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# "SUPERHEATING,"

BY

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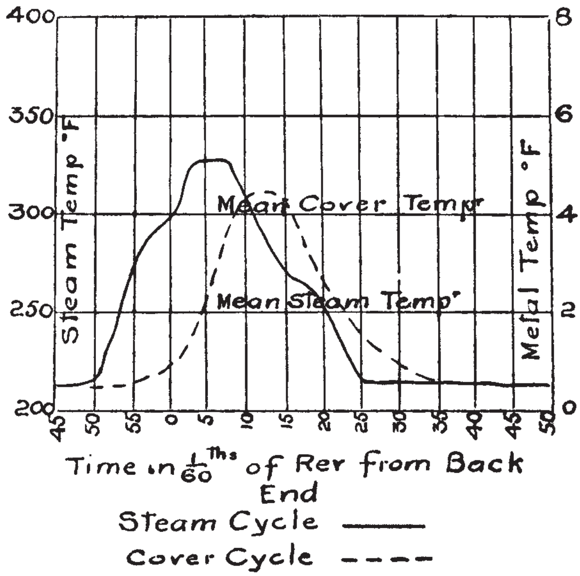
Superheating being a remedy for certain defects in the thermal working of a steam engine, the author proposed first to examine the sources of these heat losses, and incidentally to locate them on the theta-phi diagram ; then, by reference to actual tests, show how superheating remedies them. This will also be illustrated by theta-phi diagrams. It will therefore be observed that these remarks are practically confined to the steam cylinder.

As he had not carried out any trials on the use of superheat, the results which were submitted had been of necessity compiled from various published accounts of trials ; these will be acknowledged in their proper places.

Commencing with the losses in the cylinder, the author had the good fortune to discover a paper, in the " Proceedings of the Institute of Civil Engineers," by Professors Callendar and Nicolson, dealing with the thermal changes which take place during the stroke of a piston. Both these measurements were performed in the steam and the cylinder wall by the aid of electrical thermometers, but it would be outside the subject of this paper to describe the method of obtaining them. From that paper were chosen two diagrams, which would well repay close attention for a short time. The first represents the temperature changes which take place in the steam and cover wall (at a distance of $\frac{1}{25}''$ from the inner surface) during one revolution, commencing with positive, *i.e.*, the working stroke. It will be noticed that the vertical ordinates denote $\frac{1}{60}$ of a revolution ; also that zero is placed about a fourth of the length (representing a revolution) from the end of the diagram. A moment's consideration will decide that this is perfectly correct, for the positive stroke (and therefore the revolution) commences soon after admission, and compression in reality takes place during the later portion of the preceding negative stroke ; this is shown on the diagram.

Fig 1

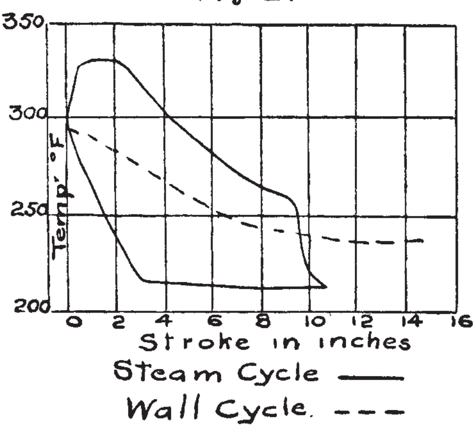



Very little need be said about the horizontal ordinates which denote temperatures, except to call attention to the fact that the figures on the left denote the steam temperature, while those on the right designate to a greatly extended scale the temperature variations, and *not* the actual temperature, of the metal cycle.

Following first the steam cycle, it will be noticed that the temperature gradient is fairly steep during compression, then at admission it falls off slightly, but rapidly recovers itself, and rises gradually until, soon after the commencement of the positive stroke, it suddenly changes into a level, in which it continues until cut-off is reached ; then it immediately falls with a decided incline, which, however, becomes somewhat less precipitous towards release, after which it descends more rapidly than ever into the constant temperature line, the latter being reached a short time before the end of the stroke, and continuing right up to the commencement of compression.

Observing the metal cycle, it will at once be seen that it does not even approximately follow the steam cycle, but clings closely to about 300° F. (note the cover mean temperature line); on examining it more closely it is seen that the temperature rises and falls with about equal

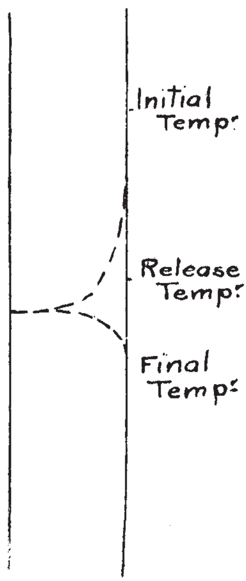
Fig 2.




gradients, that the temperature is still rising after cut-off has taken place, due probably to condensation, and that a maximum is reached at about a quarter stroke, after which it falls gradually until shortly after the negative stroke has commenced, the temperature then remaining constant until compression commences.

Fig. 2 shows the temperature gradient along the side of the cylinder. The engine used for this experiment was of a single-acting, simple, non-condensing type. In this diagram the vertical ordinates represent the stroke in inches, and the horizontal ordinates the temperature of both steam and walls. It will be noticed that the steam cycle is similar to the former one—indeed, practically the only difference being that it is plotted to form a closed figure, and that the wall temperature, shown by the dotted line, like that of the cover, does not follow the steam in temperature at all closely. This is probably partially explained by the state of things represented by Fig. 3, which shows a cross section of a cylinder wall ; the two dotted lines enclose an area, the vertical height measuring the change in temperature in the plane at any particular distance from the inner surface. Thus on the surface the temperature alters greatly during a revolution, but as the observations are taken farther from the surface, the periodic variation becomes less and less, until just before the centre is reached it ceases altogether, and the temperature then gradually falls towards the outer surface. This being so, the region of non periodic change of temperature acts as a jacket, at certain parts of the stroke, to the region of the periodic change, and heats it up to a temperature above that which it would have had if there were no region of constant temperature. From these diagrams it may be seen that at certain periods of the double stroke the steam is much hotter than the walls, and consequently parts with some of its latent heat in the endeavour to heat the metal to its own temperature ; but long before it attains its object the steam temperature has fallen far below that of the walls, and immediately absorbs some of the heat, which it formerly supplied.

Fig. 3




The diagrams also show that it most inconsiderately performs these evolutions at inconvenient moments, for the steam is parting with its heat just at the time it is most needed, *i.e.*, during admission, and it takes up heat again during exhaust, when all heat in excess of that required to keep it in the state of vapour is useless. Then again, considering the peculiar nature of saturated steam—*i.e.*, that with the slightest fall of temperature some steam is robbed of its latent heat, and is deposited in more or less fine globules on the walls of the clearance spaces, and that when the pressure falls at release, these globules are boiled off the walls (for the boiling point depends upon the pressure), and are thus carried away by the exhaust steam and irrevocably lost as far as the engine is concerned—it is doubly improvident that the temperature cycle should behave as it does. Thus, not only is heat lost by conduction, first from the admission steam to cylinder walls, and then from the cylinder walls to the exhaust steam, but also by the cylinder acting first as a condenser and then as an evaporator.

TABLE I.

| | Square Feet. | Mean Temperature °F. | Heat absorbed B.T.U. per minute. |
|---|---|---|---|
| Cover Face, 10·5 dia. ... | ·60 | 305 | 68 |
| Cover Side, 3″ ... | ·70 | 305 | 79 |
| Piston Face, 10·5″ dia. ... | ·60 | 295 | 110 |
| Piston Side, ·5″ ... | ·11 | 295 | 20 |
| Barrel Side, 3″ ... | ·70 | 297 | 123 |
| Counter Bore, ·5″ ... | ·11 | 290 | 28 |
| Ports and Valve ... | ·90 | 305 | 102 |
| TOTAL ... | | | 530 |
| Or Estimating 55 B.T.U. per minute up to ·25 stroke ... ... TOTAL ... | | | 585 |

Table I. was calculated by Messrs. Callendar and Nicolson for their experimental engine, and shows most clearly that the clearance walls may claim the doubtful honour of being responsible for about 90% of the initial condensation losses. The Table speaks for itself, 585 T. U. being the amount of heat absorbed by the cylinder walls, and of this amount the clearance accounts for 530 T. U., *i.e*, about 90%.

Other fruitful sources of heat losses are the piston and valve rods. For these are at one moment plunged into the seething hot steam, condensing some of this on their surface, and the next moment exposing their wet, hot surface to the comparatively cold outer air, where the pressure being less than within the steam chest and cylinder barrel, the water is quickly evaporated, abstracting heat from the metal in so doing, and thus adding to the loss by radiation. The walls of the steam chest are also sources of loss. Being on the one side exposed to the hot live steam, and on the other to the cooler air, some steam is condensed, and the resulting water mingling with the steam renders the latter wet. Then, again, the back of the unbalanced slide-valve and the parts, having been cooled by the exhaust steam, also condense some live steam. Now, although it has not been thoroughly investigated, it is known that the viscosity of water at any particular temperature is much less than that of steam at the same temperature, so that although a valve may be perfectly steam-tight it is not necessarily water-tight as well, therefore, in all probability, some heat is lost by the condensed water creeping through into the exhaust. Indeed, Professor Callendar believes that the greater part of the steam not accounted for by an indicator diagram is lost in this way. Priming may also be classed with the various other sources of heat losses attributed to wet steam ; and, with them, can be remedied by superheat. On the other hand, slow piston speed, too early cut off, contracted steam passages and badly set valves are all causes of cylinder losses, which, although they may be alleviated by superheat, may be totally or partially remedied by purely mechanical means ; so that it would be out of place to discuss them.

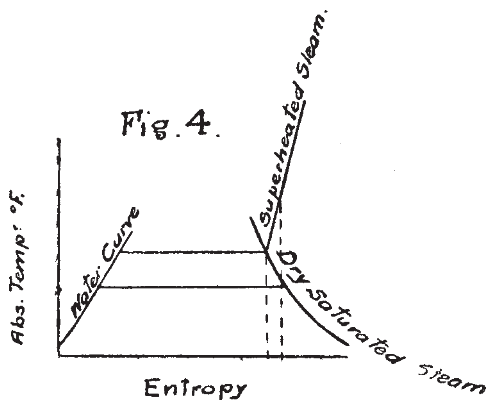
Before discussing the beneficial effect of superheat, it is desirable to describe in some detail a method of graphically representing the thermal changes which take place in a heat engine. It is done on a diagram called a theta-phi diagram, the area representing work done or to be done in units of heat. By convention its vertical dimensions are temperature, and horizontal dimensions denote entropy.

Entropy—a word of rather unfamiliar sound—is the name given to as distinct a physical property of a body as that body's volume, pressure or temperature, and just as work cannot be done without change of volume, so heat cannot enter or leave a body without change of entropy. The analogy goes even farther, for, as work done on or by a body is measured by the change in volume multiplied by its pressure, so the measure of heat entering or leaving a body is given by the change in entropy multiplied by its absolute temperature. Thus entropy can conveniently be defined as the quantity of heat in a body divided by its absolute temperature.

Co-ordinate with entropy is absolute temperature. It is measured from a point on any temperature scale, at which, theory, based on the laws dealing with the relation of volume, pressure and temperature, says that at this temperature a body would have no volume, and is, therefore, supposed to be absolute zero of temperature.

Now, to describe the method of constructing the theta-phi diagram. First, because the quantity of heat in a body varies as the mass of that body, it is necessary to assume some unit of weight for the material of which the entropy is required. In English-speaking countries 1 lb. is the unit used. Then if the total heat in 1 lb. of water at any absolute temperature be divided by that absolute temperature its entropy

Fig. 4.




is obtained, and if, on a piece of squared paper, having entropy as horizontal dimensions and absolute temperature as vertical dimensions, points of kindred entropy and absolute temperature be plotted, a curve will ultimately be drawn such that if two points be taken on it, the area beneath the portion of the curve between them down to absolute temperature is the measure in thermal units of the quantity of heat added after the lower point on the curve was reached ; this curve is called " the water curve " (*See* Fig. 4.)

Now at any particular temperature, floating above the water is an atmosphere of vapour in which the quantity of heat exceeds that contained by the water by the latent heat of steam at that temperature. If then this latent heat be divided by the corresponding absolute temperature, a number is obtained which when added to the entropy already contained by the water at that temperature gives the entropy of steam at that particular temperature. If this total entropy be plotted on its rightful absolute temperature line, a point is obtained on a curve which when complete represents, on the theta-phi diagram the expansion of saturated steam. This is called the " steam curve." Then the rectangle on a line having its end on this curve, the temperature being constant, down to absolute zero, represents the heat units available for doing work, *i.e.*, the area beneath F G (*see* Fig. 6) down to absolute zero is the quantity of heat which it is possible to convert into work, F G being the maximum temperature.

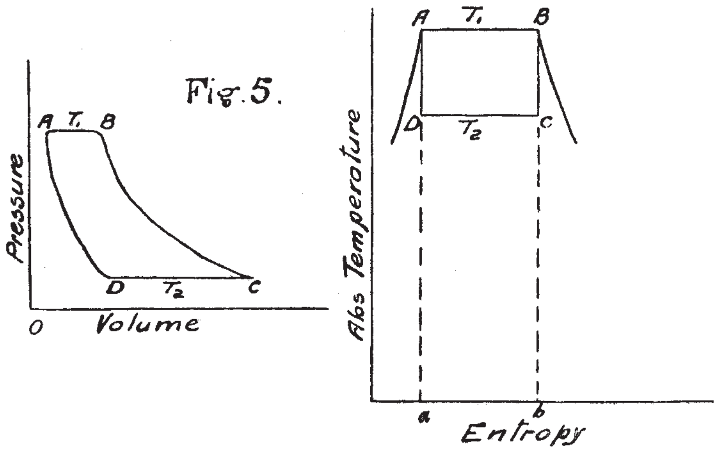
So far, saturated steam has been dealt with alone. The quantity of heat in any body equals the mass × specific heat × temperature. Suppose that steam at $T_1°$ is superheated to $T_2°$, weight being 1 lb., and specific heat being ·48, then the quantity of heat $\mathbf{Q}^1$ required = ·48 $(T_2° - T_1°)$. Entropy is $\frac{\mathbf{Q}}{\mathbf{T}}$, *i.e.*, quantity of heat divided by absolute temperature, but seeing that when the steam is being superheated the heat is neither all received at $T_1°$ nor $T_2°$, it is assumed that the heat is all received at the mean between $T_1°$ and $T_2°$, so that the entropy is given by ·48 $(T_2° - T_1°)$, divided by $\frac{T_1 + T_2}{2}$ and this must be added to the entropy of the steam at $T_1°$. When a sufficient number of corresponding entropies and absolute temperatures have been plotted, a curve is obtained similar to that marked " superheat " in Fig. 4.

Carnot's Cycle on a theta-phi diagram may be to advantage described at this point. The efficiency of this cycle is the difference between the initial and the final absolute temperatures divided by the initial absolute temperature—

$$\frac{T_1 - T_2}{T_1}$$

It is represented on a " P.V." diagram as in Fig. 5. Take 1 lb. of steam of volume O A (O being the intersection of the pressure and volume lines). Pressure P and Temperature T evaporate it isothermally, *i.e.*, at constant temperature until the volume becomes O B. Expand it adiabatically to C, *i.e.*, without supplying or abstracting any heat. Let pressure at C be $P_2$ and temperature $T_2$. Next compress steam isothermally until volume becomes O D, abstracting the heat thus generated by a refrigerator, so that the temperature still remains constant. Then adiabatically compress the fluid until it becomes of the same volume, pressure and temperature, as at commencement A. This is represented on the theta-phi diagram thus : Starting at temperature $T_1$ and entropy A, expand isothermally until B is reached. Being isothermal expansion the temperature is constant, but the entropy increases so that this expansion is represented by a horizontal line, and the area beneath this down to $T_0$ denotes heat supplied, *i.e.*, area A B b a adiabatic expansion takes place until condition C is attained. During this part of the cycle, heat is neither supplied or taken away,

Fig. 5.



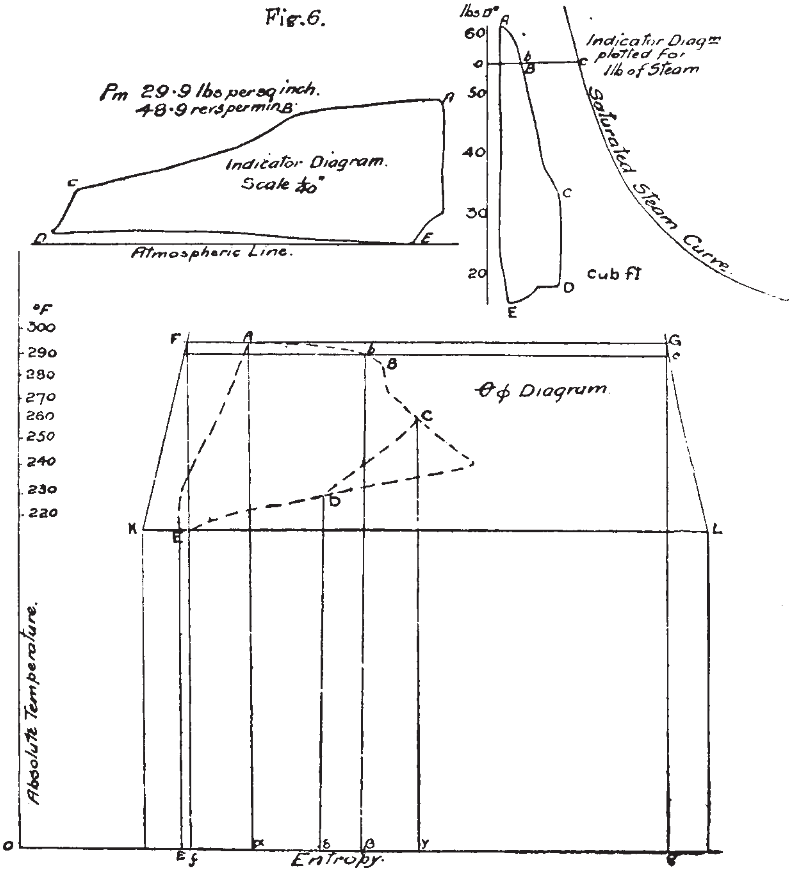

but inasmuch as heat, corresponding to the work done, must be lost, some of the steam is condensed, and thus at the end of the expansion C, the entropy is exactly the same as at its commencement B. This portion of the cycle is therefore represented by a vertical line. The temperature is now $T_2$. Next, following the cycle, isothermally compress the fluid until it is in condition D. The heat generated during this part of the cycle must be abstracted by some means, so that although the temperature is constant, the entropy decreases, therefore, this part of the cycle is represented by the horizontal line C D, and the heat abstracted by the area beneath it down to absolute zero, *i.e.*, area C D a b. Lastly, compress the fluid adiabatically so that it ultimately becomes in the same condition as it was at the commencement. This adiabatic compression is represented by a vertical line similarly to the adiabatic expansion, the only difference being that it is drawn from below upwards, while the expansion was drawn in the opposite direction. It has been shown that area A B b a represents heat supplied and C E a b heat rejected, hence their difference ; A B C D represents work done. And as the efficiency of a heat engine is the ratio of heat used in doing work to heat received, it is represented by—

$$\frac{\text{Area A B b a - area D C a b}}{\text{Area A B b a}}$$

or in other words $\frac{T_1 - T_2}{T_1}$

Carnot's Cycle has been drawn on the theta-phi diagram for the reason that later it will be seen that superheat materially assists steam to approximate to this cycle.

It is now necessary to briefly explain how the heat used in a steam engine can be plotted on a theta-phi diagram. It is really, but not directly, plotted from the mean indicator diagram. Not directly, for theta-phi diagrams are meaningless unless a certain weight of steam is specified, and one pound being taken as the standard amount, the indicator diagram must be plotted to scales of volume and pressure as though one pound of steam passed through the cylinder at every stroke. To do this the volume which has actually passed through must be known, and from this the required volume may be calculated for any pressure on the original diagram, and then plotted on the diagram for one pound of steam, this being, of course, plotted in conjunction with its proper pressure. Having done this, plot also on the same diagram the curve for one pound of saturated steam. This is shown in Fig. 6. Examining the diagram plotted for one pound of steam it will be seen that the actual expansion curve falls more or less to the left of the theoretical curve ; that is to say, the actual volume of the steam at any particular point is less than it should be theoretically on account of condensation. The ratio of the actual volume to the theoretical volume at any particular point is called the dryness fraction of the steam at that point. This is represented for point B by the ratio ab ÷ ac. This fraction may also be defined in thermal units, for it is the ratio of the heat actually used in doing work to the heat which might have been used providing there was no condensation. If, then, the temperature lines on the theta-phi diagram be divided in the same ratio as their corresponding pressures in the " P V " diagram are divided by their respective dryness fraction, there will be ultimately




plotted on the theta-phi diagram a figure representing the work done by the steam in thermal instead of work units.

And now, finally, the value of the theta-phi diagram may be discerned in its capability of locating and approximately measuring the heat losses which take place during each revolution of the engine. This is conveniently shown by the diagram Fig. 6. The area

F G g f representing the total heat supplied. The area A B *B* g G the heat loss due to walls during admission ; the area C D E *e y* the heat loss due to exhaust and back pressure ; the area B C *y B* the heat returned to the steam from the walls during expansion ; and the area A B C D E the work done in thermal units.

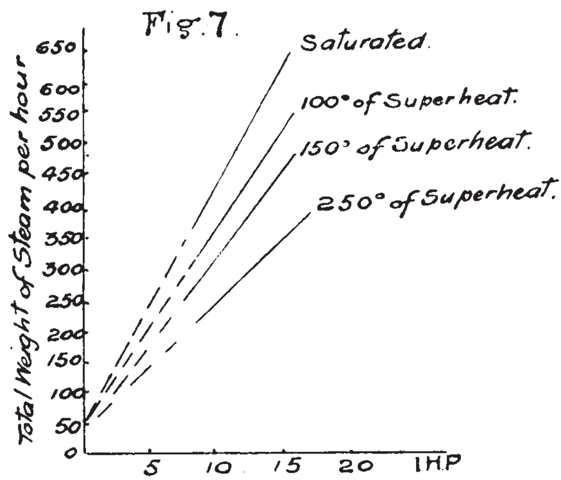
Passing on to consider the effects of superheating, the results now to be considered were all obtained by Professor Ripper. The engine was a " Schmidt Motor," a horizontal single acting, simple, non-condensing engine, somewhat similar in construction to a gas-engine. It had two cylinders, and no piston or valve rods. The valve-gear consisted of two plain piston valves, one being the inlet-valve and the other the exhaust valve for both cylinders. The engine was regulated automatically by a simple type of shaft-governor, altering, as required, the position on the shaft of the eccentric driving the inlet-valve. The clearance was rather large, and, therefore, against the engine being economical in the use of steam so that the results obtained were higher than they should have been. Yet as excessive clearance is detrimental to the use of superheated as well as saturated steam, the figures obtained showed the relation between the use of superheated and saturated steam, to the advantage of the former.

Table II.

| Boiler Pressure lbs. sq. in. | Feed Water Temp. Deg. Fah. | Deg. Fah. of Superheat above Saturation Temp. | B.T.U. added as Superheat. | Per cent. of Total Heat of Saturated Steam Represented by Superheat B.T.U.'s. | B.T.U. converted into work. | Net Gain per cent. by Using Heat as Superheat Instead of an Augmented Boiler Surface. |
|---|---|---|---|---|---|---|
| 107·6 | 206·2 | 0 | 0 | 0 | 64·2 | 0 |
| 104·3 | 199·1 | 254 | 122 | 12 | 108·9 | 51·4 |
| 105·1 | 200·7 | 321 | 154 | 15·2 | 126·5 | 70·9 |

Turning, then, to Table II., a set of figures are there given which are the results of a series of trials carried out under practically the same conditions, except the degree of superheat. Taking the first and last figures in the three last columns, it will be seen that with the addition of 15% of the heat required to evaporate the steam at the boiler temperature, the number of heat units converted into work increase from 64·2 to 126·5, a gain of 97%, or, as shown in the last column, a gain of 70·9%, even if the heat used in superheating had been used on an extended boiler surface. This gain is entirely due to the fact that the steam was dry both at cut-off and at release, which is, indeed, an ideal condition, for then no heat is lost in evaporating the water on the damp cylinder walls during exhaust.

Fig. 7 follows as a corollary to Table II,, showing that with the




increased thermal efficiency, less steam is used per hour for the same H.P. developed. But another fact shown by this diagram is that as the H.P. decreases, the advantage of any particular degree of superheat decreases also, so that when no power is developed there is little to choose between steam superheated 250° F., and steam at saturated temperature. This is no doubt partly due to the fact that with the lower loads cut-off occurs so early that the degree of superheat obtainable within workable limits is not sufficient to cope with the increased condensation due to early cut-off, and so some of the steam must also be condensed. Indeed, with this engine working with a fairly low number of expansions, namely, 2·5, the steam was superheated somewhat above 200° F. to obtain dry steam at cut-off, and as the temperature of the steam was then 565° F., it will be noted at once that with the present materials for making the parts in contact with the steam and for lubricating the piston, the limit of safety is very nearly reached. Another reason why the highly superheated steam curve converges towards the saturated steam curve is that just before the piston moves, and therefore before any power is developed, a certain amount of heat is required to warm the clearance walls to the maximum temperature, and, as this happens to be a rather large amount, and as the superheat is only about 15% of the total heat in the steam, some steam must of necessity be condensed. This occurs in any case, but when the amount of steam used per stroke becomes considerable, the superheat in the steam, if any, soon gains a mastery over this initial condensation, and accordingly reduces the amount of steam used per hour.

Table III.

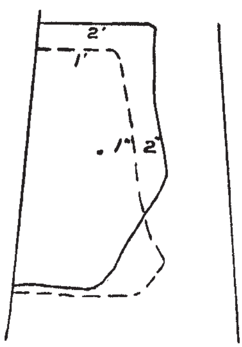
| Load 320 Lbs. | | | | |
|---|---|---|---|---|
| Press. Entering Engine. | I.H.P. | Superheat Deg. Fahr. | Steam per I.H.P. per Hour. | Abs. Thermal Efficiency. |
| lbs. per sq. ft. | | | | |
| 102·7 | 19·7 | 0 | 38·5 | 6·6 |
| 97·9 | 19·6 | 86·6 | 34·3 | 7·0 |
| 130 | 19·8 | 217·9 | 20·2 | 11·3 |
| 131·6 | 19·8 | 326 | 17·0 | 12·8 |

Table III. is closely allied to Fig. 7, being, indeed, the tabulated results of the steam per I.H.P. per hour for a series of degrees of superheat roughly corresponding to those in Fig. 7. The I.H.P. is approximately the same in each case. Here with 325° F. superheat, and the steam used per I.H.P. per hour falling from 38·5 lbs. with no superheat to 17·3 lbs., a saving of over 50%, this degree of superheat represents only a small percentage of the total heat of the steam ; so that the saving is greatly in favour of superheat.

Then, examining the next column, it will be seen that the thermal efficiency increases from 6·6 to 12·8, a gain of about 96%. Part of this, however, is due to the higher initial pressure of the steam on entering the engine, as shown in the first column.

Examining the three theta-phi diagrams—Figs 8, 9 and 10—it must be remembered that the top horizontal line represents the mean initial temperature, and the more nearly its end approaches the steam curve on the right the drier the steam is at cut off ; also that the vertical line represents the dryness fraction at any particular point ; that the point where this suddenly falls away towards the water curve is the point of release ; and that the longer this line is before it reaches the back temperature line the greater is the loss due to

Fig. 8.

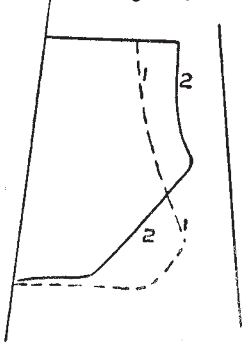



| SATURATED. | | |
|---|---|---|
| No. | Steam per I.H.P. per hour. | Load. lbs. |
| 1 | 44·7 | 140 |
| 2 | 38·5 | 320 |

incomplete expansion. The area surrounded by these lines represents the work done. In these diagrams no notice is taken of loss due to clearance, because it is unaffected by superheat.

The curve No. 1 on Fig. 8 shews that the steam is very wet all through the expansion, and that the load being light (140 lbs. on the brake) the cut-off occurs early, so that there is little loss due to low ratio of expansion. Now, on examination of curve No. 2, the load being increased $2\frac{2}{7}$ times, it is seen that the initial temperature is higher than in the former case, and that the steam is drier all through the expansion, but that there is a greater loss due to incomplete expansion, for with this higher load the steam is cut off later, to retain the speed about the same in all the trials. Also with the later cut-off, higher initial temperature and drier steam is obtained. Consequently, as only a limited amount of heat is required to heat the clearance walls to their maximum temperature, a smaller fraction of the steam supplied is condensed, resulting in higher mean initial pressure and drier steam. These latter advantages more than compensate for the loss due to

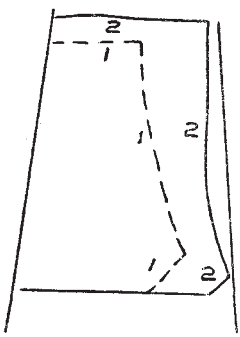
Fig. 9.



| SUPERHEATED. | | | |
|---|---|---|---|
| No. | Degree Fahr. Superheat. | Steam per I.H.P. per hour. | Load in lbs. |
| 1 | 87·5 | 37·9 | 140 |
| 2 | 86·6 | 34·3 | 320 |

late cut-off ; that is to say, more of the heat in one pound of steam is converted into useful work with the higher load ; or, putting it in yet another way, less steam is used per I.H.P. per hour ; this quantity being, in the case of No. 1, 44·7 lbs. per I.H.P. per hour, and in the case of No. 2, 38·5 lbs. per I.H.P. per hour, a saving of 14·5%.

Fig 10



| SUPERHEATED. | | | |
|---|---|---|---|
| No. | Degree Fahr. Superheat. | Steam per I.H.P. per hour. | Load in lbs. |
| 1 | 87·5 | 37·9 | 140 |
| 2 | 272·1 | 22·8 | 140 |

Fig. 9 shows similar results when using steam superheated to approximately the same temperature. The saving is, however, not so great as with saturated steam. This is probably due to a lower boiler pressure in the case of No. 2, and the superheat being insufficient to compensate for this. In Fig. 10 the load is constant, but the degree of superheat varies, and in this case a great saving is effected in the steam used per I.H.P. per hour. Not only is the steam much drier all through the expansion, but also the expansion commences appreciably earlier. This is plainly shown by the right-hand bottom corner of the diagram. Thus the higher the steam is superheated, the nearer the diagram approaches to the diagram representing Carnot's cycle, which is well illustrated by the diagrams. Indeed, it will be agreed that theta-phi diagrams are invaluable when it is desired to ascertain how, where, and why any particular method of working a steam engine is more economical thermally than other methods.

Difficulties in the use of superheated steam may now be mentioned. The chief of these is, or rather was, the great difficulty of properly lubricating the piston and piston rod, but in these days of graphite and high flash-point mineral oils, this obstacle is removed, and the difficulty of effective packing has been now overcome, by the use of metallic packing. It has been advanced as a disability of superheated steam that as it is superheated at constant pressure the volume increases, so that larger steam pipes are required and consequently a larger surface is exposed to the cooler atmosphere around it. This is no doubt true, but it must not be forgotten that the cross-section of the pipe increases as the square of the diameter, while the circumference increases as the first power of the diameter only, so that the increased surface is partially, if not wholly, counteracted by the increased volume within. It is also asserted that superheat is very quickly lost if the steam has to travel some distance, but the author was not aware of any experiments to prove this. Dry steam is an indifferent conductor of heat, so it is difficult to believe that it loses its heat at all rapidly. But even if it is so, then proper lagging ought to reduce this loss within reasonable limits. By far the most serious difficulty is the fact that the behaviour and strength of the materials used in the construction of the cylinder, etc, are not known at the high temperatures which successful superheating necessitate, so that there is always an unknown factor in designing the engine.

All the difficulties may be overcome by experience and ingenuity, but as yet superheating is an unexplored land, rich in possibilities, by no means impenetrable.