PRACTICAL MECHANISM.

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In the experiment referred to in our last, the valve had (in the first instance, when it had no lap) one sixteenth inch of lead so as to give that amount of exhaust opening when the piston was at the end of the stroke. In the second instance, however, when the valve had $\frac{5}{16}$ of steam lap added to it, it was set so as to have not more than $\frac{1}{64}$ of lead, the author being convinced that, when a valve has sufficient lap to give a moderately free exhaust, there is more to be lost by back pressure from excessive lead than to be gained by the small amount of assistance it lends towards making the exhaust more free. If a valve has no lap at all, it may with advantage be given an amount of lead that would otherwise be decidedly detrimental. It would appear that, in the early days of steam engineering, one of the advantages due to adding lap to the valve (a free exhaust) was largely attributed to the lead of the valve, since sufficient lap to cut off the steam supply when the piston has traveled three quarters or even more of its stroke will give a sufficiently free exhaust, even supposing that the valve has no lead at all.

Referring again to the advantage in economy due to using (or, as it is commonly called, working) the steam expansively, it is self-evident that, if we have steam at a gage pressure of 50 lbs. per inch, (that is, above the pressure of the atmosphere) and permit its escape at any pressure above that of the atmosphere, we shall not have extracted from it all the power it contains, because it may be used at the initial pressure of 50 lbs. per inch during a certain portion of the stroke, and, by then being permitted to expand itself before being exhausted, may be employed to perform duty as steam of 49, 48, 47, etc. lbs. per inch, and so on down to that point at which the indicating needle or hand of the steam gage will stand at zero, denoting that there is no longer any pressure in the steam. This last, however, is not actually the case, since the pressures marked on the gage are in each case 15 lbs. per inch less than the actual pressure of the steam when the needle stands at that point, which 15 lbs. serves in a high pressure engine to overcome the atmospheric pressure: which, in consequence of the exhaust port being open to the atmosphere, acts upon the exhaust side of the piston as back pressure, and therefore has to be overcome by an equal pressure of steam on the opposite side of the piston; so that, when a high pressure engine uses its steam expansively, so that it exhausts at the gage pressure of zero, it has extracted from the steam all the useful effect possible in such an engine, but at the same time not all the useful effect or power which the steam contains, as will be hereafter explained. This leads us naturally to another consideration, which is that, if steam be used expansively in a high pressure engine to an excessive extent, the result is an actual loss of power, because, if the steam on the one side of the piston is at a pressure less than the atmospheric pressure on the other, the latter acts of course as a retarding force to the advancing piston.

The steam passages between the valve seat and the cylinder bore, and the clearance between the piston (when it is at the end of its stroke) and the cylinder cover, are spaces which have each to be filled, during each revolution of the engine, with live steam; and if the engine is not worked expansively, this live steam escapes without giving any of its power to the engine, and is lost, except in so far as it was necessary to fill those spaces. If, however, the engine is worked expansively, the expansive force of such live steam is extracted from it and applied as useful effect upon the piston, the result being an appreciable gain in the economy of steam, especially in those engines which, by reason of having the valve seat in the center of the cylinder, have very long steam passages, not merely because of the length of such passages, but also because in such cases the steam port serves alternately as the exhaust port, and has therefore to be made of larger proportions than it would need to be if employed as a steam port only, since an exhaust port always requires to have a larger area than a steam port. Hence the content of such passages, together with the clearance before referred to, bears a large proportion to the whole contents of the cylinder; and to extract power from the steam contained in them, by utilizing its expansive force, is a considerable gain to the engine.

From what has been already said, it will be perceived that a high pressure engine, to work to the greatest possible advantage and economy, should work its steam expansively to such a degree that it will be exhausted at zero of the pressure gage, or in other words at a pressure of 15 lbs. per inch, that being equal to the pressure of the atmosphere on the exhaust side of the piston. The point in the stroke at which it may be necessary to cut off the supply of steam to the cylinder, in order to effect such an amount of expansion, will vary according to the pressure of the initial steam and the length of the stroke of the engine, and must hence be determined according to those conditions.

An approximate calculation, as to what extent the steam in a cylinder is working expansively and its pressure at the termination of each inch of piston stroke, may be made by making the whole distance the piston has moved (under both live and expansive steam) the denominator and the distance it has moved under expansive steam the numerator of a fraction, and then multiplying the initial pressure by the numerator and dividing by the denominator of the fraction; then subtract the quotient from the initial pressure, the last product being the pressure of the steam. Thus: Supposing the initial pressure of the steam admitted to a cylinder to be 60 lbs. per square inch, the length of the piston stroke to be 20 inches, and the supply of steam to the cylinder to be cut off by the valve when the piston has traveled 5 inches of its stroke, what pressure of steam will there be in the cylinder when the piston is at the end of the tenth and twentieth inches of its stroke, respectively: Here the tenth inch of stroke —whole distance moved by the piston $=10$, distance moved by the piston under expansive steam $=5$, hence the fraction $\frac{5}{10}$; then the initial pressure $60 \times 5 = 300 \div 10 = 30$; then $60 - 30 = 30 =$ the lbs. pressure on the piston when it had arrived at the end of the tenth inch of its stroke.

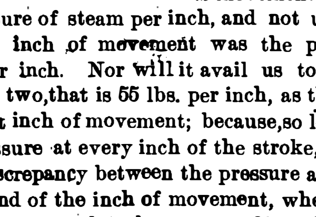
Again: Whole distance moved by piston $=20$ inches, distance moved by the piston under expansive steam 15 inches, hence the fraction $\frac{15}{20}$; then the initial pressure of the steam $60 \times 15 = 900 \div 20 = 45$; then initial pressure $60 - 45 = 15 =$ the pressure of the steam in pounds per inch at the end of the twentieth inch of the stroke or piston movement.

By making such a calculation for every inch of the piston movement and setting the figures in a column and adding them together, and dividing their sum total by the number of inches in the stroke, we arrive at a tolerably accurate estimate of the average pressure of the steam upon the piston throughout the stroke.

A review of the above calculations discloses that, as before stated, the pressure of the steam has decreased in precise ratio to the increase of the space it occupied, that is to say, when the piston was at the end of its fifth inch of stroke (the steam supply being cut off) there was five inches of the length of the cylinder filled with steam at a pressure of 60 lbs. per inch; and when the piston was at the tenth inch of its stroke and the steam had expanded so as to occupy ten inches of the length of the cylinder, the pressure was reduced to 30 lbs. per inch; and the same rule applies to the twentieth inch of stroke, for the steam then occupied four times the space it did as live steam, and had therefore fallen to one fourth of its original or initial pressure. It is to be noted, however, that while such a calculation is absolutely correct as applied to any one definite point of the stroke (making no allowance for the steam in passages and clearance) it is not entirely correct in its results if we take a number of such points to obtain therefrom the actual average pressure of steam throughout the stroke, for the following reason: Suppose we calculate (by the given rule) the pressure of the steam per inch upon the piston when it had concluded its sixth inch of stroke. Here the whole distance moved by piston $=6$ inches, distance moved under expansion $=1$ inch, therefore the fraction is $\frac{1}{6}$; then the initial pressure $=60 \times 1 = 60 \div 6 = 10$, then again initial pressure $60 - 10 = 50 =$ pressure of steam per inch upon the piston at the termination of its sixth inch of stroke. Now while 50 lbs. per inch accurately represents the pressure of steam upon the piston at the termination of its sixth inch of movement, it in nowise represents the average pressure of steam per inch during the whole inch of movement, because the piston commenced that inch of its movement or stroke under 60 lbs. pressure of steam per inch, and not until it had concluded that inch of movement was the pressure reduced to 50 lbs. per inch. Nor will it avail us to take the mean between the two, that is 55 lbs. per inch, as the average pressure for that inch of movement; because, so long as we calculate the pressure at every inch of the stroke, we shall have the same discrepancy between the pressure at the beginning and at the end of the inch of movement, whether it be at the fifth, sixth, or seventh inch, or at $5\frac{1}{2}$, $6\frac{1}{2}$, or $7\frac{1}{2}$ inches of the stroke. To get a more nearly correct result, we must take a greater number of points in the stroke such as every half or quarter inch of the piston movement; the more points taken, the more nearly correct will be the result obtained. It is, however, generally considered as sufficiently correct for practical purposes to take as many points as there are inches in the piston stroke.

With a common slide valve, it is not practicable to cut off the steam supply to the cylinder sufficiently early in the stroke to effect so large a degree of expansion; because, in the first place, it would require the valve to have an excessive amount of steam lap, and the exhaust would take place too early in the stroke, thus causing the piston to travel a large proportion of the latter part of the stroke without having any pressure of steam behind it; and because in the second place, when there is the large amount of steam lap on the valve necessary to cut off earlier in the stroke than at two thirds (that is, carrying full steam two thirds of the stroke) the admission, expansion, and exhaust of the steam to, in, and from the cylinder becomes very irregular in the forward as compared to the backward stroke of the engine, which irregularity will be shown and treated upon in connection with the piston movement, steam supply, etc. To obviate the defect (above referred to) of a too early exhaust, the valve may have lap added to its exhaust side, that is to say, the exhaust port of the valve may be made narrower than the width between the two nearest together edges of the steam ports of the cylinder face, as shown in Fig. 51, C being the exhaust

Fig. 51.



port of the valve and from A to B being the lap on the exhaust side. Such lap is, however, only possible when there is a good deal of lap on the steam side of the valve.

The amount of exhaust lap is at all times to be governed by the speed at which the engine is to run. A fast running engine, cutting off its steam supply at about one half stroke (which is the extreme limit of expansion permissible with a slide valve), may have exhaust lap to half the amount of the steam lap; a slow running engine may have exhaust lap to nearly three quarters of the amount of the steam lap. The reason of the difference is that, as the exhaust lap retains the steam in the cylinder longer, it, to that extent, cramps the exhaust; and as a quick running engine requires a more free exhaust than a slow running one, the latter may have its exhaust more covered by the exhaust lap when the piston is at the end of its stroke.

The objection to a valve having clearance is the open communication permitted between the steam and exhaust ports, which, though it exists for only a comparatively insignificant space of time, is a radical defect, especially when it is borne in mind that, as we have already shown, a slide valve should always have steam lap, and therefore will always have a proportionate amount of exhaust opening, in addition to that given to it by the lead of the valve. Clearance, then, is an expedient which should never be resorted to, it being a blunder applied merely to remedy a blunder. Clearance to a valve having much lap on its steam side is altogether inadmissible, since it is not requisite to give a more free exhaust, while it assists in letting the exhaust steam escape earlier in the stroke; and by this means, it adds to a defect inherent in slide valves having much steam lap, which is a too early exhaust.

A slide valve is sometimes given what is called clearance, that is to say, it is made wider in its exhaust port than are the two nearest together edges of the steam ports, so that (referring to Fig. 51) the port, C, of the valve would overlap the steam ports to the amount of the clearance, giving to them both an open communication with the port, C, and therefore with each other during the instant of time at which the valve is in the center of its travel. Clearance on the exhaust side is therefore the very opposite of lap on the exhaust side of a valve. The object of clearance is to give the valve a more free exhaust, and it is therefore only resorted to in cases where, the valve having little or no steam lap, the exhaust steam cannot freely escape.

Common slide valves, however, work to better advantage when the lap is so proportioned as to cut off the steam at from two thirds to three quarters of the stroke than at any other point, because of the comparatively long stroke of the valve (and hence large eccentric) necessary when much steam lap is brought into requisition, and because of the large amount of friction between the valve and cylinder faces in consequence of the pressure of the steam on the back of the valve. There are of course many devices for balancing such valves and some for reducing the pressure to a minimum, but none have as yet appeared whose benefits have proved such as to cause their general adoption for locomotives or small stationary engines, to which the application of the common slide valve is now almost universally confined.

To reduce the friction to a minimum, that part of the cylinder face upon which the face of the slide valve works may be raised above the general face upon which the steam chest beds, as is shown in Fig. 51, so that the steam lap of the valve may have the steam on the under as well as the outer side, and be to that extent relieved of the outer pressure. In such case, the width of the projecting faces (marked D in Fig. 51) should not be any wider than is the bridge (of the cylinder face) between the steam and exhaust ports; otherwise the wear of the face of the bridge will be the greatest and the valve seat of the cylinder face will wear hollow, the valve springing (to fit such face) from the steam pressure on its back. Especially is this the case where a high pressure of steam is employed. It is not uncommon to cut away these faces, leaving them full only around the edges of the ports, which cutting is performed by a slotting drill.

It is advantageous to make the steam ports long and narrow rather than short and wide, so that, when the valve commences to open, whether it be on the steam or exhaust side, a small amount of opening will present a comparatively large area for the ingress or egress, as the case may be, of the steam; hence the supply and exhaust of the steam to the cylinder will be larger in proportion to the valve movement, and therefore more instantaneous. A long port will of course entail a broader valve surface, and hence increased pressure of the valve to its seat; but this is compensated for by the decrease in the stroke of the valve (and hence in the diameter and stroke of the eccentric) permissible with the long port.

The rule sometimes given by which to calculate the required area of a steam port is, say, for a fast running engine: One eighth the area of the piston is the proper area of the steam port; the employment of such a rule, however, gives a result bearing no definite relation to the piston speed, and leaves a wide margin of difference, since either 300 or 600 feet of piston travel per minute is a fast running engine; whereas the amount of steam required to pass through the port for the one speed (supposing both pistons to be of equal diameter) is double that required for the other; while if the port area is larger than necessary, it causes a serious loss of steam; whereas if it is too small, it wiredraws the steam and fails to supply steam at full pressure to the cylinder. The following rule, given by Mr. Bourne, appears to meet the exigencies of the case, by giving the port an area proportionate to the quantity of steam required to pass through it. The rule is; Multiply the area of the cylinder in square inches by the speed of the piston in feet per minute, and divide the product by 4,000; the quotient is the area of each steam port in square inches.