The

Una-Flow Steam-Engine.

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Preface.

All new doctrines are viewed with suspicion until some inquiring mind finds out that there is really nothing fundamentally novel in them. The principles on which the doctrines are based are proved to be old, but the fact remains that the Application of the principles is new. In the case of the uni-directional flow engine, or as it has been called for brevity, the "una-flow" engine, the above remarks apply very fully. After successful trials with my various designs of una-flow engine, friends and critics have been very ready and kind in pointing out that "something very like it has been done before". The great point is that nothing *exactly like it* has been done and none of the previous attempts have been attended with anything like the practical success which has attended my introduction of the una-flow engine about three years ago.

Of the earlier attempts, preference ought possibly to be given to Mr. J. L. Todd, who not only took out several patents in Britain for his improvements, but made actual tests with his engine. It is interesting to note the lines of development which Mr. Todd followed. He started off in his early British patent No. 7301 of 1885, with a very clear statement of the thermal conditions prevailing in an engine cylinder when the inlet was at the ends and the exhaust ports arranged in the centre of the cylinder and controlled by the piston. The features he emphasised were the "hot" inlet and the "cold" exhaust. From the very commencement of his investigation, Todd showed the tendency to move along wrong lines. He suggests that the excessive compression will be minimized by heat given up to the cylinder walls during compression which walls, he infers in order to perform this function, should not be jacketted. In fact J. L. Todd went so far as to heat the cold end instead of the hot end by placing the steam chest over the exhaust belt. These suggestions show the initial error Todd made, and it was probably this initial error which led Todd to depart from the pure una-flow engine and pass over to what he called the "dual" exhaust engine, viz, the mixed una-flow and counter-flow engine. The actual locomotive test made with Todd's engine was with the "dual" exhaust type and not with the *pure* una-flow type.

Another important point to bear in mind is that some ten years elapsed between Todd's first proposal and investigation of the una-flow engine and his introduction of the dual exhaust engine. It would appear therefore, that Todd had a hard struggle with the pure una-flow before he abandoned it for the "dual" exhaust. I am not aware that the dual exhaust engine was adopted to any large extent.

Many other inventors appear to have been fascinated with the uni-directional flow type, but I need only refer to Dr. Wilhelm Schmidt of Wilhelmshöhe. Dr. Schmidt does not appear to have gone so far as Todd. Like Mr. Todd Dr. Schmidt proposed to admit steam at one end of the cylinder and exhaust it through ports controlled by the piston. He made some proposals which can scarcely be taken as serious and practical, however interesting they might be otherwise. For instance, he proposes an automatic valve engine which depends upon varying throttling action and varying piston velocities for its operation. Then again, he does not always abide by the use of the cold annular exhaust belt which is connected by wide ports to the engine cylinder. This is an all important factor in the una-flow engine. Todd was ahead of Dr. Schmidt in this respect. In fact in view of the information gleaned from practical tests made with una-flow engines according to my design, it would appear that Todd, although he never obtained great success, was working on more correct lines than Dr. Schmidt.

I have to thank my critics for pointing out to me the work of these inventors and investigators, but would state that my investigations were entirely independent. Probably if I had advised myself fully of the work done by these gentlemen, I might also have been led astray. My investigations, however, have been entirely untrammelled.

I shall now briefly state the lines followed by me in my investigations. I set out to do in *one cylinder* what is usually done in *several cylinders* and I resolved to do this after the manner of a steam turbine, where the steam goes in hot at one end and has its energy extracted as is passes axially, always in the same direction, to the cold exhaust. Tackling the problem along these lines resulted in the forms of una-flow engine described in the pages of this book. It led to the following basic principles: — cut off early, use of a large ratio of expansion — keep the hot end hot — and the cold end cold. I have not departed from the first principle and in fact have only been more and more convinced of its soundness when followed to its logical limits. My various designs all have the main object in view of satisfying these basic conditions in as full a manner as possible.

What are the facts about the una-flow engine? Briefly these. — I do in one una-flow cylinder, what others do in two or three counter-flow cylinders (compound or triple). The results in steam consumption are the same, if not better. The cost for building and lubricating the una-flow engine is much less.

Expounding ideas is one thing — convincing one's fellow men to a sufficient degree to influence them to carry out the ideas into practice, is an entirely different and very much more difficult thing. My thanks are due to Mr. Smetana, manager of the Ersten Brünner Maschinenfabrik of Brünn for being the first whose convictions led to action. To quote this gentleman's own reply to my contentions "Your arguments are good — that I cannot deny — so I propose we put them to the test".

So the first una-flow engine was built by the Erste Brünner Maschinenfabrikgesellschaft in Brünn (Austria) in accordance with my design. It proved to be a full success and had a steam consumption equal to that of a good compound engine.

The first una-flow locomotive was built by the Kolomnaer Maschinenbau-A.-G., to the order of Mr. Noltein, the well known manager of the Moscow Kazan Railway, and proved eminently satisfactory. Since the time of its introduction, I have had no reason to complain of the slowness in advance. In fact at the end of July 1911, there were engines with a total output of over half a million horse power working or in actual construction.

This rapid development has entailed a vast amount of work in adapting the una-flow engine to all kinds of purposes. I am much indebted to my assistants in helping me to tackle this work and I would also specially mention Mr. Rösler of Mülhausen in Alsace, Mr. Arendt of Saarbrücken and Mr. Bonin of Charlottenburg, all of whom rendered me valuable assistance for which I thank them.

I must also express my thanks to all those gentlemen who allowed themselves to be convinced to the extent of action. In addition to Mr. Smetana already mentioned I have to thank Mr. Noltein of the Moscow Kazan Railway, Mr. Hnevkovsky of Brünn, Mr. Lamey of Mülhausen, Geheimrat Müller of Berlin and Mr. Schüler of Grevenbroich, for their assistance and support in the early stages of the development of the una-flow engine.

I have finally to thank Mr. P. S. H. Alexander of Messrs Mathys & Co., of 43 Chancery Lane, London for his services in the translation and preparation of this work in English.

Charlottenburg, Germany.

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Chapter I.

The general thermal and constructional features of the una-flow steam engine.

As the name indicates, the energy of the steam is extracted in the case of the "una-flow" or "unidirectional" flow steam engine without causing the steam to return on its path, that is to say the steam passes always in one direction through the steam cylinder. As shown in fig. 1, the working steam enters from below into the hollow cover, heats the surfaces of the cover and then passes by the valve in the upper part of the cover into the cylinder; the steam then follows the piston whilst giving up its energy and after it has been expanded, it passes out, at the end of the piston stroke, through the exhaust ports arranged in the middle of the cylinder and controlled by the piston. In ordinary steam engines on the other hand, the steam has a counter flow action, that is to say it enters at the cylinder head, follows the piston during the working stroke, and then returns with the piston on its return stroke to exhaust through ports opening near the cylinder head. The counter flow or reversal of the exhaust steam causes considerable cooling of the clearance surfaces owing to their contact with the wet exhaust steam. This cooling action results in considerable initial condensation when the boiler steam is again admitted to the cylinder at the next working stroke. In the una-flow engine, all cooling of the clearance surfaces is almost entirely avoided and hence cylinder condensation is to a great extent eliminated as also is the necessity of employing several expansion stages. Una-flow engines therefore may be made with a single expansion stage, whilst the steam consumption will not exceed that of compound and triple expansion steam engines.

By eliminating all cooling of the clearance surfaces by the exhaust steam a somewhat similar effect is obtained as is got by superheating. In the ordinary engine, superheating is employed to overcome the above mentioned difficulties caused by the cooling of the clearance surfaces. If now, this cooling is avoided, it would appear that all necessity for superheating the steam is removed.

The use of a ring of exhaust ports or slots in the cylinder enables the area of the exhaust passage to be made three times as great as the port-area obtained by the use of slide or other valves. The result of this large exhaust area is that the end pressure in the cylinder is that of the condenser, especially when the use of long and narrow pipe connections between the condenser and the cylinder is avoided. In other words, if the condenser is arranged close up to the cylinder and the exhaust passage has a large cross section, it is possible to bring the cylinder pressure down to that of the condenser. In order to form a proper idea of the dimensions of the exhaust ports, one should imagine a piston valve of the same size as the working piston and a valve casing of the same size as the

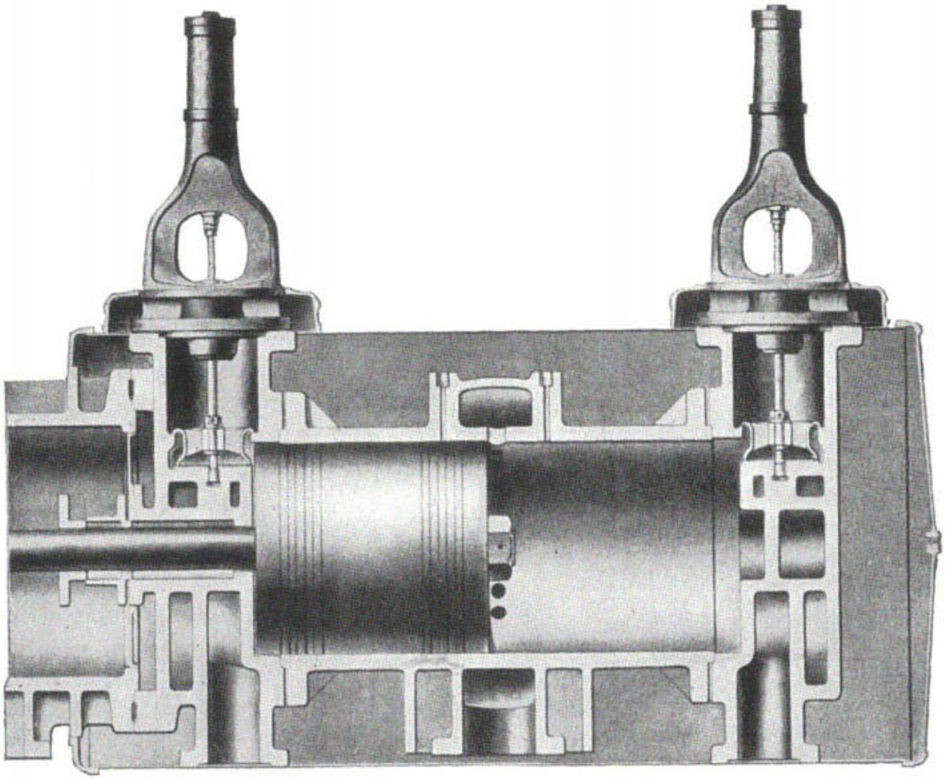


Fig. 1.

working cylinder, the piston valve being moved by an eccentric having the same throw as the engine crank. On an average, release takes place after $^{9}/_{10}$ ths of the stroke and consequently compression begins after $^{1}/_{10}$ th of the return stroke has been completed, or in other words, compression extends over $^{9}/_{10}$ ths of the stroke.

It will be evident that, by substituting exhaust ports or slots in the cylinder for the usual exhaust valve, all leakage losses at the exhaust valve and all the added clearance space and surfaces, which necessarily follow from the use of a special exhaust valve, are avoided.

The indicator diagram shows the expansion line to be an adiabatic for saturated steam and the compression line to be an adiabatic for superheated steam.

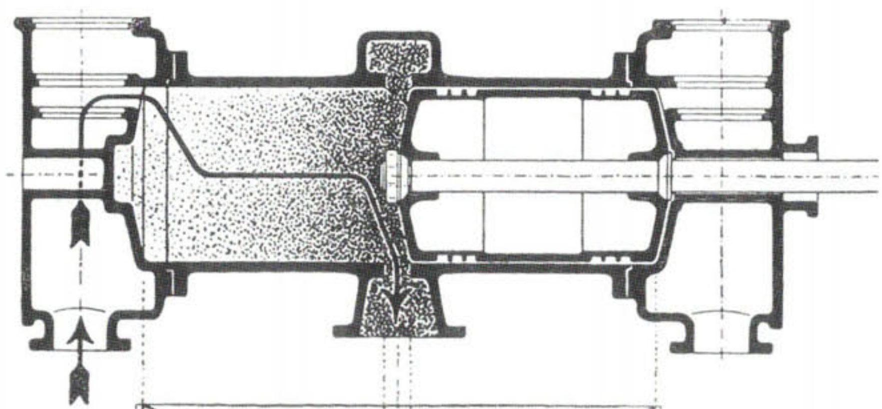
This is the best proof of the excellent thermal action of this engine. The excessive initial condensation, in an ordinary counter-flow engine using saturated steam, causes the expansion line to follow approximately the law of Mariotte. In the una-flow engine, using saturated steam, there is practically no initial condensation, so that the resulting expansion line is necessarily an adiabatic and all the more so if the steam is superheated.

Owing to the adiabatic expansion, the dryness fraction of the steam after expansion is very low. Thus in the case of steam having an initial temperature of 300 ^{0} C and an initial pressure of 12 atmos. expanding down to an end pressure of 0·8 atmos., the dryness fraction is 0·93, that is to say, the steam contains 7% of water. In reality the temperature of the steam at cut-off is liable to be a little less than the above owing to heat losses during admission. The result of these unavoidable heat losses during admission is that the expansion commences at a lower temperature and ends with a lower dryness fraction.

On the other hand, the heating jacket on the cover regenerates the steam during expansion. During expansion the cover jacket exercises considerable heating action owing to the considerable temperature difference between the cover and the steam, this heating action being transmitted principally to the steam immediately in contact with the cover. The steam immediately following the piston has a definite fall in temperature and increase in wetness owing to the adiabatic expansion. The greatest wetness is therefore found in the layer of steam immediately following the piston. In the layers between the piston and the cylinder cover, the wetness decreases until, in the case of the steam near the cover, some superheat may be present. Immediately on release, the wettest steam is forced out through the ring of exhaust ports in the cylinder walls. The steam which received heat during the entire time of expansion and exhaust and was subjected to the action of the full temperature difference between the expanding steam and the heating jacket is trapped by the piston and compressed, which compression will now approximate very closely to that of the adiabatic for superheated steam. This approximation to the adiabatic for superheated steam is still further assisted by the fact that during the first part of the compression, further heat is transmitted from the cover to the compression steam (fig. 2). Owing to the complete removal of all the moisture at each stroke, the well known heat losses, caused by the presence of water, in ordinary engines are avoided. Water hammer in the cylinder is in this way absolutely impossible.

Experimental investigation of steam jacketting on triple expansion engines shows, (I) in the case of a high pressure cylinder, no advantage, (II) in the case of the intermediate pressure cylinder, a small advantage, and (III) in the case of the low pressure cylinder, considerable advantage is obtained, in spite of the great losses which necessarily follow in the ordinary form of steam engine with a counter flow action of the steam. Counter-flow action necessarily involves the abduction of a considerable amount of heat from the jacket by the exhaust steam passing to the condenser. This will be appreciated when it is considered that, on the opening of the exhaust valve, a considerable amount of pressure energy in the steam is transformed into velocity energy, producing steam velocities in the ports and pipes between 350 and 400 metres per second. The wet exhaust steam sweeps over the clearance surfaces with this high velocity and deposits water of condensation on these surfaces. The result inevitably is that considerable re-evaporation takes place on account of the sudden fall in pressure and of the heat present in the walls of the clearances. The heat taken up by the clearances from the admission steam at each fresh charge is thus rapidly extracted during exhaust. A brief consideration will give a very fair idea of the uneconomical conditions in the ordinary steam engine, both as regards the loss of heat from the clearances and the loss of heat from jacketting. It should also be noted that the heat flow from the jacket

Fig. 2.



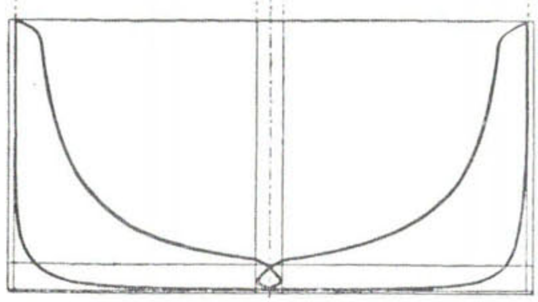


Fig. 3.

in an ordinary engine is greatest at the most unfavourable time, that is from the point where release commences to the point where compression begins or approximately during one half of a revolution, because it is during this time that the greatest temperature difference exists between the steam and the heating jacket. During the remaining half of the time of one revolution, the rate of heat flow from the jacket is less, as also is the velocity of flow of the medium over the hot surfaces. In spite of these disadvantages, the best results of jacketting are obtained in the low pressure cylinder. This may be explained by the fact that in the case of the low pressure cylinder, the heating jacket works with the greatest temperature difference. It follows from the above that in the case of a una-flow steam engine cylinder, a very efficient heating action must result as the heating in this case, as in the case of the low pressure cylinder of a triple expansion engine, works with the full temperature difference between the more or less fully expanded steam and the live steam. In addition to working with maximum temperature difference, it should be remembered that in the una-flow engine, the wasteful counter-flow of the ordinary engine is replaced by a uni-directional flow, so that not one single unit of heat is carried of from the jacketting by the exhaust steam passing through the exhaust ports. The exhaust steam, as a glance at fig. 2, will show, never passes over jacketted surfaces. The steam which comes in contact with the hot wall of the cover passes at most up to the neighbourhood of the exhaust ports, without however passing out through these ports. In consequence, the jacket heat can never be lost. The advantages of hot-jacketting the low pressure cylinder of a triple expansion engine must therefore be obtained in a much greater degree with a una-flow engine, because the great heat losses associated with the counter-flow action of the steam are wholly avoided by the new construction.'

In the above, it has been assumed that the jacketting is limited to the cover or cylinder head, and that the cylinder proper is not at all jacketted (fig. 1 and 2). It is preferable to extend the cover jacketting to the point, where cut-off usually occurs, so that the clearance walls are most effectively heated on one side by the hot steam jacket and on the inside of the cylinder by the superheated compressed steam. The end temperatures which may be obtained will be more clearly realized from the following numerical example. — Dry saturated steam compressed from 0·05 atmos. absolute to 12 atmos. absolute, gives an end temperature of 807° C, according to the adiabatic for superheated steam. This example shows that for reducing the detrimental effect of the clearance surfaces it is not necessary to compress up to the admission pressure, but that a medium compression is quite sufficient. For reducing the detrimental effect of the clearance space it is best to make same so small that the compression runs up as high as the initial pressure, but for low condenser pressures it is mostly impossible to get the clearance space so small as to suit this requirement. (See chapter IX.)

It can be shown by comparison with van der Kerchove's construction that the una-flow engine is based on sound principles. The favourable results obtained by van der Kerchove are traceable to the special arrangement of the steam inlet valve in the cover and the heating action on the cylinder ends thereby secured. The advantages obtained by van der Kerchove are considerably enhanced in the case of the unaflow engine, owing to the working piston replacing the exhaust valve employed by van der Kerchove, and further, owing to the increased cover heating action and the reduced clearance spaces and surfaces resulting from the una-flow system. Kerchove's exhaust steam chamber at the end of his cylinder completely nullifies his hot steam chamber, and even goes further — it actually cools his admission steam, owing to its juxta-position to the steam chest.

The losses due to the clearance spaces and surfaces may be further reduced by reducing the metal thicknesses and by carefully machining the surfaces.

Although the entire cycle is carried out with steam in contact with the clearance surfaces, the cooling action in the una-flow engine is small, firstly because of the comparative stillness of the molecules of steam, owing to the exhaust outflow being relegated to the entirely opposite end of the cylinder space, secondly owing to the absence of any re-evaporation, and thirdly on account of the combined heating action of the cover jacketting and the high compression.

The thermal mix-up peculiar to the ordinary or counterflow engine is wholly avoided by the una-flow system. The cylinder consists really of two single acting cylinders set end to end with their exhaust ends common. The two diagrams are separated in proportion to the length of the piston. *The two ends are hot and remain hot, the common exhaust belt is cold and remains cold.* From the hot ends to the cold centre, there is, on either side, a gradual diminution of temperature which, when

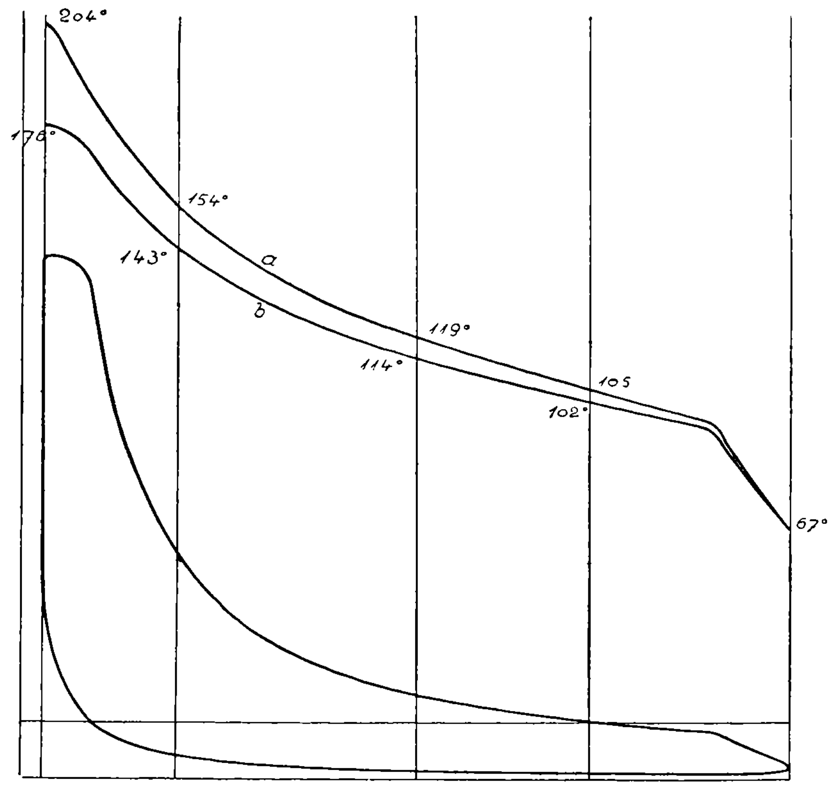


Fig. 4.

plotted out for each position of the piston, gives a line lying a little above the temperature line of the steam during admission and expansion. Fig. 4, shows the two lines plotted together, the upper one *a*, representing the temperature of the cylinder walls taken from actual measurements on a una-flow cylinder (non-jacketted cylinder, jacketted covers), whilst the lower one *b*, represents the steam temperature for the top or expansion line of the diagram.

This is in marked contrast to the ordinary or counter-flow engine where the two lines, as will be shown afterwards, have a very unfavourable situation and the exhaust end of one diagram projects pretty well into the inlet end of the other. In fact the entire engine is completely mixed up from a thermic point of view.

Owing to the disposal of the exhaust ports and the exhaust belt around the centre of the cylinder, the lowest temperature is secured in this part, where the piston velocity is greatest. This cooling is materially aided by dispensing with a jacket over the neighbouring parts of the cylinder. The piston has a large bearing surface and consequently a low specific bearing pressure. The cylinder is very simple in construction, being free from all cast-on casings or parts, so that local heating and casting difficulties may be practically eliminated, which results in the reduction or even elimination of internal strains and distortions. The large bearing surface of the piston, the cool exhaust belt and the simple form of the cylinder render the use of a tail rod quite unnecessary (see the designs of Gebr. Sulzer in Winterthur; Maschinenfabrik Grevenbroich; Globe-Iron Works in Bolton; Badenia in Weinheim; Gutehoffnungshütte). The piston is provided with two groups of rings, each group having usually three rings. Both groups of rings, that is usually six rings in all, are effective in packing the piston when the greatest pressure differences exist between inlet and exhaust. The steam pressure has usually fallen to about 3 atmos. when only a single group of rings is effective, that is when the other group has overrun the exhaust ports.

Experience with a large number of una-flow engines has shown that, even with very high superheating, the piston does not cause the slightest trouble if material and workmanship have been good and efficient lubrication is provided. Should the cylinder be scratched or scored owing to defective workmanship, bad material or bad lubrication, it is a very simple and quick operation to replace it, and owing to the simple form of the cylinder, the cost of such renewal is very low.

It is possible, without making any modification in the construction described, to work with superheats far in excess of those usually employed at present. Even with the highest initial temperatures the last part of the expansion line falls well below the saturation line, so that even with a hot steam jacket on the cover, reasonable and practicable working temperatures are obtained for the cylinder and piston.

Consequently the new construction opens up a further field of development by using higher temperatures. The new una-flow engine is all the more suitable for use with superheated steam as the superheat is favourable for the entire cycle, whilst in the ordinary counter-flow engine with multi-stage expansion, the superheat is excessive for the first cylinder and is too little in the succeeding cylinders. The excellent operation of the una-flow engine with superheated steam does not refute the fact, that it is also excellently suited for use with saturated steam. In reality the una-flow engine gives almost the same economic results with both kinds of steam. A further field of possibilities is opened up by the fact that higher initial boiler pressures may be employed. Pressures up to 30 atmos. should be quite as practicable in the una-flow engine with atmospherie exhaust as in the Diesel engine. A third field of development is opened up by combining high pressures and high temperatures.

The thermal, constructional and working advantages of the una-flow engine are such that the steam consumption for saturated and superheated steam is the same as that of good compound or triple expansion engines.

Chapter II.

The relation of the una-flow engine to the condenser.

A good condenser vacuum is of great advantage to the una-flow engine. The various compression lines for different initial pressures and the same end pressures are shown in figure 5, the clearance space being correspondingly altered for each case.

From these lines, it will be seen how the area of the diagram is considerably increased by employing a high vacuum. It is also possible, by suitable proportioning of the clearance space, to obtain any desired compression pressure and temperature. Working with a good vacuum, the clearance space may be very small, for instance the clearance space may be less than 1% of the working space of the cylinder with a vacuum of 0·05 atmos.

On comparing the duration of exhaust in the case of a una-flow engine, in which compression extends over $^9/_{10}$ths of the stroke, with the duration of exhaust in an ordinary counter-flow engine, it is found that the periods are approximately in the ratio of 1 to 2. The working steam in the una-flow engine must therefore be cleared out of the cylinder and passed to the condenser in one half of the time available in the case of the counter flow engine. It is a fact that in the present form of counter flow steam engine, a considerable pressure difference exists between the interior of the cylinder and the condenser, which pressure difference may be traced to the energy necessary to overcome the resistance of the ports and passages which are in almost all cases too narrow. As in the case of the una-flow engine the duration of the exhaust is reduced by one half, it is doubly essential to reduce, as far as possible, the resistance to the flow of steam to the condenser and this may be done by means of passages of large area and short length. These requirements are met to a considerable extent by the fact that the area of the piston-controlled exhaust ports in a una-flow engine may be three times as great as the exhaust port area of an ordinary counter-flow engine employing an exhaust valve. If the cross sectional area of the other passages are large in proportion to the large exhaust port area, and if the length of the connections from the exhaust port to the condenser is reduced as far as possible, all the requirements are satisfied for obtaining an end pressure within the cylinder equal to the pressure in the condenser. Experience proves this to be the case.

In figures 6 and 7, a una-flow engine is shown in cross section and longitudinal section. As can be seen the exhaust belt opens with its full breadth and diameter directly into the spray condensing chamber arranged below the cylinder. The spray is effected by means of a horizontal spraying tube introduced into the condenser chamber. As shown in the drawing, this construction provides an extremely large area for the flow of the exhaust steam whilst at the same time the length of the passages is reduced to the minimum. The result of this construction is equality in pressure within the cylinder and condenser when the exhaust ports are fully open.

In figures 8 and 9, a una-flow steam engine cylinder is illustrated in combination with a spray condenser of the Westinghouse-Leblanc type. The plant

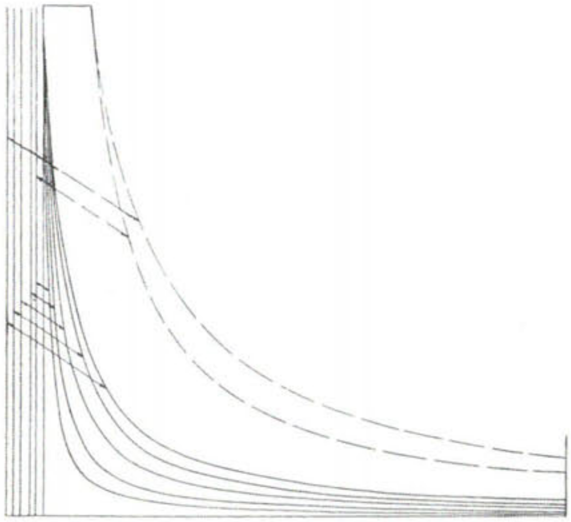


Fig. 5.

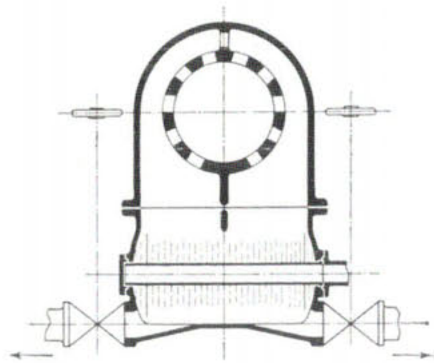


Fig. 6.

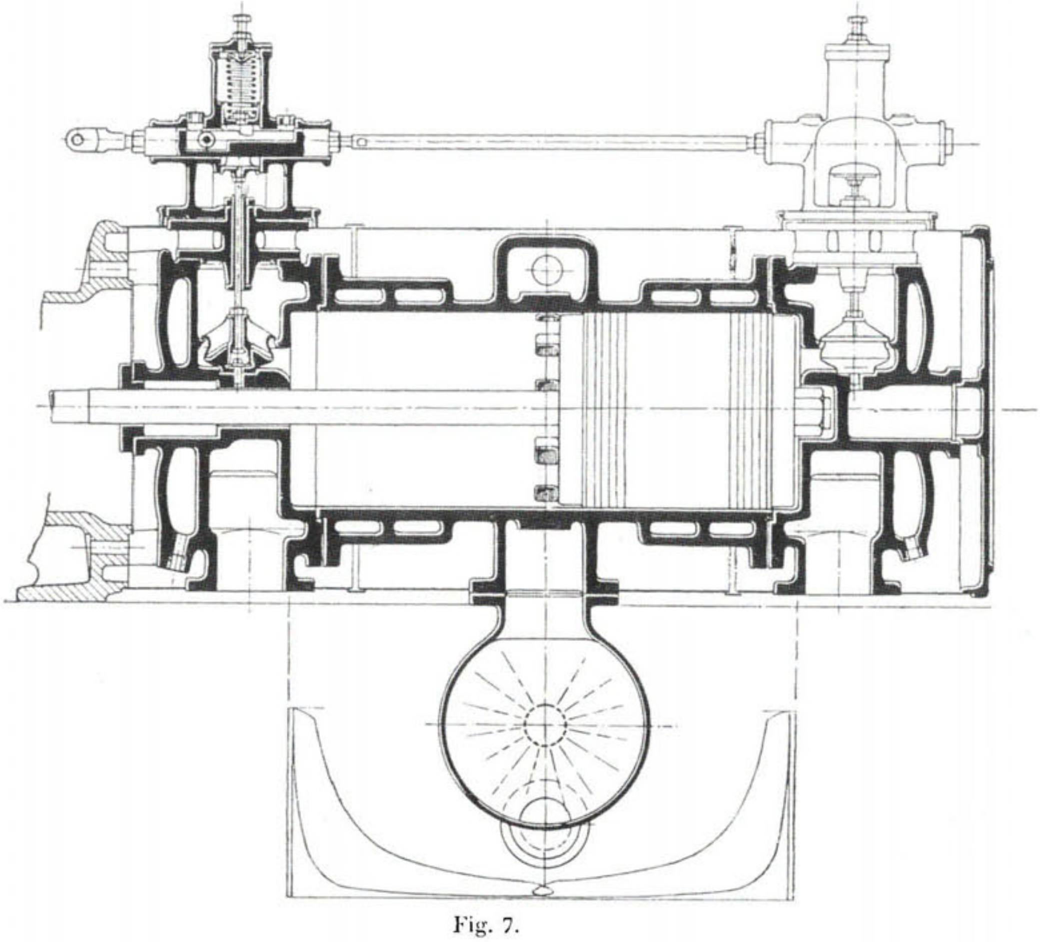


Fig. 7.

is so arranged that the condenser is also a supporting column for the una-flow engine cylinder. In this construction, as in the case of figures 6 and 7, a large cross sectional area is obtained for the flow of the exhaust steam, whilst the connections are extremely short. Here also there is complete equality between the end pressure of the steam in the cylinder and the condenser pressure. This complete equalisation of both pressures enables compression to start at the lowest possible point. The result is a considerable gain in the area of the diagram, a corresponding reduction of the clearance and the clearance surfaces, as well as a con-

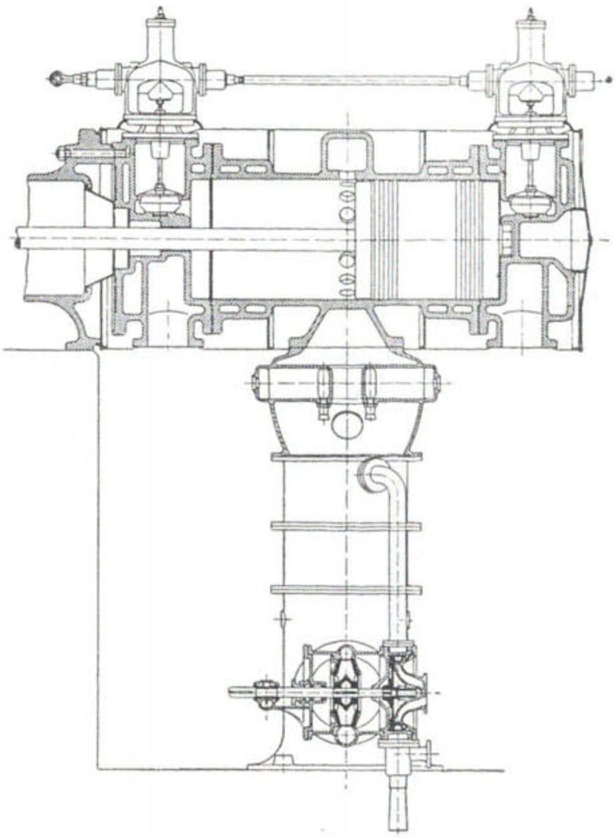


Fig. 8.

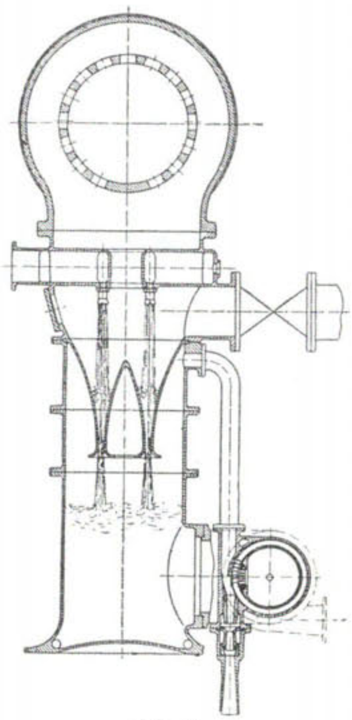


Fig. 9.

siderable improvement of the thermal efficiency (fig. 5). The reduction of the duration of the exhaust, with piston-controlled ports, also causes a corresponding reduction of the cooling of the cylinder when in communication with the condenser, as compared with ordinary counterflow engines. As soon as the exhaust ports are closed, the condenser cooling action ceases completely and during the entire remainder of the stroke the heating action of the jackets is effective. This heating action is not reduced or rendered inefficient by the nature of the exhaust or by any circumstances connected with the exhaust.

It is very bad practice to introduce anything in the nature of an oil separator, switching valve, reheater or bends in the connection between a una-flow steam engine and the condenser. Such parts cause very considerable resistance and should therefore be wholly avoided, if their introduction is not rendered absolutely necessary by other very important circumstances. The ports may also be formed as nozzles as shown in figure 10, in which case the nozzles are in the form of pipes opening into a steam turbine. This steam turbine would then work with intermittent impulses or puffs at each exhaust from the cylinder, the pressure-energy remaining in the steam being transformed into velocity-energy and extracted in the turbine. In this way a considerable part of the energy, represented by the toe of the diagram, which is cut off, may be usefully employed. A very good construction would be to employ three rows of exhaust ports, the two outer rows

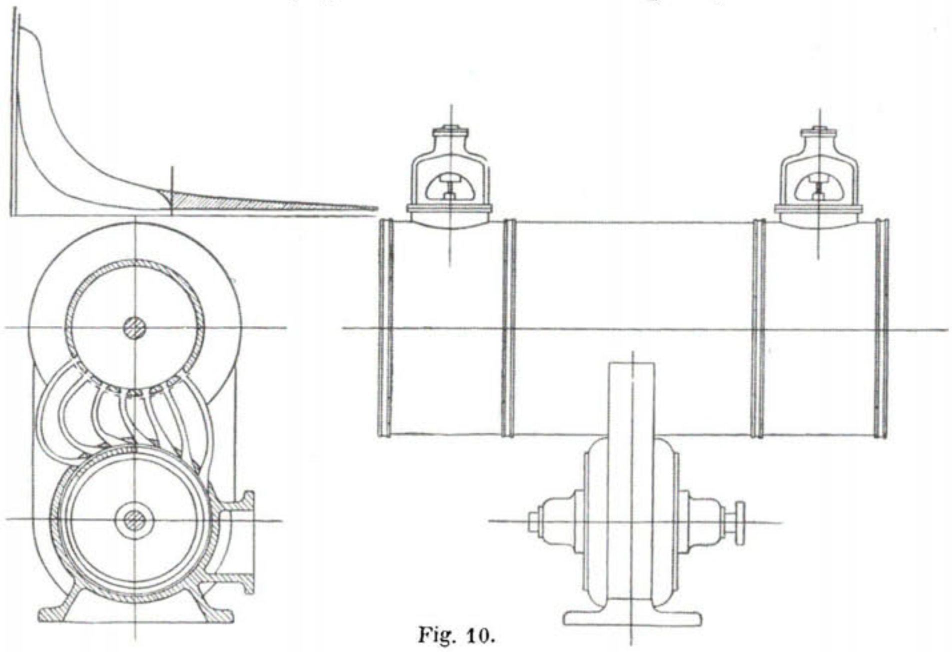


Fig. 10.

being connected as described to a turbine and the centre group of ports being connected directly to the condenser. The direct connection of the middle row of ports to the condenser produces the highest possible vacuum in the cylinder.

The connections to the atmosphere should be by way of the condenser. In such a case when the air pump, which is also connected up to the condenser, is shut down, the condenser will act as a silencer (see fig. 6).

Chapter III.

The steam jacketting.

During expansion, a very active transference of heat takes place between the cover and the layer of steam immediately in contact therewith, in consequence of the great temperature difference between the jacket and the expanding or expanded steam. The steam entering the cylinder contains less heat than the jacket steam and becomes wetter by the expansion. The greatest amount of condensation takes place in the layer of steam immediately following the piston. The moisture of the steam decreases in the layers between the piston and the cover, whilst that layer immediately in contact with the cover may at the end of expansion be dry or even superheated. During the exhaust, the wet steam is passed out of the cylinder through the ports. The quantity of steam therefore which, during the entire period of expansion, has been receiving heat from the cover at the full temperature difference between the expanded steam and the heating jacket, is trapped by the piston. This trapped residual steam is used for compression, and the compression will approximate very closely to the adiabatic for superheated steam. This approximation is still further promoted by the fact that during the first part of the compression more heat is given up by the cover to the steam being compressed. Experiments have shown that jacketting increases in importance as the temperature of the working steam approaches the saturation temperature. The importance of the jacketting therefore decreases proportionately as the steam is superheated. This will be seen from figure 11, which shows a unaflow steam engine cylinder with suitable jacketting for working with steam of a temperature of 350° C and over. In this case the jacketting is confined wholly to the cover. In figure 12, the construction of the jacket is illustrated for an engine working with steam at a temperature of 250° to 350° C, and in figure 13, the corresponding construction for saturated steam is given. On comparing the three figures, it will be seen that the jacket is extended from the end towards the middle proportionately as the temperature of the working steam approaches that of saturated steam. Even in the case of saturated steam, it is advisable to leave a neutral zone between the heating jacket and the cooling jacket, the exhaust belt being assumed to act as a cooling jacket. This neutral zone is neither heated nor cooled. It is also preferable to take the steam for the cylinder jacketting from the upper end of the cover where, in the case of superheated steam, the temperature is considerably reduced in consequence of the heat given up by the cover jacket. A further reduction in temperature takes place in this cylinder jacket, which of course does not exercise such a strong heating action as the cover jacket. The neutral zone,

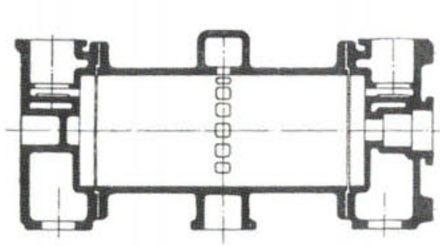


Fig. 11.

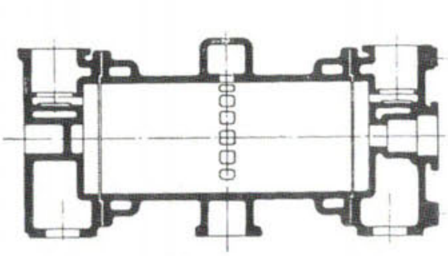


Fig. 12.

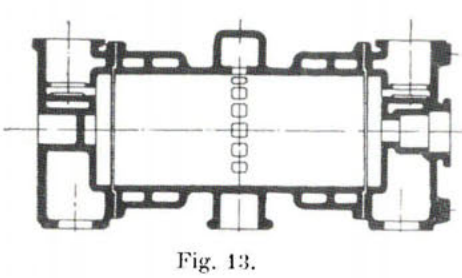


Fig. 13.

which is neither heated nor cooled, lies next to the moderately heated part, and then in the middle of the cylinder there is arranged the cool zone, represented by the exhaust steam belt, which is in the form of an annular casing around the central exhaust ports. In this way the principle of graduating the heating temperature is carried out in such a manner that the heating temperature, from the cover end towards the exhaust ports, decreases with the temperature of the working steam in the cylinder. The cooling jacket in the middle of the cylinder has the practical purpose of obtaining the best possible working conditions for the piston. At this point, where the piston velocity is greatest, the working temperature of the walls is lowest. The temperature of the cylinder wall is so fixed by the heating jackets that the surface temperature at the cover is greater than the temperature of the working medium, and in the middle of the cylinder the surface temperature is below the temperature of the working medium. The efficiency of the heating jacket of a steam cylinder, neglecting all unavoidable subsidiary losses, is given by the difference between the gross advantage of the heating and the loss in heat carried off from the cylinder by the exhaust. If the cover only is heated, a heating jacket is obtained in which, neglecting all unavoidable subsidiary losses, the gross gain is equal to the nett gain, that is to say the loss to the exhaust is nil. In the case of the cover jacket, it may be assumed that the entire heat from the cover is taken up by the steam layer adjacent to the cover. This quantity of steam corresponds very closely to the residual steam trapped in the cylinder immediately after the exhaust ports are closed by the piston on its return stroke. With a cover heating jacket working in such a manner, the steam which flows over the heated surfaces never passes out at the exhaust, so that no heat can ever be lost to the exhaust. The conditions in figure 12 are somewhat less favourable, and still less favourable are the conditions according to figure 13. In these two figures (12 and 13) the quantity of heat, which is carried off to the exhaust and is thereby lost, increases, because part of the steam which passes over the heated surfaces, goes off to exhaust. Experiments have shown that the culmination point, as regards advantage of jacketting, is passed, even for the case of saturated steam, when the heating jacket on the cylinder is extended right up to the exhaust belt. The jacketting is decidedly disadvantageous if it extends over the exhaust zone. The great loss incurred in such a case is easily explained by the great temperature difference and by the great velocity of flow of the exhaust. It is therefore preferable to have, in all cases, a neutral zone, the length of which should be determined according to the temperature of the working steam, that is to say, the neutral zone should be long in the case of highly superheated steam and short in the case of saturated steam.

Tests made with a cover-jacket, as shown in figure 11, showed a fall of 30° C, in the temperature of the superheated steam after passing through the jacket; that is to say, with steam entering at the inlet below with a temperature of 300° C, the temperature at the top, where the steam had completed its heating action in the cover, was 270° C, so that about 15 Centigrade heat units are given up to the working steam in the cylinder for each kilogram of steam used. If it is assumed that this quantity of heat is wholly taken up by the residual steam which is trapped in the cylinder after the piston closes the exhaust ports on its return stroke, or in other words, if it is assumed that none of the jacket heat is carried off by the exhaust, and assuming further that the rate of heat transference from the hot jacket to the working steam is directly proportional to the temperature difference between the jacket and the working fluid in the cylinder, figures 14 and 15 will represent on the entropy-temperature ($\Phi\tau$) diagram the thermic changes which take place. For each kilogram of saturated steam expanding from the initial pressure of 12 atmos.

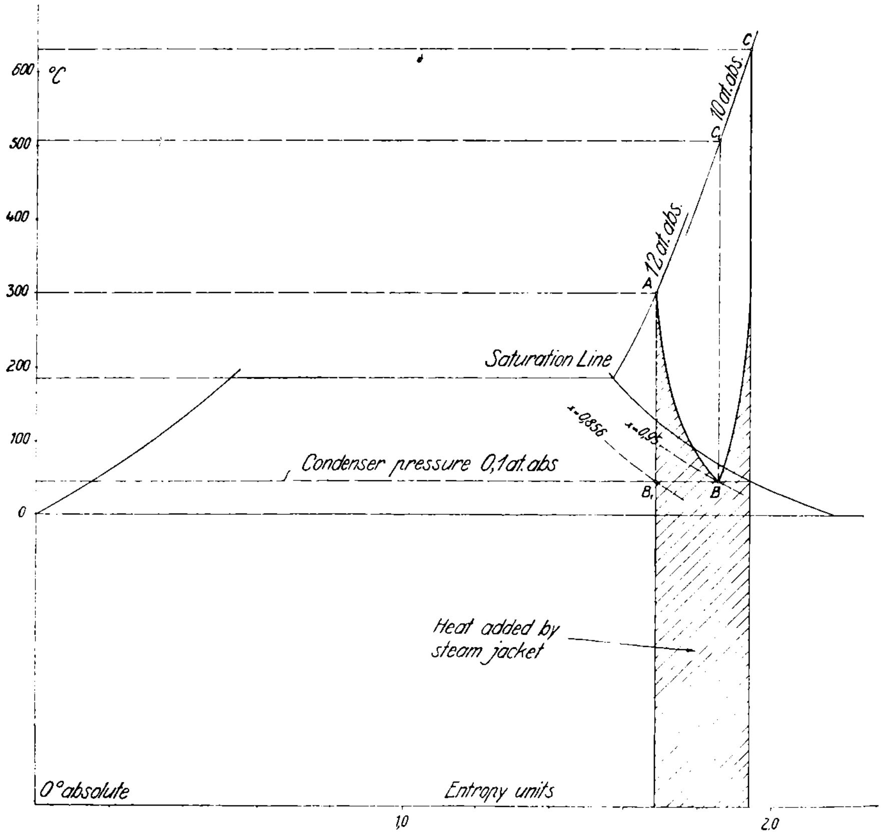


Fig. 14.

to the condenser pressure of 0·1 atmos., the residual steam trapped in the cylinder after exhaust will be 0·122 kg. With steam of the same initial and final pressures, but superheated to 300° C, the weight of the residual steam per kg. of working steam would be 0·142. In order to simplify the representation of the stages of the thermic changes in the diagram, the heat quantities have been worked out for the case of 1 kg of residual trapped steam. These quantities amount, in the case of saturated steam, to 82 C heat units, and in the case of superheated steam to 106 C heat units. With steam at 300° C, the heat transferred during expansion to the mass of steam, which will be trapped in the cylinder at the end of the next exhaust, is given by the line $A\ B$ from which it will be seen that there is a considerable increase in entropy. It is here assumed that the initial pressure is 12 atmos. and the final pressure 0·8 atmos. absolute in both cases, and also that the expansion continues during exhaust so that the point B corresponds to the condenser pressure of 0·1 atmos. The dryness fraction (x) of the residual steam is in the case taken 0·95 (B) instead of 0.856 (B_1). During the next compression stroke more heat is transferred to the steam, and the entropy is correspondingly increased. During this transference of heat the temperature of the compressed steam rises, so that the rate of trans-

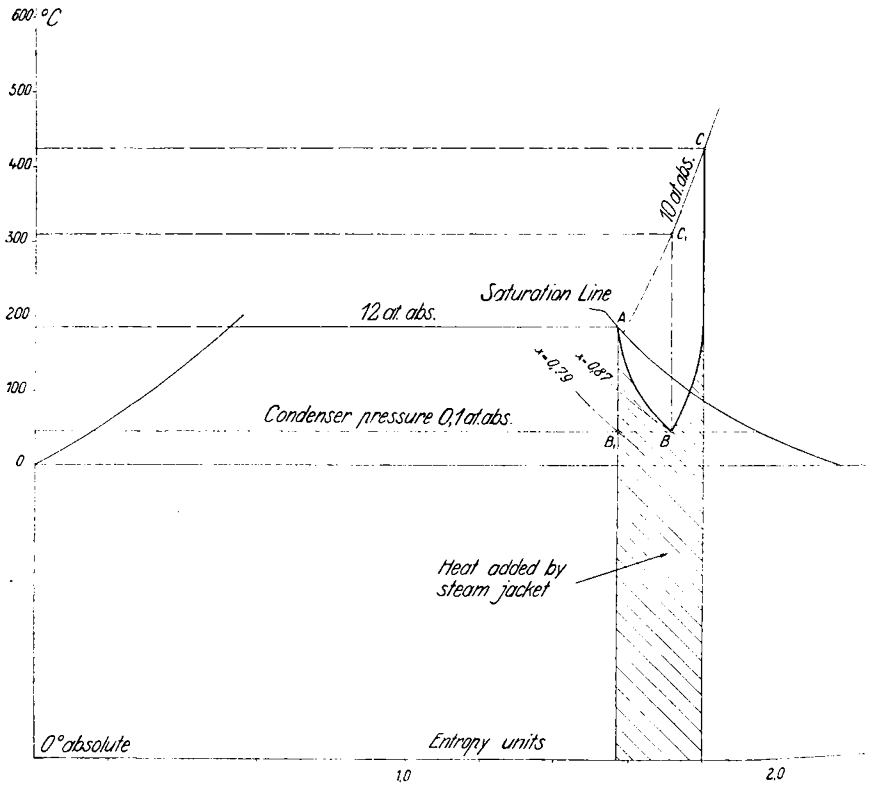


Fig. 15.

ference of heat from the jacket to the compressed steam decreases proportionally to the temperature difference. As soon as the compression temperature rises to the initial temperature (300° C) no further heat is transferred from the hot cover to the residual steam. The further compression is thus pure adiabatic, corresponding to the vertical line to C. The end temperature of compression at C is about 630° C. *The secret of the great efficiency of the una-flow engine resides, to a considerable extent, in this extraordinary rise in temperature.* Due to this great rise in temperature, the internal surfaces are most effectively heated to prepare them for the next charge of steam.

The high temperature at C also counteracts the conditions obtained at B. Thus although, practically, there is 5% moisture present at B, condensation or moistening of the cylinder walls at this point is prevented immediately by the heating action of the jacket and the adiabatic compression from B to C.

In figure 15, the same thermic changes are represented for the case of saturated steam at 12 atmos. initial pressure expanding down to 0·8 atmos. in a una-flow engine with a hot steam cover jacket. At the end of expansion, the steam which will be trapped (i. e. of course the layer nearest the hot jacket) has x (dryness fraction) $= 0{\cdot}87$ instead of $0{\cdot}79$, whilst the end temperature of the adiabatic compression is about 425° C, at the point C. It will be seen therefore that even in the case of saturated steam, there is a very considerable heating of the internal

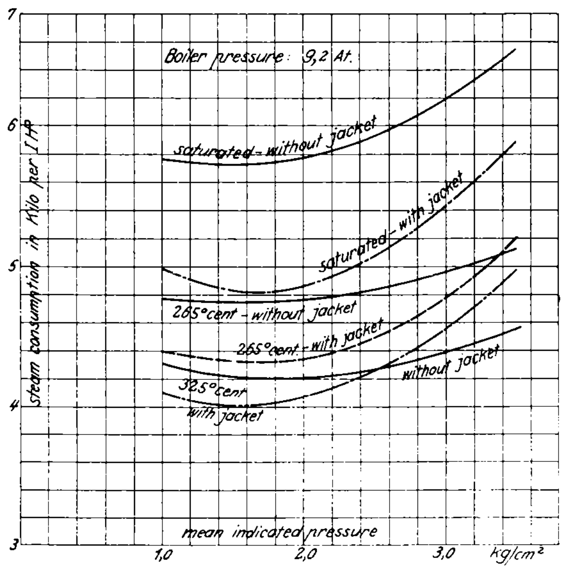


Fig. 16.

surfaces of the cylinder by the compressed steam. The heating action is so considerable that, even in the case of saturated steam, cylinder condensation or moistening of the walls or clearance surfaces is scarcely possible.

The shaded parts of each of the figures 14 and 15 represent the amount of heat transferred from the hot jacket to the working steam, this amount being 82 Centigrade heat units in the case of saturated steam (fig. 15) and 106 Centigrade heat units in the case of superheated steam at 300° C (fig. 14). Although other conditions present may influence the thermic changes which have been described, figures 14 and 15 nevertheless give some idea of the chief outstanding thermic changes, as is confirmed by tests made by Prof. Hubert and M. A. Duchesne described more fully later with reference to figures 33—38.

The above considerations and deductions are in part confirmed by the steam consumption tests of a una-flow engine shown graphically in figure 16. These tests were made with a una-flow steam engine provided with a cover-jacket similar to that which is made the basis of the above investigations. In addition, jackets were arranged over the cylinder at each end, and the centre part of the cylinder was cooled by the exhaust belt. Between the exhaust belt and the cylinder jackets there were neutral zones which were neither heated nor cooled. The cylinder jackets were arranged so that they could be cut out. During the experiments the cover jacket was *always* in use, but the cylinder jackets were sometimes put in and sometimes cut out. The results with the cylinder jackets in use are marked "with jacket"; those where the cylinder jackets were cut out are marked "without jacket". First of all, the results show that the advantage of jacketting is less as the temperature increases. In the case of saturated steam, the surprising difference of 1 kg. in favour of jacketting is obtained, whilst the saving is barely $\frac{1}{2}$ kg. in the case of steam superheated to 265° C, and $^2/_{10}$ ths kg. in the case of steam superheated to 325° C. All the above figures are taken at the points where the steam consumption per H.P. was lowest.

For the case of the most economicl cut-off, (considering also the first cost of the engine) with steam superheated to 325° C, there is no difference in the steam consumption per I. H.P. with and without cylinder jacketting. In all cases the steam used in the jacket was taken into account and added to the steam used in the cylinder.

With a mean pressure of 2·5 kgs. per □ cm, the consumption per I. H.P. with cylinder jacketting was the same as the steam consumption per I. H.P. without jacketting, in both cases the steam being superheated to 325° C. With steam at 265° C the corresponding mean pressure is about 3·4 atmos., that is the mean pressure at the point of intersection of the two curves. The point of intersection of the two curves in the case of saturated steam is still further to the right in figure 16, and corresponds to a still higher mean pressure.

On comparing the steam consumption curves for a una-flow engine with those for a multiple expansion steam engine, it is found that in the case of the una-flow engine the steam consumption does not vary so much with the load. This will be noticed chiefly in the case of the curves without jacketting, where no material variation occurs in the steam consumption between 1 and 3 atmos. mean pressure when the superheat is high. In the curve for saturated steam, little difference in the steam consumption occurs between 1 and 2·4 atmos. when a jacket is provided on the cylinder.

It is to be noted further, that with steam at the moderate pressure of 9·2 atmos. and superheated to 325° C the lowest steam consumption is very nearly 4 kgs. per I. H.P. *This is to say, in the case of a una-flow engine of scarcely 300 H.P. the same steam consumption was obtained as was attained by the 6000 H.P. triple expansion engines of the Berlin Electricitätswerk in Moabit. According to the figures published by Herr Datterer the lowest steam consumption obtained by these engines was 4 to 4·1 kgs. per H.P.*

The corresponding consumptions for the cases taken in figure 16, are represented in figure 17 in heat units. This figure shows that at the most economical cut-off the heat consumption is the same, with moderate superheat to 265° C and cylinder jacketting, as with high superheat at 325° C either with or without jacketting.

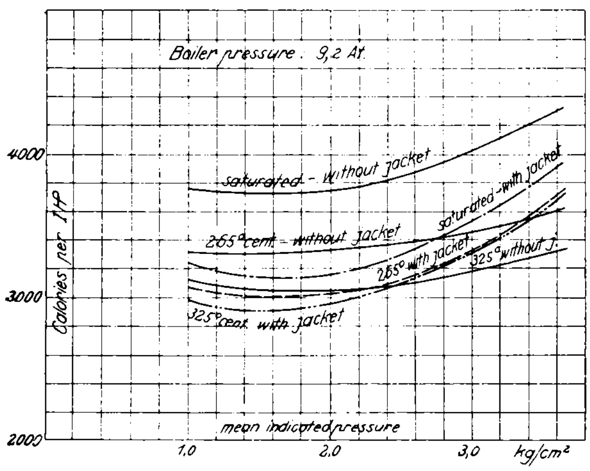


Fig. 17.

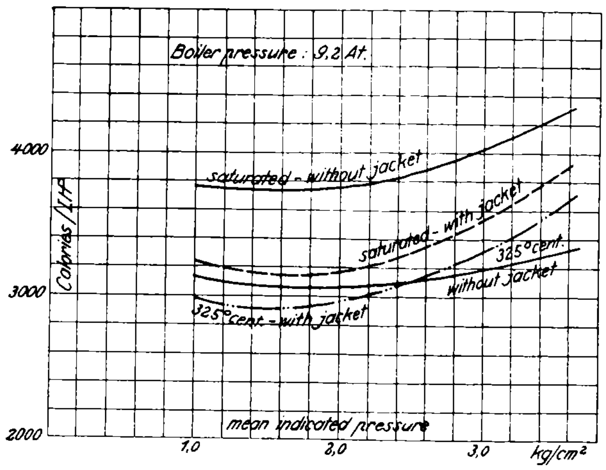


Fig. 18.

In figure 18, the heat curves corresponding to figure 17, are repeated, but the curves for moderate superheat to 265° C, have been omitted. In this case, as in figure 17, all recovery of heat from the water of condensation passing from the jacketting has been neglected. It is nevertheless evident from figure 18, that the minimum heat consumption, in the case of saturated steam working with the cylinder jacket, approximates very closely to the lowest heat consumption in the case of highly superheated steam working without any jacket on the cylinder walls.

In figure 19, the heat curves given in figure 18, are shown as corrected for the heat recovered from the water of condensation passing from the jacket. This figure shows that, when the curves are corrected in this way, the lowest heat consumptions for saturated steam with jacketting, and for superheated steam at 325° C without jacketting, are practically the same in the same engine.

Taking the Berlin conditions, viz, that 1 kg. steam corresponding to 725 Centigrade heat units costs 0·225 pfennigs, and that for superheated steam $^2/_3$ gr of oil per I. H.P. hour is required = 0·05 pfg. (taking oil at 75 marks per 100 kgs.) whilst for saturated steam only $^1/_3$ gr of oil per I. H.P. hour = 0·02 pfgs. (taking oil at 60 marks per 100 kgs.) is required, and calculating out these oil charges in heat units (160 centigrade heat units for superheated and 64 centigrade heat units for saturated steam) the results shown in figure 20 are obtained. It will be seen from figure 20 that the steam and lubrication costs taken together are almost the same at the most economical cut-off in both cases, that is in the case of saturated steam working with cylinder jacketting, and superheated steam without cylinder jacketting. In both cases of course there is cover-jacketting. *The above clearly shows that with a una-flow engine it is almost immaterial, as regards economy, whether the steam is superheated or not. It should be pointed out, however, that in this case the "saturated" steam was superheated a few degrees to ensure its dryness.*

In the above investigations no account has been taken of the fact that heat radiation and leakage losses in the pipes and steam distributing parts are much higher in the case of superheated steam than in the case of saturated steam. This increase in these losses is due to the decreased density of the superheated steam and to theg reater strain on the working parts. In the engine investigated, all the distributing parts were tight in both cases.

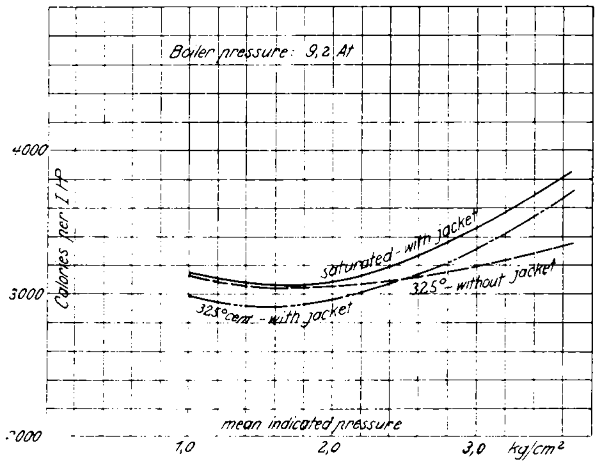


Fig. 19.

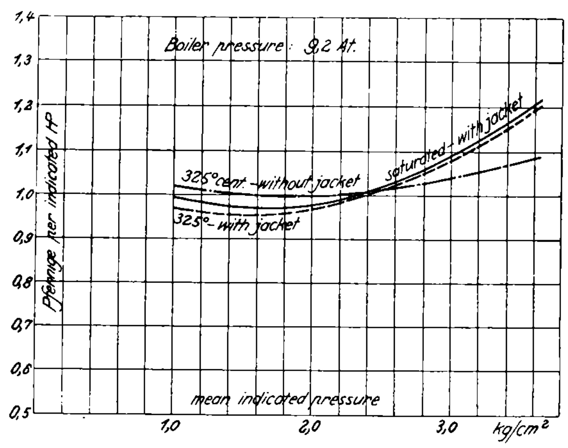


Fig. 20.

It may also be seen from consideration of the character of the adiabatic expansion and compression curves, that the una-flow steam engine must give almost the same economic efficiency with superheated as with saturated steam. This fact leads one to conclude that there is such absence of heat losses that the extraction of energy from the steam is almost exactly represented on the corresponding entropy temperature diagram. A comparison of the entropy temperature diagram for saturated and superheated steam, between the same pressure limits, shows only a very small advantage in favour of superheating. This small advantage may be, however, entirely removed if the heat in the water of condensation, passing from the heating jacket, in the case of the saturated steam cycle, is returned to the boiler to be recovered therein for useful purposes. The great mistake in steam engines of the ordinary construction resides in the counter-flow action. By reversing the flow of the steam there is always considerable cooling of the clearance surfaces by the cold exhaust current, which places these surfaces in the worst