

will introduce into the main sewer quantities far above those just calculated, especially in the branch Bab-Azoun. But the flushing will have a great effect in hot weather and times of low water.

We may have an idea of this by comparing the delivery of the three branches with the quantity introduced by the flushing. I have not as yet gauged the quantity passed in sewer during the low stage, but it may be approximately calculated as follows :

The volume entering the city in summer is about 3924 cub. yds. for twenty-four hours, or, say, 7.7 Imp. galls. per second. We may admit that the loss from consumption, evaporation, &c., is compensated by the quantity derived from wells and cisterns, and adopt 7.7 galls. as representing the combined delivery of all the sewers of Algiers, in the low stage.

To make a just distribution of these 7.7 galls. among the three sewers, we must take as a basis not the general basins of each branch, but only the inhabited portion of each basin, for the uninhabited portions do not feed the sewers, since there are no rains, and because the ravines are dry at that time. Taking the inhabited parts of the basins, we have the following approximate surfaces :

	Acres.
Branch Bab-Azoun,	88.9
“ Bab-el-Oued,	44.4
“ Marine,	14.8
Sewer emptying directly outside of port,	14.8
	<hr/>
	162.9

Dividing the 7.7 galls. proportionately to these surfaces, we have for the delivery of each branch in the low stage, the following quantities per second :

	Cub. yds.
Branch Bab-Azoun,	272
“ Bab-el-Oued,	179
“ Marine,	64

During the flushing, these volumes will be each increased by 1700 cub. yds.; that is to say, that during this time, the volume of the branch Bab-Azoun will be multiplied by 7.25; that of the branch Bab-el-Oued, by 10.50; and that of the branch Marine, by 27.50. These results prove the advantage of flushing in hot weather.

It will be well to reduce the period of flushing in the branches Bab-el-Oued and Marine, or what is equivalent, to flush more frequently in the branch Bab-Azoun.

For the Journal of the Franklin Institute.

Upon Balancing Horizontal Direct-acting Screw Engines.

By ALBAN C. STIMERS.

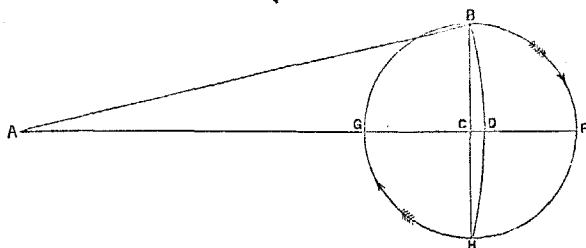
Great attention has been paid by engineers during the past few years to the subject of properly balancing *horizontal direct-acting screw engines*. The parts considered necessary to be balanced in this description of engine, are those which tend by their gravity to depress the crank-pin when the piston is at either end of its stroke, and, ordinarily, this is equal to the weight of the crank-pin, half the weight of the connecting-rod, and half the weight of the two cranks.

The practice of engineers has not materially differed about the amount of counter-balance which is required in each case, it being usual to apply sufficient weight upon the side of the shaft directly opposite the cranks to prevent the engine, when in a state of rest, having any tendency to move in either direction when the piston is at the end of the stroke. The improvements which have been made, have been in placing this counter-balance in the most favorable position.

The object of this paper is to show that an engine perfectly balanced

when in a state of rest is not balanced when in motion, and that if it is run in one direction the weight of the usual counter-balance is too great, while if it is run in the other, it is not sufficient.

Let A B, in the following diagram, represent the connecting-rod of a steam engine, of which B C is the crank and B F G the circle in which the crank-pin moves. The diameter of this circle, or G F, represents the stroke of the piston.



Now, if from the centre, A, and the radius, A B, equal to the length of the connecting-rod, the arc B D H is described; the point D will represent the position of the piston in the stroke G F, when the crank is in the vertical position, C B. This, it will be observed, is at a sensible distance from the middle of the stroke C, and if the engine is put in motion in the direction of the arrows, the half revolution, B F H, is made by a movement of the piston equal to 2 D F, and the other half revolution H G B by a movement of the piston equal to 2 D G, and as the times and piston pressures are equal during each of these half revolutions, the piston power available to drive the crank-pin from its highest to its lowest position, is less than that available to drive it from the lowest to the highest by 4 C D multiplied into the pressure transmitted from the piston to the crank-pin; and to remove this irregularity in the powers of the two half revolutions, it is necessary to have a preponderance at the crank-pin, which may be determined in any given case

by the following formula :

$$w = \frac{p \times 2 \text{ C D}}{s}$$

where w = the preponderance required at the crank-pin, p = the total effective piston pressure, and s = the stroke of piston. Because $p \times 2 \text{ C D}$ is the amount of power which must be deducted from one-half revolution and added to the other, and if this power is obtained by adding unbalanced weight to the crank-pin, it is manifest that this weight moves through a perpendicular distance equal to the stroke of the piston, and the foot-pounds to be overcome, divided by the feet through which the weight acts, gives the number of pounds necessary.

To show that this quantity is not an insignificant one, we will take the case of the U. S. Frigate *Merrimack*, of which I happen to have the necessary data.

It may, with propriety, be premised that the connecting-rods, &c., of the engines of this vessel were above the average in weight compared to diameter of cylinder and length of stroke; that the piston pressure

was less than usual; and that the ratio of length of connecting-rod to length of stroke was fully equal to the average.

There were two horizontal back-action engines connected direct to the screw shaft.

Diameter of cylinder,	72 inches.
Stroke of piston,	3 feet.
Length of connecting-rod,	7½ "
Mean effective pressure per square inch on the piston, being the average of more than 200 indicator diagrams taken during the year 1859,	10.23 lbs.

From the above length of connecting-rod and stroke of piston, the distance *CD* is easily found by first obtaining the base *AC* of the right angled triangle *ACB*, which is $(\sqrt{7.5^2 - 1.5^2}) = 7.34847$, and then deducting this from the length of the connecting-rod, or *AC*, and we have $(7.5 - 7.34847) = 0.15153$ as the value in feet of *CD* in the engines of the *Merrimack*.

From the mean effective pressure on the piston as given above, should be deducted that which is required to overcome the friction of the piston, piston rods, crosshead guides, and connecting-rod bearings; and as the air pump pistons are connected directly with the steam pistons by a rod passing through two stuffing boxes, there is the friction of the pump piston and piston rod, and the direct pressure upon the pump piston. This last was often obtained by means of indicator diagrams from the pumps themselves, and the usual average was 8 per cent. of the total effective pressure on the steam piston. It amounted, therefore, to $(10.23 \times .08) = 0.8184$ lbs. per square inch of the steam piston, which, deducted from the total, leaves $(10.23 - 0.8184) = 9.41$ lbs., and if we assume that 1.41 lbs. per square inch of the steam piston pressure will overcome the loaded friction of the parts above enumerated, we shall probably not err much. This leaves 8 lbs. per square inch transmitted to the crank-pin, and to obtain the value of *p* in the foregoing formula the area of the piston must be multiplied by this amount. The area of a piston 72 inches diameter is 4071.5 square inches, therefore $p = 4071.5 \times 8 = 32572$ lbs., and as $s = 3$, we have

$$w = \frac{32572 \times 2 \times 0.15153}{3} = 3290 \text{ lbs.}$$

This is the preponderance which it is necessary there should exist at the crank-pin of the *Merrimack's* engines, to maintain them in a perfectly balanced state when running in the direction marked by the arrows of the foregoing diagram.

To determine now how much counterbalance there should be for running them in the direction shown, it is necessary to ascertain how much preponderance there is at the crank-pin without any balance.

The following are the weights which tend to depress the crank-pin when the piston is at the end of the stroke.

Half of one connecting-rod, complete,	1950 lbs.
Half of two cranks,	1315 "
One crank-pin,	485 "

Total, 3750 "

The counterbalance required, then, upon the side of the shaft directly opposite the cranks, and at the same distance from its centre as the crank-pin, is $(3750 - 3290 =)$ 460 lbs.

By a similar process of reasoning, it may be shown that if the engine is run in a direction contrary to that marked by the arrows, the preponderance, instead of being required at the crank-pin, must be placed on the opposite side, and that in the case of the engines of the *Merrimack*, the counterbalance would require to be $(3750 + 3290 =)$ 7040 lbs.

It so happens that the position of the two engines of this vessel is such that while the forward one runs in the direction shown by the arrows of the diagram, when the vessel is being driven ahead, the after one runs in the contrary direction.

The counterbalance for both engines is in one piece, and is placed, not only abaft the engines, but upon the after part of the loose coupling connecting the line of shafting, running out through the stern to the screw, with the crank shaft.

On account of its great distance from the forward engine it probably had very little influence upon it, and it is worthy of remark in this connection that this engine always worked much more smoothly and regularly than the after one.

It is quite the common practice now to run the engines of our screw steamers in the direction contrary to that shown by the arrows in the diagram, because the oblique action of the connecting-rod upon the crosshead guides is upward when running in this direction, whereas, when running the other way, this action is downward, the friction in the latter case being the oblique force of the rod *plus* the weight at the crosshead, and in the other direction the friction is only the *difference* between these quantities.

The engines of the *Merrimack* afford an excellent opportunity to judge of the importance of this subject, the oblique force of the connecting-rod pressing down upon the forward crosshead at the same time that it lifts the after one, and if the fears of constructing engineers were well founded, the former would heat and give trouble, while the latter would work to a charm. The reverse of this, however, was the case in fact. It was the forward crosshead that never gave trouble, while the after one required frequent attention, as it was important that there should be as little play in the guides as possible, because when a few inches of the stroke had been performed, the lifting force of the connecting-rod became sufficient to raise the crosshead against the upper guide, and when it arrived at a corresponding distance from the other end of the stroke, it would drop again to the lower slide, and it may easily be imagined that when heavy engines like those under consideration were making forty or fifty revolutions per minute, this lifting and dropping process was not a very gentle affair. The only way in which the blow could be softened was to make the distance through which the crosshead rose and fell as small as possible, and sometimes this would be reduced a trifle too much, and heating ensue as a natural consequence. Whereas the forward crosshead

always worked perfectly smooth and never gave the slightest trouble from heating or any other cause.

It appears preferable, therefore, to run the engines in the direction of the arrows for reasons entirely independent of any consideration of the counter-balance, and when to this is added the great convenience of having the engine balanced without the application of any counter-balance whatever, there should certainly be very little hesitation about it.

That to apply no counter-balance would be the proper course to pursue, is evident from an examination of the case of the *Merrimack*, which we have taken as an example, for it would only have required an increase of steam pressure upon the piston of $1\frac{1}{4}$ lbs. to have caused the engines to run in a perfectly balanced state, supposing them to be run in the direction of the arrows, and if the pressure was still more increased, there would require to be weight added to the crank-pin to maintain the balance perfect, but even in engines carrying much higher steam than those of the *Merrimack*, engineers would very properly hesitate to do this on account of the great irregularity with which the engines would run when reversed.

Steam Engineering in 1859. Recapitulation.*

Concluding Remarks.

The first paper of this series appeared in the May number, and was intended to point out the avoidable difference existing between the present practice of steam engineering, and that which, if generally adopted, would result in many and undoubtable advantages.

It was intended also to indicate that steam engineers were practically neglecting to appreciate the only true principles in steam engineering, on which all economy must rest, and that with few exceptions, we are literally wasting money and time by allowing the manufacture of steam engines to degenerate into a mere trade.

During the past six months, several instances have occurred, in which the possibility of an indicated horse power being obtained from the consumption of $2\frac{1}{2}$ lbs. of coal, has been more than verified, and these instances have, in many particulars, confirmed fully the truth of all we have stated in reference to the economy of steam power.

Paddle engines of considerable power are now regularly working with a consumption less than $2\frac{1}{2}$ lbs. of coal per hour, and this is accomplished with all the disadvantages arising from the use of salt water. This creditable result is due, solely, to an intelligent appreciation, by the designer, of those truthful principles we have endeavored to commend.

In another case of recent improvements in steam power, by an improved class of boiler, the use of distilled water, conservation of heat, and extensive development of the expansive principle, an indicated horse power has been realized by a consumption considerably less than 2 lbs. of coal per indicated horse power per hour.

*From the Lond. Artizan, Dec., 1859.