FIRST REPORT TO THE STEAM-ENGINE RESEARCH COMMITTEE.*

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The Steam-Engine Research Committee was constituted at the instance of the late Mr. Bryan Donkin to investigate and carry out research upon the Initial Condensation in Steam-Engine Cylinders. With this object a French firm undertook to construct an engine embodying the ideas of Mr. Donkin, and to lend it to the Committee for a series of research experiments. This engine was provided with three valves—first, one to admit steam during lead; secondly, one to open at the commencement of the stroke and to act as the admission valve, and thirdly, an exhaust valve. By supplying the first and second valves from separate boilers Mr. Donkin hoped that the steam required to replace initial condensation might be separately measured. The French firm at the last moment found that the engine had been so costly in construction that they were unable to fulfil their promise, and this form of experiment had to be abandoned. The Committee then decided to accept the offer of the authorities of King's College, London, to provide an engine of more ordinary design and place it at their disposal for experimental purposes. Through the kindness of Messrs. Marshall, Sons and Co., of Gainsborough, a special engine was obtained at cost price, and erected in the Engineering Laboratory at King's College.

* Incorporating the results of experiments at King's College, London, on Jacketed and Unjacketed Cylinders.

Description of Engine.—The engine thus obtained is a horizontal compound engine. The cylinders, which are side by side, are $6\frac{1}{2}$ inches and $11\frac{1}{4}$ inches diameter by 14 inches stroke, and the connecting-rods drive cranks placed at right angles. The arrangement of the engine and connections is shown in Fig. 1 (page 175).

Each cylinder is separately jacketed on the barrels and the ends, the supply and drain from the ends being separate from that for the barrel. Each cylinder is fitted with a Meyer expansion-valve adjustable by hand so that the cut-off can be separately varied between one-quarter and five-eighths of the stroke. By grid valves on admission and exhaust sides, either cylinder can be arranged as a simple engine, and by blocking either of the Meyer plates the engine can be made single-acting. In any of its varied adjustments the engine is as nearly as possible an ordinary commercial engine, clearance volumes and contact surfaces being kept as small as possible. The clearance volumes for the high-pressure cylinder are 0.025 of a cubic foot at the front, and 0.033 at the back; and for the low-pressure cylinder 0.053 of a cubic foot at the front, and 0.056 at the back.

On the trials here recorded the low-pressure cylinder was not in use, the engine being run as a single-cylinder engine exhausting into a condenser, with the vacuum reduced to 2 to 3 inches of mercury. In this way the back pressure in the cylinder was kept constantly at atmospheric pressure.

The volume swept through by the high-pressure piston is 0.269 of a cubic foot at the back end, and 0.254 at the front, after deducting the volume of the $1\frac{1}{2}$ -inch diameter piston-rod. The clearance space is therefore 12.4 per cent. of the cylinder volume at the back, and 9.8 per cent. at the front end.

The flywheel is 5 feet in diameter and water-cooled. To ensure steady running at slow speeds, a second flywheel, 7 feet in diameter, was provided and fitted in halves, so that at speeds above 100 revolutions it could be readily removed. But after the preliminary trials the larger flywheel was found unnecessary and was removed, as, within the range of speeds required, the engine ran perfectly steadily without it.

Boiler, Condenser, etc.—The boiler available was a Davey-Paxman semi-portable locomotive working up to 150 lbs. pressure per square inch, and capable of evaporating about 1,000 lbs. of water per hour. The condenser is a cylindrical copper condenser with 50 square feet of cooling surface, the tubes being externally cooled. It was kindly presented by Messrs. J. I. Thornycroft and Co. and the trustees of the late Mr. Donaldson, the Worthington feed- and air-pumps being presented by the Worthington Pumping Engine Co., and the steam traps by Messrs. Geipel and Lange, and the Lancaster Trap Co.

Scheme of Experiments.—The Committee decided that the first series of trials should be made with the engine arranged as a single-cylinder high-pressure engine, non-condensing jacketed, the second series being a repetition of the first series but without jackets. After a careful study of previous experiments, it was further determined that the variables should be temperature and speed, cut-off and all other conditions being kept constant. Temperatures of 245° F., 280° F., 315° F., and 350° F., were chosen, so as to give equal intervals between successive trials of the series. These temperatures correspond to about 27, 49, 84, and 135 lbs. per square inch absolute respectively at the engine steam-chest. The chosen speeds were 50, 100, 150, 200, and 250 revolutions per minute, corresponding to piston speeds of about 117, 234, 350, 467, and 584 feet per minute respectively. For reasons given below, it was not found possible to adhere absolutely to these speeds and temperatures.

Preliminary Trials.—A large number of preliminary trials were made to determine the best cut-off, exact speeds, and temperatures, which could be accomplished with the plant, and to find the best methods of adjusting and working the measuring apparatus, etc. It was found that it was impossible to extend the range of speeds to the lowest limit of the engine, namely, 25 revolutions per minute, without interfering with the upper limit, and it was likewise found that at the lowest temperature and pressure the auxiliary pumps could not be worked. At the upper limit of pressure and lower

limit of speed, and at the lower limit of pressure and upper limit of speed, trouble was found in keeping the conditions quite steady. After numerous attempts, it was finally found necessary to work the boiler at a slightly higher pressure than that actually required, and to get the exact and steady temperature in the steam-chest by slight throttling at the engine stop-valve.

At the lowest temperature of the series and in one or two other cases this throttling was sufficient to cause superheating, to which further reference will be found below. The cut-off which was required to obtain the largest range was found to be three-eighths of the stroke, and this was adhered to throughout both series.

Arrangement of Plant.—The arrangement of the plant is shown diagrammatically in Fig. 1. Steam is supplied to the engine from the boiler above described through a steam pipe of 3 inches internal diameter. The total distance between the flange of the main stop-valve on the boiler and that on the engine was 47 feet. The whole of the pipe including all flanges is lagged for half its length with Leroy's covering $1\frac{3}{4}$ inches thick and the remainder with magnesia $1\frac{1}{2}$ inches thick. The pipe is sloped downwards from the boiler towards the engine, so that any moisture may be deposited and carried in the same direction as the steam. A $\cap$ bend is inserted between the end of the pipe and the engine stop-valve, and one arm of the $\cap$ is continued downwards beyond the junction with the steam-pipe, so as to form a collecting chamber for any moisture which is travelling along the pipe.

A gauge-glass is inserted in this separator, and a drain cock at the bottom opens communication with a steam trap, or with a measuring vessel. After experiment it was found most satisfactory to pass the discharge from the separator through a coil of pipe placed in a cooling bucket, so as to prevent the escape of vapour and ensure the whole of the moisture being measured. It was thus possible to measure with extreme accuracy any accumulation of water in the separator, and no trial in which this was above $2\frac{1}{2}$ per cent. of the whole steam discharged from the engine was accepted as satisfactory. Beyond the $\cap$ bend a wire-drawing calorimeter was inserted, and a

secondary measurement made at this point so as to check the readings of the separator. Arrangements were at first made to put a heating coil under the steam-pipe, so as to ensure dry steam being supplied, but the measurements made with the separator and the calorimeter showed that the steam supply from the boiler was so uniform that the heating coil was unnecessary.

FIG. 1.

Diagram showing General Arrangement of Experimental Plant, King's College.

Exhaust.—The trials being non-condensing, the air-cock on the condenser was opened so as to admit air and reduce the vacuum to the required extent. The discharge from the condenser was measured in a tank placed on a platform weighing machine, and records were taken every five minutes. Experiments were frequently made to determine

any leakage in the condenser, and after the tubes had been specially packed no difficulty was experienced in keeping the condenser tight. The pressures were measured by Bourdon gauges inserted one at the back of the main stop-valve and one on the steam-chest, and by a mercury gauge placed upon the condenser. All steam-gauges used were calibrated at intervals under steam on the mercury column in the laboratory, and the readings given in the Tables are the corrected readings after allowing for the errors so found.

Thermometers were placed in the steam-pipe and in the condensed water, and these were also calibrated in the laboratory. The indicator gear was attached as shown in Fig. 1 (page 175), the cylinders being positively driven, without the intervention of a spring, by steel piano wire passing round a drum on a rocking shaft. An indicator was placed on each end of the cylinder, and the connecting-pipe was made as short as possible. The indicator springs used were all calibrated under steam on the mercury column, and the indicated horse-powers have been worked out to the corrected scale of the springs used. All the diagrams were taken with soft lead pencil upon glazed paper so as to reduce as far as possible friction on the pencil point. Throughout the trials the low-pressure piston-rod was uncoupled, so that the high-pressure engine ran entirely independently without having to drive the low-pressure piston. The brake gear consisted of a rope wrapped completely round the water-cooled flywheel, the fall being attached to a spring balance which was frequently calibrated during the experiments by standard weights. The brake was lubricated by paraffin oil applied between the observations as uniformly as possible. Speed and not H.P. being fundamental, at each pressure the load and the lubrication were adjusted so as to keep the speed as nearly constant as possible. The revolutions per minute were recorded by a positive counter and checked at intervals.

Method of Conducting the Trials.—Observers were placed at the boiler gauge, the engine gauges, the separator and calorimeter, at the indicator, at the brake, at the condenser, and at the condensed-water tank, so that observations of the readings of all the instruments

could be taken at frequent intervals. Steam was raised in the boiler to the required pressure, and the engine having been started and run for some time until it was running steadily under the required conditions of speed and steam pressure, the trial was commenced and observations were taken every five minutes for half-an-hour. If the conditions varied during the run to any considerable extent the trial was continued until a steady run of twenty minutes to half-an-hour was obtained. If the trial then seemed generally satisfactory, the conditions were altered and a further trial was run.

It was not found possible to carry out more than about two trials on any day, owing to the demands upon the staff at King's College, and the large number of students who had to be dealt with in the laboratory. For the same reason the trials could not be conducted from day to day, but intervals of a week and more had often to elapse between consecutive trials. If on working out any trial any of the conditions or observations were found to vary beyond defined limits, the trial was rejected and repeated. In most cases the trials were divided into two, so that a careful comparison of the duplicate results could be made. Any trial which did not show uniformity throughout and accord well with the required conditions was not accepted.

Over 100 trials have thus been carried out and, of these, 38 have been selected as most closely satisfying the conditions laid down. Detailed observations and calculated results of these 38 finally selected trials are given in the appended Tables 13 and 14 (pages 252 to 255).

Explanation of Tables.—The capital letters A, B, C, and D, have been chosen to denote the four series of trials at different steam temperatures.

A	denotes the series at	245° F.
B	„	280° F.
C	„	315° F.
D	„	350° F.

The suffix attached to each capital letter denotes the speed at which each trial of a series was carried out.

The Suffix 1 corresponds to 50 revolutions.

„	2	„	100	„
„	3	„	150	„
„	4	„	200	„
„	5	„	250	„

Each unjacketed series is denoted by the same letter and suffix as the corresponding jacketed series, the letter being repeated in the unjacketed series for distinction.

A₁ therefore means a trial at 245° F. 50 revolutions, jacketed.

AA₁ „ „ 245° F. „ unjacketed.

and so on for the whole series.

FIG. 2.

Specimen Indicator Diagrams.

Trial D₂ (Jacketed ; 100 Revolutions per minute ; 150 lbs. Pressure.)

Trial A₅ (Jacketed ; 250 Revolutions per minute ; 35 lbs. Pressure.)

Method of working out Results.—From the whole of the diagrams taken on each trial a mean card has been constructed, and upon the mean diagram all calculations have been based. The diagrams were in most cases remarkably uniform, so that a card upon which the pencil had been allowed to trace a continuous diagram for half-a-minute shows hardly any measurable variation from a single card. Half-minute diagrams were thus taken in all cases so as to guide in the construction of the mean diagram. Copies of the actual diagrams and of mean calculated diagrams are shown on Fig. 2 and Figs. 3 to 12 (pages 180 to 189).

A few of the diagrams show negative areas, but these are confined to diagrams obtained on the lowest speed trials of the jacketed series. They do not occur with the unjacketed series, as, with jackets on, initial condensation is less, and the dryness-fraction at release is greater than without jackets. Less weight of steam is taken into the cylinders, and the release pressure is therefore lower than for the corresponding unjacketed trial. At the lowest speeds and pressure therefore the release pressure is atmospheric, and in one or two instances lower than atmospheric pressure. The negative area is so small as not to destroy the value of the trials and they have been included in the selected series.*

Saturation Curve.—From the condensed water measured from the exhaust a saturation curve has been plotted on each mean diagram representing the volume of the mean steam passing into the exhaust per stroke plus clearance steam. It has hitherto been usual to assume that this measured exhaust steam has actually passed through the cylinder, and that the difference between the expansion curve of the diagram and the saturation curve for the exhaust steam thus plotted represents condensation in the cylinder; but it is evident that any steam leaking through the slide-valve or piston to the exhaust will not be shown on the expansion curve of the indicator diagram, and that any other leakage through either piston or slide-valve will affect the

(Continued on page 190.)

* The errors due to backlash and friction on an indicator pencil occur in opposite senses on the upward and downward stroke. When there is a negative area on an indicator diagram and the exhaust line cuts the expansion curve at any considerable distance from the end of the stroke, a small difference in the pressure at which this intersection occurs will make a large difference in the relative magnitudes of the positive and negative areas.

In such a case backlash and pencil friction will make the expansion curve too high and the exhaust line too low, so that these errors will have a double effect in falsifying the point of intersection. The resulting error in the nett area of the diagram is therefore out of all proportion to the corresponding error on a diagram where exhaust and expansion do not cut. In the present case the negative is so small a percentage of the whole area that the errors fall within the limits of accuracy obtainable with the remainder of the results.

FIG. 4.—Mean Indicator Diagrams.

TRIALS AA₁, BB₁, CC₁, DD₁.
Unjacketed.

	Revs. per min. appo.	Mean Press.	I.H.P.	Steam Chest Press.	Press. at Inlet Valve.
AA ₁	50	13.04	1.53	Lbs. Abs. 37.6	Lbs. Abs. 51.5
BB ₁	50	29.23	3.54	56	57.8
CC ₁	50	61.93	7.24	99.4	105.2
DD ₁	50	92.51	11.51	143.5	160.4

FIG. 5.—Mean Indicator Diagrams.

TRIALS A₂, B₂, C₂, D₂.
Jacketed.

	Revs. per Min. approx.	Mean Press.	I.H.P.	Steam Chest Press.	Press. at Inlet Valve.
A ₂	100	9.27	2.06	Lbs. Abs. 37	Lbs. Abs. 39.2
B ₂	100	25.88	5.87	55	67.6
C ₂	100	59.71	13.84	98.7	102
D ₂	100	96.50	22.44	149.7	157.3

FIG. 6.—Mean Indicator Diagrams.

TRIALS AA₂, BB₂, CC₂, DD₂.
Unjacketed.

	Revs. per min. appo.	Mean Press.	I.H.P.	Steam Chest Press.	Press. at Inlet Valve.
AA ₂	100	12·36	3·00	Lbs. Abs. 35·7	Lbs. Abs. 49·5
BB ₂	100	28·12	6·50	55·7	57·5
CC ₂	100	55·04	12·35	90·25	93·7
DD ₂	100	88·52	21·49	139·4	150·9

FIG. 7.—Mean Indicator Diagrams.

TRIALS A₃, B₃, C₃, D₃.
Jacketed.

	Revs. per min. appro.	Mean Press.	I.H.P.	Steam Chest Press.	Press. at Inlet Valve.
A ₃	150	8.23	2.86	Lbs. Abs. 31.05	Lbs. Abs. 44.7
B ₃	150	27.11	9.39	56.74	74
C ₃	150	55.38	19.63	93.9	107.7
D ₃	150	94.80	31.14	146.5	151.5

FIG. 8.—Mean Indicator Diagrams.

TRIALS AA₃, BB₃, CC₃, DD₃.
Unjacketed.

	Revs. per min. appo.	Mean Press.	I.H.P.	Steam Chest Press.	Press. at Inlet Valve.
AA ₃	150	11·81	4·10	Lbs. Abs. 38·7	Lbs. Abs. 48·4
BB ₃	150	25·68	9·12	55	57·8
CC ₃	150	57·42	19·80	98·2	105·2
DD ₃	150	88·83	29·08	137·9	144·1

FIG. 9.—Mean Indicator Diagrams.

TRIALS A₄, B₄, C₄, D₄.
Jacketed.

	Revs. per min. appo.	Mean Press.	I.H.P.	Steam Chest Press.	Press. at Inlet Valve.
A ₄	200	9.6	4.45	Lbs. Abs. 33.2	Lbs. Abs. 47.1
B ₄	200	23.52	10.93	54.44	69.14
C ₄	200	55.3	25.25	96	101.7
D ₄	200	85.9	38.10	139.3	157.5

FIG. 10.—*Mean Indicator Diagrams,*TRIALS AA₄, BB₄, CC₄, DD₄.
Unjacketed.

	Revs. per min. appo.	Mean Press.	I.H.P.	Steam Chest Press.	Press. at Inlet Valve.
AA ₄	200	13.22	5.93	Lbs. Abs. 40	Lbs. Abs. 59.7
BB ₄	200	25.22	11.39	55	57.7
CC ₄	200	52.08	22.20	88.6	90.9
DD ₄	200	77.03	32.62	125.8	135.3

FIG. 11.—Mean Indicator Diagrams.

TRIALS A₅, B₅, C₅.
Jacketed.

	Revs. per min. appro.	Mean Press.	I.H.P.	Steam Chest Press.	Press. at Inlet Valve.
A ₅	250	11.35	6.77	Lbs. Abs. 35.5	Lbs. Abs. 44.2
B ₅	250	26.75	14.67	56.4	66
C ₅	250	46.8	26.82	84.5	92.7

FIG. 12.—Mean Indicator Diagrams.

TRIALS AA₅, BB₅, CC₅.
Unjacketed.

	Revs. per min. approx.	Mean Press.	I.H.P.	Steam Chest Press.	Press. at Inlet Valve.
AA ₅	250	12.40	6.96	Lbs. Abs. 37	Lbs. Abs. 48.9
BB ₅	250	25.07	15.01	54.2	61.7
CC ₅	250	47.74	27.24	87.5	105.2

diagram so as to modify conclusions drawn without taking such leakage into account.

Messrs. Callendar and Nicolson* in their extremely suggestive Paper have shown that such leakage may be considerable, and in the present trials an attempt has been made to measure, analyse, and allow for the various leakages which occur. This analysis confirms the conclusions at which they arrived, that without such allowance no correct determination can be made of the relative weight of steam and moisture present in the cylinder.

Thermal Units used per H.P. Minute.—To calculate the thermal units per H.P. consumed by the engine, it has been assumed that the heat supplied to the engine is the difference between the heat units in the steam at the steam-chest pressure and temperature selected and the heat units in the condensed water. This is the same as assuming that the boiler and connections are outside the engine, and the losses in these external to it. The heat units calculated on this basis are given in cols. 19, 20 and 21, Table 13 (page 253); and col. 18, Table 14 (page 255).

In several trials, as already noted, the steam-valve was set so as to produce throttling and slight superheating at admission. The moisture in the steam entering the stop-valve was less than 2 per cent., and under these conditions the superheating cannot in any case have been greater than 4° F. In calculating the thermal units supplied to the engine the superheating calculated on this basis has in every case been taken into account.

Standard of Comparison.—The standard of comparison taken is the Rankine engine working between the extreme limits of temperature. The upper temperature for the standard Rankine engine has been taken as the temperature corresponding to the pressure in the steam-pipe on the boiler side, but close to the engine stop-valve.

The British thermal units required by a Rankine engine working between the above limits is given in col. 23, Table 13; and col. 19,

* The Proceedings, Institution of Civil Engineers, 1897-8, vol. cxxxi, page 147.

Table 14. The efficiency ratio or the ratio between the heat required by the Rankine engine and the heat required by the experimental engine has been calculated and is given in cols. 25 and 26, Table 13; and col. 21, Table 14.

For comparison with other records in which a Carnot engine has been taken as the standard of comparison, the heat units required per H.P. minute for a Carnot engine working between the same limits of temperature as the Rankine engine has been given in col. 24, Table 13; and col. 20, Table 14, and the efficiency ratio of the experimental engine to the Carnot engine is given in col. 27, Table 13; and col. 22, Table 14.

For each trial a temperature entropy diagram has been constructed, and the entropy diagrams for each temperature series and each speed series are shown superimposed upon one another in Figs. 13 to 30 (pages 192 to 207) so that a direct comparison can be made of the thermal units required at constant temperature and at constant speed and at each temperature and at each speed.

Leakage through Slide-valve.—There are two obvious methods of experimentally determining the leakage through the slide-valve. Either a special valve can be used with sufficient lap to cover the steam-port throughout the valve stroke or the steam-ports themselves can be blocked. In either case the leakage will be determined by admitting steam to the slide-chest, driving the engine by external power and measuring the steam escaping into the exhaust.

The first method involves a special slide-chest, and the steam-ports being open the leaking steam will emerge partly through the exhaust and partly through the cylinder. Errors will consequently result in the measurement of the leak owing to the large areas on which unmeasured condensed steam will be deposited. Moreover, if the leakage in the case of flat plates sliding on one another is affected by the width of surface in contact, the additional overlap of the special valve will affect the leakage and vitiate the comparison of the experimental results with the leakage

(Continued on page 208.)

FIG. 13.
Temperature Entropy Diagrams.

FIG. 15.
Temperature Entropy Diagrams.

FIG. 14.
Temperature Entropy Diagrams.

FIG. 16.
Temperature Entropy Diagrams.

FIG. 17.

Temperature Entropy Diagrams.

FIG. 18.

Temperature Entropy Diagrams.

FIG. 19.

Temperature Entropy Diagrams.

FIG. 20.

Temperature Entropy Diagrams.

FIG. 21.

Temperature Entropy Diagrams.

FIG. 22.

Temperature Entropy Diagrams.

FIG. 23.

Temperature Entropy Diagrams.

FIG. 24.

Temperature Entropy Diagrams.

FIG. 25.

Temperature Entropy Diagrams.

FIG. 26.

Temperature Entropy Diagrams.

FIG. 27.

Temperature Entropy Diagrams.

FIG. 28.

Temperature Entropy Diagrams.

FIG. 29.

Temperature Entropy Diagrams.

FIG. 30.

Temperature Entropy Diagrams.

which obtains during the actual running of the engine under steam. Further, in actual working the steam-ports will, during admission and expansion, be full of steam at varying pressure, and leakage from the cylinder into the exhaust past the exhaust lap of the valve will take place in addition to the direct leak from the steam-chest to exhaust, while under experimental conditions with the special valve the cylinder will be always approximately at atmospheric pressure, so that leakage into the cylinder through the ports will be greater than under working conditions.

By the second method, namely, blocking the steam-ports and running with the normal working valve, the differences between

FIG. 31.

Four upper views show by arrows Leakage under Working Conditions.

experimental and working conditions are either eliminated or greatly reduced. Certain points of difference, however, will still remain, and the diagrams, Fig. 31, will help to explain these.

I. During admission under working conditions the flow of steam into the cylinder will to a certain extent check the tendency to leak from the steam-port past the valve into the exhaust.

With the blocked ports there will be no such check, and the leakage into the exhaust will therefore be slightly greater than under working conditions.

II. During expansion and also during compression under working conditions there will be leakage from the steam-chest past the lap into

the cylinder and from the cylinder past the valve into the exhaust. The amount of leakage into exhaust will therefore depend upon the difference between the expansion or compression pressure and the exhaust pressure.

With blocked ports there will be leakage past the lap into the clearance over the stop in the ports, and from that clearance, which will rapidly fill with steam at or about steam-chest pressure, to the exhaust.

The leakage into exhaust will therefore depend upon the difference between the steam-chest and the exhaust pressures, modified by the small amount of expansion which takes place in the clearances over the blocked ports. The leakage to the exhaust will therefore be greater with blocked ports than under working conditions.

Messrs. Callendar and Nicolson have pointed out that the leakage past the valve is most probably in the form of moisture condensed on the valve-face and re-evaporated, rather than a direct leakage of steam, and it will be seen that the leakage experiments detailed below confirm this conclusion. But this being the case, it is important to note that to imitate working conditions the cylinder walls, slide-face, and surrounding metal should be kept as near to the working conditions of temperature as possible. As the cylinder with blocked ports will not, as in working, be kept warm by the admission of steam at every stroke, the leakage found with both barrel and ends of cylinder jacketed has been accepted as the nearest approximation to the leakage taking place both in the jacketed and unjacketed series of trials.

The leakage trials described in the present Report were made on the high-pressure cylinder with the ordinary valve and with blocked ports, the engine being driven at the several trial speeds by external power and steam of the several trial pressures admitted to the steam-chest.

In blocking the steam-ports considerable difficulty was at first experienced, and numerous devices were tried before a steam-tight joint was obtained. The method ultimately adopted was to scrape a metal fitting piece flanged at each end to an absolutely tight fit in the ports, and then to screw this block down with a red-lead joint into recesses cut in the slide-face at each end of the port. This was found to give a tight joint.

TABLE 1.—*Pressures &c. during Measurements of Leakage past Slide-Valve with Steam-Ports Blocked.*
(Results given in Table 2.)

Average Pressures in lbs. per square inch absolute.				Differences in Pressure in lbs. per square inch.	
Steam-pipe before inlet.	Steam-chest.	Exhaust.		Steam-chest minus exhaust.	
		Engine standing, Valve in mid position.	Engine running at all speeds.	Engine standing, Valve in mid position.	Engine running.
39·25	35	—	16·25	—	18·75
50·5	45	15·5	16·75	29·5	28·25
66·25	63	15·75	17	47·25	46
96·25	93	15·75	18·25	77·25	74·75
152	145	17·75	19·75	127·25	125·25

(Table 1 continued in Table 2.)

N.B.—On all the above trials the sight feed lubricator was supplying oil to the steam-pipe at the rate of about two drops per minute.

TABLE 2.—*Leakage past Slide-Valve with Steam-Ports Blocked.*

Leakage in lbs. per hour through Slide-Valve.									
Syphon Lubricator on Steam-Chest stopped.						Syphon Lubricator on Steam-Chest supplying freely.			
Cylinder ends jacketed only.		Ends and Barrel of Cylinder Jacketed.							
Engine standing, Valve in mid position.	At 50 revs.	Engine standing, Valve in mid position.	At 50 revs.	At 156 revs.	At 252 revs.	Engine standing, Valve in mid position.	At 50 revs.	At 156 revs.	At 250 revs.
—	—	—	—	14·1	13·5	7·7	11·6	11·25	10·5
—	21·7	16·1	18·0	—	—	—	—	—	—
19·7	24·2	14·4	18·4	23·4	22·5	14·6	20·6	19·5	19·5
25·5	38·6	19·5	28·9	30·9	29·4	22·5	30·7	27·25	26·6
38·2	46·5	32·4	38·1	40·3	—	37·25	45·0	42·75	41·2

N.B.—On all the above trials the sight feed lubricator was supplying oil to the steam-pipe at the rate of about two drops per minute.

The exhaust was connected to a condensing coil, so that all leakage into the exhaust could be measured with great accuracy. This arrangement was found to act admirably, but it was impossible to adjust this coil with such nicety as to prevent the escape of steam and at the same time to prevent some accumulation of pressure in the short exhaust branch. This back-pressure has been in every case recorded in Tables 1 and 2 (pages 210 and 211), which give the mean results for a number of concordant experiments measured under varying conditions.

FIG. 32.

Curves plotted from Values of Table 2, showing Slide-Valve Leakage varying with Pressure.

The mean values of the last three columns of Table 2 (page 211) have been plotted on Fig. 32, and have been used for the corrections for leakage which follow.

Effect of Warm and Cold Cylinder.—On analysing these results it will be seen that, under given conditions, the leakage when the barrel of the cylinder was warmed by the jacket was considerably less than

when it was cold. From what has previously been said, the proper correction to employ will be that obtained with the cylinder ends and barrel warmed.

Effect of Lubrication on Leakage.—In all except the 50-revolution trials there is a distinct reduction in leakage when the sliding surfaces are well lubricated, over the corresponding leakage with scant lubrication. This effect is not however so constant or so large as to warrant any definite quantitative analysis of the results. It is of course difficult to ensure uniformity in the lubrication of the surfaces, and the discrepancies are no doubt largely due to breaks in continuity of the oil film. The cases in which such a reduction was observed were however so numerous in individual trials as to definitely prove that, where the oil film is uniform and continuous, leakage is appreciably reduced. The curves on Fig. 32 show this clearly, although the reduction diminishes with increase of pressure. It is interesting to note on the curve for the well-lubricated jacketed trials that, where in one instance the lubrication failed, the leakage at once runs into and above the curve for scant lubrication.

Effect of Pressure on Leakage.—The leakage, as would be anticipated, rises with an increase of pressure between steam-chest and exhaust. Where the valve is stationary in mid position, the leakage increases approximately as the pressure. Where the engine is running, however, the leakage does not increase so rapidly as the pressure, and the higher the speed the larger does this divergence become. It is evident that this bears a striking resemblance to the laws governing condensation and re-evaporation on the cylinder walls of steam-engines, and when the amount of the leakage and its reduction by jacketing is considered the conclusion arrived at by Messrs. Callendar and Nicolson is irresistible, that much of the leakage must be in the form of moisture condensed on the valve face and re-evaporated as it passes over into the exhaust.

Effect of Wire-Drawing on Leakage.—As bearing upon the above point, experiments were made to determine what effect wire-

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drawing, and therefore superheating the steam previously to its entering the steam-chest, has upon leakage. The following Table 3 shows the results obtained :—

TABLE 3.

Table showing the Effect upon Leakage through Slide-Valve of wire-drawing Steam at the Inlet-Valve. Lubrication on Steam-Chest flowing freely. All jackets on.

Condition of Trial.	Pressure, Lbs. per sq. in. absolute.					Leakage Lbs. per Hour.	Ratio. Leakage { $\frac{\text{Superheated}}{\text{Saturated}}$.
	Steam-Pipe before Inlet.	Steam-Chest.	Exhaust.	Differences.			
				Steam-Pipe minus Chest.	Steam-Chest minus Exhaust.		
Engine standing, valve in mid position. . .)	59.25	58.25	17	Lbs. 1	41.25	15.9	—
Engine standing, valve in mid position. . .)	129.75	62.25	17	67.5	45.25	12.6	$\frac{12.6}{15.9} = 0.8$
Engine running at 50 revs. . .)	61.25	59.75	19	1.5	40.75	28.9	—
Engine running at 50 revs. . .)	132	63	17.75	69	45.25	20.4	$\frac{20.4}{28.9} = 0.7$

There is therefore a reduction of about 25 per cent. in the leakage, if wire-drawing to the above extent be resorted to.

Effect of Speed upon Leakage.—At the same pressure and under the same conditions of lubrication and jacketing, leakage

diminishes with increased speed of sliding. There is considerably less leakage when the engine and therefore the valve is stationary in mid position, but when the valve is moving there is appreciably less total leakage per hour when the engine is making 250 over the leakage at 50 double-strokes per minute. The leakage is consistently less at the high speeds than at the low where the cylinder walls are jacketed and the surfaces lubricated. This may partly be due to a more perfect spreading of the oil film at the higher speeds, or it may be due to the reduced time allowed for condensation and re-evaporation, or to a combination of the two. This clearly demonstrates that leakage was not in the present instance produced by the lifting of the valve in working, but by a more or less steady flow of moisture or steam between the valve and slide-face. The persistent and considerable difference between the leakage, when the engine is still with the valve in mid position and when it is running, points strongly to the conclusion that the amount of overlap and its variation has an important effect upon the leakage.

To elucidate this point, a series of leakage measurements were made with the engine standing at nine different positions of the crank, corresponding to nine successive positions of the valve during a revolution, the ports being blocked as before. The following Table 4 (page 216) gives the results of these measurements. They were made at the highest pressure of the boiler.

It is interesting to note that the mean value of the leakage at the nine positions is 45.95 lbs. per hour as compared with 45 lbs., the value obtained when running at 50 revolutions; also that at mid-position the mean of the two measurements is 36.9 lbs. per hour or practically the same as the mean of the values previously found.

Further, the variations in leakage occur with corresponding variations in the combined overlap at the two ends of the valve, the variations being very closely in the inverse ratio to the overlap. The following illustrations help to make this clear, Fig. 33 (page 217).

Analysis of Leakage and Method of applying Correction.—Assuming with Messrs. Callendar and Nicolson that the leakage varies as the perimeter round which escape from a higher to a lower pressure can take place, and to the difference of pressure, and inversely as the

TABLE 4.
Leakage past Slide-Valve, Engine standing at different Valve-Positions.

Position of Valve.	Position on Diagram, Fig. 33.	Steam pipe before inlet. Lbs. per sq. in.	Pressures absolute.		Difference Chest minus Exhaust. Lbs. per sq. in.	Leakage. Lbs. per hour.
			Steam-chest. Lbs. per sq. in.	Exhaust. Lbs. per sq. in.		
Mid position . . .	1	151	146	20·1	125·9	37·8
$\frac{1}{2}$ th forward stroke of valve—crank at 45° }	2	150·5	145·5	22	123·5	46·8
Front end full open to steam—crank 60° end of valve stroke . }	3	150	144	23·2	120·8	47·6
Expansion valve at cut off—crank 90° . }	4	151·5	145·75	21·25	124·5	56·4
During expansion (front end)—crank at 120° }	5	151	145·25	20	125·25	46·8
Valve in mid position—crank 180° . }	6	154	148·5	20	128·5	36
Back end full open—crank 240° . }	7	152	146	21	125	41
End of valve stroke. expansion valve at cut off—crank 270° }	8	151·25	144·5	21·75	122·75	52·8
During expansion (back end)—crank 300° . }	9	152	146	19	127	43·3

overlap of the valve over the ports, the measurement of the actual leakage under given conditions will enable the factor to be determined, by which that ratio must be multiplied to obtain the mean rate of leakage at any pressure. The present experiments show that the factor so found is not a constant.

The leakage is not directly proportional to the pressure for all speeds, and there is reason to doubt whether it is exactly inversely proportional to the overlap. But these assumptions are sufficiently accurate, if suitable values of the factor are introduced, to enable an approximate correction for leakage to be made by their aid.

The total leakage found with jacketed and lubricated surfaces and given in the latter columns of Table 2 (page 211) is, as already pointed out, larger than the leakage actually taking place in the

FIG. 33.

Diagrams showing position of Main and Expansion Slide-Valves at nine points of Stroke, referred to in Table 4.

engine under working conditions. By the help of the following diagrams a fairly close estimation of the relative leakages under the two sets of conditions may be made.

Fig. 34 (page 218) represents on an empirical scale the leakage which will take place under working conditions, if the assumptions above are approximately true.

Fig. 35 (page 219) similarly represents the leakage with blocked steam-ports. Of the total leak about 76 per cent. appears to be leak

FIG. 34.—Leakage through Slide-Valves. Probable Leaks during Working Stroke.

FIG. 35,
Probable Leakage into Exhaust when running with Blocked Steam-Ports.

direct to the exhaust, and the remainder to be steam which passes into and out of the cylinder. This latter quantity therefore is apparent on, or absent from, the indicator diagrams according to the point of the stroke at which examination is made. The leakage into the cylinder during expansion appears to be about 22 per cent. of the total leak, and the leakage out of the cylinder during the same period about 10 per cent. of the total leak. The net leakage into the cylinder during the expansion period will therefore amount to about 12 per cent. of the total leakage.

In comparative units (*see* Fig. 34, page 218) the leakage to exhaust from both chest and cylinder equals $\frac{12.4}{13.18}$ of the corresponding values on Fig. 35. Therefore the total leakage up to release under actual working conditions will be about 0.95 times the leakage measured under test conditions. This quantity should therefore be deducted from the total measured exhaust before the weight of steam actually present in the cylinder up to release will be known.

To determine similarly the weight of steam present in the cylinder up to cut-off, from the total measured exhaust must be deducted all the steam which leaks into the exhaust from the chest, together with all the steam which leaks into the cylinder, after cut-off. That is, on Fig. 34, the areas $O + P + Q + R + U + V$ (14.34 in comparative units); so that up to cut-off the steam to be deducted from the total measured exhaust, in order to determine the steam which is in the cylinder at cut-off, will be $\frac{14.34}{13.18} = 1.1$ times the total measured leak with steam-ports blocked. These values have been employed in determining the corrected dryness-fractions given in Table 8 (page 228).

It will be interesting to determine the value of the factor C by which $\frac{\text{Difference of pressure} \times \text{perimeter}}{\text{Mean overlap}}$ should be multiplied in order to give the mean rate of leakage, and so to compare with the corresponding values found by Messrs. Callendar and Nicolson. The perimeter measured to the mean overlap round the steam-port and on the side and ends of the exhaust-port for one end of the slide-valve for the

valve of the experimental engine is 22.5 inches—the mean overlap is 1 inch.* The mean value of $\frac{\text{overlap}}{\text{perimeter}}$ is therefore 0.045. Taking mean values of the leakage as plotted on Fig. 32 (page 212), the value of the factor C will be as follows:—

TABLE 5.

Pressure in Chest. Lbs. per sq. in. absolute.	Difference of Pressure. Chest minus Exhaust in lbs.	Syphon Lubricator on Steam-Chest stopped.	Syphon Lubricator on Steam-Chest running freely.
35	18.75	0.032	0.026
63	46	0.022	0.019
93	74.75	0.018	0.016
145	125.25	0.0142	0.0145
Averages		0.0215	0.019

As before stated, C is not a constant for the range of pressures experimented upon, although its mean value (namely, 0.02) is identical with the value found by Messrs. Callendar and Nicolson.

As already stated, its variation is probably due to the approximate character of the assumptions above made.

Percentage of Leakage to Steam passing through Cylinders.—If Tables 1, 2, 13 and 14 (pages 210, 211, and 252–255) be compared, it will be seen that the leakage, under given conditions of pressure, is equal to a maximum of 20 per cent. of the total steam used on the A₁ trial down to a minimum of 4 per cent. of the total steam used on the DD₄ trial. It is therefore in no case negligible.

Piston Leakage.—The leakage through the piston was measured by blocking the port at one end of the cylinder and running the engine single-acting. Cards of the working and blocked ends were

* For dimensions of valves and ports, see Appendix II (page 258).

also taken. One of these is shown on Fig. 36. It was found that the piston leak was independent of revolutions per minute, and was proportionate to the admission pressure and area of indicator diagram on the working side of the piston. The piston leak was found at its worst to be less than 2 per cent. of the steam consumption of the engine.

FIG. 36.—Diagram showing Piston Leak.

SERIES I.—JACKETED TRIALS.

In the jacketed series the barrels and both ends of the cylinders were jacketed with steam at boiler pressure, the discharge from the jackets being collected and measured.

Analysis of Results.—Table 13 (pages 252 and 253) shows the records obtained on the nineteen jacketed trials. At the highest pressure and greatest speed it was found impossible to obtain a consistent satisfactory trial, owing to the limits of the boiler used for the trial being reached. After a number of attempts therefore all the trials under these conditions being found to be unsatisfactory were discarded.

On a detailed examination of the records the following points are noteworthy :—

1. *Indicator Diagrams.*—(a) Admission line. For the series of trials at the lowest admission temperature, it was found necessary, as already mentioned, to raise the boiler pressure above that required in the cylinder so as to give a margin for driving the feed-pumps. In this series therefore the inlet-valve was used as a reducing valve, and this is clearly shown on the indicator diagrams by the considerable drop which there is between the pressure at the boiler side of the inlet-valve and the admission pressure on the diagrams. The wire-drawing resulting from this use of the valve caused superheating in some of the trials of this series.

The maximum difference between the pressure of the boiler side of the inlet valve and the steam-chest pressure occurring on the A_4 trial at 200 revolutions per minute, when a reduction from 47.1 to 33.2 lbs. per square inch was recorded, corresponding to a drop in temperature of 21.2° F., and on the B_1 trial where the pressures were 75.8 and 55.1 respectively, corresponding to a drop in temperature of 21° F., and the D_1 trial with a pressure before the inlet valve of 164.7 lbs. and in the steam-chest of 141 lbs. corresponding to a drop in temperature of 13° F. The minimum recorded difference was on the A_2 trial at 100 revolutions per minute, the pressures being 39.2 and 37 lbs. per square inch, giving a difference of 3° F. only. With the single exception of the B_1 trial, there was in no case superheating at cut-off.

(b) *Expansion Line.*—As the ratio of expansion was kept constant for the whole of the trials, the release pressure increased concurrently with the admission pressure. The constant volume rejection to exhaust therefore represents a larger and larger proportion of the whole range of pressures as the admission pressure rises. This is due to the fact that the ratio of expansion necessary for the lower pressure trials is not the most economical for the higher pressure trials. On the other hand, the high temperature at release on the higher pressure trials must ensure a temperature for the cylinder walls and end and piston surfaces, and probably throughout the exhaust, higher than in the low-pressure trials. The surface temperature and the critical dry temperature of the walls and surfaces therefore will be higher the higher the temperature of admission.

TABLE 6.

*Jacketed Trials.**Range of Temperature.*

Trial.	Initial Pressure.	Release Pressure.	Initial Temperature.	Release Temperature.	Difference.	Difference between Temperature at Release and 212° F.	Difference between Initial Temperature and 212° F.
	Lbs. per sq. in. abs.						
A ₁	31	15	252	212	40	0	40
B ₁	44	25	285	239	46	27	73
C ₁	99	49	327	279	48	67	115
D ₁	147	64	356	296	60	84	144
A ₂	35	15	259	212	47	0	47
B ₂	56	24.5	288	238	50	26	76
C ₂	100	45	327	274	53	62	115
D ₂	153	63	360	295	65	83	148
A ₃	32.5	15.75	254	209	45	—3	42
B ₃	59	24.5	291	238	53	26	79
C ₃	100	40	327	267	60	55	115
D ₃	140	63	358	295	63	83	146
A ₄	33	15.75	255	209	46	—3	43
B ₄	55	23	287	235	52	23	75
C ₄	99	40	327	267	60	55	115
D ₄	145	55	355	287	68	75	143
A ₅	37.5	17	263	219	51	7	51
B ₅	57.5	24.5	289	238	51	26	77
C ₅	88	35	318	259	59	47	106

The trials have been arranged in order of speeds.

Table 6 (page 224) will help in comparing the trials from this point of view.

In it are given the admission and release temperatures, the range of temperatures between admission and release, and the range of temperatures between admission and exhaust, for the whole of the jacketed trials.

Exhaust and Compression.—With regard to the exhaust and compression portions of the stroke little need be said, as these are practically uniform for the whole series of trials.

SERIES II.—UNJACKETED TRIALS.

In this series, of which detailed results are given in Table 14 (pages 254 and 255), the indicator diagrams show much the same features as have been noticed on the jacketed series. The reduction of pressure through the inlet valve was in general more on the unjacketed than on the jacketed series. The maximum drop on any of the trials occurred in the AA series. On the AA₄ trial at 200 revolutions, the difference between the inlet-valve pressure and the admission pressure was 19·7 lbs., corresponding to a difference of temperature of 25·6° F. On the AA₅ trial at 250 revolutions, at which the pressure at the boiler side of the inlet valve was 48·9, the pressure at admission was 37, corresponding to a difference in temperature of 17·3° F. The effect of wire-drawing through the valve is not apparent on the admission line of the lower speed trials; but, as the speed increases, the admission from being practically a straight line parallel to the base slopes down considerably, and at the same time the cut-off becomes more and more rounded instead of being sharp and distinct.

The only other trials upon which any considerable reduction occurred between the pressure at the back of the inlet valve and admission pressure were at the highest speeds of the CC (that is, the 315° F. unjacketed) and DD (that is, the 350° F. unjacketed) series. On the CC₃ trial, namely the trial at 150 revolutions and 315° F., there was a drop of 7 lbs. pressure and 5° F. in temperature

between steam-pipe and admission. On the DD₁ trial, that is, the trial at 50 revolutions, 350° F., the corresponding pressures (page 254) were 160·4 in the steam-pipe and 143·5 in the steam-chest, a drop of 16·9 lbs. pressure and 9° F. in temperature. This is the trial showing the maximum difference in the DD series.

Expansion.—With a constant ratio of expansion and varying initial pressure and speeds the same effect is produced, as was noticed in the jacketed trials.

Table 7 (page 227) gives the temperature ranges and difference for the unjacketed trials.

COMPARISON OF JACKETED AND UNJACKETED SERIES.

Condensation during Admission and re-evaporation during Expansion.
—Table 8 (page 228), Columns 1 and 2 give the dryness-fractions for the whole of the trials for both series measured in the usual way, as the ratio between the steam measured upon the indicator cards up to cut-off and release and the measured condensed discharge from the exhaust and including clearance in each case. As before stated, the measured condenser discharge, including as it does all leakage, will not give a true measure of the weight of steam and moisture in the cylinder at cut-off. If the leakage for the known conditions of each trial be taken from the leakage curve, a new ratio can be found by calculation from the leakage area diagram which will represent the real dryness-fraction, for the steam actually present in the cylinder at cut-off. The values of this actual dryness-fraction for each trial are given in Columns 3 and 4 of Table 8. It will be seen that the leakage correction thus applied materially affects the estimate of condensation during admission. Similarly, the dryness-fraction at release, measured in the usual way as a fraction of the condensed exhaust plus clearance, is given in Column 2, while its true ratio, after allowing for leakage, is given in Column 4 of the same Table.

To show the re-evaporation during expansion, Columns 5 and 6 have been added. The former gives the percentage of the whole weight of mixed steam and water which would have been present

TABLE 7.

*Unjacketed Trials.**Range of Temperature.*

Trial.	Admission Pressure.	Release Pressure.	Admission Temperature.	Release Temperature.	Difference.	Difference between Temperature at Release and 212° F.	Difference between Initial Temperature and 212° F.
	Lbs. per sq. in. abs.						
AA ₁	38	19	263	224	39	12	51
BB ₁	55	32	286	253	33	41	74
CC ₁	98	50	326	280	46	68	114
DD ₁	143	66	354	298	56	86	142
AA ₂	36	18	260	221	39	9	48
BB ₂	55	29	286	247	39	35	74
CC ₂	90	45	320	274	46	62	108
DD ₂	142	62	354	294	60	82	142
AA ₃	36	17.5	260	220	40	8	48
BB ₃	54	26	285	242	43	30	73
CC ₃	99	43	327	271	56	59	115
DD ₃	141	60	353	292	61	80	141
AA ₄	39	17.5	265	220	45	8	53
BB ₄	54	25	285	239	46	27	73
CC ₄	90	40	320	267	53	55	108
DD ₄	128	53	346	284	62	72	134
AA ₅	39	17	265	218	47	6	53
BB ₅	56	24	288	237	51	25	76
CC ₅	89	36	319	260	59	48	107

TABLE 8.

Trial.	Steam present in Cylinder as fraction of measured exhaust, neglecting leaks.		Dryness-fraction of Steam present in Cylinder if measured exhaust be corrected for leakage.			Theoretical dryness-fraction at release if expansion from cut-off were adiabatic.
	At cut-off. 1	At release. 2	At cut-off. 3	At release. 4	Dryness-fraction at release, if leakage into cylinder during expansion be deducted. 5	
A ₁	0.76	0.92	0.94	1.11	1.02	0.92
A ₂	0.83	0.97	0.94	1.10	1.08	0.92
A ₃	0.87	1.00	0.93	1.07	1.04	0.90
A ₄	0.85	1.00	0.89	1.05	1.03	0.87
A ₅	0.81	0.95	0.85	0.99	0.97	0.82
B ₁	0.90	0.99	1.16	1.2	0.98	1.08
B ₂	0.8	0.91	0.88	0.98	0.93	0.84
B ₃	0.83	0.96	0.88	1.03	0.98	0.84
B ₄	0.82	0.97	0.86	1.02	0.99	0.82
B ₅	0.83	0.94	0.87	0.98	0.95	0.83
C ₁	0.71	0.84	0.82	0.95	0.86	0.78
C ₂	0.79	0.91	0.85	0.98	0.93	0.81
C ₃	0.80	0.88	0.84	0.92	0.89	0.81
C ₄	0.83	0.90	0.87	0.95	0.92	0.82
C ₅	0.82	0.88	0.85	0.91	0.89	0.80
D ₁	0.82	0.89	0.96	1.02	0.92	0.91
D ₂	0.80	0.85	0.86	0.92	0.86	0.83
D ₃	0.85	0.90	0.90	0.95	0.91	0.85
D ₄	0.84	0.86	0.88	0.90	0.87	0.83
AA ₁	0.46	0.60	0.53	0.68	0.64	0.52
AA ₂	0.54	0.70	0.58	0.75	0.73	0.57
AA ₃	0.60	0.75	0.64	0.80	0.74	0.62
AA ₄	0.77	0.91	0.82	0.97	0.96	0.79
AA ₅	0.71	0.84	0.74	0.87	0.87	0.73
BB ₁	0.49	0.64	0.56	0.72	0.67	0.55
BB ₂	0.61	0.74	0.65	0.79	0.76	0.64
BB ₃	0.63	0.76	0.66	0.81	0.78	0.65
BB ₄	0.67	0.81	0.70	0.84	0.80	0.68
BB ₅	0.71	0.84	0.74	0.86	0.85	0.72
CC ₁	0.52	0.65	0.59	0.73	0.66	0.58
CC ₂	0.63	0.74	0.68	0.79	0.76	0.66
CC ₃	0.68	0.80	0.71	0.84	0.81	0.70
CC ₄	0.73	0.81	0.76	0.85	0.83	0.74
CC ₅	0.72	0.81	0.74	0.83	0.81	0.72
DD ₁	0.63	0.69	0.72	0.78	0.69	0.70
DD ₂	0.68	0.78	0.73	0.84	0.79	0.71
DD ₃	0.74	0.80	0.78	0.85	0.81	0.75
DD ₄	0.74	0.82	0.76	0.85	0.81	0.74

in the cylinder at release as steam, if no leakage had taken place between cut-off and release, while Column 6 gives the dryness-fraction which would have resulted from an adiabatic expansion of the mixture present at cut-off. The difference

FIG. 37.

*Diagram showing Variation of Dryness-Fraction at Cut-off and Release ;
with Speed and Pressure.*

(Dotted lines = Dryness fraction at cut-off
Full " = " " " " Release

between the values in Columns 5 and 6, stated as a percentage of the steam and water actually in the cylinder at release, gives an approximately accurate determination of the quantity of

FIG. 38.

Curves showing Variation of Dryness-Fraction and Total Leakage up to Release (shaded area), with Pressure and Temperature.

FIG. 39.—Curves showing Variation of Dryness-Fraction and Total Leakage up to Release (shaded area), with Speed.

steam which has been formed by re-evaporation during expansion. It must therefore be a measure of the regenerative action of the cylinder walls. It will be seen that in several instances the steam is not only dry but superheated at release. Such superheating only takes place on the jacketed trials, and there is strong internal evidence to show that in the most marked case, that of B_1 , there was superheating throughout the stroke. In such a case, the leakage of the valves is very greatly reduced, and the values calculated from the leakage curves can no longer be relied on. From the curves of variation of the dryness-fraction given in Fig. 37 (page 229), it appears probable that the valve leakage on trial B_1 , if it existed at all, was very small; in such a case the values of the dryness-fraction calculated without leakage, as given in Columns 1 and 2 of Table 8, represent much more closely the actual facts than the values in 3 and 4 of that Table. The superheating at admission in trials A_1 , A_2 , A_3 , and A_4 has already been dealt with. Figs. 38 and 39 (pages 230 and 231) show the values of the corrected dryness-fraction at release, and the difference between the values in 4 and 5 has been shown hatched, the intercept between the two curves representing leakage up to release expressed as a percentage of the steam and water in the cylinder for each trial. The values have been plotted for both varying speeds and varying pressures. It will be noticed that as the leakage per hour when the engine is running is appreciably constant for all speeds, the leakage, expressed as a percentage of the mixture present in the cylinder, diminishes with speed for any pressure and increases with pressure for any speed. It will also be seen that the dryness-fractions for jacketed and unjacketed trials converge towards one another both with increase of temperature and increase of speed.

On Fig. 37 (page 229) is shown at a glance the dryness-fraction at cut-off and release for all the trials plotted horizontally on a scale of revolutions and vertically on a scale of pressures. It will again be seen that in both directions the dryness-fractions for jacketed and unjacketed trials converge towards one another and towards unity. The dryness-fractions at cut-off and release also clearly converge towards one another in both directions. There is only one marked

divergence from a straight line throughout the observations, namely, for both fractions for the B₁ (245° F. 50 revs. jacketed) trial. If the leakage correction for this trial be omitted, the fractions would, as shown, fall upon the proper straight lines. This was the trial when wire-drawing, and therefore initial superheating was a maximum, and thus the deduction already drawn, that in such a case leakage will be reduced or eliminated, is confirmed in a very interesting and instructive way.

Regenerative action of Cylinder Walls.—The area of walls and clearance surface of the cylinder up to cut-off is 1.355 square feet, and up to the end of the stroke is 2.6 square feet. To cover the surface up to cut-off with a film of moisture 0.001 inch thick would therefore require 0.0072 lb. of water. At the end of the stroke a similar film would weigh 0.0136 lb. Bearing these quantities in mind, an estimate of the thickness of the film of dew formed by condensation during admission on each trial could be made.

The following Tables 9 and 10 (page 234) give, first, the weight of steam condensed during admission, and, second, the net re-evaporation during expansion, after correcting for leakage.

Table 11 (page 235) gives approximately the corresponding average rate of interchange of heat in lb. F.° units per square foot of surface per minute, between the steam and the admission surfaces, and between the total cylinder surface and the steam during expansion.

Firstly—Condensation during Admission.—Table 9 shows that, excepting those trials at lowest speed where wire-drawing through the stop-valve has modified the results, the total condensation up to cut-off increases with increased pressure and temperature of admission. As the cut-off was constant, condensation therefore increases with the difference between the temperature of admission and exhaust. This difference of temperature is sensibly the same for the jacketed and unjacketed series, yet the condensation increases with increase of temperature in a much higher ratio on jacketed than on the unjacketed trials. For example, on the jacketed trials at 150 revolutions the condensation at 350° F. initial temperature is

TABLE 9.

Condensation on Cylinder Walls.

Suffix denoting Speed.	Revs.	Steam condensed during admission in lbs. per 1,000 strokes.							
		Jacketed Series.				Unjacketed Series.			
Admission temp. {		A 245° F.	B 280° F.	C 315° F.	D 350° F.	AA 245° F.	BB 280° F.	CC 315° F.	DD 350° F.
1	50	0.40	0	5.7	1.6	8.5	12.3	18.0	15.0
2	100	0.45	0.83	4.0	6.0	6.3	8.0	11.5	12.6
3	150	0.48	1.7	4.2	4.8	4.8	6.8	9.3	9.8
4	200	0.90	1.9	3.5	4.6	1.86	5.4	7.0	9.1
5	250	0.91	2.0	3.7	—	2.9	4.5	5.8	—

TABLE 10.

Re-Evaporation on Cylinder Walls.

Suffix denoting Speed.	Revs.	Steam re-evaporated during expansion in lbs. per 1,000 strokes.							
		Jacketed Series.				Unjacketed Series.			
Admission temp. {		A 245° F.	B 280° F.	C 315° F.	D 350° F.	AA 245° F.	BB 280° F.	CC 315° F.	DD 350° F.
1	50	0.71	0	3.3	0.4	2.2	3.2	3.7	0
2	100	1.2	0.6	3.3	1.4	2.4	2.6	3.6	4.0
3	150	0.98	2.0	2.5	2.3	1.4	2.3	3.6	2.8
4	200	1.4	2.3	2.5	1.86	1.7	2.14	2.8	3.1
5	250	0.93	1.9	2.1	—	1.62	2.3	2.5	—

ten times the condensation at 248° F. initial temperature, while on the unjacketed trials at the same speed the condensation at 350° F. is only twice that at 245° F. On the other hand, the condensation on the

TABLE 11.

Mean Rate of Interchange of Heat between Cylinder Walls and Steam during the Forward Stroke.

Suffix denoting Speed.	Revs.	British Thermal Units per sq. ft. of exposed surface per minute transmitted from steam to walls during admission.							
		Jacketed Series.				Unjacketed Series.			
Admission temp. {		A 245° F.	B 280° F.	C 315° F.	D 350° F.	AA 245° F.	BB 280° F.	CC 315° F.	DD 350° F.
1	50	65	—	1000	295	1460	2200	3100	2580
2	100	150	277	1370	1970	2250	2720	3920	4150
3	150	244	865	2190	2020	2460	3550	4850	4975
4	200	610	1320	2350	3000	1230	3580	4625	5700
5	250	795	1610	3100	—	2400	3990	5100	—
		British Thermal Units per sq. ft. of exposed surface per minute transmitted from walls to steam during expansion.							
1	50	36	—	183	25	119	180	200	—
2	100	123	62	350	145	269	276	373	448
3	150	155	318	410	348	224	423	575	423
4	200	297	490	525	270	350	460	548	605
5	250	255	480	555	—	420	635	655	—

unjacketed trial at 150 revolutions and 245° F. initial temperature is ten times that for the jacketed trial at the same temperature and speed, while at 200 revolutions and 350° F. temperature the condensation

without jackets is only twice that with jackets at the same temperature and speed. These facts bear out the conclusion arrived at below from a consideration of the variation of initial condensation expressed as a percentage of the steam entering the cylinder, that *the losses due to condensation on the cylinder walls of unjacketed engines diminish with increased initial pressure and temperature for a given ratio of expansion, quite independently of the influence of speed.* There are not wanting indications in the results here recorded, that if only the initial temperature could be sufficiently raised, jacketed and unjacketed engines would be equally efficient as regards initial condensation.

TABLE 12.

Suffix denoting Speed.	Revs.	Steam entering Cylinder up to Cut-off in lbs. per 1,000 strokes corrected for Leakage.							
		Jacketed Series.				Unjacketed Series.			
Admission temp. {		A 245° F.	B 280° F.	C 315° F.	D 350° F.	AA 245° F.	BB 280° F.	CC 315° F.	DD 350° F.
1	50	7·7	12·2	30·9	38·6	18·6	29·9	44·8	53·7
2	100	7·8	15·2	27·8	43·0	15·2	23·3	36·0	46·9
3	150	7·1	15·0	27·2	40·7	13·4	20·4	32·8	44·9
4	200	8·4	13·7	26·8	37·6	10·7	19·2	29·7	38·1
5	250	9·4	15·5	24·1	—	11·5	17·3	26·8	—

Secondly.—Total condensation during admission—This decreases with increase of speed throughout the unjacketed series of trials, with the single exception of the trial at 200 revolutions and 245° F. initial temperature. This trial has already been noted as one which was largely affected by superheating due to wire-drawing through the stop-valve. On the jacketed series, on the two higher temperature series, the same decrease of total condensation with increase of speed is observable, but on the two lower temperature-series the opposite is the case—the total condensation increasing with speed. This

apparent anomaly is probably largely due to the fact that throughout these two last-named series the weight of steam entering the cylinder is comparatively small, and increases rapidly with speed, so that as shown later the condensation expressed as a percentage of the steam entering the cylinder does not increase with the total condensation. The total condensation on these series is moreover so small that the heat transference is at a comparatively low rate, so that increase of speed does not affect the result to the same extent as where larger weights of steam are in question. It is noticeable that the actual rate of heat transference as shown in Table 11 (page 235) increases regularly with speed and with temperature. The weight of steam entering the cylinder as shown in Table 12 increases with speed (with one or two exceptions) during the 245° and 280° F. jacketed series, but decreases with speed in all other cases except those already noted as anomalous.

Thirdly.—Rate of Interchange of Heat between Steam and Cylinder Walls.—Table 11 gives the rate of interchange of heat from steam to cylinder walls during admission, and from walls to steam during expansion. It will be noticed that the rate of interchange of heat from the steam to the walls on the unjacketed trials approximates to twice that on the jacketed trials. This corresponds to the greater depth to which the cyclical temperature variation penetrates, when the mean temperature of the cylinder walls is lowered by the absence of jackets. This has previously been pointed out by Messrs. Donkin, Callendar and Nicolson, and other observers, but is here quantitatively analysed for the first time, as far as the reporter is aware.

Fourthly.—Re-evaporation during Expansion.—During expansion the total weight of steam evaporated, including that liquefied by expansion, has a maximum value of about 3.5 lbs. per 1,000 strokes. It is important to realise that this maximum appears to be sensibly the same on both jacketed and unjacketed trials. Generally the evaporation without jackets is greater than with jackets, showing that the regenerative action of the cylinder walls during expansion

is not appreciably helped by jacketing. During expansion there is not time for the higher mean temperature of the walls when jacketed to influence the evaporation by conduction. When the cylinder is unjacketed there is a much larger weight of condensed moisture to draw from than when jacketed, and it will be noticed that the evaporation on the jacketed trials in many cases approximates to the total condensation, so that no large increase could be looked for, while on the unjacketed trials this is only the case on the higher speed trials. Yet in neither series does the evaporation appear to vary at all closely with the condensation, though there are some indications of such a tendency.

Further, the mean rate of heat transmission from the walls to the moisture is in all cases extremely low compared with the rate of transmission in the reverse direction during admission, so that it seems difficult to believe that the limit of evaporation is in any case reached, even though it be assumed that the evaporation takes place chiefly towards the end of expansion, so that the actual rate is much higher than the mean.

It appears clear, therefore, that the jackets have little, if any, influence upon evaporation during expansion, and that under the conditions under which these trials were carried out the regenerative action of the walls is quite as high without jackets as with them, even at the low speeds where the jacketed engine is largely more economical than the unjacketed.

Influence of Ratio of Steam in Jackets to Steam through Cylinders upon Condensation and Re-evaporation.—From Cols. 19, 20 and 22 of Table 13 (page 253), fifthly, it will be seen that the ratio of thermal units through the jackets to the thermal units passed through the cylinder varies between very wide limits. For example, on the trial at 245° F. temperature and 50 revolutions the value of this ratio was 0.724, while on the trial at 356° F. temperature and 200 revolutions it was 0.042. The reporter has sought in vain for any indication, either in the condensation, re-evaporation, or the heat consumption that this variation has any influence which can be reduced to law. The former of these trials has condensation on the walls equal to

0.40 lb. per 1,000 strokes as compared with 0.45 lb. on the 245° F.-100 revolutions trial. The re-evaporation in the first is less than the last, being 0.71 as against 1.2 for the 245° F.-100 revolutions trial, while the efficiency ratio, excluding the jacket steam, is 0.486, as compared with 0.625 for the 245° F.-100 revolutions trial. Any indication of influence therefore is in the contrary direction to what would be expected. The variation is in the direction in which variation would have been expected from the relative conditions of the two trials, without regarding jackets at all. The same in a less degree is true of all the variations in jacket steam throughout the trials. Any excessive number of thermal units absorbed in the jackets, however, as for example is the case with the A_1 and A_3 trials (at 245° F., 50 and 150 revolutions), coincide, with an abnormal total consumption of heat per I.H.P. per minute.

Ratio of Steam present in the Cylinder at Cut-off to Total Steam and Moisture ; Dryness-Fraction at Cut-off.—The relationship between initial condensation and the steam passing through the cylinder can best be studied by examining the variations of the complement—the dryness-fraction. This can best be seen by comparing the temperature entropy diagrams for the whole series.

Temperature Entropy Diagrams.—Entropy temperature diagrams have been drawn for all the trials, and have been superimposed in two series—the one set showing variation with increase of speed, the other with increase of initial pressure. On these latter, necessary corrections for leakage have been made. So as to make them comparable with diagrams drawn without allowance for leakage, they have been plotted for 1 lb. of total exhaust condensation water. The leakage has then been deducted at cut-off and release, so that, after deducting the enclosed area, the entropy of the steam measured on the indicator diagram compared with the entropy up to the edge of the enclosed leakage area represents the real or corrected dryness-fraction. To express with scientific accuracy the actual entropy temperature changes, it would be more correct to plot the diagram upon a new scale, in which the entropy of the steam, after

deducting leakage, was the unit. Such a diagram would not however show so clearly the effect of allowance or non-allowance for leakage in altering the relationship between steam and moisture in the cylinder.

Comparing first the superimposed diagrams for constant temperature and varying speeds and for convenience dealing with the unjacketed series first,—at the lowest pressure, namely the AA series, it will be noted that the expansion lines of the entropy diagrams are roughly parallel to one another, and that the proportion of steam present both at cut-off and at release increases in a regular sequence with the speed, with the exception of the AA₄ trial, which is exceptional. In the BB series the same uniform and regular increase in the dryness-fraction occurs without exception throughout the whole series. The 100-revolution trial however, shows deviation from parallelism to the other trials on the expansion curve. The condensation is less and the re-evaporation more than the relative speed would indicate. In both the above series the improvement in the dryness-fraction due to increase of speed diminishes as speed increases, the curves for the higher speeds closing up upon one another. In the CC series this is still more marked, the expansion curves for the three higher speeds being almost identical, so that at 150 revolutions per minute and at an admission temperature of 315° F., the effect of speed upon the dryness-fraction both at cut-off and release is practically reduced to zero. On the DD series this point is still further emphasised, at 100 and 150 revolutions (the admission temperature being 350° F.) the speed seems to have no further effect in increasing the dryness-fractions at cut-off and release.

Turning to the jacketed series, on the A series of trials at lowest pressure (the mean admission temperature for all the trials being about 245° F.), the effect of the jackets is at once apparent; all the expansion lines being brought closer to the theoretical saturation curve. All the trials seem to show closely the same results as regards percentage initial condensation; the jackets have therefore produced, in the lowest pressure series, the same result of closing up trials at different speeds upon one another that was shown on the unjacketed

series in the highest pressure series, and at the same time all the trials have been brought closer to the ideal of maximum dryness fraction.

On the B series of trials the jackets have so far affected the result that percentage initial condensation seems to depend no longer upon speed but upon the minor accidental conditions under which the trials are run. The variations in dryness-fraction are smaller than those of the unjacketed series previously referred to. With regard to the C trials the same remarks may be made, and so also with regard to the D series at the highest pressure. All the trials show very nearly the same percentage of initial condensation. The jackets therefore have so far influenced the result as practically to eliminate, except at the lowest speeds and pressures, the effect of speed upon percentage initial condensation.

If the diagrams for the corresponding jacketed and unjacketed trials be compared, one or two points of importance are apparent. Generally the expansion curve on the temperature entropy diagram for the unjacketed trial is very closely parallel to that of its jacketed counterpart, showing graphically what has already been noted that re-evaporation during expansion is as great on the unjacketed as on the jacketed trial. In some cases it is greater on the unjacketed trial. It will be noticed also that as the speed increases for any pressure the diagram for the unjacketed trial closes up upon its jacketed counterpart, until at the higher pressure of any speed the two come comparatively close to one another. In no instance, however, does the unjacketed diagram reach the jacketed diagram at the speed or pressure included within the range of the trials until the jacket steam has been added to the steam passing through the cylinder.

Further, a comparison of the diagrams accentuates the fact already referred to that both total condensation and dryness-fraction generally increase with increase of admission pressure and temperature for a given speed on the unjacketed trials. This does not at first sight appear logical.

Referring to Tables 6 and 7 (pages 224 and 227) however, a probable explanation becomes apparent. Due to the constant ratio of

expansion the range of temperature between cut-off and release does not increase in anything like the same ratio as the increase of initial temperature. Taking the 50-revolutions series, the range of temperatures during expansion on the lowest pressure trial, namely, AA₁ is 39° F., on the second pressure trial BB₁ 33° F., on the third CC₁ 46° F., and on the fourth DD₁ 56° F.

This increase in range is accompanied by a very much larger increase in the difference between release and exhaust temperatures. Following the trials in the same order—on the AA₁ trial the drop from release to exhaust is 12° F., on the BB₁ 41° F., on the CC₁ 68° F., and on the DD₁ 86° F. The temperature at which the cylinder walls and end surfaces is dry will be lower than the temperature of release, but will rise as the temperature of release rises, and it may be assumed upon previous evidence, that the surfaces after once becoming dry are not reduced in temperature to any very great extent. If this be true, the range of variation in the temperature of these surfaces will not increase in anything like the same ratio as that in which admission temperature is increased. At the lower pressure trials therefore, especially at the lower speeds, the cylinder walls and end surfaces will not become dry nearly so quickly nor so early in the return stroke as on the higher admission pressure trials, even if they become entirely dry at all.

A further explanation is to be found in the fact, to which reference has already been made, that the higher the admission pressure, the larger is the weight of steam admitted per square foot of surface. On referring again to Table 12 (page 236), where the weight of steam present per 1000 strokes on the respective trials is given, it will be noted that on any speed series the weight of steam per 1000 strokes increases in a nearly constant ratio; so that if the maximum limit of condensation is reached in any given case, the dryness-fraction for any higher pressure must of necessity increase in proportion to the increase of weight of steam admitted per square foot of surface. The ratio of increase of the dryness-fraction for a given rise of pressure will also be affected by the speed; and on comparing the different speed series, very interesting information is obtained bearing upon the pregnant suggestions thrown out by Messrs.

Callendar and Nicolson. On the jacketed series, as the whole phenomena of condensation are modified by the raised mean temperature of the cylinder metal and walls, the relative effect of improved dryness-fraction due to increased temperature at admission will be diminished, and it is therefore found that on the diagrams for the jacketed series the dryness-fraction tends to vary much less and less regularly than in the unjacketed series, and the variation at the higher speeds is in many cases in the reverse direction.

Range of temperature has therefore a larger effect upon the initial condensation, as the other conditions have been brought closer into equilibrium by the introduction of an artificially raised surface temperature, and causes reduction in the dryness-fraction instead of the increase shown where the other governing conditions are more variable and more powerful.

As the speed increases, and the ranges of temperature during expansion at the various pressures close up to one another, the difference between release temperature and exhaust becomes less than at the lower speeds. Added to this, the time during which condensation and cooling can take place is reduced, and the resulting benefit is shown most markedly in those cases which at the slow speeds were worst, namely, on the low-pressure trials.

At the highest speeds, therefore, the difference due to the higher critical dry temperature will be diminished, and the indirect benefit upon initial condensation tends to be neutralised and swamped by the larger benefit from increased speed which is shared in common.

To show the re-evaporation during expansion, the adiabatic for the mixture of steam and moisture present at cut-off has been drawn upon each temperature-entropy diagram, and the projection of the toe of the diagram beyond this adiabatic line will represent the re-evaporation due to regenerative action of the cylinder walls. At the higher speeds of the jacketed trials an interesting point is incidentally shown. There is a general tendency for the dryness-fraction at cut-off for the individual trials of any speed series to follow an adiabatic curve, so that the dryness-fraction for intermediate trials of the series can be very closely predicted from that found at

either extremity of the range of pressure by drawing an adiabatic curve for the given proportions of steam and water through that extremity.

On Figs. 21 to 30 (pages 198-207) the leakage deduction has been plotted on the temperature entropy diagram, showing the effect of this necessary allowance on the dryness-fractions at cut-off and release.

Diagrams of Steam Consumption. Willans lines, etc.—In Fig. 40 (page 245) will be found curves in which the ordinates represent pounds of steam used in the cylinder per hour, the abscissæ representing revolutions per minute. The full lines represent for each pressure series the total measured steam-consumption per hour when jacketed. The dot and dash lines represent the corresponding consumption, including jacket steam, and the dash lines represent in corresponding units the consumption when unjacketed. These diagrams simply confirm the results obtained by Mr. Willans in his classical experiments, and graphically show how the weight of condensed steam increases as the mean pressure—that is to say, as the work done in the cylinder—increases. In the present case including the jacket steam, these diagrams show that at the higher pressures and temperatures the consumption of the jacketed engine is as large as the unjacketed even at the lower speeds. An approximate thermal unit scale has also been added showing the comparison on a direct heat basis. Figs. 41 and 42 give the corresponding values of steam and heat units consumed on an effective pressure base.

Fig. 43 (page 247) shows the total weight of steam per hour and the B.Th.U.'s per minute consumed by the engine on a H.P. base.

General Conclusions.—It is dangerous to draw conclusions of too general and sweeping a character from experiments upon one type of engine under one set of conditions, but it may fairly be claimed that the ground covered by this Report has never previously been surveyed with an engine showing more consistent and definite results, nor under conditions which enabled so detailed an analysis of the results

FIG. 40.

Steam Consumption in Lbs. per Hour.

FIG. 41.

Total Weight of Steam in Lbs.

to be made. It may therefore be anticipated with some confidence that the definite indications shown by the results form a real contribution to the present knowledge of the phenomena accompanying condensation and re-evaporation in an engine cylinder under given conditions of jacketing. The points which have been elucidated may be summarised as follows:—

Firstly, leakage through the slide-valve, to the importance of which Messrs. Callendar and Nicolson have drawn attention, has

FIG. 42.—Total B.Th.Units per minute.

been quantitatively determined under defined conditions, and has been shown to be nearly independent of speed of sliding surface and proportional to difference of pressure between the two sides of the valve. Further, it has been shown that the assumption that the leakage is inversely as the overlap of the valve is at least in the main well

FIG. 43.—*Lbs. of Steam (measured) per Hour.*

FIG. 44.

B.Th. Units per I.H.P. per min.

FIG. 45.

B.Th.Units per I.H.P. per min.

founded. And further, that with well fitted valves the leakage may amount to over 20 per cent. of the steam entering the cylinder, and is rarely less than 4 per cent.

Secondly, it has been shown that for an unjacketed engine with a given ratio of expansion initial condensation, expressed as a percentage of the steam in the cylinder, *diminishes* with increase of initial temperature, while the total condensation per stroke *increases* with such temperature increase.

This, though suggested by Messrs. Callendar and Nicolson's researches, has never previously been demonstrated with clearness, as if leakage is not allowed for, the results are obscured and even reversed, and the conclusions arrived at without leakage allowance are therefore unreliable.

Thirdly, it appears from the results here obtained that the re-evaporation for a given ratio of expansion is as great and sometimes greater without jackets than with them. This shows very clearly that the regenerative action of the cylinder walls with a given ratio of expansion is largely independent of their mean temperature.

No quantitative analysis of re-evaporation is possible where leakage is not taken into account, as without the necessary allowance results would be largely illusory.

Fourthly, it is possible from the results obtained to show the temperature when, for any speed of revolution with a given rate of expansion, the jackets will become unnecessary or wasteful. If the heat units per I.H.P. per minute required by the unjacketed engine for each speed of the series be plotted either on an initial pressure or a mean effective pressure base the points for each speed will be found to lie on four curves, which become closer and closer to one another as the speed increases, and all converge to a point as pressure or temperature increases. If the heat consumption for the jacketed series be likewise plotted, it will be found that the points for the different speeds at each pressure lie irregularly round a point, their exact position being determined by the accidentally slight variations of the conditions of each trial. A fair curve through the means of these points will lie below the corresponding curves for the unjacketed trials, but if the heat absorbed in the jackets be

included, the resultant curve cuts the unjacketed curves at points which for each speed indicate the temperature and pressure at which the jackets cease to be economical. Such curves are shown on Figs. 44 and 45 (pages 248 and 249). It will be seen that the full black line, which is the resultant mean for the jacketed trials, cuts both the 250 and the 200-revolution unjacketed curves within the temperatures and pressures included within the scope of the present experiments, and the temperatures and pressures where the 150, 100, and even the 50-revolution curves would cut, can very closely be predicted. As far as the reporter is aware, this is the first time when an exact determination for a given engine and ratio of expansion of this point has been diagrammatically shown.

In conclusion, the reporter wishes to thank most cordially the assistants who have so generously and ungrudgingly spared neither time nor thought in helping him in the arduous work of these trials extending over six years, and in working out and co-ordinating the results. Without their aid the work and the Report would have been impossible. To Professor H. M. Waynforth, especially, thanks are due for many excellent and fruitful suggestions, for ungrudging assistance in carrying out the trials and looking over, correcting and criticising the proofs, and to Mr. W. Mason for the care, thought, and originality he showed in carrying out the reporter's suggestions with regard to the leakage trials and in devising the empirical leakage area diagrams. The work done by the late Mr. James, whose untimely death has delayed the production of this Report, and has been a very serious loss, was admirably reliable. To Mr. Kiernan, as well as many of the reporter's students, thanks are also due.

The Report is illustrated by 45 Figs. and 14 Tables in the letterpress, and is accompanied by two Appendices (page 256).

TABLE 13. SERIES I.—Jacketed Trials (continued on next page).

252

STEAM-ENGINE RESEARCH.

MARCH 1905.

Trial	Duration. Minutes.	Pressures. - Lbs. per sq. inch.			Revs. per min.	Nett Brake Weight in lbs.	B.H.P.	I.H.P.	Steam through Cylinder in lbs.					Steam through Cylinder Jackets. Per hour.
		Steam before inlet abs.	Steam chest abs.	Mean effective.					From Cards.		Measurement by Weight.			
									Per min.	Per hour.	Total.	Per hour.	Per hour deducting Leakage.	
		1	2	3	4	5	6	7	8	9	10	11	12	13
A ₁	30	39.2	32.5	7.8	48.5	—	—	0.87	0.74	44	28	56	46	44
A ₂	20	39.2	37	9.2	97.2	—	—	2.06	1.58	91	34	102	90	14.25
A ₃	44	44.7	31	8.2	152.1	—	—	2.86	2.29	137	104	141.8	132	37.8
A ₄	20	47.1	33.2	9.6	203	—	—	4.45	3.20	192	72	216	205	18.75
A ₅	30	44.2	35.5	11.3	261.4	—	—	6.77	4.61	276	154	308	297	21
B ₁	25	75.8	55.1	26.1	44.5	79.6	1.72	2.66	1.31	78	35	84	66	18
B ₂	30	67.6	55	25.8	99.4	76.9	3.71	5.87	2.75	165	100	200	182	29
B ₃	30	74.0	56.7	27.1	151.7	81.8	6.02	9.39	4.36	261	146	292	274	18
B ₄	30	72.2	54.4	23.5	203.5	58.3	5.76	10.93	5.39	323	177	354	337	17.75
B ₅	30	66	56.4	26.7	240	44.5	8.21	14.67	6.72	403	233	466	448	36.5
C ₁	35	99.7	99.7	62.6	52.4	234.6	6.21	7.49	2.90	173	132	226.3	196	26.31
C ₂	30	102.0	98.7	59.7	104.5	217.6	10.71	13.84	5.15	309	188	376	346	42.0
C ₃	30	107.6	93.9	55.3	155.3	200.8	15.12	19.63	7.46	447	267	534	505	39.5
C ₄	30	101.7	96	55.3	200	188.0	18.24	25.25	9.42	565	335	670.5	641	33.5
C ₅	30	92.7	84.5	46.8	251	160.5	19.54	26.82	10.37	622	376	752	725	56
D ₁	20	164.7	141	93.2	56.5	373.5	10.24	12.04	4.3	258	101	303	262	18.375
D ₂	30	157.3	149.7	96.5	101.9	364.9	18.03	22.44	7.78	466	284	568	525	68.5
D ₃	20	151.5	146.5	94.8	143.9	344.0	24.01	31.14	10.83	649	248	744	702	37.5
D ₄	20	157.5	149.3	95.9	151.2	341.2	25.32	38.1	13.11	786	306	918	818	39.75

TABLE 13 (continued from opposite page).

Trial.	Duration. Minutes.	Steam per I.H.P. in Cylinder in lbs. per hour.			Upper limit of Temp. Abs. F. Tl.	Lower limit of Temp. Abs. F. Tl.	B.Th.U. for actual engine per min. per I.H.P.	B.Th.U. per min. per I.H.P. in jackets.	Total B.Th.U. cools. 19+20.	Value of ratio cylinder. Cols. 20/19.	B.Th.U. required by Standard Engine of Comparison per min. per I.H.P.		Efficiency Ratio of Actual to Standard.		
		Cards.	Measured.	Ditto if Leakage is deducted.							Rankine.	Carnot.	Cols. 23/19 to Rankine.	Including jackets. Col. 23/21.	Cols. 24/19 to Carnot.
		14	15	16	17	18	19	20	21	22	23	24	25	26	27
A ₁	30	51.5	64.6	52.8	727	670	1062	769	1831	0.724	570	510	0.486	0.311	0.480
A ₂	20	44.5	49.5	43.6	727	670	814	104	918	0.127	570	510	0.635	0.620	0.626
A ₃	44	48.0	49.6	46.1	734	670	815	200	1015	0.245	505	493	0.613	0.497	0.603
A ₄	20	43.1	48.5	46.0	738	671	798	61	859	0.076	483	471	0.605	0.562	0.599
A ₅	30	40.9	45.5	43.8	734	670	747	47	794	0.062	510	493	0.673	0.642	0.659
B ₁	25	29.4	31.6	24.8	769	670	525	104	629	0.198	345	331	0.657	0.548	0.630
B ₂	30	28.1	34.0	31.0	761	673	563	75	638	0.133	390	365	0.692	0.611	0.648
B ₃	30	27.8	31.1	29.0	767	669	517	29	546	0.056	349	330	0.676	0.639	0.638
B ₄	30	29.5	32.8	30.8	765	670	537	31	568	0.057	358	341	0.667	0.630	0.634
B ₅	30	27.4	31.8	30.4	759	670	527	38	565	0.072	374	356	0.709	0.662	0.674
C ₁	35	23.1	30.2	26.1	788	671	504	54	558	0.107	301	285	0.597	0.539	0.565
C ₂	30	22.1	27.1	25.0	790	671	453	47	500	0.104	298	283	0.658	0.596	0.623
C ₃	30	22.8	27.2	25.8	794	673	455	31	486	0.068	293	280	0.645	0.603	0.615
C ₄	30	22.3	26.5	25.5	789	672	442	20	462	0.045	303	284	0.685	0.656	0.641
C ₅	30	23.2	28.0	27.0	783	673	466	32	498	0.068	317	302	0.681	0.636	0.649
D ₁	20	21.4	25.1	21.7	827	670	494	24	518	0.048	240	223	0.522	0.463	0.450
D ₂	30	20.8	25.3	23.4	823	670	426	47	473	0.110	246	229	0.577	0.520	0.537
D ₃	20	20.8	23.8	22.5	820	673	401	19	420	0.047	252	237	0.628	0.600	0.590
D ₄	20	20.6	24.1	23.0	822	673	405	17	422	0.042	250	233	0.618	0.592	0.575

TABLE 14. SERIES II.—Unjacketed Trials (continued on next page).

Trial.	Duration. Minutes.	Pressures. Lbs. per sq. inch.			Revs. per min.	Nett Brake Weight in lbs.	B.H.P.	I.H.P.	Steam through Cylinder.				
		Steam before inlet abs. 1	Steam chest abs. 2	Mean effective. 3					From Cards.		Measurement by Weight.		
									Per min. 8	Per hour. 9	Total. 10	Per hour. 11	Per hour deducting leakage. 12
AA ₁	40	51.5	37.6	13.0	51.4	27.3	0.68	1.53	0.98	59	84	126	114
AA ₂	25	49.5	35.7	12.3	106.5	25.5	1.32	3.00	1.93	115	86	206	194
AA ₃	30	48.4	36.7	11.8	152	22.9	1.69	4.1	2.70	161	128	256	244
AA ₄	20	59.7	40	13.2	196.0	20.5	1.95	5.93	3.51	210	88	264	251
AA ₅	20	48.9	37	12.4	245.7	—	—	6.96	4.23	253	117	351	339
BB ₁	30	57.8	56	29.2	53.0	94.1	2.42	3.45	1.66	99	100	200	182
BB ₂	25	57.5	55.7	28.1	101.3	80.3	3.94	6.50	3.01	180	126	302	284
BB ₃	30	57.8	55	25.6	155.6	71.3	5.38	9.12	4.28	256	200	400	382
BB ₄	35	57.7	55	25.2	197.8	60.2	5.77	11.39	5.27	316	266	456	438
BB ₅	15	61.7	54.2	25.0	262.3	52.5	6.68	15.01	6.99	419	141	564	546
CC ₁	30	105.2	99.4	61.9	51.2	236.3	5.87	7.24	3.05	183	151	302	272
CC ₂	15	93.1	90.2	55.0	98.3	208.3	10.17	12.35	4.93	295	113	452	424
CC ₃	30	105.2	98.2	57.4	151	208.3	15.25	19.8	7.80	468	311	622	592
CC ₄	20	93.9	88.6	52.0	186.7	171.4	15.98	22.2	8.72	522	231	693	665
CC ₅	20	105.1	87.5	47.7	249.9	154.4	18.71	27.24	10.75	645	279	831	804
DD ₁	15	160.4	143.5	92.5	54.5	388.4	10.27	11.51	4.03	241	98	392	350
DD ₂	15	150.9	139.4	88.5	106.3	355.4	18.33	21.49	7.49	449	160	640	599
DD ₃	25	144.1	137.9	88.8	143.4	336.0	23.37	29.08	9.93	596	339	813	773
DD ₄	10	135.3	128.8	77.0	185.5	280.0	25.19	32.62	11.74	704	150	900	863

TABLE 14 (concluded from opposite page).

Trial.	Duration. Minutes.	Steam per I.H.P. in Cylinder in lbs. per hour.			Upper limit of Temp. Abs. F. 71.	Lower limit of Temp. Abs. F. 71.	B.Th.U. for actual engine per mht. per I.H.P.	B.Th.U. required by Standard Engine of Comparison per min. per I.H.P.		Efficiency Ratio of Actual to Standard.	
		Cards. 13	Measured. 14	Ditto if Leakage is deducted. 15				Rankine. 19	Carnot. 20	Cols. 19/18 to Rankine. 21	Cols. 20/18 to Carnot. 22
AA ₁	40	38.5	82.2	74.5	744	673	1351	455	441	0.349	0.326
AA ₂	25	38.4	68.6	65.0	741	673	1128	470	460	0.418	0.408
AA ₃	30	39.5	62.4	59.5	740	673	1026	473	465	0.460	0.453
AA ₄	20	35.4	44.4	42.3	753	672	750	414	392	0.563	0.523
AA ₅	20	36.4	50.4	48.7	741	673	828	483	465	0.570	0.561
BB ₁	30	28.2	56.4	52.7	751	671	932	414	397	0.444	0.425
BB ₂	25	27.8	46.5	43.7	751	673	768	414	397	0.539	0.517
BB ₃	30	28.1	43.8	41.7	751	672	724	414	397	0.539	0.548
BB ₄	35	27.7	40.0	38.3	751	672	661	414	397	0.572	0.600
BB ₅	15	27.9	37.5	36.3	755	671	622	400	378	0.552	0.607
CC ₁	30	25.3	41.7	37.5	792	673	695	299	283	0.430	0.406
CC ₂	15	23.9	36.2	34.3	783	673	603	317	302	0.525	0.500
CC ₃	30	23.6	31.4	30.0	792	673	523	299	283	0.570	0.540
CC ₄	20	23.5	31.2	30.0	782	672	519	319	203	0.615	0.390
CC ₅	20	23.6	30.5	29.5	792	673	503	299	283	0.587	0.556
DD ₁	15	21.0	34.0	30.5	825	673	573	249	230	0.434	0.401
DD ₂	15	20.9	29.7	27.8	820	673	499	254	237	0.508	0.474
DD ₃	25	20.5	27.9	26.5	816	673	469	259	242	0.551	0.515
DD ₄	10	21.5	27.5	26.4	811	673	462	266	249	0.575	0.538

APPENDIX I.
BEING A
REPORT TO THE
STEAM-ENGINE RESEARCH COMMITTEE,
ON PREVIOUS
PROGRESSIVE SPEED AND PRESSURE TRIALS.

BY PROFESSOR DAVID S. CAPPER.

(11 *March* 1896.)

So far as the writer has been able to discover after an extended search through French, German, English, and American records, only three series of steam-engine experiments have been carried out at progressive speeds and pressures.

Peabody.—In the first series Professor C. H. Peabody experimented in 1884-5* with a single-cylinder non-condensing engine of the Harris-Corliss type at the Massachusetts Institute of Technology. The cylinder was 8 inches diameter and 24 inches stroke. The boiler pressure was kept nominally constant throughout at 70 lbs. per square inch above atmosphere. But in the several trials variations occurred between 68 and 75 lbs. per square inch. The cut-off varied from 1.29 to 14.8 per cent. of the stroke. The speeds adopted were 23, 31, 36, 49, and 50 revolutions per minute. Each trial lasted about 1½ hour, and the water per indicated horse-power per hour was measured and compared with that shown upon the indicator diagrams.

Denton and Jacobus.—In 1888 Professors J. E. Denton and D. S. Jacobus made a more elaborate series of trials† of a simple non-condensing engine, with cylinder 17 inches diameter and 30 inches

* Transactions, American Society of Mechanical Engineers, vol. 7, page 328.

† Transactions, American Society of Mechanical Engineers, vol. 10, page 722.

stroke. The engine was built to drive one of the air compressors of the Rand Drill Co. Trials were run at 9, 11, 13, 17, 25, 60, 64, and 88 revolutions per minute, with a boiler pressure of 90 lbs. per square inch above atmosphere. The cut-off was not kept constant, but varied from one-eighth to half stroke.

A further series of trials with varying pressures was run at a constant speed of 60 revolutions per minute, the pressures being 90, 60, and 30 lbs. per square inch above atmosphere, and the cut-off varying from 5 per cent. to full stroke. The lowest consumption obtained was 27 lbs. of water per brake horse-power hour, and the highest 39 lbs. per brake horse-power hour.

Neither of the above series strictly furnishes examples of progressive speed and pressure trials with only one element varying at a time; and in neither case could condensing trials be made.

Willans.—The trials of the late Mr. Willans cover very fully the field of the Committee's proposed investigations. In 1887–8 non-condensing trials* were made with both simple and compound engines of the Willans type indicating up to 45 horse-power. Trials were made at 400, 200, and 100 to 120 revolutions per minute with the simple engine, and at pressures of 50, 70, 90, and 110 lbs. per square inch above atmosphere. With the compound engine trials were likewise run at each of the above speeds, and at pressures of 90, 110, and 130 lbs. per square inch above atmosphere. The ratio of expansion however was altered for different pressures, and varied from 2·174 to 4·8 expansions.

In 1891–2 a much more extended series of condensing trials† at different speeds was made with a compound engine. The speeds chosen were 400, 300, 200, and 100 revolutions per minute, with pressures of 180 to 35 lbs. per square inch above atmosphere. Sets of experiments with varying pressures at each speed were carried out with each of the following expansions—5, 10, 12·36, and 15·55. But the only series which was nearly complete, that is, where trials

* Proceedings, The Institution of Civil Engineers, 1887–8, vol. xciii, page 128.

† Proceedings, The Institution of Civil Engineers, 1892–3, vol. cxiv, page 2.

at nearly all pressures were made for each speed, was with 5 expansions, the pressures being 130, 100, 80, 60, and 35 lbs. per square inch above atmosphere. The observations made were very complete, but no measurement of brake horse-power was attempted.

The proposed trials of the Committee therefore will cover much the same ground as those of Mr. Willans, but at lower speeds. They will however be made with a different type of engine; the cut-off should be kept constant throughout the whole of the experiments, condensing and non-condensing; and all results should be measured as a function of effective power,* so as to eliminate the serious uncertainty which attaches to indicator readings.

APPENDIX II.

MAIN DIMENSIONS OF HIGH-PRESSURE VALVES, &c.

(See Fig. 46, page 259.)

2 Expansion Plates, each $2\frac{5}{8}$ inches wide.

Throw of Main Valve Eccentric, $2\frac{1}{4}$ inches.

Throw of Expansion Valve Eccentric, $2\frac{3}{4}$ inches.

Angle of Advance of Main Valve Eccentric, 30° .

Angle of Advance of Expansion Valve Eccentric, 90° .

Length of Connecting-rod of Engine, 3 feet $10\frac{1}{2}$ inches.

Stroke of Engine, 14 inches.

* In actual working it was found that at the lowest speeds and pressures the effective horse-power was so small that differences in lubrication of the brake blocks, etc., made a relatively large difference in the power. Moreover on some of these trials the large fly-wheel which was disconnected was found slightly to have touched the end of the shaft at one point, so that the brake horse-power was affected. The brake horse-power on these trials therefore has been omitted. After careful consideration and examination of the preliminary trials it was evident that at the lowest powers the specially calibrated indicators were giving records which were more reliable than those of the brake and spring balances, and it was therefore decided to express results in terms of indicated instead of brake horse-power.

FIG. 46.

*Diagram giving dimensions of Ports.**Expansion Valve Face
of Main Valve.**Face of Main Valve which works
on Engine Valve Face.* *Discussion on 17th March 1905.*

The PRESIDENT said the members had met to hear the First Report to the Steam-Engine Research Committee by Professor Capper. This Committee was appointed nearly nine years ago, and then consisted of the following gentlemen:—Professors Kennedy and Capper, Messrs. Bryan Donkin, Longridge, Mair-Rumley and Maw. The present members of the Committee were Mr. Maw,

(The President.)

Professor Capper, Mr. Davey, Professor Kennedy, Mr. Michael Longridge, Mr. Mair-Rumley and Captain Sankey. He anticipated that the Report would be received with great attention, and would give rise to a good discussion. He thought that, from the applause that had greeted the reading of the Paper, the members would agree with him that a most hearty vote of thanks was due to Professor Capper for his able and most interesting Report.

The resolution of thanks was carried by acclamation.

Captain H. RIAL SANKEY, in opening the discussion, remarked that the author had said that Mr. Macfarlane Gray had called Fig. 48 a temperature-entropy chart; this was an error, he had named it the $\theta \phi$ chart. Professor Capper had suggested it should be called a heat chart, but in reality it was more than this so soon as the constant-volume lines were added, because the area between these lines gave the work done by or on the substance during transformation from one state to another, while the area between the adiabatics gave the heat change during the transformation. The author had explained (page 243) how the re-evaporation during the expansion could be exhibited on the chart, and it would be interesting to see how this actual re-evaporation compared with the theoretical re-evaporation. Willans in his Paper on Steam-Engine Trials* had shown how the theoretical re-evaporation line could be found. In Fig. 47, if A B were the initial temperature of the steam, and A₁ B₁ were the exhaust temperature, and the point C represented the dryness or quality of the steam at cut-off, the line C R would be the re-evaporation line required; it was a curve similar to the water line of the chart, inverted. It would be seen that the point R was approximately at the foot of the adiabatic drawn through B, and for most practical purposes a straight line drawn from C to the foot of the adiabatic was sufficiently accurate. This construction had been applied to Fig. 48, the point B being taken as the top left-hand corner of the leakage diagram, and the adiabatic had been drawn down to the exhaust temperature corresponding to the line A₁ B₁ in

* Proceedings, The Institution of Civil Engineers, 1892-3, vol. cxiv, page 2.

Fig. 47. The approximate re-evaporation lines were shown by the dotted lines in Fig. 48; and it would be seen that in the first case (trial DD₂) the actual re-evaporation line lay somewhat to the right of the theoretical at its lower extremity; in the second case (trial CC₂) the two lines were fairly close together; in the third case (trial BB₂) the actual re-evaporation line fell short of the theoretical at the beginning of the expansion, but it came up to it again at the

FIG. 47.

FIG. 48.

Temperature Entropy Diagrams.
Trials AA₂, BB₂, CC₂, DD₂ Unjacketed.
(See Fig. 24, page 201, with dotted lines added.)

moment of release. In the last case (trial AA₂) the actual re-evaporation was considerably less than the theoretical during the whole of the expansion. Strictly speaking, the actual re-evaporation should always be less than the theoretical, and in general it was so. In the present case, however, owing to the engine being worked with the same number of expansions for all the trials, the release pressure was high in the first two trials, causing a considerable drying action on the walls and assisting the re-evaporation. In trial AA₂,

(Captain H. Riall Sankey.)

however, this drying action did not occur; there was obviously a considerable amount of water in the cylinder, and the re-evaporation was sluggish. As minor reasons, there was the leakage into the cylinder during expansion to be taken into account, this leakage would be greater at the higher pressures, as there would then be a greater drop of pressure in the cylinder during expansion. Lastly, the leakage correction was stated by Professor Capper to be somewhat too great, which would cause the point R to be situated too much to the left, and the error in the approximate method employed for drawing the theoretical re-evaporation line was in the same direction. It would be found that similar results to the above would be obtained by drawing the evaporation lines on Figs. 22, 26, 28 and 30 (pages 199 to 207). The construction shown in Fig. 48 (page 261) could also be applied to the jacketed trials, and it would be seen that in every case the re-evaporation extended far beyond the theoretical, more particularly so in the trials at low pressures, trial A₂ for instance. The reason was obvious: there was less condensation, and the heat from the jacket drew the expansion line more to the right. In the case of trial A₂ there was another reason—the steam had been considerably throttled, therefore eddies would be produced, and the energy in these eddies on reappearing as heat would have the same effect on the expansion line as if re-evaporation had taken place. The same remark applied to trial B₂ in a less degree.

Returning to Fig. 47 (page 261), if an adiabatic were drawn from C to cut the constant-volume line a triangle would be cut off (shaded vertically) which would represent the heat units gained by the re-evaporation. If the condensation had been less, as for example as represented by the point C', the theoretical re-evaporation would have been less as shown by the line C'R, but the release would have been at a lower pressure on the assumption that the ratio of expansion was the same. Since the volume at C' was greater than at C the point of release could be found by the intersection of the line C'R of the lower volume line shown in Fig. 47 at the point D', and the area shaded horizontally represented the gain in heat units due to having less condensation. He thought that the above showed very clearly

that although a little good was obtained by re-evaporation, the way to get an economical engine was to reduce the condensation. (*See page 276.*)

Mr. MICHAEL LONGRIDGE, Member of Council, was sure all the members would agree with him that the Report was a very difficult one to follow, and required more study than he and perhaps many other members had been able to give to it. He therefore proposed to limit his remarks to one point, namely the first of the author's conclusions (page 246) that leakage through the slide-valve, to the importance of which Messrs. Callendar and Nicolson had drawn attention, had been quantitatively determined under defined conditions, and had been shown to be nearly independent of speed of sliding surface and proportional to difference of pressure between the two sides of the valve. The leakage question had taken up a very large part of the Report, and he thought rightly so, because upon the acceptance or denial of the figures given depended the whole results of the Report. He did not wish to question the accuracy of Professor Capper's observations, but he would like to point out that the statement that leakage was proportional to difference in pressure was hardly confirmed by the measurements recorded in the Report. These showed that the observed leakages were nearly proportional to the square root of the difference of pressure. At a meeting some time ago he had occasion to speak of the theory of leakage, both of saturated and superheated steam, and he suggested that the velocity of flow through any leakage opening would, like the velocity of flow in any other case, be proportional to the square root of the head. He thought the curves on Fig. 32 (page 212) confirmed that view. These curves were practically parabolas, the leakage in the first case, namely, when the ends of the cylinders only were jacketed, being equal to 4.1 times the square root of the difference of pressure; and in the second, when the cylinders were completely jacketed, equal to 3.46 times the square root of the difference of pressure. He thought the curves were more reliable guides than the figures in the Report, because they eliminated the irregularities which the measured leakages showed. The observed leakages had been used to calculate Callendar

(Mr. Michael Longridge.)

and Nicolson's factor C which was equal to $\frac{\text{overlap}}{\text{perimeter}} \times \frac{\text{leakage per hour}}{\text{pressure difference}}$ and the results of the calculations were given in Table 5 (page 221). They varied by about 100 per cent., namely from 0.032 to 0.014. If in the above formula they substituted square root of pressure difference for pressure differences, they would find that the resulting values for C became practically constant, though of course differing from Callendar's value 0.02. That, however, was merely a question of formulæ, and not very important. The important point was whether such leakages as 20 per cent. actually did exist under ordinary conditions; and he must say that notwithstanding the evidence brought forward in the Report, and in Callendar and Nicolson's Paper, he could not believe it. He thought that in the present case the leakages were to a certain extent abnormal, because there was no current of steam passing through the steam-chest in which the valve worked. In that case a good deal of water would collect in the chest, would lie about the valves, and be gradually sucked in by the small leakage between the valve and its face; and whether the velocities varied as the square roots of the pressures or as the pressures, the quantity going through the leakage opening would be very much larger when the material going through was water than when it was steam, because the weight of a given volume of water was many times that of an equal column of steam.

In the Annual Report of the Council just presented it was stated that the question as to whether the research should be continued and, if so, in what direction, would be considered after the discussion of this Report. He sincerely hoped the work would be continued; and if he might do so without appearing to dictate to the Research Committee, he would use this occasion to suggest what the next step should be, so that members, having the needs of the Committee put before them, might be induced to assist in providing the necessary apparatus. In his opinion this leakage question barred the way to further progress and required further investigation; and the best way of investigating it would be, he thought, to have a number of valve-chests of various descriptions made, each chest to have a port leading into a chamber in which leakage might be collected, a valve—Corliss, piston, slide, or double-beat, as the case might be—with

sufficient lap to work without uncovering the port, and a nozzle at each end, with a flange to bolt to a steam pipe. Such experimental valve-chests, placed in circuit in a steam main supplying an ordinary engine, and experimented on with a current of steam always flowing through, would not only give evidence as to the extent of leakage but would show how leakage was affected by workmanship, and which kind of valve leaked least.

Mr. R. H. WILLIS thought that for purposes of reference it was handy to call the diagram to which Captain Sankey had referred a theta-phi diagram rather than a temperature-entropy diagram, and it was shorter to write. The remark applied perhaps almost more to gas-engine work than to the subject under discussion. It was just as handy to refer to for heat purposes as the ordinary indicator card was for power, but it was rather more difficult to plot. There was no question at all as to its value, and it seemed to be coming more and more into use. He thought the Report was quite an epoch-making one, and, as the author's results so far bore out Professor Callendar's, there was no doubt that the leakage question was the one which had to be considered. The most interesting point would be to see how it referred to piston-valves, because modern piston-valves had such very large surfaces relative to the surface in the cylinders.

Mr. EDWARD C. DE SEGUNDO said that there were many points in the Report which were highly suggestive, and which deserved very full consideration, and he cordially agreed with one of the speakers that it was impossible to discuss usefully such a Paper without having had more time to digest the contents thereof than had been afforded to members. Nevertheless, if he might be allowed to refer to one point in it, he would like to say that he was glad to see the author had tabulated the efficiency of the engine on the basis of the Carnot cycle as well as on that of the Rankine cycle. It savoured rather of heterodoxy to call into question the decision of the Committee which was appointed by the Institution of Civil Engineers to report upon the Thermal Efficiency of Steam-Engines, at the same time he could not help thinking that for the purposes of busy men, the Carnot

(Mr. Edward C. de Segundo.)

cycle was a very useful basis upon which to calculate the efficiency of an engine. He admitted that the Rankine formula more accurately represented the conditions under which steam was used in an engine, but although with high temperatures the discrepancy between the efficiency calculated on the Rankine basis and on the Carnot basis was very marked indeed—in the case of the gas-engine particularly so—still, after all, the Carnot cycle represented perfection of action between given temperature limits and was alluringly simple, whereas the Rankine formula strove to approximate to actual conditions but was decidedly complicated in comparison.

With regard to the theta-phi diagram, he would like to ask Captain Sankey if he (the speaker) understood correctly that in such a diagram the action of the boiler was taken into account as well as the action of the engine. It appeared to him that in the diagram they were dealing with the change of state of one pound of the working fluid through the whole cycle from the feed-water through the boiler, through the engine, and back to the feed-water temperature. Many people had a sort of idea that the theta-phi diagram was a sort of modification of the ordinary indicator diagram and that it referred only to the action of the steam in the engine, but that was not the case.

Captain SANKEY said it was not quite the case.

Mr. SEGUNDO said that that was an interesting point. He wished to be allowed to add his tribute of praise to that which had already been given to the author for his very interesting and important Paper; and to record his opinion, as a man who had some experience of experimental work, that it was quite impossible from a perusal of the Paper to form any adequate idea of the immense amount of time and trouble involved in obtaining and collating the results of the experiments, and for these reasons he thought Professor Capper and his assistants deserved the very hearty thanks of the Members.

Mr. DRUITT HALPIN exhibited a series of photographs and blue prints of an experimental engine at Montreal which had four cylinders. Several details were shown on Plate 6. One of the peculiarities of the engine was that the steam could work through the cylinders in any sequence, and that the cylinders could be worked jacketed or unjacketed, or that any ratio of expansion could be obtained. The angle between the cranks of the two engines could also be altered. Some of the principal parts of the engine were shown on the blue prints on the wall, but there was one peculiarity to which he wished to refer. An arrangement was shown on the first sheet by which the volume of the clearance was altered. The clearance could be altered from 1.2, the normal amount, up to 7.25 per cent., by variation of the position of the piston in the false clearance cylinder.*

The PRESIDENT thought the Report was so important that it would be better to postpone the discussion to an extra Meeting on 31st inst., which would give an opportunity to the Members, he was going to say to understand, but at any rate, to digest this very interesting Paper.

Discussion on Friday, 31st March 1905.

Mr. WILLIAM H. MAW, Past-President, in resuming the discussion, said he was very glad to have the opportunity of saying a few words. In the first place, he emphasised the remarks of the President on the last occasion, as to the indebtedness of the Institution to the author for all he had done in connection with the Report. Everyone knew that when a man undertook the office of a Reporter to a Research Committee, he let himself in for a very large amount of work. But in the present case circumstances had arisen which rendered the labour quite exceptional. In the first place, as stated in the Report itself, the original programme laid down under the Chairmanship of Mr. Donkin had to be modified, in consequence of the Committee

* See Proceedings 1905, Part 1, page 82.

(Mr. William H. Maw.)

not obtaining the experimental engine which it was originally expected would be available. Professor Capper then came to the rescue by placing the engine at King's College at the disposal of the Committee, and it was necessarily felt that the rearrangement of the experiments must be left largely in his hands. Then when the researches were commenced, the author found great difficulty in consequence of the leakage through the valves, and that led him to carry out the very extensive series of experiments on valve leakage which were embodied in the Report. Those experiments involved an enormous amount of personal work to the author. Then the Committee, as the Members knew, lost the services by death of their friend Mr. Bryan Donkin, which again deranged the programme to a very large extent, and eventually it came to pass that practically the whole work of carrying out the experiments rested with the author, and the Report was essentially a one-man Report, that one man being Professor Capper. The Institution thus owed, he thought, an exceptional debt of gratitude to the author for the work he had so ably carried out. Last year, when the Committee resumed its meetings, and he (Mr. Maw) was elected Chairman, it was found that Professor Capper had accumulated an enormous amount of data, and it was the duty of the Committee not to arrange the experiments, but, in consultation with the author, to determine how much of that experimental data should be published. The Report, as the Members had seen, was a very extensive one, and dealt with an enormous number of experiments, but it only represented a very small amount of the information which the author had collected. He thought that no one, except perhaps the Members of the Committee who had had the opportunity of going through the whole of the original documents, had any idea of the amount of work that had been expended in obtaining the results laid before the Members in the Report.

Reference was made at the previous meeting to the fact that in the last Annual Report of the Council, it was stated that the future of the Steam-Engine Research Committee depended largely upon the results of the discussion on the Report now before the meeting. It thus followed that the Report should be discussed from two points of

view; first, from what he might call the author's point of view, namely, a discussion of the facts set forth in the Report, and criticisms on them—and he was sure that nothing would delight the author of the Paper more than that they should be very fully and thoroughly discussed—and secondly, a discussion from what he might call the Committee's point of view, that is, an expression of opinion as to the way in which the work of the Research Committee could be best carried on in the future. The Report, extensive as it was, practically only touched the fringe of the great subject of Steam-Engine Research. Apart from the experiments on valve-leakage loss, which formed such an important portion of it, it consisted really of experiments on the high-pressure half of the engine at King's College, and of that half used only as a non-condensing engine. It thus by no means exhausted the amount of information which might be obtained from that particular engine, while it left quite untouched the further information that could be obtained from similar experiments carried out on engines of larger size.

In speaking of the leakage experiments at the last meeting, he thought Mr. Longridge rather threw doubt on the absolute accuracy of a leakage of 20 per cent. being possible in the slide-valves. He thought for the moment Mr. Longridge had overlooked the fact that that loss of 20 per cent. took place when the engine was being worked at an exceptionally low power, only $\frac{7}{8}$ H.P., and the consequence was that the leakage, which was almost a constant leakage per hour under a certain pressure, then formed a very large proportion of the whole steam used in the engine. If reference was made to the results recorded in Table 13 (pages 252–253), it would be found that for the A series of experiments the difference between the steam passed through the cylinders per hour, and the steam per hour after deducting the leakage, was almost exactly 10 lbs. in all cases. That 10 lbs. formed a very large percentage in the case of the low-power trials, and was roughly about 20 per cent. In the A₅ trials, that leakage (which was just about the same in that case per hour) was only about $3\frac{2}{3}$ per cent. He thought at the moment Mr. Longridge did not take that particular fact into consideration. Mr. Longridge had rather doubted whether experiments on valve

(Mr. William H. Maw.)

leakage conducted in what he might call a dead end, without any flow through the chest containing the valve, were quite to be depended on. It was a matter which he himself had at first had some doubt about, but looking at the experiments he felt bound to say that those doubts had disappeared. He thought the experiments, with some trifling exceptions, were very consistent, and it did not seem as if, with proper drainage of the steam-chest, the fact of there not being a flow through was any disadvantage.

There was one thing in the leakage experiments that had struck him as rather remarkable, and he would like the author to kindly say a word or two more about it in reply, namely, the very marked difference between the leakage with the valve lubricated and the valve not lubricated. Where there was a flow of steam through a steam-chest, it was not difficult to get a fairly good distribution of the lubricant over the valve, but where the valve was working in the steam-chest, without any flow, it seemed to him difficult to get the lubricant distributed over the valve-face. The fact that in the particular series of experiments under discussion there was such a very marked difference between the lubricated valve and the non-lubricated valve, showed that somehow or other the lubrication had been effectively realised, and he would like to ask the author kindly to say how the lubrication was arranged.

There was another question he would like to direct the author's attention to, namely, the question of the steam through the jackets. He thought perhaps it would be a desirable addition to the Table 13 (pages 252-253) if the pressure in the jackets during each series of experiments was stated. He thought it had been given in some of the original papers, but it was not stated in the Report as published, and he thought the information had better be supplied. Apparently there was rather an exceptional amount of steam used in the jackets in the case of Experiment A₁, namely, 44 lbs. per hour, while in the next experiment, A₂, it was only 14½ lbs.; 44 lbs. seemed rather an exceptional figure, but there might be some reason for it. He wished especially to draw the attention of the Members to Fig. 45 (page 249), which, he thought, was one of the most valuable diagrams they had ever had put before them. It summarised the author's deductions in the most admirable way.

Passing to what he might call the Committee's view of the Report, he would like to say—it was merely an individual opinion, not the opinion of the Committee, because he had not had an opportunity of consulting them—that the Report now under consideration pointed to the desirability of amalgamating the Steam-Engine Research Committee with the Steam-Jacket Committee. As the members knew, a Committee of the Institution had been in existence for some years dealing with experiments on steam-jackets, and he believed that a further Report from the Committee would be forthcoming in the course of a few months. After that Report was issued, he thought it would be well worth the consideration of the Institution whether the two Committees should not be amalgamated. The work of the two Committees overlapped in a number of points, and he thought it would be difficult for them to carry on their work independently without a considerable waste of time and energy. That was a point upon which Mr. Davey, who was the Chairman of the Steam-Jacket Committee, might like to say something later.

Then the question arose as to the future of the Steam-Engine Research Committee. As he had already said, the Committee had only just touched on one very small portion of the researches open to them, and it was quite evident that the resources of such a Committee, as it at present existed, were not equal to carrying out a thorough series of researches in anything like a moderate time. He therefore thought it was open to question whether they should not endeavour to enlist the assistance of some workers who had other experimental engines available, and see whether they could not arrange for some simultaneous series of organized experiments to be carried out, which should be submitted to a Central Committee for consideration. There were a number of experimental engines in the country, and if they could get the gentlemen in charge of those engines to carry out, independently, different branches of research, the whole operations of the Committee might be very greatly shortened.

Mr. C. E. STROMEYER said he would like to touch on the point last raised by Mr. Maw, and to ask whether it would not be possible

(Mr. C. E. Stromeyer.)

to continue the trials on a large scale. He felt sure that some manufacturers would be willing to lend their engines for such purposes. Some experiments on very large marine-engines had already been carried out by Sir John Durston and Mr. Oram. They were tried with and without jackets, from 8,000 up to 19,000 I.H.P., and the results were that for the low power the non-jacketed engine had an advantage of 1 per cent. in the steam consumption, and for the high power an advantage of 5·6 per cent. With regard to the present Report, he wished to say that he had come to the meeting to inquire why it was that piston-leakage experiments were carried out on the lines indicated on page 222. He had gone very carefully into the diagrams, and plotted a continuous diagram of the weight of steam in the blocked end of the cylinder, which only received it by leakage, and the results were very unsatisfactory. He found that when the pressure on one side of the piston was about 105 to 106 lbs., then the weight of steam in the blocked end of the cylinder was gradually diminished, which would make it appear as if there were a flow of the steam from the low-pressure towards the high-pressure. That diminution of weight continued to the end of the stroke, and then, when the leakage should be from the high-pressure in the blocked end into the exhaust, and when a decrease of weight should have occurred, there was a steady increase. He thought that there must have been leakage from the blocked end to the atmosphere. Some steam seemed to have got away and got back again. It therefore did not seem to him that this particular method was an accurate one for determining what the amount of piston-leakage was. The calculated steam weights and losses were given in Table 15 (page 274).

He had also gone very carefully into the figures which the author had given for the slide-valve leakage, and these he had plotted on a logarithmic diagram, Fig. 49 (page 273). The results were very consistent, except that the experiments at a pressure at about 29 lbs. seemed to be wrong. The figure showed that the leakage was not directly proportional to the pressure, but to a small power of the pressures; thus, in the case of the unlubricated jacketed cylinder, he found the power to be 0·553. If the leakage had been steam or water, then the index should have been $\frac{1}{2}$, which meant that the

flow should have been proportional to the square root of the difference of pressure of the steam on the one side and on the other. In the case just mentioned, it was slightly higher than that, it was 0.553, and the coefficients for that particular engine for the three speeds were 2.8, 2.7, and 2.6 lbs. per hour.

In the case of the jacketed and lubricated cylinder, he found that the index was 0.72, and the constants were 1.35, 1.25, and 1.15

FIG. 49.

Logarithmic representation of Results shown in Tables 1 and 2 (page 210).

The abscissæ and ordinates have been marked from a slide-rule.

lbs. per hour. The diagram also showed that the two leakages overlapped, and that at the low speeds the lubricated slide-valve certainly had the advantage; it did not allow so much steam to pass as the unlubricated slide-valve, but for the higher pressure the lines crossed, showing that reverse results were obtained. He noticed that the quantities estimated from the leakage experiments

(Mr. C. E. Stromeyer.)

TABLE 15.

Position of Crank.	Steam Pressures in open Cylinder End.	Blocked Cylinder End.			
		Steam Pressures.	Differences.	Steam Weights.	Flow per $\frac{1}{2}$ Revolution.
	Lbs. per sq. inch.	Lbs. per sq. inch.	Lbs. per sq. inch.	Lbs.	Lbs.
0+24	105	12	+93	0.0095	+0.0006
1	106	13.0	+93	0.0101	-0.0001
2	106	13.5	+92.5	0.0100	-0.0001
3	106	14.2	+91.8	0.0099	-0.0004
4	105	15	+90	0.0095	-0.0005
5	103	16.3	+86.7	0.0090	-0.0008
6	86	17.5	+68.5	0.0082	-0.0010
7	70	19	+51	0.0072	-0.0013
8	60	20.8	+39.2	0.0059	-0.0014
9	53	22	+31	0.0045	-0.0013
10	48	23	+25	0.0032	-0.0010
11	45	23	+22	0.0022	-0.0003
12	?	23	say -5	0.0019	+0.0002
13	18	22	-4.0	0.0022	+0.0006
14	18	19.6	-1.6	0.0028	+0.0009
15	17	17.5	-0.5	0.0036	+0.0010
16	17	16.3	-0.7	0.0047	+0.0012
17	17	15.5	+1.5	0.0059	+0.0010
18	17	14.6	+2.4	0.0070	+0.0009
19	16	14	+2.0	0.0078	+0.0007
20	16	13.3	+2.7	0.0085	+0.0006
21	16	13	+3.0	0.0091	+0.0002
22	18	12.5	+5.5	0.0093	+0.0001
23	29	12	+17.0	0.0094	+0.0001
24	105	12	+93	0.0095	—

had been deducted from the power trials, and he feared that this had not been done to the best advantage, and different results might be arrived at if his figures were adopted.

He might perhaps refer to another difficulty which he did not notice the author to have dealt with, except to mention it, namely, the lag of the indicator drum and other imperfections of the indicator. A Paper * was read by Mr. A. W. Brightmore before the Institution of Civil Engineers in 1885, and he (Mr. Stromeyer) subsequently read a Paper † at the Institution of Naval Architects in 1894, dealing with the question of the lag of the indicating cylinder, and in his case he came to the conclusion that for an indicator string of about 5 feet, one ought to allow about $\frac{1}{8}$ inch at the end of each diagram, that is, the starting ends of the diagrams should be lengthened $\frac{1}{8}$ inch. According to Mr. Brightmore's Paper, it would be seen that the correction depended very much on the number of revolutions and on the weight of the indicator drum, and various other matters. In his own Paper he had plotted down on a continuous diagram the weight of steam that was in the cylinders, and he noticed that during the expansion period, when of course the weight of steam should be constant, there was always a slight rise. That slight rise was evidently the difference resulting from re-evaporation and leakage of the steam into the cylinder less the leakage into the exhaust port and past the piston; but seeing that there were four factors which affected the curve, it was quite impossible to use it for the determination of any one of them separately.

Captain H. RIAL SANKEY apologised for again taking part in the discussion, but he had been asked to describe some experiments on the leakage of piston-valves and piston-rings which he carried out in 1897 and 1898. The experiments on piston-valves were made with a small Willans engine, a partial section of which was shown in Fig. 50 (page 276). It would be seen that there were no pistons

* Proceedings, The Institution of Civil Engineers, 1885-6, vol. lxxxiii, page 20.

† Transactions, Institution of Naval Architects, 1894, page 407.

FIG. 50.

Arrangement for measuring Leakage past Piston-Valves.

FIG. 51.

Leakage past Piston- and Slide-Valves.

fitted; there was only the hollow trunk, in which the piston-valve worked, and the arrangement was made to reciprocate at any desired speed by means of an electric motor. The object of the experiments was to determine the leakage past the piston-valve and past the steam gland surrounding the outside of the trunk. The trials were made with the engine standing and also running at revolutions varying from 20 to 500. When the engine was standing the leakage was erratic, but as soon as it revolved there was practically no difference in the leakage, whether the speed was 20 revolutions a minute or 500, which, it should be noted, agreed with Professor Capper's results. With the standard Willans valve, consisting of two spring rings separated by a dummy, the dummy having a clearance of about $\frac{2}{1000}$ of an inch, the leakage was found to be very small, as shown by the lower boundary of the shaded area in Fig. 51. The dummy ring was then made smaller so as to get a greater clearance; the split between the rings was made larger so as to get a greater leak in that way, and the splits were placed in prolongation instead of opposite each other. These various changes singly and combined increased the leakage, but not seriously, since in every case the leakage was within the shaded area.

The standard piston-valve was then replaced by a solid plug valve, made as tight as it would run without seizing, namely with a clearance of about $\frac{1}{1000}$ of an inch; and the leakage found was given in Fig. 51. Each of the black spots was the mean of several readings. Then the plug valve was ground $\frac{1}{1000}$ th inch smaller, making a clearance of $\frac{2}{1000}$ ths, then another $\frac{1}{1000}$ th, making a clearance of $\frac{3}{1000}$ ths inch, and so on. It was found that the leakage at all pressures increased almost proportionately with the clearance, when the experiments were with dry steam, as would be seen from Fig. 51. On one occasion a considerable amount of oil was put into the engine when the clearance was $\frac{4}{1000}$ ths inch, and immediately the leakage dropped to what was shown by the dotted line, namely, about the same as with $\frac{2}{1000}$ -inch clearance, which answered a question put by Mr. Maw (page 270). A coil was placed in the steam top through which water could be passed. In that way a portion of the steam was condensed, so that there was a shower of rain coming down on to the

(Captain H. Riall Sankey.)

piston-valve. The leakage was much increased, as shown in Fig. 51, for the case where the clearance was $\frac{5}{1000}$ ths inch. The leakage was also tried with water, by forcing it in with a pump, and in that case the leakage was enormously increased, being from 8 to 16 times as great as with dry steam in corresponding cases. The leakage of the steam-gland was also measured. It was found that the results were not so consistent as with the piston-valve, probably because water would form on the top of the gland-box and be forced through in gulps; but on the average the leakage of the gland was about $2\frac{1}{2}$ times greater than that of the piston-valve in the corresponding cases, which fairly corresponded to the differences in dimensions.

He had calculated the factor C given in the Report, and found that for the standard piston-valve (lower line of shaded area) it was 0.003, which was $\frac{1}{4}$ th less than the factor the author had found with a slide-valve. The author's factor corresponded almost exactly with the lowest leakage line for the plug-piston ($\frac{1}{1000}$ -inch clearance). He had also worked out what the leakage would be expressed in lbs. per I.H.P. per hour with Professor Capper's slide-valve and with the standard piston-valve, calculated on the normal speeds of the respective engines, and the result was given in Fig. 51 (page 276), from which it would be seen that the latter was about $\frac{1}{10}$ th of the former.

He next referred to experiments made to determine the leakage past piston-rings. The apparatus was shown diagrammatically in Fig. 52, and it consisted of a small Willans engine driving a dynamo, in which the usual steam distribution by means of the central valve was replaced by a D slide-valve, as shown. Experiments were made with mean pressures varying from 4 lbs. to 46 lbs. per square inch. Obviously the leakage past the piston-rings would find its way into the space marked B, and carried by the pipe F to the condenser D, so that there was quite an easy means of finding the rate of leak past the piston-rings. The trials were made at various speeds, and it would be observed that so far as leakage was concerned the piston-rings were working under normal conditions. The first trial was made with the standard rings in a cylinder of 10 inches bore. Those rings, when free, expanded one-tenth of an inch; the result was the lower leakage line marked B in Fig. 53 (page 279), in which mean

FIG. 52.—Apparatus used for measuring Leakage past Piston.

FIG. 53.—Leakage past the Piston.

x 2

(Captain H. Riell Sankey.)

results for the trials at 500 revolutions were expressed in lbs. of leakage per I.H.P. per hour, plotted on a mean pressure base. Then the cylinder was bored out approximately $\frac{4}{1000}$ ths of an inch, and the leakage was slightly increased (curve E). Some patent rings, which had been sent for trial, were next tried before the cylinder was bored out again; these rings were the correct size for the bored-out cylinder. The leakage was far greater than with the standard rings (see leakage line D).

The cylinder was next bored out to 10.105 inches. In that condition the standard rings were just a fit in the bore, and were fully expanded; it would be seen the leakage was considerably increased (leakage line F), especially at very light loads, the reason being that the rings were steam-packed, and when the load was very light there was not sufficient steam to pack them. The leakage with the patent rings was given by line G for this larger cylinder, and line C represented the result with a single Ramsbottom ring; the piston body was not sufficiently deep to admit of more than one ring, so that the leakage line was not a fair representation of the leakage when there were two or more Ramsbottom rings.

Mr. VAUGHAN PENDRED thought the Report was so full of matter for discussion that, instead of occupying ten minutes, he could speak about it for two or three hours. The first point to which he wished to call attention had been raised to a certain extent by Mr. Maw, namely, the future of the Steam-Engine Research Committee. He did not want to say anything which was severe, or which could be regarded in the slightest degree as personal, but a question he wished to ask was: What was the value of the Report under discussion? He saw around him a very large number of young men, most of whom hoped the time was not very far distant—and he hoped so too—when they would be called upon to design something in the way of an engine: either a mill engine, an electric-light engine, or a marine engine. He could not imagine any of the members sitting in a drawing office, turning to the Report, and seeking successfully in it for information which would be of use to them. He fully endorsed everything that had been said in regard to the author and the work

he had done. He (the speaker) had had in one way or another nearly fifty years' experience of the steam-engine in all possible forms, and engineers had the results of hundreds or even thousands of experiments that had been carried out. The literature of the subject was enormous, but he did not believe anybody had done better experimental work than that which the author of the Report and his assistants had accomplished. But what did it all come to? It was not the fault of the author, or of the Research Committee, or of any engineer, that an inquiry of such a kind must be futile; it was the fault of the steam-engine. Every steam-engine was a law unto itself. If the clearance space and the conditions generally of the engine with which the author had been working were altered, an entirely different set of results would be obtained.

If any of the members would take the trouble to go into the library of the Institution, they would find on its shelves the Transactions of the American Society of Mechanical Engineers. He wished to draw particular attention to a Paper in those Transactions on the "Promise and Potency of High-Pressure Steam,"* illustrated by the triple and quadruple expansion experimental engines at Sibley College, Cornell University, by Professor Thurston. He also wished to call attention to another Paper in the Transactions of the American Society, namely, "The Laws of Cylinder Condensation,"† by Arthur L. Rice. Nothing more elaborate and nothing more careful had ever been carried out than those particular experiments. Thurston's Paper was a splendid treatise on what went on inside an engine. Going back over the history of the steam-engine, it would be found that almost as far back as the memory of man went the merits and demerits of the jacket had been discussed. One man said that jackets ought to be used and another man said they ought not; and in their arguments in connection with them they resorted to nearly everything short of bloodshed. But the result was that, after all the investigations and discussions which had taken place, engineers were in precisely the same position now as they were in those earlier times.

* Transactions, A.S.M.E., 1897, page 160.

† Transactions, A.S.M.E., 1897, page 950.

(Mr. Vaughan Pendred.)

Going back 32 or 33 years, they had the trials made of portable engines by the Royal Agricultural Society when the racer was built. The racer was a very remarkable engine in that it was built for a particular purpose, and was made the subject of the most elaborate investigations. He only wished that his old friend Sir Frederick Bramwell could have been present and told the members what he knew about it. At the time the speaker happened to be brought intimately in connection with the subject, because he had some very close personal friends who built racers. A racer had to be built 6 or 12 months before the time of trial; the engine had to be placed in a special shed, and worked by a special staff, and every possible effort was made to get the highest possible result out of it. What was the result sought for? The competitor declared when he came to the trial at the Show ground what power his engine was going to work at, the number of revolutions and the brake load; 14 lbs. of coal were allowed per declared horse-power, and the engine that was able to run for the longest time won the prize. Nearly all the engines were fitted with jackets, so that steam could be turned into them if necessary. If the members referred to the Journals of the Royal Agricultural Society, it would be found that not one single engine which admitted steam into the jacket during a competition had taken a prize. Taking the case of the Reading Co.'s engine at Cardiff and also the case of Paxman's compound engine at Newcastle-on-Tyne, they were tried first for long periods with steam in the jacket, and out of the jacket; and the winning engines in the show-yard did not take steam into the jacket. That was equally true with 50 lbs. and with 80 lbs. steam.

He intended to have said something about slide-valve leakage, but he had time only to ask one question about it, which perhaps the author could answer. A good deal was heard about the water. He wanted to know what that water was supposed to represent. If it came from the boiler as steam, and condensed in going under the valve, that steam had to leave its heat behind it in the engine in some way. They certainly could not for a moment regard the loss of water under those circumstances from the same point of view that the loss of steam would be regarded. He simply asked that question

for information. The question of evaporation and re-evaporation had been raised, and unless the Paper was read very carefully, it was slightly confusing. The point was apparently missed that whatever went into the cylinder of a steam-engine as steam must leave it as steam, and whatever went into it as water must leave it as water. There was no getting away from that fact under any possible circumstances. Supposing that the re-evaporation did not equal the condensation, then the cylinder would keep on getting hotter and hotter and storing up heat until condensation ceased. Thurston remarked in the course of one of his Papers, "The exhaust steam contains precisely the amount held in suspension on entering the valve chest, plus the condensation due to waste by external conduction and radiation and its conversion into work." Supposing for a moment that the whole heat coming into a cylinder was 100; supposing that the radiation was 10, and that that due to the performance of work was 10, the re-evaporation would be 80; the rest would go out of the cylinder as water. There was no getting away from it; it was impossible to elude the question in any shape or form; it was a physical fact that must hold good. As for the distribution of the periods of re-evaporation, whether during expansion or exhaust, that was quite another affair, and an extremely variable quantity. In that connection he might cite many experiments that had been carried out, as for instance those on the Pawtucket pumping-engine,* which were to be found in the Transactions of the American Society of Mechanical Engineers. It would be found that the quantity re-evaporated was exceedingly variable, and sometimes ridiculously small.

Not a word had been said in the Report with regard to the effect of radiation and conduction from the cylinder itself. One of the standing arguments that had been raised against the jacket was that the surface was so far increased that radiation and conduction represented a very considerable loss of heat. Taking an engine when it was working to $\frac{2}{10}$ ths of 1 horse-power or something like that, the surfaces, the heat and the temperature were the same as

* Transactions, A.S.M.E., 1890, page 328.

(Mr. Vaughan Pendred.)

when very much more power was developed; what was the loss by radiation? In an ordinary engine an engineer would say he could neglect that; but in carrying out scientific experiments it was just as well to have the results accurate.

There was one remark he wished to make in conclusion. The Paper contained a series of experiments which gave particular results, and the author had reasoned from them that it was possible to get precisely the same results with another engine. If this was not true, then the Report had no general application. Before sitting down he would cite two cases to show what the probabilities were; and Captain Sankey could correct him on one if he was in error; and Mr. Davey could correct him on the other. If his memory was correct, Mr. Willans in a Paper read before the Institution of Civil Engineers cited a memorable and noteworthy case in which a quantity of water equal to about one tablespoonful lodged in the bottom of a cylinder and made a most material difference in the whole performance of the engine. The other instance was that of a very large pumping-engine at Staveley put up by Mr. Davey, which had a 10-foot stroke, and a 72-inch low-pressure cylinder, pumping at one lift 812 feet. That engine did not make the number of strokes in a minute it was required to make under the contract, because the cylinders were not fitted with drain cocks. Drain cocks were fitted to the engine—there was no other difference that he was aware of—and the engine then ran something like 2 strokes a minute over its contract number. It would thus be seen that very small details made enormous differences in the performance of an engine. The fatal difficulty with all research questions from the beginning to the end was that what was true of one engine was not necessarily true of any other engine, and was probably not true of it at all.

Professor F. W. BURSTALL said it was a particular source of gratification to him to speak on the Report, for the simple reason that the research was commencing when he was an assistant of the author's, and he knew to some extent what was in Professor Capper's mind at the time. He considered that a Paper of such a kind was

quite inadequate for discussion anywhere, there was such an immenso amount of information that it would take one quite a considerable time to fully digest the whole of it. He proposed to deal with the subject which seemed to have attracted a good deal of attention, namely, that of leakage through the valves. He need hardly say that it was a most important question. The last speaker had asked how far such things influenced the question of the design of the engine. He should say it meant everything. If Professor Capper's Report, and the Paper of Professors Callendar and Nicolson were to be trusted, all valves leaked, whether they were slide-valves or

FIG. 54.

Slide-Valve Leakage varying with Temperature.

(Compare Fig. 32, page 212.)

Corliss valves, and although they did not say so, double-beat valves, such as were in use in the Sulzer and Cornish engines. The members had no information about that. If there was a definite law of valve leak, he thought it was of the highest importance that the engine designer should know what that law was, because it would help him very considerably in his design. He was afraid he could not see that the experiments of Callendar and Nicolson proved that there was a definite law of valve leak, that was to say, it was either proportional to the difference of pressure, or proportional to the overlap. He had taken the figures out in every form: he had plotted the leakage against the pressure as it was given in the Report, and then as against the temperature, Fig. 54, and the curious

(Professor F. W. Burstall.)

thing was that when the valve leak was plotted against the difference of temperature between the valve-chest and the exhaust, the points lay upon a straight line, that was, taking the whole number of experiments given in the Report. He would be sorry to have it thought that that was in any way universal. The number of experiments in the Report were not sufficient to warrant one in stating a law. Before stating a law of valve leak, it would require an immense number of experiments before one could justify the fact that the valve leak was proportional to the difference of temperature between the valve-chest and the exhaust. But it was worth while to consider what were the consequences if it was proportional. He took it that, if the leak was proportional to the difference of temperature, it rather pointed to the leak being due to a distorted valve. Those who had experimented on slide-valves, working first with saturated and then with superheated steam, would have noticed that when they took the flat valves out and tested them for truth after experiments on the superheated steam, that they had invariably warped. That was a practical fact. If that warping took place with superheat, and was apparent when the valve was cold, why should they deny the fact that it was possible the valve warped and returned to its shape when it became cold, even when dealing with saturated steam? He thought Captain Sankey's experiments definitely went to show that a warping valve was the cause of the leak, because if the members would notice, his experiments were made on Willans engines, with spring rings. If there was one form that was less susceptible to warping than anything else, it was a purely circular form. The slide-valve was a more or less complex casting, in which there were initial stresses which might or might not be released by the action of the steam, whereas with the spring ring none of those conditions existed. Captain Sankey's diagrams went to show that the spring ring gave a very much smaller coefficient, 0.003, as against 0.21 in the case of the flat valve. He would be sorry to have those things carried too far, because he was perfectly well aware that Professor Capper would say at once, "How do you account for it that engines with flat valves do as well, or even better, than those with piston-valves?" Which of course was a perfectly

straightforward answer. In fact the actual tests showed most distinctly that engines with almost any kind of valve would do as well as those with any other, proving that it was not altogether a question of valve leak that affected the economy of the engine.

He should like to say, in agreement with Mr. Maw, that he did not think the vexed question of valve leakage, the effect of speed, and the many problems which depended on so many simultaneous factors, would ever be determined by one single individual. It was only by co-operation of those people who had engines at their disposal, and who could make really accurate tests, that the work would be carried out. If those people placed their material at the disposal of the Steam-Engine Research Committee, he thought something like laws might be arrived at which were independent of a particular engine; and, speaking personally, he should be only too pleased to put any of his engines at the disposal of the Committee.

Apart from that, the particular point to which he wished to direct attention was the question of the dryness-fractions mentioned on page 228. It would be noticed that the dryness-fractions were given in the last column on that page, as if the expansion from cut-off were adiabatic. Perhaps the author would be good enough to tell him before he made a mistake, whether the steam at cut-off was taken free of leakage, or whether the leakage had been deducted.

Professor CAPPER stated that it was taken free of leakage.

Professor BURSTALL said he understood it was the gross feed.

Professor CAPPER replied in the affirmative.

Professor BURSTALL, continuing, said that if the members considered the gross feed there, they would see that even in the unjacketed trials a very considerable amount of re-evaporation was obtained. Taking the previous column, that is, the actual dryness-fraction obtained by deducting leakage, in some cases it would be found the steam was superheated at the jacketed trials. That struck him as a curious result. He did not say it was not correct, because

(Professor F. W. Burstall.)

he had made sufficient experiments with steam-engines never to be surprised at anything he got; but he did consider that dryness-fractions varying from 1 down to the last figure, 0.69 in the case of the full load unjacketed trials, showed extremely dry steam; and it would be interesting to know whether any experiments had been made to determine what the wetness of the steam was before it entered the cylinder at all. Of course such experiments were very difficult to make, and the results were perhaps not very reliable; but still, granted that it was a locomotive boiler, which he believed it was, and the length of steam-pipe to the boiler of the engine was something like 40 feet, his experience of experiments on locomotive boilers and long steam-pipes was, that the steam would initially contain something from 5 to 10 per cent. of water, and he did not quite see how it could get so dry as 1, taking it that the leakage was accurate.

Mr. Pendred (page 282) alluded to a point as to how the steam passed away in the form of water with regard to giving up its heat. That was a point that had greatly troubled him. The steam was supposed to condense on one side of the slide-valve, which passed as water to the exhaust slide—he was speaking generally—and there it re-evaporated. Looking at the problem physically, he did not see why the steam should condense, to begin with, and in the next place that heat had to be conducted from the steam side to the exhaust side, in order to provide the heat necessary to re-evaporate that water into steam in the exhaust. Why should the heat travel along the slide-bar at all? There was very little difference of temperature from a conductivity point of view, and physically speaking he could not see why that heat should travel along the bar at all. Therefore, if the steam leaked through, he could not see why it should go through as water and not as steam. Of course the reason for taking it as water and not as steam was, that if the velocities of dry steam passing through were calculated, figures were obtained which were wholly absurd, and therefore they must say that, if it leaked at all, it leaked as water. At the same time he did not feel satisfied that the valve leakage had been made out to be a definite law, on which engineers could rely, with any description of engine. What

engineers really required now was a series of experiments, not necessarily made on an engine cylinder, but made to determine the law of leak in valves of all forms, and valves of different materials. All the present experiments had been made on cast-iron, which was a notoriously warping material. He would suggest that, if possible, experiments should be made on cast-iron valves, on cast-brass valves, and on a forging, and see whether the result was due to distortion, or whether the steam really flowed as a water film. He did not think there was anything more important for the steam-engine makers in this country, than to receive from such a great Institution as the Institution of Mechanical Engineers, some definite information about which there should be no shadow of doubt, as to what law they were to follow in designing valves, and what the valve leak would be.

Mr. A. L. MELLANBY thanked the President for the privilege of allowing him to address the meeting, and also for the opportunity of being allowed to congratulate the author upon the Report he had presented. He thought Professor Capper was to be specially congratulated, because he had drawn attention to a point which must be settled before any important gain could be made in steam-engine economy. Since the Paper of Messrs. Callendar and Nicolson, the subject of valve leakage, which they so strongly brought forward, seemed to have been forgotten, and engine builders had gone on imagining that the whole of the missing quantity in any steam-engine was due to initial condensation. The Report now under discussion was a direct confirmation of Messrs. Callendar and Nicolson's work, and ought to draw the attention of engineers to the fact that they should direct themselves to the design of valves which could be used in everyday work, and which could be depended upon not to allow the steam to leak past them. He had been asked by Dr. Nicolson, who was unable to be present, to read some remarks he would like to offer to the meeting, and with the President's permission he would do so. [Dr. Nicolson's remarks will be found on page 330.]

To himself personally the valve-leakage question was one of much interest, because when he was at McGill University as an

(Mr. A. L. Mellanby.)

1851 Research Scholar he thought he was the first to make experiments which showed that a slide-valve, absolutely steam-tight when stationary, so soon as it started to move had a very considerable leakage. The experiment was made upon the engine which he thought was described by Mr. Druitt Halpin on the last occasion. The experimental engine at the Manchester School of Technology was specially arranged for leakage experiments. Instead of plugging the ports, they had arranged the valve to work on a loose face, and between the loose face and the cylinder casting a brass plate was fitted. Two brass plates were provided, one with holes, corresponding to the steam and exhaust ports for ordinary working, and the other blank, except for the exhaust port, for leakage experiments. This arrangement enabled one to run leakage experiments with very little trouble, and to be perfectly sure about the joints being tight. They also had a small engine, on which they carried out leakage experiments in the manner described by the author. In this engine they had to block up the ports, and experienced the difficulty which Professor Capper had described in making the joints tight.

If he might offer some criticisms on the Report, he thought it was a pity the author did not take the temperatures not only of the clearance surfaces but also of the valve-faces. If that had been done, some evidence would have been forthcoming to show whether the valve-faces were cooler in the valve-leakage experiments, that is, the port plugged experiments, than they were under ordinary working. In the experiments which he had made himself, with small thermometers in the metal of the steam-ports, he had not been able to detect any lowering of the average temperatures; in fact in some cases the temperatures were higher. Not only that, but on taking by means of a delicate thermometer the temperature of the steam as it leaked through the valve, he had found that in almost all cases it was superheated. When the engine was working, the exhaust steam was not superheated, and it seemed to lower the temperatures of the ports.

It was very remarkable to notice the difference that speed had upon these temperatures. If there were a thermometer in the metal of the clearance surfaces, that is, just at the ends of the cylinder, it

would be noticed that that thermometer remained practically stationary for all speeds of the engine, say from 50 to 300 revolutions per minute; but a thermometer in the metal of the ports which was influenced by the rushing past of the steam would drop considerably with high speeds. It therefore seemed to him that the high speeds lowered the temperatures of the ports, and therefore increased the leakage.

The author mentioned (page 238) that the re-evaporation from the jacketed surface was less than from the unjacketed, and explained it by saying that the regenerative effect was largely independent of the temperature. It was extremely difficult to him to see why that should be so. That was to say, it was difficult to see why, if there were any water present in the cylinder, less should be evaporated from the hot walls than from the cold walls. He thought the true explanation was that, in most cases by the time the piston got to release, there had been complete re-evaporation. In the jacketed engine, he would say that the steam in every case was perfectly dry, and probably superheated before release, and that, even in an unjacketed engine, in a great many cases the steam was also perfectly dry at release. That perhaps might be considered rather a strong statement by those who had carried out engine trials, but it seemed to him to be true. The evidence for it was, first, the Callendar-Nicolson experiments which either had to be accepted or refused. Looking at the condensation areas in their Paper, it would be seen that for admission pressures of about 100 lbs. per square inch they were comparatively small, and that soon after cut-off the re-evaporation area was equal to the condensation area, and therefore any steam that was initially condensed would soon be re-evaporated. In order to obtain further evidence upon this point he made a special experiment, by placing in the exhaust-pipe of the engine an ordinary drain-pipe, into which he put a large number of coils of iron wire. There were two thermometers in the pipe, one on the cylinder side of the wire and one on the condenser side. The arrangement was shown in Fig. 55 (page 292). When the engine was running, those two thermometers gave practically the same reading, but by passing an electric current through the wires, the wires were heated, and

(Mr. A. L. Mellanby.)

re-evaporated any water that was present. So long as there was any water at all left in the steam the two thermometers read exactly the same, but immediately all the water was re-evaporated the second thermometer commenced to rise. Knowing the ampères and volts

FIG. 55.—*Experimental addition of Heat to Exhaust-Steam.*

used, he could easily calculate the energy put into the steam, and could therefore calculate the amount of water that had been mixed with the steam. In one particular instance he remembered, at release, according to the ordinary methods it appeared as if there were a missing quantity of about 200 lbs. per hour, and it would

have been generally said that there were 200 lbs. of water per hour present at release. The experiment he was mentioning showed that instead of there being 200 lbs. of water at release there were only 12 lbs., an amount that was practically made up by the radiation between the engine and the pipe with the coil in it. He thought that showed at once that if not perfectly dry at release the steam was very nearly dry. If they accepted this view, then it followed that the working leakage was much greater than the valve-leakage experiments showed, and the difficulty of explaining why there should be less apparent re-evaporation for jacketed than unjacketed trials disappeared. It might then be said that there was less apparent re-evaporation, because there was no water in the cylinder left to re-evaporate. He had taken trials No. C₂ jacketed and CC₂ unjacketed, and from them constructed Table 16, correcting the quantities for 100 revolutions.

TABLE 16.

—	C ₂ Jacketed.	CC ₂ Unjacketed.
Indicated weight at cut-off (lbs. per hour)	287	286
Missing quantity „ „ („ „ „)	73	174
Indicated weight at release („ „ „)	325	330
Missing quantity „ „ („ „ „)	35	130
Increase between release and cut-off	38	44

It would be seen that the apparent re-evaporation between release and cut-off was 38 lbs. per hour for the jacketed and 44 lbs. per hour for the unjacketed. The missing quantity of release for the unjacketed was 130 lbs. per hour, and for the jacketed 35 lbs. per hour. He had no doubt whatever in saying that, neglecting the leak into the cylinder, for the jacketed the whole of the 34 was valve leakage, and that the steam was perfectly dry at release. His own opinion also was that for the unjacketed the greater part of the 130 lbs. missing at release was leakage, and that the steam there was also practically dry at release. This showed that one of the

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(Mr. A. L. Mellanby.)

advantages of the jacket was that it reduced the valve leakage, probably by warming up the valve faces. He thought in that respect it was more useful than in reducing the condensation. The effect upon the condensation, if the engine was jacketed by steam of its own initial pressure, was very little, but the effect upon the valve leakage due to the heating of the ports by conduction was very great. In conclusion, he wished to say that if more engineering colleges were to take up work of the kind described in the Report, instead of devoting themselves solely to the teaching of rudimentary engine-testing, the manufacturers of the country would take a more active interest in the advancement of technical education.

Mr. R. H. CABENA said he wished to draw attention to the point which had been touched upon by almost every speaker, namely, the question of leakage; and in that connection he would refer to the leakage curves. On the last occasion Mr. Longridge pointed out that the curves on Fig. 32 (page 212) were parabolic; in other words the leakage was proportional to the square root of the pressure. He would first ask the author whether the zero point on the curves was obtained experimentally, or whether it was merely assumed that the zero pressure difference was a point on the curves.

Professor CAPPER said the point was assumed.

Mr. CABENA, continuing, said that in that case the portion of the curve between zero and the first plotted point was extrapolated, and this was also true of that portion of the curve to the right. There were thus remaining in the curve three experimental points, and it would be found that a straight line would satisfy those three points equally well. With respect to the point marked "lubrication failed," he would say that as the intermediate dotted curve represented the conditions of steam in all jackets, and stopped lubrication, they should expect to find that point on the intermediate curve. Neglecting the last point, on the intermediate curve it would be found that the plotted points also satisfied a straight line. Finally, a straight line would satisfy equally well the points in the upper curve. All of these lines cut the vertical at some point,

indicating an added constant quantity independent of the pressure difference. He suggested that the constant represented mechanical leakage, especially so on account of the experiments being conducted with blocked steam-ports. There was a clearance or pocket formed just over the blocking piece according to the diagrams in Fig. 33 (page 217), and they might reasonably expect the condensation on the cylinder face to be brought along towards the blocked steam-port

FIG. 56.

*Slide-Valve Leakage, plotted from Values in Table 2 (compare Fig. 32, page 212).
Steam in all jackets. Syphon Lubricator supplying freely.*

by the port in the main valve. The exhaust edge of the valve passing over the steam-port would remove that leakage water to the exhaust. These suggestions were of course speculative: there were other ways in which a mechanical leakage could take place. Plotting the leakage and pressure difference from the figures in Tables 1 and 2 (pages 210 and 211)—and not by taking the average values of the last three columns as was done in Fig. 32—he found that they were all straight lines, Fig. 56.

(Mr. R. H. Cabena.)

Taking the lowest line it would be found that it cut the vertical axis at a point representing a mechanical leakage of 5 lbs. per hour. Choosing a point on the axis of pressure difference at 34 lbs. and drawing a perpendicular to meet the plotted line, it would be found that a leakage of 15 lbs. per hour was obtained. He took a point at 34 lbs. pressure, because he found from figures of Dr. Nicolson's that for a difference of 34 lbs. pressure on the valve he obtained a coefficient of 0.02, for a $10\frac{1}{2}$ -inch cylinder (which was the nearest size of cylinder for which he had been able to find data).

If they applied these figures for pressure difference and leakage to determine the coefficient C, they would find leakage divided by pressure difference, multiplied by the overlap, upon the perimeter, gave exactly 0.02. That might be a mere coincidence, but it was peculiar that the same pressure gave the same result for these experiments. The total leakage obtained by Dr. Nicolson was greater than that obtained in the present case, but the valve was for a $10\frac{1}{2}$ -inch cylinder against that for a $6\frac{1}{2}$ -inch cylinder; and therefore the perimeter would be greater. He had not obtained the data for the perimeter of the larger valve, but he estimated that it would be approximately about as much greater as the leakage per hour was greater.

Another point to which he wished to draw attention was in connection with Table 3 (page 214), where in the last two lines the effect of leakage through the valve due to wire-drawing was given. Instead of comparing the second line from the bottom with the lower line, he proposed to take the third line of figures from Tables 1 and 2 (pages 210 and 211 to compare with the lower line of Table 3). Taking those figures, and setting them in Table 3, the following results given in Table 17 were obtained :—

It would be noticed that the leakage 20.6 was not very different from the leakage given in Table 3, 20.4; yet there was 66 lbs. difference in the wire-drawing, and the pressures on the valve were 46 and 45.25. If it were legitimate to take those figures from Table 1 for comparison, then it did not seem to prove that wire-drawing reduced the leakage. There was the remarkable fact about those figures that the pressure upon the valve was practically

TABLE 17.

Revs.	Steam Pipe before Inlet.	Steam Chest.	Exhaust.	Steam Pipe minus Chest.	Steam Chest minus Exhaust.	Leakage. Lbs. per hour.
50	66.25	63	17	3.25	46	20.6
50	132	63	17.75	69	45.25	20.4

the same, and that the leakage was also practically the same. It would be remembered that Dr. Nicolson found that, for a non-balanced valve with 34 lbs. pressure, he obtained a leakage of 41 lbs. per hour; for a balanced valve with 80.5 lbs. pressure he obtained a leakage of 267 lbs. per hour, pointing to the fact that not merely steam-chest pressure minus exhaust, but the actual pressure between the valve and the cylinder face, was a determining factor. He referred to that as being an indication of the reason why the leakage was the same in both cases, namely, that the pressures were equal, and not as a result of wire-drawing.

Mr. HENRY DAVEY, Member of Council, wished to refer in the first place to some remarks which fell from Mr. Maw at the opening of the discussion. He himself had occupied the position of Chairman of the Steam-Jacket Committee since its inception, which he was sorry to say was a long time ago. The Committee set to work immediately it was formed and had done some very useful work. One Report had been read at the meetings, but in the Proceedings of the Institution there were three Reports* which he ventured to say contained more information on the practical value of the steam-jacket than could be found elsewhere in any one publication. It appeared to him that a proposal was now being made to alter the line of research, and to revert to what the Committee started with. The Committee started experimenting with actual

* Proceedings, 1889, page 703; 1892, page 418; and 1894, page 535.

(Mr. Henry Davey.)

engines under actual working conditions, testing them with and without jackets, and finding out the differences; and that was the only rational way to get at facts in practical engineering.

He thoroughly joined with other speakers in their appreciation of the value of the present Report. It was valuable not only from a scientific point of view, because it pointed to the direction in which improvements might be effected in the steam-engine, but because it also pointed out defects which many practical engineers perhaps were not quite conversant with. The leakage question, the discussion of which had occupied the whole of the present evening and the previous meeting, was an incidental matter in the Paper, not of the supreme importance of some other facts which had been brought out. He quite agreed with Mr. Maw that it was useless for two Committees to be working on what was practically the same subject. There ought to be an amalgamation; but, when that was brought about and before any future experiments were undertaken, the line of investigation should be thoroughly discussed and laid down with some definiteness. He might mention that Professor Hudson Beare was now carrying out, with an apparatus which he had at his disposal in Edinburgh, some experiments bearing on the value of the steam-jacket; and probably when his report came to be presented it might, to a certain extent, clash with some of the author's investigations.

Having said that much with reference to the two Committees, he would venture to make one or two remarks on the Report itself. The leakage question seemed to be a very vexed one; and he began to wonder when he heard of the enormous amount of leakage which was supposed to take place through valves, how it was that engineers managed to get from steam-engines any efficiency at all. There was leakage of the pistons, leakage of the valves, initial condensation, and all sorts of things; but when one considered that some of the best steam-engines gave 70 per cent. of efficiency on the Rankine formula, one wondered how it was that there was so much leakage, and yet the engines gave such good results. He was convinced that in practical work there was not a leakage from valves that one would be led to expect from the very small experiments carried out. A

theory had been propounded that the valve ran on a film of water, that the water escaped into the port, that the valve would leak water easier than it would leak steam, and that more steam got away in the form of water. On reading the Report and observing that the leakage was somewhat in proportion to the increase in pressure, one wondered why increase in pressure did not squeeze out the film and so make it very much thinner, and if it got thinner the leakage would be very much less. It struck him that the leakage was more with the higher pressure because the valve was not tight to begin with, and never had been tight; it was probably warped. It was natural enough when there was a leakage space, that the leakage should increase with the pressure; it was mere common sense. The investigation which the author had been carrying out had shown how such a leakage might affect the general results. Professor Capper had been quite justified in getting at the leakage first, because his results would be vitiated if he did not know what the leakage was.

With regard to the value of the investigations, he thought they were of a purely scientific value. They were facts that had been ascertained with reference to a particular engine under particular circumstances, and they did not generally apply to engines in actual work; but nevertheless they had their scientific value. There were many anomalies with reference to the economy of steam-engines. It was generally supposed that the faster the engine worked, the better its economy. He thought it was pretty well known that pumping engines, working with a piston speed not exceeding 200 feet and making only 30 revolutions per minute, had given results in economy much higher than anything that had been produced by any other steam-engine. As he had said before, the Paper must be read from a scientific point of view, and hasty conclusions must not be drawn from it and applied in practice.

The gist of the Report appeared to be at the end. In Figs. 44 and 45 (pages 248 and 249) there was some information which was to a certain extent new, and which he thought contained some of the most valuable information in the Report. It appeared from the results that the jacket ceased to be of value when a certain number of revolutions had been attained

(Mr. Henry Davey.)

with a certain mean pressure in the cylinder. With regard to the value of the jacket, if the members referred to the three Reports of the Steam-Jacket Committee which had been published in the Proceedings of the Institution, they would find that very elaborate experiments were made with a triple-expansion engine at the East London Water Works, both with and without jackets. Those experiments were carried out with the greatest care and accuracy, and were repeated four or five times with consistent results. The difference between working the engine thoroughly steam-jacketed and working it without any steam-jackets in use was about 3 per cent. As a general fact, the more economical the engine was the less the percentage economy of the jacket. If an engine was using about 13 lbs. of steam per I.H.P. with an initial pressure of about 120 lbs. the value of the jacket became exceedingly small. With an old-fashioned single-cylinder engine, working with only 30 or 40 lbs. of steam, and cutting off at half-stroke, the percentage value of the jacket would be a very different matter altogether.

Mr. ROGER T. SMITH thought that one among the many very interesting facts brought out in the Report was that shown in Fig. 37 (page 229), indicating that as the steam-chest temperature was increased the dryness-fractions all tended to converge to one point. The same thing occurred in Fig. 39 (page 231), and in Fig. 44 (page 248), which had been referred to as one of the most satisfactory and novel results of the Report, exactly the same thing happened. As the initial pressure was increased, that is to say, as the temperature of the steam, in the unjacketed trials, increased, the thermal units per I.H.P. per minute for engines at different speeds, all tended towards one point. The results were all obtained with saturated steam, and through the kindness of Messrs. Belliss and Morcom he had been able to exhibit a diagram showing analogous results from a number of quick revolution engines of different sizes, all of them using superheated steam. On Fig. 57 (page 302) was plotted the variation with superheat of the lbs. of steam per kilowatt-hour supplied to seven engines, each coupled to a dynamo, ranging in output from 220 to 1,500 kw. They were all

non-jacketed condensing engines, and the lbs. of steam per kilowatt-hour had been plotted (from the saturation line as zero) for various degrees of superheat above the temperature corresponding to saturated steam at stop-valve pressure. All the engines were tested at full load, one of them being also tested at three-quarter load; and all the water measurements were made from the condensed steam. The interesting result was that if all the curves were produced sufficiently far, they met very nearly at one point, namely, 400° of superheat,* showing as a result that if one could only use enough superheat all engines of this type, of whatever size, were about equally economical. From a large number of experiments on the engine marked "A" on Fig. 57 (page 302), a series of curves had been plotted, Fig. 58 (page 303), giving the lbs. of steam per B.H.P. passing through the engine at all loads up to full load for saturated steam, and also for 50° , 100° , 150° , 200° , 250° , 300° , and 350° F. superheat above the temperature of saturated steam. These curves got flatter as the superheat increased, showing that, when sufficient superheat was used, an engine of this type tended to become equally economical at all loads.

He also desired to say a few words with regard to the question of valve leakage. The leakage measurements from the King's College engine had probably come out at as nearly an accurate result as could be obtained. He would like, however, to point out that the effect of leakage, in any given engine, on its economy depended on the size of the ports and the volume of the steam in the cylinder. The area of the port through which steam was admitted should be designed to suit the velocity of that steam. Being anxious to find out how the size of the little engine experimented on would compare with other engines in this respect, he had examined several engines of various sizes and types, and taken out the ratio between the area of the steam port and the volume of the cylinder plus clearance. The ratio had no value in itself whatever; it only indicated the sort of

* To show this result more clearly, all the engines should be using steam at the same stop-valve pressure. This was not the case, but it has been thought preferable to plot the actual experimental results rather than to rearrange them for one uniform stop-valve pressure.

(Mr. Roger T. Smith.)

FIG. 57.—*Non-Jacketed Quick-Revolution Triple-Expansion Condensing Engines, using Superheated Steam.*

Experiments on Messrs. Belliss and Morecom's Engines.

Set.	Kw. Output of Generator coupled to Engine.	Load at Test.	Stop Valve Steam Press. Lbs. per sq. inch.	Vacuum at Engine. Inches of Mercury.	Date of Test.
A	208	Full	155	26	Jan. 1904.
B	220	"	175	25	Nov. 1902.
C	308	"	190	25	Dec. 1902.
D	362	"	162	25.8	Feb. 1903.
E	500	"	150	26	Mar. 1904.
F ₁	700	"	190	27	Jan. 1905.
F ₂	580	"	190	27	Feb. 1905.
G	1456	Full	183	26	July & Aug. 1903.

FIG. 58.—*Non-Jacketed Quick-Revolution Triple-Expansion Engine.*
(Engine A of Fig. 57), showing effect of Superheat on Steam-Consumption at Varying Loads.

The percentage figures indicate the increase in lbs. of water per B.H.P.-hour over full load consumption.

(Mr. Roger T. Smith.)

relation in size between the space over which leakage could occur and the volume of the cylinder. The results were shown in Table 18 (page 305).

He had not sufficient particulars to work out the leakage perimeter and its ratio to the volume of the cylinder, and had taken the port area as the numerator, since this was applicable to all engines. The Sulzer engine, it should be said, worked a Riedler pump, having the pump valves mechanically shut and opened, and its piston speed was about double what was usual in a pumping engine. He merely mentioned those figures to show that, although the amount of leakage might be as much as 20 per cent. of the steam passing through the cylinder in the small engine used at King's College, when one came to a pumping engine, as Mr. Davey pointed out, the corresponding effect of leakage, owing to the enormous volume of the cylinder compared to the leakage area, was a very different matter altogether, and possibly some of the good results obtained by pumping engines might be due to that. He would also like to mention a couple of results of actual measurements of leakage from a Belliss engine using piston-valves. One of the engines (marked F_1 and F_2 in Fig. 57), a 1,200 H.P. engine, was first tested without any rings in the piston-valve at all, and it did not then give very good steam-consumption results; it was obvious, according to the indicator card, that there was leakage. It was then tested with spring rings in the piston-valve, and there was an increased efficiency in the steam consumption of the engine of 12 per cent. It is right to point out however that this improvement was not all due to reduction of leakage, as the low-pressure valves had been altered, giving a better steam distribution. He would like also to mention another actual leakage test made at Messrs. Belliss and Morcom's works with a small unjacketed high-speed engine, using steam at 80 lbs. worked under various conditions. The piston-valve was fitted with solid rings, which, in the first instance, were actually ground into the valve-chest so that they were as tight as possible. The water-consumption per indicated H.P. was obtained with the engine under those conditions, both non-condensing and condensing, with saturated and with slightly superheated steam, but the actual

TABLE 18.

Table of various sizes of Engines, showing Ratio of Space through which Leakage could occur to Volume of High-Pressure Cylinder and Clearance.

Type of Engine.	H.P.	Cylinder Diameter.			Stroke.	Revs. per min.	Piston speed.	Port Area. ÷ H.P. Cylinder Volume.
		H.P.	I.P.	L.P.				
		ins.	ins.	ins.	ins.		Ft. per min.	
King's College. . .	{ Up to 40 }	6½	—	—	14	{ 50 to 250 }	{ 116 to 584 }	1½
"Robb" used in Callendar and Nicolson's experiments. . .	—	10½	—	—	12	250	500	1½
Direct-acting beam Pumping . . .	75	16½	—	25	{ 40 and 60 }	20	{ 133 and 200 }	1½
Ditto	150	18½	—	31	{ 44 and 66 }	20	{ 147 and 220 }	1½
Direct-acting horizontal pumping with Sulzer valves	650	29·9	—	43·3	41·35	60	412	2½
Belliss and Morcom Quick-Revolution Engine (Size F ₁ and F ₂ on Fig. 57, p. 302) piston valves	1200	18½	27	40	20	250	835	1½
J. and H. McLaren Quick-Revolution Engine, slide valves	2500	25	39	62½	24	200	800	1½

(Mr. Roger T. Smith.)

amount of superheat was not measured. Six thousandths of an inch were then turned off the rings, and the engine was tested again under exactly the same conditions of load and of initial pressure, the result being an increased water consumption per indicated H.P. of $22\frac{1}{2}$ per cent. in the case of the saturated steam experiment, and of $10\frac{1}{2}$ per cent. with the slightly superheated steam experiment, both non-condensing. This showed the immense effect of even slight superheat in diminishing leakage, as referred to in Professor Capper's Table 3 (page 214). With the engine condensing, only one experiment was made with saturated steam, and there the increase of water consumption due to the reduction in the valve ring was $14\frac{1}{2}$ per cent.

Mr. Maw seemed to invite members to say what they thought the Committee should do in the future; and he should like to ask if it would be possible for the Committee to test an engine with drop-valves or Cornish valves. The drop-valve had this very great advantage, especially as far as leakage was concerned, that it could only be used as an admission valve. The exhaust valve must necessarily be at a different part of the cylinder; and it was impossible to have any leakage between the admission and exhaust ports such as occurred with the slide-valve. In that way the drop-valve must simplify results. Also, if the drop-valve was tight cold, one could easily test if it was tight hot; and no leakage, supposing it was tight and remained tight under the heat of the steam, could occur through the valve during the whole of the expansion part of the stroke. Therefore with a drop-valve engine it would appear (if the above reasoning was correct) that a simple experiment could be made by testing the leakage when standing before-hand, and if it existed, the valve could be re-ground so as to eliminate the leakage. He was sure there were a number of pumping-engines round London and other places working hour by hour and day by day under constant loads with drop-valves, some of them single-acting engines, on which leakage experiments could be easily made; and if a large commercial engine of that kind could be experimented with under such conditions, the results would be of advantage to engineers in general.

Discussion on Friday, 14th April 1905.

Mr. MARK ROBINSON, Member of Council, thought that after two evenings of a very interesting discussion the members might approve of shortening the proceedings by agreeing to take as said the compliments which the Report well deserved. He thought the author, after all the praise which had been offered to him, might not less appreciate a little criticism. On page 233 it was stated that "the total condensation up to cut-off increases with increased pressure and temperature of admission. As the cut-off was constant, condensation therefore increases with the difference between the temperature of admission and exhaust." That was the accepted view, well established by Mr. Willans' trials, and many others, namely, that the greater the range of temperature the greater the condensation, though in fact it was but another form of the truism that it took more heat to warm up a given weight of metal to a high temperature than it did to a low one. But on page 236, with the added dignity of italics, it was said that "expressed as a percentage of the steam entering the cylinder, the losses due to condensation on the cylinder walls of unjacketed engines diminish with increased initial pressure and temperature for a given ratio of expansion, quite independently of the influence of speed." The words "for a given ratio of expansion" made the preceding statement true, but he was afraid they also made it largely valueless, and even dangerous, by tending to mislead the learner. The main deduction which *seemed* to be drawn in that paragraph was that the condensation diminished with increasing temperature. Of course it diminished, but it was not *because* of the increasing temperature; it was *in spite* of the increasing temperature. He was aware that the data for this statement would be found in the Paper, and of course the author was under no misapprehension, but he thought the way the facts were presented tended to mislead a reader not well up in the subject. The higher range of temperature, supposing other things to be equal (as the "other things" ought to be, in all experimenting), *increased* the condensation; but since the higher temperature was merely the result

(Mr. Mark Robinson.)

of throttling the steam less, and so passing a much larger quantity through the engine, the condensation, which was more or less a fixed quantity depending on the temperature and not on the quantity of the steam, of course diminished rapidly as a percentage of the quantity of steam used. He could not help thinking that the author might have pointed out, not that the percentage diminished with the rise of temperature, but that it increased with it, although in the present case the increase was neutralised by the merely accidental interference of other causes.

If the work done in a shop was increased, so that the tools were more fully employed, the standing charges would also increase, but in a less proportion than the value of the work turned out; hence they were easier to bear than when the shop was working with only half the tools employed. But he would be a bad accountant who set forth that fact to support a general statement that standing charges diminished when more tools were running, unless he had laid more stress upon the qualification than upon the statement.

With regard to valve leakage, he presumed that question had been fully dealt with by Captain Sankey at the last Meeting, although he had not the pleasure of hearing him; but he could not help inquiring why, in carrying out experiments of a kind in which any serious valve leakage must be fatal to accuracy, such an imperfect instrument as a slide-valve should be used. Heated up on one side with live steam and cooled on the other by exhaust, a slide-valve necessarily warped and twisted, and let the steam through. With Corliss valves and piston-valves to fall back upon, he could not understand why the makers of experimental engines should use slide-valves rather than adopt something better.

Mr. DRUITT HALPIN was greatly surprised to hear from Mr. Pendred (page 282), in speaking on the question of jackets, that none of the racers that had won first place were jacketed. He had spoken to him about it, and Mr. Pendred had said they were jacketed, but that their jackets were out of use during the time of the tests. If that was so the matter ought to be threshed out, and it ought to be known now before it became a matter so far back in history as to

be untraceable. Taking the engines that ran in 1872 at Cardiff, Clayton's engine certainly was most elaborately jacketed, not alone on the barrel, covers, and valve-chest, but everywhere, the whole thing being put even into the smoke-box. He did not clearly see therefore how it could have been more jacketed. If it was the fact, as Mr. Pendred said, that those jackets were shut off, it would be a valuable thing to know.* In 1887 they ran at Newcastle, and Mr. Pendred said that the jackets were shut off there in the engine which was given the prize. But there were several others that ran, in which he knew the jackets were not shut off. In 1888 they ran in Glasgow, and there he was positive the jackets were not shut off in the winning engine, because he designed the engine himself, and it was so designed that the jackets could not be shut off. The value of jacketing had been discussed a great deal in the Institution, as also had been the increasing their efficiency by ribbing the jackets, and one Professor proved completely to his own satisfaction that the whole thing was nonsense, that it retarded the transmission of heat into the cylinder. But when the amount of heat actually transmitted was absolutely measured by weighing water, and using thermometers, it was found instead of retarding the transmission of heat, the transmission of heat was increased in the ratio of 3.4 to one. He had seen scores of those engines, common, commercial, portable engines, without superheating and without condensing, developing I.H.P. with 19.7 and 19.5 lbs. of water.

With regard to experimental engines, there was one at Nottingham College where the jacketing was carried out from the piston-rod right round back to the piston-rod, including both covers and valve-chests. There was a cover at the end of each cylinder and a false cover outside, so that the whole of the flanges of the covers were also jacketed, and the jackets were ribbed all over and the covers were also ribbed all over. As much heat was got through as possible, and the result was a small engine indicating 30 H.P.

* Mr. Halpin subsequently wrote to Messrs. Clayton and Shuttleworth asking them whether a steam-jacket was actually in use when their engine was tested in Cardiff in 1872, but they answered that all the people connected with the trial were dead, and unfortunately they had no record of the matter.

(Mr. Druitt Halpin.)

with 65 lbs. of steam doing its work on 17·4 lbs. of water without super heating the steam.

The question had been raised of slide-valve leakage. It was in 1870 or 1871 that he first saw a slide-valve working, and the leakage could be seen. He exhibited at the Meeting a model of one that he had at work in a locomotive, 17-inch engine, Fig. 59 (below). The valves were in the open ended steam-chest. The valve-rods went through guides at the back end and the valves leaked. They were scraped absolutely true, but still they leaked. People did

FIG. 59.—*Smart's Balanced Slide-Valve.*
Model exhibited in South Kensington Museum.

not know as much 35 years ago as they knew now, and they could not discover the reason of the leakage. A bucket with heater was used and a fire was put under it, and the water kept boiling, and one of the valves was taken by tongs and put in the bucket, so that it was known that the temperature was at 210° or 212° F. It was taken out quickly, and put on the surface plate again and the whole thing could be seen. Three or four layers of tracing paper were put under the valve at various positions, and the whole valve was found to be buckled up from heat. It might be said that that was an unfair way to look at it, because a grid-iron valve taking steam through it was more complicated than the ordinary flat-valve, and for that reason it might be more sensitive to changes of temperature.

Mr. C. H. WINGFIELD said that reference was made (page 174) to some separators consisting of U tubes, and the result of some experiments which were kindly made by Messrs. Willans and Robinson for him might be of interest. He could not give the exact figures, as they were made for another Committee and he was not at liberty to publish details. Fig. 60 represented the apparatus and showed at A a tube arranged on the same principle as those used by the author. The steam, and whatever water there was at the bottom of the main steam-pipe, entered where the arrow indicated, struck

FIG. 60.

Separation of the Water.

FIG. 61.

Dryness-Fraction.

against the side of the vertical pipe, and the steam went upwards, the water being supposed to drop down to the bottom. He had great doubts as to the efficiency of such a device and, in order to ascertain if these were well founded, a second separator B was fitted in series with A. If the separator marked A was perfectly efficient, there would be no water in B except such as was condensed between the two. This could only be minute in quantity, as the whole apparatus was very carefully lagged. The result of the experiments was that, when steam was passing slowly, rather more water was caught in A than in B, and when it was made to pass quickly more water was caught in B than in A.

He wished to say a few words about the "dryness-fraction" as,

(Mr. C. H. Wingfield.)

although of great importance in the thermodynamics of steam, its meaning was sometimes misunderstood, and the author was not quite so clear in his words as he knew him to be in his mind on this point. In an indicator diagram, such as shown in Fig. 61 (page 311), horizontal distances corresponded, as was well-known, to volumes, when measured on a suitable scale. Let two saturation curves $a b$ and $c d$ be drawn; the former through b (the point where the exhaust-valve closed) and the latter at such a distance from $a b$ that, if any horizontal line were drawn through the two curves, the length of the intercept between them would represent the volume which the weight of steam and moisture discharged through the exhaust pipe at each stroke would occupy (if it were all in the form of saturated steam), at the pressure corresponding to the height of the horizontal line in question.

Similarly, the horizontal width of the shaded area at any point was a measure of the volume of saturated steam (at the pressure corresponding to the height of the selected point), which was equal in weight to that of the volume $e b$ of "play-steam" * shut up in the cylinder and clearance when the exhaust-valve closed at b . Although not measured with the condensed discharge from the exhaust, this "play-steam" was present in the cylinder and clearance throughout the expansion, and its weight must therefore appear both in the numerator and denominator of the "dryness-fraction." At any one pressure, such for instance as the pressure p of release, horizontal distances might equally well be taken as measures of weight as of volume, and the figure was marked accordingly. In the terms shown on the diagram, the dryness-fraction at "release" was obtained by dividing w by W , and this agreed with the author's definition on page 239, which he would take the liberty of somewhat amplifying, namely, the "ratio of steam present in the cylinder at cut-off" (or any other point in the expansion-curve) "to the total steam and moisture" then present in the cylinder. (Of course in the term "cylinder" the clearance volume was included.)

* A name introduced by Mr. Macfarlane Gray.

Another definition was, however, given on page 226 (in which the "play-steam" did not appear in the denominator) corresponding to w/v_1 in Fig. 61 (page 311), and he suggested this should be corrected in the two places where the slip occurred. [This has been done.]

Yet another definition had been given in a popular text-book,* which was quite incorrect. It would correspond with v/v_1 in Fig. 61, if the compression line followed the saturation curve as the author of the book appeared to have assumed. This was by no means necessarily the case, however. The fact that the speaker had often found confusion to exist with regard to the "dryness-fraction" must be his excuse for dwelling upon it at such length.

On page 225 it was mentioned that in some of the indicator diagrams the admission line sloped considerably. As far as he could see, the author had taken the highest point as the admission pressure. Mr. Willans, in the first of his classical papers, took the *mean* admission pressure, because he considered that the results obtained in calculation on that basis were more consistent than when he took the top of the line. In his second Paper, however, he abandoned the first practice in favour of the second.

On page 175 he noticed that the vacuum was regulated by opening an air-cock to the condenser. As a matter of interest he should like to know whether this cock wanted much attention, in order to keep the vacuum constant. A rather interesting action took place when air was mixed with steam. It might be known, but he did not think it was as well known as it should be. When even a very small quantity of air went into a condenser it was astonishing, at first sight, to find how much the vacuum was affected. What happened, he believed, was that the air and steam mixed up with more or less uniformity, probably less. Particles of steam charged with air came into contact with the cooling surface, and were condensed, each leaving its partner of air behind it. Gradually, enough air was accumulated to make a cushion of approximately non-conducting material, which reduced the efficiency of the condensing surfaces,

* H. A. Golding's "Theta-phi diagram" (1898 edition), page 22.

(Mr. C. H. Wingfield.)

until a puff of steam came and blew it off, and then the action began again. On this account he anticipated the air-cock might want a good deal of regulation to maintain a uniform vacuum.

On page 190 the author referred to a standard of comparison, in which he took the same upper limit of temperature as that which was fixed by a committee of the Institution of Civil Engineers, namely, the temperature at the stop-valve. When dealing with the actual engine however, the preceding paragraph showed he took the temperature of the steam-chest as his upper limit: this being less was rather in favour of the engine. The lower limit for the standard engine was not given, but in the case of the actual engine the author took the temperature of the condensed water issuing from the condenser. The Institution of Civil Engineers fixed it as the temperature of the steam issuing from the exhaust branch of the engine. No doubt the author had good reasons for departing from their standard in these respects, and it would be interesting to know what they were.

In conclusion, he did not think sufficient attention had been called to diagrams, Figs. 44 and 45 (pages 248 and 249). They were most valuable because, while they showed (what everyone, more or less, thought they knew) that at high pressures jackets began to lose efficiency; they also showed there was a lower limit, and that the jacket was only useful between those limits. The author had done some very good work in that, and it ought to be appreciated.

Mr. MICHAEL LONGRIDGE, Member of Council, desired to say one or two words about steam separators, because evidently from the diagrams and from many designs of separators which he had seen, there was a considerable misapprehension about the way they worked. Most of the separators designed proceeded on the principle that there was a quiet atmosphere within them, whereas as a matter of fact there was a gale of wind blowing through them, and a very heavy gale too. If one tried to heat the water out of the steam by directing the stream against a surface, the water was broken up into a spray of small drops which were easily blown away by the gale of wind and carried forward in the steam; but if one took advantage of the

momentum of these drops and allowed them to continue their course and deflected the steam, most of the water would be separated from the steam. Probably most of the members had played the game of billiards, and they all knew that if a ball was directed against the cushion it would rebound obliquely, the paths before and after contact making approximately equal angles with the normal to the cushion at the point of contact. In the same way a particle of water coming along the pipe, as sketched, Fig. 62, would strike the surface of the pipe and would come off at an equal angle to the normal to that

FIG. 62.—*Steam Separators.*

surface and go down the pipe, whereas the steam would find its way straight up. That he thought was the proper principle on which to make a separator.

Mr. H. E. WIMPERIS said that the idea was to have as few variables as possible (page 173), and that the two should be temperature and speed, the cut-off and all other conditions being kept constant. That seemed to him a most excellent thing. The Paper was a sort of quarry in which other people might work. Mr. Willans had given a somewhat similar Paper many years ago, and many people had been at work on it, prominent amongst them being Professor Perry; since then a Paper on Gas-Engine Experiments had been read by Professor Burstall and now there was this Paper by Professor Capper. All

(Mr. H. E. Wimperis.)

those Papers were most useful in furnishing material to investigators, especially to those of the student class, as it familiarised them with the ideas that underlay steam-engine and gas-engine working. There was a very important statement on page 177, and one to which but little attention had been paid. It was said: "If on working out any trial any of the conditions or observations were found to vary beyond definite limits, the trial was rejected and repeated." Doubtless Professor Capper in replying would mention what those limits were. It was absolutely necessary in certain cases to work on a rule of that kind, but it was easy to see that great danger was incurred unless it was worked on the most conservative lines.

On page 179 were given the errors due to backlash in the indicator, and he confessed he did not understand the explanation given at the bottom of the page. If the backlash was constant on the up-stroke and on the down-stroke, seeing that the indicator pencil must move up as much as it moved down and move to the right as much as to the left, one would imagine that the area of error would have been the same for all size of diagrams, and although more important for small ones would not be affected by the considerations stated in the footnote. On page 222 the piston leak was found to be less than 2 per cent. of the steam consumption of the engine. How did that agree with a calculation based on the customary formula? If he might do so without presumption, he would like to congratulate Professor Waynforth on being so happily associated in so important a work and in the privilege of working under so distinguished a chief.

Professor CAPPER, in reply, said he would not detain the meeting long, but would begin by thanking those who had spoken so kindly about the work embodied in the Report. It had been an arduous work, but it had been a work which had been intensely interesting. One of the great difficulties had been the mass of information accumulated, and it had been extremely difficult to compress it within the limits of a single Paper. For that reason there was no doubt that some of the criticism that had been called forth had only too great justification. Dealing with points raised in discussion he

might perhaps deal with those which had been made that evening first and reply to them in the reverse order of the speakers.

Mr. Wimperis (page 316) asked how far the piston leak found by experiment agreed with formula? He was not quite clear to what formula Mr. Wimperis was referring; he thought it was probably the formula by Callendar and Nicolson for slide-valve leakage. But that formula did not really apply in the case of the piston leak. He did not moreover think the formula was a final one, as he would show later on. Mr. Longridge had given a very lucid and interesting description of how to make separators, which was extremely helpful, as Mr. Longridge's suggestions generally were. Mr. Mark Robinson had attacked one passage in the Report, and added that no doubt it would be intelligible if the Paper were carefully studied. To pick out a clause as Mr. Mark Robinson did and to say that that clause might be misunderstood, if the text was not read, was perhaps a little bit unfair to one who was trying to make lucid what was a very complicated subject. He really could not write a Paper on such a subject that would be clear to those who did not take the trouble to read it.

With regard to the dryness-fraction, Mr. Wingfield criticised the definitions in the Paper as ambiguous, but, as far as he could gather, Mr. Wingfield's definition was exactly the same as his own. If the members would turn to page 226, they would see that what he said was not in the least that the dryness-fraction was the ratio between the steam consumed by the engine as measured on an indicator and the condenser discharge, but the ratio between the steam measured upon the indicator card up to cut-off or release and the measured condenser discharge from the exhaust. That was exactly the ratio $\frac{w}{W}$

Mr. WINGFIELD said he was speaking of the denominator W .

Professor CAPPER said that w was the whole steam up to the saturation curve and therefore gave the same ratio as Mr. Wingfield. He would alter the wording before the final reprint so as to avoid any possible ambiguity.

(Professor Capper.)

With regard to trial B_1 (page 198), if the Paper was read, a full explanation of the irregularity in the dryness-fractions of B_1 would be found. This trial was dealt with at considerable length. The steam at admission on this trial was dryer than on the others, because in B_1 there was a considerable additional throttling, and therefore the steam when admitted was dryer than in the other cases. That was the reason why B_1 showed irregularity with regard to dryness.

Mr. Pendred drew attention to the fact that an experimenter was very apt, however strenuously he might endeavour to avoid the fault, to be led away in a scientific laboratory experiment from the usual conditions of working practice. In order to make experiments of the sort described, as anyone who had tried would know, one must of necessity prevent a large part of the accidental discrepancies that occurred in ordinary practical working, or else one could not get the information one was seeking at all. Therefore when it was urged that during the experiments ordinary working conditions did not obtain, one could only say humbly that some of the variables must be cut out or else scientific information could not be got at all. When Mr. Pendred said that the author did not make himself quite clear with regard to the question of re-evaporation he thought it would be possible—although he should be sorry to do so—to retort that Mr. Pendred still more laid himself open to attack when he said (page 283) that “whatever went into the cylinder as steam must leave it as steam, and whatever went into it as water must leave it as water,” and made that bold statement without any limitation whatever. That statement was perfectly untrue, if it was not very carefully qualified. It was only true with a closed cycle. In the experiments under discussion the author was not concerned with what happened after opening to exhaust. After opening to exhaust, such very variable conditions existed that any exact determination of what was taking place was extremely difficult, and even if this were determined, except as regards the main factors, such as temperature pressure, etc., the results were not of much practical value to the particular enquiry under discussion. The work in the present experiments had chiefly been devoted to trying to determine the

laws governing initial condensation. This was the original object of the enquiry. Therefore, when in the Report he spoke of "re-evaporation," he spoke of re-evaporation in its usual acceptation, that is, up to the point of release. After that point there was of course a great deal of re-evaporation into the condenser, but they were not concerned with that phenomenon.

With regard to jackets, Mr. Pendred had said (page 282) that many experiments had been made thirty-two or thirty-three years ago on jackets, and it was found in all cases that the engines without jackets were more economical than those with. He did not wonder at that. Countless experiments could be adduced to show that and countless others to show the contrary. Very little was known about the behaviour of the jackets thirty-three years ago. It all depended on whether the jackets were used and worked soundly or not, and he hoped he had possibly added a little to that information by this Paper. If one used jackets (as was generally the case thirty-three years ago, and was sometimes the case even now) without providing any possibility for the steam to circulate in the jackets, one might just as well put the cylinder into the condenser at once; and consequently one was likely under those conditions to find the non-jacketed engine more efficient than the jacketed. If enquiry was made into many of those cases where the jackets were detrimental, it would be found that instead of being steam-jackets the jackets were really water-jackets, and for that reason a good many of the comparative experiments were worthless.

The main portion of all the remarks that had been made had been devoted to valve leakage, as might have been expected. It was an extremely important subject, and an enormous debt of gratitude was owing to Messrs. Callendar and Nicolson for having first drawn attention to it. But an enormous grudge was also owing to them from the experimenters who came after them, because they had enormously increased the labours of anybody who followed after them. It was perfectly impossible to get any reliable results as to initial condensation and re-evaporation, without measuring the valve leakage. Therefore it was desirable to so add up and pile up experiment on experiment on different types of valve, on different ways

(Professor Capper.)

of working valves, and so on, that knowledge might be gained of what leakage was likely to be obtained under any given set of conditions. He hoped nobody would think that he pretended these experiments were in any way exhaustive. They had, as Mr. Maw said, only just touched the fringe of the subject, and he hoped that others working on the same lines would add vastly more information. The law of leakage suggested by Messrs. Callendar and Nicolson, which he had accepted tentatively, and only tentatively, was thrown out as a working hypothesis. It seemed to him that the best way to try and add to knowledge was to take the hypothesis of one's predecessors and see how far one's own results concurred with this hypothesis. Therefore, as he stated in the report, he had adopted the constant suggested by Messrs. Callendar and Nicolson. The present experiments showed that the value of C , which on their hypothesis should be a constant, varied so much that it was not a constant at all. That showed that the assumptions made must be modified, and that the suggestion of Messrs. Callendar and Nicolson was only a tentative one. Mr. Longridge and other speakers had rightly pointed out that there were certain indications—he would not say more because the experiments were not numerous enough to be sure about it—but there were indications that the variation of leakage was possibly not directly as the pressure but as the square root of the pressure. Mr. Cabena at the last Meeting rightly pointed out that the results on Fig. 32 (page 212) really lay on a straight line and that the first part was extrapolated. If that straight line were produced back to the origin of pressure differences, it would be found that at zero pressure there was still a leakage which Mr. Cabena attributed to mechanical leakage. That was perfectly true, and he himself was struck with it when he first plotted the curve. He drew out the curve in all sorts of ways varying with temperature, with pressure, and with difference of pressure.

Professor Burstall pointed out (page 286) that the curves were straight lines which came to an origin of temperature, but it was curious that if the points were drawn as straight lines they also came to an origin of pressure, not of difference of pressure but of

absolute pressure. Starting from zero of pressure instead of a zero of the difference of pressure, a perfectly straight line was obtained from those points. He failed to see any physical explanation of this fact, but it was suggestive, possibly as Mr. Cabena said, of mechanical leakage taking place.

Mr. Cabena had also remarked on the lubrication failure. That was rather an interesting point, and perhaps in his desire in the Report to be succinct he had not been absolutely clear. Not only did lubrication fail in that case, but as a matter of fact the valve seized and was scored. The valve was taken out and he made some subsequent experiments to see what the effect of the scoring was, and the interesting point was that after the burr had been taken off—there was one pretty deep score in the valve—the leakage seemed to be reduced by that score. That no doubt, Mr. Longridge would say, was just as he expected. The effect of having a score which came quite close to the edge of the valve was, as far as it had any effect at all, to reduce the leakage, partly probably owing to the fact that there was improved lubrication and possibly owing to what Mr. Longridge maintained, that a valve which had no lap at all except a line would be really more tight than one which had lap. As he had shown in the Paper this view was not borne out by experiment.

Mr. Stromeyer had referred to Fig. 36 (page 222) and asked why the experiments were carried out under such bad conditions; and in his usual thorough manner, after working out the weight of the steam at each point of the stroke from the diagram, said that there must have been some communication with the atmosphere. As a matter of fact there was, because the leak was being measured at the time, and naturally there was a communication through a suitable trap with the atmosphere, and that was of course what upset the diagram from Mr. Stromeyer's point of view. He thought Mr. Stromeyer took it as though the blocked end was simply shut up, in which case of course there would not have been the variation he alluded to. Mr. Stromeyer's figures bore out the view that the leakage was negligible from the piston under all conditions.

He had to thank Mr. Maw especially for his kind remarks (page 267). Mr. Maw asked him to explain how the lubrication of the

(Professor Capper.)

valve was carried out (page 270). It was carried out in the following manner. There was a sight-feed lubricator which entered the steam-pipe after the stop-valve. There was an ordinary siphon lubricator on the top of the steam-chest, and the sight-feed lubricator was run in all cases as stated in the Tables at about two drops per minute. The difference between the two top curves of Fig. 32 (page 212) and the bottom ones was that in the bottom ones the siphon lubricator on the steam-chest was running freely, while in others the siphon lubricator on the steam-chest was stopped. Where there was not a large quantity of steam passing through, as, for instance, when the only steam passing was the steam that had leaked past the valve, one would expect to get a more perfect lubrication than would be obtained with the full quantity of steam running through, as would be the case when an engine was in full work. Mr. Maw had also asked at what pressure the jackets were supplied with steam. In all cases the jackets were supplied with steam at steam-chest pressure.

Professor Burstall had pointed out (page 286) that the leakage of the slide-valve was probably to a large extent due to the warping of the valve. That was possibly right. He also pointed out as already noted that the leakage varied as the temperature, if the diagrams were drawn on a scale of temperatures. He was also surprised at the dryness of the steam. That again was due to the difficulty of thoroughly reading the Report in the short time at the disposal of the members. It was fully explained in the Report that the steam was dry partly owing to the fact that a separator was used somewhat of the form, but not quite the form, of that suggested by Mr. Longridge, and secondly because it was found impossible to get steady and uniform pressures in the steam-chest from the boiler without throttling. As a result, the steam was in all cases dried to a certain extent before coming into the cylinder by that throttling.

Mr. Mellanby had made some very interesting and instructive additions to the discussion both on his own behalf and that of Dr. Nicolson, whose opinion on the question was of course of great value. He said they were making experiments in the laboratory at Manchester with a loose slide-face and a plate underneath, and from these experiments they found the leakage in actual running was twice as much as the

experimental leakages with blocked ports. He knew Dr. Nicolson's skill and care in experimenting would not allow him to do so, but it was possible one might not get a perfectly tight joint with a face of that sort, and it was possible that there might be an addition to the leakage in that way. He could not believe that there was generally leakage to the extent suggested in ordinary practice.

Mr. Pendred had remarked that these experiments had been made on one engine, and generalisations made from it for all engines (page 284). He did think that he really had protected himself against criticism of that sort. He must repeat again that he did not say that the results he had got from the one engine were universally applicable. But individual experiments after all were of value, and if those individual experiments were multiplied on a number of engines then real progress in knowledge would be made. To that end he claimed that this research was a real contribution. The lines on which the Committee might very usefully make further investigations would be partly on the lines which Mr. Maw suggested, namely, making experiments on a number of different valves, if one could get large engines placed at one's disposal. Also there was a very large amount of information he should like to be able to obtain on the increased range of temperatures due to running the engine condensing, and also on the effect of changes in the ratio of expansion. The whole of the results were limited by the fact that there was a constant ratio of expansion throughout.

Captain Sankey's results (page 277) on the Willans engines were extremely important, and Mr. Roger Smith's contribution on the effect of superheating was also of great value and very instructive. Captain Sankey had also kindly pointed out the slip of the tongue by which he had attributed the name temperature-entropy to Mr. Macfarlane Gray instead of $\theta\phi$ which he intended to say. He thanked all who had spoken for the kind way in which they had received his researches, and he hoped no one would consider that the results were final but only suggestive, and that they were suggestive for good and not for evil. He trusted that the Report would prove a means by which future workers might find help in elucidating the problems better than he had been able to explain them.

On the motion of the PRESIDENT, a hearty vote of thanks was accorded to Professor Capper.

Communications.

Mr. E. A. FORWARD wrote that with regard to the valve-leakage diagram, Fig. 32 (page 212), the author had stated that the last point on the lowest curve was too high owing to the failure of the lubrication, and that the valve actually seized. The curves were, however, plotted from the mean of the figures given in the last three columns of Table 2 (43 lbs.), and the writer would like to know whether the lubricant failed in all three experiments, as the lowest figure 41.2 lbs. was also very much above the full line curve. His own opinion was that valve leakage was by no means so serious as these experiments appeared to show, and he believed that the method of testing with blocked ports was not sufficiently representative of the actual working conditions of the engine. It seemed to him that while the steam was flowing into one or other of the ports very little would leak under the valve, and it was only when this flow ceased that the conditions became in any way similar to the experimental ones. He would suggest that leakage experiments should be carried out on a cylinder provided with an ordinary three-ported valve-face and also extra exhaust-valves. The ordinary valve should be replaced by a flat plate, and the engine worked under steam as usual. Something like the true leak would then take place from the steam-chest to the dummy exhaust.

Mr. A. T. J. KERSEY wrote that the discussion had hitherto been almost wholly confined to valve leakage. This was, of course, a most important question, since the accuracy of other conclusions depended upon the correct deductions being made for leakage, but it must be regarded as merely incidental to the main object of the research, that is, to determine how initial condensation and

re-evaporation were affected by the range of temperature in the cylinder and the speed of the engine. The deductions made for valve leakage appeared to be sufficiently approximate to enable conclusions to be drawn as to condensation in the cylinder, but further extensive experiments appeared to be necessary in order to determine how much of the leakage was due to condensation and re-evaporation and how much to distortion of the valve due to the high temperature of the steam.

With regard to the effect of range of temperature on condensation, it appeared to be quite hopeless to expect to obtain any simple law connecting these. In the first place, the temperature of the steam diminished continuously from cut-off to release. There was then a drop of temperature, and the return stroke took place with exhaust steam of a constant temperature until compression took place, raising the temperature before admission. In this connection it might be pointed out that the effect of compression was of greater importance in the A trials than in the D trials. Thus far more heat would be lost by the clearance spaces during exhaust than during expansion, so that, apart from the effect of time on the variation of temperature of the clearance surfaces, it would be seen that the initial condensation could not be expressed as a simple function of the total range of temperature of the steam. It seemed to him that, if the results were to be of comparative value and any quantitative laws were to be deduced from the research when completed, some mathematical theory based upon the thickness and conductivity of the metal and the cyclical variation of temperature expressed as a Fourier series was desirable before a series of experiments of this kind was attempted, in order that there should be no complication due to more than one variable in any one set of experiments. Such a theory would be highly complicated by the fact that all parts of the cylinder walls were not subjected to the same range of temperature, but in the absence of a theory extreme care was necessary in settling the variables, it being evident that mere range of temperature was not a simple variable. Now in this series of trials the range of temperature was changed by altering the initial pressure, and this complicated matters, since, as stated in the

(Mr. A. T. J. Kersey.)

Report, the initial condensation was affected by alteration of initial pressure, owing to the exposed surface per lb. of steam admitted being thus altered; and more satisfactory conclusions might have been drawn if it had been possible to keep the initial pressure constant and vary the back pressure in order to vary the temperature range. This could be accomplished by means of a loaded valve on the exhaust, and he did not see any special difficulties which could not be overcome. He would like to make it clear that these criticisms were not offered in any carping spirit. The general conclusions drawn from the Report were most valuable, but there was a difficulty in analysing the results owing to the complication of variables.

Since the back pressure was constant in all the experiments, the writer had endeavoured to deduce laws connecting condensation with speed and initial pressure. In the absence of a theory such laws were, of course, not necessarily of general application, and he had not then access to other experimental results in order to check them. They might, however, be worth a certain amount of consideration. The following was a summary of the conclusions arrived at for the unjacketed trials, and it would be found that they agreed fairly well with the experimental results.

$$(1) \text{ Wetness-fraction at cut-off (deducting leakage)} = \frac{6.37}{\sqrt[3]{p.n}} = y_0.$$

From this the initial condensation per stroke might be deduced as follows:—

Volume of indicated steam per stroke = $c + x$

where c = clearance volume

x = volume traced by piston at cut-off.

Let w = weight of 1 cubic foot of steam,

$$\text{Then, total weight of steam per stroke} = \frac{w(c+x)}{1-y_0},$$

Therefore weight of condensed steam per stroke

$$\begin{aligned} S_c &= w(c+x) \left\{ \frac{1}{1-y_0} - 1 \right\} \\ &= \frac{w \cdot y_0 \cdot (c+x)}{1-y_0} \end{aligned}$$

Taking the value of y_0 given in (1), then

$$S_c = \frac{6.37 w (c + x)}{\sqrt[3]{p.n} - 6.37}$$

From data given $c = 0.029$ cubic feet; $x = 0.093$ cubic feet.

For the range of pressures given

$$w = 0.0022 p + 0.009.$$

$$\text{Therefore } S_c = \frac{0.00178 p + 0.00717}{\sqrt[3]{p.n} - 6.37}$$

Owing to wire-drawing, compression, etc., a coefficient of 0.85 has to be applied, and the following expressed the actual weight of condensed steam per stroke

$$(2) \quad S_c = \frac{0.0015 (p + 4)}{\sqrt[3]{p.n} - 6.4}$$

The ratio of re-evaporation to condensation is fairly well expressed by

$$(3) \quad \frac{S_e}{S_c} = 0.07 \sqrt[3]{n}$$

Therefore net condensation per stroke

$$(4) \quad = \frac{0.0015 (1 - 0.07 \sqrt[3]{n}) (p + 4)}{\sqrt[3]{p.n} - 6.4}$$

This expression was less complicated than the probable correct expression taking all the conditions into account.

Other approximate laws were

$$(5) \text{ Missing steam per I.H.P. hour} = \frac{9500}{p \sqrt[3]{n}}$$

$$(6) \text{ Indicated steam per I.H.P. hour} = 18 + \frac{420}{p - 16}$$

$$(7) \text{ Therefore } \frac{\text{Missing steam}}{\text{Indicated steam}} = \frac{528}{p \sqrt[3]{n}} \times \frac{p - 16}{p + 7.3}$$

He had much diffidence in presenting these conclusions, but it seemed impossible to express them as simple functions of the temperature range.

It was a great pity that so many isolated sets of experiments were being carried out by other observers, without any co-ordination of purpose or even knowledge of each other's work. It would be of

(Mr. A. T. J. Kersey.)

immense value if a series of tests exactly similar to these could be carried out simultaneously on say half-a-dozen engines of different types, so that the idiosyncrasies of a particular engine might be eliminated, or at least discovered, from the general conclusions drawn.* He was in thorough agreement with the remarks of Mr. Mellanby (page 294) as to the desirability of furnishing engines in technical institutions which provided something more than instruction in ordinary commercial testing routine. He suggested that a committee of the Institution could do much in the way of advice and criticism, if experimenters could be induced to put their schemes before this committee with a view to a scientific sequence of experiments on determined lines.

Mr. WILLIAM MASON wrote that during the discussion on Messrs. Callendar and Nicolson's Paper on "The Law of Condensation of Steam,"† Mr. Druiitt Halpin mentioned that he had tested a slide-valve of the balanced type for warping. The face of the valve when cold was trued to a surface plate; but there was serious warping shown when the valve was made hot in boiling water and again tested on the surface plate. The valve of the experimental engine when similarly tested had no warping which could be detected by ordinary methods of measurement.

It was interesting to estimate the mean thickness of the film of water leaking past the valve. This film would vary in thickness all round the valve, and also as the valve moved. But assuming, for the purpose of estimation, that it was uniform, and 0.0004-inch thick round a perimeter of 22.5 inches, the velocity of the water (if water only and no steam transpired) worked out at from 1 to 4 feet

* To take one instance, let them consider another engine with the same cylinder-volume, but different diameter and stroke. For the same cut-off the same weight of steam would be admitted, but the exposed surface would be much different, and this would affect the initial condensation. For example, a cylinder 8 inches diameter and 12 inches stroke would have 30 per cent. more exposed surface per unit volume than a cylinder 10 inches diameter and 7½ inches stroke, taking cut-off at 0.25 in each.

† Proceedings, The Institution of Civil Engineers, 1897-S, vol. cxxxi, page 212.

per second for the leakages measured. The figure 0.0004 inch was chosen because this was the minimum thickness of oil film calculated by Professor Osborne Reynolds for one of Mr. Beauchamp Tower's experiments on Journal Lubrication; a value the order of which was afterwards checked by direct measurement.

Professor Burstall has pointed out (page 286) that if the diagram, Fig. 32 (page 212), of the Report, be plotted with differences of temperature as abscissae instead of with differences of pressure, then the leakage curves became straight lines. If this was done for the full-line curve, which represented the most reliable and consistent series of the trials, the straight line passed almost exactly through the origin. The point was worth further investigation; for on temperature or temperature difference depended (1) warping of the valve, (2) condensation on valve surface, (3) the viscosity of water.

Professor Capper had not recorded in the Report what the condition of cylinder lubrication was during the trials. It was well understood that an abundance of lubricant in the cylinder diminished condensation a great deal. Messrs. Callendar and Nicolson mentioned that examination of the cylinder after sundry of their trials always showed very little trace of oil. Professor Nicolson has expressed the opinion that leakage under working conditions will be more than that measured with blocked-up steam ports. This was remarkable, because in the Paper above referred to, the values of the condensation calculated by the method of "condensation areas" agreed closely with those found from the feed and indicator diagrams, after allowing for leakage measured with blocked ports. The greatest divergence between the two sets only corresponded to 3 per cent. difference in the measured leakage, so that this close agreement was apparently a check both on the "method of condensation areas" and upon the assumption that the working leakage agreed with that measured.

Mr. EDMUND L. MORRIS, referring to his remarks on the Paper by Mr. Barr on "American Pumping Engines,"* wrote that some

* Proceedings 1905, Part I., page 40.

(Mr. Edmund L. Morris.)

further experiments had been made on No. 7 Engine at Hornsey, which might be of interest:—

- (1) Without steam in the cylinder-jackets or reheaters.
- (2) With steam in the cylinder-jackets, but not in the reheaters.
- (3) With steam in the cylinder-jackets, and also in the reheaters.

The cylinder jacketing alone gave a saving of 4·2 per cent. over the unjacketed engine, and the reheaters in addition a further saving of 2·5 per cent., making in this engine a total saving of 6·7 per cent. by the system of jacketing adopted.

Dr. J. T. NICOLSON wrote that he thought that his former colleague Professor Callendar and himself had every reason to be gratified with Professor Capper's results, which might be said to have broadly confirmed the conclusions they had reached in 1896 with regard to the leakage of slide-valves. These conclusions had been met with a good deal of scepticism, especially on the part of steam-engine makers, who were usually unwilling to believe that their valves were capable of any leakage at all, much less the very considerable quantities which actually passed them. Only a few men like Colonel Crompton and Captain Sankey had given the matter any attention so far as he (Dr. Nicolson) was aware; and had come to the conclusion that, instead of aiming at reduction of cylinder condensation or reduction of clearance volumes and small refinements of that order, the thing that wanted doing was the improvement of the valves as to their tightness when in operation; either by better design or by heating the valve faces by jacketing in a suitable manner or by superheating the steam in the steam-chest. Perhaps they would, in future, as one consequence of Professor Capper's Report, hear less about cylinder condensation, Hirn's analysis and sharp points of cut-off, and more about valve leakage and the difference between engine feed and cylinder feed.

This Report was worthy of a much more prolonged and closer study than he (the writer) had as yet been able to give to it; and what he now wished to say had reference only to one or two of the more salient features. First he would desire to congratulate Professor Capper, if he might, on the admirable way in which he

had presented his data. He thought that in this respect the Report left nothing to be desired. He only wished, however, that Professor Capper had made some use of the method of condensation areas by observing the temperature of at least his clearance surfaces. He would then have been able to deduce the amounts of valve leakage under different conditions by subtracting the estimated condensation from the measured engine feed, instead of relying upon the assumption that leakage during an actual trial was practically the same as that during a valve-leak experiment. This was an important point, and one upon which he found himself unable to

FIG. 63.—*Simple Non-Condensing Engine Trials.*
McGill College, 1894-1899.

agree with Professor Capper's conclusions. He himself had found the working rate of leak was always greater and often very much greater than the experimental rate.

In support of this statement he had prepared a diagram, Fig. 63, giving the results of a series of careful trials he had made during the years 1894-1899 at McGill University with the same engine as used by Professor Callendar and himself on their experiments, to which Professor Capper had made reference. The results he now gave were, however, for double-acting trials, whereas in their Paper in the Proceedings of the Institution of Civil Engineers the engine was run only single-acting for the reasons there given. The engine was a high-speed simple automatic cut-off engine of $10\frac{1}{2}$ inches diameter and 12 inches stroke, normal speed

(Dr. J. T. Nicolson.)

TABLE 19.

Date, December 1897. 100 lbs. per square inch. Unjacketed.

1		M _o .	J _o .	K _o .	L _o .
2	Revolutions per minute	60	100	200	300
3	Cut-off	0.4	0.4	0.4	0.4
4	Steam-pipe press. (abs.) lbs. per sq. in.	118.4	121.5	117.3	111.2
5	Corresponding temperature . F. ^o	340	342	339	335
6	Observed steam-pipe temperature F. ^o	336	339	336	331.2
7	Dryness by throttling	0.98	0.98	0.97	0.97
8	Steam-chest pressure (abs.) lbs. per sq. in.	110	114.4	108.4	104.8
9	Corresponding temperature . . .	—	—	—	—
10	Observed steam-chest temperature F. ^o	334.6	337.5	333.5	331
11	Clear surface temper. (cover end) F. ^o	321	320	321.3	322
12	" " " (crank end) F. ^o	309	309	308	308
13	Exhaust-pipe press. (abs.) lbs. per sq. in.	—	15.5	16.0	—
14	Corresponding temperature . F. ^o	—	214.7	216.3	—
15	Observed exhaust-pipe temperature F. ^o	—	214.3	215	—
16	{ Condenser pressure (abs.) = barometric pressure }	15.0	15.0	15.0	14.7
17	I.H.P.	20.1	34.9	66.6	84.6
18	Steam per I.H.P. per hour . lbs.	55.1	44.9	34.7	33.4
19	Total steam per hour . . . lbs.	1108	1536	2308	3075
20	Cushion steam per hour . . . lbs.	—	88	184	290
21	{ Cylinder feed per hour by indicator at cut-off lbs. }	—	886	1662	2364
22	Missing quantity per hour at cut-off lbs.	—	738	830	1001
23	{ Estimated condensation per hour at cut-off lbs. }	—	110	133	142
24	Estimated valve leakage per hour lbs.	—	628	697	859
25	Leakage coefficient	—	6.28	6.90	9.54
26	Estimated cylinder feed per hour .	—	919	1626	2234
27	" " " " revolution	—	0.153	0.156	0.123
28	{ Efficiency relative to Clausius cycle allowing for leak }	—	0.63	0.69	0.67
29	{ Efficiency relative to Clausius supposing no valve leak }	—	0.38	0.49	0.525

275 revolutions per minute. The trials were made at $\frac{1}{8}$, $\frac{1}{6}$, $\frac{1}{5}$, 0·3 and 0·4 cut-off, and at the three speeds, 100, 200 and 300 revolutions for each cut-off. Table 19 (page 332) gave the results, and Fig. 63 (page 331) gave three sets of curves representing them. The uppermost curves were ordinary Willans lines co-ordinating total engine feed (line 19 of Table) plotted on a base of I.H.P. (line 17). The lowest lines were valve leakages per hour, and the middle the differences.

Curves JKL referred to 0·4 cut-off,

„	GHI	„	0·3	„
„	DEF	„	0·2	„
„	ABC	„	0·125	„

and the 3 sets of points on each curve were for 100, 200 and 300 revolutions in the order named.

The leakages were obtained by finding the cylinder condensation at cut-off, by observation of wall temperatures, and the method of condensation areas, adding this to the indicated weight, and deducting this sum (which ought properly to be called the *cylinder feed*) from the whole amount used by the engine, called the *engine feed*, (allowance being, of course, made for cushion steam). It would be seen that the amount of this leak varied from 500 to 800 lbs. per hour; the steam used by the engine varying from 900 to 3,000 lbs. per hour. Upon reference to the Paper by Professor Callendar and himself, it would be found that the measured leak of the same valve, when tested with the ports plugged, and at the same pressure difference of 100 lbs. between steam-chest and exhaust pipe, was 300 lbs. per hour, so that in that instance, at all events, they had not found that the leak during actual working was the same as that during the valve leakage trial. * It was two or three times as great. Professor Callendar and he had found that the temperature of the valve face had a very marked effect on the rate of leak; * and as, from actual observation of the temperature of the metal of the steam-chest face during both leakage and load trials, they had found that in the latter case this metal was much cooler,

* Proceedings, The Institution of Civil Engineers, 1897-8, vol. cxxxi, page 235.

(Dr. J. T. Nicolson.)

(as a result, probably, of the current of wet exhaust which flowed over it); this explained why the leak was greater during actual working than during a valve leak test with the ports plugged.

He wished Professor Capper had drilled some holes in his steam-chest face and noted the temperatures during leakage trials as compared with actual running. It would have been most instructive. Professor Capper's Table 5 (page 221) of the values of the leakage-coefficient for his valve gave for the mean value the figure 0.02. This was in surprisingly close agreement with the value for the two types of valve Professor Callendar and he had tested in Montreal, namely, 0.021 for the balanced valve of the small high-speed engine, and 0.02 for a valve with Meyer expansion plates on the low-pressure cylinder 18 inches diameter of the experimental engine designed by Mr. Druitt Halpin. Another valve of that engine which had been tested for leakage gave the same figure 0.019. It really would appear as if the leakage of slide-valves of small size and simple design was subject to the very simple law which Professor Callendar and he had found; and which might be stated thus: In slide-valves whose average "overlap" (or breadth of face opposing leakage) was one inch, there would be a leak of one-fiftieth of a pound of steam per hour per pound difference of pressure between steam-chest and exhaust pipe, for every inch of perimeter round the exhaust port, or about $\frac{1}{4}$ lb. per hour for every foot. This was the law of leakage of saturated steam across a moving valve with steam-ports blocked. In actual work he had produced evidence, Fig. 63 (page 331), that the leak would be twice as great; and that half a pound per hour would leak across every foot of perimeter having an overlap 1 inch wide, for a pressure difference of 1 lb. per square inch. The perimeter of the valve in question was 6 feet, and its overlap $\frac{1}{2}$ inch, so that according to this simple rule it ought to have leaked $\frac{1}{2} \times 6 \div \frac{1}{2} = 6$ lbs. per hour per pound difference of pressure, or 600 lbs. per hour at the pressure at which these trials were made.

They would also observe from the diagram that the leakage increased considerably with the power developed, whether that was obtained by diminishing the ratio of expansion or by increasing the

speed. This seemed reasonable, as in either case more exhaust steam would pass across the valve face per unit of time, producing a greater cooling effect at large than at small powers. This result also was contrary to Professor Capper's deduction that leakage *diminished* with speed (page 215). No doubt speed would have an effect on the laminar motion of the leaking water, similar perhaps to that of journal speed on lubrication, where there was found to be a critical speed below which the journal would not carry its oil into the bearing. But such a phenomenon would, in his opinion, be obliterated by the much more powerful contrary effect due to greater cooling of the faces at higher speed.

FIG. 64.

Indicator Diagrams affected by Speed.

He had also prepared three superposed indicator cards, Fig. 64, from this engine, showing the effect of variation of speed on the shape of the card at constant cut-off. The smaller weight of cylinder feed taken in at the high speeds which was obvious from these diagrams was not accompanied by a correspondingly rapid reduction of mean effective pressure; so that notwithstanding the increase of rate of valve with leakage speed, there was on the whole a diminution of the consumption with increasing speed, to a minimum which for this engine was reached at from 250 to 300 revolutions with late and early cuts-off respectively.

Professor Capper wrote in reply to the communications as follows:—Mr. Forward (page 323) had very rightly asked if the failure of lubrication occurred in all these cases of highest difference

(Professor Capper.)

of pressure, Fig. 32 (page 212), as the figure was plotted from the mean of all these experiments. His answer was that the leakage at the point where the valve seized could not of course be measured, but all these experiments were carried out directly after each other, and just at the end the valve seized. The scoring had probably been going on throughout all these measurements, and all these would therefore be probably affected by the burring up and raising of the valve. He could not say positively that this was so, as it was only when the valve seized that the effect was finally noticed, but, as there was a good deal of trouble experienced in getting regular results for some time before the seizure, it probably was so. As to the question of blocked ports versus experiments designed as Mr. Forward and indeed other speakers had suggested, he would not now fully enter. But he was distinctly of the opinion, after the careful investigation and experiments he had made, that any arrangement such as that suggested by Mr. Forward would lead to much greater errors in measurement than would be found to exist when blocked port experiments were corrected by the help of leakage area diagrams, such as those given in Figs. 34 and 35 (pages 218 and 219), though he admitted that these diagrams were drawn on tentative assumptions only. Yet both sets were drawn on the same hypothesis, so that the ratio between them was probably very closely correct. He did not think the flow of steam in actual working was likely to affect the comparative results more than shown on those diagrams. Experiments to determine whether this was so or not would be of great value, but were extremely difficult to carry out without introducing disproportionate errors.

In reply to Mr. Mason (page 329), during the whole series of experiments the lubrication in the cylinder was that described in the leakage tests as "siphon lubricator flowing freely" in addition to the sight-feed lubrication. To Dr. Nicolson's valuable remarks he had already alluded in his verbal reply. He would only add that the trials were well under way when Messrs. Callendar and Nicolson's Paper was published, and the whole of the trials would have had to be repeated if the scheme of them were to be altered, so as to work upon the basis of condensation areas. He quite agreed that it would have

been a very valuable addition if he had taken temperature at a number of points in the cylinder, but at the time the trials were begun it was not found practicable to do so. He acknowledged the omission, but claimed that this would have so largely complicated the experiments as to make their scope and period of incubation—already too large—extend beyond reasonable dimensions. He would endeavour to run special typical trials in the future under those conditions, so as to supplement the information contained in the present research.

To Mr. Kersey's very valuable contribution (page 324) he would only say that the laws he deduced and the results obtained by him from those laws, together with his comparison of these with the actual results obtained, were instructive. Their value would be greatly enhanced, if Mr. Kersey could find time to apply them to other experiments and see how far a general law could be evolved by so doing. He cordially agreed with the suggestion made that co-ordination of experiments was what was needed at the stage at which they had now arrived.

Details of L.P. Cylinder,
showing Valves, Jackets, and Willans' Coil
in bottom cover for boiling off any deposit
water.

Clearance Pots,
2 for L.P. Cyl.

Capacity-Index
on Spirals

STEAM-ENGINE RESEARCH.

Plate 6.

Mechanical
Engineers 1905.