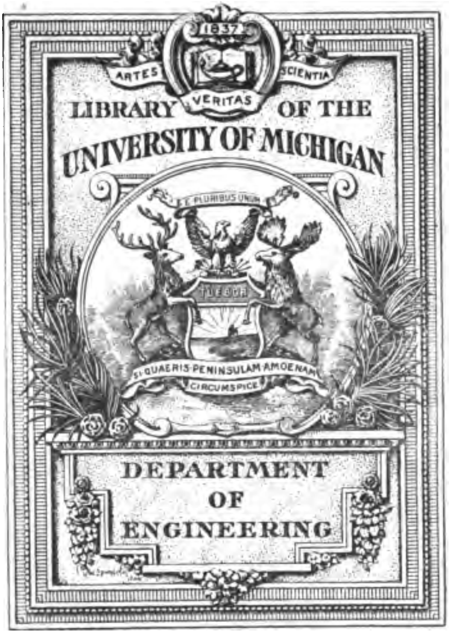
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From the Institution of Civil Engineers. October, 1898

REPORT OF COMMITTEE

ON THE

THERMAL EFFICIENCY OF STEAM-ENGINES.

The Institution of Civil Engineers.

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REPORT OF THE COMMITTEE

APPOINTED ON THE 31st MARCH, 1896,

TO

CONSIDER AND REPORT TO THE COUNCIL UPON THE SUBJECT OF THE DEFINITION OF A STANDARD OR STANDARDS OF THERMAL EFFICIENCY FOR STEAM-ENGINES.

WITH AN INTRODUCTORY NOTE.

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THE INSTITUTION OF CIVIL ENGINEERS.

THE THERMAL EFFICIENCY OF STEAM-ENGINES.

Report of the Committee appointed on the 31st March, 1896, to Consider and Report to the Council upon the Subject of the Definition of a Standard or Standards of Thermal Efficiency for Steam-Engines: with an Introductory Note.

Members of the Committee.

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INTRODUCTORY NOTE.

Prepared at the Request of the Council by Captain SANKEY.

The following Report assumes a competent knowledge of thermodynamics, and some explanation may therefore be of use to those who have not this knowledge, but yet are interested in steam-engine economy. For this purpose the following Introductory Note has been drawn up to show what it has been desired to measure and to compare.

A steam plant in its simplest form normally consists of:

(1) The boiler, with its feed arrangements, to produce the steam.

(2) The pipes to conduct the steam to the engine.

(3) The engine to transform a portion of the steam energy into work upon the pistons.

Further the plant may include:

(4) A condenser with an air-pump or equivalent apparatus, which must be added if the engine is to be relieved from the back pressure due to the atmosphere.

(5) An economiser may be added to the boiler to improve its performance by transmitting to the feed-water some of the heat which would otherwise escape up the flue.

(6) A feed-heater may be used in the case of a non-condensing engine, to recover some of the waste heat in the exhaust steam.

(7) Steam-jackets and reheaters are added to the engine under certain circumstances.

No portion of a steam plant is perfect, and each is the seat of losses more or less serious. If therefore it is desired to improve the steam plant as a whole, it is first of all necessary to ascertain separately the nature of the losses due to its various portions; and in this connection the diagrams in Plate 5 have been prepared, which it is hoped may assist to a clearer understanding of the nature and extent of the various losses.

The boiler; the engine; the condenser and air-pump; the feed-pump and the economiser, are indicated by rectangles upon the diagram. The flow of heat is shown as a stream, the width of which gives the amount of heat entering and leaving each part of the plant per unit of time; the losses are shown by the many waste branches of the stream. Special attention is called to the one (unfortunately small) branch which represents the work done upon the pistons of the engine.

It is obvious that with so many channels of loss, it is of the utmost importance that the circumstances under which they arise, and their amount, should be ascertained and considered separately. It is also especially useful to ascertain the exact amount of those losses which are inevitable according to the laws of Nature, and to distinguish them from other losses, also to some extent inevitable but yet capable of reduction by improvement in design or in material. It is shown in the diagram, by the stream which goes from the engine towards the condenser, that by far the greatest loss occurs in the engine; and this circumstance, coupled with the fact that the design and manufacture of steam-engines form a distinct and almost separate branch of engineering, give ample ground for isolating the engine and considering it apart from other portions of the steam plant.

For this reason the Report does not deal with the steam plant as a whole, but only with the steam-engine proper, which is defined in the Report (p. 9) as being "everything included between the boiler side of the engine stop-valve and the exhaust flange."

Much diversity of opinion has existed as to the method of stating the thermal economical value of an engine, and the standard with which it should be compared. The work of the Committee has been directed to these points.

The upper diagram in the Plate represents, in the main, the trial of the Louisville Leavitt Pumping Engine[1] as described in the Transactions of the American Society of Mechanical Engineers, Vol. XVI., and the data are sufficiently full to enable the flow of heat to be stated at all points with fair accuracy. The numerals represent British thermal units flowing per minute; the streams are drawn to such a scale that each inch of width represents a flow of 100,000 B.T.U. per minute. Temperatures are also given.

Starting at the fire-grate it is shown that 183,600 B.T.U. are produced per minute by the combustion of the coal, and that 131,700 of these go direct into the water of the boiler, 10,000 are lost by boiler radiation and leakage, and the remainder, viz., 41,900, pass away with the flue gases. On the way to the economiser, 1,000 B.T.U. are lost by radiation, but in the economiser itself 15,750 B.T.U. are diverted into the feed-water, 5,000 B.T.U. are dissipated by radiation, and finally 20,150 B.T.U. pass out of the economiser and into the chimney, and are lost to the steam plant. The heat entering the economiser with the feed-water is 5,450 B.T.U., which is added to the 15,750 B.T.U. diverted from the flue gases, thus giving a flow of 21,200 B.T.U. in the feed out of the economiser. Radiation however reduces this flow to 20,950 B.T.U. per minute at the entry to the boiler, where a further addition is made of 6,600 B.T.U. returned by the jacket water.

The steam produced by the boiler is thus seen to derive its heat from three streams, as shown on the diagram; the steam finally leaves the boiler with 159,250 B.T.U. per minute. Before this heat gets to the engine, however, 3,100 B.T.U. are lost by radiation and leakage from the steam-pipes, so that the flow of heat is reduced to 156,150 B.T.U. per minute, which is the gross supply of heat to the engine; the net supply is less, because there are certain returns of heat to the boiler to be deducted. In the first place credit has to be given to the engine for the heat which could be imparted by means of the exhaust steam to the feed-water, inasmuch as the exhaust is theoretically, and very nearly practically, capable of raising the temperature of the feed to the exhaust temperature. On this basis, 7,400 B.T.U. should be credited to the engine, although the actual return to the boiler,

[1] This trial is also the subject of Example No. 4 in Appendix VI of the Report of the Committee.

or rather to the economiser, is only 5,450 B.T.U. The difference is due to excess of circulating water, which lowers the hot-well temperature, and also to radiation, in the feed arrangements, which are a necessary portion of the steam plant, independent of the engine proper, and which are not applied to correct any fault of the engine. The loss of 1,950 B.T.U. should therefore be debited to the feed and condensing arrangements, and not to the engine. It will be seen that the 7,400 B.T.U. credited to the engine is equal to the water-heat of the feed at the temperature of the exhaust, which is in agreement with the rule given at p. 10 for calculating the "heat supplied to the engine," and also with the third recommendation of the Committee (p. 21). In the present case a special credit has to be given to the engine, because 6,750 B.T.U. leave the engine in the jacket water to return to the boiler; but 150 are lost by radiation on the way, so that the net return is 6,600 B.T.U. Only the net return to the boiler should be credited to the engine, because the jackets are applied to correct a fault of the engine, namely, cylinder condensation; hence any loss to the steam plant that may be entailed by the use of jackets (*i.e.*, the radiation in question) should be reckoned as a loss due to the engine, although it takes place outside the engine. In the case under consideration the jacket water was returned direct to the boiler; but if, as is frequently the case, the jacket water is drained into the hot well, the heat in it cannot be credited to the engine, for the reasons given above.

The net supply to the engine can therefore be obtained as follows:—

	B.T.U.	B.T.U.
Gross supply		156,150
Less returnable by feed	7,400	
„ returned by jackets	6,600	
		14,000
Net supply per minute		142,150

On referring to the Plate it will be seen that only 27,260 B.T.U. are utilized as work upon the pistons. The ratio of the heat so utilized to the net heat supplied is called in the Report the "Thermal Efficiency of the Engine,"[1] and it is to be observed that this is the usual meaning applied to this term, although some writers, notably Mr. Willans, applied the term to another

[1] Report, p. 9.

ratio, as mentioned on p. 8. In the numerical case under consideration the thermal efficiency is $\frac{27{,}260}{142{,}150} = 0 \cdot 15$.

The engine under discussion indicates 643 HP.; the heat supplied "per minute per I.HP." is therefore $\frac{142{,}150}{643} = 221$ B.T.U., and it is this figure which, as recommended in the Report,[1] should be used to express the thermal economical value of a steam-engine. This recommendation is made with a view of replacing the present usual method of stating the performance in lbs. of feed-water per I.HP. per hour, although the latter statement is useful for other purposes.

It will be seen that 1870 B.T.U. are deducted from the 27,260 utilized as work on the pistons, as engine friction, so that the effective or brake power of the engine corresponds to 25,390 B.T.U. per minute; in other words, the brake HP. of the engine is 599—so that the "B.T.U. per minute per B.HP." are $\frac{142{,}150}{599} = 237$—another figure which the Committee recommends should be quoted whenever it is possible to ascertain the brake efficiency.[2]

Let it now be supposed that the actual steam-engine is replaced by an ideal steam-engine, forming part of an idealized steam plant working according to the "Rankine cycle"; using for this cycle the name recommended by the Committee.[3] It must be assumed that the I.HP. of the "ideal" engine is the same as that of the "actual" engine, namely 643 HP., and that the admission and exhaust temperature are the same, namely, 359° F. and 100° F. respectively. The flow of heat in the ideal steam plant is represented by the lower diagram in the Plate, and it will be observed that all the losses have disappeared, except the flue loss and the heat carried away by the condensing water. There are no jackets, but there is an economiser, because although the boiler is perfect, 31,400 B.T.U. are sent into the flue, since the gases cannot be cooled down below the temperature of the water in the boiler, and a portion of this heat can be saved by heating the feed to the steam temperature. The heat returnable by the feed is, of course, less than in the case of the actual engine, owing to the reduced weight of steam passing through the engine. The heat supplied to the ideal engine is therefore:—

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[1] Report, p. 16. [2] Report, p. 16.

[3] Report, footnote 1, p. 11.

	B.T.U.
Gross supply	100,900
Less returned by feed	5,600
Net heat supplied per minute	95,300

The I.HP. of the ideal steam-engine being the same as that of the actual engine, the same number of B.T.U. per minute are utilized as work upon the pistons, namely, 27,260. The thermal efficiency of this ideal steam-engine is therefore $\frac{27{,}260}{95{,}300} = 0 \cdot 285$, and the B.T.U. per minute per I.HP. are $\frac{95{,}300}{643} = 148$, as against 221 for the actual engine. The actual engine, therefore, requires 73 B.T.U. per minute per I.HP. more than the ideal engine; these heat-units are lost on account of imperfections in the actual engine, and can be looked upon as a measure of these imperfections.[1] The ideal steam-engine thus becomes the "standard of comparison," as recommended and defined by the Committee.[2]

The number of B.T.U. per minute per I.HP. required by the ideal engine depends on the temperature of the steam at admission and its temperature at exhaust, and also on whether the steam is saturated or superheated. *Figs. 3* and *3a* in the Report have been drawn to enable this number of B.T.U.'s or of Calories to be read off directly in all cases likely to occur in practice.

If the number of B.T.U. per minute per I.HP. required by the ideal engine be divided by the number needed by the actual engine, there is obtained what has been called the "efficiency ratio" by the Committee. In the numerical case under consideration this ratio is $\frac{148}{221} = 0 \cdot 67$.

In order to connect this term with recent writings on the steam-engine, it should be stated that Mr. Willans called this ratio the "thermal efficiency;" he, however, employed exactly the same standard of comparison as that now recommended by the Committee. Professor Osborne Reynolds has called this ratio "the percentage of theoretical efficiency," but his standard of comparison was different. Captain Sankey has called this ratio the "standard thermal efficiency," and has also suggested a somewhat different standard of comparison.

In conclusion, the result of the Committee's work can shortly be said to consist in the recommendation that the thermal economical value of steam-engines should be stated in B.T.U. per I.HP. per minute; in defining the standard engine of comparison; and in giving precise directions for the observations to be taken, and for the calculations to be made, in regard to the foregoing.

[1] Report, p. 16. [2] Report, p. 11.

REPORT OF THE COMMITTEE.

(*Adopted by the Council, 26 April, 1898.*)

On the 12th April, 1897, your Committee submitted a preliminary report which embodied the gist of the conclusions they had arrived at. This report is given in Appendix I.

Your Committee have now the honour to forward their final report.

There are two methods of stating the thermal economical value of a steam-engine.[1] In the first, which is principally used for scientific purposes, it is expressed as a ratio, and it is in this connection that a so-called "standard" of thermal efficiency is needed; in the second and more ordinary method, an absolute statement is used, such as lbs. of feed-water per I.HP., or heat-units supplied per I.HP. per unit of time.

Thermal Economical Value Expressed as a Ratio.

There are several such ratios relating to quite distinct orders of ideas, and the term "thermal efficiency" has been indiscriminately used for all of them. For instance, the term has been applied by some authors to the ratio which the work done upon the pistons of the engine, and shown by the indicator, bears to the heat supplied to the engine; by others to the ratio of the performance of the engine, as above defined, to that of some ideal engine; and by others even to some quantity not a ratio at all, although more or less representing an economical result.

With the view of avoiding risk of misunderstanding in future, and of obtaining the great advantage which would result from the adoption of a uniform system of nomenclature, your Committee submit and recommend for use the following definitions:—

The expression "thermal efficiency" as applied to any heat-engine should mean the ratio between the heat utilized by that engine and the heat supplied to it, or

$$\text{thermal efficiency} = \frac{\text{heat utilized as work on the pistons}}{\text{heat supplied to the engine}}.$$

In the case of any particular actual steam-engine this ratio can

[1] For the purposes of this enquiry a steam-engine will be taken to include everything between the boiler side of the engine stop-valve and the exhaust flange, so that neither the boiler, the feed arrangements, nor the condenser, if there be one, are taken into consideration.

be determined by calculations based on actual experiment in the following manner:—

The heat-units utilized as work are to be obtained by measuring the indicator diagrams of the engine in the usual way to ascertain the mechanical work done.

The heat supplied to the engine is to be calculated as the heat required to produce at constant pressure the steam that enters the engine, from water at the temperature of the engine exhaust.[1] This rule applies whether the steam be saturated or superheated; also whether the engine be non-condensing or condensing.[2]

In the case of an ideal engine working between certain temperatures according to some theoretical cycle, the term "thermal efficiency" will equally represent the ratio of heat utilized to heat supplied.

For many important purposes it is required to know both the thermal efficiency of a particular engine and the calculated thermal efficiency of some ideal engine more or less closely corresponding to it. The ratio between these two efficiencies is a matter of special importance. This ratio between two efficiencies has often been itself called an efficiency, but your Committee think that this use of the word has led to no little confusion, and consider it undesirable.

The ideal steam-engines thus contemplated are the "standards of thermal efficiency" which form the subject of the reference to your Committee; but, as in the opinion of your Committee the use of the words "thermal efficiency" is undesirable in this connection, they recommend that such an ideal steam-engine be called the "standard steam-engine of comparison," and that the ratio of the thermal efficiency of any actual steam-engine to the corresponding standard steam-engine of comparison be called the "efficiency ratio."

It will be seen from the foregoing that two ratios are required to fully express the thermal economical value of a steam-engine,

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[1] The above is equivalent to the following statement:—The heat supplied to the engine per minute is equal to the total heat of the steam entering the engine less the water heat at the temperature of the engine exhaust of the same weight of water. The temperature of the engine exhaust is taken because, theoretically, the feed can be raised by the exhaust to this temperature, although the result is not usually realised in practice.

[2] Special cases occur when steam is abstracted from the engine before final exhaust to heat the feed or when the steam is "reheated" in the receivers. Subtraction or additions to the heat-units supplied must then be made, as shown in Appendix VI.

namely, the thermal efficiency, which gives the proportion of the heat utilized as work upon the pistons to the heat supplied, and the efficiency ratio, which is the measure of the extent to which the actual engine fails to reach the physical possibilities of the conditions under which it is working.

The next step is to decide what the standard steam-engine of comparison should be.

STANDARD STEAM-ENGINE OF COMPARISON.

As is well known, a heat-engine working on the Carnot cycle would give the maximum utilization of the heat-supply; but it is impossible for a steam-engine to work on this cycle when the working substance is saturated steam, unless it be provided with a dynamical feed-heater, an arrangement probably outside any future development of the steam-engine. For an engine using superheated steam produced at constant pressure, the Carnot cycle is theoretically impossible, even with a dynamical feed-heater.

An ideal steam-engine working on the cycle defined by Rankine, Clausius and others[1] gives the maximum utilization of the heat-supply when the working substance is steam, either saturated or superheated, when produced at constant pressure, and is to be regarded as the perfect steam-engine. In this connection it is to be observed that every working substance has its own cycle of maximum utilization of the heat-supply depending on the method in which heat is applied to it.

Actual steam-engines fall far short of the perfect steam-engine, owing principally to incomplete expansion, to the use of conducting materials for cylinders, and to the clearances that have to be observed for mechanical and other reasons. Ideal cycles can be devised to take account of these practical necessities for each particular type of engine, but if such a cycle were devised for general use, arbitrary limitations would be required, which your Committee do not see their way to recommend.

For saturated steam, within the temperature limits at present in use, the difference between the thermal efficiency of the Carnot cycle and of the Rankine cycle is inconsiderable. For instance, if the higher temperature limit is 370° F. and the lower temperature limit is 110° F., the thermal efficiency of the Carnot cycle is 0·313

[1] This ideal steam-engine has often been called the Clausius engine, but, as appears in Appendix VII, Rankine was the first to describe its cycle. Your Committee therefore prefer to call this cycle the Rankine cycle, and it is defined on page 12.

and the thermal efficiency of the Rankine cycle is 0·283. If the lower limit of temperature be increased to 212° F. (as in a non-condensing engine) the thermal efficiency of the two cycles is still closer, viz., 0·19 and 0·178. In view of the fact that the efficiency ratio rarely reaches 0·7 in the case of a condensing engine, or 0·8 with a non-condensing engine, there would seem to be little practical objection to using the Carnot cycle instead of the Rankine, while there would be advantage in the former in simplicity of calculation.

For superheated steam, however, there is a much greater difference between the thermal efficiencies of the two cycles. For instance, if the pressure is 185 lbs. per square inch absolute, and the admission temperature of the steam 650° F., and the lower limit of temperature is 100° F., it will be found that the thermal efficiency of the Carnot cycle is 0·495 while the thermal efficiency of the Rankine cycle is only 0·31. Comparison with the Carnot engine might in this case give rise to undue expectations as to the possibilities of superheated steam.[1]

After careful consideration, your Committee recommend an ideal steam-engine working as part of the Rankine cycle as the standard steam-engine of comparison, both for saturated and for superheated steam-engines.

The definition of the Rankine cycle is as follows:—It is assumed that all the component parts of the steam plant are perfect, and that there are no losses due to initial condensation, leakage, radiation, or conduction, and that there is no clearance in the cylinder. The feed-water required is taken into the boiler at the exhaust temperature, and its temperature is gradually raised until that corresponding to saturated steam is reached. Steam is then formed at constant pressure until dry saturated steam is produced,

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[1] The following numerical example also throws some light on this point. Suppose that an engine working with superheated steam of 120 lbs. per square inch absolute pressure and 300° F. superheat at admission is stated to require 7·6 lbs. of feed-water per I.HP. per hour. The temperature of the exhaust is 120° F. Can the statement be true? On calculation it is found that the thermal efficiency of the engine is $\frac{42 \cdot 4 \times 60}{1,242 \times 7 \cdot 6} = 0 \cdot 27$. The thermal efficiency of the Carnot cycle for the limiting temperatures of 641° F. and 120° F. is 0·473, so that the actual engine only performs 57 per cent. of what the corresponding Carnot cycle engine could do. But the thermal efficiency of the Rankine cycle for this superheated steam is 0·268, so that the actual engine is claimed to do over 100 per cent. of what the Rankine cycle engine could do, or, in other words, does more than is physically possible. Clearly the Carnot cycle helps little in discovering a misstatement of this kind, whilst the Rankine cycle exposes it at once.

after which, if the steam is to be superheated, heat is added at constant pressure and at increasing temperature, until the required temperature of superheat is reached. The steam is introduced into the cylinder at constant pressure, displacing the piston, and performing external work equal to the absolute pressure multiplied by the volume swept through by the piston up to the point of cut-off. Beyond that point expansion takes place adiabatically, doing work until the pressure in the cylinder is equal to the back pressure against which the engine is working. The steam is then completely exhausted from the cylinder at constant pressure corresponding with the lower limit of temperature, work being done on the steam by the engine during exhaust, equal to the absolute back pressure multiplied by the total volume swept through by the piston. The steam is thus removed from the cylinder and the cycle is complete. This definition is exhibited on the entropy chart in *Figs. 1* and *2*.

The heat supplied to the standard engine of comparison is shown by the area aA_1ABb for saturated steam, and by the area aA_1ABSs for superheated steam, and the heat utilized is indicated by the shaded area in each figure. To calculate the heat supply in any particular case, the only data required are the higher and lower limits of temperature, and for superheated steam the admission pressure as well.[1]

Higher Limit of Temperature (t_a) *and of Pressure.*—In the ideal Rankine cycle engine just described, the temperature and pressure are the same in the cylinder at cut-off as in the boiler or superheater; but in an actual engine, the temperature and pressure are always less in the cylinder than in the boiler. There are five different places at which the measurements can be taken:

(1) The boiler.
(2) The stop-valve.
(3) The valve-chest.
(4) The cylinder during admission.
(5) The point of cut-off.

The temperature and pressure of steam in the boiler, owing to the losses in the connecting steam-pipes, are of necessity higher than at the stop-valve of the engine, and it would be unfair to the engine to reckon these losses against it. Moreover cases may arise where the steam is superheated in a separately fired superheater close to the engine. It is therefore undesirable to measure the upper limit of temperature and pressure at the boiler.

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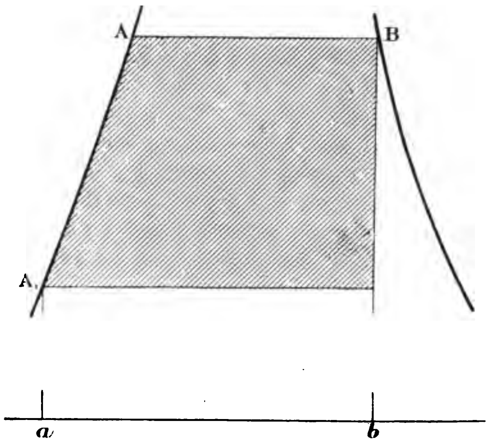
[1] The formula for the thermal efficiency of the ideal steam-engine working according to the Rankine cycle is given on p. 17.

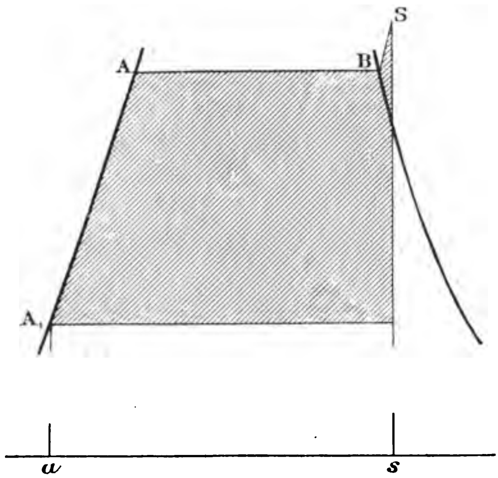
On the other hand, to take the temperature and pressure in the cylinder during admission, and still more to take them at the point of cut-off, would favour the engine by ignoring some of its defects.

Fig. 1.



ENTROPY OR $\theta\phi$ DIAGRAM SHOWING RANKINE CYCLE FOR SATURATED STEAM.

Fig. 2.



ENTROPY OR $\theta\phi$ DIAGRAM SHOWING RANKINE CYCLE FOR SUPERHEATED STEAM.

Taking the temperature and pressure at the valve-chest also favours the engine. In many cases there are considerable variations of pressure in the valve-chest during the stroke, as was shown by experiments undertaken for the purpose and described in Appendix II.

Your Committee, therefore, recommend that the upper limit of pressure and temperature be taken close to, but on the boiler side of, the stop-valve.

Lower Limit of Temperature (t_e).—It has been less easy to come to a conclusion as to the lower limit of temperature, many views having been expressed on this subject, for the most part tending to fix the limit, either in accordance with the conditions affecting the exhaust of the actual engine, or at some constant but arbitrary value.[1]

Experiments were made to ascertain the degree of accuracy with which the temperature of the steam in the exhaust of an engine could be measured. These are described in Appendix IV, and the conclusion drawn from them is that this temperature can be obtained with sufficient accuracy, either directly by a thermometer placed in the exhaust pipe, or indirectly by observing the pressure in the exhaust pipe and taking the corresponding steam temperature, provided a vacuum gauge of the mercurial barometric type, or an indicator, be used to measure this pressure.

After a considerable amount of discussion, your Committee decided to recommend that the lower limit of temperature should be measured in the exhaust pipe outside, but close to, the engine.

This recommendation covers the special cases of an engine working against a high back pressure, and of an engine working, at a considerable terrestrial elevation, in an atmosphere of low barometric pressure.

Thermal Economical Value expressed in Heat-Units.

Referring to p. 9, the thermal economy expressed in heat-units has now to be considered. It has been frequently pointed out, even as far back as 1881 by Mr. Mair-Rumley,[2] that the usual method of expressing the thermal economy of a steam-engine in lbs. of feed-water per hour is inaccurate, because the number of heat-units required to produce a pound of steam depends upon the circumstances of its production. The error is unimportant so long as we are dealing with saturated steam, but when using super-

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[1] The examples given in Appendix III illustrate numerically the effect of various lower limits of temperature.

[2] "On the Independent Testing of Steam-Engines, etc." Minutes of Proceedings Inst. C.E., vol. lxx. p. 313.

heated steam the error may be as much as 10 per cent. or even 15 per cent., and cannot therefore be neglected.

If the heat supplied to an engine per minute, as calculated by the rule given on page 10, is divided by the indicated HP., we obtain a number which is a true measure of the thermal economy of the engine, namely, the number of British thermal heat-units supplied per minute per Indicated HP., and a similar number can be obtained per Brake HP. Further, the B.T.U. required per Indicated HP. by the standard steam-engine of comparison can also be calculated, if the limiting conditions are given.

The complete statement can be tabulated as follows, two numerical examples being added by way of illustration:—

		Numerical Examples.	
	DATA.		
	(1) Stop-valve pressure (absolute) lbs. per square inch	158	185
	(2) Stop-valve temperature, °F.	362	650
	(3) Exhaust temperature, °F.	213	100
	(4) I.HP.	500	1,300
	(5) B.HP.	450	1,155
	(6) Weight of steam entering engine per minute, lbs.	171	222
	RESULTS.		
Actual Engine.	(7) B.T.U. supplied per minute per I.HP.	345	215
	(8) " " " B.HP.	385	242
Standard Engine of Comparison.	(9) B.T.U. theoretically required per minute per I.HP.	250	136
	(10) Efficiency ratio	0·725	0·633

The "B.T.U. supplied per minute per I.HP.," together with the "B.T.U. supplied per minute per B.HP.," give all the information as regards economy needed by the user, and correspond with the "feed-water per I.HP." and "per B.HP." at present in use. The addition of the B.T.U. per minute required by the standard engine of comparison gives the further information needed for scientific purposes.

It will be seen that the number of B.T.U. lost by imperfections in the actual engine can be obtained by the subtraction of line 9 from line 7, which is a valuable and suggestive piece of information.

The estimation of the B.T.U. supplied per I.HP. from the usual observations made at trials is easy. For instance, suppose the following observations are made with an engine using saturated steam:—

I.HP.	520
Weight of steam entering engine per minute	108·3 lbs.
Stop-valve pressure	170 lbs. per sq. in. abs.
Exhaust pressure	2 lbs. " "

The heat supplied per lb. of feed, as defined on page 10, is 1,194 − 94 = 1,100 B.T.U. Hence the B.T.U. per minute per I.HP. $= \dfrac{108 \cdot 3 \times 1{,}100}{520} = 229$.

The B.T.U. per HP. for the standard engine of comparison can be calculated as follows:— .

The formula[1] for the thermal efficiency of the Rankine cycle for saturated steam is

$$\frac{\left(T_a - T_e\right)\left(1 + \frac{L_a}{T_a}\right) - T_e \text{ hyp log } \frac{T_a}{T_e}}{L_a + T_a - T_e},$$

in which formula the increase in the specific heat of water at higher temperatures affects the numerator and denominator nearly equally.

The B.T.U. per minute per HP. for the standard engine of comparison is 42·4 divided by the thermal efficiency of the Rankine cycle thus:—

For saturated steam, the B.T.U. per minute per HP. for the standard engine of comparison is:—

$$\frac{42 \cdot 4\,(L_a + T_a - T_e)}{\left(T_a - T_e\right)\left(1 + \frac{L_a}{T_a}\right) - T_e \text{ hyp log } \frac{T_a}{T_e}},$$

and similarly for superheated steam it is:—

$$\frac{42 \cdot 4\,\{L_a + T_a - T_e + 0 \cdot 48\,(T_{as} - T_a)\}}{\left(T_a - T_e\right)\left(1 + \frac{L_a}{T_a}\right) + 0 \cdot 48\left(T_{as} - T_a\right) - T_e\left(\text{hyp log } \frac{T_a}{T_e} + 0 \cdot 48 \text{ hyp log } \frac{T_{as}}{T_e}\right)}.$$

It will be noticed that the usually accepted figure of 0·48 for the specific heat of superheated steam at constant pressure has been taken, although this figure is open to much doubt.

These formulas being somewhat complex, the curves shown in *Fig. 3* have been prepared,[2] from which the B.T.U. per minute per HP. in the case of saturated steam can be read off directly.[3]

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[1] Meaning of the letters used in the following formulas:—

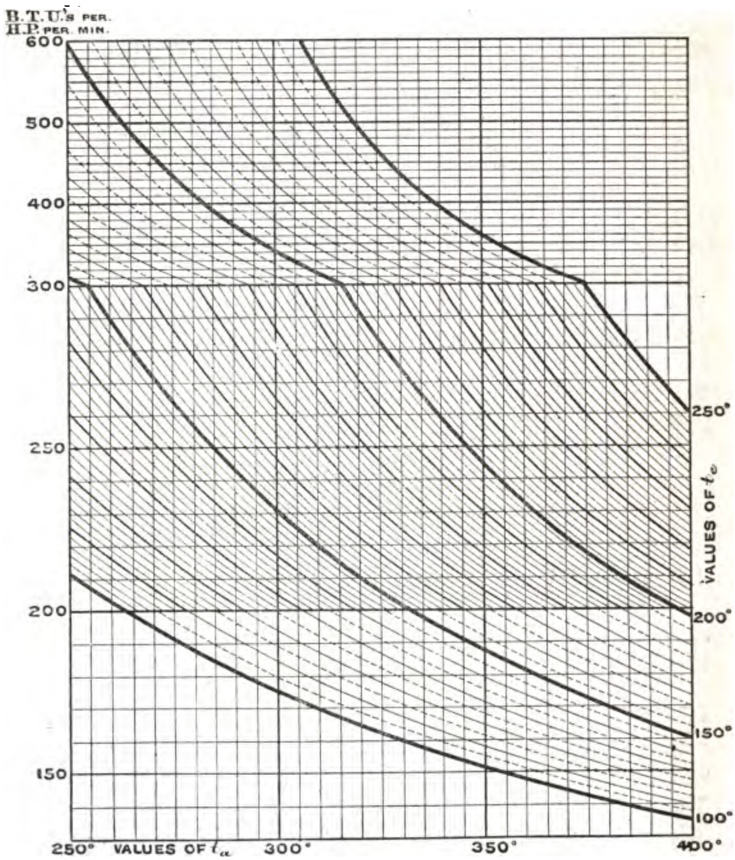
T_a absolute temperature of saturated steam at stop-valve pressure.
T_{as} „ „ of superheated steam at stop-valve.
T_e „ „ in exhaust.
L_a latent heat of steam at temperature T_a.

[2] From *Fig. 3a*, which has been added by request of the Council, the calories per minute per kilowatt can be read off.

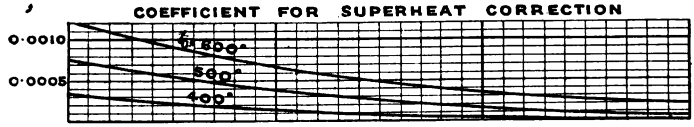
[3] Thus, for $t_a = 350°$ and $t_e = 212°$, the figure is 265.

Fig. 3.

Curves Showing British Thermal Units expended per Minute per I.HP. by the Standard Engine of Comparison, viz., the Ideal Steam-Engine forming part of the Rankine Cycle.



COEFFICIENT FOR SUPERHEAT CORRECTION

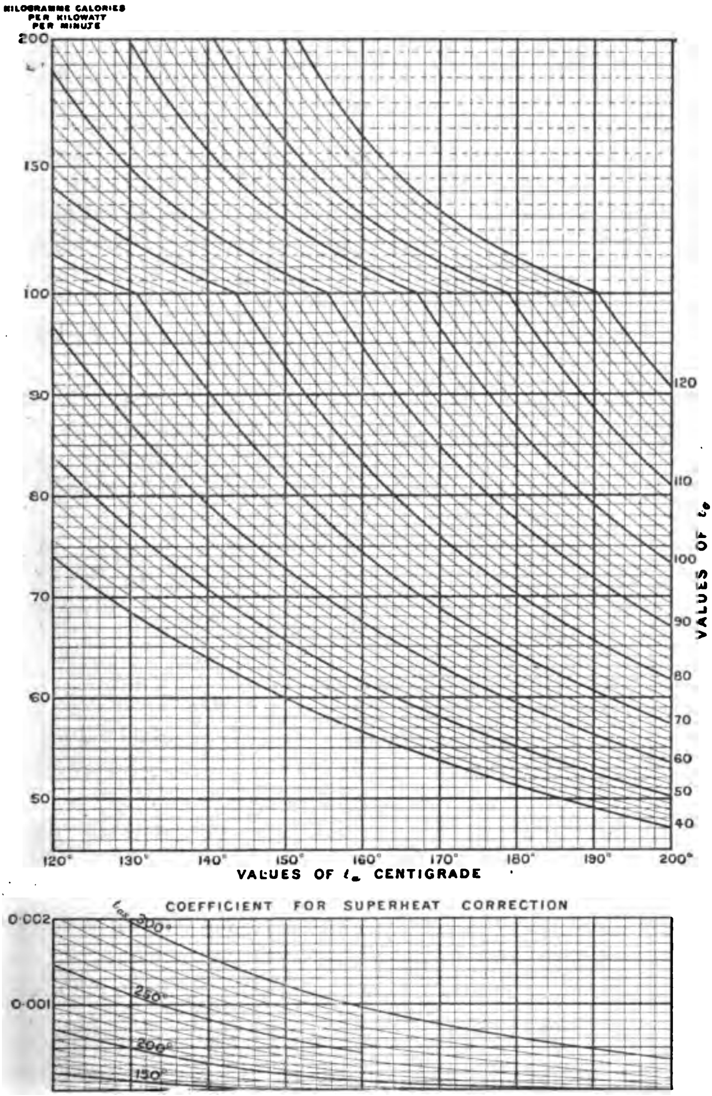


Temperatures are expressed in degrees Fahrenheit.

It will be noticed that the upper and lower portions of the diagram are to different scales; this is in order that the lower and more important part may be read more easily, and accounts for the cusps in the curves.

Fig. 3a.

CURVES SHOWING KILOGRAMME-CALORIES EXPENDED PER MINUTE PER KILOWATT BY THE STANDARD ENGINE OF COMPARISON, VIZ., THE IDEAL STEAM-ENGINE FORMING PART OF THE RANKINE CYCLE.



Temperatures are expressed in degrees Centigrade.

In the case of superheated steam, the figure is obtained by applying a correction to that got for saturated steam. At the bottom of *Fig. 3* is a set of curves marked "coefficient for superheat correction." Against the temperature of saturation corresponding to the stop-valve pressure, and on the curve corresponding to the temperature of the superheated steam (t_{as}), is found a coefficient. This coefficient multiplied by the exhaust temperature, and by the B.T.U. already found, gives the deduction to be made from these B.T.U.[1]

Further numerical examples are given in Appendix III.

It is further to be observed that this method of statement is applicable to all heat engines whether using steam, gas, oil, or air. Calculations in respect of fuel consumption are more readily made than when the feed per I.HP. is given, as will be seen on referring to Appendix V.

Relation between Thermal Economy expressed in Heat-Units and as a Ratio.

The thermal efficiency, as defined on page 9, is proportional to the inverse of the B.T.U. required per minute per indicated HP., and if we divide the energy of 1 HP. per minute expressed in thermal units, namely 42·4, by the B.T.U. required per indicated HP. per minute, we obtain the thermal efficiency.[2]

The efficiency ratio, as defined on page 10, is equal to the B.T.U. supplied per minute per indicated HP. to the actual engine divided by the B.T.U. theoretically required per indicated HP. per minute by the standard engine of comparison.[3]

Recommendations by the Committee.

Having considered the various questions connected with the subject, as detailed in the Report, your Committee recommend :—

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[1] Thus, let $t_{as} = 500°, p_a = 135$ lbs. (so that $t_a = 350°$), and $t_e = 212°$. Against 350°, and on the curve for 500°, we read the coefficient 0·00015. This gives the correction thus: $0·00015 \times 212 \times 265 = 8·5$, and $265 - 8·5 = 256·5$, the number of B.T.U.'s required.

[2] Thus in the first numerical example, p. 16, the thermal efficiency is $\frac{42·4}{345} = 0·123$.

[3] In the numerical example for saturated steam, p. 16, this efficiency ratio is $\frac{250}{345} = 0·725$, and in the example for superheated steam it is $\frac{136}{215} = 0·633$.

1. That "thermal efficiency" as applied to any heat-engine should mean the ratio between the heat utilized as work on the piston by that engine and the heat supplied to it.

2. That the heat utilized be obtained by measuring the indicator diagrams in the usual way.

3. That in the case of a steam-engine, the heat supplied be calculated as the total heat of the steam entering the engine less the water-heat of the same weight of water at the temperature of the engine exhaust, both quantities being reckoned from 32° F.

4. That the temperature and pressure limits, both for saturated and superheated steam, be as follows:—

Upper limit: the temperature and pressure close to, but on the boiler side of, the engine stop-valve, except for the purpose of calculating the standard of comparison in cases when the stop-valve is purposely used for reducing the pressure. In such cases the temperature of the steam at the reduced pressure shall be substituted. In the case of saturated steam the temperature corresponding to the pressure can be taken.

Lower limit: the temperature in the exhaust-pipe close to, but outside, the engine. The temperature corresponding to the pressure of the exhaust steam can be taken.

5. That a standard steam-engine of comparison be adopted, and that it be the ideal steam-engine working on the Rankine cycle between the same temperature and pressure limits as the actual engine to be compared.

6. That the ratio between the thermal efficiency of an actual engine and the thermal efficiency of the corresponding standard steam-engine of comparison be called the efficiency ratio.

7. That it is desirable to state the thermal economy of a steam-engine in terms of the thermal units required per minute per Indicated HP., and that, when possible, the thermal units required per minute per Brake HP. be also stated.

8. That, for scientific purposes, there be also stated the thermal units required per minute per HP. by the standard engine of comparison, which can readily be obtained from a diagram similar to that given in *Fig. 3*, and from which the efficiency ratio can be deduced.

Your Committee would also suggest that in Papers submitted to the Institution bearing on steam-engine economy, Authors be invited to conform to the above recommendations.

[APPENDIXES.

APPENDIXES.

APPENDIX I.

PRELIMINARY REPORT OF THE COMMITTEE APPOINTED TO CONSIDER AND REPORT TO THE COUNCIL UPON THE DEFINITION OF A STANDARD OR STANDARDS OF THERMAL EFFICIENCY FOR STEAM-ENGINES.

Your Committee beg leave to report that they have now practically come to an agreement on the subject of the reference to them, and that the draft report has been drawn up. It was hoped that this report would have been ready to submit to you in time for the Annual Meeting, but unforeseen delays have occurred.

Your Committee, however, wish now to state that the gist of the conclusions they have come to is as follows:—

(1) That the statement of the economy of a steam-engine in terms of pounds of feed-water per I.HP. per hour is undesirable.

(2) That for all purposes, except those of a scientific nature, it is desirable to state the economy of a steam-engine in terms of the thermal units required per I.HP. per hour (or per minute), and that if possible the thermal units required per brake HP. should also be given.

(3) That for scientific purposes the thermal units that would be required by a perfect steam-engine working under the same conditions as the actual engine should also be stated.

The proposed method of statement is applicable to engines using superheated steam as well as to those using saturated steam, and the objection to the use of pounds of feed-water, which contain more or less thermal units according to conditions, is obviated, while there is no more practical difficulty in obtaining the thermal units per I.HP. per hour than there is in arriving at the pounds of feed-water.

For scientific purposes the difference in the thermal units per I.HP. required by the perfect steam-engine and by the actual engine shows the loss due to imperfections in the actual engine.

A further great advantage of the proposal is that the ambiguous term "efficiency" is not required.

Your Committee will do themselves the honour of submitting their complete Report at an early date.

On behalf of the Committee,

(Signed) ALEX. B. W. KENNEDY, *Chairman.*
H. RIALL SANKEY, *Hon. Secretary.*

Dated 12 April, 1897.

APPENDIX II.

REPORT ON EXPERIMENTS MADE IN CONNECTION WITH THE HIGHER LIMIT OF TEMPERATURE (t_a) AND INCIDENTALLY TO DETERMINE THE RELIABILITY OF GAUGE AND INDICATOR READINGS OF STEADY AND VARYING STEAM PRESSURES.

The experiments were made at Ferry Works, Thames Ditton, on the 18th June 1896, by Captain Sankey, and on the 25th September, 1896, by Professor Beare, Mr. Bryan Donkin and Captain Sankey.

Steam was taken from a pipe fed by three boilers, which were supplying steam at the same time to the shop engines and pumps, and to engines under test.

The engine was a single-throw compound Willans central-valve engine, driving a dynamo. It was run at various speeds from 195 revolutions to 480 revolutions per minute, and also at various loads.

The steam passed through a separator before it reached the steam-chest of the engine. Indicators and steam-gauges were fitted both to the separator and to the steam-chest. There was a valve on the boiler side of the separator and also between the separator and the steam-chest. If the former valve was full open and the latter partially closed, the separator formed a reservoir for the engine to draw from. If, on the other hand, the valve on the boiler side of the separator was partially closed, and the other full open, the effective volume of the steam-chest was increased by that of the separator, and the pressure of the steam in it remained far more constant.

The experiments at various loads and speeds consisted in taking indicator cards and gauge readings from both separator and steam-chest simultaneously. The gauge readings were taken, first with the cock full open and the pointer oscillating, the highest and lowest readings being noted; and secondly, with the cock nearly closed so as to steady the pointer, generally a steady reading could be obtained.

When the regulation was effected by the steam-chest stop-valve, the separator indicator diagram was practically a straight line, but the steam-chest diagram was of considerable area, varying but little with the load, but greater the higher the revolutions. *Figs. 4, 5* and *6* are examples.[1]

The separate indicator diagrams have been combined into one diagram with the same atmospheric line.

The pressure acting on the piston up to the point of cut-off has in each diagram been shown by a thicker line, and this pressure, it will be seen, is in many cases far less than the stop-valve pressure.

When the regulation was effected by the separator stop-valve, the diagram of the steam-chest was much reduced in area, owing to the increase of its volume, as illustrated by *Fig. 7*, and there was a small diagram in the separator itself.

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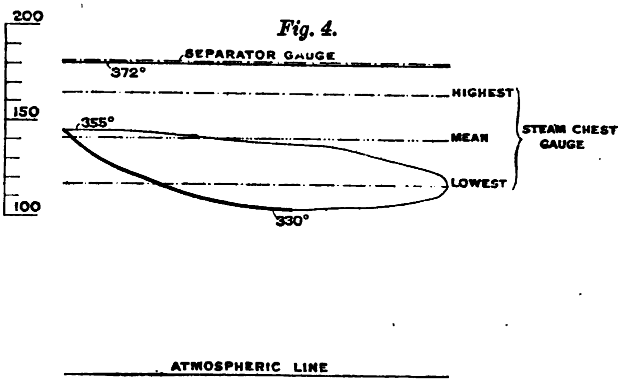
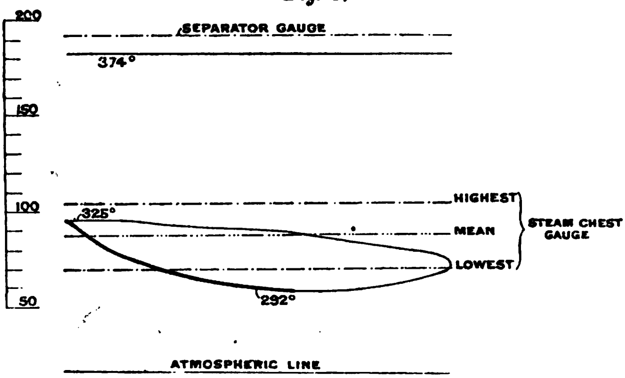
[1] In all the figures the vertical scales represent pressure in lbs. per square inch and the temperatures are given in degrees Fahrenheit.

Fig. 4.



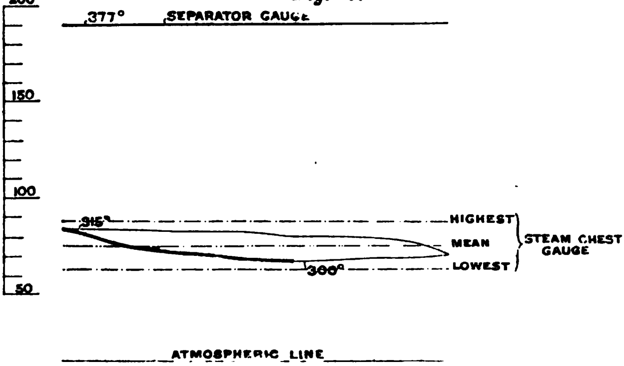
REGULATION BY STEAM-CHEST STOP-VALVE. 452 REVOLUTIONS.

Fig. 5.



REGULATION BY STEAM-CHEST STOP-VALVE. 195 REVOLUTIONS.

Fig. 6.

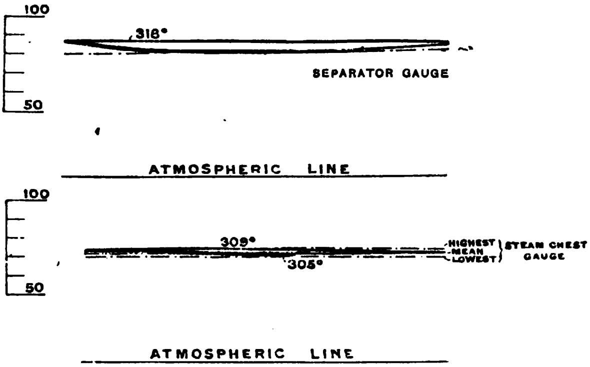


REGULATION BY STEAM-CHEST STOP-VALVE. 480 REVOLUTIONS.

The average pressures of the steam-chest indicator diagrams were obtained as follows:—

A B C D, *Fig. 8*, is the diagram drawn by the indicator pencil, and X Y is the atmospheric line drawn by the same. Verticals to the atmospheric line are drawn at A and C; the planimeter is then run first round the figure A D C Y X, and then round A B C Y X. The reading so got, divided by twice X Y, and multiplied by the pressure scale, gives the "average pressure of indicator

Fig. 7.

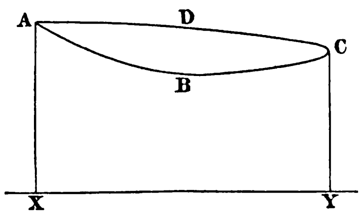


REGULATION BY SEPARATOR STOP-VALVE. 210 REVOLUTIONS.

diagram," shown on *Fig. 9*, after being corrected for the error of the indicator spring, as determined by comparison with a standard.

An endeavour was also made to see whether the steam-gauge reading represented in any way the results obtained by the indicator diagram, but no relation could be discovered. The arithmetic mean of the gauge readings, when oscillating, did not agree with the comparatively steady reading of the gauge

Fig. 8.

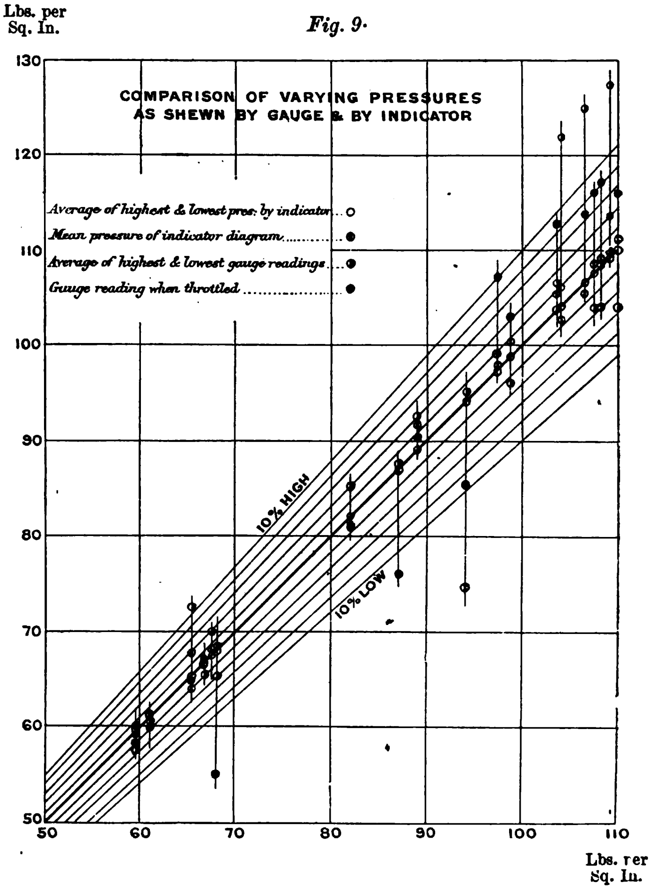


when throttled to prevent oscillation, nor was there any agreement with the average pressure of the indicator card, obtained as just described.

Fig. 9 gives in a graphic form the results of all the experiments in this respect. It is to be observed that both the indicators and the steam-gauges were compared against a standard, so that the want of agreement is not due to errors in calibration. The inference is that steam-gauges are quite unreliable, even if calibrated, for reading rapidly varying pressures.

The diagrams hitherto referred to were taken from a single-acting engine which drew steam from its steam-chest but once in a revolution—an unusual arrangement in practice. *Fig. 10* shows a better state of things, resulting from having two similar single-acting engines drawing steam from a common steam-chest, somewhat as an ordinary double-acting engine does; and *Fig. 11* shows a

Fig. 9.



fair constancy of pressure resulting from having three similar engines with cranks at 120°, drawing steam from one common steam-chest.

Figs. 12, 13 and *14* are steam-chest diagrams, taken by Mr. Bryan Donkin, of a slow-speed engine under varying conditions stated on the *Figs.* It is interesting to note that the steam-chest diagram increases with the smaller loads and earlier cut-off; this is apparently due to the fact that with the earlier cut-off the initial condensation is much increased, causing an increased draught of steam.

Fig. 10.

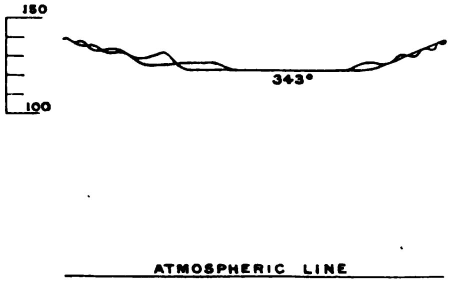


Diagram taken from the Steam-Chest of a two-throw Willans Engine. Revolutions about 450.

Fig. 11.

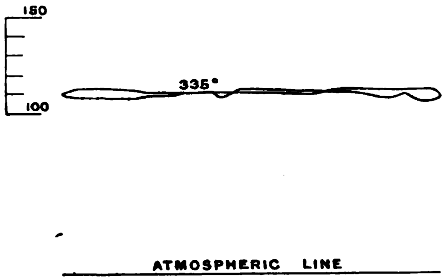
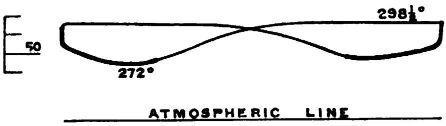


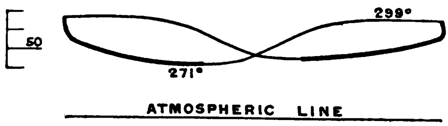
Diagram taken from the Steam-Chest of a three-throw Willans Engine. Revolutions about 340.

Fig. 12.



Beam-Engine driving Shafting only. Cut-off ¼. Revolutions 34. Boiler temperature 298½°.

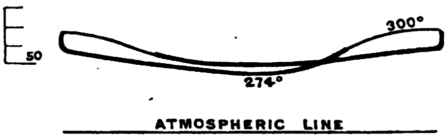
Fig. 13.



Beam-Engine doing ordinary Work. Cut-off ⅜. Revolutions 34. Boiler temperature 300°.

The principal inference to be drawn from all these experiments is, that the stop-valve pressure and temperature should be taken for the upper limit of the

Fig. 14.



BEAM-ENGINE DRIVING FOUNDRY FAN. CUT-OFF ⅜. REVOLUTIONS 35.
BOILER TEMPERATURE 301°.

standard of comparison. The pressure and temperature in the engine is less, as these experiments show, but it is distinctly the fault of the engine or of its valve-gear.

APPENDIX III.

FOURTEEN EXAMPLES OF STEAM-ENGINES WORKING UNDER VARYING CONDITIONS.

Referring to p. 15, the accompanying Table gives an abstract of the working out of some examples of steam-engines working under various conditions. The following points are worth noticing:—

Examples 1 and 2.—Comparing examples 1 and 2, it will be seen that according to col. IV both these engines are doing about 56 per cent. of the theoretical possibility, due to their environment, that is, they both have an efficiency ratio of 0·56. The thermal efficiency of the first example is only 0·12, and in the second it is as much as 0·17. The improvement in work return is due to the higher boiler pressure and better vacuum. If, however, these two examples are compared by the figures given in col. I it would appear that the "Iona's" engine is making better use of its opportunities in the proportion of 58 to 41. The former comparison, viz., 56, is surely the more accurate; but it must not be argued from this that if the "Ville de Douvres" engines were put to work with 167 lbs. stop-valve pressure and 0·7 lb. per square inch exhaust pressure, that they would still have an efficiency ratio of 0·56. It will, of course, be a less ratio, inasmuch as the engines would not then be suitably designed for the new conditions.

Examples 4 and 7.—These examples show that the engine was in each case arranged so as to make the same use of its opportunities, namely, 0·74 and 0·73 efficiency ratio, which is reasonable, seeing that the cut-off had been adjusted by Mr. Willans in each case to give the best results. The thermal efficiency in the first case is, however, greater, owing to the better conditions under which the engine of example 4 is working.

Examples 2 and 6.—The thermal efficiency is greater in example 6 than in example 2, because the engine makes better use of its opportunities, which are not quite so good in example 6 as they are in example 2, owing to the exhaust being at a higher pressure. This example 6, together with example 2, illustrates also the disadvantage of taking the temperature of injection as the lower temperature. If this temperature had been chosen, the efficiency ratio for

Data abstracted from Table issued to the Members of the Committee on the 23rd December, 1896.							Thermal Efficiency.	Efficiency Ratios.				Proposed Method of Comparison in British Thermal Units.		
A	B	C	F	J	K	N		Fixed Lower Limit of Temperature.[1]		Lower Limit of Temperature varying with conditions of Running.[1]				
Case.	Engine.	Reference and Authority.	Temperature of Steam at Stop Valve.	Temperature of Steam in Exhaust Pipe just outside Engine.	Temperature in Hot Well.	Lbs. of Feed-Water per I.HP. per Hour.		I. Non-condensing, 212° F. Condensing, 100° F.	II. All Steam-Engines 100° F.	III. Hot-Well Temperature (col. K).	IV. Temperature in Exhaust just outside Engine (col. J).	B.T.U. per I.HP. per Minute.	B.T.U. per B.HP. per Minute.	B.T.U. per HP. per Minute for Standard Engine of Comparison.
			° F.	° F.	° F.									
1	S.s. "Ville de Douvres"	Dr. Kennedy, Proc. Inst. Mech. E.; Prof. Hudson Beare's Report . . .	(338)	160	160	20·8	0·12	0·41	0·41	0·56	0·56	367	..	205
2	S.s. "Iona" .	Ditto ditto ditto	(367)	91	75	13·4	0·17	0·58	0·58	0·52	0·56	254	..	141
3	Willans (experimental engine)	Willans, Non-Condensing Trials; Min. Proc. Inst. C.E., vol. xciii. . $C\frac{80}{3\cdot 2}$	(322)	213	..	26·2	0·10	0·72	0·34	..	0·74	436	506	321
4		$C\frac{160}{6}$	(370)	212	..	19·2	0·13	0·73	0·40	..	0·74	325	356	239
5	" . .	Willans, Condensing Trials; Excerpt Min. Proc. Inst C.E., vol. cxiv. . $\frac{T\,75}{20\cdot 6}$	(312)	91	(70)	15·3	0·15	0·60	0·60	0·51	0·57	285	458	162
6		$\frac{T\,175}{12\cdot 36}$	(375)	105	(70)	12·7	0·18	0·60	0·60	0·53	0·61	238	268	145
7	Willans II-S Engine, driving "Holmes" dynamo	Dr. Kennedy, Tests made at Thames Ditton on the 20th May, 1893 . . .	(362)	213	..	20·5	0·12	0·72	0·38	..	0·73	345	383	250
8			(362)	193	(190)	(19·6)	0·13	0·40	0·40	0·65	0·66	337	374	220
9			(362)	135	(130)	18·2	0·13	0·43	0·43	0·50	0·51	330	367	168
10	"Milwaukee" pumping-engine	Thurston, Trans. Am. Soc. Mech. Eng., vol. xv. .	350	126	105	11·7	0·20	0·69	0·69	0·71	0·79	213	235	168
11	" H.P.C.	Ditto ditto ditto	350	277	..	38·4	0·07	(0·41)	(0·21)	..	0·80	603	..	481
12	" I.P.C.	Ditto ditto ditto	277	216	..	39·7	0·06	(0·76)	(0·26)	..	0·81	649	..	527
13	" L.P.C.	Ditto ditto ditto	216	126	105	29·3	0·08	(0·62)	(0·62)	0·52	0·64	515	..	328
14	Imaginary steam-engine	.. ..	338	162	160	13·0	0·19	0·64	0·64	0·90	0·91	228	..	208

[1] In all cases the stop-valve temperature was chosen as the higher limit of temperature. Various values were chosen for the lower limit of temperature.

example 2 would have been 0·51, and 0·50 for example 6, which would have made it appear that engine 6 is not quite so good as engine 2, whereas in reality it is better.

Examples 7, 8, and 9, have very nearly the same thermal efficiency notwithstanding the varying conditions as regards back-pressure; the efficiency ratio gives the reason. This particular engine had been designed to give the best results when non-condensing, and its efficiency ratio under these circumstances is 0·73 (columns III and IV). It was then put on the condenser without any change, not even of cut-off, and being then unsuited to its environment gave only 0·66 with a bad vacuum, and only 0·51 with a moderate vacuum. If in this latter case, without changing the design of the engine in any way, but merely altering the proportion of the cylinders and the cut-off, the efficiency ratio when condensing would have risen to about 0·65.

Example 8.—In this example, if treated as a condensing engine with 100° F. for the lower limit of the standard engine of comparison, the efficiency ratio (column I or II) comes out at only 0·40, but when the vacuum improves, and the engine is still less suited to its environment, example 9, a better efficiency ratio of 0·43 is obtained. This is obviously not a helpful comparison.

Example 10 gives the results of one of the best trials on record, and it is interesting to note that even this engine is only able to do 79 per cent. of the possibilities. This may be looked upon as the high-water mark of present engine economy with saturated steam.

Examples 11 and 12 show that the HP. and the intermediate cylinders of this engine are making rather better use of their opportunities that the LP. cylinder. A considerable portion of the loss of the LP. cylinder is of course due to the necessary cutting off of the toe of the diagram in this cylinder.

Example 14.—The efficiency ratio in columns III and IV is given as 0·90 and 0·91. This efficiency ratio is so high that it ought to excite suspicion as to the accuracy of the test. The thermal efficiency is 0·19, which, although high, is not so good as that in example 10. In columns I and II the efficiency ratio (with fixed lower limit of temperature of 100° F.) is only 0·64, and this figure would not call attention to there being anything wrong with the test. If the matter is inquired into a little further, however, and the engine is compared with one working on an ideal cycle, having for its lower temperature limit the temperature of the exhaust in the LP. cylinder, and with the toe cut off at 2 lbs. above the exhaust, it will be found that the efficiency ratio is no less than 0·96 or 0·97; that is to say, this engine would be actually doing within 3 per cent. of the real theoretical possibility. In other words, it is practically an impossible engine. One object of the efficiency ratio is to detect error. This it does, as is seen above, if the lower limit of temperature is taken as that in the exhaust, but it fails to do so if a fixed lower limit of temperature of 100° F. is taken. It is further interesting to note that, according to column I, engine 14 is not so good as engine 10, whereas in reality it is so much better that it is impossible.

Cases will occur in which it is of scientific interest to determine the efficiency ratio of any particular cylinder of a compound or of a triple-expansion engine. This can be readily done if the temperatures are taken as in column IV (see examples 11, 12 and 13), but obviously absurd results are obtained if a fixed lower temperature is taken, as in columns I and II.

APPENDIX IV.

Experiments were made by Prof. Beare and Capt. Sankey to determine the degree of accuracy with which the lower limit of temperature could be obtained. The same engine was used for these experiments as for the first set of those described in Appendix II. The pressure in the exhaust pipe was measured by a water U-tube in the case of the non-condensing experiments, and by a mercury vacuum gauge when the engine was running condensing. The thermometer used was subsequently calibrated at University College by Prof. Beare.

After trial No. 15, the speed of the engine was gradually reduced—but no change occurred in the readings. The engine was then stopped; there was no change for about one minute, then the thermometer began to fall very slowly.

The results are embodied in the following Table:—

Trial.	Speed of Engine.	Mean Gauge Readings.		Temperature in ° F.		
				Corresponding to Pressure shown by—		Mean Readings of Mercurial Thermometer.
		Mercurial Gauge.	Bourdon Gauge.	Mercurial Gauge.	Bourdon Gauge.	
1	455	*27·58*	27·9	*107·8*	102·45	*105·0*
2	455	*23·90*	23·9	*141·6*	141·6	*143·8*
3	455	*21·22*	21·4	*156·05*	155·3	*158·9*[1]
4[2]	455	..	18·5	..	167·2	*170·9*
5	455	*14·9*	15·0	*179·5*	179·2	*181·5*
6[3]	455	*11·2*	11·0	*189·6*	190·1	*191·7*
8	455	*8·55*	8·0	*195·7*	196·9	*196·1*
7	455	*11·3*	11·0	*189·3*	189·95	*190·7*
8	200	*7·5*	7·2	*198·1*	198·7	*191·5*
9	450	*8·35*	..	*196·2*	..	*195·7*
10	250	*8·5*	..	*195·9*	..	*188·8*[4]
11	445	*27·1*	27·5	*114·1*	108·4	*111·5*[5]
12[6]	150	*27·6*	28·0	*107·5*	101·7	*116·7*
13[7]	450	*27·3*	..	*111·5*	..	*111·4*
14	450	*27·2*	27·6	*112·0*[8]	107·6	*110·0*
15	450	*−0·1″* water press.	..	*211·7*	..	*211·8*[8]

[1] Gauges and thermometer varying slightly, but in time with each other.

[2] Flying readings.

[3] Flying readings.

[4] As engine slowed down the temperature fell and the vacuum increased.

[5] At slow speed the exhaust took time to cool from previous trial, but cooled quickly on raising speed.

[6] Speed dropped from 445 to 150. At first the temperature fell to 108° and the vacuum rose to 27·6″ by the mercurial gauge. Afterwards without any change in the vacuum the temperature slowly rose to 116°·7.

[7] Speed raised quickly. Vacuum fell slightly, and temperature also fell to 110° quickly.

[8] Non-condensing. The temperature rose at once on starting engine.

APPENDIX V.

NUMERICAL EXAMPLE OF METHOD OF CALCULATING THE FUEL CONSUMPTION WHEN THE B.T.U.'S PER MINUTE PER I.HP. ARE GIVEN.

It is stated at p. 20 that calculations in respect of fuel consumption can be readily made when the thermal economy of a steam-engine is expressed in B.T.U. per I.HP. The following example is given by way of illustration:—

A steam-engine requires 235 B.T.U. per minute per I.HP. The coal used has a calorific value of 12,000 B.T.U. per lb., and 70 per cent. of the heat in the coal is transmitted to the steam, or, in other words, 8,400 B.T.U. per lb. of coal burnt are transmitted to the steam. Hence the coal required per I.HP. per hour, is $\frac{235 \times 60}{8,400} = 1 \cdot 68$ lb. This calculation is, however, based on the supposition that the feed is at the temperature of the exhaust of the engine, since the heat supplied to the engine is calculated from this temperature. The feed temperature may be higher or lower according to circumstances, and an allowance must be made. This allowance is easily made in the case of saturated steam because the total heat required per lb. of feed does not differ much from 1,000 B.T.U., so that the correction is sensibly one-thousandth of the difference of temperature between the exhaust and the feed multiplied by the coal consumption obtained as above. Thus, supposing the feed in the above numerical example is 100° F. above the exhaust temperature (owing, say, to the use of an economizer), the coal consumption will be $\frac{100}{1,000} \times 1 \cdot 68$ less, or $1 \cdot 68 - 0 \cdot 17 = 1 \cdot 51$ lb. per I.HP. per hour. Or, again, supposing that in the case of a non-condensing engine the coal consumption works out to 2·32 lbs. and the feed is at 62° F., owing to there being no economizer or feed-heater, the correction will be $\frac{1}{1000}(212 - 62)\, 2 \cdot 32 = + 0 \cdot 35$, and the coal consumption becomes 2·67 lbs. per I.HP. per hour.

In the case of superheated steam, the correction becomes

$$\frac{\text{feed temperature} - \text{exhaust temperature}}{1,000 + 0 \cdot 48 \times \text{degrees of superheat}}.$$

Thus, taking the second example on page 16, and supposing that 9,000 B.T.U. per lb. of coal burnt are transmitted to the steam, the coal consumption is, if the feed is at exhaust temperature, $\frac{215 \times 60}{9,000} = 1 \cdot 43$ lb. of coal per I.HP. per hour. If the feed temperature is, however, 320° F., the correction according to the rule becomes $\frac{100 - 320}{1,000 + 0 \cdot 48 \times 275} = - 0 \cdot 19$ nearly, and the coal consumption is $1 \cdot 43 - 0 \cdot 19 = 1 \cdot 24$ lb. per I.HP. per hour.

APPENDIX VI.

NUMERICAL EXAMPLES OF THE METHOD OF CALCULATING THE B.T.U. PER MINUTE PER I.HP.

The principal supply of heat from the boiler to a steam-engine is effected by means of the steam entering the cylinders, but heat can also be supplied by the intermediary of jackets or reheaters. A portion of the heat thus entering the engine is returnable to the boiler. For instance, the exhaust steam from the cylinders can theoretically heat the feed-water to the temperature of the exhaust, and in some cases live steam from a receiver is taken to heat the feed. In the latter case the heat returned to the boiler per minute is clearly the weight of feed-water per minute multiplied by the number of degrees through which the feed is raised in temperature.

The heat returned by jackets and reheaters can be calculated if the weight of steam passing through them, and the temperature at which the resulting water is returned to the boiler, are known.

The expression "heat returnable" has been advisedly used, because in practice the heat actually returned to the boiler is generally less, owing to a variety of causes over which the engine has no control. For instance, in the case of a non-condensing engine, a feed-heater may not have been fitted, or if there is one it may be unable to raise the feed to exhaust temperature. Such losses should not, however, be charged against the engine, but against the feed arrangements.

The following numerical examples show how the various quantities of heat referred to above can be calculated, and the heat-supply to the engine obtained.

With one exception the data have been obtained from the actual published tests referred to in each case. In many instances the published figures do not give the "temperature of the steam at the stop-valve," or the "temperature in the exhaust close to the engine," as recommended by the Committee. In such cases either the temperature in the boiler or in the condenser have been taken as equivalent for the corresponding temperatures recommended by the Committee.

EXAMPLE 1.—NON-CONDENSING NON-JACKETED COMPOUND ENGINE.

Willans Trial C $\frac{80}{3\cdot2}$ (Minutes of Proceedings Inst. C.E., vol. xciii.).

Data—

I.HP.	24·9
Weight of steam entering engine, lbs. per minute	10·86
Temperature at stop-valve, ° F.	322
„ in exhaust, ° F.	213

Result—

Total heat of 10·86 lbs. of steam passing through the cylinders	12,800
Less heat returnable in the feed from exhaust	1,980
Net heat supply per minute	10,820
B.T.U. per I.HP. per minute $\left(\frac{10,820}{24\cdot9}\right)$	435

Example 2.—Condensing Non-Jacketed Compound Engines.

Engine of s.s. "Colchester" (Proceedings Inst. Mech. Eng., 1890, No. 2).

Data—

I.HP.	1,980
Weight of steam entering engine, lbs. per minute	717
Temperature of steam at stop-valve, ° F.	325
" " in exhaust, ° F.	135

Result—

Total heat of 717 lbs. of steam passing through cylinders	847,000
Less heat returnable in the feed from the exhaust	74,100
Net heat supply per minute	772,900
B.T.U. per minute per I.HP. $\left(\frac{772,900}{1,980}\right)$	391

Example 3.—Condensing Jacketed Compound Engines.

Engines of s.s. "Iona" (Proceedings Inst. Mech. Eng., 1891, No. 3).

Data—

I.HP.	645
Weight of steam through cylinders, lbs. per minute	136·5
" " H.P. jacket (discharged into feed-tanks)	6·2
Temperature at the stop-valve, ° F.	373
" in the exhaust, ° F.	90

Result—

Total heat of 136·5 lbs. of steam passing through cylinders	163,000	
Less heat returnable in the feed from the exhaust	7,920	
		155,080
Total heat of 6·2 lbs. of steam passing into HP. jacket	7,420	
No heat is returned by jacket, water being discharged into feed-tanks	0	
		7,420
Net heat supply per minute		162,500
B.T.U. per minute per I.HP. $\left(\frac{162,500}{645}\right)$		252

Example 4.—Condensing Compound Engine with Jackets and Reheater.

Louisville Leavitt Engine (Trans. American Soc. M.E., vol. xvi.).

Data—

I.HP.	643
Weight of steam through cylinders, lbs. per minute	109
" " jackets, " "	12
" " reheater " "	10
Temperature of steam at engine stop-valve, ° F.	359
" " in exhaust, ° F.	102
" " jacket and reheater water at return to boiler, ° F.	328

Result—

Total heat of 109 lbs. of steam passing through the cylinder	129,950	
Less heat returnable in the feed from the exhaust	7,400	
		122,550
Total heat of 12 lbs. of steam into jackets .	14,300	
Less heat returned (at 328°)	3,600	
		10,700
Total heat of 10 lbs. into reheater	11,900	
Less heat returned (at 328°)	3,000	
		8,900
Net heat supply per minute . . .		142,150
B.T.U. per minute per I.HP. $\left(\frac{142,110}{643}\right)$		221

EXAMPLE 5.—ENGINE IN WHICH STEAM IS ABSTRACTED FROM INTERMEDIATE RECEIVER TO HEAT THE FEED.

Imaginary Case. Condensing, non-jacketed.

Data—

I.HP.	1,000
Weight of steam in lbs. per minute passing into engine .	300
Temperature at engine stop-valve, ° F.	370
,, in exhaust pipe, ° F.	120
Rise in temperature of feed due to steam taken from receiver, ° F.	100

Result—

Total heat of 300 lbs. passing into engine		359,000
Less heat returnable in the feed from the exhaust	26,400	
Less heat returned from receiver steam by the feed	30,000	
Total heat returned		56,400
Net heat supply per minute . . .		302,600
B.T.U. per minute per I.HP. $\left(\frac{302,600}{1,000}\right)$		302·6

EXAMPLE 6.—SIMPLE NON-CONDENSING ENGINE USING SUPERHEATED STEAM SCHMIDT MOTOR.

No. 6 Trial (Minutes of Proceedings Inst. C.E., vol. cxxviii.).

Data—

I.HP.	19·85
Weight of steam in lbs. per minute passing through the cylinders	5·64
Pressure of steam at engine, lbs. per square inch (absolute)	131·6
Temperature of steam at engine, ° F.	674
,, in exhaust, ° F.	217

Result—

Total heat of 5·64 lbs. of saturated steam passing through cylinder	6,700	
Superheat of 5·64 lbs. of steam passing through cylinders	880	
Total heat supplied to engine per minute	7,580	
Less heat returnable in the feed from the exhaust	1,050	
Net heat supply per minute		6,530
B.T.U. per minute per I.HP. $\left(\frac{6{,}530}{19\cdot85}\right)$		329

APPENDIX VII.

Note on the Rankine Cycle.

The recommendation of the Committee that the cycle usually known as the cycle of the "Clausius engine" in this country should in future be called the "Rankine cycle" is based on the following:—

Rankine described the cycle in question, in his Paper "On the Geometrical Representation of the Expansive Action of Heat, and the Theory of Thermodynamic Engines," read before the Royal Society on January 19th, 1854, and published in the Philosophical Transactions for that year in 1855. This Paper is contained in his "Miscellaneous Scientific Papers," and on p. 400, in par. 46, entitled "Efficiency of a Vapour Engine without Compression," Rankine takes the case of a Carnot cycle, and finds that the result of dispensing with the adiabatic compression is to add to the heat expended and also to the work done. He finally expresses the thermal efficiency as follows:—

$$\frac{\text{Power developed}}{\text{Heat expended}} = \frac{\tau_1 - \tau_2}{\tau_1 - \kappa} - \frac{\mathrm{K_L}(\tau_2 - \kappa)\left\{\text{hyp log}\,\frac{\tau_1 - \kappa}{\tau_2 - \kappa} - \frac{\tau_1 - \tau_2}{\tau_1 - \kappa}\right\}}{\mathrm{L}_1 + \mathrm{K_L}(\tau_1 - \tau_2)},$$

where $\tau - \kappa$ signifies absolute temperature. His formula for the work done is—

$$\mathrm{L}_1\frac{\tau_1 - \tau_2}{\tau_1 - \kappa} + \mathrm{K_L}\left\{(\tau_1 - \kappa) - (\tau_2 - \kappa)\left(1 + \text{hyp log}\,\frac{\tau_1 - \kappa}{\tau_2 - \kappa}\right)\right\}.$$

Clausius described the same cycle in a memoir published in *Poggendorff's Annalen* in March and April, 1856, and contained in his "Mechanical Theory of Heat." He considers a continuous process with a boiler, cylinder, condenser and feed-pump, and also takes a mixture of steam and water at the end of the isothermal admission. His formula for the work done is—

$$\frac{1}{\mathrm{A}}\left\{m_1 r_1 \frac{\mathrm{T}_1 - \mathrm{T}_0}{\mathrm{T}_1} + \mathrm{M}c\left(\mathrm{T}_1 - \mathrm{T}_0 + \mathrm{T}_0 \log\frac{\mathrm{T}_0}{\mathrm{T}_1}\right)\right\}.$$

where A is the thermal equivalent of the unit of work.
m_1 is the weight of steam at the end of admission.
M is the weight of steam and water.
r_1 heat required to vaporize the unit mass at the temperature T_1.
c mean specific heat of the liquid.

After this formula Clausius inserts a footnote, which appears in Hirst's translation of the edition of 1867 (p. 161), but not in the German edition of 1876. This footnote is as follows:—

"The above equations, representing the amount of work under the two simplifying conditions introduced at the close of Art. 19,[1] were developed by me some time ago, and publicly communicated in my lectures at the Berlin University as early as the summer of 1854. Afterwards, on the appearance in 1855 of the Philosophical Transactions for the year 1854, I found therein a memoir of Rankine's 'On the Geometrical Representation of the Expansive Action of Heat and the Theory of Thermodynamic Engines,' and was surprised to learn that at about the same date Rankine, quite independently and in a different manner, arrived at equations which almost entirely agreed with mine, not only in their essential contents, but even in their forms; Rankine, however, did not take the circumstance into consideration, that, when entering the cylinder, a quantity of liquid is mixed with the vapour. By the earlier publication of this memoir I lost, of course, all claim to priority with respect to this part of my investigation; nevertheless, the agreement was so far satisfactory as to furnish me with a guarantee for the accuracy of the method I had employed."

It is also important to observe that this cycle is called the "Rankine Cycle" in America, by Professor Thurston, for instance.

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[1] *Viz.* no losses in ports, &c.

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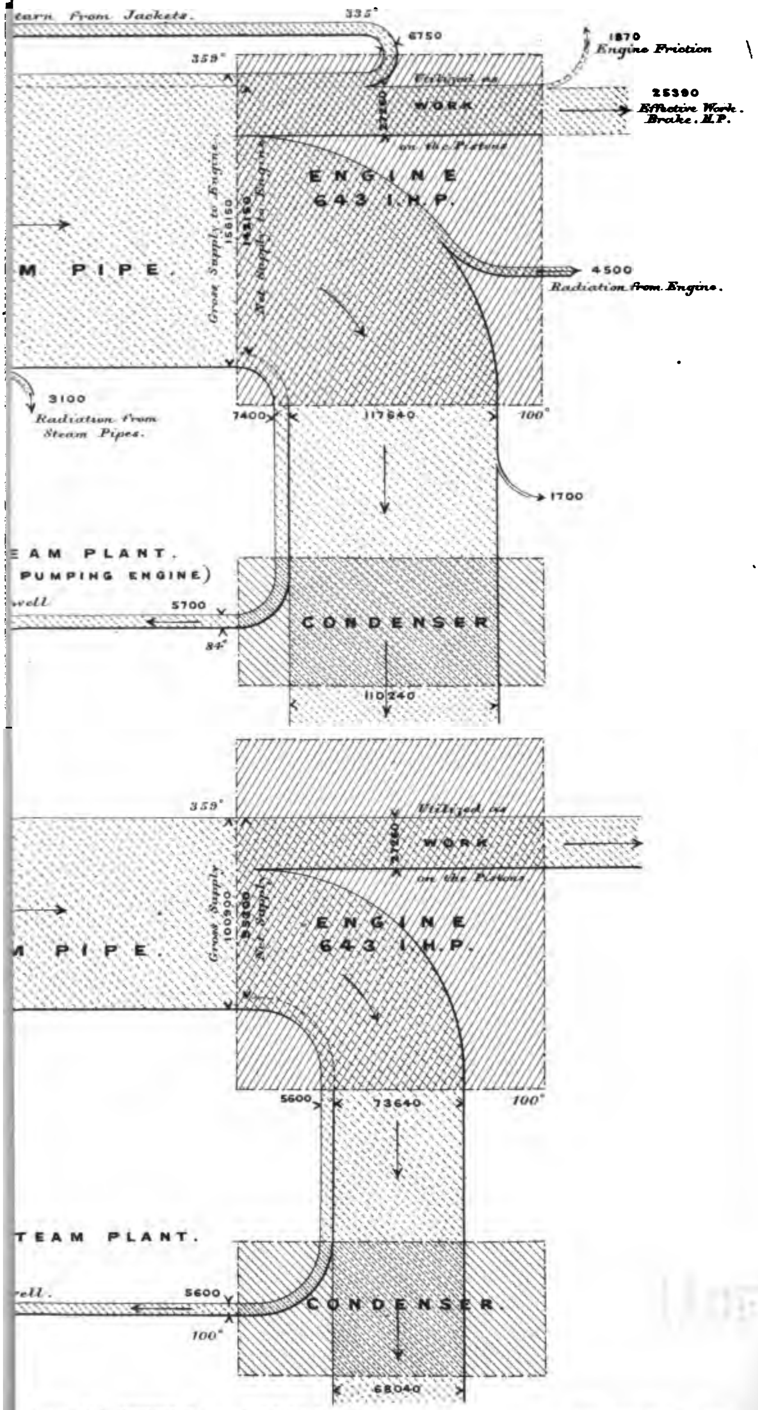

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