increases more rapidly than the entropy. As an illustration, the entropy of a pound of steam at 150 pounds absolute pressure, superheated 100° F., is 1.632.

The temperature of saturated steam at 150 pounds is 818° absolute.

The entropy of the liquid at 818° absolute is 0.513.

The entropy of a pound of dry steam at 818° is 1.565.

The increase in entropy due to the 100° superheat is .067.

During a reversible adiabatic expansion the entropy remains constant. This plot is a great help in solving for the final condition of a mixture after an adiabatic expansion. In Problem (6) on the non-conducting engine, steam at 144 pounds absolute pressure with one per cent priming was expanded adiabatically to 126° F. or 585.5° absolute. It was found by a numerical calculation that $x_2 = 0.79$. This value may be found at once by the chart. Follow along at the temperature level corresponding to 144 pounds until $x =$

Measurement of Dry Steam by the Flow through an Orifice

An empirical formula known as Napier's or as Rankine's gives very accurate results.

The orifice should have a rounded edge at entrance.

W = the weight of steam flowing per second.

P_1 = the absolute pressure in pounds per square inch on the entrance side.

P_2 = the absolute pressure in pounds per square inch on the exit side.

A = area of the orifice in square inches.

Where P_1 is equal to or greater than $\frac{5}{3}P_2$

$$W = A\frac{P_1}{70}$$

Where P_1 is less than $\frac{5}{3}P_2$

$$W = A\frac{P_2}{42}\left\{\frac{3(P_1 - P_2)}{2P_2}\right\}^{\frac{1}{2}} = 0.0292\,A\,(P_1P_2 - P_2^2)^{\frac{1}{2}}$$

As P_2 approaches P_1 more steam goes through the orifice than this formula gives.

This second formula is not to be recommended as accurate within 8 per cent when $\frac{P_2}{P_1}$ bears the ratio 0.85 or higher.

Design of a Turbine Nozzle for Complete Expansion

By gradually increasing the diameter of a nozzle beyond the throat or smallest section, the velocity of the steam in the nozzle may be increased as the pressure drops, till at the end of the nozzle a velocity of from 3,600 to 4,000 feet per second may be realized if the back pressure is low.

By complete expansion is meant a drop in pressure in the nozzle from the highest to the lowest pressure; that is, there is no drop after leaving the nozzle.

The method commonly used in calculating a nozzle is given in the following pages, but the derivation of the formulæ used is omitted.

Let the subscript i denote conditions and values of the high pressure steam at entrance to the nozzle; the subscript t similar conditions at the throat, and the subscript e at exit.

$H_i = q_i + x_i\, r_i$ for saturated steam.

$H_i = q_i + r_i + C_p$ (degrees of superheat) for superheated steam.

$H_t = q_t + x_t\, r_t$ for saturated steam.

$H_t = q_t + r_t + C_p$ (degrees superheat) for superheated steam.

$H_e = q_e + x_e\, r_e$.

The values of x_t and x_e are read from the temperature entropy plot, assuming adiabatic expansion from the condition x_i. If the steam is superheated to start with, the chart is used in the same way after locating the starting position. Call V_t the velocity at the throat in feet per second and V_e the velocity at the exit.

$$V_t = 224\sqrt{(H_i - H_t)}$$
$$V_e = 224\sqrt{0.85(H_i - H_e)}$$

The area of the throat and the exit sections are calculated thus: The volume of one pound of steam at the throat is

$$v_t = x_t\,(s_t - 0.016) + 0.016$$

where s_t is the volume of one pound of dry steam at the throat pressure. Should the steam be superheated at the throat, the volume of a pound would be calculated by the formula given in the earlier part of this chapter.

$$\frac{v_t \times \text{weight per second}}{V_t} = \text{area of throat in square feet.}$$

In finding V_e, 85 per cent of $H_i - H_e$ was used because a friction loss amounting to 15 per cent of $H_i - H_e$ was assumed to occur in the nozzle. The friction loss up to the throat is small and is not considered in this calculation. A small allowance is sometimes made for it, however.

The effect of this friction and of the conduction of heat by the nozzle is to make the steam more nearly dry at exit than it would have been after an expansion at constant entropy. The increased dryness may be found by $\dfrac{0.15\ (H_i - H_e)}{r_e}$

This added to x_e gives x_f the final condition leaving the nozzle.

$$v_e = x_f\ (s_e - 0.016) + 0.016$$

$$\frac{v_e \times \text{weight per second}}{V_e} = \text{area of exit in square feet.}$$

Problem (8). A de Laval turbine of 350 H. P. rated capacity is supplied with seven nozzles. The pressure of steam at entrance is 200 pounds absolute, the steam being superheated 35°. The exit pressure is 2 pounds absolute. Assume the friction loss in the nozzle to be 15 per cent. Assume also that 65 per cent of the kinetic energy of the steam is utilized by the wheel. Find steam per H. P. per hour and the diameters of each nozzle at exit and at the throat.

Refer to temperature entropy chart, 200 pounds pressure, 35° superheat. The entropy is 1.57. Follow down on 1.57 until the temperature corresponding to $0.6 \times 200 =$ 120 pounds pressure is reached. Read $x_t = 0.99$. Continue on 1.57 until the temperature corresponding to 2 pounds is reached. Read $x_e = 0.80$.

$$H_i = 354.3 + 843.5 + (0.60 \times 35) = 1218.8$$

$$H_t = 312.3 + (0.99 \times 876.9) = 1180.4$$

$$H_e = 94.2 + (0.80 \times 1021.9) = 911.7$$

$$V_t = 224\sqrt{1218.8 - 1180.4} = 1393$$

$$V_e = 224\sqrt{0.85\ (1218.8 - 911.7)} = 3618$$

$$\frac{0.15\ (1218.8 - 911.7)}{1021.9} = 0.045$$

$$x_f = x_e + 0.045 = 0.80 + 0.045 = 0.845$$

The kinetic energy per pound of a jet issuing with a velocity of 3,620 feet per second is $\frac{3620 \times 3620}{2 \times 32.2}$. As 65 per cent of this is utilized, the energy received by the wheel per second per pound of steam is

$$\frac{0.65 \times 3620 \times 3620}{64.4} \text{ foot-pounds.}$$

The energy needed per second to develop 350 H. P. is

$$350 \times \frac{33000}{60}$$

hence the number of pounds of steam which must be supplied per second is

$$\frac{350 \times 33000 \times 64.4}{0.65 \times 3620 \times 3620 \times 60} = 1.455$$

The steam per H. P. hour is

$$\frac{1.455 \times 3600}{350} = 14.96 \text{ lbs.}$$

The steam per nozzle per second is

$$\frac{1.455}{7} = 0.208 \text{ lbs.}$$

The volume of a pound of mixture at the pressure and the condition at the throat is $0.99 (3.723 - 0.016) + 0.016 = 3.686$ cubic feet. The volume of a pound at exit is $0.845 \times (173.1 - 0.016) + 0.016 = 146.2$ cubic feet. The area of the throat in square feet is

$$\frac{3.686 \times 0.208}{1390}; \text{ or } 0.32 \text{ inches diameter.}$$

The area of the nozzle at exit in square feet is

$$\frac{146.2 \times 0.208}{3620}; \text{ or } 1.33 \text{ inches diameter.}$$

CALCULATING THE SIZE OF A STEAM MAIN

The indicator when applied to the steam chest as explained in Part II, page 107, sometimes shows that the steam pipe is not large enough to supply the engine.

PERFECT GASES

The characteristic equation of a perfect gas or the equation giving the relation between the absolute pressure, the volume and the absolute temperature is

$$\frac{Pv}{T} = \frac{P_1 v_1}{T_1} = \frac{P_2 v_2}{T_2}, \text{ etc.}$$

This relation was determined experimentally.

The volume of a pound of air at atmospheric pressure and at freezing point has been determined experimentally to be 12.39 cubic feet. That of hydrogen, 178.2 cubic feet.

Atmospheric pressure is 14.7 pounds on the square inch, or 2116.3 pounds on the square foot, equivalent to 29.92 inches of mercury or 760 mm. of mercury. The temperature T is absolute; as has been stated, this is found by adding 459.5 to the reading of a Fahrenheit thermometer.

A few examples will best illustrate how use is to be made of this equation.

(1). What will be the volume of one pound of air at 100 pounds absolute pressure and at 139.3° F.?

$$\frac{14.7 \times 12.39}{491.5} = \frac{100 \times v}{459.5 + 139.3}$$

$$v = 2.22 \text{ cu. ft.}$$

(2). What will be the weight of a cubic foot of air at this pressure and temperature?

$$\frac{1}{v} = \frac{1}{2.22} = 0.45 \text{ lbs.}$$

(3). An air compressor draws in 100 cubic feet of free air per minute at 14.6 pounds pressure (absolute) and at 60° F. The air is compressed to 200 pounds absolute and leaves at 120° F. What is the volume of the air discharged?

$$\frac{14.6 \times 100}{459.5 + 60} = \frac{200 \times v}{459.5 + 120}$$

$$v = 8.14 \text{ cu. ft.}$$

(4). A balloon of 10,000 cubic feet capacity, weighing together with car, sand bags, etc., 550 pounds, has 9,000 where W is in foot-pounds. P_1 is the absolute pressure on the square inch at the beginning of compression or at the end of expansion; v_1 is the volume in cubic feet at this pressure. P_2 is the absolute pressure on the square inch at the end of an isothermal compression or at the beginning of an isothermal expansion. The heat which must be abstracted during an isothermal compression or added during an isothermal expansion is $\frac{W}{778}$.

Adiabatic line. The equation in terms of pressure and volume representing an adiabatic expansion or compression of a perfect gas is

$$P\, v^{1.405} = P_1\, v_1^{1.405}$$

The temperature along an adiabatic change may be calculated by combining this equation with the characteristic equation for gases

$$\frac{P\, v}{T} = \frac{P_1\, v_1}{T_1}$$

The work done by an adiabatic expansion of a perfect gas or required for an adiabatic compression is

$$W = \frac{144\, P_1\, v_1}{0.405} \times \left[1 - \left(\frac{v_1}{v_2}\right)^{.405}\right]$$

where W is in foot-pounds. P_1 is the absolute pressure on the square inch at the beginning of expansion or at the end of compression; v_1 is the volume at this pressure. v_2 is the volume after the expansion or at the beginning of compression. The volumes are measured in cubic feet.

This value $\frac{v_1}{v_2}$ will always come out less than unity. Suppose for illustration $\frac{v_1}{v_2} = 0.3$. This is to be raised to the 0.405 power. The logarithm of 0.3 is $9.47712 - 10$. Write this $999.47712 - 1000$ and multiply by 0.405 thus:

$$(999.47712 - 1000) \times 0.405 = (404.78823 - 405.) \text{ or}$$
$$9.78823 - 10$$

The number corresponding to this log. is 0.6141.

where P_s is the absolute pressure per square inch at end of the suction stroke, and P_d is the absolute delivery pressure; and v is the cubic feet of free air.

For a three-stage compressor this formula becomes

$$0.0633\ P_s\, v \left\{ \left(\frac{P_d}{P_s}\right)^{.077} - 1 \right\} = \text{H. P.}$$

The air at entrance to a compressor is slightly rarefied, thus making P_s less than atmospheric pressure, and the volume displaced by the piston of the compressor has to be greater than the free air on this account and also on account of the clearance. The ratio of the volume of free air per minute to the piston displacement is called the displacement efficiency. For the very best compressors with mechanically operated valves this is approximately 95 per cent. In the expression just given for calculating H. P., a value of 90 has been used for the displacement efficiency as this is more nearly correct for the ordinary compressor.

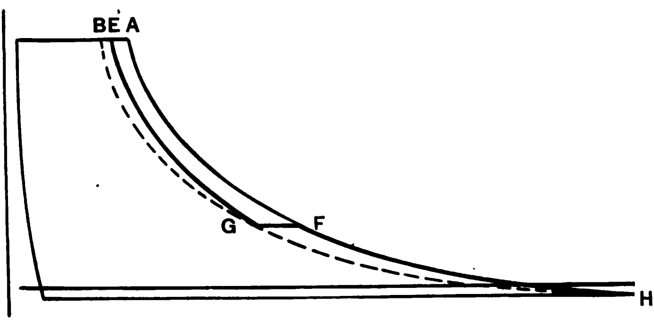


FIG. 58

The water jackets of a compressor are not very efficient in abstracting the heat of compression. By dividing the compression up between two or more cylinders or by compressing in stages it is possible to cool the air between stages down to its original temperature by passing it through inter-coolers placed between the compressor cylinders. This greatly reduces the work of compression as is shown by the illustration, Fig. 58.

The vertical line represents the cylinder head. H A represents the entire compression as taking place in one cylinder along a line $P v^{1.3}$ = a constant. The dotted line H B represents an isothermal compression. Air compressed along the line H A would shrink in volume from A to B as it cooled to its original temperature at intake. If the compression had been divided into two stages with an inter-cooler between the two cylinders the air delivered by the first stage at F would shrink in the inter-cooler to the volume at G before entering the second stage. The compression in the second stage is along a line G E of equation $P v^{1.3}$ = a constant, and the air at delivery shrinks from E to B. Evidently there has been a saving of work equal to that represented by the area F G E A. Against this saving is to be charged the extra mechanical loss due to the friction of the second cylinder. For a two-stage compression the absolute pressure on the square inch P_F is equal to $\sqrt{P_H \times P_A}$ the pressures at H and A being absolute. For a three-stage compression the absolute pressure at the end of the first stage should be $\sqrt[3]{P_H^2 \times P_A}$ and at the end of the second stage the pressure should be $\sqrt[3]{P_H \times P_A^2}$; the pressures P_H and P_A being absolute.

Problem on the compressor. 1,000 cubic feet of free air per minute are compressed to 265.3 pounds gage pressure in a two-stage compressor which is steam driven. The displacement efficiency of the compressor is 90 per cent. The pressure in the first cylinder at the end of the suction stroke is 14.0 pounds absolute, or 0.7 pounds below the atmosphere. The compression is along a line $P v^{1.3}$ = a constant. Calculate the H. P. needed at the compressor cylinders, and assuming a mechanical efficiency of 85 per cent, what is the

I. H. P. needed at the steam cylinders? What should be the pressure at the end of the compression in the first stage?

$$\text{H. P.} = 0.0422 \times 14. \times 1000 \left\{ \left(\frac{265.3 + 14.7}{14.0} \right)^{.1154} - 1 \right\} = 244.$$

$\frac{244}{0.85} = 287$ H. P: for steam cylinders. Absolute pressure at end of first stage $= \sqrt{280 \times 14} = 62.6$; or 47.9 pounds gage pressure.

PROBABLE HORSE-POWER OF AN ENGINE

The method of finding the number of expansions was explained in Chapter I, at page 11. As far as the power developed by a compound or triple expansion engine is concerned, the total number of expansions worked through by the steam might be made in the low pressure cylinder. By dividing the expansion between two or three cylinders the total cylinder condensation is reduced and a better rotative effect is obtained.

The expansion line of an indicator card does not vary much from a rectangular hyperbola having the equation $Pv =$ a constant, and such an equation is commonly assumed.

The entire cycle of expansions being assumed to take place in the low pressure cylinder, the M. E. P. for the entire cycle is figured for the low, and this M. E. P. multiplied by a constant between 0.7 and 0.9, which makes allowance for the losses in area and in M. E. P. due to the rounding of the card at cut-off and at release, and to the loss due to compression. For a Corliss valve gear these losses are small and the multiplier 0.9 would be used. For a plain slide valve a multiplier as low as 0.7 would be used.

The expression for M. E. P. is

$$\text{M. E. P.} = \frac{P_1}{n} + \frac{P_1}{n} \times 2.3026 \text{ log. } n - P_2$$

REFRIGERATION

A refrigerating machine of the compression type draws in heat on the suction stroke of the compressor, and discharges, at a higher temperature, the heat drawn in plus the heat equivalent of the work of compression.

The substance used as the refrigerant acts simply as a heat carrier the same as steam is used as the heat carrier to transfer to the engine the heat of the coal burned under the boiler.

Various substances have been used as refrigerants: Ammonia, NH_3, Carbonic Acid, CO_2, Sulphuric Acid, SO_2, Ethylchloride, C_2H_5Cl, water vapor and air.

In America probably 90 to 95 per cent of all refrigerating plants use ammonia in either compression or absorption machines.

Liquid ammonia, like water, can be made into vapor like steam and into superheated vapor like superheated steam. There are tables of the properties of ammonia similar to those giving the properties of steam, the tables of Keyes and Brownlee and those by Goodenough being the most reliable.

The pressures of saturated ammonia vapor at a few temperatures will give one an idea of the suction pressures which must be maintained in order to hold a certain temperature in the refrigerator.

Press. Ab.	Temp. °F	Vol. 1 lb. Saturated Vapor	Heat of Liquid	Latent Heat	Heat Content Dry Vapor
15	-27.2	17.8	-67.5	605.1	538.3
20	-16.7	13.7	-55.5	595.6	541.0
25	- 8.1	11.1	-45.9	588.0	543.0
30	- 0.8	9.3	-37.6	581.4	544.5
35	5.7	8.1	-30.3	575.7	545.9
40	11.3	7.1	-23.7	570.6	547.0
45	16.6	6.4	-17.7	565.8	548.
160	82.1	1.9	57.8	498.3	556.1
170	85.8	1.8	62.2	494.2	556.4
180	89.3	1.7	66.3	490.2	556.5
190	92.7	1.6	70.3	486.2	556.5

Generally 5 cubic feet piston displacement per minute may be taken as a fair average rating per ton and 1.5 H. P. at compressor per ton. The displacement efficiency of a compressor may be taken as 80 to 85 per cent.

As ammonia gas acts on composition the indicators used on the compression cylinder must be made of steel or other metal unaffected by the gas.

With carbonic acid machines where pressures of from 800 to 1200 pounds are required, it is necessary to use indicators with small diameter of piston.

The pressure ranges for sulphurous anhydride are about the same as with ammonia. This gas does not act on composition.

Dense air refrigerating machines are used on the battleships in the United States navy and water vapor machines in the French navy. While these machines are much larger than the other types referred to, there is no danger to the men, should any part of the system be broken and the working substances escape into the air.

The dense air machine consists of a compressor, a cooler, an expanding cylinder and a coil of pipe immersed in the brine which is to be cooled and circulated.

The compressor, the expanding cylinder and the coil are charged up to 65 pounds pressure with compressed air. The air compressor discharges air at 180 to 260 pounds pressure into the cooler, where the temperature of the air is reduced by cooling water; this air now goes behind the piston in the cylinder known as the expanding cylinder and expands along what is approximately an adiabatic line down to 65 pounds, when it is delivered at low temperature into the coil immersed in the brine.

The air picks up heat from the brine and enters the suction side of the compressor and is again compressed.

cylinder up to cut-off and up to release. This is called the total weight of mixture.

The weight of steam in the cylinder at compression is found by assuming that the space between the piston and the head of the cylinder, including port passages, is filled with dry steam of the absolute pressure at compression.

The weight at compression is equal to the (per cent clearance plus the per cent compression) times the piston displacement and times the weight of a cubic foot of steam at the absolute pressure at compression. (The weight of a cubic foot of steam is the reciprocal of the volume of a pound.)

Call this weight at compression M_0.

The weight of mixture in the cylinder per stroke is $M + M_0$.

The volume at cut-off is equal to the (per cent clearance plus the per cent of cut-off) times the piston displacement. Call this V_1.

This volume is filled by $(M + M_0)$ pounds of mixture at the absolute pressure of steam at cut-off as obtained from the diagram.

It has previously been shown (page 127) that the volume of one pound of mixture is $V = x\ (s - 0.016) + 0.016$ where x is the per cent steam by weight.

Hence V_1, the volume of $(M + M_0)$ pounds, must equal $V_1 = (M + M_0)\ x\ (s - 0.016) + (M + M_0)\ 0.016$. The volume s of one pound of steam can be found from tables or from a chart.

x, the only unknown term, is obtained by solving this equation.

The percentage of the mixture accounted for as steam at release is found in a similar way. The volume V_1 is replaced by V_2, the volume including clearance and the piston displacement up to release.

s is taken at the absolute pressure at release.

The percentage of mixture accounted for as steam at release frequently is as low as 0.6.

The indicator by itself does not show that there is any water with the steam in the cylinder. If the steam consumption of an engine is figured from the indicator card, assuming dry steam at release and dry steam at compression, a result as low as 18 pounds may be obtained when the actual consumption, as measured by the steam condensed in the condenser, is as much as 30 pounds.

An eccentric and an eccentric rod are equivalent to a crank and a connecting rod; the length of the crank being the distance from the center of the shaft to the center of the eccentric, called the eccentricity.

An eccentric is simply a crank with a crank pin so large that it takes in the shaft. The diameter of the eccentric has nothing to do with the travel given by it; this depends solely on the eccentricity.

The eccentric is generally set from 95° to 110° ahead of the crank. The excess over 90° is called the *angular advance.*

The eccentric rod is so long in proportion to the eccentricity of the eccentric that the travel of the valve is not affected appreciably by the slight angular movement of the eccentric rod and it is customary to assume that the valve moves in harmonic motion.

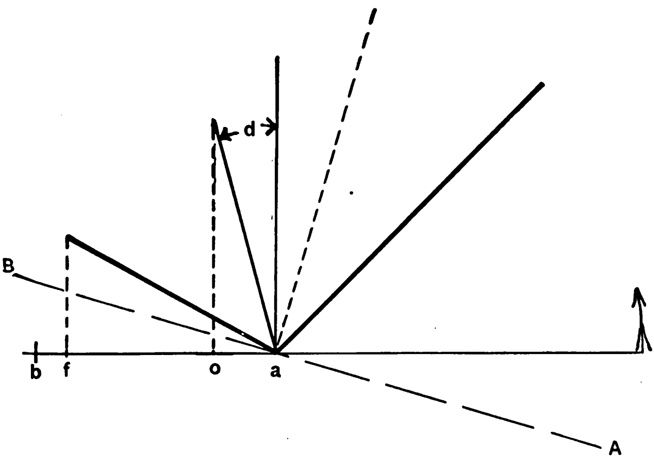


Fig. 59

In Fig. 59 the light lines show the crank at the center and the eccentric set with an angular advance *d.*

$(180° - d)$. From this point on, the valve is similarly displaced to the right of its mid position.

A section through a plain slide valve and its seat is given below. It is seen that when the valve is in mid position, or in the center of its travel, the outer edge of each end of the valve overlaps the outer edges of the ports. This overlap is called the outside lap. It may or may not be the same on the two ends of the valve. The outer edges of the valve govern the admission of steam and the cut-off of steam.

The inner edges of the valve overlap the inner edges of the port in the same way. This distance is called the inside lap. The inside edge of the valve governs the release of steam and compression of steam.

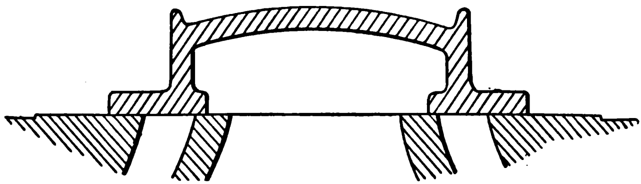


FIG. 61

Considering now the Zeuner diagram and the valve. When the crank is at the position O M $(360° - d)$ the valve is in mid position, as shown in Fig. 61. As the crank reaches the position O A the valve is displaced to the left a distance O E. The arc O E is drawn with a radius equal to the outside lap on the right-hand end of the valve. At this crank position the outer edge of the valve is on the outer edge of the port and admission of steam is about to begin; see Part I, Chapter I, page 6. When the crank gets to the dead point, the displacement of the valve is greater than the outside lap by an amount f, which is called the *lead.* This lead varies from 0.01 of an inch to 0.38 of an inch in different engines.

When the crank gets to an angle ($90° - d$) the valve is displaced its maximum amount to the left; the outer edge of the valve is now to the left of the outer edge of the port by a distance equal to the eccentricity minus the outside lap.

The valve then begins to move back to its mid position and at a crank angle O C the outer edge of the valve is on the outer edge of the port, since the displacement is equal to O E, the outside lap, and cut-off occurs. It is seen that the displacement of the valve is the same at cut-off and at admission; at admission the displacements are increasing and at cut-off decreasing.

After cut-off, the steam in the cylinder expands as the valve moves to mid position at the crank angle O M = ($180° - d$) and then to the right of the mid position till the crank angle O R is reached.

At the crank angle O R the valve is displaced to the right of the mid position a distance O Y, which is equal to the inside lap on the right-hand end of the valve. (*Note.* This is drawn out of proportion on the diagram in order to avoid confusion.) The inner edge of the valve is now on the inner edge of the port and steam is about to escape from the cylinder over the bridge into the exhaust port. This is release. Steam is exhausted from the cylinder from the crank position O R to the crank position O K when the inner edge of the valve is again on the inner edge of the port, giving compression.

The displacement of the valve at release and at compression is the same in amount; the displacements at release are increasing and those at compression decreasing.

Consider now the left-hand end of the valve, which controls the distribution of steam to the left-hand end of the cylinder.

Evidently the valve must be displaced to the right for admission and cut-off and this displacement must be equal in amount to the outside lap. The arc $o\ z$ is drawn with a radius when a valve has a clearance on its inner edge, to bring this edge to the inner edge of the port the valve must be displaced to the same side of the mid position that it was for cut-off. This means that the arc drawn on the Zeuner diagram to represent the inside clearance would be on the same side as that for the outside lap.

To lay out the seat for a valve: The width of the steam ports can be figured by assuming a velocity of steam as 100 feet per second, and the length of the port as about 0.8 the diameter of the cylinder.

Begin at the end of the valve which has the smaller outside lap. Starting at the outer edge of the port, measure off the outside lap. This outer edge of the valve will move to the right and to the left a distance in each direction equal to the eccentricity.

To allow the valve to over-travel its seat, the seat is depressed from a point about $\frac{1}{8}$ inch inside of the extreme travel out to the ends of the chest.

The travel of the valve in the other direction brings the outer edge over onto the bridge. The inner edge of the bridge should be about $\frac{1}{8}$ inch beyond this point. The inner edge of the bridge being now determined, and the width of the steam port having been figured, the width of the bridge itself is known.

To find the width of the exhaust port: Lay off the greater inside lap from the inner edge of the port onto the bridge. It may happen that the greater inside lap does not come on this end of the valve. The correct lap may be put on later after the width of the exhaust port is found. Measure from this lap towards the center of the chest a distance equal to the eccentricity, then add the width of the steam port; from this point, which is the inside edge of the bridge of the other end, lay off the bridge and then the steam port. Now lay out the laps.

The engine is now started up and a set of cards taken. The release and compression are adjusted by lengthening or shortening the links between the wrist platc and the exhaust valves. Lengthening the link increases the exhaust lap and delays release and hastens compression; shortening it hastens release and delays compression. The cut-off is adjusted last. This is done by varying the length of the rods from the governor to the knock-off tappets and may be done while the engine is running. After setting the cut-off the engine should be tested with the governor pushed up against the top collar to see if the tappets keep the claws from engaging, and tested also with the governor at its lowest position, not resting on the starting block, to see if the safety tappets prevent the claws from catching.

EXPLANATION OF LOGARITHMS AND HOW TO USE THEM

The common logarithm of a number represents the power to which 10 must be raised to equal the number. Thus the log. 100 = 2 because 10 must be raised to the second power to equal 100. The log. 1000 = 3. The log. 10 = 1. The log. 1 = 0. Evidently the logarithm of a number between 1 and 10 will be between 0 and 1; between 10 and 100 will be between 1 and 2; between 100 and 1000 between 2 and 3. The logarithm of a number less than 1 will be considered later.

It is known that $a^2 \times a^3 = a^5$ and that $a^5 \div a^3 = a^2$. If log. 3 = .4771 and log. 5 = .6990 then $10^{.4771} = 3$ and $10^{.6990} = 5$.

$$5 \times 3 = 10^{.6990} \times 10^{.4771} = 10^{.6990 + .4771} = 10^{1.1761}$$

$$5 \div 3 = 10^{.6990} \div 10^{.4771} = 10^{.6990 - .4771} = 10^{.2219}$$

To multiply two numbers, add their logarithms and this sum is the logarithm of the product; to divide one number by another, subtract the logarithms of the numbers, and the result is the logarithm of the answer.

Example: Multiply 5 × 4 × 20 × 3;

log. 5 =	.6990
log. 4 =	.6021
log. 20 =	1.3010
log. 3 =	.4771
	3.0792

The number corresponding to this logarithm is 1200.

The part of the logarithm to the left of the decimal point is called the mantissa. The value of the mantissa deter-

The number corresponding to .8634 is 730; 18 − 20 is the same as 8 — 10; this indicates that the left-hand figure is in the second decimal place, or 0.0730 = *Ans.*

To raise a number to any power, multiply the log. of the number by the exponent of the power and the result is the log. of the answer.

Examples: $(15)^2$

$$\begin{aligned} \log. 15 &= 1.1761 \\ & \,2 \\ \log. Ans. &= 2.3522 \\ Ans. &= 225. \end{aligned}$$

$(1.83)^3$

$$\begin{aligned} \log. 1.83 &= .2625 \\ & 3 \\ \log. Ans. &= .7875 \\ Ans. &= 6.13 \end{aligned}$$

$\sqrt[4]{1.83} = (1.83)^{\frac{1}{4}}$

$\log. 1.83 = .2625$;

multiply by $\frac{1}{4}$ or divide by 4,

$$\begin{aligned} \log. Ans. &= .0656 \\ \log. 1.16 &= \underline{.0645} \\ & 11 \end{aligned}$$

In the table of proportional parts, on page 172, at the right of this line in which .0645 is found, 11 corresponds to 3, so the next figure is 3 and the answer is 1.163.

$$\sqrt[5]{0.00373} = (0.00373)^{\frac{1}{5}}$$

$$\log. 0.00373 = 7.5717 - 10$$

this may be written 47.5717 — 50; divide by 5;

$$\begin{aligned} \log. Ans. &= 9.5143 - 10; \\ Ans. &= 0.3268 \end{aligned}$$

TABLES

LOGARITHMS

Nat. Nos.	0	1	2	3	4	5	6	7	8	9	Proportional Parts 1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	4	8	12	17	21	25	29	33	37
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	3	6	8	11	14	17	20	22	25
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	[illegible]
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	[illegible]

LOGARITHMS

Nat. Nos.	0	1	2	3	4	5	6	7	8	9	Proportional Parts 1	2	3	4	5	6	7	8	9
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
82	9138	9143	9149	9154	5159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4

AREAS AND CIRCUMFERENCES OF CIRCLES

ADVANCING BY EIGHTHS

Diam.	Area.	Circum.
1-64	.00019	.04909
1-32	.00077	.09818
3-64	.00173	.14726
1-16	.00307	.19635
3-32	.00690	.29452
1-8	.01227	.39270
5-32	.01917	.49087
3-16	.02761	.58905
7-32	.03758	.68722
1-4	.04909	.78540
9-32	.06213	.88357
5-16	.07670	.98175
11-32	.09281	1.0799
3-8	.11045	1.1781
13-32	.12962	1.2763
7-16	.15033	1.3744
15-32	.17257	1.4726
1-2	.19635	1.5708
17-32	.22166	1.6690
9-16	.24850	1.7671
19-32	.27688	1.8653
5-8	.30680	1.9635
21-32	.33824	2.0617
11-16	.37122	2.1598
23-32	.40574	2.2580
3-4	.44179	2.3562
25-32	.47937	2.4544
13-16	.51849	2.5525
27-32	.55914	2.6507
7-8	.60132	2.7489
29-32	.64504	2.8471
15-16	.69029	2.9452
31-32	.73708	3.0434
1.	.7854	3.1416
1-16	.8866	3.3379
1-8	.9940	3.5343
3-16	1.1075	3.7306
1-4	1.2272	3.9270
5-16	1.3530	4.1233
3-8	1.4849	4.3197
7-16	1.6230	4.5160
1-2	1.7671	4.7124
9-16	1.9175	4.9087
5-8	2.0739	5.1051
11-16	2.2365	5.3014
3-4	2.4053	5.4978
13-16	2.5802	5.6941
7-8	2.7612	5.8905
15-16	2.9483	6.0868
2.	3.1416	6.2832
1-16	3.3410	6.4795
1-8	3.5466	6.6759
3-16	3.7583	6.8722
1-4	3.9761	7.0686
5-16	4.2000	7.2649

Diam.	Area.	Circum.
2 3-8	4.4301	7.4613
7-16	4.6664	7.6576
1-2	4.9087	7.8540
9-16	5.1572	8.0503
5-8	5.4119	8.2467
11-16	5.6727	8.4430
3-4	5.9396	8.6394
13-16	6.2126	8.8357
7-8	6.4918	9.0321
15-16	6.7771	9.2284
3.	7.0686	9.4248
1-16	7.3662	9.6211
1-8	7.6699	9.8175
3-16	7.9798	10.014
1-4	8.2958	10.210
5-16	8.6179	10.407
3-8	8.9462	10.603
7-16	9.2806	10.799
1-2	9.6211	10.996
9-16	9.9678	11.192
5-8	10.321	11.388
11-16	10.680	11.585
3-4	11.045	11.781
13-16	11.416	11.977
7-8	11.793	12.174
15-16	12.177	12.370
4.	12.566	12.566
1-16	12.962	12.763
1-8	13.364	12.959
3-16	13.772	13.155
1-4	14.186	13.352
5-16	14.607	13.548
3-8	15.033	13.744
7-16	15.466	13.941
1-2	15.904	14.137
9-16	16.349	14.334
5-8	16.800	14.530
11-16	17.257	14.726
3-4	17.721	14.923
13-16	18.190	15.119
7-8	18.665	15.315
15-16	19.147	15.512
5.	19.635	15.708
1-16	20.129	15.904
1-8	20.629	16.101
3-16	21.135	16.297
1-4	21.648	16.493
5-16	22.166	16.690
3-8	22.691	16.886
7-16	23.221	17.082
1-2	23.758	17.279
9-16	24.301	17.475
5-8	24.850	17.671
11-16	25.406	17.868
3-4	25.967	18.064
13-16	26.535	18.261
7-8	27.109	18.457
15-16	27.688	18.653

Diam.	Area.	Circum.
6.	28.274	18.850
1-8	29.465	19.242
1-4	30.680	19.635
3-8	31.919	20.028
1-2	33.183	20.420
5-8	34.472	20.813
3-4	35.785	21.206
7-8	37.122	21.598
7.	38.485	21.991
1-8	39.871	22.384
1-4	41.282	22.776
3-8	42.718	23.169
1-2	44.179	23.562
5-8	45.664	23.955
3-4	47.173	24.347
7-8	48.707	24.740
8.	50.265	25.133
1-8	51.849	25.525
1-4	53.456	25.918
3-8	55.088	26.311
1-2	56.745	26.704
5-8	58.426	27.096
3-4	60.132	27.489
7-8	61.862	27.882
9.	63.617	28.274
1-8	65.397	28.667
1-4	67.201	29.060
3-8	69.029	29.452
1-2	70.882	29.845
5-8	72.760	30.238
3-4	74.662	30.631
7-8	76.589	31.023
10.	78.540	31.416
1-8	80.516	31.809
1-4	82.516	32.201
3-8	84.541	32.594
1-2	86.590	32.987
5-8	88.664	33.379
3-4	90.763	33.772
7-8	92.886	34.165
11.	95.033	34.558
1-8	97.205	34.950
1-4	99.402	35.343
3-8	101.62	35.736
1-2	103.87	36.128
5-8	106.14	36.521
3-4	108.43	36.914
7-8	110.75	37.306
12.	113.10	37.699
1-8	115.47	38.092
1-4	117.86	38.485
3-8	120.28	38.877
1-2	122.72	39.270
5-8	125.19	39.663
3-4	127.68	40.055
7-8	130.19	40.448

AREAS AND CIRCUMFERENCES OF CIRCLES

For Diameters from $\frac{1}{10}$ to 99, advancing by Tenths

Diam.	Area.	Circum.	Diam.	Area.	Circum.	Diam.	Area.	Circum.
15.0	176.7146	47.1239	20.0	314.1593	62.8319	25.0	490.8739	78.5398
.1	179.0786	47.4380	.1	317.3087	63.1460	.1	494.8087	78.8540
.2	181.4584	47.7522	.2	320.4739	63.4602	.2	498.7592	79.1681
.3	183.8539	48.0664	.3	323.6547	63.7743	.3	502.7255	79.4823
.4	186.2650	48.3805	.4	326.8513	64.0885	.4	506.7075	79.7965
.5	188.6919	48.6947	.5	330.0636	64.4026	.5	510.7052	80.1106
.6	191.1345	49.0088	.6	333.2916	64.7168	.6	514.7185	80.4248
.7	193.5928	49.3230	.7	336.5353	65.0310	.7	518.7476	80.7389
.8	196.0668	49.6372	.8	339.7947	65.3451	.8	522.7924	81.0531
.9	198.5565	49.9513	.9	343.0698	65.6593	.9	526.8529	81.3672
16.0	201.0619	50.2655	21.0	346.3606	65.9734	26.0	530.9292	81.6814
.1	203.5831	50.5796	.1	349.6671	66.2876	.1	535.0211	81.9956
.2	206.1199	50.8938	.2	352.9894	66.6018	.2	539.1287	82.3097
.3	208.6724	51.2080	.3	356.3273	66.9159	.3	543.2521	82.6239
.4	211.2407	51.5221	.4	359.6809	67.2301	.4	547.3911	82.9380
.5	213.8246	51.8363	.5	363.0503	67.5442	.5	551.5459	83.2522
.6	216.4243	52.1504	.6	366.4354	67.8584	.6	555.7163	83.5664
.7	219.0397	52.4646	.7	369.8361	68.1726	.7	559.9025	83.8805
.8	221.6708	52.7788	.8	373.2526	68.4867	.8	564.1044	84.1947
.9	224.3176	53.0929	.9	376.6848	68.8009	.9	568.3220	84.5088
17.0	226.9801	53.4071	22.0	380.1327	69.1150	27.0	572.5553	84.8230
.1	229.6583	53.7212	.1	383.5963	69.4292	.1	576.8043	85.1372
.2	232.3522	54.0354	.2	387.0756	69.7434	.2	581.0690	85.4513
.3	235.0618	54.3496	.3	390.5707	70.0575	.3	585.3494	85.7655
.4	237.7871	54.6637	.4	394.0814	70.3717	.4	589.6455	86.0796
.5	240.5282	54.9779	.5	397.6078	70.6858	.5	593.9574	86.3938
.6	243.2849	55.2920	.6	401.1500	71.0000	.6	598.2849	86.7080
.7	246.0574	55.6062	.7	404.7078	71.3142	.7	602.6282	87.0221
.8	248.8456	55.9203	.8	408.2814	71.6283	.8	606.9871	87.3363
.9	251.6494	56.2345	.9	411.8707	71.9425	.9	611.3618	87.6504
18.0	254.4690	56.5486	23.0	415.4756	72.2566	28.0	615.7522	87.9646
.1	257.3043	56.8628	.1	419.0963	72.5708	.1	620.1582	88.2788
.2	260.1553	57.1770	.2	422.7327	72.8849	.2	624.5800	88.5929
.3	263.0220	57.4911	.3	426.3848	73.1991	.3	629.0175	88.9071
.4	265.9044	57.8053	.4	430.0526	73.5133	.4	633.4707	89.2212
.5	268.8025	58.1195	.5	433.7361	73.8274	.5	637.9397	89.5354
.6	271.7164	58.4336	.6	437.4354	74.1416	.6	642.4243	89.8495
.7	274.6459	58.7478	.7	441.1503	74.4557	.7	646.9246	90.1637
.8	277.5911	59.0619	.8	444.8809	74.7699	.8	651.4407	90.4779
.9	280.5521	59.3761	.9	448.6273	75.0841	.9	655.9724	90.7920
19.0	283.5287	59.6903	24.0	452.3893	75.3982	29.0	660.5199	91.1062
.1	286.5211	60.0044	.1	456.1671	75.7124	.1	665.0830	91.4203
.2	289.5292	60.3186	.2	459.9606	76.0265	.2	669.6619	91.7345
.3	292.5530	60.6327	.3	463.7698	76.3407	.3	674.2565	92.0487
.4	295.5925	60.9469	.4	467.5947	76.6549	.4	678.8668	92.3628
.5	298.6477	61.2611	.5	471.4352	76.9690	.5	683.4928	92.6770
.6	301.7186	61.5752	.6	475.2916	77.2832	.6	688.1345	92.9911
.7	304.8052	61.8894	.7	479.1636	77.5973	.7	692.7919	93.3053
.8	307.9075	62.2035	.8	483.0513	77.9115	.8	697.4650	93.6195
.9	311.0255	62.5177	.9	486.9547	78.2257	.9	702.1538	93.9336

AREAS AND CIRCUMFERENCES OF CIRCLES

For Diameters from $\frac{1}{10}$ to 99, advancing by Tenths

Diam.	Area.	Circum.	Diam.	Area.	Circum.	Diam.	Area.	Circum.
45.0	1590.4313	141.3717	50.0	1963.4954	157.0796	55.0	2375.8294	172.7876
.1	1597.5077	141.6858	.1	1971.3572	157.3938	.1	2384.4767	173.1017
.2	1604.5999	142.0000	.2	1979.2348	157.7080	.2	2393.1396	173.4159
.3	1611.7077	142.3142	.3	1987.1280	158.0221	.3	2401.8183	173.7301
.4	1618.8313	142.6283	.4	1995.0370	158.3363	.4	2410.5126	174.0442
.5	1625.9705	142.9425	.5	2002.9617	158.6504	.5	2419.2227	174.3584
.6	1633.1255	143.2566	.6	2010.9020	158.9646	.6	2427.9485	174.6726
.7	1640.2962	143.5708	.7	2018.8581	159.2787	.7	2436.6899	174.9867
.8	1647.4826	143.8849	.8	2026.8299	159.5929	.8	2445.4471	175.3009
.9	1654.6847	144.1991	.9	2034.8174	159.9071	.9	2454.2200	175.6150
46.0	1661.9025	144.5133	51.0	2042.8206	160.2212	56.0	2463.0086	175.9292
.1	1669.1360	144.8274	.1	2050.8395	160.5354	.1	2471.8130	176.2433
.2	1676.3853	145.1416	.2	2058.8742	160.8495	.2	2480.6330	176.5575
.3	1683.6502	145.4557	.3	2066.9245	161.1637	.3	2489.4687	176.8717
.4	1690.9308	145.7699	.4	2074.9905	161.4779	.4	2498.3201	177.1858
.5	1698.2272	146.0841	.5	2083.0723	161.7920	.5	2507.1873	177.5000
.6	1705.5392	146.3982	.6	2091.1697	162.1062	.6	2516.0701	177.8141
.7	1712.8670	146.7124	.7	2099.2829	162.4203	.7	2524.9687	178.1283
.8	1720.2105	147.0265	.8	2107.4118	162.7345	.8	2533.8830	178.4425
.9	1727.5697	147.3407	.9	2115.5563	163.0487	.9	2542.8129	178.7566
47.0	1734.9445	147.6550	52.0	2123.7166	163.3628	57.0	2551.7586	179.0708
.1	1742.3351	147.9690	.1	2131.8926	163.6770	.1	2560.7200	179.3849
.2	1749.7414	148.2832	.2	2140.0843	163.9911	.2	2569.6971	179.6991
.3	1757.1635	148.5973	.3	2148.2917	164.3053	.3	2578.6899	180.0133
.4	1764.6012	148.9115	.4	2156.5149	164.6195	.4	2587.6985	180.3274
.5	1772.0546	149.2257	.5	2164.7537	164.9336	.5	2596.7227	180.6416
.6	1779.5237	149.5398	.6	2173.0082	165.2479	.6	2605.7626	180.9557
.7	1787.0086	149.8540	.7	2181.2785	165.5619	.7	2614.8183	181.2699
.8	1794.5091	150.1681	.8	2189.5644	165.8761	.8	2623.8896	181.5841
.9	1802.0254	150.4823	.9	2197.8661	166.1903	.9	2632.9767	181.8982
48.0	1809.5574	150.7964	53.0	2206.1834	166.5044	58.0	2642.0794	182.2124
.1	1817.1050	151.1106	.1	2214.5165	166.8186	.1	2651.1979	182.5265
.2	1824.6684	151.4248	.2	2222.8653	167.1327	.2	2660.3321	182.8407
.3	1832.2475	151.7389	.3	2231.2298	167.4469	.3	2669.4820	183.1549
.4	1839.8423	152.0531	.4	2239.6100	167.7610	.4	2678.6476	183.4690
.5	1847.4528	152.3672	.5	2248.0059	168.0752	.5	2687.8289	183.7832
.6	1855.0790	152.6814	.6	2256.4175	168.3894	.6	2697.0259	184.0973
.7	1862.7210	152.9956	.7	2264.8448	168.7035	.7	2706.2386	184.4115
.8	1870.3786	153.3097	.8	2273.2879	169.0177	.8	2715.4670	184.7256
.9	1878.0519	153.6239	.9	2281.7466	169.3318	.9	2724.7112	185.0398
49.0	1885.7409	153.9380	54.0	2290.2210	169.6460	59.0	2733.9710	185.3540
.1	1893.4457	154.2522	.1	2298.7112	169.9602	.1	2743.2466	185.6681
.2	1901.1662	154.5664	.2	2307.2171	170.2743	.2	2752.5378	185.9823
.3	1908.9024	154.8805	.3	2315.7386	170.5885	.3	2761.8448	186.2964
.4	1916.6543	155.1947	.4	2324.2759	170.9026	.4	2771.1675	186.6106
.5	1924.4218	155.5088	.5	2332.8289	171.2168	.5	2780.5058	186.9248
.6	1932.2051	155.8230	.6	2341.3976	171.5310	.6	2789.8599	187.2389
.7	1940.0042	156.1372	.7	2349.9820	171.8451	.7	2799.2297	187.5531
.8	1947.8189	156.4513	.8	2358.5821	172.1593	.8	2808.6152	187.8672
.9	1955.6493	156.7655	.9	2367.1979	172.4735	.9	2818.0165	188.1814

AREAS AND CIRCUMFERENCES OF CIRCLES

For Diameters from $\frac{1}{10}$ to 99, advancing by Tenths

Diam.	Area.	Circum.	Diam.	Area.	Circum.	Diam.	Area.	Circum.
60.0	2827.4334	188.4956	65.0	3318.3072	204.2035	70.0	3848.4510	219.9115
.1	2836.8660	188.8097	.1	3328.5253	204.5176	.1	3859.4544	220.2256
.2	2846.3144	189.1239	.2	3338.7590	204.8318	.2	3870.4736	220.5398
.3	2855.7784	189.4380	.3	3349.0085	205.1460	.3	3881.5084	220.8540
.4	2865.2582	189.7522	.4	3359.2736	205.4602	.4	3892.5590	221.1681
.5	2874.7536	190.0664	.5	3369.5545	205.7743	.5	3903.6252	221.4823
.6	2884.2648	190.3805	.6	3379.8510	206.0885	.6	3914.7072	221.7964
.7	2893.7917	190.6947	.7	3390.1633	206.4026	.7	3925.8049	222.1106
.8	2903.3343	191.0088	.8	3400.4913	206.7168	.8	3936.9182	222.4248
.9	2912.8926	191.3230	.9	3410.8350	207.0310	.9	3948.0473	222.7389
61.0	2922.4666	191.6372	66.0	3421.1944	207.3451	71.0	3959.1921	223.0531
.1	2932.0563	191.9513	.1	3431.5695	207.6593	.1	3970.3526	223.3672
.2	2941.6617	192.2655	.2	3441.9603	207.9734	.2	3981.5289	223.6814
.3	2951.2828	192.5796	.3	3452.3669	208.2876	.3	3992.7208	223.9956
.4	2960.9197	192.8938	.4	3462.7891	208.6017	.4	4003.9284	224.3097
.5	2970.5722	193.2079	.5	3473.2270	208.9159	.5	4015.1518	224.6239
.6	2980.2405	193.5221	.6	3483.6807	209.2301	.6	4026.3908	224.9380
.7	2989.9244	193.8363	.7	3494.1500	209.5442	.7	4037.6456	225.2522
.8	2999.6241	194.1504	.8	3504.6351	209.8584	.8	4048.9160	225.5664
.9	3009.3395	194.4646	.9	3515.1359	210.1725	.9	4060.2022	225.8805
62.0	3019.0705	194.7787	67.0	3525.6524	210.4867	72.0	4071.5041	226.1947
.1	3028.8173	195.0929	.1	3536.1845	210.8009	.1	4082.8217	226.5088
.2	3038.5798	195.4071	.2	3546.7324	211.1150	.2	4094.1550	226.8230
.3	3048.3580	195.7212	.3	3557.2960	211.4292	.3	4105.5040	227.1371
.4	3058.1520	196.0354	.4	3567.8754	211.7433	.4	4116.8687	227.4513
.5	3067.9616	196.3495	.5	3578.4704	212.0575	.5	4128.2491	227.7655
.6	3077.7869	196.6637	.6	3589.0811	212.3717	.6	4139.6452	228.0796
.7	3087.6279	196.9779	.7	3599.7075	212.6858	.7	4151.0571	228.3938
.8	3097.4847	197.2920	.8	3610.3497	213.0000	.8	4162.4846	228.7079
.9	3107.3571	197.6062	.9	3621.0075	213.3141	.9	4173.9279	229.0221
63.0	3117.2453	197.9203	68.0	3631.6811	213.6283	73.0	4185.3868	229.3363
.1	3127.1492	198.2345	.1	3642.3704	213.9425	.1	4196.8615	229.6504
.2	3137.0688	198.5487	.2	3653.0754	214.2566	.2	4208.3519	229.9646
.3	3147.0040	198.8628	.3	3663.7960	214.5708	.3	4219.8579	230.2787
.4	3156.9550	199.1770	.4	3674.5324	214.8849	.4	4231.3797	230.5929
.5	3166.9217	199.4911	.5	3685.2845	215.1991	.5	4242.9172	230.9071
.6	3176.9043	199.8053	.6	3696.0523	215.5133	.6	4254.4704	231.2212
.7	3186.9023	200.1195	.7	3706.8359	215.8274	.7	4266.0394	231.5354
.8	3196.9161	200.4336	.8	3717.6351	216.1416	.8	4277.6240	231.8495
.9	3206.9456	200.7478	.9	3728.4500	216.4556	.9	4289.2243	232.1637
64.0	3216.9909	201.0620	69.0	3739.2807	216.7699	74.0	4300.8403	232.4779
.1	3227.0518	201.3761	.1	3750.1270	217.0841	.1	4312.4721	232.7920
.2	3237.1285	201.6902	.2	3760.9891	217.3982	.2	4324.1195	233.1062
.3	3247.2222	202.0044	.3	3771.8668	217.7124	.3	4335.7827	233.4203
.4	3257.3289	202.3186	.4	3782.7603	218.0265	.4	4347.4616	233.7345
.5	3267.4527	202.6327	.5	3793.6695	218.3407	.5	4359.1562	234.0487
.6	3277.5922	202.9469	.6	3804.5944	218.6548	.6	4370.8664	234.3628
.7	3287.7474	203.2610	.7	3815.5350	218.9690	.7	4382.5924	234.6770
.8	3297.9183	203.5752	.8	3826.4913	219.2832	.8	4394.3341	234.9911
.9	3308.1049	203.8894	.9	3837.4633	219.5973	.9	4406.0916	235.3053

AREAS AND CIRCUMFERENCES OF CIRCLES

For Diameters from $\frac{1}{10}$ to 99, advancing by Tenths

Diam.	Area.	Circum.	Diam.	Area.	Circum.	Diam.	Area.	Circum.
75.0	4417.8647	235.6194	80.0	5026.5482	251.3274	85.0	5674.5017	267.0354
.1	4429.6535	235.9336	.1	5039.1225	251.6416	.1	5687.8614	267.3495
.2	4441.4580	236.2478	.2	5051.7124	251.9557	.2	5701.2367	267.6637
.3	4453.2783	236.5619	.3	5064.3180	252.2699	.3	5714.6277	267.9779
.4	4465.1142	236.8761	.4	5076.9394	252.5840	.4	5728.0345	268.2920
.5	4476.9659	237.1902	.5	5089.5764	252.8982	.5	5741.4569	268.6062
.6	4488.8332	237.5044	.6	5102.2292	253.2124	.6	5754.8951	268.9203
.7	4500.7163	237.8186	.7	5114.8977	253.5265	.7	5768.3490	269.2345
.8	4512.6151	238.1327	.8	5127.5819	253.8407	.8	5781.8185	269.5486
.9	4524.5296	238.4469	.9	5140.2818	254.1548	.9	5795.3038	269.8628
76.0	4536.4598	238.7610	81.0	5152.9973	254.4690	86.0	5808.8048	270.1770
.1	4548.4057	239.0752	.1	5165.7287	254.7832	.1	5822.3215	270.4911
.2	4560.3673	239.3894	.2	5178.4757	255.0973	.2	5835.8539	270.8053
.3	4572.3446	239.7035	.3	5191.2384	255.4115	.3	5849.4020	271.1194
.4	4584.3377	240.0177	.4	5204.0168	255.7256	.4	5862.9659	271.4336
.5	4596.3464	240.3318	.5	5216.8110	256.0398	.5	5876.5454	271.7478
.6	4608.3708	240.6460	.6	5229.6208	256.3540	.6	5890.1407	272.0619
.7	4620.4110	240.9602	.7	5242.4463	256.6681	.7	5903.7516	272.3761
.8	4632.4669	241.2743	.8	5255.2876	256.9823	.8	5917.3783	272.6902
.9	4644.5384	241.5885	.9	5268.1446	257.2966	.9	5931.0206	273.0044
77.0	4656.6257	241.9026	82.0	5281.0173	257.6106	87.0	5944.6787	273.3186
.1	4668.7287	242.2168	.1	5293.9056	257.9247	.1	5958.3525	273.6327
.2	4680.8474	242.5310	.2	5306.8097	258.2389	.2	5972.0420	273.9469
.3	4692.9818	242.8451	.3	5319.7295	258.5531	.3	5985.7472	274.2610
.4	4705.1319	243.1592	.4	5332.6650	258.8672	.4	5999.4681	274.5752
.5	4717.2977	243.4734	.5	5345.6162	259.1814	.5	6013.2047	274.8894
.6	4729.4792	243.7876	.6	5358.5832	259.4956	.6	6026.9570	275.2035
.7	4741.6765	244.1017	.7	5371.5658	259.8097	.7	6040.7250	275.5177
.8	4753.8894	244.4159	.8	5384.5641	260.1239	.8	6054.5088	275.8318
.9	4766.1181	244.7301	.9	5397.5782	260.4380	.9	6068.3082	276.1460
78.0	4778.3624	245.0442	83.0	5410.6079	260.7522	88.0	6082.1234	276.4602
.1	4790.6225	245.3584	.1	5423.6534	261.0663	.1	6095.9542	276.7743
.2	4802.8983	245.6725	.2	5436.7146	261.3805	.2	6109.8008	277.0885
.3	4815.1897	245.9867	.3	5449.7915	261.6947	.3	6123.6631	277.4026
.4	4827.4969	246.3009	.4	5462.8840	262.0088	.4	6137.5411	277.7168
.5	4839.8198	246.6150	.5	5475.9923	262.3230	.5	6151.4348	278.0309
.6	4852.1584	246.9292	.6	5489.1163	262.6371	.6	6165.3442	278.3451
.7	4864.5128	247.2433	.7	5502.2561	262.9513	.7	6179.2693	278.6593
.8	4876.8828	247.5575	.8	5515.4115	263.2655	.8	6193.2101	278.9740
.9	4889.2685	247.8717	.9	5528.5826	263.5796	.9	6207.1666	279.2876
79.0	4901.6699	248.1858	84.0	5541.7694	263.8938	89.0	6221.1389	279.6017
.1	4914.0871	248.5000	.1	5554.9720	264.2079	.1	6235.1268	279.9159
.2	4926.5199	248.8141	.2	5568.1902	264.5221	.2	6249.1304	280.2301
.3	4938.9685	249.1283	.3	5581.4242	264.8363	.3	6263.1498	280.5442
.4	4951.4328	249.4425	.4	5594.6739	265.1514	.4	6277.1849	280.8584
.5	4963.9127	249.7566	.5	5607.9392	265.4646	.5	6291.2356	281.1725
.6	4976.4084	250.0708	.6	5621.2203	265.7787	.6	6305.3021	281.4867
.7	4988.9198	250.3850	.7	5634.5171	266.0929	.7	6319.3843	281.8009
.8	5001.4469	250.6991	.8	5647.8296	266.4071	.8	6333.4822	282.1150
.9	5013.9897	251.0133	.9	5661.1578	266.7212	.9	6347.5958	282.4292

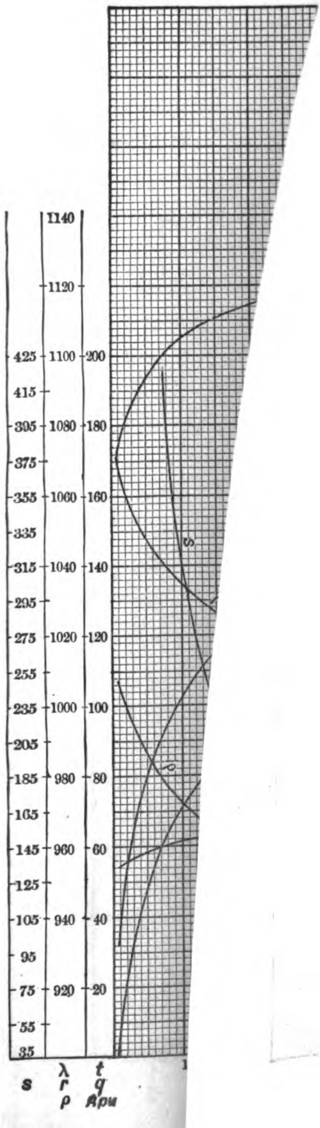
AREAS AND CIRCUMFERENCES OF CIRCLES

For Diameters from $\frac{1}{10}$ to 99, advancing by Tenths

Diam.	Area.	Circum.	Diam.	Area.	Circum.	Diam.	Area.	Circum.
90.0	6361.7251	282.7433	93.0	6792.9087	292.1681	96.0	7238.2295	301.5929
.1	6375.8701	283.0575	.1	6807.5250	292.4823	.1	7253.3170	301.9071
.2	6390.0309	283.3717	.2	6822.1569	292.7964	.2	7268.4202	302.2212
.3	6404.2073	283.6858	.3	6836.8046	293.1106	.3	7283.5391	302.5354
.4	6418.3995	284.0000	.4	6851.4680	293.4248	.4	7298.6737	302.8405
.5	6432.6073	284.3141	.5	6866.1471	293.7389	.5	7313.8240	303.1637
.6	6446.8309	284.6283	.6	6880.8419	294.0531	.6	7328.9901	303.4779
.7	6461.0701	284.9425	.7	6895.5524	294.3672	.7	7344.1718	303.7920
.8	6475.3251	285.2566	.8	6910.2786	294.6814	.8	7359.3693	304.1062
.9	6489.5958	285.5708	.9	6925.0205	294.9956	.9	7374.5824	304.4203
91.0	6503.8822	285.8849	94.0	6939.7782	295.3097	97.0	7389.8113	304.7345
.1	6518.1843	286.1991	.1	6954.5515	295.6239	.1	7405.0559	305.0486
.2	6532.5021	286.5133	.2	6969.3106	295.9380	.2	7420.3162	305.3628
.3	6546.8356	286.8274	.3	6984.1453	296.2522	.3	7435.5922	305.6770
.4	6561.1848	287.1416	.4	6998.9658	296.5663	.4	7450.8839	305.9911
.5	6575.5498	287.4557	.5	7013.8019	296.8805	.5	7466.1913	306.3053
.6	6589.9304	287.7699	.6	7028.6538	297.1947	.6	7481.5144	306.6194
.7	6604.3268	288.0840	.7	7043.5214	297.5088	.7	7496.8532	306.9336
.8	6618.7388	288.3982	.8	7058.4047	297.8230	.8	7512.2078	307.2478
.9	6633.1666	288.7124	.9	7073.3033	298.1371	.9	7527.5780	307.5619
92.0	6647.6101	289.0265	95.0	7088.2184	298.4513	98.0	7542.9640	307.8761
.1	6662.0692	289.3407	.1	7103.1488	298.7655	.1	7558.3656	308.1902
.2	6676.5441	289.6548	.2	7118.1950	299.0796	.2	7573.7830	308.5044
.3	6691.0347	289.9690	.3	7133.0568	299.3938	.3	7589.2161	308.8186
.4	6705.5410	290.2832	.4	7148.0343	299.7079	.4	7604.6648	309.1327
.5	6720.0630	290.5973	.5	7163.0276	300.0221	.5	7620.1293	309.4469
.6	6734.6008	290.9115	.6	7178.0366	300.3363	.6	7635.6095	309.7610
.7	6749.1542	291.2256	.7	7193.0612	300.6504	.7	7651.1054	310.0752
.8	6763.7233	291.5398	.8	7208.1016	300.9646	.8	7666.6170	310.3894
.9	6778.3082	291.8540	.9	7223.1577	301.2787	.9	7682.1444	310.7035

DECIMAL EQUIVALENTS OF FRACTIONS OF ONE INCH

1–64	.015625	17–64	.265625	33–64	.515625	49–64	.765625
1–32	.03125	9–32	.28125	17–32	.53125	25–32	.78125
3–64	.046875	19–64	.296875	35–64	.546875	51–64	.796875
1-16	.0625	**5-16**	.3125	**9-16**	.5625	**13-16**	.8125
5–64	.078125	21–64	.328125	37–64	.578125	53–64	.828125
3–32	.09375	11–32	.34375	19–32	.59375	27–32	.84375
7–64	.109375	23–64	.359375	39–64	.609375	55–64	.859375
1-8	.125	**3-8**	.375	**5-8**	.625	**7-8**	.875
9–64	.140625	25–64	.390625	41–64	.640625	57–64	.890625
5–32	.15625	13–32	.40625	21–32	.65625	29–32	.90625
11–64	.171875	27–64	.421875	43–64	.671875	59–64	.921875
3-16	.1875	**7-16**	.4375	**11-16**	.6875	**15-16**	.9375
13–64	.203125	29–64	.453125	45–64	.703125	61–64	.953125
7–32	.21875	15–32	.46875	23–32	.71875	31–32	.96875
15–64	.234375	31–64	.484375	47–64	.734375	63–64	.984375
1-4	.25	**1-2**	.50	**3-4**	.75	**1**	1.



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counter it is required to *move the whole mechanism,* and to do this intermittently and with the varying resistance of one or all of the figure-wheels.

In a Revolution Counter which shall be *reliable, durable,* and free from liability to serious injury, the actuating force must, through proper mechanism, be transmitted *directly* and with *certainty* to the figure-wheels, and this can best be done by means of a crank. It matters not in the Crosby Counter whether the movement of the crank is rotary or merely oscillatory, it will count just the same.

The Crosby Improved Revolution Counter, only, employs the crank principle, applied through other simple mechanical motions, so as to record with certainty the operations of any machine, and at the same time obviate all danger of injury to the counter itself or the machine to which it is attached.

This counter is adapted to either right or left hand rotary or reciprocating motions, and is capable of 500 revolutions per minute with safety to the machine and accuracy in the enumeration.

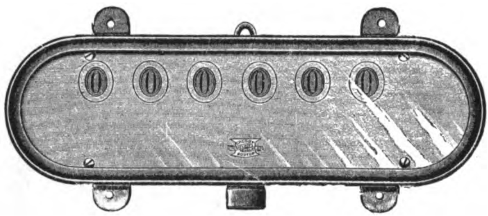
The shaft through which the actuating force is applied may extend from the counter either on the right-hand or left-hand side as desired.

It is made in the following sizes :

12 inch dial	8 wheels
12 " "	7 "
12 " "	6 "
10 " "	8 "
10 " "	7 "
10 " "	6 "
8½ " "	8 "
8½ " "	7 "
8½ " "	6 "
6¾ " "	6 "
6 " "	6 "

CROSBY SQUARE COUNTER

The actuating mechanism of this counter is positive and employs the principles just described, as used in the Crosby Revolution Counter. It is a strong and useful instrument, compact in form, durable and accurate. It may be provided with a re-setting device and also with a padlock if desired. When required for rotary motion, it should be stated



whether it is to be used for right-hand or left-hand rotation.

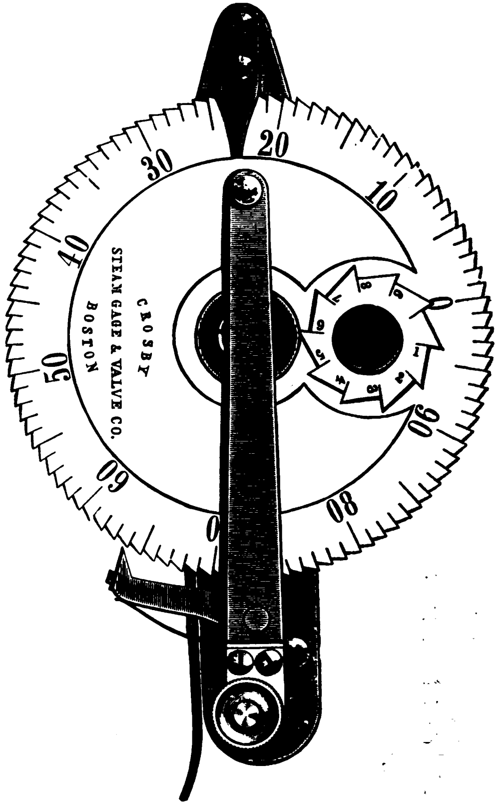
This counter is made in the following sizes :

$4\frac{5}{8} \times 1\frac{1}{2}$ in. dial	4 figures
$5\frac{1}{4} \times 1\frac{1}{2}$ " "	5 "
$6 \times 1\frac{1}{2}$ " "	6 "
$7\frac{1}{4} \times 2\frac{1}{4}$ " "	4 "
$8\frac{1}{4} \times 2\frac{1}{4}$ " "	5 "
$9\frac{1}{4} \times 2\frac{1}{4}$ " "	6 "
$10\frac{3}{8} \times 2\frac{1}{4}$ " "	7 "

THE CROSBY LOCOMOTIVE COUNTER

For High Rotative Speeds

The cut on page 188 shows the Locomotive Counter. It is designed particularly for use on locomotives and high speed engines, and is a valuable auxiliary to the steam engine indicator. The arm which moves the ratchet is connected by a cord with some reciprocating part of the engine, or with the drum motion, so as to give it about 1½ inches swing back and forth during each revolution of the shaft. It is



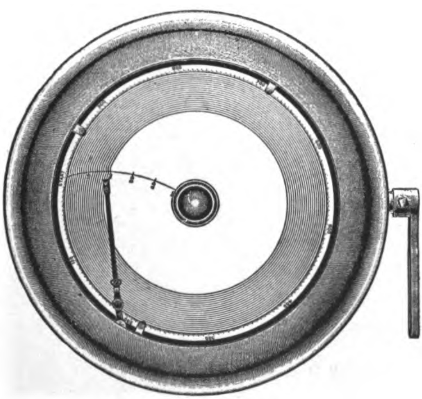
provided with a convenient starting and stopping device, so that it can be made to begin or stop counting at any instant.

CROSBY RECORDING COUNTER

Patented

This instrument furnishes a chart record of the revolutions or strokes of any engine, pump, or moving part. It is designed with remarkable genius for its special purpose, and it will record the highest speeds with mathematical accuracy. Every counter is tested to at least two thousand revolutions per minute, and will give positive results at much higher speeds without slip or error. Each regular chart affords a pen record up to fifty thousand consecutive strokes or revolutions, and the exact total number, or the elapsed count between any two noted periods, can be read with certainty. The highest working speeds found in mechanical operations are within the range of this device.

The Crosby Recording Counter is not a tachometer, but a chart-recording instrument, occupying a field by itself of peculiar importance. There is no other instrument like it, or that gives similar results. Its applications are varied and universal. It will be found especially valuable in making permanent records of the performance of machinery or engines either under special test or in daily service, and it is



of the greatest usefulness and importance to every engineer, designer, and user of power. The chart is 8 inches in diameter and easily read; the mechanism is well constructed, durable, and accurate; it cannot get out of order or adjustment. All like parts are interchangeable and suitably designed to give proper wear and service. It is adapted for both revolutions and reciprocating motion without alteration. It is simple to attach and no skill is required to operate it.

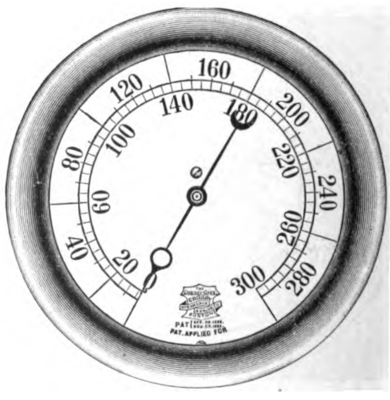
A smaller recording counter capable of registering at the highest speeds upon a chart reading to 5,000 revolutions or strokes, and adapted to be attached to the Crosby Reducing Wheel, is described on page 73.

CROSBY PRESSURE AND VACUUM GAGES

Important

Accuracy is the essential feature in all gages, whether pressure or vacuum. The principle of construction of Crosby gages is correct and they embody important improvements in many essential details.

In the Crosby Improved Pressure Gage the tube springs are connected at each end with their respective parts by screw



threads, without the use of any soldering material whatever, thus insuring tight joints under all conditions of heat and pressure.

The index mechanism and the dial are mounted upon an extension of the socket, thus rendering the entire operating parts of the gage independent of the case and free from any errors arising from its distortion or from external heat.

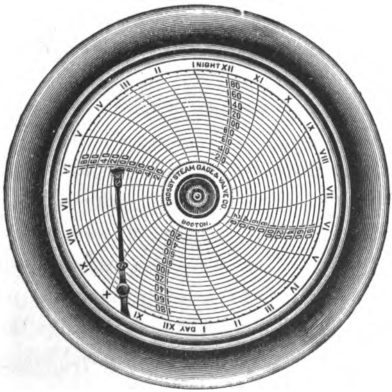
The method by which Crosby gages are tested and graduated will insure a truthful and reliable gage. Each one is tested under steam pressure and subjected to pressures accurately measured by standardized weights, and the gage is graduated to such absolute pressures and not by comparison only with another gage.

An equally accurate method is used in the testing of vacuum gages; each one is tried, marked, and adjusted by the direct readings of a mercury column, by means of an apparatus in which the successive stages of vacuum are actually produced.

Every gage used to indicate the pressure of steam should have a siphon or some other device which will furnish water to and completely fill the tube springs to keep them cool. Be sure that the connections between the gage and the siphon are perfectly tight.

CROSBY PRESSURE RECORDER

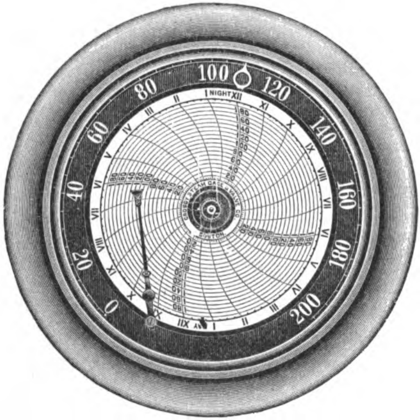
Patented



The Crosby Pressure Recorder records the pressure of any fluid during a certain period of time.

CROSBY PRESSURE RECORDER AND GAGE

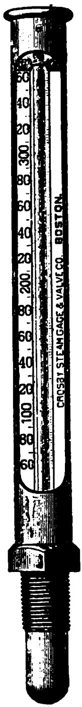
Patented



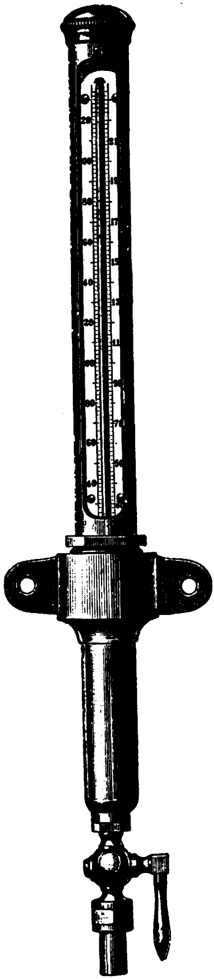
The Crosby Pressure Recorder and Gage, in addition to recording the pressure of any fluid during a certain period of time, has an index hand and an outside circle of figures to show the actual pressure recorded on the chart at the moment of observation; in this respect it operates like an ordinary steam gage.

The chart rotates once in 24 hours, and by the employment of a special clock movement, the rotation of the chart may be made to conform to any period of time from one hour to one week, thus adapting the instrument to almost any conditions to be found, in which a record of pressures is desirable or necessary; and the range of pressures which may be recorded is practically unlimited.

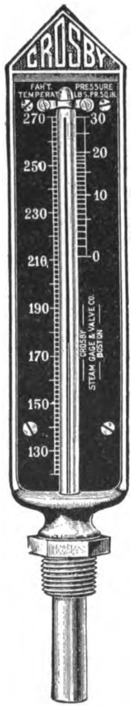
The reading of the chart, as shown above, is 110 pounds pressure at 6.30 o'clock A.M. Supposing the instrument to be properly connected to a steam boiler, or other receptacle, then during the next 24 hours, a red line is traced by the pen completely around the dial, showing by its deviations from a true circle, the variations of pressure which take place during the whole of that time.



STEAM PIPE THERMOMETER



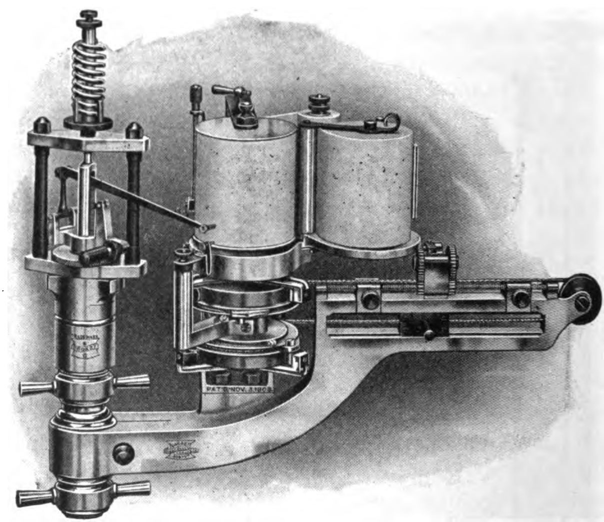
HOT WELL THERMOMETER



HOT WATER THERMOMETER

THE LANZA CONTINUOUS DIAGRAM APPLIANCE WITH CROSBY INDICATORS

Patented



The Lanza Continuous Diagram Appliance is not in itself an indicator, but displaces the ordinary drum as a means for supplying the paper for taking indicator cards, and any indicator may be combined or adapted for use with it. It is assembled upon a bracket, or frame, which is designed to support also the indicator and its connections so that these parts may be rigidly fixed in proper mutual relation. Upon this bracket are mounted the spindle for receiving the new roll of paper, the drum which feeds the paper forward, and upon which the pencil point bears in making the record, and the spool upon which the paper is afterward wound.

The drum is rotated continuously in one direction by the alternate engagement of two series of clutches controlled by a cord passing over the pulley at the extreme end of the bracket arm and actuated by a cross-head block which is diagram, but with this instrument all such uncertainty and error is avoided.

Determination of M. E. P. by Means of a Planimeter or an Integrator

Having taken a diagram corresponding to a certain number of revolutions of the engine by means of the Lanza Continuous Diagram Appliance, we must first draw through the points which mark the ends of the several strokes lines perpendicular to the atmospheric line.

If we use an ordinary planimeter, we need only to planimeter the positive and the negative areas separately, and then to subtract the sum of the negative areas from the sum of the positive areas, to obtain the total area of the given series of cards. Having obtained the resultant area in this way, we obtain the M. E. P. for this diagram by first dividing it by one-half of the length RZ (that is, the total length divided by two, in any case), and then multiplying this average height by the scale of the indicator spring.

Time can be saved if an integrator is used. It will be convenient to set the track of the integrator approximately parallel to the atmospheric line. To obtain the area of the diagrams corresponding to the successive revolutions of the engine, we can start the pencil of the integrator anywhere on the pressure line. In the forward motion we must drag the pencil of the integrator along the line drawn by the indicator pencil for every forward stroke, and along the atmospheric line for every return stroke, while in the return motion we must drag the pencil of the integrator along the line drawn by the pencil of the indicator for every return stroke and along the atmospheric line for every forward stroke. The resultant area can then be read off on the integrator. Any line parallel to the atmospheric could be used in this operation instead of the atmospheric, if for any reason it were more convenient.