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CORLISS ENGINE





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Edwin P. Bradbury
April 1909

The Corliss Engine,

BY

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THE CORLISS ENGINE.

CHAPTER I.

Since the issue of the patent, in the year 1849, to Geo. H. Corliss, for certain improvements in the working of steam-engines, and covering the admission of steam to the cylinder by the combined action of a governor, to determine the point of cut-off at which a liberating valve-gear shall act, and thus allow a certain amount of expansion to take place in the cylinder before the end of a stroke is completed, I think it will be conceded by all fair-minded engineers, when we come to look over the ground carefully, covered by the proposition, that no improvement has been made since that time, up to this date, in the economy of working steam expansively as exemplified by this system. This assumption may be questioned even now by a few zealous persons.

The gradual development and appreciation of the Corliss system during the past thirty-six

years, has grown to such proportions as to trace this Corliss principle in the design and build of a large proportion of the engines used in our manufacturing industries in this and foreign countries, and I may say that its use for maritime purposes is better appreciated to-day, and will be still better in years to come, by the few years of experience that it has been subjected to to determine its value over other systems now in use for that purpose.

During this long period of increasing usefulness, it has seen the rise and fall of the most sanguine expectations of many inventors for its honors. Sufficient evidence has been gathered by steam-users throughout, I may say, the civilized world, as a criterion of its merits; and this has been established by facts covering economical performance for years, rather than by claims based upon theory.

STEAM-JACKETING.

When we consider the value that steam-jacketing the cylinders of an engine offers for economizing fuel, we enter into many problems of an interesting character. The mere fact of steam-jacketing a cylinder, or, in other words, subjecting the working barrel to a con-

stant circulation of steam direct from the boiler, inclosed in a separate chamber, does not imply a direct or definite saving in fuel equally available to all types of engines.

In one case it is an improvement of a very decided character, while for another, working under a different range of expansion, the additional expense for its provision would not be justifiable when we consider the question of fuel and care necessary for the proper working condition of the jacket. Steam-jacketing is a decided advantage when a liberal amount of expansion is carried out in the cylinder, or when a wide range in temperature exists between the initial temperature of the steam admitted and the final temperature at which it is exhausted into the condenser or atmosphere. Therefore its economical application is governed entirely by these two elements.

To steam-jacket a slide-valve engine, when the steam is allowed to follow nearly full stroke before cutting off, the saving in fuel, for this case, would not pay the difference for the extra cost of such practice, and is not advisable.

The shorter the cut-off in any engine, the more efficient a system of steam-jacketing the

cylinder becomes; from the fact that the amount of cylinder condensation which takes place at each admission of steam to the cylinder is reduced to a minimum, by maintaining the walls of the cylinder at a uniform temperature.

Many people suppose that this question of admitting steam to a cylinder to overcome a given amount of resistance is like unto measuring beans in a bushel, and of a practically definite quantity, whereas, the fact is, that when steam is admitted to the cylinder sufficient heat has to be imparted by such entering steam to the walls of the cylinder to bring its temperature up to a point equal to that of the entering steam, resulting in a possible amount of cylinder condensation equal in quantity to that required to overcome the work of that stroke.

It is therefore essential that when we consider the question of the economical performance of any proposed engine, we thoroughly consider the effect that an efficient system of steam-jacketing offers for that purpose. Its application is well worthy of the object, and may safely be stated to save, for engines of approved character using steam expansively,

from 8 to 10 per cent. of the fuel used. It may be claimed by a few that the speed of the engine has a marked influence upon the amount of cylinder condensation, from the theory that the walls of the cylinder are exposed for a greater length of time between each alternate stroke of the engine, for strokes of 5 to 6 feet, and about 60 revolutions per minute, and consequently more condensation takes place during that period than would otherwise occur for a higher speed of rotation and shorter stroke. This assumption, although existing as a fact for slow speeds, ceases to have much importance for what may be termed long-stroke stationary engines, say from 5 to 6 feet, or considering other circumstances of equally practical moment, for speeds even as low as 50 to 60 revolutions per minute.

At this speed, I believe, for equal points of cut-off or range of expansion, that the difference in fuel, due to cylinder condensation between this speed, of say 55 revolutions per minute, and for engines of a higher speed of rotation, say 100 revolutions per minute, a jacket is of no practical value, as I believe, for the disadvantages attending such a high speed

of engine more than neutralize any benefit to be derived from an assumed decrease in the cylinder condensation in the working of the two systems.

CHAPTER II.

INDICATOR-CARDS.

I do not consider that the engine making the finest-looking indicator-card, as that term is generally considered to-day by engineers, is necessarily working to the best advantage, regarding friction and economy.

I am aware that this impression has been given by some engineers, who have suggested to manufacturers that such must necessarily be the case under all circumstances. I am free to admit that I think there are other conditions to be fulfilled, in the satisfactory working of an engine, equally as important as giving an indicator-card: "fulfilling all the conditions necessary for economy" with their square corners, excessive compression, and plumb-line induction, representing the admission of steam to the cylinder.

To my eye, the best-looking card is that showing the least amount of frictional resistance to

be overcome for a given amount of work performed, and compression sufficient to gently affect the moving parts as they come to rest; to turn the center with an expansion-line following well-established rules, and a movement of the exhaust-valves to allow in the least possible time, during the first part of the return stroke, the piston to have the full benefit of the vacuum where a condensing-engine is employed.

If these conditions are fulfilled, I believe that the engine is accomplishing good work economically, with a minimum amount of friction for the power developed. All these conditions may be obtained, notwithstanding we may have a card where the steam-line falls off as in Fig. 1. To some, this may seem a very serious defect. If we wish to drive machinery economically, assuming we have tools to do it with, we should so adjust its condition of working that it may be able to produce that result with as little effort to move itself as possible. This implies an admission of full boiler pressure to the cylinder after the crank has passed the center to help turn the crank-shaft rather than retard it. This will be the case if we attempt to make a perfectly plumb steam-admission line on the card.

To bring about this early admission of steam, or steam-lead so called, to produce a steam-admission line at right angles to the line of motion it is necessary to so place the eccentrics relative to the crank as to have the steam-valves which are operated by it in such a position that boiler pressure may be upon the

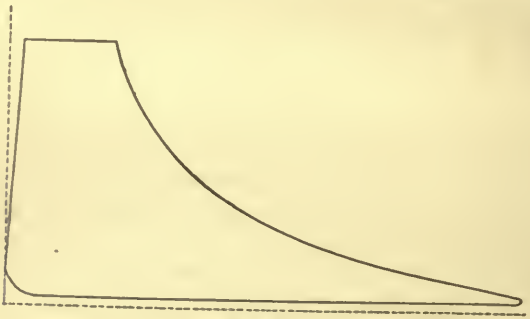


FIG. 1.

piston the instant the direction of motion is changed. To accomplish this implies an opening of the steam-valve before the crank comes up to the center, as shown in Fig. 2. Now any such steam pressure admitted to the cylinder with the crank in that position is a detriment, causing an increase in the friction of the en-



FIG. 2.

gine by a longer application of pressure upon crank and cross-head pins, and main bearing ; also, a diminution in the energy of the wheel, which, necessarily, has to be restored by additional steam at the next stroke of the engine, and a generally debilitating effect upon all parts of the engine.

I believe that the time to admit full steam pressure to an engine, leaving aside the question of handsome indicator-cards, is when the crank has arrived at such a point in its travel (see Fig. 3) as to be influenced by such pressure, with the effect, as I have said before, to momentarily hasten rather than retard the action of the driving pulley.

Of course, I do not wish to be understood as favoring extremes, even in this direction ; but I am willing to accept Fig. 1 as a basis for my indicator-card. That, in my judgment, is best suited for the majority of engines as ordinarily run, producing the least friction in accomplishing a given amount of work, and in the easiest running condition, all things considered, for doing that work.



FIG. 3.

CHAPTER III.

I have seen, a great many times, the folly and evil effect of allowing a fine-looking indicator-card to be the ruling spirit governing valve adjustments, where questions of a practical character have not been considered of sufficient importance to justify thought, so

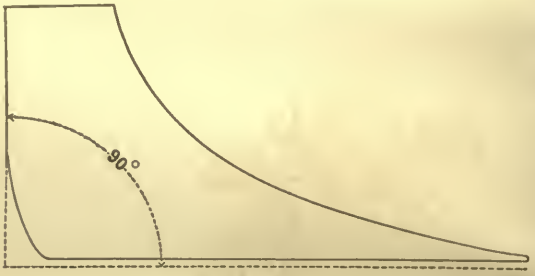


FIG. 4.

long as a "fine-looking" card, as shown in Fig. 4, is obtained. This state of affairs requires adjustment, so that the engine may show an indicator-card worthy of consideration, and be given a chance to do its work, untrammeled by steam-lead or the excessive compression that is now "fashionable." High compression is assumed to be a necessity for all high-speed

engines, from the fact that it is much used in locomotives.

If we compress steam in the cylinder by an early closing of the exhaust-valve, up to a point about equal to the terminal pressure, we have reached the limit desirable for condensing-engines; for the majority of non-condensing engines the compression should be about 5 lbs. in excess of the terminal pressure. These limits are suggested by a consideration of practical questions equally as important to the manufacturer as those of a theoretical nature, and are applicable for points of cut-off in the two systems best adapted for economy, coming within the range of about 1-5th and 1-7th cut-off, respectively.

THE GOVERNOR.

The function of a governor is to act in accord with each variation of load, and to so limit the quantity of steam to be admitted to the cylinder as to overcome the resistance of the load, and thus maintain a uniform speed of rotation of the engine-pulley. This cannot be properly done unless the action of the governor is untrammeled by unnecessary friction, so as to instantly meet changes in the load as they occur from time to time.

We should, therefore, see that all of the working surfaces of the governor are in proper running condition, and a quality of oil used for lubrication that will not "gum up" after it has been applied for a time. We should, also, see that the oil, or dash-pot, is in good working order, with a constant supply of oil to gently retard any sudden fluctuation in the movement of the regulator.

On engines that have been speeded up to relieve their working, and bring about an earlier cut-off (by reason of an overgrown mill), we may possibly find that the speed of the regulator has been allowed to remain the same as before the change, with a possible chance for the overseer of weaving to suggest that the speed is not quite so uniform as before the engine was speeded up.

To anticipate any such complaint we should, while we are charging the speed of engine, change also that of the regulator, and to overcome the effect of a higher speed of rotation upon the elevation of the balls of the regulator (and its equivalent effect upon the point of cut-off), we should weight the regulator down (as shown in Fig. 5) until the requisite speed is at-

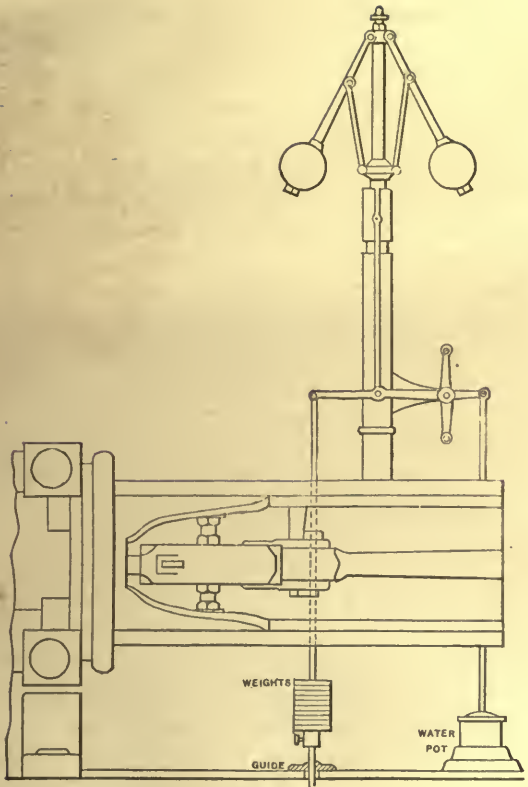


FIG. 5.

tained for the engine. By doing this we may divide up the interval of time, which such variation and speed of engine has upon the governor, and its consequent reaction upon the cut-off mechanism of engine, to a shorter degree, and thus control the point of cut-off during one revolution of governor in place of one and one-half revolutions, which would be the case if the governor was allowed to run slow on an engine that had been increased in speed.

After this has been done, especial care should be taken to see that all bearings are kept thoroughly clean and free, for the lubricant to act, for this speeding up is attended with more or less risk, on account of imperfect lubrication and non-adaptability of the engine and its parts for such a change in the speed.

CHAPTER IV.

In engines of this type having a detachable valve gear where the motion for working the valves is derived from the action of an eccentric, it follows that when there is no lap on the valve to be worked off or the steam valve set edge to edge with the port opening, that the eccentric will be at its half-throw, or at

right angles to that of the crank when it is on its center.

During the rotation of the shaft, the eccentric would therefore arrive at its greatest throw and the opening motion of the steam valve would cease, and thus the detaching mechanism remain inoperative, while the crank or piston reaches, practically embraces, half stroke, and any liberation of the cut-off gear actuated by the governor or other means must take place, if at all, before this point is reached in crank travel, or before the eccentric commences its return stroke. If this action has not taken place the steam valve then commences to close positively at a speed governed by that of the eccentric. In order to have a safe working lap of the steam valve before the exhaust port upon that end is open to atmospheric pressure, or the condenser, and thus prevent any blowing through during the relalative time of closing the steam and opening the exhaust valves, it is essential that the valves admitting steam to the engine should be given a definite advance in their movement relative to that of the exhaust.

This is commonly called lap ; this and the

amount of it increases with the size of the engine. To neutralize this effect upon admission of steam to the engine, and have the valve gear in readiness to act for that purpose, when the crank comes up to the center, it is evident that the position of the eccentric must be more than 90° with relation to the crank; or at a position beyond its center of half throw, in order to remove the lap given to the steam valve, and to take a position to admit steam to the cylinder. This, of course, reduces the range of expansion that steam may be carried for that end of the cylinder, by allowing less time during the travel of the eccentric before it reaches its full throw for the cut-off to act, which must occur, in this case, before half stroke is reached, or upon the opening motion of the steam valve. Occasionally, we may see an indicator card where the cut-off had apparently taken place after more than half stroke of the engine had been covered, which is a very bad condition, however, considered economically. At this point in the piston travel the velocity is at its maximum, while that of the valve-gear, actuated by the eccentric nearing its dead point of full travel, is at its

minimum. These elements, while working, as a matter of fact, with a positive ratio of action to one another, admit at this high velocity a certain amount of piston travel to take place after the point of detachment has been reached by the valve-gear before motion can be imparted to the valve and its connection sufficient to cover a full port, and prevent further admission of steam to the cylinder. This apparent cut-off taking place after half stroke has been reached is more prominently defined on the card as we increase the lap of the steam valve, and, by increasing this lap we are enabled to open the exhaust valve sooner for the return stroke, thus giving a free opening for the exhaust in relation to the former, and necessarily enforcing a later closing of the valve, reducing the amount of compression; this may be considered by a few objectionable, and we will have something to say further on about it.

CHAPTER V.

SETTING VALVES.

Upon the wrist-plate of the engine will be found three marks; one representing the position at half-travel, or center of motion, while

the remaining two represent the extreme motion of the wrist-plate as actuated by the eccentric.

Upon the wrist-plate stand, or pin, as the case may be, will be found a mark coinciding with those on the wrist-plate, designating the extreme motion and center of motion or travel,

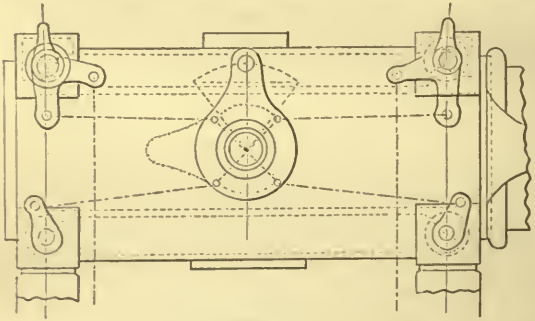


FIG. 6.

as per Fig. 6. These are to be our guides in arranging for the relative time of action between the four valves.

Upon removing the back bonnets from our engine we shall see, also, a mark upon each face of the valve-port, showing the location and width of port openings in relation to the cylinder; thus furnishing a starting point for

the setting of the four valves without the trouble of removing them from the port opening.

Upon the end of the valves will be found, also, a mark in line with the opening edge of the valve, as shown by Fig. 7.

In preparing to set valves the first operation

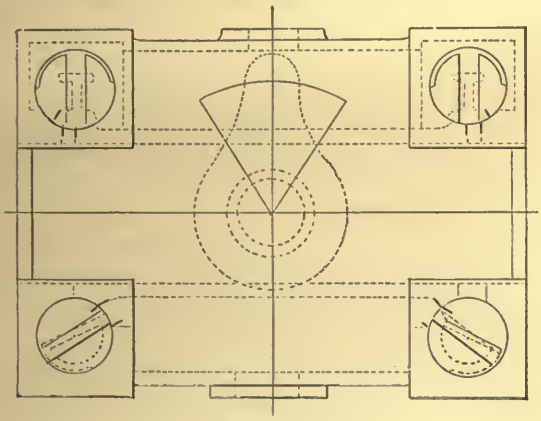


FIG. 7.

is to place the wrist-lever at half-throw, which, for horizontal engines, is represented by the center of carrier and wrist-lever pin being plumb, but for the majority of beam engines the center of motion is represented by the

dotted and circular marks, as indicated upon Fig. 6.

At this position, designated above, the marks on the wrist-lever and pin should correspond, and the whole mechanism be secured in that position, by placing pieces of paper between the washer on the end of the pin and wrist-lever, so as to produce friction sufficient to hold the whole in the desired position when the nut on the end of the pin is screwed up.

This being accomplished, we may consult the annexed table giving the lap of the steam-valve, and the relative position of exhaust-valve when the wrist-lever is at its center of motion, and thus fix upon the lap for the steam-valve and the position of exhaust-valve desirable for the size of engine under consideration.

By lengthening or shortening the connections leading from this wrist-lever to steam-lever we bring the position of the opening edge of the steam-valve to correspond with the amount of lap fixed for that case, which, of course, should be the same for each end of the cylinder.

For the position of the exhaust-valve, by con-

sulting the table herewith we find for a 20′ engine $\frac{1}{16}$th of an inch opening.

TABLE I.

POSITION OF STEAM AND EXHAUST-VALVES WITH WRIST-LEVER AT ITS CENTRE OF MOTION.

Size of engine.	*Laps of steam valve.*	*Exhaust valve open.*
12″	$\frac{1}{4}$″	$\frac{1}{32}$″
14″	$\frac{5}{16}$″	$\frac{1}{32}$″
16″	$\frac{5}{16}$″	$\frac{1}{32}$″
18″	$\frac{3}{8}$″	$\frac{1}{16}$″
20″	$\frac{3}{8}$″	$\frac{1}{16}$″
22″	$\frac{3}{8}$″	$\frac{1}{16}$″
24″	$\frac{7}{16}$″	$\frac{3}{32}$″
26″	$\frac{7}{16}$″	$\frac{3}{32}$″
28″	$\frac{7}{16}$″	$\frac{3}{32}$″
30″	$\frac{1}{2}$″	$\frac{1}{8}$″
32″	$\frac{1}{2}$″	$\frac{1}{8}$″
34″	$\frac{1}{2}$″	$\frac{1}{8}$″
36″	$\frac{1}{2}$″	$\frac{1}{8}$″
38″	$\frac{9}{16}$″	$\frac{3}{16}$″
40″	$\frac{9}{16}$″	$\frac{3}{16}$″
42″	$\frac{9}{16}$″	$\frac{3}{16}$″

This amount for each end of the cylinder is obtained by lengthening or shortening of the connection leading from wrist-lever to exhaust-arm, while the wrist-lever still remains at its center of motion, until we bring the marks on the end of the exhaust-valve to the distance required from the closing edge of the exhaust-port.

We have now established the relative time desirable for the opening of the steam, and closing of the exhaust-valves, that each should act relatively to one another, and they should remain so without further adjustment being allowed upon the steam or exhaust connections until sufficient reason is established by practical working to warrant a modification in adjustment, and then only upon a full consideration of all the facts attending the working and the service for which the engine is used. When we change any one of these connections we alter its time of movement, and destroy the unison of action between the closing of one valve and the opening of another. If we shorten one steam connection, the effect is to allow that end to open quicker than its neighbor, but at the expense of a reduction in the safe working lap that it should have with relation to the exhaust-valve. If we lengthen the connection, we increase the lap of the valve, with the effect of opening it later, which condition would call for a greater advance of the eccentric with relation to the crank.

Again: if we lengthen the connection leading to exhaust-arm, we add lap to the valve, and

increase the amount of compression at one part of its movement, while for the other part its time of opening will be later relative to the movement of the piston, and retard the exhaust to the condenser.

If we shorten the connection, we increase the opening when the stud-plate is on its center of motion, and consequently decrease the amount of compression thereby, with a corresponding earlier opening of the exhaust-port.

This additional opening given to the valve by shortening connections reduces, also, the lap of the valve over the port that the exhaust-valve should have, during the period of time coincident with that of the steam valve when upon the point of opening for the admission of steam to the cylinder.

CHAPTER VI.

SETTING VALVES CONTINUED.

After we have "squared the valves," to use a shop phrase, it is necessary to look after the carrier and eccentric rods, and see that their travel is equidistant from an established center line of motion; giving our attention first to the rod leading from eccentric to carrier-arm.

and termed the eccentric-rod. This rod is the first acted upon, and after it is once adjusted will not be affected by any future adjustment found necessary for the carrier, or rod leading from carrier-arm to wrist-lever. This might be found necessary if we should reverse the operation, and thus make it necessary to go over the work a second time.

After we have found that the throw of the carrier-lever is equidistant from an established plumb-line, in its extreme travel each way, brought about by adjustments in the length of the stub end, in the end of the eccentric-rod, until such a result is accomplished, we may repeat the operation of turning the engine by hand for the benefit of the carrier-rod, or the connection leading from the wrist-lever to the carrier-arm, and so adjust its length that the wrist-lever will travel the same amount each way from the center of motion fixed by the marks upon the wrist-lever and pin.

As we now have the valves adjusted relatively to one another, and also the throw of the carrier-arm and wrist-lever equally divided from center of motion, we place the crank upon one center or the other, and roll the ec-

centric around on the shaft in the direction that the engine is to run, until we bring the opening edge of the steam-valve on that end of the engine that is next to take steam, $\frac{1}{32}$ of an inch beyond the line on the valve face representing the opening edge of the port, and secure the eccentric in position.

It is well then to bar the engine around to the other center, and note if a similar opening is obtained, which will be the case if the rods are properly adjusted.

After satisfying ourselves of the correctness of the movements, we may replace the back bonnets, and proceed to adjust the cam-rods leading from governor to the detaching levers on the valve-gear.

In adjusting these cam-rods we should first block the regulator up to its extreme point of travel, and secure it for a time in that position. Then lengthen or shorten the rods leading from the governor to the valve-gear on each end of the cylinder so as to bring the detaching apparatus into action, and allow the valve to be unhooked with the regulator in that position, when we roll the valve around by means of a starting-bar placed in the wrist-

lever, until the steam-port is uncovered about 1/8th of an inch. This adjustment will prevent the engine getting beyond the control of the governor if the load is suddenly removed from it by breaking a shaft or belt in the mill, or by extreme variation of the load, as in rolling-mill practice and similar service.

After the adjustment of cam-rods has been made, we may lower the governor down to its lowest position, to see that the valve-gear will not be detached at that level. In this position of the governor steam should follow full stroke, and the valves should not be liberated until the governor has reached an elevation corresponding to nearly the normal speed of the engine and the load carried.

CHAPTER VII.

LUBRICATION.

The question, which is the best method of lubricating crank and cross-head pins of engines running continuously is not positively settled in my mind.

There is one thing certain: a plan must be devised that will admit of putting any amount of lubricant at pleasure upon the crank and

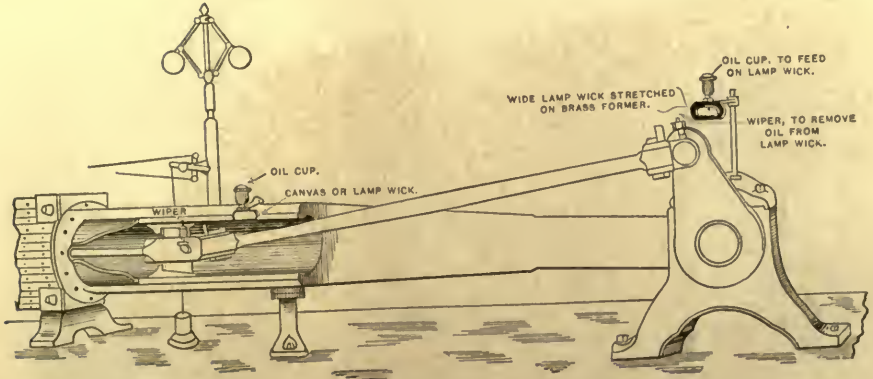

OIL CUP. TO FEED ON LAMP WICK.
WIDE LAMP WICK STRETCHED ON BRASS FORMER.
WIPER, TO REMOVE OIL FROM LAMP WICK.
OIL CUP.
CANVAS OR LAMP WICK.
WIPER

FIG. 8.

cross-head pins, while the engines are running at their regular speed; either by first dropping it upon a ribbon of canvas and, in turn, taking off at each revolution by a wiper, secured to the mouth of an oil-cup on the crank end of the connecting rod, and also for the cross-head, as shown by Fig. 8.

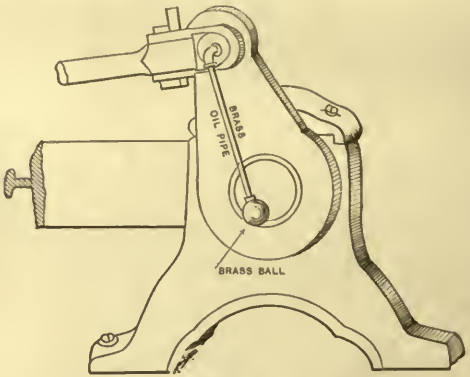


FIG. 9.

This, I think, is a good plan; probably equal to any yet introduced for that purpose, although, in being exposed to all the dust in the room, it is, I think, more destructive to the bearings than the oil-pipe having its opening in line with that of the main shaft and con-

nected to the crank-pin, as in Fig. 9. When engines are required to run only 11 or 12 hours per day, with an interval at noon of 10 or 15 minutes to fill up the cups, I think the most satisfactory method of lubrication for moderate speeds is oil cups, and they should be of a sufficient capacity to run all day with good feed.

Main and back bearings may be lubricated by cups having a sight-feed, and should be protected by glass casing around them, to prevent the dust from working down into the bearings. Or we may employ grease in boxes on the main bearings, having covers to protect them. In many cases this is an excellent lubricant for heavy journals, although this method is more expensive to maintain than with oil, from the fact that the drippings from those journals cannot be used over again, and are, of necessity, thrown away. It is a most excellent plan to provide drip-pans under the crank, main and back bearings, eccentric, and end of slides, to catch all waste oil, not only as a means to promote cleanliness, but as a matter of economy. Such drips should be afterwards strained through cotton-waste, and sets of sieves having bottoms made of wire-gauze

of different degrees of fineness, as shown by Fig. 10. After being passed through the strainers and removing all grit that may have been gathered, the oil may be used over again upon the heaviest of bearings, without any fear of trouble. This will materially reduce the oil ex-

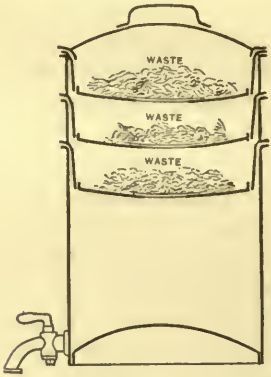


FIG. 10.

penses. Cases may be cited where the waste oil from the hangers on line shafts of mills has been collected and treated in this manner, and afterwards used upon heavy journals that were subjected to high temperatures in inclosed engine-rooms; it worked quite satisfactorily.

This plan is certainly worth considering, where an eye is had to the most economical method of lubrication. Another plan of lubricating bearings for heavy main lines of shafting is to provide rings made of iron or brass about 3/8ths of an inch wide, and 3/8ths thick, revolving upon the top of the shaft through recesses cored in the top and bottom shells, and of a sufficient diameter to reach from the top of the shaft to near the bottom of an oil cavity formed in the lower part of the pillow-block. This plan maintains at all times a constant and uniform oil-bath for the journals, as the oil is carried up by the revolving ring, actuated by the shaft from the oil-pan beneath, using the oil over and over again, without any perceptible loss. It is needless to say that these rings and oil-pans should be protected from the dust, and a ring provided at each end for a bearing of moderate length, while for a bearing of excessive length one may be provided in the middle, and thus insure a uniform flow of oil throughout its entire length.

As a matter of interest bearing upon this subject, I would refer the reader to a paper prepared by the writer for the American Society of

Mechanical Engineers, and read at its meeting held at Atlantic City, May 28, 1885, wherein is contained a tabular statement of the quantity of oil used for cylinder and general lubrication in an engine-room, and for a number of mills.

CHAPTER VIII.

AIR-PUMPS.

Trouble is sometimes experienced with the air-pump of the condensing apparatus connected to engines; for it seems as though determined to make the most disagreeable noise possible by continual hammering of its valves at each change in the direction of motion. This state of things is most likely to occur where valves of large area are operating at a speed beyond their capacity and endurance. Within the past few years, in order to meet the demands of changes in the mill, it has become common to speed up the engine, but in overcoming the want of power there is a possibility of encountering trouble in the air pump. The only remedy is in relieving the valves and preventing their hammering while working.

Horizontal pumps are more susceptible to this complaint than vertical pumps having a

short stroke of the bucket, and a multiple of small valves for the discharge of the water of condensation.

I have yet to see a long-stroke air-pump that will run smoothly to the satisfaction of the majority of engineers at, say, 55 to 60 revolutions per minute, where there is but one large valve provided for the discharge of the water from the condenser. I may go still further and insist upon a number of small valves (in place of one large one), on the system first inaugurated by the late Henry R. Worthington on his pumping-engines for Water-Works service.

This plan is used by but few makers of air-pumps, but it is correct both in theory and practice, and should be universal if satisfactory ends are to be accomplished.

This plan of distributing a given area of discharge among a number of small valves in connection with a short stroke of the air-pump bucket as we increase the speed of the engine, will insure a most satisfactory and smooth-running pump.

For the ordinary speed of engines driving cotton-mills, from 60 to 65 revolutions per minute, we may safely employ pumps of 12-inch

stroke ; for 75 revolutions of the engine 10-inch stroke, and for 100 revolutions 7½-inch stroke should not be exceeded.

In determining the proper capacity of an air-pump we may make it about ⅙th of the capacity of the steam cylinder, for ordinary condensing engines where the air-pump is single-acting.

This proportion is ample and should always be provided for to meet the demands of a moderate increase in the speed of the engine, if such should occur from any cause.

To relieve the hammering of air-pump valves I should recommend drilling into the space directly under the delivery-valve, inserting a ¼" pipe and valve, to be regulated at will for the admission of a slight quantity of air under the delivery-valve.

As this will be between the foot-valve and delivery-valve, and communication with the condenser is checked by the foot-valve and the water upon it, it will have no effect upon the vacuum in the condenser or engine. As to the regulation of the valve for the admission of air, experiment will soon demonstrate the best adjustment for working conditions.

CHAPTER IX.

CARE OF MAIN DRIVING GEARS.

Gears, like all other parts of machinery, require looking after, to see that a proper bearing is maintained throughout the length of the tooth and that they are working in their most advantageous positions regarding the pitch lines of one another.

While advising a system of periodical inspection we should, at the same time, insist upon a thorough lubrication at frequent and stated intervals.

I am willing to admit my preferences for gearing for the majority, and I may say all cases, for transmitting heavy powers to first movers in mills, on the ground of less revolving weight, shorter main shafts, and the general compactness of engines, and, I think, when all things are considered, the first cost will be in favor of the geared plant.

But to come back to my text: this matter of lubrication for gears is a very simple affair, requiring but a short time every other day for the work, which if carried out, will reduce to a minimum any liability to abrasion, or teeth

getting out of shape ; resulting in a most satisfactory record, as to working, for a long term of years.

This, of course, upon the assumption that the gears were properly made and proportioned to the work to be performed in the first instance. My experience leads me to believe that the most satisfactory method of lubricating any gear is to apply the lubricant to the driving side of the tooth by a paint-brush, as the gear is moved slowly around by hand. This method insures placing the grease where it will do the most good, and not on the ends of the teeth, where, it is more than likely, the greater portion will be wasted if applied during working hours by pouring it on. The proportion of the mixture which I have found to give good results is as follows :

One pint of Carolina pitch-tar ; one pound of plumbago, four pounds of tallow ; melt together and thin down with one pint of raw linseed oil.

In the course of their journeys many, no doubt, have seen gears where the utmost care has been taken of them, and others so bad that if any quantity of grease should be applied

several times a day, it would be fruitless to remedy any defects of construction ; and to hope for smooth running gears, under such conditions, would be useless. Then again, we come across gears which have been in constant service for many years, with no perceptible wear upon the surfaces of the teeth, not even upon those of the jack-shaft-gear, which is, of course, exposed to the most wear. I well remember a set of gears, which have been in constant operation for the past seventeen years. The wear is scarcely perceptible upon the teeth, notwithstanding the fact that they have transmitted about seven hundred horse power for the first five years and about fourteen hundred for the remaining twelve ; they are doing excellent work to-day.

Gears with wood and iron teeth working together are a very good institution, where the noise of iron gears would be objectionable in a room, and it is surprising, sometimes, what an amount of power they will transmit without interruption, if properly proportioned to the work done, for a great number of years with but little wear.

I remember an instance where, in the year

1847, there was a steam plant put into a cotton mill having for its driver an internal gear with wooden teeth. This gear was 12 feet diameter, working into one of 4 feet, the smaller having iron teeth. Each gear had a 10 inch face, and 3.14 inches pitch.

Since they were first started there have been put into the large gear less than a half dozen new wooden teeth, and this was only brought about by repairs to a segment, broken by bricks falling from a part of the foundation. Up to this date the original wood is running in the same gear, and has been in constant service; it is driving to-day about 170 horse power, at 72 revolutions of engine. This, I think, is a hard record to beat, and speaks well for the management, as well as for the lifetime of gears having wooden teeth.

The wood in this case was hickory, and confirms the opinion that it is quite a difficult question to answer, as to what is the lifetime of wooden gears if proper care is taken of them while in operation, and well proportioned in the first instance.

Foundations for Gears. — There have been many excellent gears ruined when first started

by being placed upon shafts much too small for the requirements, or from being fastened upon timber work subject to shrinking and swelling, and, consequently, getting out of line from being fastened to such unstable work. If the locality is such as to make it necessary to support gears on, or from timber work, the bearings for the gears should be first secured to a heavy cast-iron plate, and entirely independent of such wood-work. This plate should be, in turn, fastened to the timbers by bolts independent of those for the bearings, admitting of a slight adjustment of the gears without disturbing the alignment or level of the foundation plate. This same plan may be adopted when it is necessary to remove one or both gears, for repairs to them, or shafting. What is still better, and I think the only proper way, is to provide a good stone and brick foundation of ample weight, when the conditions will admit of it.

In that case the bearings may be placed directly upon the stone, properly dressed off, and securely bolted to the bottom stones in foundation.

If this plan is carried out we may rest as-

sured that, with gears of proper dimensions, suitable to the work to be done, they will run smoothly with a minimum cost for repairs, friction and maintenance for a long term of years.

CHAPTER X.

HEATING MILLS.

In the majority of cotton manufactories, steam for heating and drying purposes is required in more or less quantities in the different departments. To supply the heat for this work necessarily involves an expenditure of money, or its equivalent, coal, and it becomes a question as to the best method of supplying the heat in the most economical manner. If a limited quantity of steam is required, at intervals only, not equal to what would be discharged on one exhaust of the engine, the most economical plan would be to heat with steam direct from the boilers, and to run the engine condensing. Upon the other hand, if we have use in the different departments for all the exhaust steam that would be discharged from the engine, we may divide the exhaust-chest into two independent chambers, as in Fig. 11, midway between the two ports, and so arrange

our conditions of running as to be able to exhaust from one end of the cylinder direct into the condenser, while, for the other end, the exhaust may go to the heating pipes in the mill.

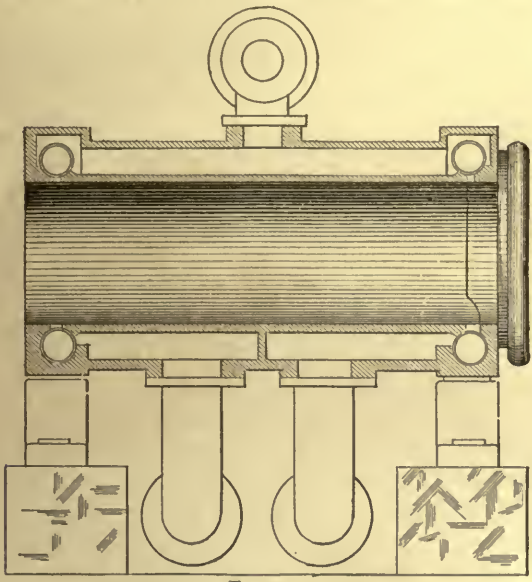


FIG. 11.

Or, we may run wholly condensing or wholly high pressure, as required by the varied conditions of the seasons.

In the event of having use for all of the exhaust steam from the engine, we should, by all means, avail ourselves of the advantages which a non-condensing engine offers over a condensing, or even a compound engine, for similar work, from the fact that we are able to utilize to better advantage the heat in the steam passing from the cylinder after it has done its work; it is equally as available for heating purposes.

Woolen mill having dye-houses attached would come under this head, and for that, and similar service, the best compound system or condensing engine that could be devised, would prove inferior to a plain non-condensing engine; all the conditions being taken into account.

In one case we have a plant for driving machinery only, with an expenditure of money for fuel for that purpose; in addition we have that required for generating steam in sufficient quantities to meet the demands for heating purposes direct from the boilers, whereas in the other case we install a high pressure engine, and while we are running our machinery and using steam expansively for that purpose,

we are discharging sufficient exhaust steam at a pressure of from 1½ to 3 lbs. in cotton mills, and from 4 to 8 lbs. for woolen mills. This is available for heating water, drying, etc., etc.

In heating by live steam we accomplish but one result, with no benefit to be derived from its expansive force, but by first putting the amount of steam required for heating purposes through an engine, we derive benefit from its expansive force. We shall find that, for the same fuel, we are enabled to not only heat, but to furnish a certain amount of power. This, at first sight, may appear strange reasoning, but it is attested by well established facts and confirmed by the experience of manufacturers who have established the economy of using exhaust-steam over live steam for heating purposes, both in woolen and cotton mills.

The whole secret of using exhaust-steam successfully is to provide ample main-pressure lines, and it is then available a long distance from the source of supply.

In the ordinary construction of cotton mills it is well to first carry the main exhaust line to the upper room as soon, and in as direct line as possible, and there locate a back-pressure

valve under the roof, admitting of a change of pressure at will.

From this point, we run to the center of the mill and across to each wall. Thus, in the upper story we have two main lines for the distribution, descending from this point to the basement. From these upright lines we branch off to each end of the mill, and with a system of piping on each side of the room as we descend, a uniform reduction in the size may be made, after each heating system has been provided for.

The drip from the circulations should also return to the basement, where it should be trapped and collected in a tank for feeding the boilers. At the upper end of each drip-pipe means should be provided for blowing out any air that may be in the pipes.

CHAPTER XI.

ENGINE FOUNDATIONS.

In dealing with this subject the probabilities are that a practical mason would be the most proper person to give advice, regarding the best way and plan of preparing an engine foundation, and to consider the most econom-

ical and satisfactory method of treating each individual case, so that in the end it will be the most suitable, and cover all the requirements to which it is to be subjected when finished. It is this knowledge of a full understanding of the necessities of the case that first prompts me to hold my peace regarding this subject for abler hands, but upon reflection suggestions may be in order, covered by an experience sufficient to form an approximate judgment upon the subject.

In preparing a bottom for foundations of engines many different characters of material are to be met with, and each requiring for their treatment quite a different mode of operation, so that one plan of a successful character, carried out at one place, and in one material, may be quite inadequate to meet demands when carried out in a material of a different character in another locality.

The easiest solution of this problem, next to rock foundation, is when a general bottom is found, below the surface. This state of things being established, we may excavate about 18 inches below the bottom of engine foundation, and after being leveled off it should be

thoroughly sprinkled with water, and well tamped down with a heavy beater. This being done, fill in about 3 inches in depth of the same material (gravel), and honestly repeat the operation, until there is a thoroughly compact mass of about 18 inches deep. Upon this base we may commence directly, without any further material, the brickwork of our foundation laid in good cement.

The width of base should be from 2 to 3 feet more than the width and length of the engine foundation proper, as shown by Fig. 12.

This foundation, or base, we may carry up about 5 courses in height before we come to the pocket-hole under the bottom stones, for the nuts and washers of the foundation-bolts. The absence of heavy footing stones, upon which to start our brickwork, may be looked upon with some distrust by a few, but facts have proven this arrangement to fully withstand the vibration, and to cover all of the requirements for that purpose.

Hard-pan, or clay-bottom, may be treated in the same manner. The most troublesome of all material to deal with, and to sustain a foundation upon, is quick-sand impregnated

with springs of water, It is here that considerable thought should be given to the situation

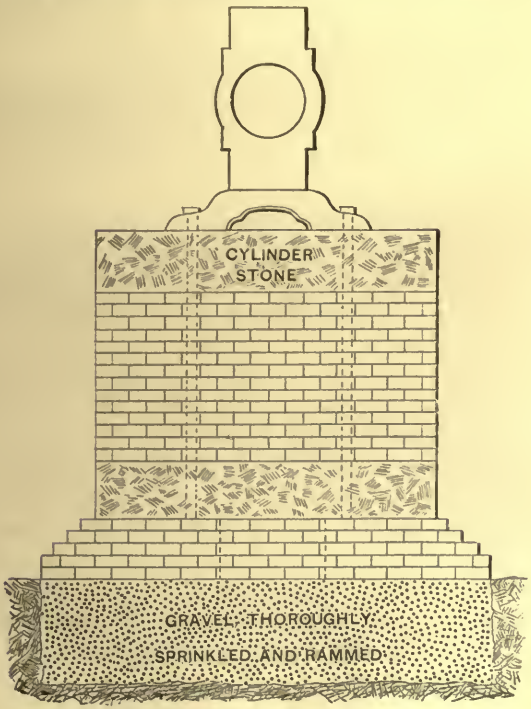


FIG. 12.

and means provided to safely accomplish, in an economical manner, a first-class piece of work.

The most successful method of dealing with this class of material, quick-sand, that has come to our notice is that devised by Mr. Geo. H. Corliss for the bottom of the foundation for one of his pumping-engines, and its building upon the bank of a river. This plan consisted of driving down any amount of stone, or other material, of small size, where required by the lines of the building and space covered by foundation of engine, by a weight in shape of a ball weighing about 4,000 lbs. falling any convenient distance, varying from 10 to 25 feet. At each blow of the ball, rolling from a cradle upon which it had been hoisted, the loose stone, dumped from carts, were driven into the sand, spreading out as they descended, and finally making, as the work progressed by repeated addition of material and subsequent blows from the ball, a compact mass of sufficient solidity able to stand the test of years, without the least sign of settlement in any part of the building, or foundation of pump and engine.

This plan of such a novel and successful

character is susceptible of application to many localities where similar material is to be contended with, and may be applied with the assurance of affording an economical and, at the same time, a successful means of producing foundation for almost any construction.

CHAPTER XII.

Another plan is to excavate to the proper depth required between and under sheet piling, driven down as the excavation progresses until the depth is reached, where a bed of concrete, formed by cement, sand, broken stone or brick bats, is placed; the depth of same varying for each case to meet the requirements, usually from 24″ to 36″, and of such a dimension of base as to present a large bearing surface upon the sand. Upon this bed of concrete, after sufficient time has elapsed to allow the material to harden and settle, is commenced the brick work of the foundation.

If this plan is properly carried out, and care taken not to cover too much space at a time, so as to allow the mass to harden while the process is in operation in putting down the

concrete bed, very good results may be expected for most constructions.

Crib-work, forming a heavy flooring made of timber, is sometimes resorted to in foundations, but as the bond between such material and cement is not of the best, and what would be desired for such work, we cannot consider this plan as efficient as those already referred to. Aside from the difficulty of getting an even and solid bearing for a timber platform upon the sand, any settlement of a part of this yielding material is likely to cause trouble in the foundation, from the lateral movement which may take place.

MATERIAL FOR FOUNDATIONS.

The most satisfactory materials, so far as my judgment serves me, to build engine foundations of, is a good quality of hard burned brick, well laid in cement mortar, composed of one part cement to one of sand, with plenty of water used upon the brick, and while laying.

At the top, for the parts of the engine to rest upon, granite blocks of good size or iron plates may be bedded in cement, granite being by all means the most desirable.

Similar blocks of granite should be provided

for the bottom and for the bearing surface of the foundation bolts. This is far superior to any wholly stone foundation that can be built, for equal cost.

As a guide for the location of all bolt-holes, throughout the work of building the foundation, it is a very good plan to first prepare a wooden templet, carefully laid out from the drawings, and to cover all of the holes required in the foundation. This templet may be made of one inch by six inch boards, as shown by diagram, Fig. 13, securely fastened together, and permanently suspended from the roof overhead. From this templet may hang at all times plumb-bobs over the center of each bolt-hole.

When the brick laying is commenced above the bottom stone short, wooden boxes, three inches square and about eighteen inches long, may be used to build the brick around, to form the bolt-holes, pulling the boxes up and centering them anew as the work progresses.

Another plan to locate the position of bolt-holes in a foundation is first to establish the center line of shaft on the side walls of the buildings, by securing targets made of ⅞

boards six inches wide, and upon this plainly mark the position of shaft, relatively to walls of building. Upon the end walls of building

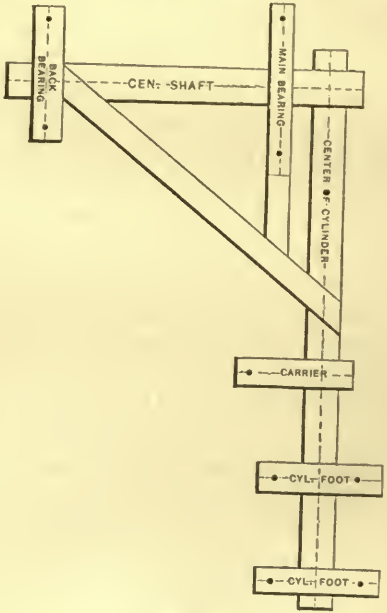


FIG. 13.

fasten similar targets, one to represent the center line of engine and one the center line of

back bearing. These two centers, of course, being exactly at right angles to that representing center line of main shaft.

From this center line of engine, measure off on the wall at each end of the building the distance each way from this center that is required for the cylinder-foot bolts, and also, from this same center line of engine measure off the offset for the main bearing. On the sides of the building we measure off an equal distance each way from the line established for the center of the shaft, equal to that required for the holes in the main and back bearings.

Also, from center of shaft we measure off the distance required to the first cylinder-foot bolt, and the distance between center of front and back cylinder bolts.

From these different positions on target, where, it is assumed, nails are driven, we stretch stout lines, and at their intersection with their corresponding line we hang plumb-bobs, from which we may work, throughout the whole height of foundation, for centering the piece of joist, or box, which we use to form the bolt-holes, a method already referred to.

CHAPTER XIII.

This plan will look something like Fig. 14. Of course these cross lines should be placed

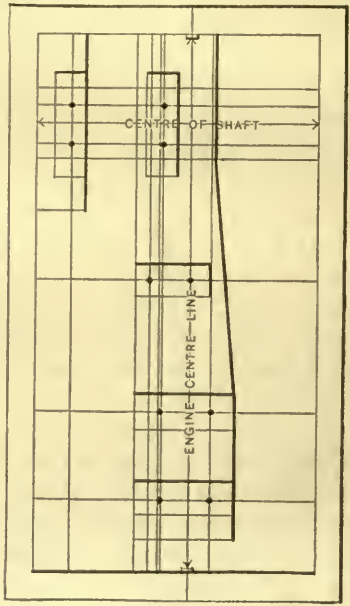


FIG. 14.

high enough to be out of the way in walking round. If the location is such as to make it

inconvenient to carry out this plan, a substitute for the walls of the building may be had by building a fence, as it were, around the space to be occupied by the foundation, and upon which the various center lines may be marked and lines stretched accordingly, from side to side thereon. This plan is much more convenient, requiring less strength of line (which is an advantage) than when long and wide engine-rooms are to be met with.

In bedding the top as well as the bottom stones down, in cement, it is a good plan to first lower the stone to its place to establish its level, and be centered each way, and afterwards hoisted up about three inches, and blocks put under each corner at that height.

After mixing the quantity of cement required of the proper consistency, the operation of placing it under the stones, and leveling off the whole surface, should be done as expeditiously as possible, and the stone lowered into position. A heavy wooden mall or piece of joist should be used endwise to bring it to a bearing and approximate level.

In setting the top stones it is well to leave them about $\frac{3}{16}$ of an inch high, so as to allow

"bushing" down to a proper level all around, when the engine comes to be located.

Above all things, in building an engine foundation, give no heed to the person who suggests economizing in the material to be used in it. It is very natural to suggest lime-mortar in place of cement, but if done, we shall have about as poor a foundation for the purpose as could be conveniently built.

Aside from the extremely long time required for mortar to dry out, it will not stand the strain and jar to which it is subjected without cracking after a time, as the sand is not nearly so firm with lime as with cement-mortar. The difference in the benefit to be derived by the use of the best quality of cement will not allow, for a moment, a comparison between the two materials. Instances may be cited where engine foundations have been built in this manner (with lime-mortar) that became a source of annoyance and trouble when strain was put upon the holding-down bolts, by the mass giving way and settling; so that it was difficult to tell where the engine was, as to line. Resort had to be made to shims placed here and there, between the engine and stone,

to make up for the deficiency caused by the stone taking a different level when the engine was screwed down.

This is the result of trying to save the difference between the cost of lime and cement, a very small item where so much depends upon the work for good running engines.

As we approach the top of foundation, pieces of joist 4″ by 6″, should be built into the brickwork at regular intervals, where most convenient, to which we may fasten the floor-planks. And at the side of foundation, spaces should be left to receive the ends of the floor-beams, for the engine-builder is the most proper person to say where these floor-beams should be set so as to be least in the way of the exhaust and condenser pipes. These should be located from the plan furnished with that object in view, so as to readily admit making anew any joint about the engine coming under the floor.

DRESSING DOWN TOP OF FOUNDATION.

After allowing sufficient time for the foundation to season and dry out, we may commence the operation of "bushing" the tops of foundation-stones to bring them to the proper level at the places covered by the machinery. To

do this properly we should provide ourselves with a three-foot and a fourteen-foot straight-edge, and a good spirit-level. After satisfying ourselves as to the lowest place in the different stones that is to be covered by the bearing surface of our machine, we use that as a starting-point, from which we are to determine the proper level for the remaining portion of the surfaces of the different stones in top of foundation.

This is best done by providing short blocks, say two inches square, upon which, at different points, we rest the ends of our straight-edge, one block being located at the starting-point, while the other is set at that part of the stone to be worked upon, dressing down with the bushhammer until we reach the proper level.

After a number of spots at the proper level have been established upon each stone, we may work off the surplus between these places to the best advantage, and finally, smooth down the work to a bearing by a judicious use of the bushhammer, using as a guide our short straight-edge, having one edge rubbed with red chalk, which being applied to the stone determines the high places that are to be removed.

MANAGEMENT OF THE CORLISS ENGINE.

CHAPTER I.

The cotton manufacturing company, by whom I have been employed for the past eight years, is located on one of the New England rivers, which furnishes very good water for steam purposes. We have one large battery of boilers in which the pressure is 100 pounds for the simple condensing engines, and in line with these there is a battery of six boilers, in which the usual pressure is 125 pounds for the compound engine.

All of these boilers are return tubular, with overhanging fronts, and are 60 in. diameter, with 76 tubes 3¼ in. diameter and 21 feet long. The distance from grate to shell of boiler is 24 in.

I am particular to give these dimensions for the reason that I have found them very good proportions, with the exception of distance from grate to boiler, which should be something more, on account of the gaseous nature of the Cumberland coal that we use.

We have a small independent steam pump

connected to tanks, for weighing feed water, and other instruments for testing the evaporative efficiency of the boilers, and trying different kinds of fuel.

Numerous tests have proved that the best method of burning bituminous coal is the one mentioned in *The Engineer* of October 18, 1890, p. 90, "*The Smoke Nuisance*," as the slow-firing method. I allow but three shovels full of coal to each furnace door at a time, and alternate from right to left hand door at each firing. This does not prevent all smoke by any means, but there is not an objectionable amount appearing at top of the chimneys at any time.

The method of coking the coal in front of the fire, I have found very objectionable when tried in connection with an evaporative test. The fireman cannot see the body of his fire and there are sure to be thin places where cold air rushes through ; and besides this it is very destructive to the front of the furnace.

Leaving the fire door open a little for a few moments after each firing reduces the amount of smoke, but gives a lower result of water evaporated on a week's trial. I have also carefully tried introducing air through perfor-

ated plates in the bridge wall. This appeared to promise success, judging by the appearance of the fire as seen through sight-holes on back end of boiler, but I found by my testing apparatus that this view of the fire was very misleading.

The best results will be found with moderately thick fires, and while burning from 12 to 13 pounds of coal per square foot of grate per hour.

I have met many engineers who seem to think that slow combustion means more perfect combustion. I have found the reverse to be true. A high firebox temperature, a clean boiler, and low temperature of uptake, is what is wanted.

All engineers who possibly can should have apparatus for weighing feed water and making evaporative tests, as the amount of steam used for power and other purposes is liable to considerable variation, and the weight of coal alone does not always give a correct result.

By careful trials one can readily find out what is best in any given case, and will undoubtedly find the unexpected sometimes, and thereby make a handsome saving in the coal

bills. These tests should not be less than one week in duration.

I have found it best to leave the air passages in furnace doors open at all times, while burning bituminous coal. The draught is much better regulated by an automatic damper regulator than by hand.

My experience with shaking grates has not been very favorable. By a careful trial of one week, of one of the best rocking grates in the market, I failed to find any gain in economy of coal; in fact a slight reduction in pounds of feed water evaporated. This was probably owing to the reduced grate surface, as the maker put dead plates three inches wide on sides and back of furnace.

We have for a number of years used plain grates in sections about 6 inches wide and 21 inches long, air opening ½ inch, width of metal bars ½ inch. I see no reason why they should not last ten years.

In boiler rooms where there is shafting, the Davis double plunger feed pump has proved to be very reliable and durable; 6 inch plungers should not be run over 25 strokes per minute. I would say avoid single plunger pumps, both

power and steam, on account of the pulsations of the water in the pipes, and especially so if you have a fuel economizer or large feed water heater, as all the water contained in these must come to rest at each change of stroke, and must be made to move on again as the plunger advances, while the duplex pump keeps the water steadily advancing all the time.

The feed pipe should not be less than two inches in diameter, and is best when made of brass, and introduced into the boiler on the top at back end over the steam space. Where the draught is sufficient the fuel economizers, so-called, are true to name. The reduction in draught will amount to about 5-10oths of an inch of water, and at this mill heats the feed water from 140 to 220 degrees.

All valves on water pipes should be of the straight-way pattern. I use a gate-valve for both steam and water, and provide all steam valves over 3 inches with a ¾ inch by-pass, by which steam can be let into the pipe without shock, which is very destructive sometimes.

The following is a copy of one of the many

tests that I have made : There was no special preparation made for this trial, and no allowance made for moisture in the coal. The coal includes that used to get up steam from cold water and banking fires :

Duration of test	2 days.
Boiler pressure in pounds	101
Temperature of feed in degrees	130
Temperature escaping gases	392
Water evaporated in pounds	83,034
Coal consumed in pounds	8,397
Weight of combustible in pounds	7,891
Coal per square foot of grate per hour	13.1
Water evap. per pound coal, (actual conditions)	9,888
Water evap. per pound combustible	10,522
Water evap. per pound coal from and at 212	11,173
Water evap. per pound combustible from and at 212	11,889
Percentage of water in steam	0.5

CHAPTER II.

In the las. chapter nothing was said about the quality of steam produced. This is a point too frequently overlooked by engineers. If you have upright boilers which moderately superheat the steam, you are all right in this respect, but with horizontal tubulars there is a strong probability of there being too much

moisture in the steam. This can be determined by the calorimeter test, which has been fully explained in engineering papers. I do not consider that high evaporative results from horizontal boilers are of any value, unless accompanied with the calorimeter test, showing that the steam produced is quite dry.

This tendency of horizontal boilers to furnish wet steam is the reason why I prefer well designed uprights when high steam is required for compound engines. One gauge of water while engines are running, and continuous feed, will prevent an excess of water passing away with the steam in boilers of good design. I depend entirely on gauge cocks, having no glass gauges whatever. These boilers in my charge are over 10 years old, and, all told, there are over 22 hundred tubes; not one of them has ever leaked from any cause.

We are all aware that properly covered pipes contribute largely to the economic results. I have found that hair felt, with asbestos mill board next to the pipe, makes a good covering, but for a short time only. The asbestos does not prevent the felt from burning on top of pipe; in 6 months it will be found nearly,

if not quite, destroyed, and on the bottom it will be hanging loose from the pipe. On flanged pipe I use one inch each of Nos. 1 and 2 asbestos cement, and outside of this, one inch of hair felt. This brings the covering out even with the flanges; then all is covered with heavy cotton cloth, and for the boiler room it is whitewashed; in the engine rooms it is covered with a jacket of Russia iron, and brass bands to break joints.

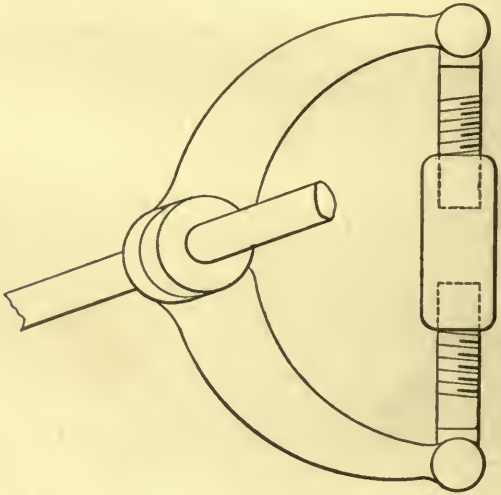
In regard to the size of steam pipe to the engine, I would say: Do not be governed by the capacity of opening into the cylinder or the valve furnished by the builders. The high piston speeds now so common require larger pipes, and if they are long, considerably larger diameter should be used. The Corliss cross compound engine, of which I am about to write, has cylinders 22″ + 40″ by 60″ stroke, and runs 60 revolutions per minute. The throttle is 7″ and the steam pipe about 200 feet long. I enlarged from the throttle with a short piece of cast-iron pipe up to 10″, and the balance is made of 10″ boiler tubing in long lengths, with heavy cast-iron flanges riveted on. All bends are made of copper. Such a

steam pipe as this is far more reliable for high pressures than cast-iron or common wrought pipe with screwed joints.

The cylinders of this engine are proportioned in this way on account of the unusual amount of steam required for use in the mills, which is taken from the receiver. Besides seven slashers (which turn off something over ninety thousand pounds of yarn each week), there are eighty vapor-pots in the weaving rooms, all of which are supplied from this source. The usual pressure in receiver is five pounds. This engine has been running two years, and is connected to ten water-wheels of 180 h. p. each. There are no regulators on the water-wheels, the engine governor controlling the speed of all. From this and other causes, the load is quite variable, and with full river the engine is sometimes underloaded.

As left by the builders, I at first had trouble to maintain an even pressure in the receiver, and furnish steam for use in the mills, as the point of release on both cylinders was determined by the governor, and no way was provided to change the cut-off on l.p. cylinder. This would be correct under certain conditions,

but not so here, as the amount of steam taken from the receiver is subject to considerable variation, and this, together with changes in power required, made it necessary to make an alteration. The regulator shaft connecting one



cylinder with the other was cut in two and connected as shown by sketch.

By turning the nut between the levers to the right or left, the l.p. cylinder is made to take

more or less steam, as the case may be, and at the same time it is controlled by the regulator.

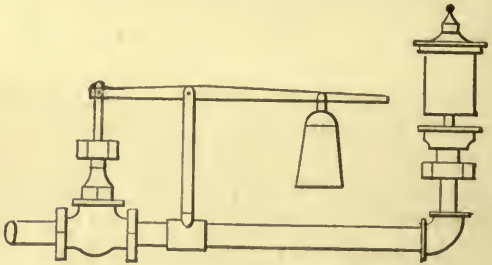
During a large part of the time the l.p. cylinder is made to cut off somewhat earlier than would be the case if a low steam consumption by this engine alone was desirable. It is the total coal used that interests the stockholders. During the season of low river, when other engines are furnishing the steam for use in the mills, and this engine is running under favorable conditions, the coal consumed during a test of one week, including that used for banking fires at night, was 1.73 lbs. per h.p. per hour.

The exhaust steam from the duplex feed pump on this trial was used to heat the feed water for another set of boilers than those from which the engine steam was taken. As running at present—with an overflowing river, and steam taken from the receiver for the mills, as mentioned above—the coal used per h.p. per hour varies from 1.9 to 2.1 lbs.

The variations in load sometimes caused the receiver pressure to fall quite low, unnoticed by the engineer. This made bad work

and loss in the slasher room. To prevent this loss, and give us timely notice, I connected a small whistle to pipe leading to slashers, as shown by sketch.

A pressure of 5 pounds holds the valve down to its seat, but at 4½ pounds the weight lifts the valve, and blows the whistle before there is any cause for complaint in the mill, and the



engineer has time to make the proper adjustment of the cut-off on the l.p. cylinder, or open the by-pass valve into receiver.

The air pump is the regular Corliss pattern, 34 in. by 12 in. stroke. A part of the overflow, with the water of condensation from the receiver, is returned to boiler room. A

¾ in. pipe from the overflow leads up to the engine room, into which a thermometer is inserted, and returns below to condenser, thus showing temperature of overflow water at all times.

A small pipe, admitting a little air into the channel way, effectually stops the pound that is so common with this kind of air pump.

As the machinery in the mills is run at as high speed as possible, it is very important that the engine regulation should be as perfect as it can be made.

This engine should make a full revolution each second; if it should fall short of $\frac{1}{100}$ of a revolution in each second throughout the day there would be a very perceptible diminution in product. If it should fall short this much during an hour it is so much lost; it should not be made up in the next hour.

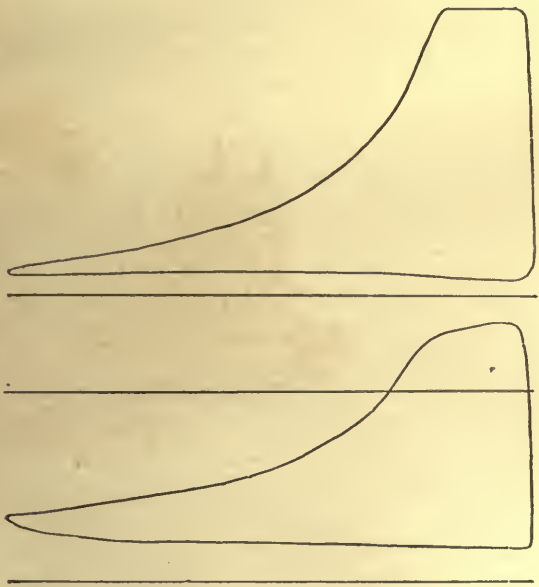
The Corliss governor, unaided, will not regulate as closely as it is desirable to run in a cotton mill.

I have found the Gale governor attachment to be a great help, but even this useful and neat-looking little device needs some attention if a strictly uniform rate of speed is desired,

I suppose the Moscrop recorder is an excellent thing to show the variations in the speed. A good many years ago, before I ever heard of a "Moscrop," or other device for this purpose, I made a little machine, which is still doing splendid service. On the shaft of the air pump rocker is fixed a small adjustable arm carrying a ratchet, which picks one tooth at each revolution of the engine in a ratchet wheel of an equal number of teeth, with revolutions desired by the engine in one minute. On the shaft of this ratchet wheel there is a pair of mitre gears, by which this movement is taken up into the engine room, where, at a convenient place, there is a clock dial with suitable train of gears to run a minute and "second hand."

After the engine is started, the hands on this dial are set to exactly agree with the engineer's time. After an hour's run, if there is a gain or loss of one second, which means with this engine one revolution, it is easily detected and corrected. This dial makes no record of the revolutions for yesterday or last week, but as it enables us to run just right all the time; it is very satisfactory. The engine

register or counter shows each day the number of revolutions, and this is recorded for each engine, with all other matter of interest, in a logbook kept for this purpose.



These cards were taken November 4. There was not sufficient load for the engine to show

its best work. The cards from the other ends are as nearly the same as can be. Springs in indicators, 60 and 12 to the inch.

CHAPTER III.

The compound from which the indicator cards in the last chapter were taken is said to be Mr. Corliss's final and perfected engine. It certainly is very economical in the use of steam, and many parts are excellent in design, but there are others which are not quite perfect.

There is no means provided to raise the pistons as they wear below the center; the steam packing rings are let into the head, there being no bull-ring. The piston can be turned around on the rod, which will make up for the wear on the packing rings, but not for the wear of the cylinders. The slides are very much too narrow; there is not sufficient bearing surface for heavy loads. The pillow blocks are all that could be desired, 24 in. length of bearing for an 11 in. shaft. The hand hole in the side, which enables the engineer to discover any heating before it has pervaded the whole mass, is a very good thing. The shaft

is 15 in. in diameter in the center, and carries a 25 ft. wheel, weighing fifty thousand pounds.

The valve gear is very peculiar, and different from any other Corliss engines made. The wrist plates are nearly as large as the sides of the cylinders, and the valve connections are, consequently, very short; this gives the valve a very quick movement. The vacuum dash-pots close the valves very quickly, as will be seen by the cards. The piston rods have U. S. metallic packings, and the valve rods are packed with Garlock's patent sectional rings.

This engine has run two years, and has been stopped but once during working hours from causes arising in engine room. One of the adjusting screws on the cross head gibs worked loose soon after the engine was left by the builders, and it was necessary to stop. I have put preventers on all of these screws, as I found that they had a tendency to work loose.

I saw an item in the daily papers of an engine of this type running away and breaking up badly; the question has been asked, why the governor did not take care of it? I should say that the cut-off rods were so set that when the governor balls were at their extreme height,

the low-pressure cylinder would still take some steam as long as there was any in the receiver, and there would be enough (with the vacuum) to do mischief.

There are some features about another engine, in my care, which may possibly be of interest to engineers, who may have similar conditions.

This is a double high-pressure condensing engine, having cylinders 26 in. by 60 in. stroke, running 58 revolutions per minute; built at the Corliss Steam Engine Co.'s Works, in 1881. It can be run ¼, ½, ¾, or all condensing.

During a few weeks of very low river in the summer we run all condensing, in order to get full power, sometimes as much as 950 h.p. I have never been able to test the steam consumption with this load, but it is not of these conditions that I am about to write.

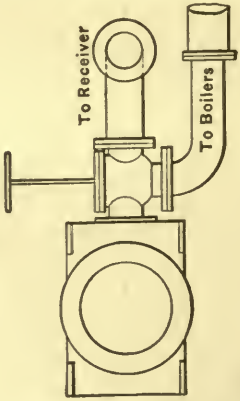
This engine runs in connection with six waterwheels of 180 h.p. each. Several years ago, during the season of high water, this engine was disconnected from the mill, the waterwheels doing all the work, the steam for the slashers and vapor for weaving rooms being furnished direct from the boilers. As there

was not sufficient power in the wheels to insure good regulation, there was a diminished product in the mill. The slashers and vapor-pots were then piped for exhaust steam, and one cylinder only of the engine was run non-condensing, with five pounds back pressure. Sufficient water was taken from the water-wheels to give the engine a load of 100 to 150 h.p. The result was very satisfactory, perfect regulation and full product in the mill, and less coal consumed than was required to do the mill work with direct steam. The gates on the wheels were so regulated that there was very little, if any, steam escaped through the back pressure valve.

For the past two years, owing to the enlargement of the mill, the load for one cylinder was such that there was about twice as much exhaust steam as the mill required. Besides this loss of steam, there were many days during the last part of the afternoon when the river was somewhat low, that the load was far too much for one cylinder, while in the morning one cylinder was ample. I then compounded this engine as shown by sketch.

A piece of 20 in. pipe was placed in the ex-

haust from the high-pressure cylinder, which answers for a receiver ; from this we piped to the two-way valve on top of low-pressure cylinder. The throttle that was taken from this cylinder was placed in this new pipe. For more than two years this engine has run

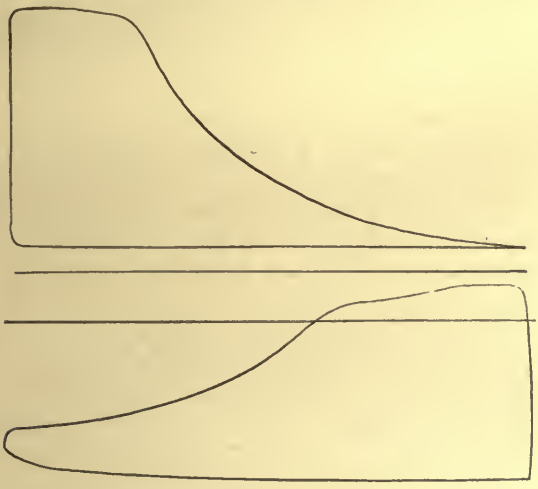


with this arrangement, giving perfect satisfaction.

Besides running very economically with light load, one important advantage is that I always have the low-pressure cylinder connected, and ready for heavy loads, and the change is made

from compound to simple condensing, while running, and without any perceptible change in the speed.

The amount of power derived from the low-pressure cylinder is quite small, from 60 to 90



h.p., but this is obtained from steam that would otherwise be thrown away, and the 5 pounds back pressure must be maintained. When running compound, with load of 400 h.p. or under,

and steam taken from the receiver for four slashers and twenty vapor-pots, the coal required is 2.25 pounds per h.p. per hour. With one cylinder non-condensing, and furnishing the same amount of steam for use in the mill, and the other cylinder condensing, the coal required is 2.69 lbs. per h.p. per hour. The cards shown were taken November 8, with light load. The springs used were 40 and 12 to the inch. I am aware that these are not perfect cards, but I think it will be generally admitted that the result, taken as a whole, is very good.

As to setting Corliss' valves, I can add nothing to the full and correct article published in *The Engineer*, by John T. Henthorn, mechanical engineer, of Providence.

When starting, as soon as the engine has made two or three revolutions, I raise the governor a little, by a thumb nut, in order to make the cut-off operate. This diminishes the liability to take water from the boilers and greatly aids the engineer in getting a vacuum.

The noise from the dashpots was somewhat harsh, and this was remedied by bolting a small box-shaped casting over the outlet holes, and from this running a 1¼ in. pipe, with cock in

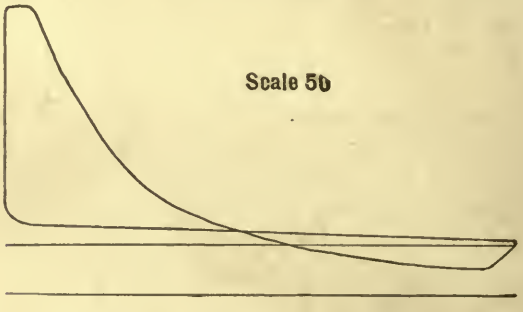
it below the floor. All the principal parts of the engines in my care are oiled by stationary sight feed cups. The cylinder oil is introduced at each end of the cylinders. The oil for other parts I usually mix 3 parts paraffine to one of sperm, and one of neatsfoot. This mixture costs 32 cents per gallon, and will run the heaviest shaft and not gum.

Usually a very small quantity of oil is sufficient for crank-pins and cross-head wrists, but my experience has taught me that main bearings should have a very generous quantity regularly applied, carefully collected in drip pans, and strained and used again. With an experience of over thirty years, I have never stopped an engine for a hot pillow-block. The cost of all the oil for the 26 in. double engine mentioned, including two jack shafts, is twenty-seven cents per day. The compound engine requires about the same quantity. A pair of 23in. × 60in. engines, running 66 revolutions per minute, are oiled for seventeen cents per day.

Some experience with the indicator, together with my observation of the running of many non-condensing Corliss engines, leads me to

the opinion that far more than one-half of these engines are running under-loaded. To speak more correctly, I would say that the majority of engineers carry a higher pressure of steam than their work calls for.

This may be equally true of other automatic engines, but I have found it so frequently the case in Corliss engine rooms, and so difficult



to convince engineers of their error, that I mention it here. The first card taken is generally like the one I have constructed for the purpose of illustration. It will be seen that the fly-wheel is driving the piston during one-half of the time. The pressure of the steam on the piston is like a series of kicks, but it is not

best to let it kick so hard that it will kick back. The strain on the crank pin is greater, and the valves work harder while running in this way.

CHAPTER IV.

Another very important reason why the engines should not be operated in this manner, is the greatly increased cylinder condensation, owing to too great range of temperatures.

The difference between the temperature of the initial pressure of this diagram and the absolute terminal pressure is about 140 degrees; whereas if the steam had been lower, say 60 pounds total, and the terminal about the same as the back pressure, the difference in the temperatures would have been only 80 degrees. It will be seen that there is a difference of 60 degrees in favor of running with the lower steam pressure; nearly as much as there is between a winter and a summer day.

I do not wish to be understood as advocating low pressures, but would say—do not allow a loop in the card while the engine is doing regular work. This may appear to many as an extreme case, but I have frequently

found the loop in the card from engines which were represented by engineers as heavily loaded.

There are a great many old Corliss engines that have the packing rings set out by set screws against elliptic springs. I would advise engineers who can't get anything better than this, to be extremely careful when setting out these screws. The best way is to stop the engine on the front centre, disconnect the main rod, and pull the piston to the back end ; then it can be ascertained by moving the piston back and forth when it is just right. It should be so easy that one man can readily move a 20 inch piston in the cylinder.

The Corliss engines of later make have a single packing ring, in small sections, let into a junk ring. This is a great improvement over the other, as it requires no setting out, and runs very satisfactorily; but judgment should be used in setting up the screws for the purpose of centering the piston head. These are frequently set up so hard as to strain the junk ring out of round. If the junk ring nearly fills the bore of the cylinder (as it should), this strain causes it to bear very hard in these four

places, causing undue friction, and sometimes injury.

The follower of this kind of piston is usually secured by large steel bolts. I have no doubt that many engineers have had serious trouble in starting these bolts. I remedy this by coating the bolts with a mixture of Dixon's graphite and cylinder oil. This mixture is just the thing for pipe and other bolts about boilers, which are sure to stick unless something of this kind is used.

We are frequently told by the newspapers that a very large proportion of the accidents in engine and boiler rooms are the result of carelessness, and right here they stop and leave the general reader with the impression that it is the engineer in charge who is responsible. Carelessness and ignorance on the part of builders, and indifference on the part of owners, cause the greater part of the accidents. The engineer is generally expected to get along with what he has, and very frequently cannot clean his boilers for lack of opportunity. I find in a copy of *The Locomotive* the monthly report of the inspectors of the Hartford Steam Boiler Insurance Co.,

the following items : "Cases of defective riveting, 1,658 ; cases of deficiency of water, 6."

Engineers and their assistants were faithfully attending to their boilers that month, and all the reports show about the same proportion.

It is very much the same in the engine room, but we have no insurance company to publish records of defects that may exist there. If we had, there would be a large number of Corliss engines which would have to be stopped and have larger piston rods put in. There are a great many of these engines running that were built some twenty-five years ago (designed for a mean pressure of 30 pounds and a piston speed of about 350 feet), that have been speeded up and the pressure increased. When one of these engines breaks down they usually send to the shop; the wise man comes and tells the manager that there was too much water in the boiler, and that the engine was blown up by too much water ; now the boiler is blown up by too little water, while it is very rarely the case that there is a grain of truth in either statement.

Boilers that are not strong enough to withstand the pressure explode, and overloaded en-

gines of faulty design and poor material are liable to accident. while a really first-class man is doing all in his power to prevent it.

The older engineers of the country will remember when the original builders of the Corliss engine used cast-iron main shafts, and the numerous breaks caused by them; and still we have cast iron cranks, cross-heads and valve arms. The spring of the cast-iron crank on a full loaded engine is quite appalling. Put the engine on the center, and work the valves by hand, and measure this deflection. Certainly, a stiffer crank is needed.

The valve arms seldom break if the valves are properly oiled. Oiling the cylinder three or four strokes with the oil pump every ten or fifteen minutes will require nearly one gallon of oil per day, and still the valves will be dry nearly half the time. The Corliss oil-pump discharges one cubic inch at each stroke, and that single stroke does not oil the cylinder and valves any better than one drop; therefore it is much better to have good automatic sight-feed cups for this purpose.

The cast-iron cross-heads are seldom sound, and the manner in which they are supported is

very faulty. The latest patterns correct this, but there are a great many of the old style in use that can be improved by the engineer in the manner shown by the illustration.

The nut under the cross head in Fig. 1 will not hold the cross head firmly, and long con-

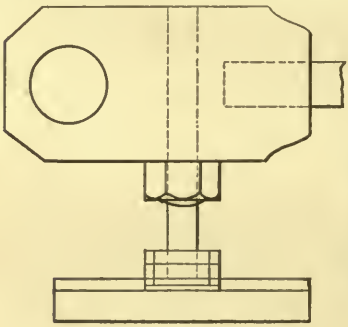


Fig. 1

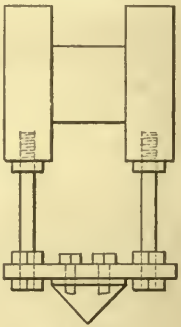


Fig. 2

tinued working will be likely to break the piston rod. This movement of the cross head at the outer end cannot be seen, but it is there all the same. The support given by the bolts from the V, as shown in Fig. 2, are adjustable, and remedy this trouble.

APPENDIX.

CHAPTER I.

PROPORTIONS.

The pressure on bearings is figured as the projected area and equals the length multiplied by the diameter.

Main bearings	..	350 lbs. per sq. in.
Crank pins	..	450 ,, ,, ,,
Guide blocks	..	150 ,, ,, ,,
Crosshead pins	..	900 ,, ,, ,,

The diameter of fly-wheels should be about four times the stroke of the engine.

The speed of fly-wheels should never exceed a circumferential velocity of 4,500 feet per minute.

The average piston speed for Corliss engines should not exceed 500 ft. per minute.

The following tables will be found useful for reference.

Horse-Power and General Dimensions

Size of Cylinder.	Indicate H. P. for 80 lbs. Boiler Pr. Cutting off at 0.2	80 lbs. Boiler Pr. Cutting off at 0.3	90 lbs. Boiler Pr. Cutting off at 0.2	90 lbs. Boiler Pr. Cutting off at 0.3	Number of Rev. per Minute.	Band Fly Wheel. Dia. ft.	Face. ins.	Weight. lbs.	Shaft. Dia. of Crank Shaft.	Centre of Shaft above Foundation Stone.	Length of Shaft from Centre Line of Cylinder to End. in feet.	Distance from Centre of Crank Shaft to back end of Cylinder Head.	Steam Pipe Dia. ins.	Exhaust Pipe Dia. ins.	Total Weight of Engine. lbs.
10 x 30	41	56	48	64	100	9	12	5,000	5	22	7.6	13.0	3	5	15,000
12 x 30	55	76	65	87	93	10	16	7,000	6	22	8.0	13 0	3½	5	16,500
12 x 36	64	88	75	101	90	10	18	9,000	6½	24	8 0	13 6	3½	5	18,000
14 x 36	82	100	96	128	85	11	20	10,000	7	24	8.0	14.0	4	5	20,000
14 x 42	93	113	109	147	80	11	20	10,000	7½	24	8 0	14.0	4	5	22,000
16 x 30	117	137	123	166	90	12	22	12,000	8	24	8 6	14.0	4	5	24,000
16 x 36	129	149	138	185	85	12	22	12,000	8½	26	8.6	14 0	5	6	26,000
16 x 48	149	168	151	204	80	14	24	18,000	8½	26	9 0	13.0	5	6	27,500
18 x 36	164	184	162	219	85	16	24	18,000	9	28	9.0	14 0	6	7	28,000
18 x 12	170	194	174	235	80	16	30	20,000	9½	28	9.6	13.0	6	7	32,000
18 x 48	180	210	185	265	70	18	30	22,000	9½	28	10.0	14 0	6	7	36,000
20 x 42	185	200	206	281	80	18	36	24,000	10	30	10.0	14.0	7	8	38,000
20 x 48	195	215	228	297	70	20	38	30,000	10½	30	10.0	13.0	7	8	39,000
22 x 42	230	250	247	330	80	20	38	30,000	11	30	12.0	14 0	8	10	42.000
22 x 48	240	260	267	360	70	20	38	30,000	11½	32	12.0	14.0	8	10	48,000
22 x 60	265	283	302	405	65	22	42	32,000	11½	32	12.0	13.0	8	10	52,000
24 x 48	265	280	311	416	70	22	42	32,000	12	34	13 0	13 0	8	10	58,000
24 x 60	308	328	360	483	65	22	42	32,000	12½	34	13.0	13.0	10	12	72,000
26 x 48	311	330	365	490	70	24	48	34,000	13	36	14 0	13.0	10	12	74,000
26 x 60	356	376	390	559	65	24	48	34,000	13½	36	14.0	14.0	12	14	77,000
28 x 48	356	376	435	580	70	22	42	32,000	14	36	14.0	14 0	12	14	80,000
28 x 60	420	450	460	600	65	24	48	34,000	14½	36	15.0	15.0	14	16	85,000
30 x 48	450	476	480	650	70	24	50	35,000	15	36	14.0	14 0	14	16	90 000
30 x 60	500	550	546	735	63	24	56	40,000	15	36	15.0	15 0	14	16	98 000

The Principal Dimensions of Corliss Engines.

		in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
Stroke in Inches		30	30	36	36	42	42	42	48	48	60	60
Diameter of Cylinder		10	12	14	16	18	20	22	24	26	28	30
Main Bearing	Diameter.	4 1/16	5 1/16	6 1/16	7 1/16	8 1/16	9 1/16	10 1/16	11 1/16	12 1/16	13 1/16	14 1/16
	Length.	10	12	14	16	18	20	20	22	22	26	26
Steam Pipe	Diameter.	3	3 1/2	4	4 1/2	5	6	6	7	7	8	9
Exhaust Pipe	Diameter.	3 1/2	4	5	6	6	7	8	9	9	9	10
Steam Ports	Width	0 3/4	0 3/4	0 13/16	0 7/8	0 15/16	1 1/16	1 1/8	1 3/16	1 3/8	1 1/2	1 5/16
	Length.	9	11	12 1/2	14 1/2	16 1/2	18 1/2	21	23	25	27	29
Exhaust Ports	Width	1 3/16	1 3/16	1 5/16	1 7/16	1 9/16	1 3/4	1 7/8	2	2 1/8	2 5/16	2 1/2
	Length	9	11	12 1/2	14 1/2	16 1/2	18 1/2	21	23	25	27	29
Crank Pin	Diameter.	2 7/16	2 15/16	3 7/16	3 15/16	4 7/16	4 15/16	5 7/16	5 15/16	5 15/16	6 7/16	7 7/16
	Length.	3 1/4	3 1/4	3 3/4	4 3/8	5	5 3/8	5 3/8	7 7/8	7 7/8	9	9
Crosshead Pin	Diameter.	2 7/16	2 15/16	2 15/16	3 15/16	3 15/16	4 15/16	4 15/16	5 15/16	5 15/16	6 7/16	6 7/16
	Length.	3 3/4	3 3/4	3 3/4	4 3/8	4 3/4	5 3/8	5 3/8	5 7/8	5 7/8	6 3/8	6 3/8
Valve Chamber	Diameter.	3 1/2	3 3/4	4 1/8	4 1/2	5	5 1/2	6	6 1/2	7	7 1/2	8 1/2
Valve Stem	Diameter.	1 3/16	1 5/16	1 5/16	1 7/16	1 9/16	1 11/16	1 11/16	1 13/16	1 15/16	2 3/16	2 3/16
Piston Rod	Diameter.	1 15/16	2 3/16	2 7/16	2 15/16	3 3/16	3 7/16	3 15/16	4 3/16	4 7/16	4 5/8	4 3/4

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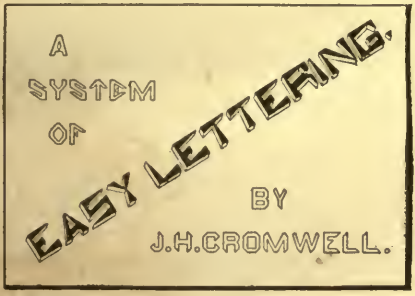
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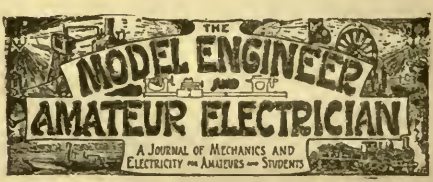
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