PRACTICAL APPLICATION OF THE INDICATOR

Practical Application

of the

INDICATOR

WITH REFERENCE TO THE ADJUSTMENT OF VALVE GEAR ON ALL STYLES OF ENGINES

BY

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DEDICATION.

This work is dedicated, with the highest regard, to my friend the working engineer, with the hope that it may in some manner benefit and assist him in his endeavors to rise to a higher standard of excellence in his calling.

THE AUTHOR.

January 1st, 1894.

PREFACE.

While engaged in my work as consulting engineer, it has been my fortune to come in contact with a large number of working engineers; some more or less experienced in the use of the Indicator, others knowing practically nothing concerning it. It was gratifying, however, to note that whenever called upon to use my Indicators, I invariably found an interested spectator in the engineer, who, in most cases, expressed a desire to learn their use. In some cases, where desired I have given personal instructions, but in many cases the engineer could not afford such an expensive method of learning. To these, I recommended various works on the subject, but found the very general complaint that most of the works on the Indicator now published are not sufficiently definite for the beginner. These circumstances, together with the fact that I have personally felt the want of more practical information on the subject than is contained in existing books, have, to a great extent, induced me to prepare this volume; and it has been my aim to cover the ground as thoroughly as possible, and produce a work which will meet the requirements of the beginner as well as the experienced engineer. As so many of our engineers have not had the benefit of even a common school education, I have endeavored to use only such wording as would be readily understood by all, and avoid all intricate formulæ which tend to confuse and mislead the reader. The importance of the Indicator is now so generally recognized that the time is rapidly approaching when no engineer will be considered competent unless he is proficient in the use of the Indicator; and if this effort on my part shall have made the subject plainer to some, or in any degree assisted those who desire to better their condition, I shall feel amply repaid for the time and labor expended in the preparation of this work.

LEWIS M. ELLISON.

CHICAGO, Jan. 1, 1894.

INTRODUCTION.

The steam engine indicator is an attachment to the steam engine, which has been too long neglected. In fact, only a few years have elapsed since it was practically unknown to the average working engineer. However, questions were continually coming up which called for a frequent resort to its use, until now, in the hands of a skillful engineer, it is a very important factor in the running of the steam plant. The first indicator, I believe, was invented by James Watt and was a very crude affair, adapted for use only on engines of low pressure and running at slow speeds. It would be wholly unfit for use on the high pressure and high speed engines of the present day. Meantime, the changes in the styles of engines necessitated changes and improvements, and the makers of indicators have endeavored to keep pace with the times; and there is no doubt that the American indicators of to-day, for simplicity, accuracy and durability, far surpasses any like devices in the world. The benefits derived from the use of the indicator becoming more and more apparent, have greatly increased the demand, and recently there have been put upon the market several new makes and styles which are duly appreciated and which show the enterprise and mechanical skill of the maker. Nevertheless, we regret that they have not aimed to produce a more reliable and accurate instrument rather than one where reliability is sacrificed to price. While some of the older makes of indicators are reasonably correct, the majority of the recent makes are practically worthless when accuracy is desired and few of them will give two readings alike. It is to be hoped that the day will soon pass when such instruments will be recognized by intelligent engineers. I wish to state here that I have no connection or interest, financial or otherwise, with any indicator or attachment, steam engine or steam appliance, and wherever comparisons are made they are used simply as facts for the benefit of the readers and without any preference or prejudice on my part. With few exceptions the diagrams used in this work were taken by myself in actual practice and are engraved as nearly as possible as taken, full size and contain all the peculiarities of the original diagram. The demonstrations are put in the simplest language, so that all can fully understand them without reference to the higher mathematics, and some of them have never before appeared in print and will no doubt prove interesting.

PRACTICAL

APPLICATION OF THE INDICATOR.

CHAPTER I.

THE INDICATOR.

The indicator is an instrument for registering on paper the varying steam pressure on the piston of an engine during both the forward and return stroke. It consists, essentially, of a small steam cylinder, having on its axis a small drum upon which the paper is rolled. This drum revolves backward and forward by a motion derived from the crosshead of the engine. The cylinder is provided with a piston whose motion is resisted by a spiral spring. Steam is admitted beneath this piston, and as it is at a greater or less pressure, so is the spring more or less compressed and the piston rises and falls accordingly. The motion of the piston is conveyed by a series of levers to a pencil which is pressed against the paper on the drum. Before using the indicator clean the bearing surfaces of the cylinder and piston and lubricate them with the best cylinder oil that is clean and does not stick or gum. Use refined porpoise jaw oil on all the other bearings. On tests of long duration, the piston should be taken out and together with the cylinder should be cleaned and oiled frequently as necessary. The central bearing upon which the drum rotates, is subject to much wear and should be frequently cleaned and oiled. After using the instrument, thoroughly clean and oil it, being especially particular in regard to the springs as a very slight rust impairs their accuracy. It is of importance that an indicator be kept in good order and to determine that it is properly adjusted and constructed, the following is a good method. With the indicator in a vertical position and without the spring (with piston attached); raise the pencil motion to its highest position and when released it should drop with freedom from friction to its lowest position. The piston, cylinder and oil must be warm, otherwise the piston is apt to stick. As a further test, again raise the pencil motion to its highest position and cover the hole through which steam is admitted to the indicator with the thumb when the pencil should sink at a slow and uniform rate through the whole range of its motion. Should there be a suspicion of a catch from any position within this range, detach the piston from the pencil motion and try the pencil motion and piston separately. If both are in good order the fault lies where the piston is attached to the pencil motion. A piston which is too free may occasion serious defects from leakage, especially in condensing engines. The pencil motion should be parallel to the drum surface and the drum be perfectly round and properly centered. By adjusting the pencil stop so that only a ray of light can be seen between the pencil point and the drum surface, the piston is moved upward (with a lead pencil) and the pencil point should be carefully watched to see that it keeps the same distance from the drum surface. With the pencil, both in the highest and lowest position, rotate the drum through its whole range by drawing out the cord and again watch

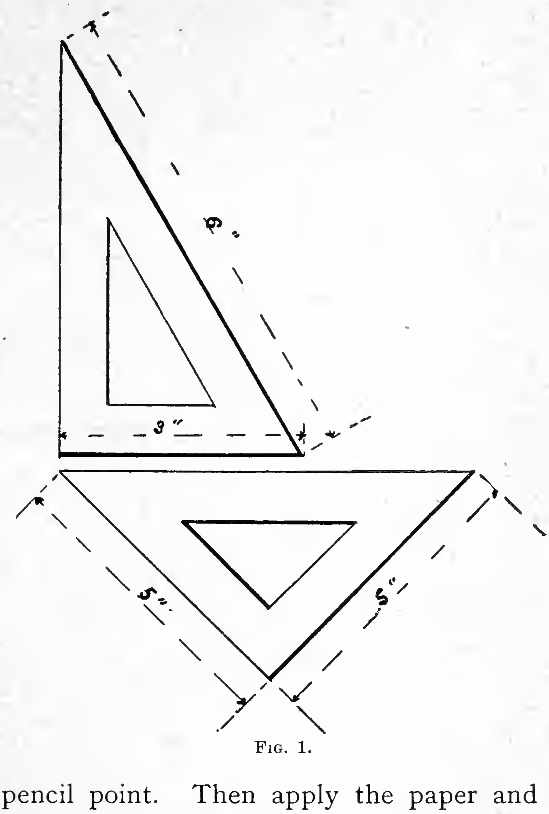

Fig. 1.

the pencil point. Then apply the paper and allow the pencil to touch the paper and repeat the operation by forming two horizontal lines and one vertical line.

If the reduction is correct, the lines drawn will be straight throughout their whole length and at right angles to each other which may be proven with the angles as shown in Fig. 1. Those made of celluloid are preferred. To determine that all joints are without play, insert a spring of high tension and carefully feel at the end of the pencil bar. In case two indicators are used they should be either both right or both left hand.

PAPER.

The paper used with the indicator should be of good quality, having a smooth surface and capable of receiving an impression with the lightest possible pressure of the pencil against the paper. Paper having a glossy surface should be avoided. The paper should reach to the top of the drum and about one inch longer than the circumference of the drum. Printed blanks containing spaces for recording various observations for future references are preferable. Those used by the writer are prepared for the particular test on which they are to be used. Those shown in Fig. 2, are shown one-half size and arranged for a

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DIAGRAM FROM________ ENGINE AT________

DIAM OF CYLINDER____	PER CENT. OF CLEARANCE____	BOILER PRESSURE.____	VAC. PER GAUGE, INCHES____
DIAM OF PISTON ROD____	STROKE IN INCHES____	INITIAL PRESSURE.____	BAROMETER READS____
DIAM OF STEAM PIPE____	REV. PER MIN.____	BACK PRESSURE.____	TEMP. HOT WELL____
DIAM OF EXHAUST PIPE____	SCALE OF SPRING____	MEAN EFFECTIVE PRESSURE.____	INDICATED HORSE POWER____

DATE____98.__TIME____

HEAD OR CRANK____

REDUCING MOTION____

INDICATOR____

No. ____

REMARKS

Fig 2.

drum two inches in diameter, will usually be sufficient for the engineer. Remember, that without all necessary data the diagram is worthless except as an indication of the valve setting. In getting the correct data in making indicator tests remember the golden rule is, never take anyone's word for anything. But note these observations yourself which must be undertaken with a degree of precision, as many a man's evidence in court has been disallowed for want of sufficient and reliable data.

The paper should be placed so that the clips hold it on the drum as smooth as a glove. To accomplish this, close to the clips, with the thumb and forefinger pinch the ends of the paper together. Never attempt this operation by pulling on the ends of the paper, as it springs the clips outward which in turn sets the paper away from the drum.

PENCIL.

Use hard lead and of good quality such as usually accompanies the indicator. It should be sharpened with fine sandpaper to a smooth point, as fine lines are preferable for their distinction. The lead should project through the socket as little as possible, so as to avoid undue twisting of the pencil bar. The pressure of the pencil on the paper should be just sufficient to make a legible mark, a greater pressure only creates undue friction. The friction of the pencil tends to lessen oscillations, but at the same time it falsifies the diagram. A more satisfactory result can be obtained by using a metallic point in place of the pencil and it, like the lead, should be as light as possible, for, its weight acting at the end of the lever where the velocity of motion is at its maximum will tend to increase oscillations. It should be sharpened to a fine point, rounded off on the end so as not to tear the paper while at the same time make a fine but distinct line. The metallic cards are chemically prepared for the metallic points.

SPRINGS.

In order to obtain a correct diagram, the height of the pencil (caused by the compression of the spring) must exactly represent in pounds pressure per square inch the pressure on the piston of the engine at any point of the stroke and the velocity of the drum must at all times bear a constant ratio to that of the piston.

In selecting a spring to suit different conditions, as to steam pressure, speed, etc., the aim should be to obtain a diagram about two inches high on slow speed engines and about one and one-fourth inches or less on high speed. This is accomplished by using a spring that is numbered one-half the highest pressure. Thus, if a 40 spring is used, forty pounds pressure per square inch will raise the pencil one inch, and eighty pounds two inches and so on for other pressures. The spring is the measuring factor of the indicator and the most vital part of the instrument. Indicator springs give to-day, as a rule, more power on an engine than they would give if the springs were correct. For tests of importance the scale of deflection should be verified before and after each test. The rule for giving the maximum pressure to which each spring is subjected varies with different makes of indicators; therefore it is not given here.

CHAPTER II.

APPLICATION OF THE INDICATOR.

The application of the indicator to an engine is a scientific experiment and should be undertaken with a degree of care and precaution against error, as the value of the diagram depends on its correctness. The value of the indicator is now so generally recognized that every concern which pretends to manufacture an engine makes provision for its application. When no provision for its application has been made, the cylinder must be drilled and tapped for not less than one-half inch pipe thread, in the side where the pantograph reducing motion is used, but usually on top for high speed and in such position in the clearance space that the holes are not the least covered over by the piston at the end of its travels. Should the clearance space be too small to allow for this, access may be had by chipping a channel in the cylinder head from the tapped hole out into the clearance space. In drilling, the piston should be taken out and the cylinder cleaned from chips and cuttings. The indicator should be connected as directly as possible to the cylinder. For the reason that it requires a difference in the pressure in order that the steam can flow from the engine cylinder into the indicator, thus causing a loss of pressure in the steam line and an increase of pressure in the expansion curve and with long connections, such as the use of side pipes connected at the middle with a three-way cock as shown in Fig. 3, or still worse a T with an angle valve at each end which is one

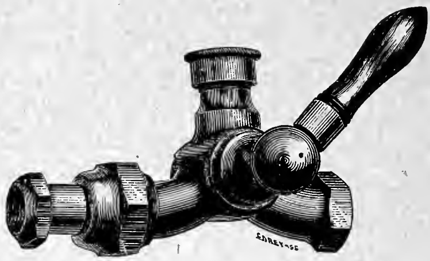

Fig. 3.

of the most objectionable forms, the results will show up later in the indicator than when it took place inside of the steam engine cylinder.

The cock should be screwed into the cylinder itself. When the cylinder is tapped upon the side this will bring the instruments horizontal. Where nipples and elbows are used to bring the indicator into a vertical position, the ends of the nipples must be reamed out, as the burr made in cutting it if not reamed wire-draws the steam. Use no leads or paints in connecting, it is not necessary and they are liable to get into the instrument. When, however, a single indicator is to be used upon both ends of the cylinder, and where engines are constantly changing their load, the three-way cock is sometimes used to equalize the work between the two ends of the cylinder. The most proper arrangement is to have an instrument upon each end of the cylinder from which simultaneous diagrams may be taken.

REDUCING MOTION.

Whenever the stroke of the engine is greater than the length of the diagram desired, some means must be provided to reproduce upon a reduced scale a perfect duplicate of the piston movement, many devices in use fail to do so, and the diagrams taken with them are incorrect. The pantograph, as shown in Fig. 4, is a correct reducing motion, and various lengths of

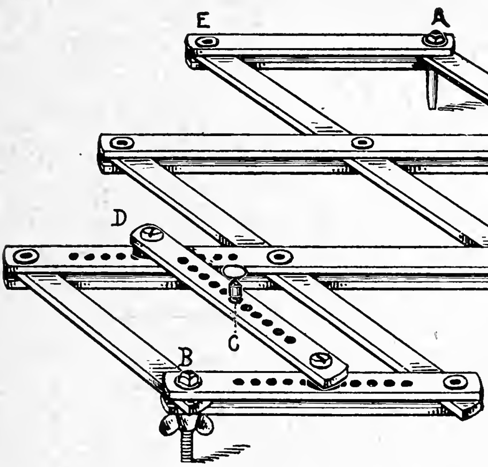

FIG. 4.

diagrams may be had by changing the position of the cross-bar D, moving it toward the end pivot A for longer diagrams, and toward the end pivot B for shorter ones. The cord pin C must be in line with the end pivots A and B. The cord pin is grooved for the application of the cord, but its diameter is usually so great that it tends to throw the cord. To prevent this it should be regrooved to a diameter of about one-eighth of an inch. The joints can readily be tightened when they are too loose, by upsetting the rivet heads. When not in use, the whole apparatus should be kept in a bath of oil which at the same time lubricates the joints. The arrangement for sup-

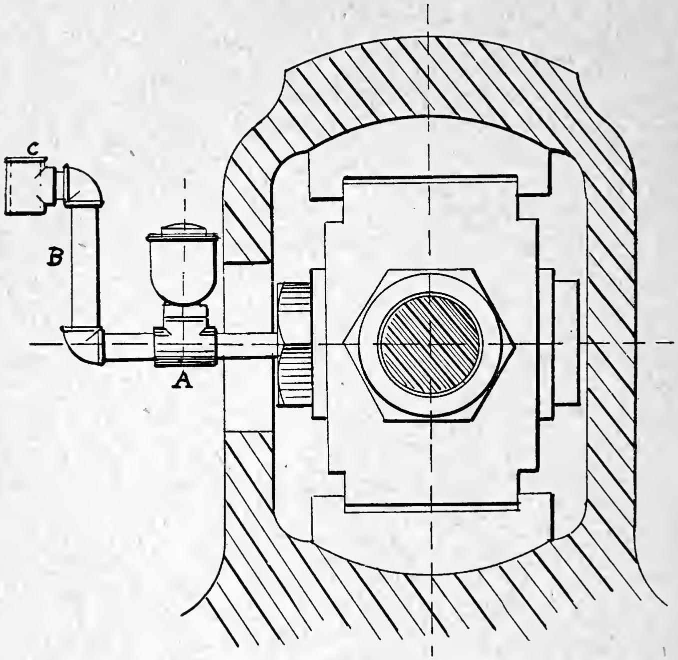

FIG. 5.

porting and attaching the pantograph differs as the ingenuity of the operator may suggest.

The simple device as shown in Fig. 5 is constructed of half-inch nipples and fittings. A is screwed into the side of the crosshead, the oil cup being removed and placed in the tee. The length of A should be about four inches so as to prevent the corners, E. E. of the pantograph from coming in contact with the crosshead. The height of B should be about four inches so as to bring the pantograph to the

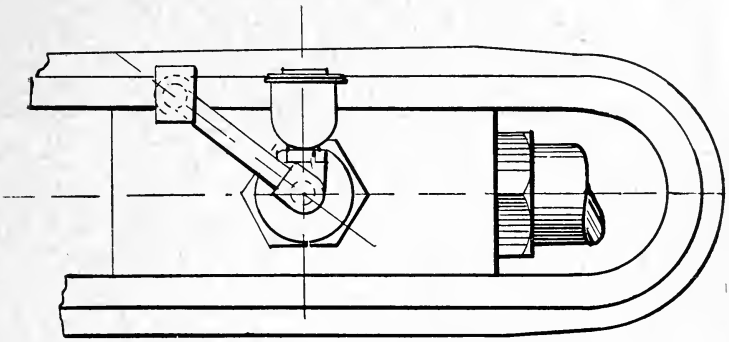

Fig. 6.

proper level. B can be turned at any angle, as shown in Fig. 6, to suit various sizes of engines. A wooden

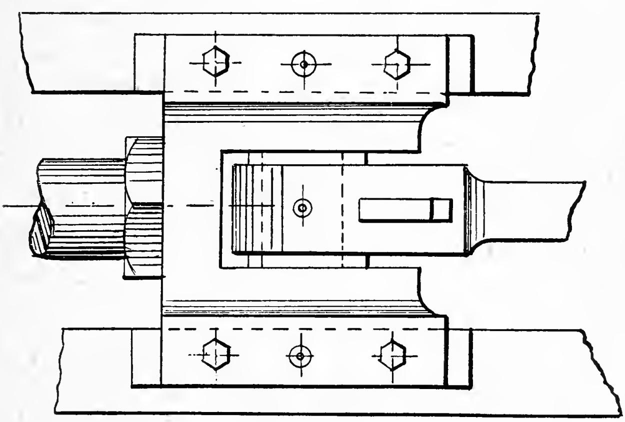

Fig. 7.

plug is driven into C and bored out to receive the end pivot A of the pantograph.

On crossheads similar in style to Fig. 7, A can be made of round iron, flattened and slotted near the end as shown in Fig. 8 when it will fit any bolt and can be clamped fast to one of the bolts of the crosshead. Fig. 9 shows the arrangement for supporting the end pivot B of the pantograph. This device is constructed of about 4 feet of $\frac{3}{4}$ inch gas pipe which can be made up with couplings so that it can be disconnected into parts, thereby rendering it convenient in moving from place

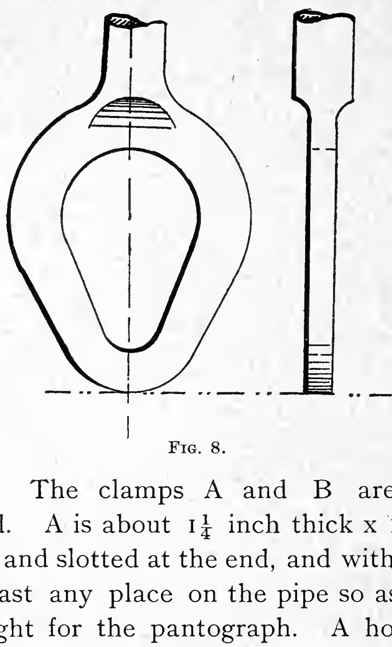

FIG. 8.

to place. The clamps A and B are made of tough wood. A is about $1\frac{1}{4}$ inch thick x $1\frac{1}{2}$ wide x 6 inches long and slotted at the end, and with a bolt can be made fast any place on the pipe so as to get the correct height for the pantograph. A hole is made near the other end for the end pivot B of the pantograph. The floor plate is made fast to the floor with screws. One brace is parallel with the engine and made fast to the floor and the short clamp B and another brace at right angles made fast near the top with a similar clamp or into a wooden plug which is driven into the top of the pipe while the other end of the brace is usually fastened to the wall or over on to the engine frame. It is immaterial whether the stationary end of the pantograph be placed in the center of motion or not so far as correctness goes and makes no difference whether the pantograph is set horizon-

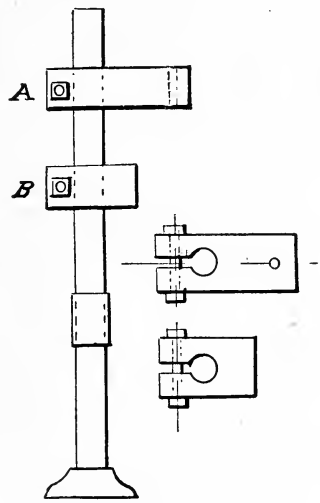

FIG. 9.

tally, perpendicularly or oblique so long as the corners E. E. of the pantograph will clear.

Some writers claim that the stationary end of the pantograph must be precisely in the center of motion, which is a mistaken idea, and has caused many engineers to measure with precaution against error to locate this point correctly. Owing to the complicated construction of the pantograph, it cannot be used on high speed engines, and is frequently replaced by a light wooden pendulum pivoted to a fixed support usually a $\frac{3}{8}$ inch gas pipe screwed into the engine guide as shown in Fig. 10 and fitted at its lowest extremity with a pin which is carried by a slot made fast

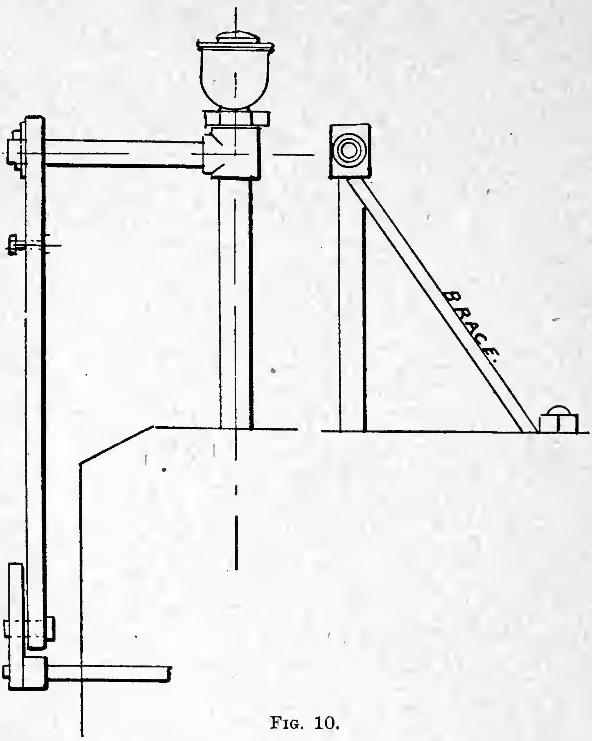

Fig. 10.

to the crosshead as shown in Fig. 11. The length of the pendulum should be about one and a half times the stroke of the engine. The cord is looped or hooked on a pin which must be in the center line of the pendulum and on a point on the pendulum to correspond with the length of diagram, and the cord must be precisely at right angles to the pendulum when it is in the middle of the stroke. The point of support must be precisely in the center of motion and must not vary to the edge of a knife blade otherwise it will

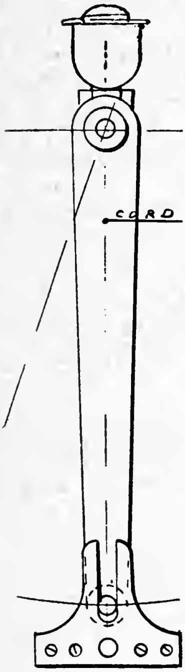

Fig. 11.

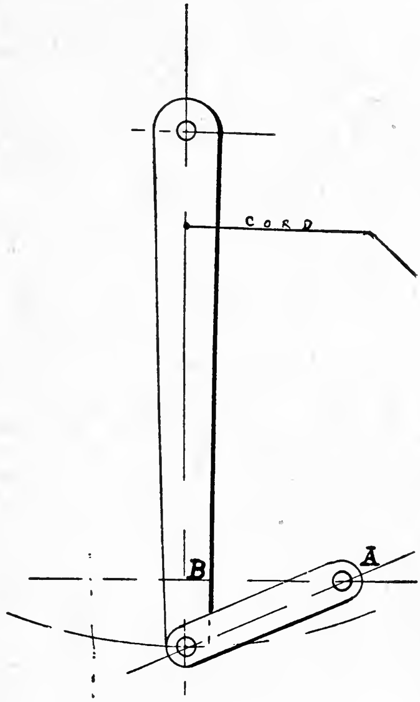

Fig. 12.

distort the diagram and the results will appear unequal if they really are equal.

When a link is used in connection with the pendulum as shown in Fig. 12, the link should be about one-half the stroke. The pendulum should be arranged so that the link will incline as much below the path of the crosshead when in the middle of the stroke as it inclines above the path when at the end of its travel. The fixed support of the pendulum will then be at a point as far from the support of the slotted pendulum as the distance from A to B. Frequently provisions are made near one end on the guide for making the support fast, and in order to have the same inclination, the relative length of the pendulum and link must be such that at midstroke the pendulum is perpendicular, when it is a horizontal engine, and at right angles to the plane in which the crosshead moves.

While both of the motions last mentioned are frequently used neither of them are correct. The use of a segment of a pulley, sometimes called the Brumbo pulley, introduces rather than eliminates an error. The correctness in the reduction of any reducing motion can readily be determined by placing the engine so that the crosshead is at either end of its travel with reducing motion and indicator properly attached. First, draw a horizontal line by pulling out the cord and without the spring make a vertical mark upon the card by raising the pencil and with the engine in this position make a mark on the crosshead and guide. Space off as many marks on the guide as the length of stroke is in inches, move the engine so that the mark on the crosshead coincides with each mark on the guide, and at each movement make a corresponding mark on the card. If the lines are equally spaced, as shown in Fig. 13, the principle of the reduction is correct, while its practicability when at full speed may be radically distorted. If so, it is due to several causes, but principally vibration of the reducing motion or indicator. Frequently cord stretching or throw of the indicator drum will cause a distortion of the diagram which can be tested by first taking the atmospheric line while the engine is run-

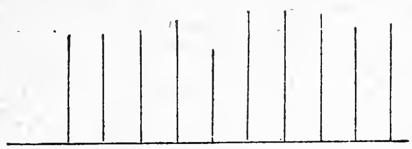

FIG. 13.

ning slow, then move the card upward about 1-16 of an inch and take another line when the engine is at full speed. If there is a difference in the length of the lines, distortion is generally due to one or more of these causes.

CHAPTER III.

CORD.

The movement of the paper drum is derived from a closely braided hemp or metallic cord, all of which are more or less elastic. The result of the elasticity "in the hemp cord" is that at the end of the stroke, the cord having to overcome the throw of the drum, the tension of the drum spring stretches the cord more or less at the commencement of the stroke. The drum loses a part of its movement at the commencement of the stroke, but gradually regains its curtailment at the end of the stroke by giving the drum a throw, thereby affecting a motion proportionate to that of the piston movement. The result of the elasticity in the metallic cord is that the drum retains its proportional movement to the end of the stroke when the throw of the drum produces an elongation of the diagram. The hemp cord is therefore the most efficient to eliminate this distortion When any great length has to be used, it is well to partly substitute metallic wire for hemp cord. The braided cord is supplied by dealers in the instruments; but before using, it should remain suspended for at least one day with a heavy weight at its lowest end to take out any undue stretch which it may contain, after which it should be treated with a light coat of beeswax, when it will not be easily affected by moisture and will longer retain its proper condition. The cord should be looped over the cord-pin on the reducing motion which permits the pin to turn easily within it, and should be hooked and unhooked at that point. Nearly every engineer at first expects a difficulty in hooking and unhooking the cord on high speed engines, which is merely supposition. The writer has taken diagrams on some of the highest speed engines built and never experienced the slightest difficulty. When a hook is used it must be light and hooked upon the cord-pin itself, or as near the instrument as possible. The device frequently used for adjusting the length of the loop and cord, and usually made of a small piece of stiff metal in which are drilled four holes for the cord to pass through, thereby forming the loop, is a device about as worthless as can be made, and when it is on the middle of the line, together with the heavy hook now so common in use, it is in its most objectionable form, and will cause the cord to whip. The length of the diagram on slow speed engines should be about four inches and about two inches high, as large diagrams will show up the defects to greater advantage. The length of diagram desired can easily be determined by taking hold of the cord in one hand and allow it to pass over a rule held in the other hand with the engine running slow. When a hook and loop are used, the length of the cord is adjusted so that the hook clears the loop about one inch at the end of the stroke, and since a drum two inches in diameter unwinds the cord about six inches, this will give a clearance of one inch to the drum at each end of the stroke, when the diagram will come in the center of the card. The proper tension of the drum spring is of importance. If too little tension it will cause the drum to throw past at the end of the stroke when the cord will be constantly whipping, and if the tension is too great the drum loses a part of its stroke, a matter which can be easily tested. With the engine running slow and with a tension on the drum spring, so that it neither causes the cord to stretch nor the drum to throw, take the atmospheric line, move the card up about 1-16 of an inch, put the engine up to speed and readjust the drum spring to the supposed proper tension and take another atmospheric line. If the last line is the longest the tension of the drum spring is too little, which caused the drum to throw, and if shorter the tension is too great.

It is desirable to avoid the use of leading pulleys as shown in Fig. 12; however, when it becomes necessary in order to lead the cord off parallel with the guides, a leading pulley must be used and from this the cord may be carried in any direction. For high speed engines the length of the diagram should be about 2 to $2\frac{1}{2}$ inches and from 1 to $1\frac{1}{2}$ inches high and it is desirable to use a lighter and smaller drum on high speed engines.

HOW TO ATTACH THE INDICATOR AND TAKE THE DIAGRAMS.

Before attaching the indicators, steam should be blown through the cocks so as to remove any grit which may be in the connections. Fig. 14 shows the indicators properly attached to a Corliss engine, for taking steam pipe and cylinder diagrams, with the

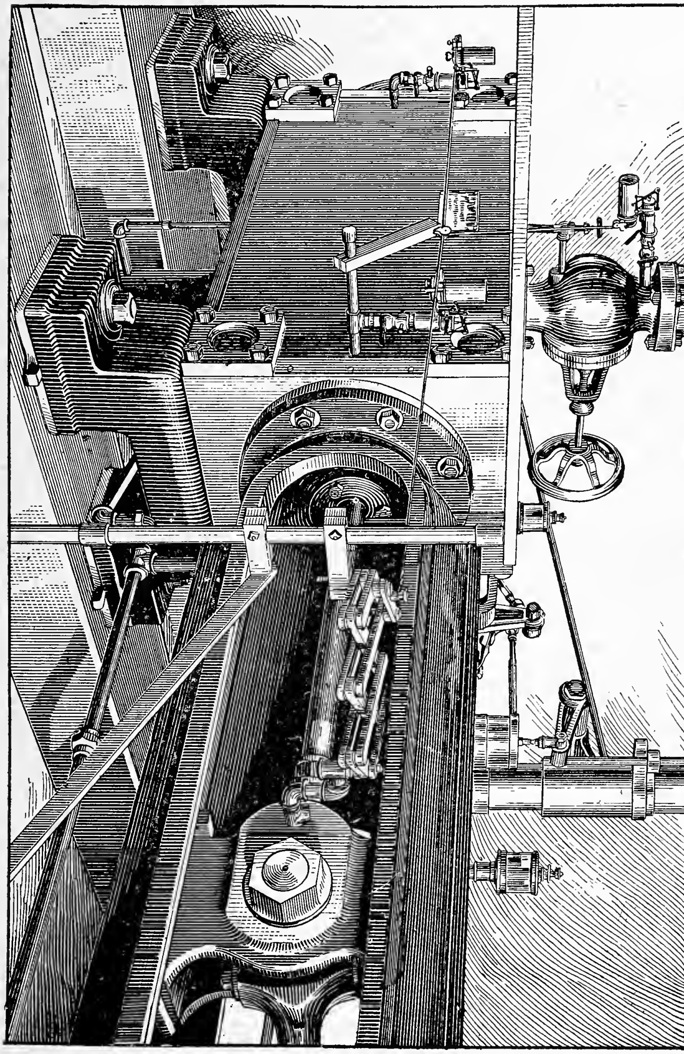

Fig. 14.

pantograph reducing motion and its attachments complete. Before taking a diagram, steam should be admitted into the indicators during several revolutions of the engine and when dry steam blows through the relief, apply the pencil during one revolution, but where the load is constantly changing and when the power is to be computed from the diagrams, the pencil should be applied during as many revolutions as is necessary in order to obtain both the lightest and heaviest load, from which the average horse power may be ascertained. After taking the diagrams, steam should be shut off from the indicators and the atmospheric line should be taken immediately, before the spring contracts, as the indicator piston rod and spring expand when in contact with steam and contract when steam is shut off, due to the difference in temperature. The result is that the expansion of the piston rod raises the pencil and the expansion of the spring lowers it, and when steam is shut off, the contraction of the piston rod and spring affects the pencil *vice versa.* Now if the expansion and contraction of the rod and spring were the same, one would balance the other, but they are not the same, consequently, the atmospheric line should be taken immediately after the diagram, so that the conditions in both rod and spring may be as nearly the same as possible. After unhooking the line, let the indicator drum revolve gently back against the stop.

TESTING STEAM GAUGE WITH THE INDICATOR.

To determine the loss of pressure between the boiler and engine, the steam gauge and indicator must agree, or if any variation, it should be known. Fig. 15 shows the indicator and steam gauge properly connected for making this test. They must be attached where there is no fluctuation in pressure. A conven-

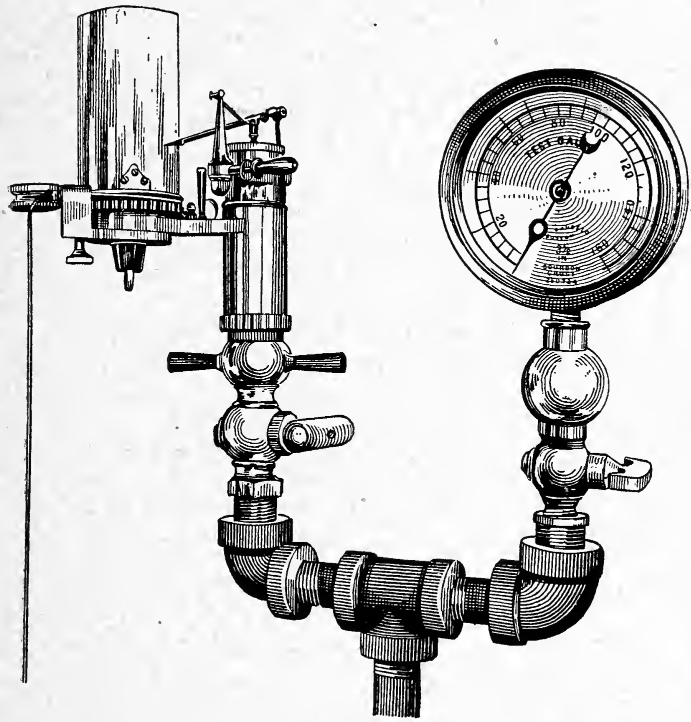

Fig. 15.

ient place is above the throttle valve when steam is shut off from the engine. To take this diagram turn steam on the gauge and indicator and when they are well heated up take the steam pressure line by drawing out the cord and at the same time note the reading of the steam gauge. Shut off steam from the indicator and take the atmospheric line. Fig. 16 shows the diagram taken with 40 spring. Measure on the scale of the spring from the atmospheric line to the steam pressure line which is 80 pounds and if the

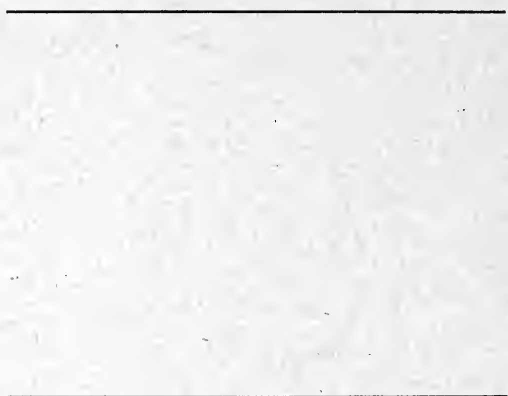

FIG. 16.

gauge registers the same they agree for this pressure, while for other pressures they may disagree, as an indicator spring and a steam gauge vary more at some pressures than at others. If the gauge registers the greater pressure it is light, and if less it is heavy, providing the indicator spring is correct.

CHAPTER IV.

DIAGRAM ANALYSIS.

An indicator diagram is the result of two movements, namely, a horizontal movement of the paper and a vertical movement of the pencil, and therefore, 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 a corresponding point in the stroke. A single diagram shows the pressure acting on one side of the piston during both the forward and return stroke. To show the corresponding pressure on the other side it is necessary to take another diagram from the other end of the cylinder.

The following definitions have been given to the different parts of the diagram proper and to lines added, as required for purpose of analysis. Fig. 17 shows the general features of a well formed indicator diagram from a Corliss non-condensing engine, the attainment of which should be the aim in setting the valves of an engine. Line A B is the admission line which indicates lead only, and steam should not be admitted into the cylinder until the piston has completed its stroke. But before the piston moves away, the steam valve should open and admit the pressure so as to erect the admission line perpendicular to the atmospheric line. This can be proven by placing one side of a right angle, parallel with the atmospheric line and the other extending a trifle beyond the admission line. Then draw line I. If the admission line is parallel with this line, the admission is perfect. If

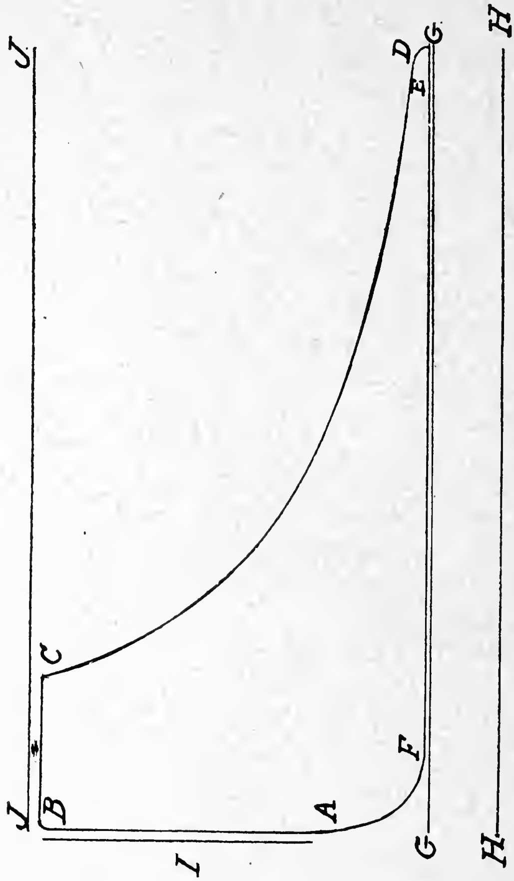

FIG. 17—SCALE OF SPRING 40.

the admission line leans inward, the admission is too late and causes loss of power and economy. If the admission line leans outward, the admission is too early, which by all means must be avoided, unless it is necessary in order to obtain the necessary cushion when the cushion from the exhaust steam is insufficient, which frequently happens with overloaded engines or condensing engines, as it forms a counter motion on the piston which also is a loss of power and economy, and were it not for the motion of the fly wheel in assisting to bring the crank beyond the center, the engine would stop. If there is any choice between a too early and a too late admission it is better to have it too late. Line B C is the steam line which indicates the stroke of the piston to the point of cut-off. An engine that can maintain the nearest to a straight steam line is the most efficient.

The initial pressure is the first pressure realized in the cylinder at the commencement of the stroke, and is measured on the scale of the spring from the atmospheric line to the highest true point at the commencement of the steam line. The initial pressure should maintain to the point of cut-off and approach the boiler pressure as nearly as possible, and if the boiler and engine are a reasonable distance apart and all steam passages are ample, there should not be a greater loss of pressure than from 2 to 3 pounds. A greater loss is principally due to insufficient area through the steam pipe, steam chest, and ports (which can be located with the indicator) and generally follows with rapid falling of the steam line which is sometimes called wire-drawing. It is also due to condensation caused by too many expansions. J J is the boiler pressure line and is located by measurement on the scale of the spring at such a distance above the atmospheric line as to represent the boiler pressure at the time the diagram was taken, and the loss of pressure between the boiler and engine is measured from the steam line to the boiler pressure line. C is the point of cut-off where the steam valve closes automatically by action of the governor, either by changes of pressure of steam or changes of load. When the valve closes it should be quick in its action so as to form the point C as sharp as possible. A valve that is slow in closing reduces the pressure at this point which is a loss of economy. After the valve has closed there should be no leakage of steam in the valve, as this leakage is a partial loss of economy on the admission side of the piston and a total loss on the exhausting side except during compression. The leakage of valves—when it exists—will be the greatest immediately after the valve closes, when the difference of pressure in the cylinder and the steam chest is the least, as the valve is not held so firm to the seat. C D is the expansion curve comparing it with what is known as Mariotte's law, according to which, its volume and pressure are inversely proportional to each other, and for elementary sake will assume that an engine has no clearance and that steam follows this law. And if 100 pounds absolute pressure be admitted into the cylinder of an engine and cut-off at $\frac{1}{4}$ stroke, when the piston has moved to $\frac{1}{2}$ stroke, the volume will be doubled and if there be no loss in pressure from condensation or gain from re-evaporation, leakage of piston, valves, etc., the pressure would be 50 pounds absolute; when the piston has completed its stroke, the volume will again be doubled from what it was at $\frac{1}{2}$ stroke and the pressure will be 25 pounds absolute, and so on for other pressures and volumes. But steam does not follow this law, as it is a known fact that it falls in temperature during expansion and rises during compression and this change of temperature augments the change of pressure so that as before assumed, 100 pounds absolute pressure be cut-off at $\frac{1}{4}$ stroke instead of falling 50 pounds at $\frac{1}{2}$ stroke will fall to a trifle below 48 pounds. The absolute pressure is the gauge pressure plus the atmospheric pressure and is the basis for all calculations for expansion and compression.

The object of expanding steam is to obtain economy and therefore it should be expanded to its greatest reasonable extent. If steam is expanded down to the atmospheric pressure so that all its force has been utilized in doing work it will lower the temperature of the cylinder walls, etc., to such a degree that it will result in too great condensation of the entering steam and will be a loss rather than a gain, and if released at too high pressure, when it could yet do useful work, will also result in loss of economy. The most suitable release pressure depends upon several conditions and should be in accordance with the initial pressure so as to effect an economical range of temperature in the cylinder.

D is the point of exhaust opening and should be as near the end of the stroke as possible. The exhaust valve should open quickly so as to release the pressure and bring it down where it belongs before the piston starts to return.

The release pressure is measured from the atmospheric line to the point D, and represents the pressure at which steam is released. The terminal pressure is the pressure at the end of the stroke. EF is the back pressure line and is measured from the back pressure line to the atmospheric line. If the exhaust passages are ample there should be no more back pressure than a fraction of a pound due to friction in the passages, etc., unless the exhaust be utilized for other purposes than that of heating the feed water, because back pressure represents power and fuel thrown away as it requires just that much more pressure on the admission side of the piston to overcome it and if carried to extreme, the engine would stop. Excessive back pressure is due principally to insufficient area in the heater, exhaust pipe, and ports and can be located with the indicator. Many exhaust pipes have not sufficient size drip to allow the water of condensation to properly escape without partly filling the pipe, thereby reducing its size and causing back pressure. I have found some that have no drip which is still worse. If there were no back pressure the back pressure line would merge into the atmospheric line.

F A is the compression curve and is formed by closing the exhaust valve at the point F before the piston completes the stroke thereby imprisoning the exhaust steam which is compressed as the piston completes the stroke. The compression is measured from the atmospheric line to the point A which effects an equal amount of cushion per square inch on the piston only at the time when the pressure on the other side of the piston is equal to that of the atmospheric which forms an elastic cushion to stop the piston and its reciprocating parts at the end of the stroke.

Engines running from 60 to 200 revolutions per minute may require from about 10 to 45 pounds cushion. The amount of cushion per square inch on the piston depends principally on the weight of these parts and the speed of the engine, and no more cushion should be given than is required to accomplish this result, as a greater amount has the same defect as too early admission. It takes an additional amount of steam on the admission side of the piston to form compression; but if there were no loss of heat from condensation during compression and expansion, and the steam be expanded to the same pressure as that when the exhaust valves closed, there would be no loss from compression. Condensation, however, exists which causes loss of economy from compression; yet if the clearance space be filled with live steam, whose temperature is greater than that of the cylinder walls, etc., it would also result in loss of economy from condensation; therefore to obtain the greatest economy direct from compression it should be at a temperature that does not exceed the temperature of the cylinder walls, etc. Cylinder condensation has led some engine builders, as well as engineers, to believe that a steam jacketed cylinder is more economical than an unjacketed one, but this is a mistaken idea and whenever tests have favored the steam jacketed cylinder, the conditions have been unfavorable for the unjacketed cylinder.

G G is the atmospheric line, from which all pressure above and vacuum below must be measured.

H H is the vacuum line and is located by measurement on the scale of the spring at such a distance below the atmospheric line as to represent the atmospheric pressure in pounds per square inch at the time and place; but the pressure is constantly changing, at sea level the average being 29.922 inches, or about 14.7 pounds and for higher altitudes, correspondingly less. The reading of the barometer in inches, multiplied by .49 will give the pressure in pounds.

CHAPTER V.

CUSHION.

The cushion is the highest unbalanced pressure per square inch on the piston at the end of the stroke, and serves to stop the piston and the reciprocating parts at the end of the stroke, so as to relieve the connections from undue strain.

Suppose it is desired to determine the cushion per square inch on the piston of Fig. 18. H represents the head end and C the crank end of a Corliss engine. The compression on the head end is 20 pounds, which is sufficient cushion per square inch on the piston for engines running at about the speed of the average Corliss engine; but the pressure on the other side of the piston when it comes to a standstill must be deducted, and should be measured on the crank end diagram, at such point as shown by the dotted line, which is 10 pounds. Therefore, the cushion is only 10 pounds and not 20 pounds, as shown by the compression. This is a mistake often made by some of our best engineers, as well as some of the so-called indicator experts. When this pressure is the same as the compression then the pressure is equal on both sides of the piston and there is no cushion. This frequently happens when the engine is overloaded. When engines are taking steam nearly full stroke, it will cause them to pound, not because they are working hard, as is so often surmised by some engineers, but from the fact that this pressure may be even greater than the compression, when the tendency is to send the piston still further, were it possible to do so. This is an important factor which has been too long ignored, and has caused

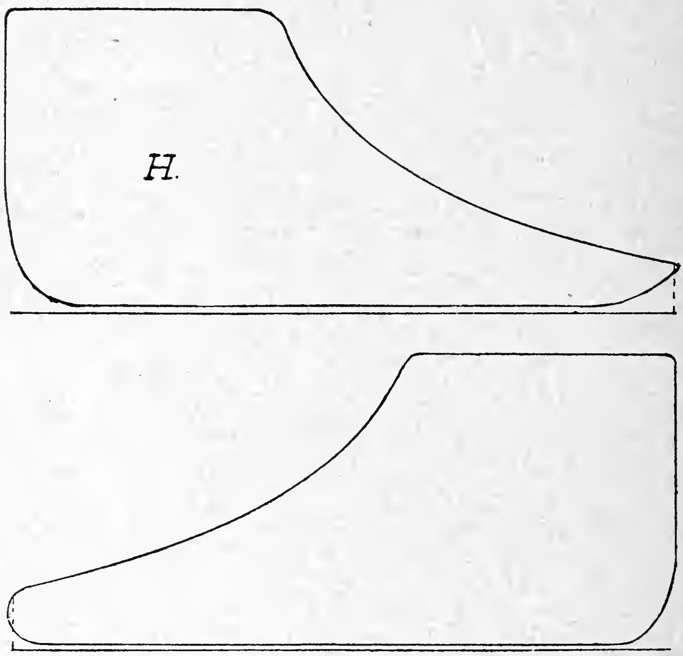

FIG. 18—SCALE OF SPRING 40.

many engines to run noisy when it could be easily prevented. The cushion on the crank end is determined in like manner, by measuring the compression of the crank end, from which deduct the pressure of the head end at a point as shown by the dotted line. If the exhaust valve rods be lengthened, so as to close the exhaust valves earlier, it will cause a higher compression, but would result in later opening of the valves; therefore, what is gained in compression by closing the valves earlier, is partly lost by late opening on the other side of the piston; besides, it robs the engine of power and economy. Right here we begin to appreciate engines that are equipped with two eccentrics; however, with the single eccentric, some means must be provided to obtain sufficient cushion to bring the piston and the reciprocating parts to a state of rest at the end of the stroke. This can be accomplished by raising the boiler pressure providing the boiler will permit, which will cause an earlier cut-off and a lower terminal pressure, as well as an increase of economy. If the boiler will not permit, the eccentric should be advanced so as to admit live steam into the cylinder just before the piston completes the stroke, which assists the compressed exhaust steam in forming cushion. This is shown when the admission line on the diagrams lean slightly outward, but pains must be taken in doing this work, so as not to admit steam too early and cause a counter motion on the piston.

Fig. 19 also has twenty pounds compression. To determine the cushion on the head end, the partial vacuum on the other side of the piston at the instant it comes to a stand still must be added. This is measured on the crank end diagram, at a point as shown by the dotted line, which is ten pounds, there being ten more pounds atmospheric pressure on the head end, to which is added twenty pounds compressed exhaust steam, giving thirty pounds cushion, or three times as much cushion as Fig. 18; yet, the exhaust valve closed at the same point. When this is carried to extreme, it has the same effect as a too early admission and will cause the engine to pound.

The cushion of the crank end is determined in like

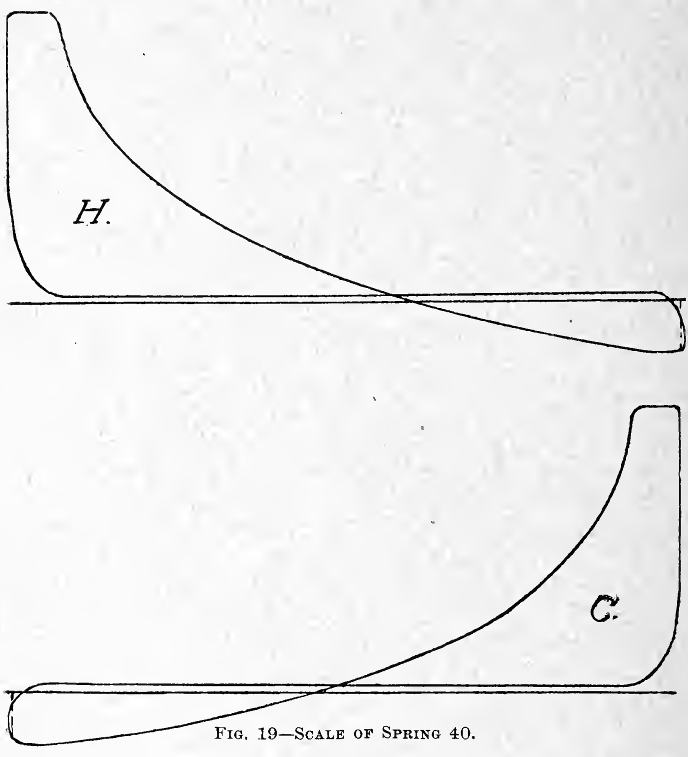

Fig. 19—Scale of Spring 40.

manner, by adding the partial vacuum of the head end to the compression of the crank end. The loop in the diagrams is caused by a too light load and the expansion curve falls below the atmospheric line, showing a loss of economy, as it takes that much more steam on the admission side of the piston to overcome it, as it forms a counter motion on the piston, and were it not for the motion of the fly wheel, the engine would stop. This can be remedied by lowering the boiler pressure until a suitable release pressure is obtained. When the back pressure is enough greater than the partial vacuum formed in the cylinder near the end of the stroke as represented by the

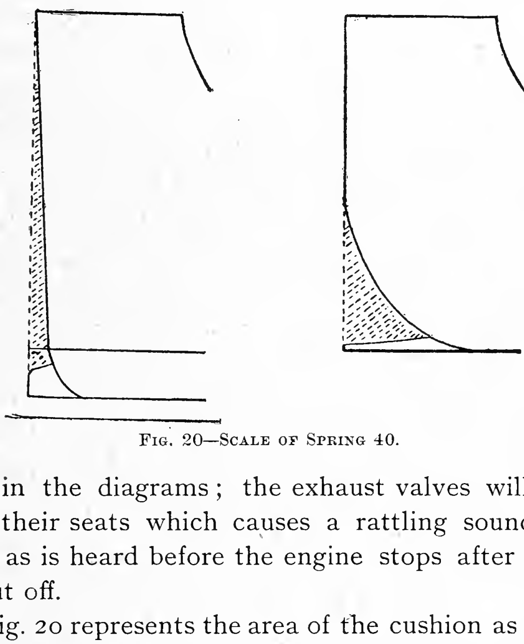

FIG. 20—SCALE OF SPRING 40.

loop in the diagrams; the exhaust valves will raise from their seats which causes a rattling sound, the same as is heard before the engine stops after steam is shut off.

Fig. 20 represents the area of the cushion as shown by the dotted portions and can be accurately determined with a planimeter; but unless the diagrams are both taken on the same card the latter end of the expansion curve of the other diagram must be transfered from the point where it should cross the comparison curve of this diagram. This can be done accurately by making a pinhole at each end of the atmospheric line of each diagram; then place the diagram to be transferred on a piece of glass and the other directly above it so that the pinholes match and when viewed from a strong light the pencil mark of the diagram underneath can be plainly seen and in this manner can be conveniently transfered.

Supposing that the same steam pressure be admitted into the cylinder of an engine during the entire stroke when it would do a certain amount of work while at one-half stroke with the same pressure it would do one-half the work and so on. So it is with the cushion as it depends as much upon the length of the area embraced in the cushion as the height. In Fig. 20 the diagram shown at the left representing a diagram from a Corliss condensing engine where it was necessary to make up the deficiency in cushion by admitting live steam into the cylinder before the piston had completed the stroke as shown by the admission line leaning outward, thus admitting the full pressure or 75 pounds cushion which is measured from the top of the termination of the release curve of the diagram from the other end of the cylinder as shown, while the cushion of the non-condensing diagram as shown at the right is 29 pounds which is formed by compressed exhaust steam. Yet the area in the shaded portion of this diagram is greater thus showing that this diagram has greater retarding power to stop the piston while the height of the cushion on the other diagram is 46 pounds greater.

To admit the full pressure before the piston has completed the stroke so as to make up the deficiency in cushion when it becomes necessary, is regarded as an objectional feature by some engineers, but it will be seen from the above illustration that it is practical to make up the deficiency in cushion with live steam when the cushion from the exhaust steam is insufficient. But as I have before stated, pains must be taken not to admit the pressure too early and cause a counter motion on the piston.

In setting the valves on Corliss engines without the indicator, the question as to whether or not to lap the exhaust valves, has been frequently discussed the world over, but like all other types of engines, it depends upon too many conditions to lay down a definite rule ; however, it depends principally upon the speed of the engine ; weight of the piston and the reciprocating parts ; pressure in the cylinder at the time the exhaust valves closed, pressure on the expanding side of the piston at the time the piston comes to a stand still ; temperature of the cylinder walls ; the amount of clearance, and with small amount of clearance ; high temperature of cylinder walls ; low pressure on the expanding side of the piston, when it comes to a stand still, and high back pressure in the cylinder at the time the exhaust valves closed, all of which will cause higher cushion and a closing of the exhaust valves at the same point.

When setting valves without the indicator, it is well to examine into these conditions, but at best, it is merely supposition and guess work.

DIAGRAMS FROM CORLISS CONDENSING ENGINE.

Fig. 21 shows the general features of well formed indicator diagrams from a Corliss condensing engine. The object of attaching a condenser to an engine is to obtain greater economy, but it will show economy

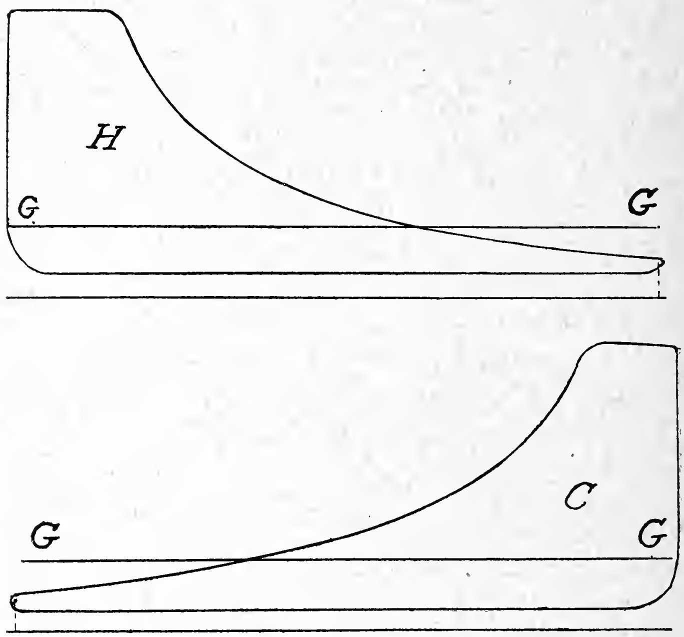

FIG. 21—SCALE OF SPRING 40.

only when conditions are favorable. To credit the condenser as the gain in efficiency equal to the percentage of the total load carried by the condenser is a blunder pure and simple and like many of the tests that are being made on steam powers, they are worthless and misleading.

The real economy of an engine is based on the amount of heat expended in furnishing it with steam and this bears no fixed relation to the percentage of the total load carried by the condenser or the amount of feed water consumed. The thermal unit basis takes into consideration all the various items, and reduces the performance of each engine to a standard that is comparable, in that credit is given for the energy expended in doing the work. It is the only true method of comparison and is especially worthy of adoption where a test is to be made in which the condenser and different types of engines are to be put into competition.

The partial vacuum in the cylinder is measured on the scale of the spring from the back pressure line to the atmospheric line G G, which is ten pounds, while the partial vacuum in the condenser is greater, as there is a loss of vacuum between the cylinder and condenser about the same as there is a loss of pressure between the boiler and engine. The exhaust valves should open quickly near the end of the stroke so as to bring the expanded pressure down where it belongs before the piston starts to return, and should remain open until it is necessary for it to close to form compression so as to obtain a vacuum in the cylinder as near full stroke as it is possible. The condenser simply removes atmospheric pressure on the exhausting side of the piston. Thus, if the mean effective vacuum is ten pounds the atmospheric pressure on the admission side of the piston is ten pounds greater, plus the mean effective steam pressure, which, we will suppose, is thirty pounds; this gives a total mean effective pressure (M.E.P.) of forty pounds.

Suppose it be desired to determine the cushion of the head end; first locate the lines of perfect vacuum on the scale of the spring. Then measure the compression on the head end diagram from the line of perfect vacuum which, in this case, is fifteen pounds absolute pressure, and is about as much as can be expected unless the clearance be extremely small, from the fact that the atmospheric pressure in the cylinder was only five pounds at the time the exhaust valve closed. Had this pressure been fifteen pounds, or three times as great, which it would, at least, have been in a non-condensing engine, the compression would also have been three times as great, or forty-five pounds, and the exhaust valve been closed at the same point. This is the reason why engines run noisy when condensing, and a readjustment of the valves is necessary. The pressure on the other side of the piston is measured on the crank end diagram from the line of perfect vacuum to a point as shown by the dotted line, which is eight pounds, and must be deducted from the compression of the head end, which gives a cushion of seven pounds; not enough for Corliss engines running at the ordinary speed, and will cause them to pound; therefore, some means must be provided to make up the deficiency. To accomplish this, some engineers have many peculiar wild-cat schemes which are oftimes amusing as well as costly. However, it may be accomplished by advancing the eccentric so as to admit live steam into the cylinder just before the piston completes the stroke, and will be shown on the diagrams by the admission line leaning slightly outward. But pains must be taken in doing this work so as not to admit the live steam too early and cause a counter-motion on the piston.

The cushion of the crank end diagram is determined in like manner by measuring the compression of the crank end diagram, from which deduct the pressure on the other side of the piston, at the time it comes to a stand still, which is measured on the head end diagram at a point as shown by the dotted line.

CHAPTER VI.

SETTING CORLISS VALVES WITH THE INDICATOR.

While the Corliss engine, as manufactured by the different makers, varies considerably in its details, a general understanding of the type as here illustrated and described, and the method of adjusting, will enable any one to readily adjust the valves in any of the different makes of this style of engine. In the manner of adjusting with the indicator it is necessary that certain preliminary adjustments be made; first, that the rocker arm A, as shown in Fig. 22, to which the eccentric rod B is connected, shall vibrate to equal distances each side of a plumb line from its center of vibration, when it is a horizontal engine and setting level. This may be determined with the engine running slow, or by revolving the eccentric on the shaft, and at the extreme positions of the rocker-arm, with a tram, make the marks C D usually on a smooth board placed near by. Then equally divide the distance between C and D as indicated by E representing a center-punch mark; next, place the engine so that the tram will span the distance between the rocker-arm and E, and adjust the length of the eccentric rod so that the rocker-arm is plumb and at right angles to the eccentric rod; after which, hook in the hook-rod F, as shown in Fig. 23, and adjust the length of hook-rod so that the mark on back hub of wrist-plate G coincides with center mark on wrist-plate stand; when the wrist-plate will be in its central position (if the plumb-bob be submerged in water it will more quickly bring it to a state of rest); then,

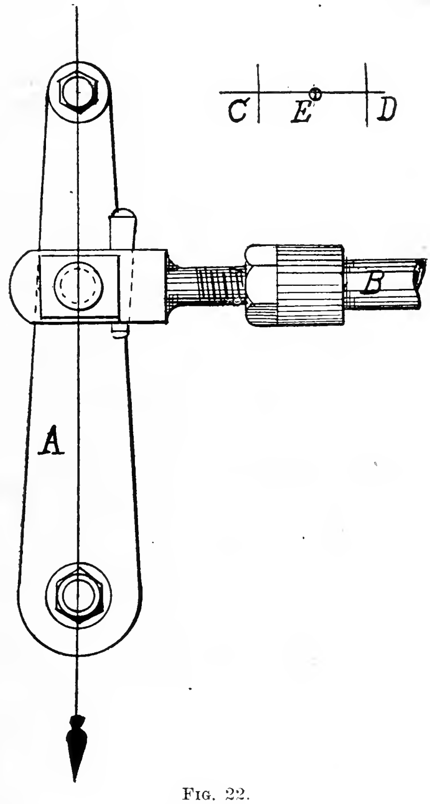

Fig. 22.

with steam valves hooked in, take off back bonnets of steam valves at the side of cylinder opposite the wrist-plate and pull valves out flush with the end of chambers and adjust the length of steam valve rods by means of the right and left thread at their opposite ends so that the steam valves will have lap according to the table, and the dividers will span the distance between the marks H and I, H representing the working edge of the port, and I the working edge of the valve. Lengthen the rods for greater lap and shorten for less. On engines with valves opening in the op-

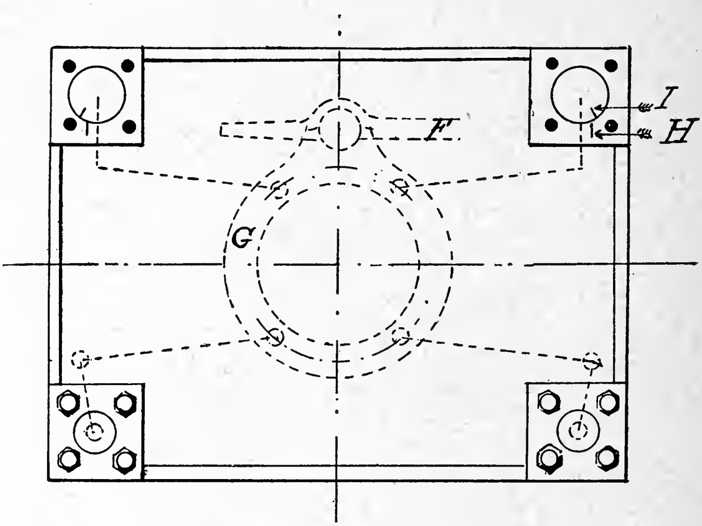

FIG. 23.

posite direction the difference in adjustment is to set the lap on the other side of the port. Any preliminary adjustment of the exhaust valves is unnecessary, because their proper positions are determined by use of the indicator, and this bears no fixed relation as to whether they are lapped, edge to edge, or open.

TABLE FOR SETTING LAP OF STEAM VALVES.

Diameter of Cylinder.	Lap of Steam Valves.	Diameter of Cylinder.	Lap of Steam Valves.
10	¼	24	⅜
12	¼	26	⅜
14	5-16	28	⅜
16	5-16	30	⅜
18	5-16	32	7-16
20	5-16	34	7-16
22	⅜	36	7-16

The amount of lap on steam valves with wrist plate on center varies among the different builders about 1-16″, more or less, from that given in the table; some giving about 1-16″ more, others about 1-16″ less. With less lap when the valves are at a point of opening, the eccentric is in a position to give the valves a more rapid opening, thus realizing a higher pressure in the cylinder; but the position of the eccentric must also be correspondingly later in order to have the same amount of lead, and this brings about a later action of the exhaust valves. With high terminal pressure, the exhaust valves in some of the engines, when equipped with a single eccentric, are not sufficiently rapid in their action to give a proper release and compression, and this loss of time more than balances what is gained in realizing a higher pressure in the cylinder from the small amount of lap. When it is found necessary to give more lap on steam valves than that given in the table in order to give proper action to the exhaust valves, the motion is not of a modern character.

With nearly all engines of present construction the amount of lap, as given in the table, will answer

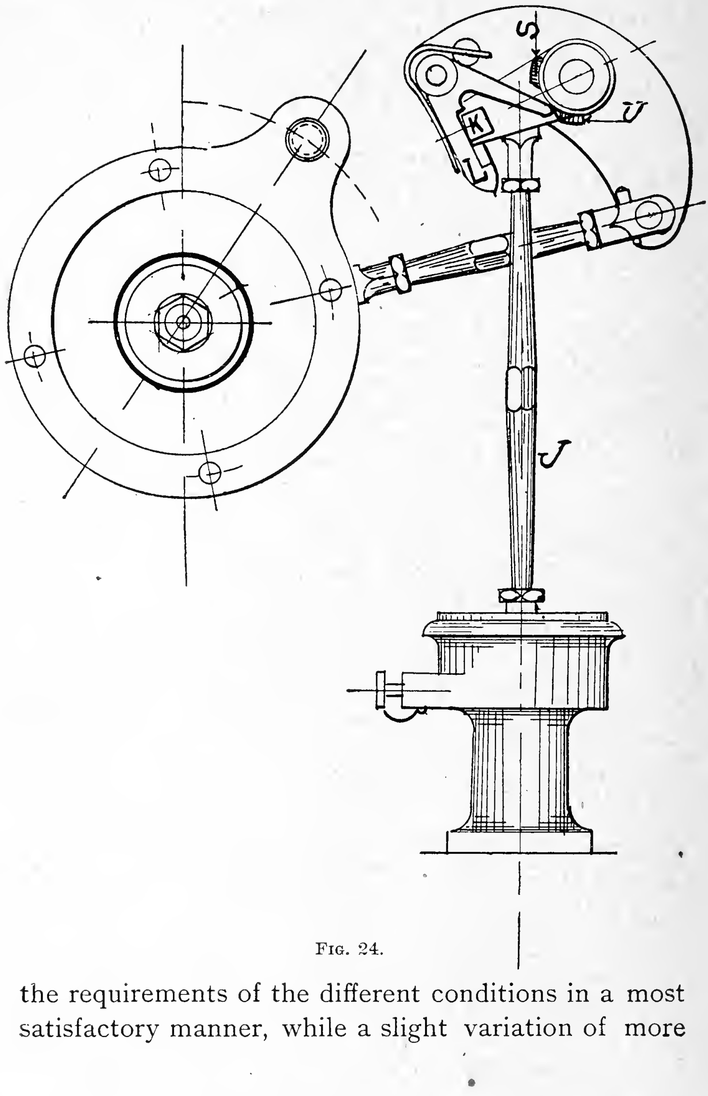

Fig. 24.

the requirements of the different conditions in a most satisfactory manner, while a slight variation of more or less lap may better suit the requirements in a few foreign conditions.

To adjust the length of the dash pot rod J, as

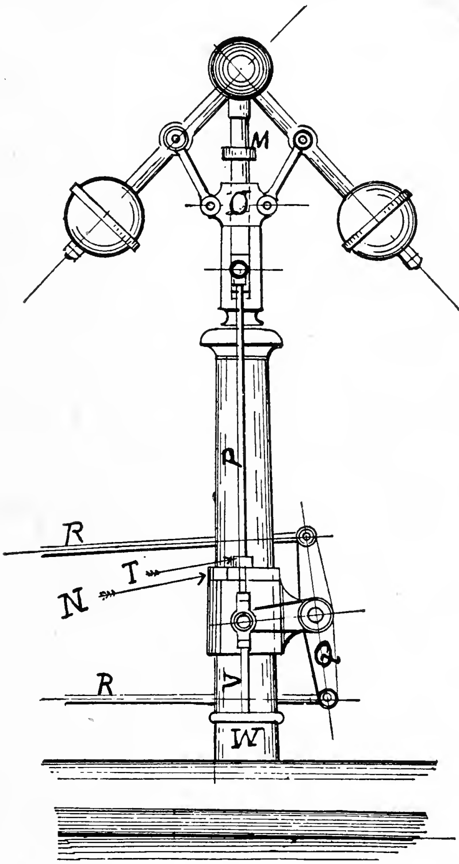

Fig. 25.

shown in Fig. 24, place the wrist-plate so that the right-hand mark on wrist-plate stand coincides with mark on the back hub of wrist-plate, and with the dash pot piston properly seated, adjust the length of the dash pot rod so that the block K will clear the hook L an equal distance on top and bottom; then place the wrist plate in the other extreme position and adjust the other rod in like manner. The collar M, as shown in Fig. 25, should be fastened at a height to allow the governor to raise enough so as to affect a sufficiently early cut-off, which will not materially increase the speed of the engine with only its friction load and highest pressure. With the governor resting on the projection of the collar N, raise the governor until the sleeve O is at a point one-half the distance toward the collar M, block it and adjust the side rod P until the double bell crank Q is perpendicular, when the governor will be the most sensitive; then lower the governor so that it again rests on the projection, and with the wrist-plate in the position as shown in Fig. 26, so that the left hand mark on the wrist-plate stand coincides with the mark on the back hub of wrist-plate, adjust the governor rod R so that the tripping cam S just touches the tail of the hook, and adjust the other rod in like manner.

The automatic safety stop consists of the collar N, about the vertical governor shaft, having on one side a projection through which is a pin to receive the groove underneath the side rod collar T. This collar is stationary on the side rod P, and rests on the projection of the collar N, when the governor is in its lowest working range; as soon as the governor raises a spring in connection with the collar N, swings the projection away from the path of the side collar T, and should the governor belt break the side collar T will clear the projection of the collar N and drop to its lowest extremity, when the safety cams

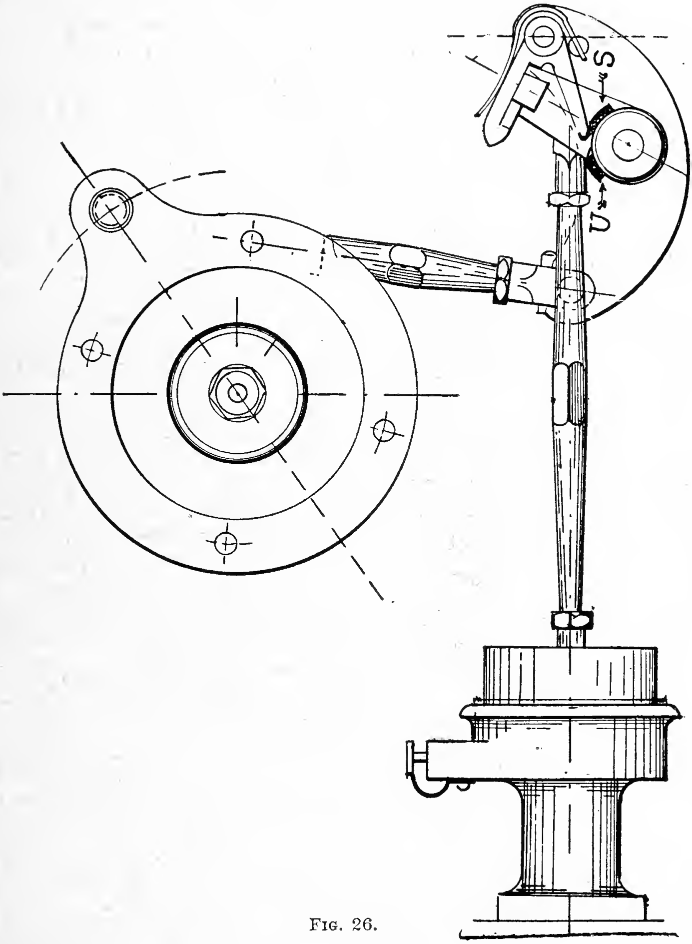

Fig. 26.

U will prevent the hooks from engaging. The steam valves will then remain closed and the engine will stop. When it is desired to stop the engine the collar N may be swung in place to receive the governor just before the engine stops; thus preventing the governor from dropping to its lowest extremity, and it will be necessary to raise it again before the engine can be started. No Corliss engine should be without an automatic safety stop, and the absence of one, no doubt, is the cause of many of the disastrous bursting of fly-wheels, causing loss of life and destruction of property. Such a device does not depend upon the memory of the engineer to pull a pin, or make some adjustment by hand, after the engine is up to speed, as is the case with many devices in order to make them effectual, To adjust the safety cams, place the wrist-plate in the position as shown in Fig. 24, and adjust the cams so that it just touches the tail of the hook, when the governor is resting on the collar N, and adjust the other cam in like manner. To test the adjustment, lower the governor to its lowest extremity and vibrate the wrist-plate in both extreme positions. Should the hooks engage, move these cams upward until the hooks disengage.

The plunger rod V of the oil pot W should be adjusted so the plunger will clear on top and bottom when the governor is in its highest and lowest positions. The substance used in the pot is usually engine oil diluted with kerosene, so as to permit the governor to more quickly respond to the variations of load. With most of the pots now in use the kerosene alone has not sufficient body to steady the governor, thus causing it to fluctuate, which make variations in the cut-off when there are no changes in load.

To reverse the engine only requires to set the eccentric so that it has the same degrees of advancement on the other side of the crank and not a half turn on the shaft, unless its previous position was 90 deg. in advance of the crank.

To increase the speed of the engine the governor pulley must be larger, and to decrease, smaller.

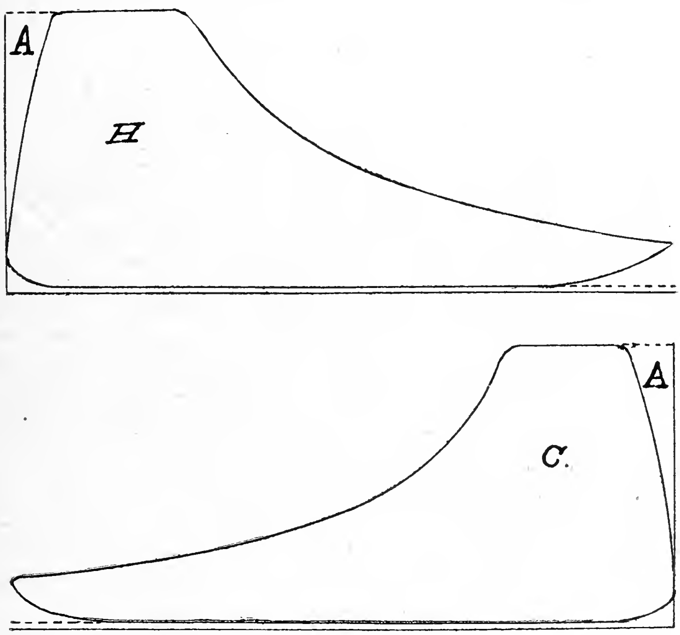

FIG. 27—SCALE OF SPRING 40.

The remaining adjustment of the valves are made by use of the indicator.

Should the results be a pair of diagrams similar to those represented in Fig. 27, H representing the head end, and C the crank end, the admission lines lean inward, which indicate insufficient lead, causing loss of power and economy. The distance along the dotted lines from the completion of the admission lines to the lines A, which is located by a right angle perpendicular to the atmospheric lines, is $\frac{1}{4}''$. The length of the diagrams is $3\frac{1}{2}''$, and the length of the stroke of the engine is $42''$.

Now, as $\frac{1}{4}=.25$, and $3\frac{1}{2}=3.5$:

$$\text{Thus } \frac{.25\times 42}{3.5}=3''$$

which is the distance the piston moved before the full pressure was admitted. The exhaust valves are correspondingly late, both in opening and closing, because the expanded pressure is not fully released until the piston has traveled on its return stroke the distance equivalent to the length of the dotted line along the end of the diagrams. The compression is only seven pounds, and the pressure on the other side of the piston at the end of the stroke is ten pounds. This is three pounds more pressure per square inch on the opposite side of the piston than the compression, and instead of having cushion, the tendency would be to send the piston still further, were it possible to do so.

Many engines are being run under just such abusing conditions, and ofttimes worse, and it is not to be wondered at that it requires liberal use of lubrication, frequently hot crank pin, etc., as well as a noisy engine. If engineers would make frequent use of the indicator the engine would require less attention, and looking at it from a commercial standpoint, at the rate of dollars and cents, for those who must pay for the fuel and repair bills, the indicator is an indispensable adjunct to every engine. The economical running of the engine and the practical application of the indicator should be thoroughly understood by every man who aspires to the calling of an engineer. It is plain to be seen that the engine is late all around, but before advancing the eccentric, attention should be given to the exhaust valves, that they are sufficiently closed before the steam valves open, so that the entering steam does not pass into the exhaust, in which case the admission would appear too late, if it really were early enough. This is indicated by absence of compression. The compression shown in these diagrams is sufficient to accomplish this result. Therefore, the first adjustment to be made is to advance the eccentric until the admission lines are perpendicular. The exhaust valves will open earlier, causing a more perfect release and an earlier closing, causing a higher compression. Should the compression be too high, and the release still a trifle late, shorten the exhaust valve rods, which will result in an earlier opening and a later closing; and should the release be too early and compression too low, lengthen the rods accordingly. Again, supposing the results should be a pair of diagrams similar to those represented in Fig. 28. The admission lines lean outward, which indicate too early lead, because steam was admitted into cylinder before the piston had completed the stroke at an equivalent distance to that represented by the dotted lines. That the admission is too early is plain to be seen, because the upper line indicates the piston going ahead and the lower line coming back, and if steam was not admitted into the cylinder until the piston had completed the stroke, as it should, neither would the admission line start to erect until at the end of the diagram. The compression is high, and an additional amount of live

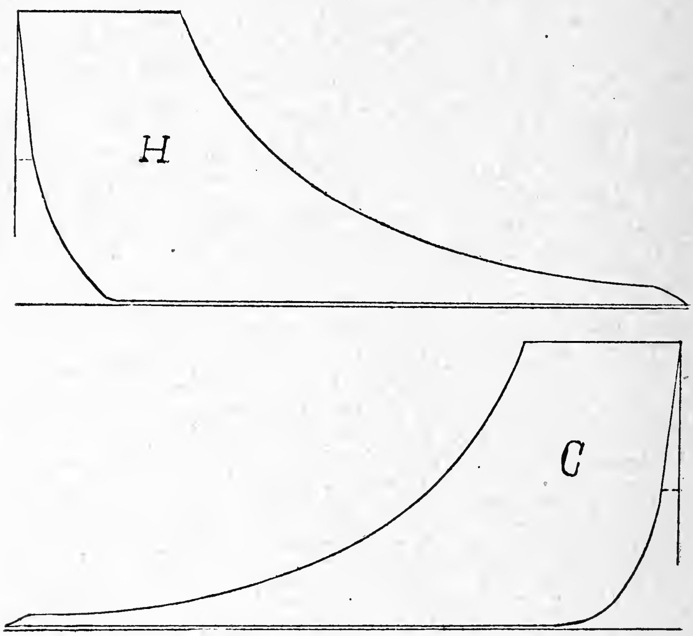

Fig. 28—Scale of Spring 40.

steam will more than stop the piston at the end of the stroke, thereby forming a counter-motion, and requires an additional amount of steam, which is a loss of power and economy. Should the exhaust valves close so early that the compression equals the highest admission pressure there would be no admission lines and the exhaust valve rods should be shortened so as to lower the compression about equal to that of Fig. 28, then the eccentric should be set back until the admission lines be perpendicular, which at the same time will lower the compression. Should the compression be too low after the eccentric has been set back so that the admission lines are perpendicular,

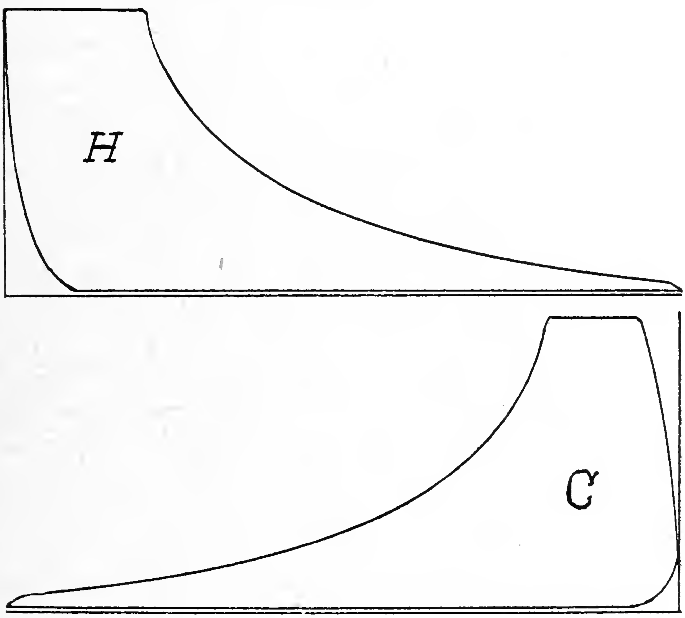

FIG. 29—SCALE OF SPRING 40.

the exhaust valve rods should be lengthened until a higher compression be obtained.

Obtaining the same form of admission lines from both ends of the cylinder cannot always be accomplished with the eccentric. Frequently the admission lines obtained will be similar to those represented in Fig. 29. The head end admission line leaning as much outward as the crank end one leans inward—principally due to the angularity of the eccentric rod. Should the exhaust opening and closing be late, the eccentric should be advanced until the admission line of the crank end is perfect. This will usually bring the exhaust valves around so they will perform their proper duties. But the admission of the head end will be still earlier, which can be remedied by lengthening the head end steam valve rod, thereby increasing the lap until the admission at this end also is perfect. This necessitates a readjustment of the governor rod and dash pot rod on the head end. Should the opening and closing of the exhaust valves be too early, the eccentric should be set back until the admission line of the head end is perfect, when the exhaust valves will be sufficiently later, but the admission of the crank end will be still later. This can be remedied by shortening the crank end steam valve rod which decreases the lap until the admission of this end also is perfect, and a readjustment of the governor rod and dash pot rod on the crank end must be made. The indication of the opening and closing of the exhaust valves from the diagrams, as shown in Fig. 29, are perfect, therefore the eccentric should remain in this position, and the steam valve rods on both ends of the cylinder should be altered, lengthening the head end and shortening the crank end.

To equalize the cut-off is the last adjustment to be made. This can be determined on the steam lines by the distance from the commencement of the diagrams to the points of cut-off. Should the cut-off on the crank end be the earlier as shown in Fig. 30, it is only necessary to shorten the crank end governor rod, which will set this tripping cam further away until the cut-off is equal. Where the load is constantly changing and when only one indicator is used, it is frequently connected with side pipes in a three-way

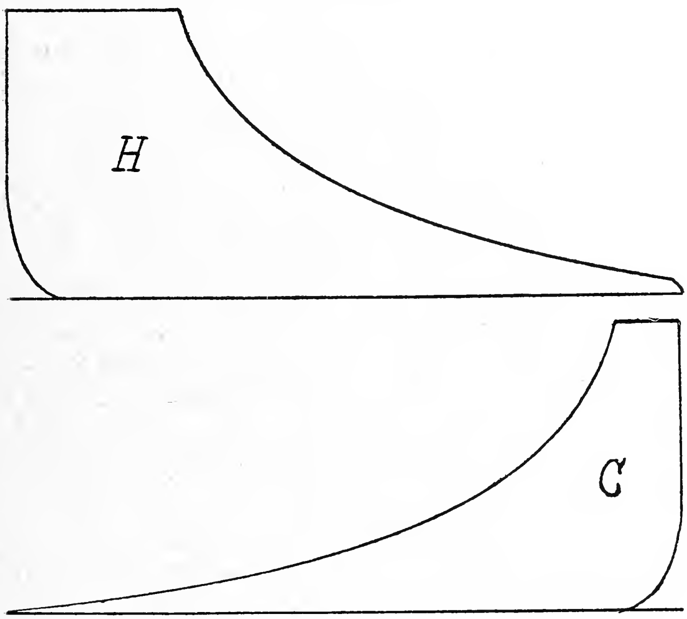

Fig. 30—Scale of Spring 40.

cock or T with an angle valve at each end and a diagram is taken from each end of the cylinder on the same card as shown in Fig. 31, and the pencil should be applied about four revolutions and about three different times each side of the piston.

In this manner the average points of cut-off are more closely determined than if all the revolutions were taken at one time on each side of the piston. Should the head end be the earlier, lengthen the head end governor rod which will have the same effect on this end. With valves opening in the opposite direction the governor rod of the head end must be shortened to effect a later cut-off and the crank end

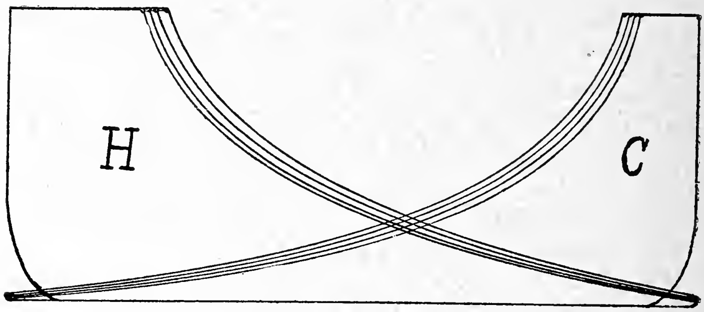

FIG. 31—SCALE OF SPRING 40.

lengthened. While this may be quickly equalized by partly closing the throttle valve until the valves trip only a part of the time and it can readily be detected if they trip and pick up equally. This will disturb the adjustment of the safety cams of the automatic safety stop and a readjustment becomes necessary.

CHAPTER VII.

DIAGRAM FROM STEAM PIPE.

Fig. 32 shows a cylinder and steam pipe diagram with boiler pressure line added. A A is the boiler pressure line which is located by measurement on the scale of the spring at such a distance above the atmospheric line as to represent the boiler pressure by gauge at the time the cylinder diagrams were taken

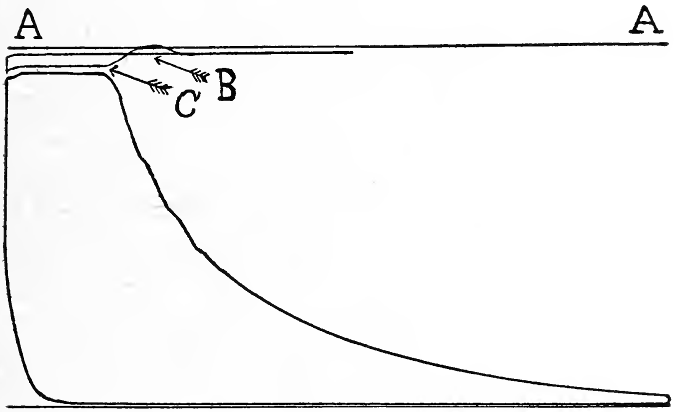

FIG. 32—SCALE OF SPRING 40.

The diagram between the boiler pressure line and the cylinder diagram is that portion of the steam pipe diagram belonging to that end of the cylinder and if taken separately, must be located by measurement at such a distance above the atmospheric line on the cylinder diagram as its distance is above the atmospheric line of the steam pipe diagram. B is taken when no steam is admitted into the cylinder and represents the highest pressure realized at the point the steam pipe diagrams were taken. The distance between B and the boiler pressure line represents the loss of pressure by radiation between this point and the boiler. C is taken during the time that steam is admitted into the cylinder and represents by its distance between it and B, the loss of pressure at that point during admission. If the indicator is attached above the throttle valve the distance between C and the boiler pressure line, represents the loss of pressure between the boiler and the point above the throttle valve during admission. If the indicator be attached below the throttle it represents the loss of pressure between the boiler and that point, and so on. The distance between C and the steam line of the cylinder diagram represents the loss of pressure between the cylinder and the point above the throttle or any other point of attachment. When steam is cut off the pressure in the pipe is restored and C rises correspondingly, and should merge into the line B, but it generally causes the pressure at this point to fluctuate above B and frequently a trifle above the boiler pressure line as shown. With long steam pipes and excessive loss of pressure between the boiler and engine the fluctuation is constantly taking place and is shown by a rising and falling of pressure at other points of the diagram about the same as when two or more engines are being supplied from the same main. For convenience the steam pipe and cylinder diagram should be of equal length, otherwise, it requires considerable figuring to learn their proportions.

ECCENTRIC TOO LATE.

Fig. 33 shows a pair of diagrams taken from a 14″ × 36″ Corliss engine, where the superintendent had never run an engine and knew practically nothing concerning it, but thought himself more proficient in the science of adjusting valves than his engineer, and he proceeded to adjust the valves on his engine by the rule of the thumb. The results are not surprising as the man attempted to do something without know-

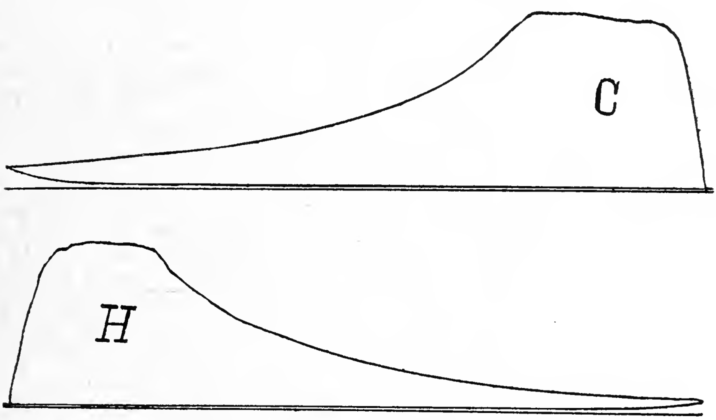

FIG. 33—SCALE OF SPRING 60.

ing how to do it. After the engine had run a few months he desired to have a pair of indicator diagrams taken from it by an expert only as a matter of interest because any readjustment of the valves he knew was unnecessary. After the valves were properly adjusted with the indicator, the fuel bill was nearly cut in two, and the superintendent was forced to acknowledge that the thumb rule for setting engine valves was neither practical nor profitable.

The admission is late and the exhaust valves are still open after the piston has started on its return stroke causing the entering steam to pass into the exhaust and the admission lines follow the back pressure line a short distance. When the exhaust valves close, thus preventing the entering steam from passing into the exhaust, the admission lines rise, but owing to the fact that piston has already moved a considerable distance the highest pressure is not realized on the crank end until at the point of cut-off. The exhaust valves are late in opening and the pressure near the termination of the expansion curve of the crank end is six pounds, which on account of having no compression, would tend to send the piston still further at the head end were it possible to do so.

The pressure near the termination of the expansion curve of the head end is four pounds, which would tend to send the piston still further at the crank end. The engine sounded like the drop of a pile driver at each end of the stroke. The connections were frequently hot and most of the engineer's time was occupied in oiling and keying up.

The cut-off is unequal and the automatic safety stop was ineffectual and the whole job needs no further description. The vibration of the rocker arm was first adjusted; then lap of the steam valves was set and other necessary adjustments made, such as the governor, dash pot rods, etc., after which the eccentric was advanced about one inch on the shaft, which brought a perfect admission and release, and the compression still being insufficient the exhaust valve rods were lengthened.

DIAGRAMS FROM THE SAME ENGINE, BEFORE AND AFTER ADJUSTING WITH THE INDICATOR.

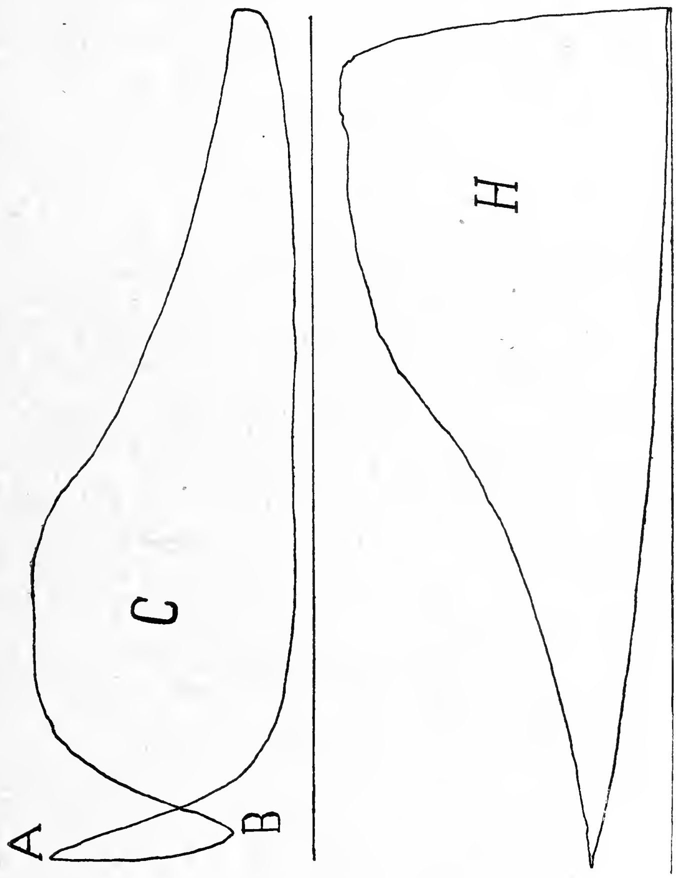

FIG. 34—SCALE OF SPRING 40.

Fig. 34 is taken from a 20″ × 42″ Corliss engine; revolutions, 60; boiler pressure, 85. This engine had been in use many years and the diagrams in question were the first ones taken from it, and their features show almost everything but economy. The release pressure is too high and a vast volume of steam at high pressure which could yet do useful work is exhausted into the atmosphere. The exhaust valves are late in opening and the minimum back pressure of the crank end is $3\frac{1}{2}$ pounds. The crank end exhaust valve closes too early and forms $54\frac{1}{2}$ pounds compression, as shown at A. When the piston begins its return stroke the steam valve remains closed and the pressure falls to B, the valve then opens, but the highest pressure is not realized until the piston has traveled about nine inches, causing a loss of 27 pounds pressure between this end of the cylinder and the boiler.

The admission on the head is also late, which is caused either by open exhaust valve (because there is no compression) or by absence of lead, and the loss of pressure between this end and the boiler is 16 pounds. The points of cut-off are poorly defined, caused by the steam valves being slow in their action while closing. The release pressure of the head end is $16\frac{1}{2}$ pounds and the minimum back pressure is not obtained until the end of the stroke.

After the valves were properly adjusted other conditions were examined and also found to be anything but economic, as the loss of pressure between boilers and engine was 14 pounds, yet the distance was short. Steam pipe diagrams were taken and the loss of pressure through the throttle valve, steam chest and ports was found to be slight. The excessive loss being in the steam pipe, which was imperfect in every detail; no provisions having been made for unequal expansion or contraction, and its small size and unnecessary bends were responsible for the loss of power and economy. This great difference of pressure tends to bring water from the boilers, and there being no separator in the steam pipe consequently the condensation and water entrained in the steam passed into the cylinder, thus affecting its efficiency as well as the lubrication. Kerosene was also used in the boilers in great quantities as a scale preventative, and the kerosene must pass through the engine cylinder, which also destroys a perfect lubrication, and it is no wonder the valves and piston were in a very leaky condition. These were put in order and the steam pipe replaced with one of ample size and proper construction, being provided with a separator which returned the water automatically to the boilers by means of a steam loop, which is the best cf the many devices known. Some steam users consider that if the engines and boilers are of modern type the rest of the plant is good enough and can take care of itself. The importance of having a properly piped steam plant has been too long neglected, and I feel safe in stating that a very large percentage of the steam plants in use to-day are in such a defective condition as to render them both costly and dangerous, and it is to be hoped that at no distant date steps will be taken toward bringing this important factor of the steam plant to a degree of perfection.

The excessive back pressure was not caused through any fault of the exhaust pipe, but was due to insufficient area of the exhaust ports, and this defect was also remedied. After the alterations were made, over which I had personal supervision, and

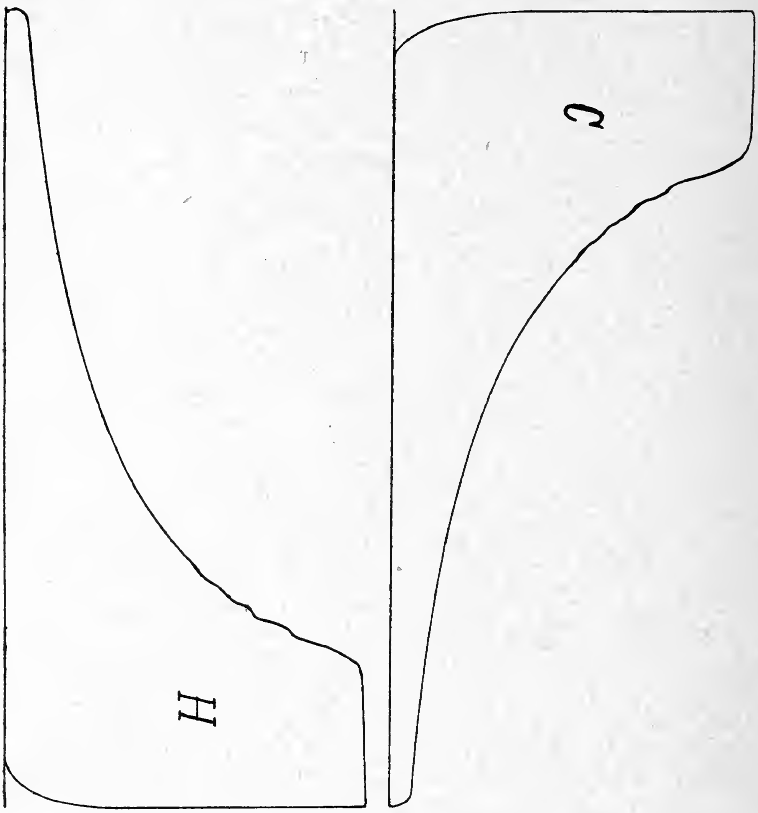

FIG. 35—SCALE OF SPRING 40.

the valves properly adjusted, all the machines of the factory were put to work at full capacity for the purpose of determining the capacity of the engine. While before it had been too small, it was now more than ample, and Fig. 35 was obtained. The loss of pressure between the boilers and engine is only 2 pounds and a fraction; the back pressure line merges into the atmospheric line and the working of the entire valve gear produced a pair of indicator diagrams that it is hard to believe come from the same engine. There are many men traveling around the country who style themselves indicator experts, and profess to make complete and accurate tests of steam plants. I have noted several instances where

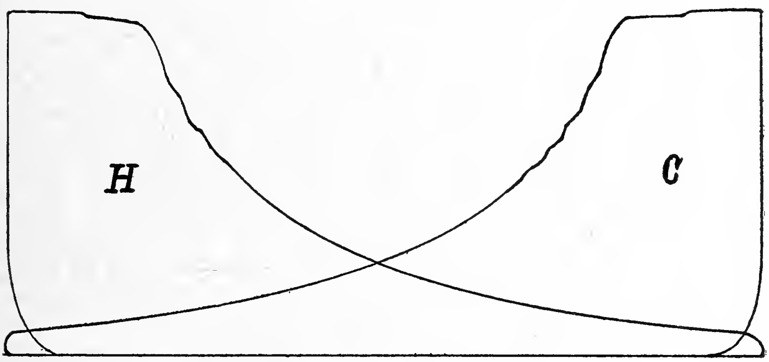

Fig. 36—Scale of Spring 40.

these so-called experts have made what they considered a complete test, and invariably found it to be the case that they had simply adjusted the valves of the engine so as to secure a diagram that would pass inspection, and had not taken into consideration any of the other important factors. Such tests are expensive as well as incomplete.

GOOD ADJUSTMENTS BY A NOVICE WITH THE INDICATOR.

Fig. 36 shows a pair of diagrams from a 16″ × 42″ Corliss engine. Diameter of piston rod, 2 15-16″;

lap on sceam valves, 5-16″; revolutions, 48; boiler pressure, 80. The valves of this engine were adjusted and the diagrams taken by a young man who had only a very limited experience with the indicator, but he rapidly perfected himself in its use by receiving personal instructions as to its practical application and the study of practical works on the subject. In the present epoch of improvement and advancement, length of service is not now such an important factor as capability. The indicator may at first seem a broad scope to the beginner, which is true, from the fact that it embraces the most valuable, important and scientific factors in the science of steam engineering. If the author presents the instruction to the student in a comprehensive, reliable and attractive manner, what to do and how to do it and why it should be done, any intelligent engineer or practitioner having a reasonable mechanical knowledge can, in a short time, perfect himself so as to be thoroughly capable of correctly applying the indicator and skillfully make all the adjustments upon the engine so as to obtain an economical departure in the working of steam, and secure a smooth running engine, as well as to make the necessary calculations from the diagrams in a scientific and systematic manner. The indicator unfolds a picture of the performance that takes place inside of the steam engine cylinder which would otherwise be enveloped in mystery, and guides the engineer accurately in setting the valves of all the different styles of engines by a method that is easily and readily comprehended by any intelligent engineer, and there is no other device that so thoroughly teaches and qualifies an engineer as a first-class indicator. The degree of perfection which this young man has attained in the handling of the steam and exhaust valves only proves what I have before stated, that any man with ordinary intelligence can, with the indicator, soon learn to keep his engine in the most economical working condition. The steam is admitted and released at the right time; the compression is moderate and the cut-off is practically equal; the steam lines falls slightly as the piston approaches the point of cut-off, and is no fault of any of the adjustments made upon the engine, but is due to insufficient area in the steam passages. The loss of pressure will be the greatest at a point in the cylinder when the crank is at right angles to the connecting rod, at which time the movement of the piston is the fastest. The small waves at the commencement of the expansion curve are caused by fluctuations of the pressure in the cylinder, due to the rapid falling of pressure as compared with the distance the piston moves away. If steam be cut off at one-fourth stroke (clearance neglected) the piston must move only from one-fourth stroke to one-half stroke when the volume is doubled, while from one-half stroke it must move to the end of the stroke, or double the distance, before the volume is again doubled from what it was at one-half stroke. The most rapid change of volume is, therefore, immediately after the earliest point of cut-off, when the piston moves the shortest distance to change the volume. The sudden change of pressure causes the pressure to fluctuate, and generally follows with oscillations of the indicator piston and forms a wavy expansion curve at this point. On high-speed engines, where the piston is moving rapidly away, the wavy expansion curve may continue to the end of the stroke. The fluctuation of pressure in the cylinder immediately after the point of cut-off, also is due to condensation, because the temperature of the steam at this pressure exceeds the temperature of the cylinder. When the steam has expanded so that its temperature does not exceed the temperature of the cylinder, the fluctuations from this cause have practically ceased. When the waves are symmetrically rounded, as they will be when taken with a sensitive instrument, they do not destroy the truth of the diagram.

DIAGRAMS SHOWING DIFFERENT EFFECTS IN CUSHION WITH AND WITHOUT THE CONDENSER.

Fig. 37 shows a pair of diagrams from a 16″×36″ Corliss non-condensing engine. Revolutions, 80. Fig. 38 is taken from the same engine, with condenser attached, and without readjusting the valves. The difficulty in this case was, as stated by the operator, that, when running non-condensing the engine run practically noiseless, but when the condenser was attached the engine pounded badly, for which they were unable to account and could not remedy. Frequently inquiries are being made as to why engines run noisy while condensing, and how it may be remedied. But few condensing engines at the present time are running smooth and with proper action of the exhaust valves, for the reason that this subject has not been thoroughly understood. Referring to Fig. 37, the cushion on both ends is 23 pounds, and in this case, when both diagrams are taken on one card, instead of measuring the compression and deducting the

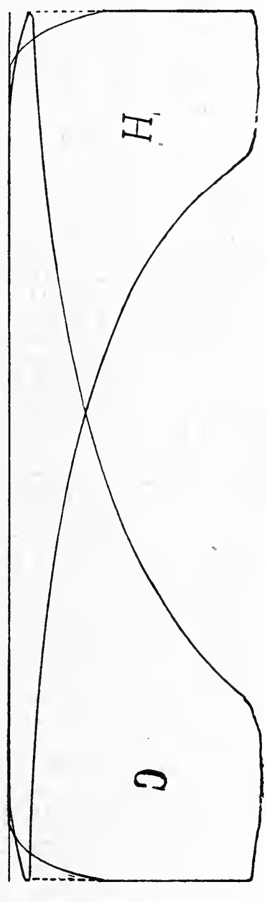

FIG. 37—SCALE OF SPRING 60.

pressure above the atmosphere, or adding the vacuum below on the other side of the piston at the instant it comes to a standstill, it is only necessary to measure from the top of the termination of the expansion curve to the point intersecting the admission lines, as shown by the length of the dotted lines.

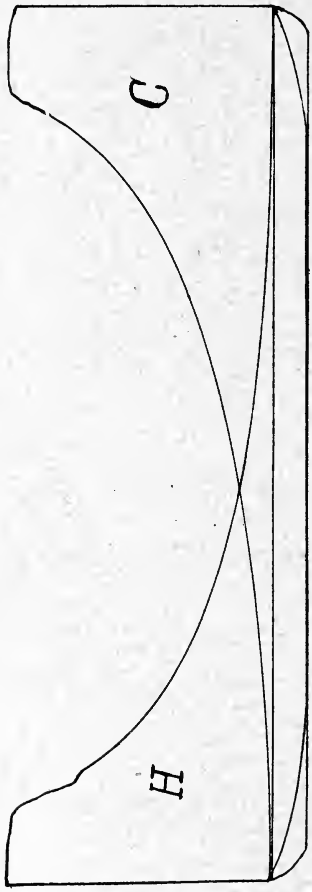

Fig. 38—Scale of Spring 60.

The absolute back pressure is 15 pounds, and the compression is 45 pounds. When the condenser is attached, as shown in Fig. 38, having 10 pounds vacuum, or a remainder of five pounds atmospheric back pressure, or one-third as great as when non-condensing. The compression can only be one-third of 45, which is 15 pounds, and in this case the terminal pressure on the opposite side of the piston is 15 pounds, therefore the pressure is equal on both sides of the piston, and there is no cushion. This is the reason that engines run noisy when condensing, and a readjustment of the valves is necessary. The necessary cushion can be obtained by advancing the eccentric until the pounding has ceased, thus admitting live steam into the cylinder a trifle before the piston has completed each stroke, and the admission lines will lean outward. This will also cause the exhaust valves to open earlier, and should they then close too early and choke off the condenser the exhaust valve rods should be shortened, which will cause an earlier opening. If this does not give proper action to the exhaust valves, and unless they have the maximum amount of lap, we would suggest an increase of the lap, and advance the eccentric accordingly, thus giving a still earlier action of the exhaust valves.

DIAGRAMS FROM OVERLOADED CORLISS CONDENSING ENGINES.

Fig. 39 shows a diagram from a $20'' \times 42''$ Corliss condensing engine. Revolutions, 70. The manipulation of the indicator in this case only proves that unless it is properly made use of, and its reading correctly interpreted, and the conclusions based upon the diagrams are judiciously made, the indicator is of no more value to the engineer than a pair of spectacles are to a blind man. The imperfect action of the exhaust valve practically robs the condenser of any value it may have. In order that the earliest release might be obtained from any adjustment made on the exhaust valve rod the compression has been lowered to such an extent that the exhaust valve is only sufficiently closed at the end of the stroke so as to prevent the entering steam from passing into the exhaust and the piston has moved a considerable distance before the condenser has

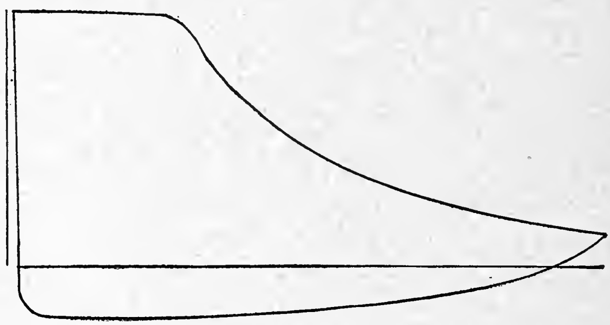

Fig. 39—Scale of Spring 40.

succeeded in lowering the back pressure equal to the atmospheric, and a reasonable amount of vacuum is not obtained until the piston has reached about one-half stroke. The release pressure on the diagram from the other end of the cylinder was considerably greater than the compression, but a part of the deficiency in cushion is obtained from the entering steam which is admitted into the cylinder a trifle before the piston has completed the stroke, as shown by the admission line leaning slightly outward. Whether the steam valves have as much lap as is possible without wire drawing the entering steam to an extent that will more than balance that gained by an earlier action of the exhaust valves, is a matter of doubt, and should this be the case, it proves that the motion is not of a modern character. It is true that the majority of engine builders are not as close readers of the indicator as they are generally supposed to be, and this is the reason that many of our modern engines are deficient in construction rather than principle. Every concern that pretends to manufacture an engine proves its experiments and substantiates its claims by the aid of the indicator, and unless the operator is proficient in its use it must necessarily follow that the engine is as incorrect as the conclusions based upon the diagram. This diagram shows that the engine is overloaded, and unless the steam valves have as much lap as is possible the lap should be increased, which would necessitate an advance of the eccentric accordingly, and this would bring about an earlier action of the exhaust valves so as to give the condenser proper action throughout the stroke, and relieve the load carried by the steam, which would bring about a more suitable release pressure and a gain in economy.

Fig. 40 shows a pair of diagrams from an $18'' \times 42''$ Corliss condensing engine. Revolutions 60. The admission lines on the diagrams lean slightly inward and indicates a trifle late admission, thus showing that the eccentric should be advanced a trifle, providing the steam valves had the proper amount of lap and that the exhaust valves performed their proper duties. But the exhaust valves are late as they do not commence to open until at the end of the stroke, and the terminal pressure at the head end is 7 pounds greater than the compression of the crank

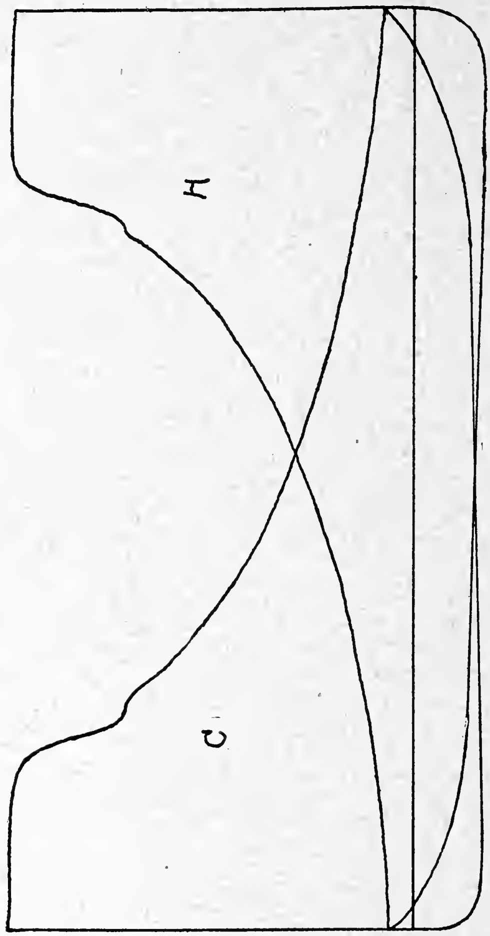

Fig. 40—Scale of Spring 30.

end, which would tend to send the piston still further were it possible to do so. Also on the right hand end of the diagrams the terminal pressure is 5 pounds greater, and since the piston speed in feet per minute is 420, the revolutions 60, and without any cushion the engine cannot help but pound. And this late opening of the exhaust valves chokes off the condenser at the commencement of its stroke, so that the condenser does not get the proper action until nearly one-half stroke.

The load is too heavy for such a pressure as the point of cut-off should take place at a point in the stroke so as to bring the expansion curve below the atmospheric line near the end of the stroke.

The steam lines are fairly well maintained which indicate an ample area in the steam passages, and a rapid opening of the steam valves which is evidently accomplished by having small amount of lap on the steam valves, in which case the eccentric has less angular advance in order to obtain the same amount of lead, and consequently gives a more rapid opening of the steam valves. But this gain cannot balance the efficiency, of which the condenser is robbed, therefore the lap on the steam valves should be increased which would necessitate advancing the eccentric, thus giving a more rapid action of the exhaust valves. But, however, with such a high terminal pressure the necessary cushion cannot be properly obtained, from the fact it would choke off the condenser too early at this end of the stroke, therefore, since the engine is overloaded, and after giving the proper amount of steam lap the eccentric should be advanced sufficiently to make up the deficiency in cushion by admitting live steam into the cylinder before the piston completes the stroke, when the admission lines will lean slightly outward.

Fig. 41 shows a diagram from a 14″ × 42″ steam jacketed Corliss pumping engine. Revolutions 24. The admission line leans inward, which indicates a late admission, and the piston moved a short distance before the full pressure was admitted. The steam line is maintained in a straight line with the atmospheric line to the absolute point of cut-off, which, however, can be readily obtained in most any Corliss engine running at such a slow speed. The point of

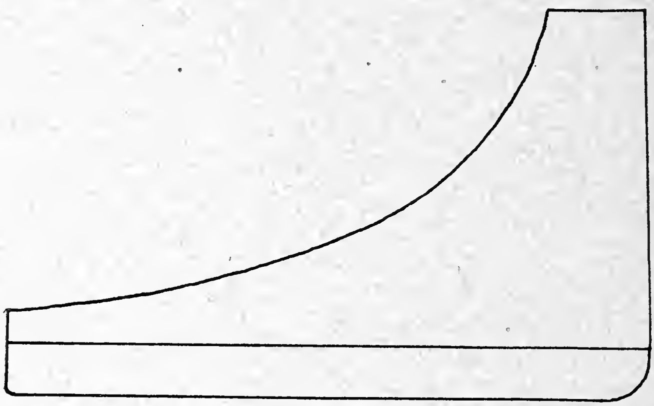

FIG. 41—SCALE OF SPRING 40.

cut-off is so sharply defined that there is no indication of a reduction in the pressure during the time that the steam valve is closing, which is also readily obtained on a slow speed engine that is equipped with an independent cut-off. The expansion curve presents a beautiful appearance but not knowing the percentage of clearance, its truthfulness can not be known. However, an approximation of the percentage of clearance not materially differing in the location of the clearance line will make but slight variation in the theoretical curve. The exhaust valve does not open until at the end of the stroke and a perfect release is fully obtained while the engine is passing its center, thus the full amount of vacuum is realized before the piston commences its stroke, as shown by the perpendicular release line. But the slow speed gives ample time for a perfect release. The vacuum line is also beautifully maintained and the exhaust valve closes at a point in the stroke to give the con-

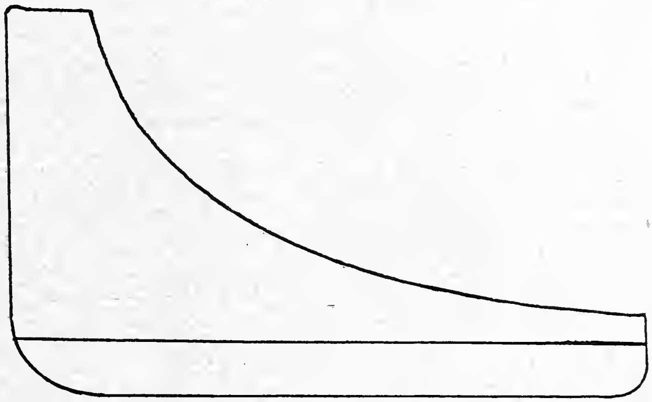

FIG. 42—SCALE OF SPRING 40.

denser proper action. The terminal pressure on the diagram from the other end of the cylinder was 4 pounds greater than the compression of this diagram, yet the engine run fairly smooth, from the fact that the piston speed in feet per minute is 168, and the revolution 24. The engine is overloaded, but had the admission line been perfect the diagram would have been a subject of beauty.

Fig. 42 shows a diagram from the head end of the cylinder. The admission line leans outward and the full pressure is admitted before the piston comes to a standstill, which was found necessary on the head end in order to secure smooth running, from the fact that the piston at this end moves faster, the crank end having no cushion. As the piston begins its stroke it creates a fluctuation in the pressure and forms the small hump above the steam line proper. Steam lines frequently have many peculiar defects and some of them are regarded as a mystery, but are in reality simple facts, caused by the disturbance of the pressure in the steam passages, and with the Indicator attached at these various places, they can be located as precisely as on the point of a needle. Frequently steam lines have a drop in pressure and again immediately rises thus forming a notch, and on a Corliss engine it is generally caused by a readjustment of the exhaust valves which after a long service have worn only that portion of the chamber in which it moves, and as the valve passes over the shoulder it gives a sudden leak and forms the notch in the steam line. When two or more engines are supplied from the same main and unless its size is ample, it may cause a wavy steam line about the same as caused by a too small steam pipe on a single cylinder engine, or an inferior Indicator may falsify a diagram and cause the engineer a great deal of unnecessary labor and thought.

DIAGRAMS SHOWING LEAKY EXHAUST VALVES.

Fig. 43 indicates a leak in the exhaust valve, as shown by a falling in pressure in the upper portion of the compression curve. Had the exhaust chamber been free from cuts and wear, it would only have been necessary to replace the valve, but when they are both cut the chamber must be re-bored and a valve made to fit. The admission line leans inward which indicates a too late admission, fiing of indicator causing a peak. The sudden drop of pressure in the latter portion of the expansion curve is also caused from a leak in the exhaust valve, because the lappage on the exhaust valve at this point in the stroke is less than in the beginning of the expansion curve. The

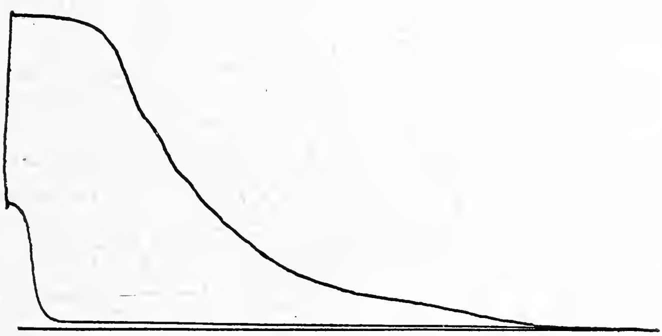

FIG. 43—SCALE OF SPRING 40.

steam valves in this case were practically free from leaks, otherwise the leakage of them into the cylinder might have balanced the leakage out of the cylinder in which case the compression curve would have indicated no leak. This falling in the compression curve may also be caused by leakage in the piston, but if so it can generally be detected by a rapid falling of the pressure in the expansion curve, immediately after the point of cut-off. The only positive method of testing the leakage of the valves and the piston is to test them without the indicator, from the fact that the leakage into and out of the cylinder may balance each other in which case the indicator would indicate no leak.

Fig. 44 shows a diagram from a $32'' \times 60''$ Corliss

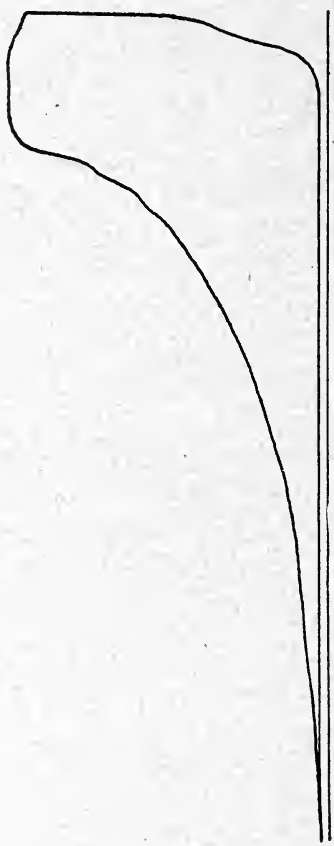

Fig. 44—Scale of Spring 40.

engine. Revolutions, 50. The admission line leans slightly outward and indicates a too early admission, but still there is a reduction in the initial pressure as the highest pressure is not realized until the piston has moved a considerable distance as shown by the loss of pressure at the commencement in the stroke and was caused from keying up the connections on the eccentric rod which disturbed its length and gave an unequal vibration to the rocker-arm in which case the steam valve on this end of the cylinder was slow in its action while opening. The upper portion of the compression curve leans outward and indicate a leak in the exhaust valve from the fact that the compression curve on the diagram from the other end of the cylinder was perfectly formed, thus showing that there was practically no leak in the piston. Had the compression curve on the diagram from the other end of the cylinder also showed a leakage out of the cylinder it would indicate a leak in both exhaust valves, or the piston or both.

A DISTORTED DIAGRAM CAUSED BY IMPERFECT REDUCING MOTION.

Fig. 45 shows a diagram taken from a Corliss engine, and is a fair representation of diagrams frequently encountered and considered as a correct record of the action of the steam in the cylinder. The absence of data and atmospheric line renders the diagram worthless, except as an indication of the valve setting. But in this case the diagram is worthless even for such a purpose, from the fact that the reducing motion was incorrect in every detail. At the point A the movement given to the paper barrel was too rapid, and the expansion curve is completed nearly in a straight line, while at other points the motion is correspondingly slower. To base any conclusions upon such a diagram is impossible, unless the error of the reducing motion be known. This can readily be determined, however, from the explanation given in Fig. 13.

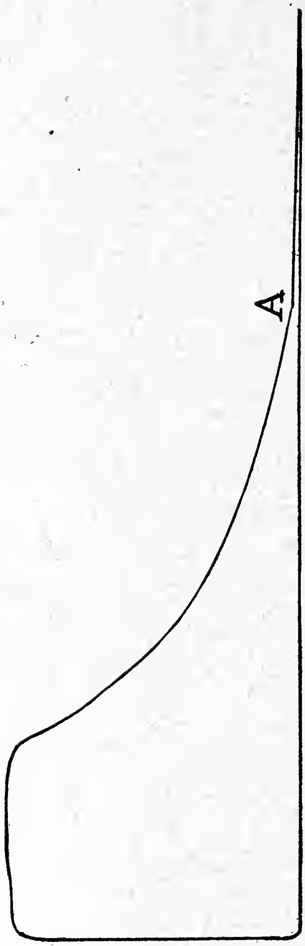

Fig. 45—Scale of Spring 40.

Indicator diagrams have many peculiar defects, and are due to several causes, chief of which are the engine, indicator and in many cases the man who manipulates the indicator. Frequently we hear engineers and manufacturers of indicators boast about their indicators taking such beautiful diagrams, but no engineer who understands the rudiments of his business will make or recognize any such claims. The value of an indicator depends upon its correctness and sensitiveness in responding to the most delicate variations in pressure, and the value of the diagram depends upon its correctness and not its beauty. Unless the utilization of steam by the engine is perfect (which case has never been found) the diagram cannot be perfect.

Some engineers consider that the "how cheap" so-called indicators are good enough. But an inferior instrument is the poorest recommendation any engineer can offer, and we would advise all engineers to follow the good old rule that "the best is none too good and always the cheapest."

In the manner of attaching, ofttimes what is wrong is considered nearly correct, and what is nearly correct is considered perfect, and the summing up amounts to a blunder that ends nowhere. A diagram is also frequently distorted by improper tension of the drum-spring, and in applying the pencil to the paper during more than one revolution it does not follow the same path of the admission line.

The length of the diagram being nearly five inches is too long for a two-inch drum, unless the engine is running very slow. A more satisfactory result could be obtained with a four-inch diagram on slow speed engines.

DIAGRAM FROM A MODERN CORLISS ENGINE SHOWING FAULTY CONSTRUCTION.

Fig. 46 shows a diagram from a 32″×60″ Corliss engine. Revolutions, 56. The Corliss engine, as manufactured by the different builders, varies considerably in the economical utilization of steam, for which this style of engine has become so prominent. But like all other styles of engines they are more or less faulty in construction and must necessarily follow

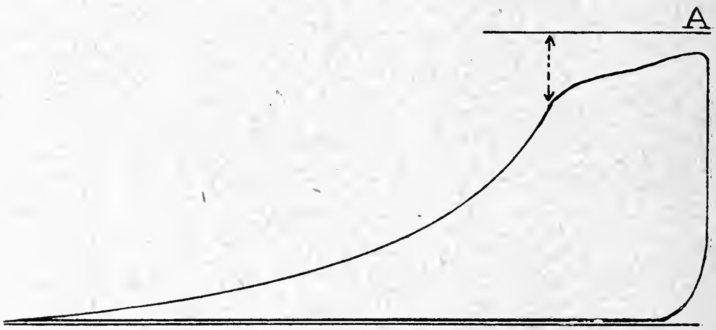

FIG. 46—SCALE OF SPRING 40.

with a corresponding loss of power and economy. The line A represents the pressure above the throttle valve and is the point from which the engine is responsible for the loss of pressure at any point on the steam line. The loss of pressure at the commencement of the stroke is four pounds, and in some cases I have found a considerably greater loss, when it should have been only a fractional part of a pound. Frequently an engine may maintain nearly a straight steam line while the loss of pressure between the point above the throttle valve and cylinder may be considerable. At the point of cut-off, as shown by the dotted line, the pressure has fallen fifteen pounds.

From time to time new styles of engines are put upon the market which we duly appreciate. But the majority of the present makes of engines are so deficient in the use of steam that if steps were first taken to bring this important factor to a degree of perfection it would be a step in the right direction.

If the Corliss engine be converted into a high-speed engine, the same as men frequently advocate, it would be simply a high-speed engine in nearly every sense of the word, and would lose all the advantages that the slow-speed and long-stroke engine gains in economy above the high-speed and short-stroke, viz.: a high initial pressure is realized and maintained in the cylinder; the points of cut-off are better defined, which also causes less reduction in pressure. The expanding pressure can be maintained nearer full stroke and be properly released. The back pressure is less, and a lower cushion is necessary. The percentage of clearance is less, which also plays an important part.

CHAPTER VIII.

SETTING AUTOMATIC RIDING CUT-OFF VALVES WITH THE INDICATOR.

While the riding cut-off engine, as manufactured by the different makers, also varies considerably in its details, a general understanding of the type as here illustrated and described and the method of adjusting, will enable anyone to readily adjust the valves on any of the different makes of this style of engine.

Fig. 47 shows a central section through the cylinder, valves and balance pistons. The left hand end being a section through one of the balance pistons I. The steam enters at D, through passages *aa* and balance pistons to the interior chambers II of the valve, as shown by the arrows, in which the steam chest pressure is maintained. The balance pistons are packed with steam metal sprung rings between it and the followers F, thus preventing the steam from leaking out around the balance pistons into the exhaust K at the bottom and faced to work steam tight on the cover plate of the valve, also preventing the steam from leaking out between the face of balance piston and the face of cover plate. When steam is shut off from the engine the coiled steel springs E serve to hold the balance pistons and the main valve to their seats. The balance pistons have only the necessary area to hold the valves to their seats during admission, and since the force of the balance pistons is constant, this force is too great after cut-off and during

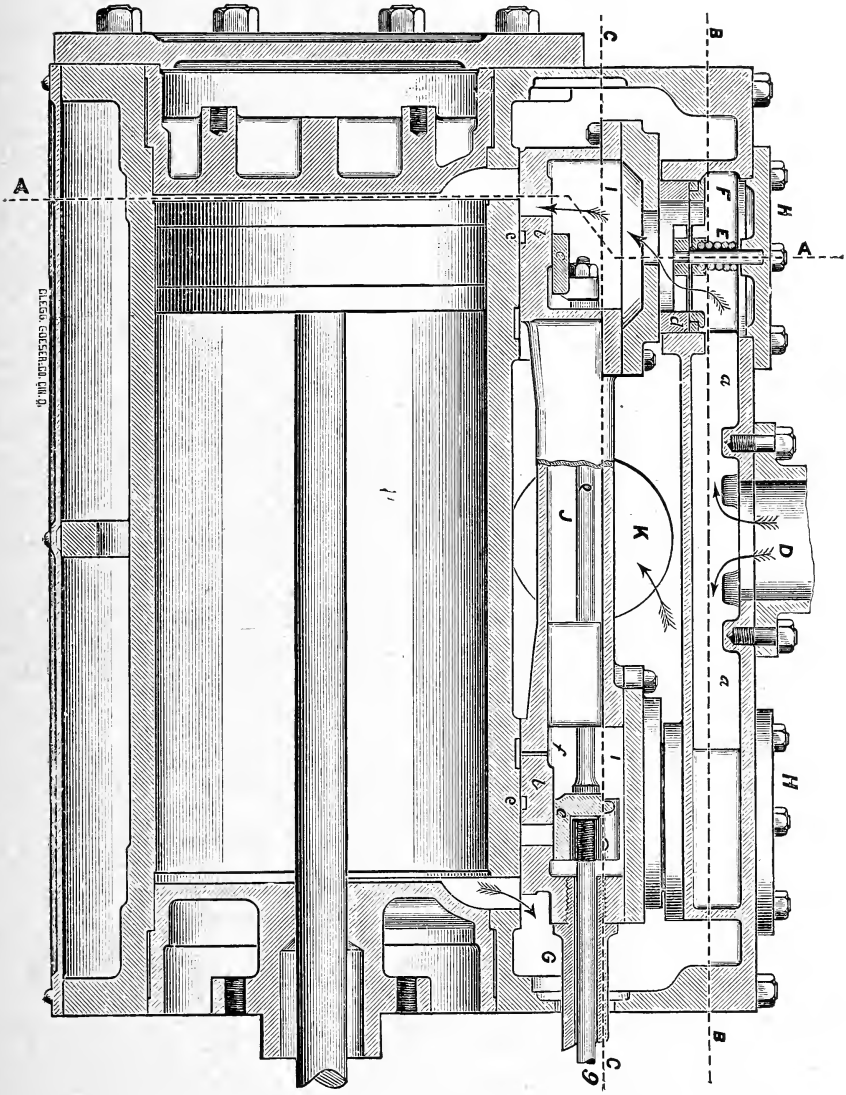

Fig 47.

release, and to counteract such excess, shallow recesses or relief chambers, as they are called, are formed in the valve seats corresponding to the cylinder ports in shape and area, and are filled with exhaust pressure, after which the steam chest pressure is admitted underneath the valve from the interior of the valve, through the small holes *f*, as shown near the crank end. This pressure is released into the exhaust by the movement of the valve a trifle before the admission, as shown at the head end. Provision also is made to permit the main valve to raise from the seats and release any undue pressure that may accumulate in the cylinder on the return stroke of the piston.

Channels *ee* are cut across the valve faces near its ports which serve as passages into the exhaust for any steam that may accumulate under the valve through any imperfection of fit, which otherwise tend to throw the valve from the seats. These must be of sufficient capacity to permit the free escape of steam that may leak into them. Long wear reduces their depth and requires deepening or otherwise enlarging.

Steam is admitted into the cylinder from the interior of the valve, as shown by the arrow as just beginning to admit steam at the head end of the cylinder and the main valve is also moving toward this end, while the other cylinder port is partly open for the exhaust, as shown by the arrow; therefore the exhaust edges are at the ends of the valve and the steam edges near their inner margins.

The cut-off valve is formed by two plates, shown at *cc*, rigidly connected by two rods and operated inside of the main valve.

Fig. 48 shows the shaft governor. It belongs to that class of riding cut-off automatics in which the function of the governor is to move the loose eccen-

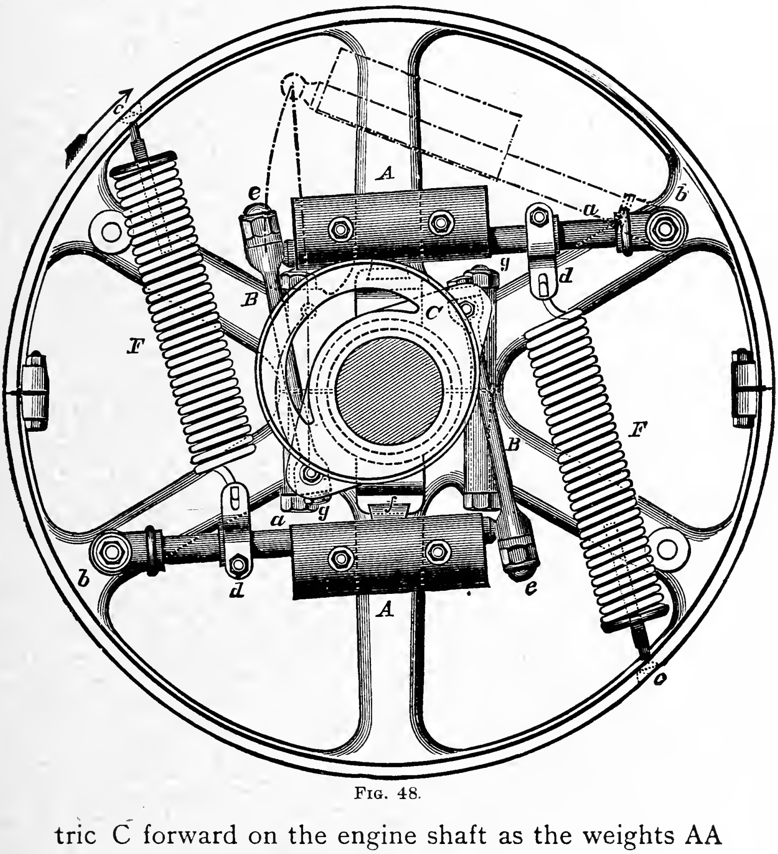

FIG. 48.

tric C forward on the engine shaft as the weights AA move outward by centrifugal force in opposition to the force of springs FF to the position as indicated by the dotted lines, near one of them, as the speed of the engine increases by the decreasing load. By this means in advancing the eccentric, it brings the cut-off valve around to meet the port of the main valve earlier, and the steam is cut off at an earlier point in the stroke. As the speed of the engine decreases by the increasing load the weights move inward and revolve the eccentric backward, and the steam is cut off later. Arms *aa* are pivoted to the governor wheel at *bb* and connected at their other ends by links BB to ears on loose eccentric.

To decrease the speed, more weight at AA is added, or the weights may be shifted farther outward on the arms, providing other parts do not interfere, and less tension is given to the springs FF which are adjusted by screws *c c*. To increase the speed, the adjustments are made *vice versa*.

The amount of tension on the springs FF to give the closest possible regulation is determined by increasing their tension until the engine shows indication of racing, then the tension should be slightly lessened. When the tension of the springs is too great in the inner half of movement range as compared with the outer half, or, in other words, that the regulation of speed is less sensitive with light loads than heavy, the auxiliary springs are applied to assist in throwing the arms outward, leaving contact with the fingers at mid-stroke on which they act. The tension of FF must generally be about one-third to one-fourth greater than could be carried without them. If too great tension is given to the auxiliary springs they will start outward at a speed too much below the normal, or by too much loss of speed at heavy loads.

The direction of motion is indicated by the arrow, and to reverse the engine, the arm as shown at the top, is removed to the unoccupied hole below it, and its side now shown, is turned toward the wheel. The other arm is removed to the unoccupied hole above in like manner, and the other parts are changed to correspond, so that the arrangement will correspond exactly with a view of the cut as it appears held up to a strong light and viewed from the back, or as it would in a looking glass. The positive eccentric is set to follow the crank with the same number of degrees on the other side of the crank, and the automatic eccentric C is set to correspond, by shifting the governor wheel around on the shaft. In the manner of adjusting with the indicator it is necessary that certain preliminary adjustments be made; first, that the rocker arm A which is carried by the positive eccentric, as shown in Fig. 49, to which the eccentric rod B is connected, shall be at right angles to the eccentric rod and vibrate to equal distances each side of a plumb line from its centre of vibration, when it is a horizontal engine and setting level. This may be determined by the explanation previously given in Fig. 22, or the governor wheel and positive eccentric may be loosened and revolved on the shaft. The vibration of the rocker arm C of the automatic eccentric is determined in like manner, when the arm D is plumb and at right angles to the valve stem, and the arm C is at right angles to the inclined eccentric rod E. When it is a vertical engine and straight connected, the centre line of the rocker arm must be level at mid-stroke and at right angles to the rods. The remaining adjustments are

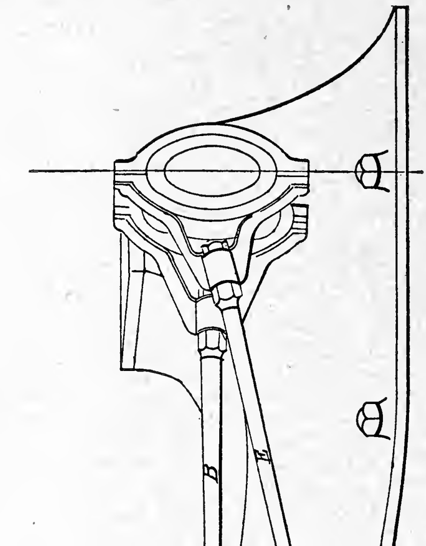

FIG. 49.

made by use of the indicator; first, equalize the compression, admission and release on the main valve stem, which are all controlled by the main valve; and supposing a pair of diagrams be obtained, as shown in Fig. 50. The compression and admission of the head end

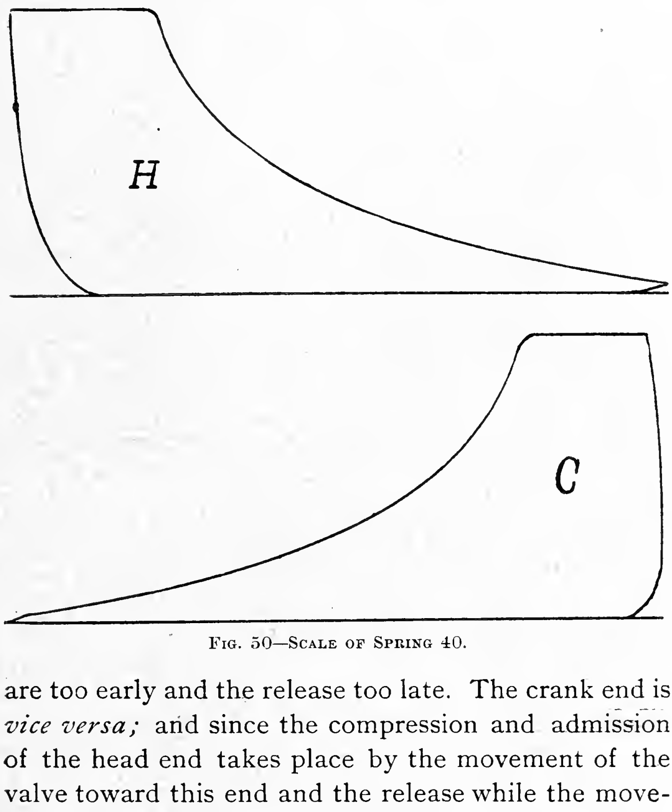

Fig. 50—Scale of Spring 40.

are too early and the release too late. The crank end is *vice versa;* and since the compression and admission of the head end takes place by the movement of the valve toward this end and the release while the movement is toward the crank end, it is plain to be seen the valve stem of the main valve is too long and should be shortened until the compression, admission and release are equal on both ends of the cylinder. Whether they are perfect, too early or too late, they should be equally so from both ends.

Should a pair of diagrams be obtained as was shown in Fig. 27, where the compression, admission and release are late on both ends of the cylinder, it only requires to advance the positive eccentric until the admission lines are perpendicular and the compression and release will also be equal and perfect, providing the valve is properly constructed. But should the results be too early, as was shown in Fig. 28, the eccentric should be set back accordingly. Sometimes with improperly constructed main valves, it is necessary to sacrifice a little of the perfect admission in order to obtain a satisfactory cushion. The cut-off is next equalized. This is accomplished by adjusting the valve stem of the cut-off valve. Should the cut-off on the crank end be the earliest, as was shown in Figs. 30 and 31, and since the cut-off on the crank end of the cylinder takes place while the valve is moving toward this end, it is plain to be seen that the valve stem is too short and should be lengthened; and should the cut-off on the head end be the earliest, the valve stem should be shortened accordingly.

The last adjustment to be made is to set the governor wheel, and it should have no more advance than is necessary to hold the engine to speed under its friction load and highest pressure. This is determined by placing the engine on either centre, mark the cross head and guide, then revolve the engine in the direction of motion about $\frac{3}{4}''$ travel on the crosshead. But this depends principally upon the amount of friction and size of engine, pressure and tightness of piston and valves, and with the weights AA blocked to their outward movement to the position indicated by the dotted lines near one of them, set the governor wheel so that the cut-off valve is just beginning to close the port of the main valve. To test the adjustment, run the engine with its friction load and highest pressure, and should the speed increase considerably above the normal, advance the governor wheel accordingly.

From these explanations it can readily be seen that the manner of adjusting riding cut-off valves with the indicator is not so intricate as generally supposed, but is really a very simple undertaking when thoroughly understood.

DIAGRAMS FROM AN ENGINE ON WHICH AN INDICATOR HAD NOT BEEN USED FOR ELEVEN YEARS.

Fig. 51 shows a pair of diagrams from a $16'' \times 32''$ automatic riding cut-off engine. Diameter piston rod, $2\frac{1}{4}''$; revolutions, 93. This engine was erected eleven years ago, at which time the valves were adjusted with the indicator, and since then no re-adjustments have been made. The unequal distribution of steam is caused by the valve being out of its proper position, which no doubt was caused by frequent keying up. The head end exhaust edge closes too early, causing a too high compression, and a too early admission follows. The full pressure is admitted into the cylinder before the piston has completed the stroke, as shown by the admission line leaning outward, consequently the pressure in the cylinder must be forced back into the steam chest as the piston completes the stroke. When the piston has completed the stroke the pressure in the clearance space and steam chest equalize and the pencil falls, causing a peak above the diagram as shown. The com-

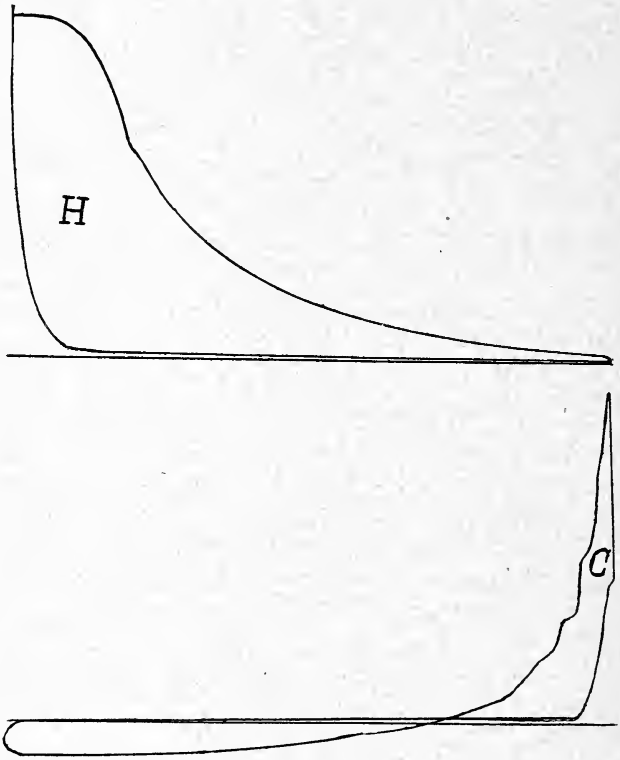

FIG. 51—SCALE OF SPRING 40.

pression is the highest pressure compressed in the clearance space; therefore, the compression in this case is measured at the highest point at the commencement of the diagram, which is $74\frac{1}{2}$ pounds, and at this time there is a partial vacuum of 5 pounds on the other side of the piston, which is measured on the crank end diagram near the termination of the expansion curve; therefore, the cushion is $79\frac{1}{2}$ pounds and is the highest unbalanced pressure realized on the piston at the instant it comes to a standstill. This will more than stop the piston before the end of the stroke, and were it not for the energy stored in the fly wheel the engine would stop. Considering the length of stroke, speed and character of the compression curve about 15 or 20 pounds cushion should be sufficient to secure smooth running.

The crank end diagram has no steam line because the cut-off took place before the piston had made any movement. The highest pressure realized at this end is only 70 pounds, as the cut-off took place even before the full pressure was realized in the clearance space. The admission is too late, as shown by the admission line leaning slightly inward. The compression rises more rapidly than on the head end, because the piston approaches nearer the crank end, due to the clearance being unequally divided. The compression curve, at the point where it intersects with the admission line goes out partly in a straight line and is caused by condensation because the temperature of this end of the cylinder is lower than the temperature of the compression. When the expansion curve reaches the back pressure line the piston has only moved about one-third of its stroke and the pressure is balanced on both sides of the piston, and were it not for the motion of the fly wheel and reciprocating parts the engine would stop at this point of the stroke. As the piston completes the stroke the expansion curve continues to fall below the atmospheric line, causing a partial vacuum on the admission side of the piston and the fly wheel and reciprocating parts must force the remaining stroke of the piston against a pressure equal to the vacuum formed on this side of the piston, plus the back pressure, above the atmospheric on the exhausting side.

The loop embraces the greater part of the diagram, consequently the countermotion is greater than the load carried by the steam, hence it takes power from the head end to drive the crank end. The unequal distribution of steam on each side of the piston caused considerable variation of speed during each revolution and the belt was constantly whipping.

DIAGRAMS SHOWING INCORRECT ADJUSTMENTS.

Fig. 52 shows a pair of diagrams from a 12″×24″, riding cut-off automatic engine. Revolutions, 104. These diagrams show the manner in which the builder adjusted the valves with the indicator. The adjustments are incorrect and the distribution of steam on each side of the piston is unequal. The compression is too high and follows with a too early admission, and the full pressure is admitted into the cylinder before the piston has completed the stroke. The compression and admission take place on the head end still earlier than the crank end, therefore the stem of the main valve should be shortened and the positive eccentric set back until the admission lines are perpendicular, which will cause a later closing of the exhaust valves and a lower cushion. The cut-off on the crank end is the earlier, therefore, the stem of the cut-off valve should be lengthened, but owing to the fact that the position of the positive eccentric must be changed, the governor wheel of the

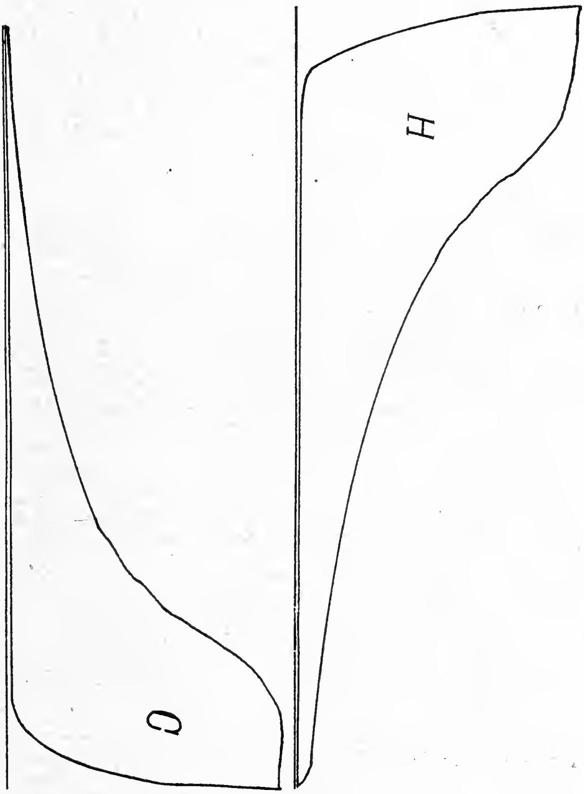

FIG. 52—SCALE OF SPRING 40.

automatic eccentric must be set to correspond.

The compression of the head end is 59 pounds and the counter-pressure is one pound, thus giving 58 pounds cushion. The cushion of the crank end is 56 pounds.

SETTING AUTOMATIC RIDING CUT-OFF VALVES BY THE SOUND OF THE EXHAUST.

Fig. 53 shows a pair of diagrams from a $16'' \times 34''$ automatic riding cut-off engine, revolutions 100. The valves were set by the sound of the exhaust, and because the engine runs smooth as indicated by a sufficient cushion, the owner thought that the valves had been properly adjusted, but when the fuel bills increased and the engine became unable to carry the load, it was a sticker in his mind and he was forced to search for the truth. Without any further guesswork he called in the man with the indicator, who upon inspecting the diagrams saw at a glance that the difficulty could not be remedied by any adjustments on the outside of the engine. Had the main valve stem been lengthened so as to make the admission lines equally late on both ends it would necessitate a considerable advancing of the eccentric in order to obtain a proper admission, which would cause a too high cushion. The steam chest cover was taken off and the valve is formed of two plates and connected by rods. It was found that the head end plate had been moved toward the crank end, and since the valve admits steam while moving away from the crank end, it is plain to be seen that this caused the late admission on the head end. When the piston commenced its stroke at the head end the steam valve at first remained closed, the admission line fell below the compression curve until the piston had moved a considerable distance, when the steam valve opened and raised the pressure, thus forming the loop. But since the piston had moved such a considerable distance before the admission commenced, it caused an undue loss of admission pressure. The

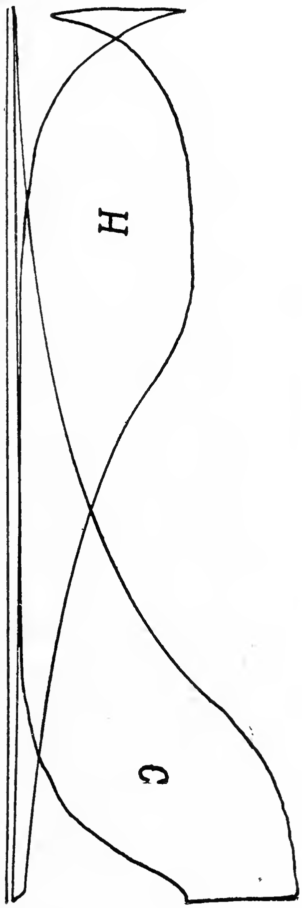

FIG. 53—SCALE OF SPRING 40.

long compression curves indicate excessive clearance or leakage out of the cylinder, which is shown in the crank end compression curve.

CHAPTER IX.

SETTING SINGLE VALVE AUTOMATICS WITH THE INDICATOR.

While the single valve automatic engines as manufactured by the different makers also vary considerably in their details, a general understanding of the type as here illustrated and described, and the method of adjusting them with the indicator, will enable anyone to readily adjust the valves on any of the different makes of this style of engine. The difference in adjusting depends chiefly upon whether the valve admits steam into the cylinder from the center of the valve or at the ends. With valves admitting steam from their ends the eccentric leads the crank, and while the piston is moving rapidly the valve is moving slowly, and *vice versa*, due to the angularity of the connecting and eccentric rod, which is well understood by all engineers familiar with the mathematics of the steam engine. This objectionable feature is obviated by admitting steam into the cylinder from the inner margins of the valve and the eccentric follows the crank, which causes the piston and valve to move in unison.

Fig. 54 shows a central section through the cylinder, steam chest and the right end of the valve; the other end being a section through the valve. The steam chest pressure surrounds the valve, and as it is moving toward the head end, the live steam edge of the valve is shown by the lower arrow as just beginning to admit steam into the head end of the cylinder, another passage being in a port at the other end of the valve and through the tubing, as indicated by the course of arrows, thus forming a double port valve. The steam is exhausted at the other end of

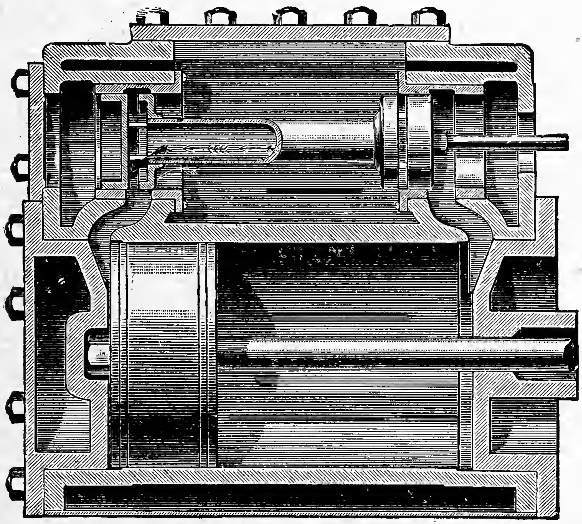

FIG. 54.

the valve, and when the piston is at the crank end the valve is also in the opposite direction.

Fig. 55 shows the shaft governor. It belongs to that class of single valve automatics in which the function of the governor is to move the eccentric across the engine shaft, altering its throw and varying the travel of the valve, either by changes of load or changes of steam pressure. It consists essentially of a governor wheel which is fixed on the engine shaft, to which are hinged the weights, 1, 1. These weights are moved outward by centrifugal force in opposition to the springs. The inner eccentric, C, having ears attached and connected with the weights by rods, 2, 2. The outer eccentric ring, D, is free to

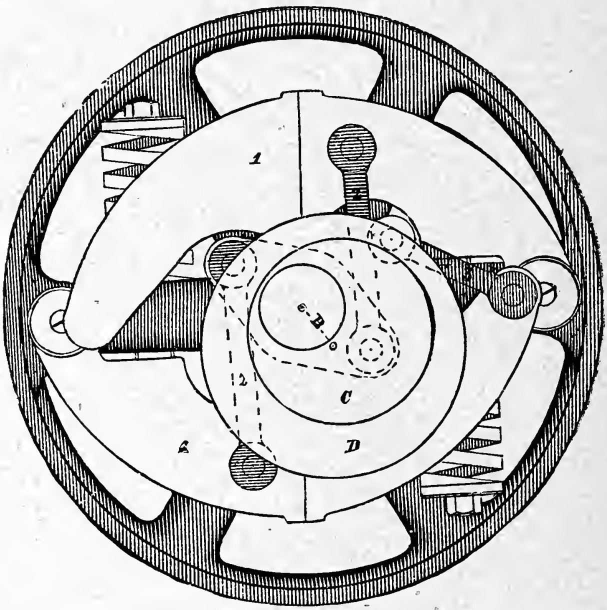

Fig. 55.

turn on the outside of the inner eccentric, and is connected to the toe of one of the weights by rod, 3.

On this outer eccentric ring is placed the usual eccentric strap (which is not shown in the cut) to which the eccentric rod is connected. When the engine has its greatest load the weights will be in the inner position, as shown in the accompanying cut; and the throw of the eccentric will be the greatest, consequently the valve must return to cut-off correspondingly later. The eccentricity of the two combined eccentrics is then the distance, as shown at B. As the speed of the engine increases, by the decreasing load, the weights overcoming the springs move out-

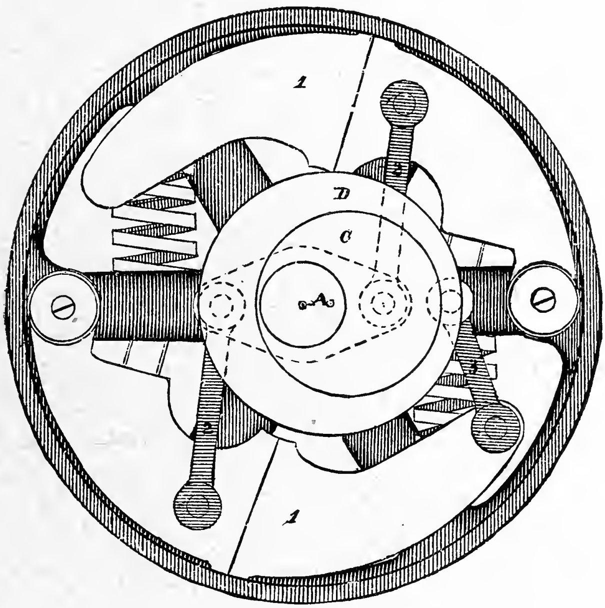

Fig. 56.

ward, which decreases the travel of the valve and the steam is cut off earlier. When the weights are in the extreme outer position, as shown in Fig. 56, the point of cut-off should be sufficiently early so that the speed of the engine does not materially exceed the normal with the friction load and highest steam pressure. The eccentricity of the two combined eccentrics is then the distance shown at A.

The manner of changing the speed and direction of motion will be about the same as was explained in Fig. 48. But in some types of governors it is necessary to use parts differently constructed. An engine that is equipped with a shaft governor not contained in the fly wheel is, of course, more convenient for altering the valves, as the eccentric can readily be changed to any position so as to obtain the proper action of the valve under various conditions, according to the engineer's own fancy. Such a device any intelligent engineer cannot fail to appreciate. However, with nearly all governors that are contained in the fly-wheel it is necessary to change the position of the fly-wheel on the engine shaft in order to alter the valve.

When single valve engines are equipped with a rocker arm it is necessary that it shall have the proper vibration, as explained in Fig. 22. The remaining adjustments are made by the use of the indicator. First, the load or steam pressure, or both, should be varied so the cut-off on both ends of the cylinder will average about one-quarter, which is about the center of distribution in a single valve automatic engine, and the cut-off varies unequally from this point, due to the angularity of the eccentric rod. Supposing a pair of diagrams be obtained, as shown in Fig. 57, the head end exhaust closure and lead is too early and the cut-off and release too late. The crank end is *vice versa*, and since the compression, admission, cut-off and release are all controlled by the same valve, and the compression and admission take place at the head end, while the movement of the valve is toward the head end, and the cut-off and release while the valve is returning, it is plain to be seen that the valve stem is too long and should be shortened until the results be equal on both sides of the piston. Whether they are too early or too late they should be equally so on both ends. Should the crank end compression and admission be too early and the cut-off and release too late, the valve stem should be lengthened accordingly. Should the results then be too late, as shown in Fig.

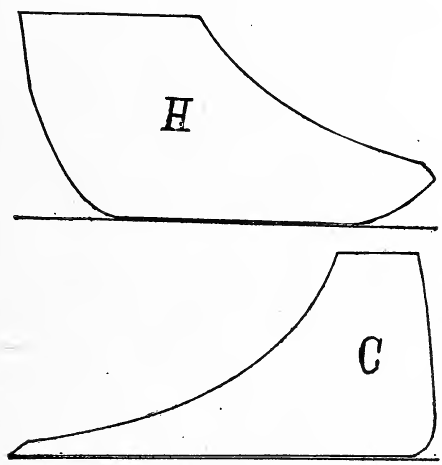

FIG. 57—SCALE OF SPRING 40.

27, the eccentric should be advanced, and if too early, as shown in Fig. 28, the eccentric should be set back accordingly.

It frequently happens that it is necessary to sacrifice the accuracy of one of the main functions, either compression, admission, cut-off or release, in order to accommodate the other three, but when the engine is overloaded it is then necessary to make up the deficiency in cushion with live steam.

AN EXCELLENT DIAGRAM FROM A MODERN HIGH-SPEED ENGINE.

Fig. 58 shows a diagram taken from a $15\frac{1}{2}'' \times 15''$ single valve automatic engine. Revolutions, 250. This cut gives the general features of a well-formed indicator diagram of a high-speed engine, and one of the most beautifully formed steam lines that scientific steam engineering can produce. The maintenance of such an absolutely straight steam line to the absolute point of cut-off is the most difficult feature to produce

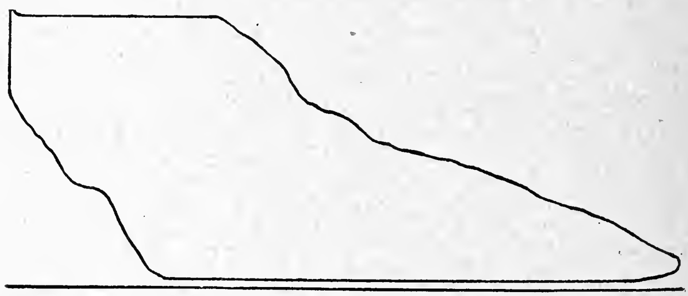

Fig. 58—Scale of Spring 50.

in a high-speed engine. This is accomplished in this engine to the highest degree of perfection. The admission pressure approaches the pressure above the throttle valve to a fractional part of a pound, and unless all other engine builders can match it in as perfect a utilization of steam, they must necessarily follow, taking this figure as an example to imitate, while endeavoring to produce an engine that may be of modern character. The point of cut-off embraces an equal degree of perfection, and is defined with the utmost rapidity. The expansion and compression curves are wavy, caused by fluctuation in pressure and oscillations of the indicator piston. The admission line also has its beauty as the full pressure was admitted into the clearance space at the instant the crank was passing the dead center, and this rapid admission caused the pressure to fluctuate above the normal and caused the peak above the diagram. The undue back pressure is caused by restricted area through the exhaust pipe. This diagram will bear a great deal of study by all engineers, especially those in charge of high-speed engines, where the diagrams depart from the prominent features as attained in this diagram. However, the engine is overloaded, and to obtain better economy and carry the same load the boiler pressure should be increased which would result in an earlier cut-off and a lower release pressure.

If engineers will compare the absolute efficiency of different styles of engines in an accurate manner, it will prove a revelation that will force them to acknowledge that many of the so-called high grade engines are so deficient in the utilization of steam that they never should have been recognized as a modern steam engine.

A DIAGRAM FROM A MODERN HIGH-SPEED ENGINE, SHOWING FAULTY CONSTRUCTION.

Fig. 59 shows a pair of diagrams from a 10″×12″, single valve automatic engine. Revolutions, 255. These diagrams were taken after the valve was adjusted by the builders, and it proves whether or not the engine is capable of the proper performance so as to bear them out in their claims. The diagrams show that the adjustments were judiciously made, as the four points are fairly equal—namely, the admission, cut-off, release and compression, and these features being accomplished the builders considered that they had a novel result. But these diagrams, upon inspection by a "card sharp," only proves what I have already stated, viz., the majority of engine builders are not as close readers of indicator diagrams as they are generally credited with being; and unless they are proficient in the application of the indicator and the close observers of the diagrams, it must necessarily follow that their engine is as deficient as the conclu-

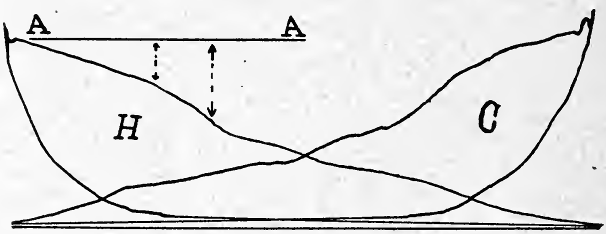

FIG. 59—SCALE OF SPRING 40.

sions based upon the diagrams. This is self-evident, when it is considered that in these diagrams all the prominent features that were attained in Fig. 58 are lacking. By a comparison of the difference of efficiency of the two diagrams mentioned it should be sufficient evidence for engineers that such engines should never be placed in the field where fuel, boiler capacity, etc., is of any value without any ifs, ands, or buts. The steam pressure from above the throttle valve, not being known, the loss of pressure at any point on the steam line can only be calculated from the initial pressure line A A, which line is located by measurement on the scale of the spring at such a distance above the atmosphere line, as is represented by the initial pressure. Had the pressure above the throttle valve been known, and should have been taken with another instrument at the same time the cylinder diagrams were taken, this loss of pressure could be accurately determined. At the point represented by the first dotted line the steam valve commences to close and the pressure has fallen 15 pounds below the initial pressure. The valve has not fairly closed until at the point shown by the second dotted line because the expansion curve at this point changes from convex to concave, and the loss of pressure is 20 pounds, and there is about an equal loss of pressure at the other end of the cylinder.

This great loss of pressure during admission generally follows with a serious loss of pressure between the highest pressure realized in the cylinder and that above the throttle valve and is a corresponding loss of power and economy. This is only one of the many objectionable features that cause steam users to discover why they lack boiler and engine capacity when, according to the engine builder's rating, the engine should be more than ample. The exhaust valves open and release the pressure before the point of release on the diagrams meet with the compression curve and is a too early release.

The ports were very long and indirect, causing a large per cent of clearance which also is an objectionable feature, and is shown by an early exhaust closure in order to produce the necessary cushion. The admission is too early, as shown by the admission lines leaning outward causing the peak above the diagrams, which, however, are common with high speed engines. This, however, may be intentional so that the valve may be more open, so as to better maintain the pressure when the movement of the piston commences.

A considerable less pressure is realized on the head end which may be accounted for from the fact that when the movement of the piston was rapid the movement of the valve was slow, and *vice versa*, because the valve admitted steam at the ends of the valve.

This faulty construction is not confined to a single make of engine but is found in many single valve engines, and the manufacturers of such engines should make the necessary changes.

SETTING SINGLE VALVE AUTOMATICS BY THE SOUND OF THE EXHAUST.

Fig. 60 shows a pair of diagrams from a 10″ × 12″ single valve automatic engine. Revolutions 285. The valve of this engine was set by the sound of the exhaust and judging from the diagrams the man who followed this method of setting engine valves displayed his ignorance. In this case the sound of the exhaust was equal from the fact that the head end was exhausting no pressure as shown by the expansion curve crossing the atmospheric line before the end of the stroke. The crank end exhaust valve closes too early and forms 77 pounds compression, thus exceeding the steam chest pressure, and when the steam valve opens, the pressure passes into the steam chest and the pencil falls, causing the loop above the diagram. The steam valve closes at a late point in the stroke and the pressure is 22 pounds when it is released. The head end steam valve remains closed and the pressure in the expansion curve is produced by the compression and the leakage of steam into the cylinder. The loops at each end are nearly as great as the other portion of the diagram and, were it not for the leakage of steam into the cylinder, it would require power

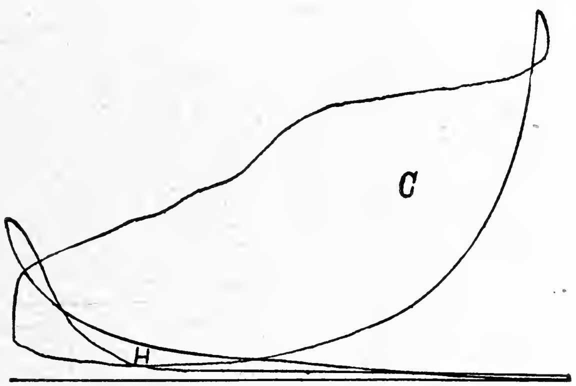

FIG. 60—SCALE OF SPRING 40.

from the crank end in order to drive the head end. The valve admitted steam at the ends of the valve therefore the first adjustment made was to shorten the valve stem. The amount saved in fuel in a few days only, more than paid the cost of adjusting with the indicator, but many of the owners of steam-plants are ignorant as to the economical utilization of steam in an engine and not until they are forced to do so will they go to the expense of having the engine properly adjusted with the indicator.

Fig. 61 is another example of setting single valve automatics by the sound of the exhaust and is taken from a 10″×12″ single valve engine. Revolutions 250. The crank end in this case is exhausting no pressure. The exhaust valve closes and forms the compression but the steam valves does not open in due time and when the piston commences to move away, the pressure falls and forms the loop at the commencement of

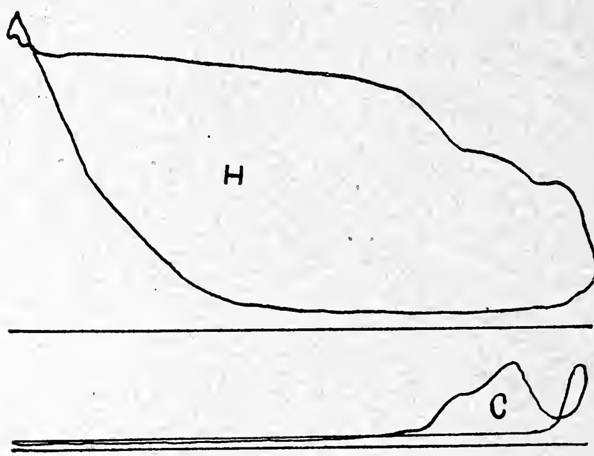

FIG. 61—SCALE OF SPRING 30

the diagram. The steam valve then opens a trifle and raises the pressure but is immediately closed and the expansion curve crosses the back pressure line at an early point in the stroke and should have expanded below the atmospheric line and its failure to do this indicates leakage of steam into the cylinder.

The head and exhaust valve closes too early and the compression rises above the steam chest pressure and when the steam valve opens the compression and the steam chest pressure equalize and form the loop.

A DISTORTED DIAGRAM CAUSED BY INSUFFICIENT TENSION OF THE DRUM-SPRING.

Fig. 62 shows a diagram from an 8″ × 10 single valve automatic engine. Revolutions 336. The diagram shows that the drum-spring of the indicator had insufficient tension and produced a fling at the end of the stroke as shown by the admission lines not merging into each other. If the pencil be applied during several revolutions and the admission line is broader than the width of a single pencil mark, it indicates a defect in the indicator, or reducing motion, due to

FIG. 62—SCALE OF SPRING 60.

vibration of the reducing motion, cord stretching, or improper tension of the drum-spring. The load of this engine varied suddenly and the pencil was applied during one revolution with the heaviest load, as shown by the late cut off, and another revolution when the load was thrown off leaving only the friction load of the engine and machinery as shown by the early point of cut-off. This decrease in load increased the speed and caused a greater fling of the drum. The steam valve closed before the full pressure was admitted into the cylinder and reduced to initial pressure. With such a pressure of steam and early point of cut-off the expansion curve should fall below the atmospheric line, and its failure to do this indicates a leak in the steam valve.

It will be seen that with a single valve automatic engine, as the load decreases the cut-off, release and exhaust closure take place at an earlier point in the stroke, while the lead should remain constant.

The compression shown in the diagram with the heavy load, as indicated by the late point of cut-off, is only 23 pounds, and since the late point of cut-off causes a higher counter-pressure and the late exhaust closing causes a lower compression, the cushion is rapidly reduced.

This is one of the most objectionable features in a single valve automatic engine of this type from the fact that the valve travel is varied by altering the throw of the eccentric which is moved across the engine shaft, due to the changes of load or changes in pressure of steam. As the load decreases the valve travel also decreases and the admission and exhaust are cut off at an earlier point in the stroke.

The compression in the diagram representing the early point of cut-off increases to 41 pounds while at the same time the counter-pressure decreases and the cushion increases rapidly.

Where the exhaust is controlled with a positive eccentric, the compression should practically remain constant and the cushion is only affected by the change in the counter-pressure caused by the variation in the point of cut-off due to changes in load or changes in steam pressure.

One of the admission lines lean slightly inward and indicates a trifle late admission. The peak at the top of one of the admission lines which extends above the steam line is caused by a fling of the pencil.

DIAGRAMS FROM AN OVERLOADED HIGH SPEED ENGINE.

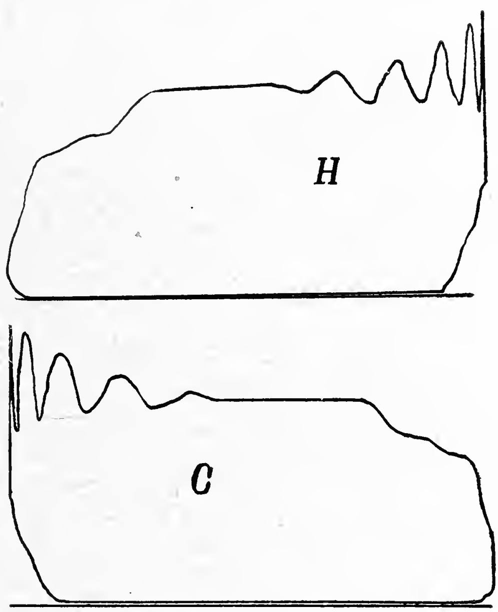

FIG. 63—SCALE OF SPRING 60.

Fig. 63 shows a pair of diagrams from a 10″ × 12″ single valve automatic engine, revolutions 283. This engine was runnning noisy and needed to be frequently repaired. The diagrams show that the high terminal pressure destroyed the cushion which caused the trouble. If the eccentric had been advanced, the lack in cushion would have been obtained by admitting live steam into the cylinder before the piston completed the stroke and the admission lines would lean outward. The engine is overloaded to the limit and should be replaced with one of proper size, which would effect a saving in fuel sufficient to pay for the engine in a short time. The irregularities in the steam lines are caused by oscillations of the indicator piston, due to the rapid admission of steam and high speed, and a spring of higher tension

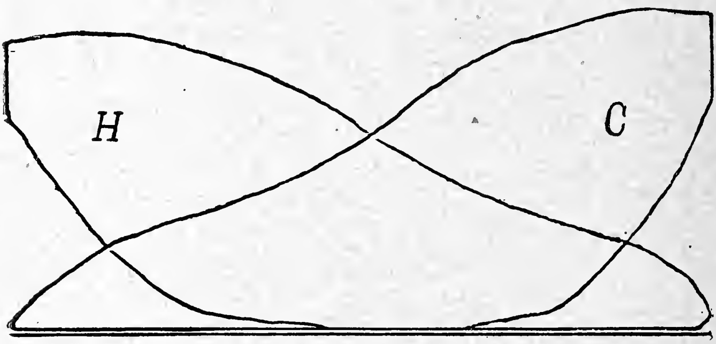

Fig. 64—Scale of Spring 40.

should have been used. The initial pressure in this case is measured from the atmospheric line to a point so that at an average height the oscillations are as much above as below.

DIAGRAMS FROM EACH END OF THE CYLINDER, SHOWING A DIFFERENCE IN THE INITIAL PRESSURE.

Fig. 64 shows a pair of diagrams from a 11″ × 16″ single valve automatic engine, revolutions 175. The initial pressure on the head end is less than on the crank end, and is caused by the valve admitting steam from the ends, in which case while the piston is moving rapidly on the head end the valve is moving slowly, and *vice versa.* Had the valve admitted steam from the inner margins the piston and valve would have moved in better unison.

The points of cut-off are poorly defined, caused by the valve being slow in its action while closing, and is a loss in power and economy.

SERRATED CURVES CAUSED BY UNEQUAL EXPANSION OF THE INDICATOR-PISTON AND CYLINDER.

While an indicator may be free from friction when

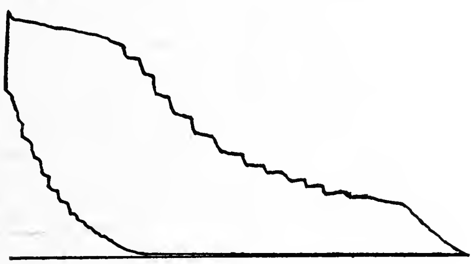

FIG. 65—SCALE OF SPRING 60.

cold, as explained in the chapter on "The Indicator," nevertheless, when in contact with steam, it may be subject to undue friction, due to unequal expansion of the piston and cylinder, and the expansion and compression curves will be serrated, as shown in Fig. 65. This can readily be tested by detaching the spring, and with piston attached raise the piston to its highest position with the pencil motion, and in this position admit steam, and when well heated shut off steam, and immediately commence to raise and lower the pencil motion to its highest and lowest positions, and if there is a suspicion of a catch it is due to this cause. But if slight it may be remedied by inserting a spring of such tension that the piston moves through the entire range of its motion and attaching the indicator to one end of the cylinder, but without attaching the paper-drum to the reducing motion allow the piston to work in communication with the steam in the cylinder a few hours.

Should this fail to remove the defect the instrument is impracticable and should not be used. If engineers will submit the different makes of indicators that they may come in contact with to these simple tests, and as they become skilled in the importance depending upon this indispensable instrument, they will indorse the statements which I have heretofore made against inferior makes of indicators, and agree as men of science that there is want of a more correct instrument than exists, even among the best makes.

THROTTLING OR SLIDE VALVE ENGINE.

The throttling or commonly called the slide-valve engine, usually admits steam at the ends of the valve, otherwise the manner of equalizing the four points, namely, the admission, cut-off, release and compression are the same as the single valve automatic explained in Fig. 57, except that the load or steam pressure need not be varied. This valve is operated with a positive eccentric, and its movement is constant. Advancing, the eccentric brings these four points earlier, and *vice versa*. The governor is placed in the steam pipe and its function is to throttle the pressure accordingly as the speed of the engine increases or decreases above the normal by the variation in load, and is called a throttling governor. It is simply a choker in the steam pipe, and such ancient and wasteful method of working steam should make a scrap-pile blush with shame.

It is frequently assumed that steam plants equipped with slide valve engines in localities where fuel is free of cost, are the proper type of engines to use, and the plant can be installed at a less cost. This is merely supposition and ignorance, as such plants are the most costly in price, first, last and always, for the

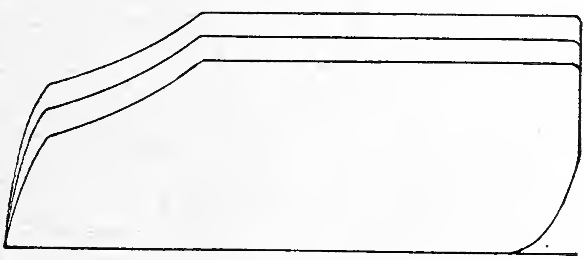

FIG. 66—SCALE OF SPRING 40.

simple reason that it requires about double the boiler capacity, and this additional expense, together with the extra cost for attendants, repairs etc., will more than balance the lesser price of the slide-valve engine as compared with any high grade automatic.

Fig. 66 shows the general features of a well formed diagram from a throttling engine, the attainment of which should be the aim in setting this valve. The release and compression are perfect, the back pressure line merges into the atmospheric line. The steam lines are maintained in straight lines and the points of cut-off which usually take place about $\frac{3}{4}$

stroke in a throttling engine are well defined. The three steam lines at different heights show the action of the governor under variations in load, while in the automatic engines the full pressure is admitted and cut-off at a distance in the stroke, according to the load, as shown in Fig. 31.

TYPICAL COMPRESSION CURVES AND ADMISSION LINES.

The compression curves *a*, *b*, *c*, as shown in Fig. 67, while differently formed, are all caused by more

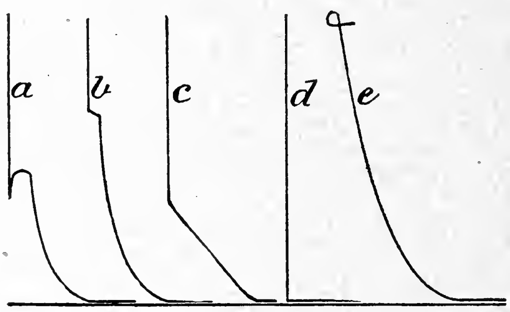

Fig. 67.

or less leak in the exhaust valves or piston; *a* is subject to a greater leak than *b*, while *b* also may be caused by a high compression whose temperature exceeds the temperature of the interior of the cylinder when condensation takes place and is completed in a partly straight line.

Long service will cause the valves in a Corliss engine to wear that portion of the chambers in which it moves, and if the valve rods are altered, the leakage will be greater when the valves rise upon the shoulder, while the valves may be tight when moving in its former position. This leak may more than balance the advantages for which the alterations were made, and the chambers should be rebored and the valves replaced with correspondingly larger ones.

If the compression curves indicate leak only at one end of the cylinder; it is generally caused by the exhaust valve, while if at both ends, it may be the piston, or both.

d has no compression, because the exhaust valve remains open until at the end of the stroke, and the

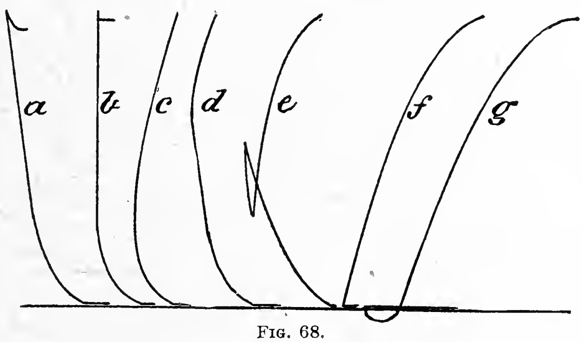

FIG. 68.

back pressure line is completed in a straight line. *e* has too high compression, because the exhaust valve closes too early and the compression exceeds the steam chest pressure, and when the steam valve opens, the pressures equalize and form the loop.

The admission line *a*, as shown in Fig. 68, leans outward and indicates a too early admission, and a peak is formed above the steam line as shown. Frequently a peak is formed as shown at *b*, yet the admission line is perpendicular and is caused by a fling of the pencil motion, which in turn is caused by momentum of the indicator piston. It also may be caused by rapid opening of the steam valve, which causes fluctuation in pressure.

c leans inward and indicates a too late admission; *d* is first too early and then too late, and is accounted for by an improper movement in the opening of the valve. This frequently happens with a Corliss engine in keying up the connections in the hook and eccentric rods when the vibration of the rocker arm becomes unequal, or in any other type of engine that is equipped with a rocker arm. In *e*, the steam valve remains closed at the commencement of the stroke and the pressure falls, forming the loop. *f* indicates no compression and late admission which may be caused by the exhaust valve being open when the entering steam blows into the exhaust and follows with loss of initial pressure.

g has no compression. At the end of the stroke the exhaust valve closes and the steam valve also remains closed at the commencement of the stroke, and as the piston moves away it creates a vacuum in the cylinder, but when the steam valve opens the pressure rises and forms the loop.

DEFECTIVE STEAM LINES.

The defects in steam lines are due to several causes, chief of which are the indicator, improper movement of the steam valves during admission, leaky piston and exhaust valves, restricted area in the steam pipe, throttle valve, steam chests or ports. Some of the peculiar defects frequently encountered, are often regarded as a mystery, but can readily be accounted for and remedied when properly traced with the indicator.

The line A as shown in Fig. 69 shows a defect frequently found, especially in slow speed engines and is generally caused by leakage of pressure out of the cylinder. It will be seen that the pressure falls rapidly at the instant the piston commences its stroke and is caused by leakage of pressure out of the cylinder, but soon rises and restores the pressure equal to

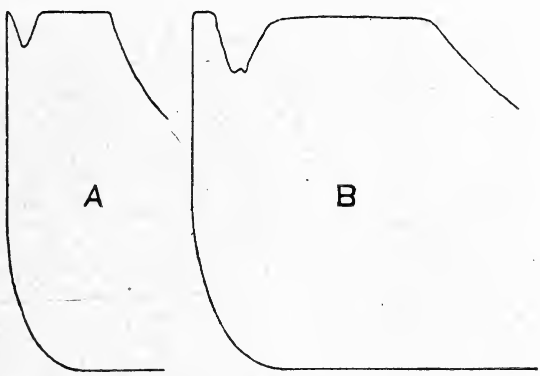

FIG. 69—SCALE OF SPRING 40.

the pressure at the commencement of the stroke which is accounted for by the fact that the steam valve at this point in the stroke is fully opened, and since the piston has only moved a short distance, the volume is comparatively small and requires a less quantity of steam to raise the pressure. Had this loss in pressure taken place at a later point in the stroke, when the volume would be greater, it would require a greater quantity of steam in order to restore the pressure and unless the engine is running slow the pressure could but slowly raise and would cause a corresponding lower pressure. But, however, many of the indicator diagrams showing defects, are a false record of the action of the steam, and may be attributed to an inferior instrument or improper application of the indicator, or both.

The steam line B has two points of cut-off and some of the reasons frequently given for this are ofttimes amusing. While some of these wildcat ideas may be the case, nevertheless, the general cause is due to the fact that in a slow speed engine, equipped with a shaft governor, and especially those having a light fly-wheel, and a short valve travel, in which case the governor needs only to slightly alter its position in order to vary the point of cut-off, and frequently causes a re-opening of the steam valve, making two points of cut-off. In this case, the engine had a light load as shown by the first point of cut-off but when the piston had moved only a short distance, an excessive load was suddenly applied which reduced the speed and the governor being sensitive in its action, instantly re-opened the valve causing the second admission, but owing to the fact that the piston had already moved a considerable distance, the pressure could not be restored equal to the first time of admission.

CHAPTER X.

ANALYSIS OF BOILER FEED PUMP DIAGRAMS.

The maintenance of an economical utilization of steam in pumps is no less an important factor than a properly constructed pump, which fact has been greatly neglected by nearly all engineers. The indicator should be applied to the steam and water cylinders of every pump in order to determine that the plunger and valves are free from leaks, and that the speed of the pump does not exceed the capacity of the suction and discharge passages, so as to create undue suction and discharge pressure in the water cylinder. The length of strokes should not vary and the steam piston should approach the ends of the cylinder as nearly as possible; the ports should be short and direct, so as to reduce the percentage of clearance to a minimum, because the steam in the clearance space in a pump is a total loss on account of there being no expansion. The water cylinders of all pumps are subject to leakage and they ofttimes lack capacity which could be traced with the indicator and remedied, when their capacity would be more than ample.

Fig. 70 shows the general features of a well formed diagram from the steam cylinder. The exhaust valve closes just in time to prevent the entering steam from passing into the exhaust, as shown by the slight compression curve at the left hand end of the diagram; and before the piston completes the stroke the steam valve opens and stops the piston with live steam, so as to prevent it from striking the head, as shown by the admission line leaning outward. With higher speed the piston moves further before it can be stopped, and this increase of stroke causes the admission line to lean outward still more.

The steam line is straight and parallel with the atmospheric line which indicates a uniform piston speed, which otherwise would be wavy.

The release line as shown at the right hand end of the diagram is perpendicular, and the exhaust pres-

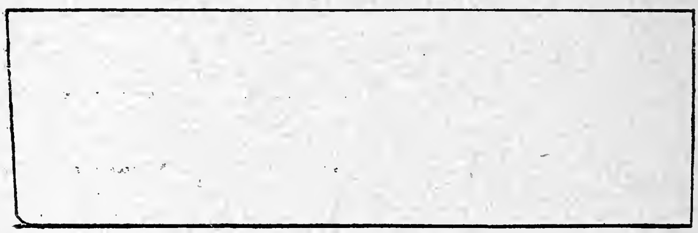

Fig. 70—Scale of Spring 40.

sure is reduced to a minimum before the piston begins to return. The back pressure line merges into the atmospheric line, which shows that the exhaust passages are ample for the speed.

Fig. 71 shows the general features of a well formed diagram from the water cylinder. The line A is the atmospheric line from which the discharge and supply pressure above, and suction below is measured. The lower line is the suction line. The suction is measured on the scale of the spring from the suction line to the atmospheric line, and represents by its distance below the atmospheric line, the height the water is lifted, resistance in passing through the suction pipe, ports, and the pressure necessary to raise the discharge valves. If the suction passages are ample for the speed, the suction line will be maintained in a straight line, and only the necessary distance below the atmospheric line to accomplish this result with freedom. When the water is supplied by pressure, this line will be elevated above the atmospheric line, and is then called the supply line, and

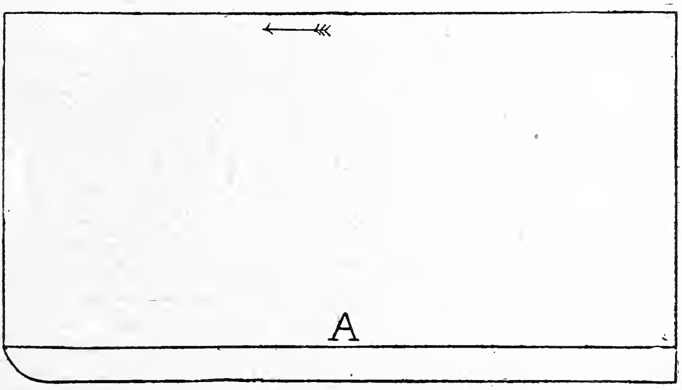

FIG. 71—SCALE OF SPRING 40.

represents by its distance above the atmospheric line the pressure at which the water enters the cylinder. The loss of pressure through the supply passages should not be greater than is necessary for the free passage of the water to the cylinder, because with the least amount of suction (when the water is lifted) or the greater distance the supply line is elevated above the atmospheric line (when the water is supplied by pressure), the greater will be the gain in power and economy.

When the plunger has completed the stroke, and if the cylinder is filled with water, and there be no leakage in the plunger, valves and discharge passages, the compression line as shown at the left hand end of the diagram will erect perpendicular when the plunger returns, because water practically, cannot be compressed nor expanded.

The upper line is the discharge line, and if the discharge passages are ample for the speed, this pressure line should not exceed the boiler pressure or any other point of discharge more than is necessary to raise the valves and resistance through the passages, in which case the steam gauge and indicator must agree, and this line should be maintained in a straight line and parallel with the atmospheric line. The release pressure line, as shown at the right hand end of the diagram, should fall perpendicular when the piston begins to return.

The capacity of the suction and discharge passages can also readily be tested by decreasing the speed of the pump, and should the suction and discharge pressure materially decrease, the area in the passages is insufficient for such a speed and should be enlarged.

DIAGRAM FROM BOILER FEED PUMP SHOWING LEAKAGE IN THE DISCHARGE PASSAGES.

Fig. 72 shows a diagram from the water cylinder of a pump $4'' \times 2\frac{3}{4}'' \times 4''$. Boiler pressure 37. The compression curve as shown at the right hand end of the diagram leans inward, and shows that the pressure commenced to release before it exceeded the boiler pressure, and indicates a leak in the plunger, suction valves on the suction side of the plunger, discharge pipe or heater. The discharge pressure proper is two pounds greater than the boiler pressure, and is a moderate loss. The release pressure curve, as shown at the left hand end of the diagram, leans inward and indicates that the pressure was partly maintained on the suction side of the plunger for a considerable distance, and indicates a leak in the discharge valves on this end of the cylinder, or in the plunger, or both. The supply pressure was 3 pounds, and the

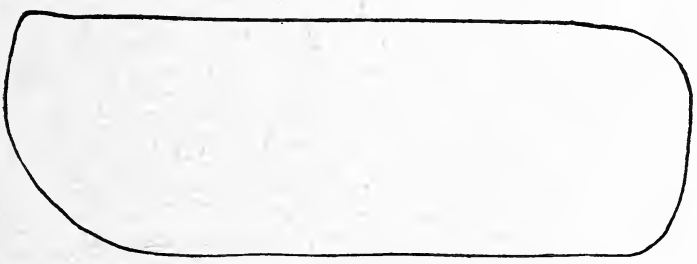

FIG. 72—SCALE OF SPRING 30.

distance between the atmospheric line and the suction line measures 2 pounds, and is also a moderate loss.

The speed of the pump was then decreased, and the discharge pressure fell to 18 pounds or 19 pounds less than the boiler pressure, in which case no water could pass into the boiler but passed through the leaks, and a search for the leakage was made at every point between the boiler and the water cylinder, as follows: The pump was run at a moderate speed and a valve on the discharge pipe next to the boiler was closed, and the pump should have stopped had there been no leakage. But the speed decreased only slightly ; another valve between the heater and next to the pump was closed, and the speed decreased fifty per cent., thus showing that there was about an equal amount of leakage in the heater and in the pump. Had the pump then stopped, all the leakage would have been in the heater, or had the speed been maintained, the leakage would have been in the pump.

Fig. 73 was taken from the same pump under a high speed, and it created a suction of 9½ pounds.

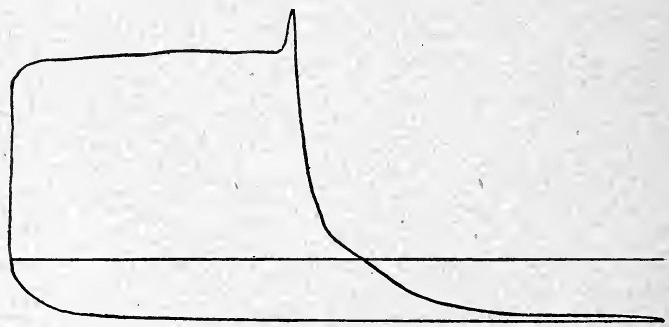

FIG. 73—SCALE OF SPRING 30.

When the plunger returns the compression curve erects at about one-half stroke, showing that the cylinder was only half full of water, and the speed was beyond the capacity of the suction passages. Had there been no leakage out of the cylinder, this compression curve should have been erected perpendicular at the point where it rises above the atmospheric line instead of forming a curve neither a concave nor a convex. Frequently pumps are replaced with larger ones, and have failed to produce any better results, from the fact that the difficulty was due to insufficient area in the suction passages. The peak above the diagram at the commencement of the discharge line is caused by a shock from the rapid rising in pressure.

DIAGRAM SHOWING ABSENCE OF ELASTIC CUSHION CAUSING IRREGULARITIES IN THE DISCHARGE LINE.

Fig. 74 is taken from the water cylinder of a pump where all passages were practically free from

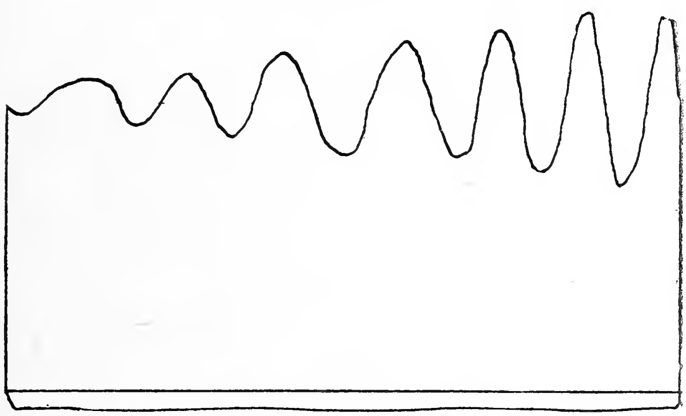

Fig. 74—Scale of Spring 40.

leakage, as is shown by the perpendicular compression and release pressure line. The air cushion chamber on the discharge was filled with water, and the absence of any elastic cushion, caused shocks and intermittent action in the discharge pressure, and formed the abrupt irregularities in the discharge pressure line. When the water was removed from the air chamber, the discharge line was straight. The absence of an air vent at the top of the air cushion chamber renders it practically impossible to drain the chamber of water, and when the chamber is filled it causes the pump to run noisy, about the same as when the cylinder is only partly filled with water.

PECULIAR COMPRESSION AND RELEASE PRESSURE LINES.

Fig. 75 shows a diagram from the steam cylinder of a $4'' \times 3'' \times 4''$ Duplex Boiler Feed Pump. The upper diagram was taken when the pump was running at a moderate speed, sufficient to supply the boiler. The release curve as shown at the right hand end of the diagram, curves inward and indicates that the exhaust port was not sufficiently open to properly release the pressure before the piston commenced to return. The back pressure in the beginning of the back pressure line is 12 pounds and gradually falls to about 6.5 pounds at about three-quarter stroke, when it gradually rises until the exhaust valve closes. The excessive back pressure is caused by insufficient area in the exhaust passages, either in the ports or in the exhaust pipe, a matter which can readily be determined by placing a gauge on the exhaust pipe or by taking an exhaust pipe diagram with the indicator. Should the back pressure in the exhaust pipe be moderate, the restricted area would be in the ports, and should the back pressure in the pipe be the same as the indicated back pressure on the diagram, the restricted area would be in the exhaust pipe.

Back pressure is a loss of power and economy because it requires an additional amount of steam on the admission side of the piston to overcome it and should the back pressure equal the admission pressure, the pump would stop.

The compression curve as shown at the left hand end of the diagram is formed by the live steam being admitted into the cylinder before the piston completes its stroke as the exhaust lap is only sufficient to prevent the entering steam from passing into the exhaust, which is the general method of cushioning a steam pump.

The speed of the pump was then decreased and the pencil again applied when the admission pressure fell to only 5 pounds as shown in the diagram between the upper diagram and atmospheric line. The boiler pressure was 75 pounds which multiplied by the area

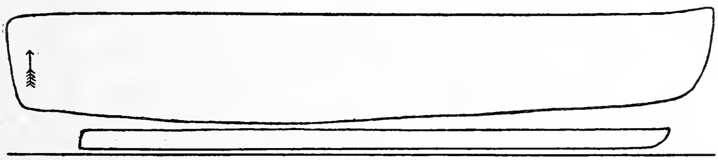

FIG. 75—SCALE OF SPRING 40.

of the plunger 7.06 inches, gives a total of 529.5 pounds resistance. And since the area of the steam piston is 12.56 inches, multiplied by the admission pressure 5 pounds, gives 62.8 pounds exerting pressure, thus showing that it was impossible to force the water into the boiler. Upon searching, it was found that the water passed through leaks in the heater. This diagram is considerably shorter and is caused by the velocity of the parts being less and is stopped at an earlier point in the stroke and this reduction in the stroke increases the percentage of clearance an amount proportionate to the difference in the length of the two diagrams.

Owing to the short stroke of the pump, no reducing motion was necessary in taking the diagrams.

Fig. 76 shows a diagram taken from the water end of the same pump and will bear considerable study. The compression line is shown at the right hand end of the diagram as curving outward and shows that the pressure rises to about one-half the discharge pressure on the suction side of the plunger while the plunger is still moving away and is caused by leakage of water

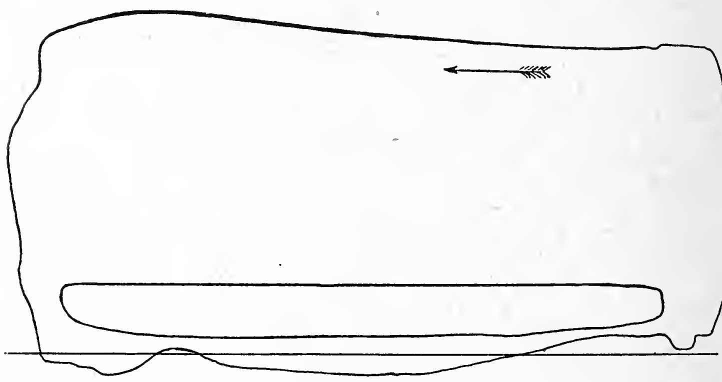

FIG. 76—SCALE OF SPRING 40.

into the cylinder through the discharge valve on the suction side of the plunger. When the plunger commences to return the line is then completed by curving inward and is caused by leakage out of the cylinder through the suction valves on the discharge side of the plunger, or the plunger or both. The discharge line should be maintained horizontal to the end of the stroke but it will be seen that the pressure falls before the end of the stroke as shown by the release pressure line curving outward and is due to the fact, that before the plunger stops, it is moving slowly and the pressure leaks out of the cylinder and the water escapes through the leaks at a lower pressure than that required to force it into the boiler. When the return stroke commences, the completion of this line should fall perpendicular but instead, it completes by curving inward and is caused by leakage into the cylinder on this suction side of the plunger and the pressure is not fully released until the plunger has moved a short distance on its return stroke.

The water was supplied by pressure but the insufficient area in the supply passages even caused it to fall below the atmospheric line and created a vacuum in the cylinder, and together with the speed not being uniform throughout the stroke it formed irregular suction line at different points in the stroke as shown.

When the speed was reduced the pencil was again applied and the smaller diagram was obtained. The discharge pressure is only 14.5 pounds and the boiler pressure was 75 pounds, thus showing that the water could not pass into the boiler and the leakage was found to be in the heater. The decreased speed of the pump reduced the length of the stroke and formed the shorter diagram as shown.

CHAPTER XI.

CALCULATING THE MEAN EFFECTIVE PRESSURE BY ORDINATES.

The mean effective pressure (M. E. P.) is the average pressure per square inch on the piston during its entire stroke, less the back pressure on its return stroke.

To calculate the M. E. P. by the ordinate process, the height in pounds of each ordinate is measured on the scale of the spring from the back pressure line and compression curve to the steam line and expansion curve, whether condensing or noncondensing. The ordinates must be equal distances apart and perpendicular to the atmospheric line. The first ordinate is located at the point of exhaust closure as shown in Fig. 77, and no other ordinate between it and the admission line is necessary, except in diagrams taken from high speed engines where the exhaust valve closes at an early point in the stroke. This ordinate measures 72 pounds, and is the pressure exerted on the piston during this part of the stroke, providing there was no compression, and that the steam line was parallel to the atmospheric line. But a deduction must be made for the compression, and to do this, another measurement is made at such a point between this ordinate and the admission line, as to represent the average pressure during this range of the piston movement, and in this case is located at such a point as shown by the dotted ordinate which measures 70 pounds ; therefore, the first ordinate proper should read 70 pounds instead of 72 pounds. The remaining portion of the diagram is divided with ordinates whose distances apart are the same as the distance between the admission line and the first ordinate proper and measures respectively, 72, 41, 30, 23, 18, 13, 9.5, 6.5, 4 pounds. The greater the number of ordinates employed, the nearer correct will be the

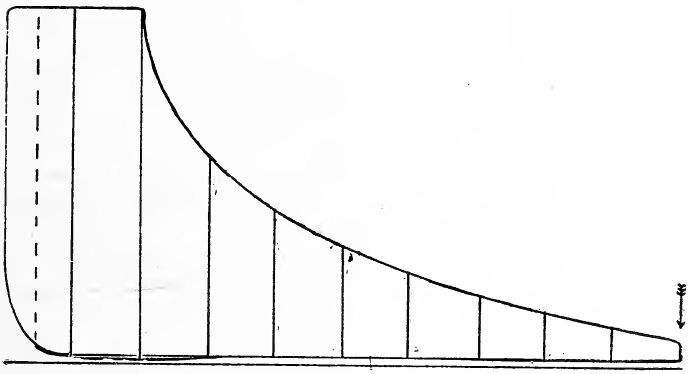

FIG. 77—SCALE OF SPRING. 40.

calculation ; however, about ten (the same as employed in this diagram) will be sufficient. Their total sum is 287 pounds. The total sum of the ordinates on the diagram from the other end of the cylinder is determined in like manner. Supposing this is 288 pounds. The two sums combined and divided by 20, which is the number of ordinates contained in both diagrams, gives the M. E. P. Then 287+288÷20= 28.75 pounds M. E. P. and is the factor used in computing the indicated horse power (I. H. P.) Had the number of ordinates been 30, the sum of the two totals should have been divided by 30, and so on.

It is of importance that the ordinates be equal distances apart, and unless they extend to the end of the diagram, as shown by the arrow, which is the tenth ordinate, the fraction must be known. This however, can readily be computed on the scale of any spring. The distance between the ordinates proper, measures 14 pounds on a 40 scale; and supposing the distance between the ninth ordinate and the end of the diagram measures 6 pounds, the fraction is $\frac{6}{14}$ths, and is reduced to decimal by dividing 6 by 14. Thus $6 \div 14 = .42$; therefore, the number of ordinates in the diagram is 9.42; and supposing the ordinates on the diagram from the other end of the cylinder also, is 9.42. Then $9.42 + 9.42 = 18.84$, and is the number of ordinates contained in both diagrams. Supposing the average height in pounds of the ordinates contained in both diagrams is 575. Then $575 \div 18.84 = 30.52$ pounds M. E. P. Should the expansion curve merge into the back pressure line before the end of the diagram, and follow the back pressure line to its end, nevertheless the total sum in pounds of the ordinates must be divided by the number of the ordinates which must extend to the entire end of the diagram, or the same as if the expansion curve was maintained above the back pressure line.

Where the load is constantly changing, the pencil should be applied during as many revolutions as is necessary, in order to obtain both the lightest and heaviest load, as shown in Fig. 78, from which the M. E. P. can be calculated, and the ordinates are located in the usual manner. The first ordinate proper, when corrected for the compression curve measures 68 pounds, and the second ordinate has two measurements, viz: the earliest and the latest points of cut-off. The first measures 56 pounds and the second 73 pounds. The remaining ordinates are measured in like manner, which are respectively, 35–44, 24–31, 16.5–22, 11–16, 7–11, 4–8, 2–5.5, 1–4, giving a total sum of 439 pounds. The sum of the ordinates on

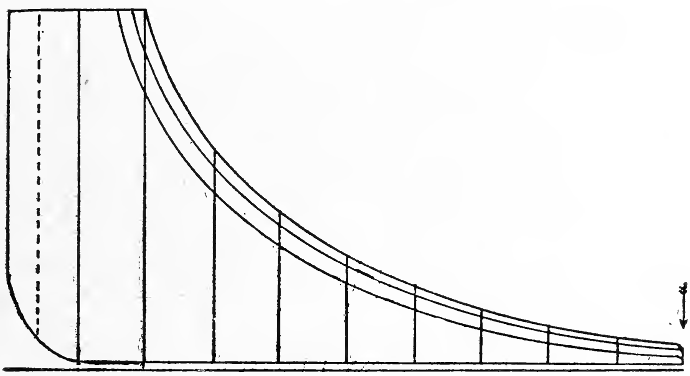

FIG. 78—SCALE OF SPRING 40.

the diagram from the other end of the cylinder is calculated in like manner. Supposing this is 221 pounds; then 439+221÷38=17.36 pounds M. E. P. It will be seen that the total sum of the ordinates from the two diagrams must be divided by the number of measurements contained in both diagrams, which is 38, as 18 of them have two measurements. The average height of the nine ordinates may also be found by making single measurements half-way between the lowest and highest expansion curves.

When a loop is formed in the diagram as shown in Fig. 79, the total sum of the ordinates contained in the loop of the diagram from each end of the cylinder must be deducted from the total sum of the ordinates contained in the exerting portion of the diagrams and the result divided by 20, which is the number of ordinates contained in both diagrams, and this gives the M. E. P. The ordinates in the loop of this diagram measures respectively, 3, 5, 7, 4 pounds,

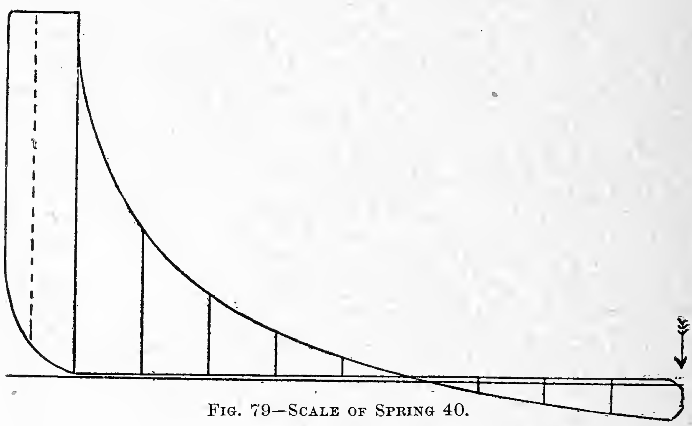

FIG. 79—SCALE OF SPRING 40.

or a total of 19 pounds; and supposing the total sum of the ordinates in the loop on the diagram from the other end of the cylinder measures 11 pounds. Then $19+11=30$ pounds. The height of the ordinates in the exerting portion of this diagram measures respectively 69, 30, 16.5, 9, 3.5 pounds, or a total of 128 pounds. And supposing the total sum of the ordinates in the exerting portion of the diagram from the other end of the cylinder is 122 pounds, plus 128, gives 250 pounds; then $250-30\div20=11$ pounds M.

E. P. The expansion curve crosses the back pressure line at the sixth ordinate, and there is no pressure above or below the back pressure line at this point, but nevertheless this point counts as an ordinate.

The M. E. P. can also be readily determined by making a continuous measurement on a strip of paper near its edge (which must be straight), and carried from one ordinate to another, the measurement of each being marked beyond the preceding one, so that the distance from the end of the strip to the last mark, represents the height of the ordinates combined. The distance as shown in Fig. 77, is 7.01 inches; and supposing the distance on the diagram from the other end of the cylinder is 7.25 inches, then $7.01+7.25=14.26$, and since the scale of the spring is 40; then $\frac{14.26\times 40}{20}=28.52$ pounds M. E. P.

CHAPTER XII.

THE PLANIMETER.

The planimeter is an instrument for measuring the square inches of area of an indicator diagram, or any figure that has a plain surface. This instrument has become almost indespensible to the engineer, and is the only accurate method of determining the M. E. P. of an indicator diagram. Its operation is very simple, and it can be readily manipulated by any person of ordinary intelligence. Fig. 80 shows the planimeter properly applied to an indicator diagram.

To calculate the M. E. P. with the planimeter: First, the diagram is made fast on the drawing-board by a thumb tack at each end, and placed as shown in Fig. 80. The surface on which the roller is traveling must be perfectly smooth and unglazed. A piece of dull cardboard serves the purpose satisfactorily, and should be placed underneath the diagram and extend sufficiently above it to give ample traveling surface for the roller. The tracing point should be pressed sufficiently into the paper so as to form an impression as a starting and stopping point. This point must be on the back pressure line, and at about the middle of the diagram. The stationary point as shown at the left is pressed fast into the board, on about an equal plane with the back pressure line and about the same distance from the tracing point as the length of the diagram. If the card on which the diagram is taken, contains spaces above the diagram for recording various observations, and they extend such a distance above the diagram as to interfere with the roller, the diagram should be placed upside down.

With the planimeter in proper position as shown,

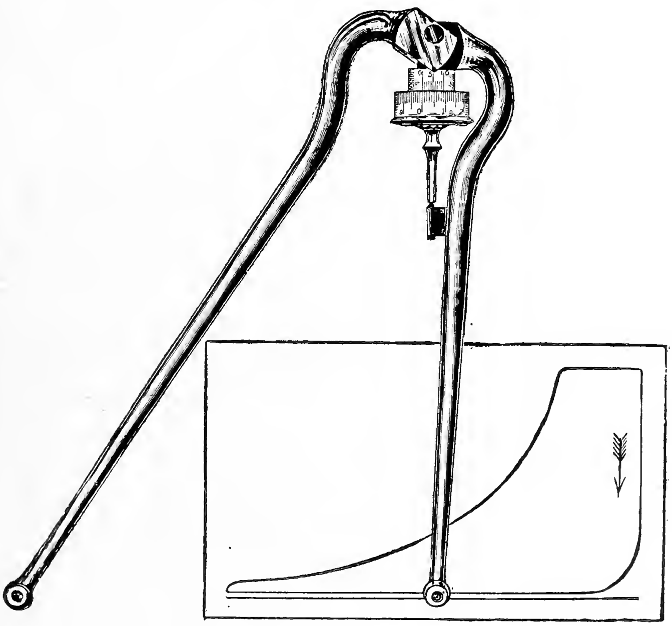

FIG. 80.

adjust the roller so that its zero coincides with the zero of the vernier, as shown in Fig. 80. Then circumscribe the diagram in the direction indicated by the arrow, or that of the hands of a watch, and with such precision that the tracing point follows the precise path of the pencil until it returns to the starting point. Supposing the reading to be the same as shown in Fig. 81. The roller is divided into ten numbered parts, and each number represents one square inch. Had figure 2 on the roller exactly coincided with the zero on the vernier, the result would have been 2 square inches without any fraction. The spaces between the numbers are divided into ten graduations, each of which represents 1-10 of a square inch; and had the first graduation coincided with the zero on

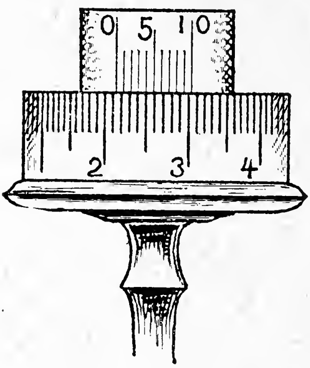

Fig. 81.

the vernier, the result would have been 2.1. The vernier also has ten graduations, and if one of them coincides with a graduation on the roller, it represents —counted from zero—so many hundredths of a square inch. In this case, the fourth graduation of the vernier coincides with one of the graduations of the roller, and the final reading is 2.14 square inches.

Referring again to the reading as indicated by the planimeter, it is plain to be seen that Fig. 2 on the roller, has passed the zero on the vernier, therefore, figure 2 may be first marked down. It also shows that there is one graduation between the figure 2 on the roller and zero on the vernier, and therefore figure 1 is next marked down, and a decimal point must be placed between it and figure 2. It also shows that the fourth graduation on the vernier coincides with one of the graduations on the roller and is the last figure marked down, and the reading is as before stated, 2.14 square inches. It can readily be seen from the above explanation that the method of reading the planimeter is a simple undertaking. Supposing the scale of the spring is 40, and the length of the diagram (which is measured on the atmospheric line) is 4 inches, and the area of the diagram from the other end of the cylinder is 2.26 inches; the M. E. P. is then found by multiplying the combined area in square inches contained in the diagram from each end of the cylinder (4.40) by the scale of the spring (40) which gives 176.00, and instead of dividing 176.00 first by 4, that being the length of the diagram in inches, and the quotient by 2, so as to obtain the average pressure on the piston from both ends of the cylinder; it is necessary only to divide 176.00 by two times four:

Example, $\frac{2.26+2.14\times 40}{8}=22$ pounds, M. E. P.

To determine the length of the diagram, a scale graduated in one-hundredths of an inch should be used, otherwise it frequently requires figuring on the scale of any other spring to learn its length. Supposing the length of the diagram is 3 inches and 30 pounds on a 40 scale. The pounds is reduced to inches by dividing it by the scale of the spring: Example, $30 \div 40 = .75$ of an inch, and the length of the diagram is then 3.75 inches. Supposing the combined area of the diagram from each end of the cylinder is 4 inches, multiplied by 40 gives 160 without any fraction, and since there is a fraction of two figures to the right of the decimal in the divisor (3.75), therefore, two ciphers should be added to the dividend, when it will read 160.00, and so on, or in other words, unless the fraction either in the divisor or dividend contain the same number of figures to the right of the decimal, ciphers should be added accordingly, the example then becomes:

$$\frac{4 \times 40}{3.75 \times 2} = 21.33 \text{ pounds, M. E. P.}$$

The planimeter is a delicate instrument and should be carefully handled. The roller should be taken out and the bearings cleaned and oiled with porpoise jaw oil, the same as used on the pencil motion of the indicator. After replacing the roller, the roller-shaft should be adjusted so that there is no end play and that the roller revolves with perfect freedom, a matter which should be occasionally tested before using the planimeter.

Where a diagram is taken from each end of the cylinder on the same card, as shown in Fig. 82. It is unnecessary to make two separate readings. After circumscribing the diagram from one end of the cylinder, continue to circumscribe the diagram from the other end of the cylinder in the direction as indicated by the arrows, and when the tracing point has again reached the starting point, the result is a double reading, which in this case is 5.38 square inches. The length of the diagram is 3.51 inches, then

$$\frac{5.38 \times 40}{3.51 \times 2} = 30.65 \text{ pounds, M. E. P.}$$

Where the load is constantly changing as shown in Fig. 78, the diagram must be circumscribed twice. Commencing with the tracing point in the usual place and circumscribe the diagrams at the earliest and latest points of cut-off, thus giving a double reading.

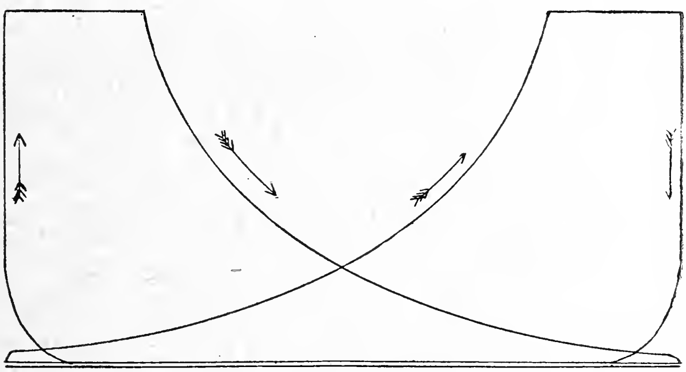

FIG. 82—SCALE OF SPRING 40.

Supposing this is 4.84 square inches, and another double reading of the diagram from the other end of the cylinder is 4.78 square inches: Scale of the spring is 40 and the length of the diagram 3.5 inches, then $\frac{4.84 + 4.78 \times 40}{3.5 \times 4} = 27.48$ pounds, M. E. P.

The reason for multiplying 3.5 by 4 is, that there are two double readings, and the easiest cut-off represents the lightest load and the latest cut-off the heaviest load.

When diagrams contain loops as shown in Fig. 83, the loops must be circumscribed in the opposite direction, as indicated by the arrows, and in this way the planimeter automatically deducts the area contained in the loops from the exerting portion of the diagrams.

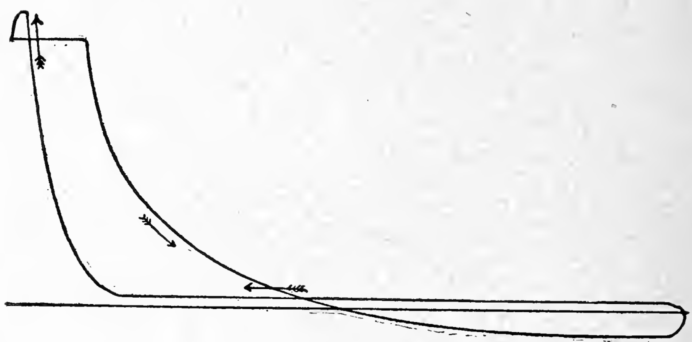

FIG. 83.

The area of the diagram from the other end of the cylinder is determined in like manner and their lengths are measured on the atmospheric line, and the process of calculating the M. E. P. is made in the usual manner.

CHAPTER XII.

TO COMPUTE THE INDICATED HORSE-POWER OF AN ENGINE.

The commonly accepted valuation of a horse-power is 33,000 pounds raised one foot high in one minute, or its equivalent.

The rule for finding the horse-power of an engine is: Multiply the area of the piston, less one-half the area of the piston-rod, by the mean effective pressure and this by the speed of the piston in feet per minute. This product divided by 33,000 will give the indicated horse-power (I. H. P.) of the engine.

Supposing it be desired to compute the I. H. P. of an engine having the following data: Diameter of piston, 20″. Diameter of piston rod, 3″. Stroke, 42″. Revolutions, 70. M. E. P., 30.5 pounds.

The area of the piston is found by multiplying the square of its diameter by .7854.

Example, $20'' \times 20'' \times .7854 = 314.16''$.

One-half the area of the piston-rod is found by multiplying the square of its diameter by .7854 and dividing by 2. Example, $3'' \times 3'' \times .7854 \div 2 = 3.53''$. Then $314.16'' - 3.53'' = 310.63''$ net area of piston. There being a piston-rod on but one side of the piston makes it necessary to deduct only one-half the area of the piston-rod.

The speed of the piston in feet per minute is found by multiplying twice the stroke in inches and this divided by 12″ (there being 12 inches to the foot) gives the feet per revolution, and this multiplied by the number of revolutions gives the feet per minute. Example, $42'' \times 2 \div 12'' \times 70 = 490$ feet. The I. H. P. is then found in the following manner:

$$\frac{310.63'' \times 30.5 \times 490'}{33000} = 140.67 \text{ I. H. P.}$$

The horse power constant is the number of I. H. P. for each pound of M. E. P., and when the piston speed in feet per minute remains constant, this is the most rapid process of computing the I. H. P. as when the horse-power constant is once found it is only necessary to ascertain the M. E. P. and this multiplied by the horse-power constant gives the I. H. P., viz., net area piston 310.63″. Piston speed in feet per minute 490. Then $\frac{310.63'' \times 490'}{33000} = 4.612$ H. P. C.

To ascertain the I. H. P. where the load is constantly changing, diagrams should be taken at about ten minute intervals for a duration of one or more hours according to the importance of the test, etc., and the pencil should be applied as many revolutions as necessary in order to obtain the lightest and heaviest load.

The length of the diagrams is found by measuring the length of one of them. But should their length vary, in which case they are distorted and in order to obtain the average length, their combined length divided by the number of diagrams gives the average length, and supposing this is 4.25″. Supposing the scale of the spring is 40, and the number of diagrams is 24, and that the point of cut-off varies so that each diagram has a double reading which is equal to 48 diagrams, and their combined area which supposing is 96.45 multiplied by the scale of the spring and the product divided by 48 times the average length of the diagrams, gives the M. E. P.

Example, $\frac{96.45 \times 40}{48 \times 4.25} = 18.91$ pounds M. E. P.

The I. H. P. for the entire duration of the test is then found in the usual manner.

To ascertain the I. H. P. of one or more machines it is only necessary to take a pair of diagrams with the machine in operation, and supposing the area of the diagrams is 4.38″. Another pair of diagrams is taken while the machine is not in operation, whose area is 3.14″. Then 4.38″ − 3.14″ = 1.24″ net area and is the factor used in computing the I. H. P. of the machine.

To determine whether the load is equally distributed on each side of the piston, it is only necessary to find the area of the diagram from each side.

To compute the I. H. P. from one end of the cylinder the M. E. P. of the diagram from the head end of the cylinder multiplied by the area of the piston, in which case no deduction is made from the head end when there is no piston rod. (But the whole area is deducted from the crank end in computing the I. H. P. from the crank end.) This multiplied by one-half the piston speed in feet per minute, and divided by 33,000 gives the I. H. P. from one end of the cylinder.

To compute the I. H. P. of a two cylinder compound engine the M. E. P. of the high pressure cylinder divided by the ratio of the piston areas plus the M. E. P. of the low pressure cylinder multiplied by the horse power constant of the low pressure cylinder gives the I. H. P. of the engine. The ratio of the cylinder areas is found by dividing the net area of the low pressure piston by the net area of the high pressure piston, in which case when the cylinders are placed tandem the low pressure piston has two piston rods, and their areas combined and divided by 2 gives the average area which must be deducted from the low pressure piston. The net area of the high pressure piston is found in the same manner as in a single cylinder engine and the horse power constant of the low pressure piston is also found in the usual manner. Supposing it be desired to find the I. H. P. of an engine having the following data: Net area high pressure cylinder 100″. Net area low pressure cylinder 300″. Then $300'' \div 100'' = 3$. Ratio: M. E. P. high pressure cylinder 40; M. E. P. low pressure cylinder 13; piston speed in feet per minute 500; horse power constant low pressure cylinder 5.

Example, $40 \div 3 + 13 \times 5 = 131.66$ I. H. P.

The I. H. P. may also be determined by computing the I. H. P. of each cylinder separately, the same as in a single cylinder engine, and their I. H. P. combined gives the I. H. P. of the engine.

To ascertain the distribution of power as between the two cylinders instead of computing the I. H. P. of each cylinder it is only necessary to divide the M. E. P. of the high pressure cylinder by the ratio and should the quotient be the same as the M. E. P. in the low pressure cylinder the load is equally distributed.

CHAPTER XIII.

TESTING THE PISTON AND VALVES FOR LEAKS.

The piston and valves of an engine are never absolutely tight. To test the steam valves of a Corliss engine, unhook both steam valves and with the dash-pots properly seated so that the valves cover the ports in the usual manner, and with the tapped holes for the indicator open, admit the full pressure into the steam chest, and if steam escapes through the tapped holes the steam-valves leak.

To test the piston and exhaust valves, place the engine on the center, supposing the head end, disconnect the crank end exhaust valve and place the valve so as to properly cover the port. Place the wrist-plate toward the crank end and with the head end steam valve hooked in admit the full pressure into the head end clearance space. If steam escapes through the exhaust pipe, the exhaust valve leaks, and if a greater quantity of steam escapes through the crank end tapped hole than the leakage of the crank end steam-valve, the piston leaks. The other exhaust valve is tested in like manner.

To test the valve of a single valve engine, revolve the engine and with a tram at each position make a mark on the valve-stem; then equally divide the distance and place the engine so that the tram spans the middle mark; when the valve equally covers the ports admit the full pressure into the steam-chest, and if steam escapes through the tapped holes the valve leaks.

To test the piston, place the engine on the center and if the valve has lead admit the full pressure into the clearance space, and if a greater quantity of steam escapes through the tapped hole on the other side of the piston than the leakage of the valve, the piston leaks.

This is the only positive test for determining the leakage of the piston and valves.

TO ASCERTAIN THE PERCENTAGE OF CLEARANCE IN AN ENGINE.

The clearance is the space embraced in the ports and between the piston and cylinder head when the engine is on the center. The piston and valves must be made absolutely tight, otherwise the result is misleading.

First, place the engine one inch travel on the cross-head, remove the cylinder-head, then pack the piston tight with asbestos, or other suitable substance. After replacing the cylinder-head make the valves tight by placing the sheet packing between the valve and seats, in which case the valve must be held tight to the seats; but, if a piston valve, remove it and place sheet packing over the port, which can be held in place by a segment made of wood, whose radius is less than the thickness of the packing. If the cylinder is tapped on the side for the indicator, extend the filling pipe above the cylinder and fill this space with water. Supposing its weight is 55 pounds, then place the engine on the center; and supposing that 30 pounds of this water is discharged, then 30 pounds is the weight of the water contained in the cylinder for each inch of the piston movement. Supposing that the weight of the water in the filling pipe is one pound, then the remaining 24 pounds is the weight contained in the clearance space. Supposing the stroke of the engine is 40″, then

$$\frac{24}{30 \times 40} = .02 \text{ (2 per cent.) clearance}$$

It will be seen from the above explanation that this is a remarkably simple, quick and accurate method of ascertaining the percentage of clearance, and that the displacement of the piston-rod is automatically deducted from the crank end of the cylinder.

METHOD OF LOCATING THE CLEARANCE LINE AND DETERMINING LEAKAGE ON THE DIAGRAM.

The clearance contained in the head end of the cylinder must be added to the head end diagram, and the clearance of the crank end to the crank end diagram. In Fig. 84 the clearance line is shown at the left of the diagram. Supposing the percentage of clearance is .025 ($2\frac{1}{2}$ per cent.) the length of the diagram 4″, then $.025 \times 4'' = .1''$, and is the distance the clearance line should be located from the commencement of the atmospheric line and at right angles. This can be reduced to pounds by multiplying it by the scale of any spring, thus, $40 \times .1'' = 4$ pounds, and this distance must be located with the same scale, and so on.

To determine from the diagram the leakage of the valves and piston seems to be a popular craze with some engineers. While demonstrations are mathematically correct, yet there are other factors to be considered which are not shown in the diagram, and to base any facts upon a false expansion curve is a misleading and costly undertaking. Chief among these factors are an incorrect indicator, leakage of the piston and exhaust valve, which may be balanced with re-evaporation; leakage of the steam-valve,

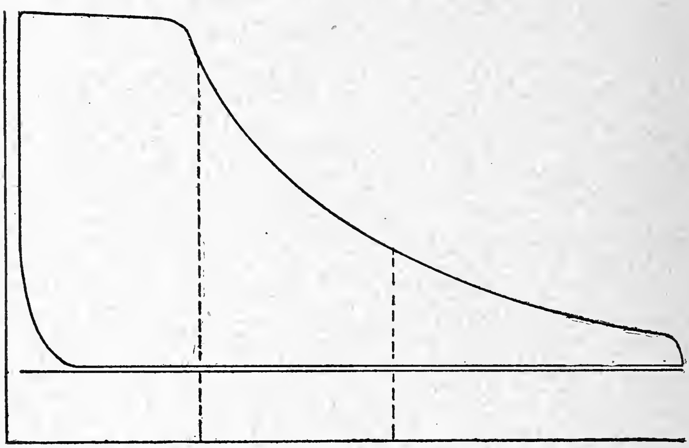

Fig. 84—Scale of Spring 40.

which may be balanced with condensation, and the leakage into and out of the cylinder may balance each other, when the engine will produce an expansion curve representing a perfect expansion of steam.

The demonstration as shown in Fig. 84 is remarkably simple, and can be readily applied to any diagram. Locate the vacuum line on the scale of the spring at such a distance below the atmospheric line as to represent the atmospheric pressure at the time and place; also the clearance line representing the percentage of clearance on this end of the cylinder. Should the clearance not be known, and for such a purpose an approximation may be made, as a reason able variation in the location in the clearance line varies but slightly in the expansion curve. Then locate a perpendicular ordinate from a point in the expansion curve, as shown by the first dotted ordinate, in which case the steam valve has fully closed, as shown by the expansion curve in changing from a convex to a concave curve, and the pressure at this point is 80 pounds absolute. Then locate another ordinate at a point double this distance from the clearance line, as shown by the second dotted ordinate, in which case the volume is doubled, and the pressure should be one-half of that at the first dotted ordinate, or 40 pounds absolute, providing that steam was a perfect gas. If the pressure is greater than 40 pounds the steam-valve has leaked and raised the expansion curve, and if less, the piston or exhaust-valve, or both, have leaked and lowered it.

CHAPTER XIV.

LOCATING THE THEORETICAL CURVE.

The theoretical curve represents a figure in which the volume is inversely as the pressure. Thus, if one cubic foot of perfect gas at 100 pounds absolute pressure be expanded into two cubic feet, the volume is

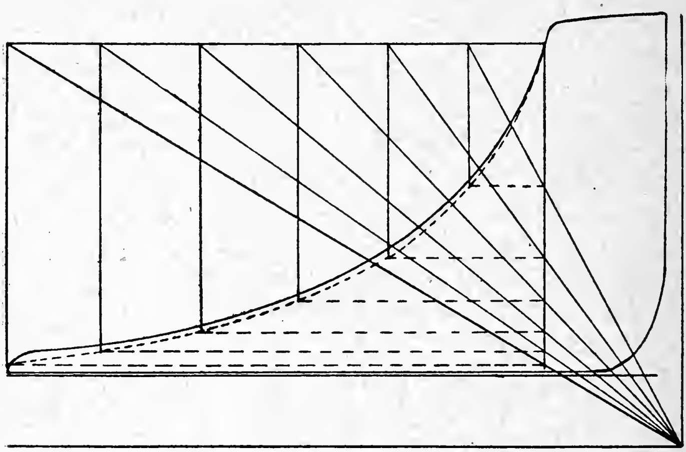

Fig. 85.

doubled, and the pressure is one-half or 50 pounds absolute, and so on. When this curve is applied to an indicator diagram it is of no more value than Fig. 84; except that it shows the corresponding pressure at any point in the expansion curve. Fig. 85 shows the method of applying it to an indicator diagram. With the clearance and vacuum lines added, locate the perpendicular line at a point beyond the cut-off, as explained in Fig. 84; and from this point where it intersects with the expansion curve, locate the upper line at right angles, as shown. Then from this line locate the ordinates perpendicular, and about the distance apart, as shown and from each of these intersections, locate the diagonal lines to the intersection of the clearance and vacuum lines. Then from the point where the first diagonal line crosses the cut-off line, draw the dotted horizontal line intersecting with the first ordinate at a point shown by the dotted curve, and so on. If the expansion curve of the indicator diagram rises above the theoretical curve it indicates leakage of steam into the cylinder, and should it fall below the theoretical curve, it indicates leakage out of the cylinder.

LOCATING THE THEORETICAL POINT OF CUT-OFF.

To locate the theoretical point of cut-off, as shown in Fig. 86; the clearance and vacuum lines must be added in the usual manner. The initial pressure line as shown above the diagram, should be located such a distance above the steam line as to allow for the friction of the indicator, because owing to the friction, the indicator gives less pressure while rising, and more while falling.

Supposing it be desired to determine the theoretical point of cut-off for the purpose of ascertaining the I. H. P. for a given point of cut-off, in which case all the leakage into and out of the cylinder must be calculated from a point just before the exhaust valve opens, which gives the point of cut-off corresponding to this pressure. Therefore, the base line is located perpendicular at a point as shown, then locate the diagonal line between the two intersections as shown, and next the horizontal line at right angles from the intersection of the base line and the expansion curve, which represents the distance of the theoretical point of cut-off, and this distance is transferred on the initial pressure line. From this point the cut-off line is located

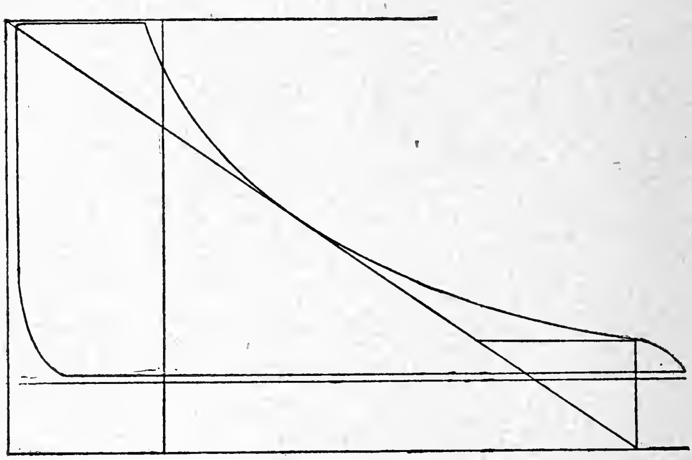

FIG. 86.

perpendicular, and represents the theoretical point of cut-off corresponding to the pressure of the base line.

COMPUTING THE WATER CONSUMPTION PER INDICATED HORSE POWER PER HOUR FROM THE DIAGRAMS.

To compute the water consumption of an engine from the diagram is a worthless and misleading method, from the fact that condensation can not be accounted for, and the leakage into and out of the cylinder, may balance each other, etc. Supposing it be desired to compute the water consumption of Fig. 87, having the following data: Diameter cylinder 20″, stroke, 42″, diameter piston-rod, 3″, length piston-rod, 42.5″, revolution, 70, clearance head end, 3 per cent., clearance crank end, 2 per cent.

One-half the piston-rod displacement in cubic inches, is found by multiplying the square of its diameter

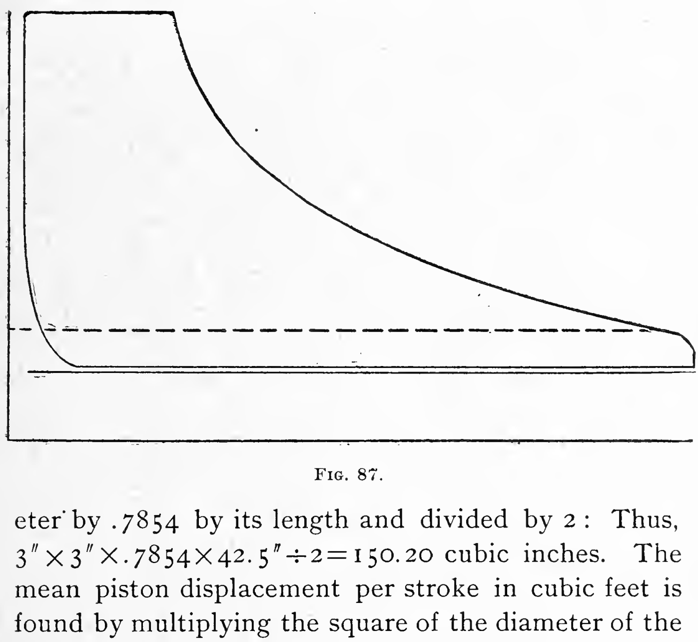

Fig. 87.

by .7854 by its length and divided by 2: Thus, $3'' \times 3'' \times .7854 \times 42.5'' \div 2 = 150.20$ cubic inches. The mean piston displacement per stroke in cubic feet is found by multiplying the square of the diameter of the cylinder in inches by .7854 by the stroke, less one-half the piston-rod displacement, and the product divided by 1728, that being the cubic inches in a cubic foot. Thus: $20'' \times 20'' \times .7854 \times 42'' - 150.20 \div 1728 = 7.548$ cubic feet of mean piston displacement per stroke.

The mean clearance is two and one-half per cent. (.025) and the length of the diagram is 3.5″. Then $.025 \times 3.5'' = .08''$, and is the distance the clearance line should be located from the commencement of the diagram, the vacuum line being located the usual distance. Next locate a line from the clearance line parallel with the atmospheric line intersecting with the expansion curve, at a point just before the exhaust valve commences to open (as shown by the dotted line), so that all the transactions taking place in the expansion curve may be accounted for in the computation. The length of the dotted line is 3.40″; and supposing the length of this line on the diagram from the other end of the cylinder is 3.20″, then the mean is 3.30″. The length of the diagram plus the clearance, is 3.58″. The absolute pressure at the location of the dotted line is 23 pounds, and supposing the absolute pressure of the diagram from the other end of the cylinder is 27 pounds, then the mean absolute pressure is 25 pounds, and weighs according to the tables on properties of saturated steam .0634 pounds per cubic foot, and since the piston displacement per stroke in cubic feet is 7.548, then $3.30'' \times 7.548 \times .0634 \div 3.58'' = .4411$ pounds per stroke, providing there was no compression. But a deduction for the compressed exhaust steam must be made, and since the mean piston displacement in cubic feet is 7.548, and the mean clearance is two and one-half per cent., then $7.548 \times .025 = .1887$ cubic feet of mean clearance per stroke. The compression is 45 pounds absolute; and supposing the compression of the diagram from the other end of the cylinder is 35 pounds, the mean is

40 pounds, whose weight per cubic foot is .0994; then $.1887 \times .0994 = .0187$ pounds per stroke.

Since the revolutions per minute is 70 or 140 strokes, the strokes per hour is 8400; and supposing the I. H. P. is 140, then $.4411 - .0187 \times 8400 \div 140 =$ 25.34 pounds water consumption per I. H. P. per hour.

CHAPTER XV.

Table No. 1.

TEMPERATURE OF VACUUM.

Pounds per square in.	Temperature.	Inches of Mercury.	Pounds per square in.	Temperature.	Inches of Mercury.
1.127	208°	2.3028	13.182	116°	26.8608
2.186	204	4.4538	13.345	112	27.1908
3.172	200	6.4638	13.493	108	27.4908
4.095	196	8.3428	13.624	104	27.7588
4.954	192	10.0948	13.743	100	28.0038
5.755	188	11.7258	13.852	96	28.2238
6.497	184	13.2418	13.948	92	28.4218
7.190	180	14.6508	14.035	88	28.5988
7.833	176	15.9588	14.114	84	28.7588
8.427	172	17.1718	14.183	80	28.8988
8.978	168	18.2948	14.245	76	29.0249
9.489	164	19.3338	14.300	72	29.1374
9.959	160	20.2928	14.350	68	29.2372
10.393	156	21.1768	14.393	64	29.3256
10.792	152	21.9928	14.431	60	29.4038
11.162	148	22.7428	14.465	56	29.4726
11.499	144	23.4318	14.495	52	29.5335
11.811	140	24.0638	14.521	48	29.5867
12.093	136	24.6418	14.543	44	29.6334
12.352	132	25.1718	14.563	40	29.6742
12.591	128	25.6548	14.581	36	29.7097
12.807	124	26.0958	14.596	32	29.7407
13.004	120	26.4968			

TABLE 2.

PROPERTIES OF SATURATED STEAM.

Absolute pressure in pounds per square inch.	Temperature in Fahrenheit degrees.	Total heat-units per pound above zero (32°).	Heat-units per pound contained in the water.	Weight in decimals of a pound per cubic foot.	Specific volume.
1	102	1145	102.1	.0030	20620
2	126.3	1152.5	126.4	.0058	10720
3	141.6	1157.1	141.9	.0085	7326
4	153.1	1160.6	153.4	.0112	5600
5	162.3	1163.4	162.7	.0137	4535
6	170.1	1165.8	170.6	.0163	3814
7	176.9	1167.9	177.4	.0189	3300
8	182.9	1169.7	183.5	.0214	2910
9	188.3	1171.4	188.9	0239	2607
10	193.2	1172.9	193.9	.0264	2360
11	197.8	1174.2	198.5	.0289	2157
12	202	1175.5	202.7	.0313	1988
13	205.9	1176.7	206.7	.0337	1846
14	209.6	1177.9	210.4	.0362	1722
14.7	212	1178.6	212.9	.0380	1644
15	213.1	1178.9	213.9	.0387	1612
16	216.3	1179.9	217.2	.0413	1514
17	219.4	1180.9	220.4	.0437	1427
18	222.4	1181.8	223.4	.0462	1351
19	225.2	1182.6	226.3	.0487	1282.1
20	227.9	1183.5	229	.0511	1220.3
21	230.5	1184.2	231.7	.0536	1164.4
22	233	1185	234.2	.0561	1113.5
23	235.4	1185.7	236.7	.0585	1066.9
24	237.7	1186.5	239	.0610	1024.1
25	240	1187.1	241.3	.0634	984.8

TABLE 2—*Continued.*

Absolute pressure in pounds per square inch.	Temperature in Fahrenheit degrees.	Total heat-units per pound above zero (32°).	Heat-units per pound contained in the water.	Weight in decimals of a pound per cubic foot.	Specific volume.
26	242.2	1187.8	243.5	.0658	948.4
27	244.3	1188.5	245.7	.0683	914.6
28	246.3	1189	247.7	.0707	883.2
29	248.3	1189.7	249.8	.0731	854
30	250.2	1190.3	251.7	.0755	826.8
31	252.1	1190.8	253.6	.0779	801.2
32	254	1191.4	255.5	.0803	777.2
33	255.7	1191.9	257.3	.0827	754.7
34	257.5	1192.5	259.1	.0851	733.5
35	259.2	1193	260.8	.0875	713.4
36	260.9	1193.5	262.5	.0899	694.5
37	262.5	1194	264.2	.0922	676.6
38	264	1194.5	265.8	.0946	659.7
39	265.6	1195	267.4	.0970	643.6
40	267.1	1195.4	268.9	.0994	628.2
41	268.6	1195.9	270.5	.1017	613.4
42	270.1	1196.3	272	.1041	599.3
43	271.5	1196.7	273.4	.1064	586.1
44	272.9	1197.2	274.9	.1088	573.7
45	274.3	1197.6	276.3	.1111	561.8
46	275.7	1198	277.7	.1134	550.4
47	279	1198.4	279	.1158	539.5
48	278.3	1198.8	280.4	.1181	529
49	279.6	1199.2	281.7	.1204	518.6
50	280.9	1199.6	283	.1227	508.5
51	282.1	1200	284.2	.1251	499.1
52	283.3	1200.4	285.5	.1274	490.1
53	284.5	1200.7	286.7	.1297	481.4
54	285.7	1201.1	288	.1320	472.9
55	286.9	1201.4	289.2	.1343	464.7

TABLE 2—*Continued.*

Absolute pressure in pounds per square inch.	Temperature in Fahrenheit degrees.	Total heat-units per pound above zero (32°).	Heat-units per pound contained in the water.	Weight in decimals of a pound per cubic foot.	Specific volume.
56	288.1	1201.8	290.3	.1366	457
57	289.1	1202.1	291.5	.1388	449.6
58	290.3	1202.5	292.7	.1411	442.4
59	291.4	1202.8	293.8	.1434	435.3
60	292.5	1203.2	294.9	.1457	428.5
61	293.6	1203.5	296	.1479	422
62	294.7	1203.8	297.1	.1502	415.6
63	295.7	1204.1	298.2	.1525	409.4
64	296.8	1204.5	299.2	.1547	403.5
65	297.8	1204.8	300.3	.1570	397.7
66	298.8	1205.1	301.3	.1592	392.1
67	299.8	1205.4	302.4	.1615	386.6
68	300.8	1205.7	303.4	.1637	381.3
69	301.8	1206	304.4	.1660	376.1
70	302.7	1206.3	305.4	.1682	371.2
71	303.7	1206.6	306.4	.1704	366.4
72	304.6	1206.9	307.3	.1726	361.7
73	305.6	1207.1	308.3	.1748	357.1
74	306.5	1207.4	309.3	.1770	352.6
75	307.4	1207.7	310.2	.1792	348.3
76	308.3	1208	311.1	.1814	344.1
77	309.2	1208.2	312	.1836	340
78	310.1	1208.5	313	.1858	336
79	310.9	1208.8	313.8	.1880	332.1
80	311.8	1209	314.7	.1901	328.3
81	312.7	1209.3	315.6	.1923	324.6
82	313.5	1209.6	316.5	.1945	320.9
83	314.4	1209.8	317.3	.1967	317.3
84	315.2	1210	318.2	.1989	313.9
85	316	1210.3	319	.2010	310.5

TABLE 2—*Continued.*

Absolute pressure in pounds per square inch.	Temperature in Fahrenheit degrees.	Total heat-units per pound above zero (32°).	Heat-units per pound contained in the water.	Weight in decimals of a pound per cubic foot.	Specific volume.
86	316.8	1210.6	319.9	.2032	307.2
87	317.6	1210.8	320.7	.2053	304
88	318.5	1211	321.5	.2075	300.8
89	319.3	1211.3	322.4	.2097	297.7
90	320	1211.6	323.2	.2118	294.7
91	320.8	1211.8	324	.2139	291.8
92	321.6	1212	324.8	.2161	288.9
93	322.4	1212.3	325.6	.2183	286.1
94	323.1	1212.5	326.4	.2204	283.3
95	323.9	1212.7	327.1	.2225	280.6
96	324.6	1213	327.9	.2245	278
97	325.4	1213.2	328.7	.2267	275.4
98	326.1	1213.4	329.4	.2288	272.8
99	326.8	1213.6	330.2	.2309	270.3
100	327.6	1213.8	331	.2330	267.9
101	328.3	1214	331.7	.2351	265.5
102	329	1214.3	332.4	.2372	263.2
103	329.7	1214.5	333.1	.2392	260.9
104	330.4	1214.7	333.9	.2413	258.7
105	331.1	1214.9	334.6	.2434	256.5
106	331.8	1215.1	335.3	.2455	254.3
107	332.5	1215.3	336	.2475	252.2
108	333.2	1215.6	336.7	.2496	250.1
109	333.9	1215.8	337.4	.2517	248
110	334.5	1216	338.1	.2538	246
111	335.2	1216.2	338.8	.2558	244
112	335.9	1216.4	339.5	.2579	242
113	336.5	1216.6	340.2	.2599	240.1
114	337.2	1216.8	340.8	.2620	238.2
115	337.8	1217	341.5	.2640	236.3

TABLE 2—*Continued.*

Absolute pressure in pounds per square inch.	Temperature in Fahrenheit degrees.	Total heat-units per pound above zero (32°).	Heat-units per pound contained in the water.	Weight in decimals of a pound per cubic foot.	Specific volume.
116	338.5	1217.2	342.2	.2661	234.5
117	339.1	1217.4	342.8	.2682	232.7
118	339.7	1217.6	343.5	.2702	231
119	340.4	1217.8	344.2	.2722	229.3
120	341	1217.9	344.8	.2743	227.6
121	341.6	1218.1	345.4	.2763	226
122	342.2	1218.3	346.1	.2783	224.4
123	342.9	1218.5	346.7	.2803	222.8
124	343.5	1218.7	347.3	.2823	221.2
125	344.1	1218.9	348	.2843	219.7
126	344.7	1219.1	348.6	.2862	218.2
127	345.3	1219.3	349.2	.2882	216.7
128	345.9	1219.4	349.8	.2902	215.2
129	346.5	1219.6	350.4	.2922	213.7
130	347.1	1219.8	351.1	.2942	212.3
131	347.6	1220	351.7	.2962	210.9
132	348.2	1220.2	352.3	.2982	209.5
133	348.8	1220.4	352.9	.3001	208.1
134	349.4	1220.5	353.5	.3021	206.7
135	350	1220.7	354.1	.3040	205.4
136	350.5	1220.9	354.6	.3060	204.1
137	351.1	1221	355.2	.3080	202.8
138	351.7	1221.2	355.8	.3099	201.5
139	352.2	1221.4	356.4	.3119	200.2
140	352.8	1221.5	357	.3139	199
141	353.3	1221.7	357.5	.3159	197.8
142	353.9	1221.9	358.1	.3179	196.6
143	354.4	1222	358.7	.3199	195.4
144	355	1222.2	359.2	.3219	194.2
145	355.5	1222.4	359.8	.3239	193

TABLE 2—*Continued.*

Absolute pressure in pounds per square inch.	Temperature in Fahrenheit degrees.	Total heat-units per pound above zero (32°).	Heat-units per pound contained in the water.	Weight in decimals of a pound per cubic foot.	Specific volume.
146	356	1222.5	360.4	.3259	191.9
147	356.6	1222.7	360.9	.3279	190.8
148	357.1	1222.9	361.5	.3299	189.7
149	357.6	1223	362	.3320	188.6
150	358.1	1223.2	362.6	.3340	187.5
151	358.7	1223.3	363.1	.3358	186.4
152	359.2	1223.5	363.6	.3376	185.3
153	359.7	1223.7	364.2	.3394	184.3
154	360.2	1223.9	364.7	.3412	183.3
155	360.7	1224	365.2	.3430	182.3
156	361.2	1224.1	365.8	.3448	181.3
157	361.8	1224.3	366.3	.3467	180.3
158	362.3	1324.4	366.8	.3485	179.3
159	362.8	1224.6	367.3	.3503	178.3
160	363.3	1224.8	367.9	.3521	177.3
161	363.8	1224.9	368.4	.3540	176.4
162	364.3	1225	368.9	.3558	175.5
163	364.8	1225.2	369.4	.3577	174.6
164	365.2	1225.3	369.9	.3596	173.7
165	365.7	1225.5	370.4	.3615	172.8
166	366.2	1225.6	370.9	.3634	171.9
167	366.7	1225.8	371.4	.3652	171
168	367.2	1225.9	371.9	·3671	170.1
169	367.7	1226.1	372.4	.3690	169.2
170	368.2	1226.2	372.9	.3709	168.4
171	368.6	1226.4	373.4	.3727	167.6
172	369.1	1226.5	373.9	.3745	166.8
173	369.6	1226.7	374.4	.3763	166
174	370	1226.8	374.9	.3781	165.2
175	370.5	1226.9	375.4	.3799	164.4

TABLE 2—*Concluded.*

Absolute pressure in pounds per square inch.	Temperature in Fahrenheit degrees.	Total heat-units per pound above zero (32°).	Heat-units per pound contained in the water.	Weight in decimals of a pound per cubic foot.	Specific volume.
176	371	1227.1	375.9	.3817	163.6
177	371.4	1227.2	376.3	.3835	162.8
178	371.9	1227.4	376.8	.3853	162
179	372.4	1227.5	377.3	.3871	161.2
180	372.8	1227.7	377.8	.3889	160.4
181	373.3	1227.8	378.3	.3908	159.7
182	373.7	1227.9	378.7	.3926	159
183	374.2	1228.1	379.2	.3944	158.3
184	374.6	1228.2	379.7	.3962	157.6
185	375.1	1228.3	380.1	.3981	156.9
186	375.5	1228.5	380.6	.3999	156.2
187	376	1228.6	381.1	.4017	155.5
188	376.4	1228.7	381.5	.4036	154.8
189	376.9	1228.9	382	.4054	154.1
190	377.3	1229	382.4	.4072	153.4
191	377.7	1229.1	382.9	.4090	152.7
192	378.2	1229.3	383.3	.4108	152
193	378.6	1229.4	383.8	.4125	151.3
194	379	1229.5	384.2	.4143	150.7
195	379.5	1229.7	384.7	.4160	150.1
196	380	1229.8	385.1	.4178	149.5
197	380.3	1229.9	385.6	.4196	148.9
198	380.7	1230.1	386	.4214	148.3
199	381.1	1230.2	386.5	.4232	147.7
200	381.6	1230.3	386.9	.4250	147.1

TABLE 3.

ONE-HALF AREA OF CIRCLES IN SQUARE INCHES.

Diameter in inches.	One-half area in square ins.	Diameter in inches.	One-half area in square ins.	Diameter in inches.	One-half area in square ins.
0—		**2—**	1.5708	**4—**	6.283
1-16	.0015	1-16	1.6690	1-16	6.481
1-8	.0061	1-8	1.7732	1-8	6.682
3-16	.0138	3-16	1.8792	3-16	6.886
1-4	.0245	1-4	1.9880	1-4	7.093
5-16	.0383	5-16	2.1000	5-16	7.303
3-8	.0552	3-8	2.2151	3-8	7.516
7-16	.7516	7-16	2.3533	7-16	7.732
1-2	.9817	1-2	2.4543	1-2	7.952
9-16	.1242	9-16	2.5786	9-16	8.174
5-8	.1534	5-8	3.7059	5-8	8.400
11-16	.1856	11-16	2.8361	11-16	8.628
3-4	.2208	3-4	2.9697	3-4	8.860
13-16	.2592	13-16	3.1063	13-16	9.095
7-8	.3006	7-8	3.2459	7-8	9.332
15-16	.3451	15-16	3.3886	15-16	9.573
1—	.3927	**3—**	3.5343	**5—**	9.817
1-16	.4433	1-16	3.6831	1-16	10.064
1-8	.4970	1-8	3.8349	1-8	10.314
3-16	.5537	3-16	3.9899	3-16	10.567
1-4	.6135	1-4	4.1478	1-4	10.823
5-16	.6765	5-16	4.3090	5-16	11.083
3-8	.7424	3-8	4.4731	3-8	11.345
7-16	.8114	7-16	4.6403	7-16	11.610
1-2	.8835	1-2	4.8105	1-2	11.879
9-16	.9587	9-16	4.9840	9-16	12.150
5-8	1.0369	5-8	5.160	5-8	12.425
11-16	1.1182	11-16	5.339	11-16	12.703
3-4	1.2026	3-4	5.522	3-4	12.983
13-16	1.2900	13-16	5.708	13-16	13.267
7-8	1.3805	7-8	5.896	7-8	13.554
15-16	1.4741	15-16	6.088	15-16	13.844

Table 4.

AREA OF CIRCLES IN SQUARE INCHES.

Diameter in inches.	Area in square ins.	Diameter in inches.	Area in square ins.	Diameter in inches.	Area in square ins.
0.		8.	50.265	16.	201.06
0.25	.0490	8.25	53.456	16.25	207.39
0.5	.1963	8.5	56.745	16.5	213.82
0.75	.4417	8.75	60.132	16.75	220.35
1.	.7854	9.	63.617	17.	226.98
1.25	1.227	9.25	67.200	17.25	233.70
1.5	1.767	9.5	70.882	17.5	240.52
1.75	2.405	9.75	74.662	17.75	247.45
2.	3.141	10.	78.540	18.	254.46
2.25	3.976	10.25	82.516	18.25	261.58
2.5	4.908	10.5	86.590	18.5	268.80
2.75	5.939	10.75	90.762	18.75	276.11
3.	7.068	11.	95.033	19.	283.52
3.25	8.295	11.25	99.402	19.25	291.03
3.5	9.621	11.5	103.86	19.5	298.64
3.75	11.044	11.75	108.43	19.75	306.35
4.	12.566	12.	113.09	20.	314.16
4.25	14.186	12.25	117.85	20.25	322.06
4.5	15.904	12.5	122.71	20.5	330.06
4.75	17.720	12.75	127.67	20.75	338.16
5.	19.635	13.	132.73	21.	346.36
5.25	21.647	13.25	137.88	21.25	354.65
5.5	23.758	13.5	143.13	21.5	363.05
5.75	25.967	13.75	148.48	21.75	371.54
6.	28.274	14.	153.93	22.	380.13
6.25	30.679	14.25	159.48	22.25	388.82
6.5	33.183	14.5	165.13	22.5	397.60
6.75	35.784	14.75	170.87	22.75	406.49
7.	38.484	15.	176.71	23.	415.47
7.25	41.282	15.25	182.65	23.25	424.55
7.5	44.178	15.5	188.69	23.5	433.73
7.75	47.173	15.75	194.82	23.75	443.01

TABLE 4—*Concluded.*

Diameter in inches.	Area in square ins.	Diameter in inches.	Area in square ins.	Diameter in inches.	Area in square ins.
24.	452.39	32.	804.24	40.	1256.6
24.25	461.86	32.25	816.86	40.25	1272.3
24.5	471.43	32.5	829.57	40.5	1288.2
24.75	481.10	32.75	842.39	40.75	1304.2
25.	490.87	33.	855.30	41.	1320.2
25.25	500.74	33.25	868.30	41.25	1336.4
25.5	510.70	33.5	881.41	41.5	1352.6
25.75	520.76	33.75	894.61	41.75	1369.0
26.	530.93	34.	907.92	42.	1385.4
26.25	541.18	34.25	921.32	42.25	1401.9
26.5	551.54	34.5	934.82	42.5	1418.6
26.75	562.00	34.75	948.41	42.75	1435.3
27.	572.55	35.	962.11	43.	1452.2
27.25	583.20	35.25	975.90	43.25	1469.1
27.5	593.95	35.5	989.80	43.5	1486.1
27.75	604.80	35.75	1003.7	43.75	1503.3
28.	615.75	36.	1017.8	44.	1520.5
28.25	626.79	36.25	1032.0	44.25	1537.8
28.5	637.94	36.5	1046.3	44.5	1555.2
28.75	649.18	36.75	1060.7	44.75	1572.8
29.	660.52	37.	1075.2	45.	1590.4
29.25	671.95	37.25	1089.7	45.25	1608.1
29.5	683.49	37.5	1104.4	45.5	1625.9
29.75	695.12	37.75	1119.2	45.75	1643.8
30.	706.86	38.	1134.1	46.	1661.9
30.25	718.69	38.25	1149.0	46.25	1680.0
30.5	730.61	38.5	1164.1	46.5	1698.2
30.75	742.64	38.75	1179.3	46.75	1716.5
31.	754.76	39.	1194.5	47.	1734.9
31.25	766.99	39.25	1209.9	47.25	1753.4
31.5	779.31	39.5	1225.4	47.5	1772.0
31.75	791.73	39.75	1240.9	47.75	1790.7

CHAPTER XVI.

ANALYSIS OF DIAGRAM FROM AMMONIA COMPRESSOR.

Fig. 88 shows the general features of a well formed diagram from an ammonia compressor, the attainment of which should be the aim in running an ammonia machine. The following names have been given to the different parts of the diagram for the purpose of

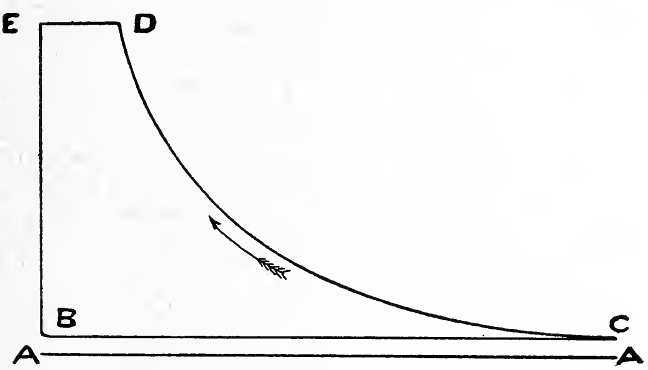

Fig. 88—Scale of Spring 80

analysis, and it will be seen that the transactions taking place in forming this diagram are *vice versa* to those taking place in forming the diagram from a steam engine cylinder, or in other words an ammonia compressor is simply a pump. The line AA is the atmospheric line and is taken after the ammonia is shut off from the indicator. The line BC is the suction line and represents by its distance above the atmospheric line the suction pressure in the compressor at any point in the stroke and there should not be a greater loss of pressure than is required to raise the suction valves and the minimum friction in the passages, as this is a corresponding loss of duty and capacity. The indicated suction pressure which is measured on the scale of the spring from the atmospheric line to the suction line, when deducted from the suction pressure by gauge, gives the loss of pressure through the passages. When the piston has completed its stroke as shown at the point C the suction valves close and the ammonia gas is imprisoned and as the piston is returning the pressure rises and forms the compression curve C D as shown by the arrow and according to Mariotte's law "a perfect gas is inversely proportional to its volume," and assuming that ammonia gas is a perfect gas and that the pressure at the end of the stroke is 20 pounds absolute which is the atmospheric pressure plus the pressure above the atmospheric and at one-half stroke (clearance neglected) the volume is one-half and the pressure is doubled or 40 pounds, and at three-quarter stroke the volume is again one-half of that at one-half stroke and the pressure is again doubled or 80 pounds absolute and so on. But should the absolute pressure at one-half stroke plus one-half the clearance be greater than double the absolute pressure at the commencement of the return stroke it indicates leakage into the compressor through the discharge valves; and if less, leakage out of the compressor through the suction valves or piston, or both. But, however, the leakage into and out of the compressor may balance each other and would indicate no leakage, therefore this test should be made on the machine itself and under the actual pressure.

When the compressing pressure exceeds the condensing pressure as shown at the point D, the discharge valves open and form the discharge line DE; the discharge pressure is measured on the scale of the spring from the atmospheric line to the discharge line and the indicated discharge pressure should not exceed the condensing pressure more than is required to raise the discharge valves, and the minimum friction in the passages, as this loss in pressure is another loss in duty and capacity. And supposing the indicated discharge pressure is 145 pounds and the condensing pressure by gauge is 140 pounds, it shows a loss of 5 pounds. Should there be a serious loss in the suction and discharge pressures the speed of the machine is beyond the capacity of the passages, a matter which can be tested by varying the speed.

The line EB is the release pressure line, and if there is no leakage into the clearance space through the discharge valve and the clearance is small, this line should fall nearly perpendicular and at right angles to the atmospheric line but when there is a leak into the clearance space or excessive clearance, this line forms a curve by curving inward and is another loss.

Such parts of the indicator as come in contact with the ammonia gas must be constructed of steel as the steam engine indicator being made of brass will not withstand the action of the ammonia gas.

In attaching the indicator the same precautions must be taken as on a steam engine cylinder.

DIAGRAMS FROM AMMONIA COMPRESSOR SHOWING LEAKS AND OTHER DEFECTS.

Fig. 89 shows a diagram from a 14″×28″ Ammonia Compressor, revolutions 50; suction and discharge pressure respectively 11 and 160 pounds by gauge.

The suction pressure measures 6 pounds and shows a loss of 5 pounds in the suction passages. The discharge pressure measures 150 pounds and shows a

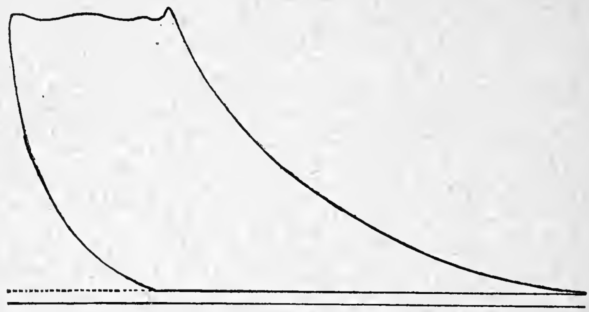

FIG. 89—SCALE OF SPRING 100

loss of 10 pounds in the discharge passages. It is evident the speed of the machine was beyond the capacity of the passages and did not allow a free passage for the ammonia gas.

In determining the loss of pressure as in all similar tests, the indicator and gauges should be compared according to the explanations given on pages 30, 31 and 32. The discharge line is wavy and is caused by a fluctuation in the pressure. When more than one machine is discharging into the condensor, it frequently causes abrupt irregularities at different points in the discharge line.

The release pressure line curves greatly inward and the pressure is not reduced to the suction pressure until the piston has moved a distance proportionate to the length of the dotted line, which measures .76 inches, divided by the length of the diagram, 3 inches, shows $25\frac{1}{3}$ per cent. loss of pumping capacity in the compressor, due to the expansion of the clearance-space gas, providing there was no leakage. But in

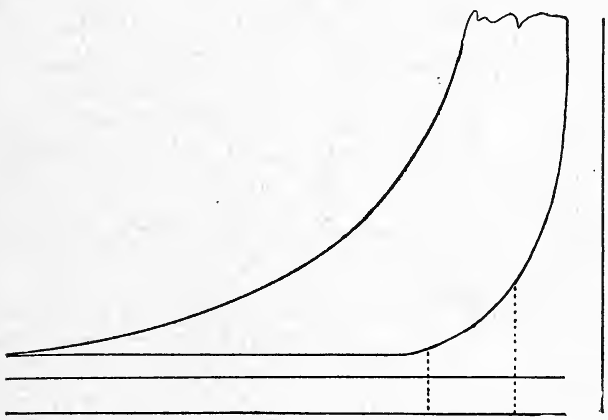

FIG. 90—SCALE OF SPRING 80.

this case the discharge valve leaked and prolonged the curve and followed with a too rapid rising in the compression curve. And since this leakage is taking place throughout the entire revolution except during the time of discharge, it results in a still greater loss in capacity.

Fig. 90 shows another diagram representing leaks and is taken from a 14″×24″ Ammonia Compressor. Revolutions, 40. The release pressure line as shown at the right hand end of this diagram also curves greatly inward and would indicate excessive clearance providing there was no leakage into the compressor. This can readily be determined on the diagram. First locate the vacuum line on the scale of the spring at such a distance below the atmospheric line as to represent the atmospheric pressure at the time and place which at sea level averages about 14.7 pounds. The lower line is the vacuum line and the line between it and the suction line is the atmospheric line. Then from the vacuum line locate an ordinate at about a point on the release pressure curve as shown by the first dotted ordinate at the left. Next locate another ordinate at such a point that its height is double the height of the first ordinate as shown by the second dotted ordinate and the distance between these ordinates is the distance the clearance line should be located from the second ordinate, and represents by its distance between it and the end of the diagram, the percentage of clearance as compared with the length of the diagram. This distance is .19 of an inch, divided by the length of the diagram 2.90 inches gives $6\frac{16}{29}$ per cent. Example, $.19'' \div 2.90'' = .06\frac{16}{29}$. The percentage of clearance multiplied by the stroke in inches gives 1.57 inches and is the clearance in inches as compared with the stroke in inches. Example, $.06\frac{16}{29} \times 24'' = 1.57''$. This represents a greater clearance than is contained in the machine and denotes a leakage into the compressor through the discharge valves. Were the clearance less than is contained in the machine it denotes a leakage out of the compressor through the suction valves, piston, or both.

This demonstration can also be applied to the compression curve of a diagram from a steam engine cylinder.

To determine the leakage from the compression curve, which in a diagram from a steam engine cylinder would be the expansion curve, the demonstration given in Fig. 84 may be applied, in which case if the pressure at the ordinate at the left is greater than double the pressure at the ordinate at the right it indicates leakage into the compressor, and, if less, leakage out of the compressor. When the theoretical curve is applied the leakage will also be *vice versa* to that shown in a diagram from a steam engine cylinder.

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