the performance of actual engines, and as setting forth the advan­tage of high-pressure steam from the thermodynamic point of view.

70. As a contrast to the ideally perfect steam-engine of § 68 we may next consider a cyclic action such as occurred in the early engines of Newcomen or Leupold, when steam was used non- expansively,—or rather, such an action as would have occurred in engines of this type had the cylinder been a perfect non-conductor of neat. Let the cycle of operations be this : —

(1) Apply A and evaporate the water as before at P1. Heat taken in = L1.

(2) Remove A and apply C. This at once condenses a part of the steam, and reduces the pressure to P2.

(3) Compress at P2, in contact with C, till condensation is complete, and water at τ2 is left.

(4) Remove B and apply A. This heats the water again to τ1 and completes the cycle.

Heat taken in=*h*1-*h*2.

The indicator diagram for this series of operations is shown in fig. 15.

Here the action is not reversible. To calculate the efficiency, we have Work done = (P1 - P2)(V1 - Q∙Q17)

Heat taken in J(L1 + Λ1-∕⅛)

The values of this will be found to range from 0·067 to 0·072 for the values of P1 which are stated in § 69, when the temperature of condensation is 60° F.

71. In the ideal engine represented in fig. 14 the functions of boiler, cylinder, and condenser are combined in a single vessel ; but after what has been said in chap. II. it is scarcely necessary to re­mark that, provided the working substance passes through the same cycle of operations, it is indifferent whether these are performed in several vessels or in one. To approach a little more closely the conditions that hold in practice, we may think of the engine which performs the cycle of § 70 as consisting of a boiler A (fig. 16) kept at τ1, a non-conducting cylin­

der and piston B, a surface con­

denser C kept at τ2, and a feed-pump

D which restores the condensed

water to the boiler. Then for every

pound of steam supplied and used

non-expansively as in § 70, we have

work done on the piston =(P1- P2) V1 ;

but an amount of work has to be ex­

pended in driving the feed-

pump =(P1-P2) 0·017.

Deducting this, the net

work done per lb of steam

is the same as before, and

the heat taken in is also

the same. An indicator diagram taken from the cylinder would give the area *efgh* (fig. 17), where *oe* = P1, *ef* = V1, *oh* = P2 ; an indi­cator diagram taken from the pump would

give the negative area *hjie,* where *ei* is the

volume of the feed-water, or 0·017 cub. ft.

The difference, namely, the shaded area, is

the diagram of the complete cycle gone

through by each pound of the working

substance. In experimental measurements

of the work done in steam-engines, only

the action whieh occurs within the cylinder is shown on the indi­cator diagram. From this the work spent on the feed-pump is to be subtracted in any accurate determination of the thermodynamic efficiency. If the feed-water is at any temperature τ0 other than that of the condenser as assumed in § 70, it is clear that the heat taken in is H1 - *h*0 instead of H1 - *h*2*.*

72. We have now to inquire how nearly, with the engine of fig. 16 (that is to say, with an engine in which the boiler and condenser are separate from the cylinder), we can approach the reversible cycle of § 68. The first stage of that cycle corresponds to the *admission* of steam from the boiler into the cylinder. Then the ]∣oint known as the point of *cut-off* is reached, at which admis­sion ceases, and the steam already in the cylinder is allowed to expand, exerting a diminishing pressure on the piston. This is the second stage, or the stage of *expansion.* The process of expansion may be carried on until the pressure falls to that of the condenser, in which case the expansion is said to be complete. At the end of the expansion *release* takes place, that is to say, com­munication is opened with the condenser. Then the return stroke begins, and a period termed the *exhaust* occurs, that is to say, steam passes out of the cylinder, into the condenser, where it is condensed at pressure P2, which is felt as a *back pressure* opposing the return of the piston. So far, all has been essentially reversible, and identical with the corresponding parts of Carnot’s cycle.

But we cannot complete the cycle as Carnot's cycle was com­pleted. The existence of a separate condenser makes the fourth stage, that of adiabatic compression, impracticable, and the best we can do is to continue the exhaust until condensation is com­

plete, and then return the condensed water to the boiler by means of the feed-pump.

It is true that we may, and in actual practice do, stop the exhaust before the return stroke is complete, and compress that portion of the steam which remains below the piston, but this does not materially affect the thermodynamic efficiency ; it is done partly for mechanical reasons, and partly to avoid loss of power through clearance (see chap. IV. ). In the present instance it is supposed that there is no clearance, in which case this compression is out of the question. The indicator diagram given by a cylinder in which steam goes through the action de­scribed above is shown to scale in fig. 18 for a par­ticular example, in which it is supposed that 1 cubic foot of dry saturated steam is admitted at an absolute pressure of 90 lb per square inch, and is expanded twelve times, or down to a pressure of 5·4 lb per square inch, at which pressure it is dis­charged to the condenser.

As we have assumed the cylinder to be non-conducting, and the steam to be initially dry, the expansion follows the law PV1·135 = constant. The advantage of expansion is obvious, that part of the diagram which lies under the curve being so much clear gain.

73. To calculate the efficiency, we have Work done per lb during admission = P1V1 ;

P V - P. rV

,, ,, during expansion to volume rV1=-i-^ι~ — -,

(by § 36), = (P1V1 - P2*r*V1)∕0 135 ;

Work spent during return stroke = P2*r*V1 ;

,, „ on the feed-pump = (P1 - P2)0·017 ;

Heat taken in = H1-*h*0.

74. These expressions refer to complete expansion. When the expansion is incomplete, as it generally is, the expression given above for the work done during expansion still applies if we take P2 to be the pressure at the end of expansion, while the work spent on the steam during the back-stroke is P*br*V1 and that spent on the feed-pump is (P1 - P*b*)0·017, P*b* being the back pressure. Incomplete expansion is illustrated by the dotted line in fig. 18.

It is easy, by the aid of §§ 64 and 67 to extend these calcula­tions to cases where the steam, instead of being initially dry, is supposed to have any assigned degree of wetness. The efficiency which is calculated in this way, which for the present purpose may be called the theoretical efficiency corresponding to the assumed conditions of working, is always much less than the ideal efficiency of a perfect engine, since the cycle we are now dealing with is not reversible. But even this theoretical efficiency, short as it falls of the ideal of a perfect engine, is far greater than can be realized in practice when the same boiler and condenser tem­peratures are used, and the same ratio of expansion. The reasons for this will bo briefly considered in the next chapter ; at present the fact is mentioned to guard the reader from supposing that the results which the above formulas give apply to actual engines.

75. The results of § 68 have been turned to account by Rankine and Clausius for the purpose of deducing the density of steam from other properties which admit of more exact direct measure­ment. Let the perfect steam-engine there described work through a very small interval of temperature ∆τ between two temperatures τ and τ - ∆τ. The efficiency is ∆τ∕τ, and the work done (in foot-lbs. ) is JL∆τ∕τ. The indicator diagram is now reduced to a long narrow strip, whose length is V -0·017 and its breadth ∆P, the difference in pressure between steam at temperatures τ and τ-∆τ. Hence the work done is also ∆P(V - 0·017), and therefore

V-0∙017 = JL/τ·∆τ/∆P·

Here ∆τ/∆P, or(in the limit) dτ/dP, is the rate of increase of tempera­ture with increase of pressure in saturated steam at the particular temperature τ. It may be found roughly from Table II., p. 484, or more exactly by differentiating the equation given in § 57. L is also known, and hence the value of V corresponding to any assigned temperature may be calculated with a degree of accuracy which it would be difficult to reach in direct experiment. The volumes given in the Table are determined in this way.@@1

@@@1 The result of § 75 may be applied as follows to give the formula of § G7 for the adiabatic expansion of wet steam. For brevity we may write V—0∙017 = *u*. In adiabatic expansion the work done is equal to the loss of internal energy, or

*Pd*(*qu*) *=—*J*d*I*=—*J*d(h+qρ).*

Since *dh=dτ,* and ρ=L — P*u*/J, this may be written J*dτ +* J*d(q*L*)-qudP =* 0.

By § 75, *ud*P*=* JL/*τdτ*; hence 1 + *d*/*dτ*(*q*L)-*q*L/*τ* =0;

and by integration,

logετ+*q*L,∕τ=constant=loge τ1+*q*1L1∕τ1, which is the equation of § 67.