EXPERIMENTAL INVESTIGATION OF THE SOLID INJECTION PROCESS IN A HIGH COMPRESSION OIL ENGINE.

-oOo-

By

A. Murray Burdon, B.Sc.

-oOo-

26th., February 1930.

INDEX.

------ -oOo- ------

INTRODUCTION

-oOo-

INTRODUCTION.

The object of the experimental work described in the following paper was to investigate the process of injection of fuel oil into the engine combustion chamber, and to relate it if possible to that of combustion. The injection process was taken to consist mainly of

(a) the motion of the fuel pump plunger,

(b) the variation in the oil pressure behind the fuel valve nozzle,

(c) the motion of the fuel valve spindle,

(d) the temperature of the oil behind the nozzle,

(e) the rate of injection of fuel into the combustion chamber, and

(f) the pressure in the combustion chamber.

It was obvious that items (a), (b), (c), and (d) were directly determinable from engine tests and the necessary apparatus was accordingly designed. For item (e) however, an indirect method of determination was decided upon. Preliminary engine tests were run and measurements made of items (a),(b),(c) and (d). With a knowledge of the working limits of these items an apparatus was designed whereby the discharge through the engine fuel valve at any setting of spindle lift could be measured for various conditions of pressure, density, viscosity and temperature of the liquid being discharged, and of pressure, density and viscosity of the 'medium' into which the discharge took place. Numerous experiments on discharge (forming the first part of the thesis) were carried out with this apparatus and from the results of these, curves were obtained which were used in conjunction with items (b) (c) and (d), as well as with the cylinder or combustion chamber pressure (f)(Which directly/

directly affects, but is hardly part of the process of injection), to determine item (e), the rate of injection for any instant during the injection period.

Item (d), the temperature of the oil behind the nozzle was investigated in a separate series of engine tests and the readings thus obtained for full load conditions and normal jacket temperature determined roughly the temperature to which the fuel oil was heated during the discharge experiments.

Item (a), the stroke of the fuel pump plunger when once determined was kept constant throughout the remaining tests.

With the main object of obtaining reasonably accurate indicator diagrams of cylinder pressure, the motion of the indicator drum, which was found to be considerably affected by engine speed, was investigated.

Three series of engine tests were then run for various conditions of brake load and fuel valve spring load. For the first two series the injection process was only investigated with respect to items (b) (c) and (f). The third series was carried rather further in that not only was item (e) investigated but sufficient readings were taken to enable a complete $T\phi$ analysis of the indicator cards to be made. For this analysis, readings (apart from the usual test readings) were taken of air consumption and by thermocouple, exhaust temperature. The air flow was measured by means of air boxes and an orifice or throttle plate, and as a check on the accuracy of this method of measurement, a series of earlier experiments are given showing the results obtained by the air boxes compared with direct measurement by means of a gasometer.

By/

By means of the $T\phi$ analysis already mentioned and a knowledge of the heat loss during the working stroke an attempt was made to determine the true rate of combustion of the fuel. In this last determination an approximate allowance was made for the lag of the cylinder pressure indicator and a fairly close estimate of the 'ignition lag' thus obtained.

The final part of the thesis describes tests carried out in order to determine roughly to what extent the presence of inert gases within the combustion chamber might be responsible for the lags in ignition and combustion indicated by the preceding experiments. As a supplement to this last section some results are given of increase of ignition lag produced by throttling the air flow to the engine.

It seems worthy of mention here, that before the experiments described in the following pages were begun, numerous preliminary tests were carried out on the engine with the object of obtaining a knowledge of the general character of the high compression oil engine cycle.

From various approximate methods of indicator diagram analysis it appeared that the combustion of the fuel practically occupied the total period of the working stroke, and the consumption of fuel was excessive. This was thought to be possibly due to an unnecessarily long period of injection. At about full brake load the overflow through the byepass valve of the fuel pump was found to be 56% of the flow through the fuel valve nozzle, and for this reason a method of reducing the injection period was employed first by steepening the face of the pump cam toepiece and then reducing its effective height.

Tests/

Tests were run with these variations of toepiece and as the effective height was diminished, improvement in both consumption and exhaust temperature was observed. With the lowest toepiece the consumption was most favourable but the running rather erratic owing to the flow through the byepass valve being too small for good governing.

With a view to checking the air flow to the engine as measured by means of air boxes and an orifice or throttle plate, and ascertaining whether combustion was complete, a considerable amount of exhaust sampling was done. The samples were collected over acidulated water and analysed either by means of a burette, or occasionally by a Bone and Wheeler apparatus. Considerable experience of this apparatus was acquired in connection with the complete analysis of coal gas, as it was then the intention to use it later for samples taken of cylinder contents during combustion. The analysis, by means of the burette, of the gas collected over acidulated water showed principally that, for a sample allowed to stand for some time before analysis, although the oxygen and nitrogen contents remained constant, the carbon dioxide content rapidly diminished. In no sample, even when immediately analysed, was the CO_2 content found to be quite as much as desired. This might have been due to the presence of traces of CO, CH_4(Methane) and other hydrocarbons, which the simple burette method of analysis, unlike the Bone and Wheeler, was unable to detect. Owing to this uncertainty of the CO_2 content it was found that much better results, which checked very closely with the air box 'air consumption' readings, were obtainable by calculating the 'air to fuel' ratio on a basis of the O_2 content of the exhaust gas analysis instead of the CO_2 content, as was first attempted.

- - - - - - - - - - - -

DESCRIPTION OF ENGINE.

-oOo-

DESCRIPTION OF ENGINE.

The engine used in the experiments was a horizontal, high compression oil engine of the automatic solid injection type as manufactured by Messrs Ruston and Hornsby Ltd. Under full load, and at a normal speed of 300 R.P.M. it developes about 14 B.H.P. It relies entirely on the heat of compression for its ignition and starts readily from cold by means of compressed air.

The load is applied to the engine by means of an ordinary rope brake fitted on a special water cooled drum on one end of the crankshaft.

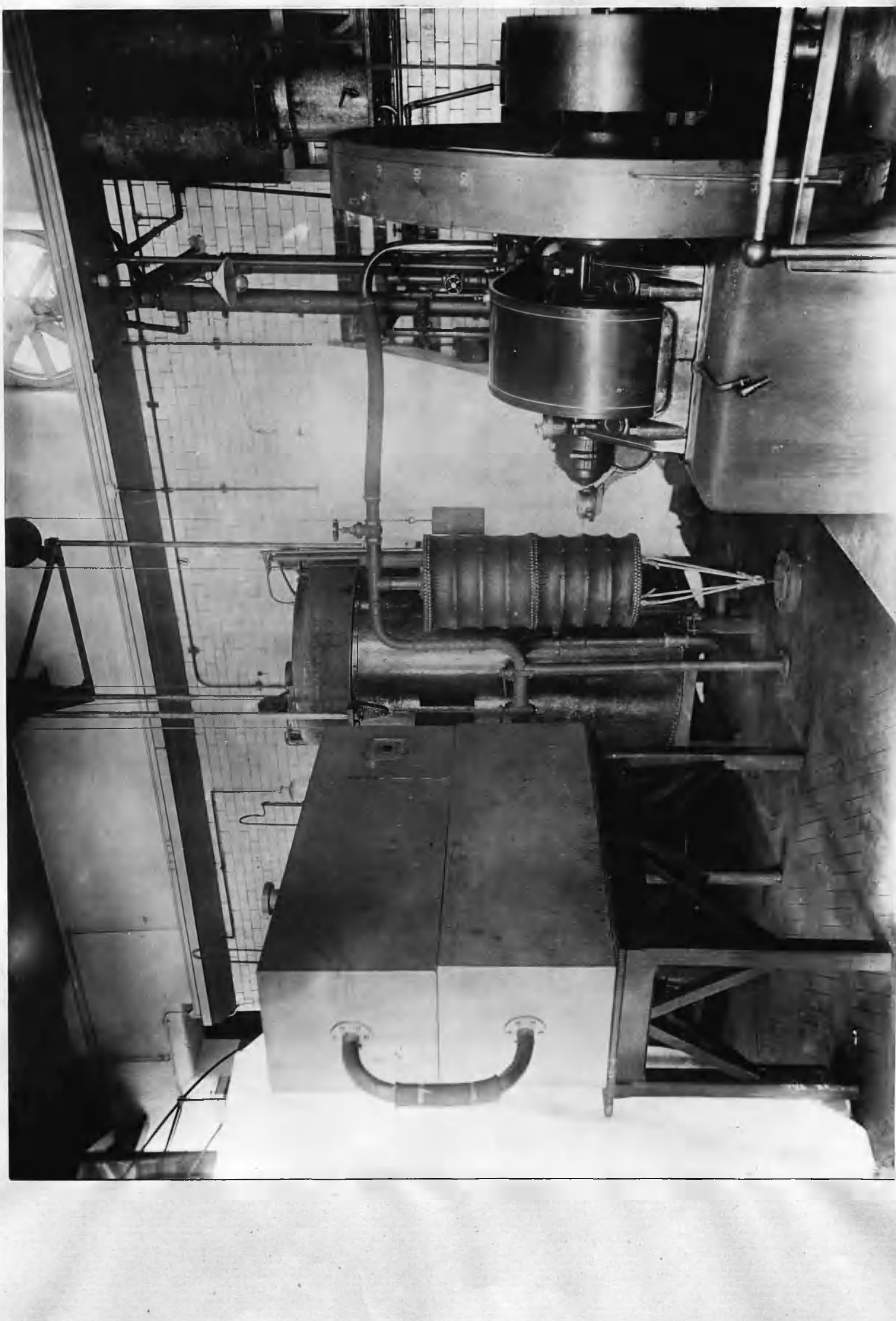
Fig. 1, p.6 gives a general view of the engine with its normal experimental apparatus. Fig.2,p.7 gives a view of the back of the engine showing particularly the positions of the fuel valve (with experimental apparatus) and the engine indicator.

Leading Dimensions of Engine:-

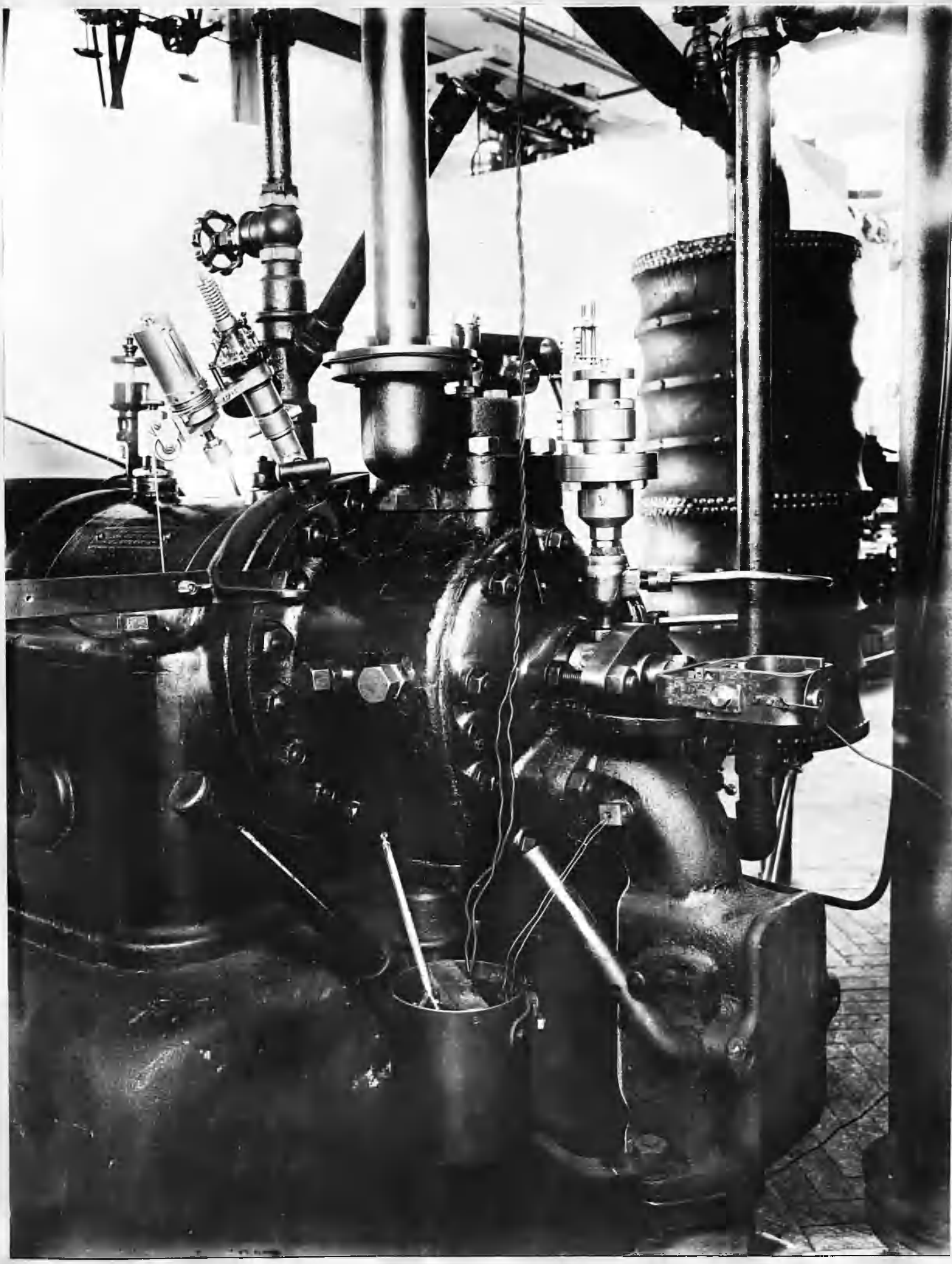
Dia. of Cylinder	$= 7\frac{1}{2}$"
Stroke	= 14"
Stroke Volume	= 0.35793 ft^3.
Clearance Volume	= 0.02816 ft^3.
Length of Connecting Rod	= 31.5".

Cylinder Head.

Figs. 3 and 4, pp. 8 and 9 , show the general form of the Cylinder Head incorporating the Combustion Chamber. In these sketches, for simplicity, only the essential features are given. Fig.3(a) is a section taken vertically along the centre line of the engine. It shows the piston at the inner centre position. The air inlet valve above and the exhaust valve below are shown in their closed positions, and also at the opposite end of the combustion chamber from the piston is indicated the position of/



OIL ENGINE AND NORMAL EXPERIMENTAL APPARATUS. FIG. 1.



Back View of Engine.

FIG. 2.

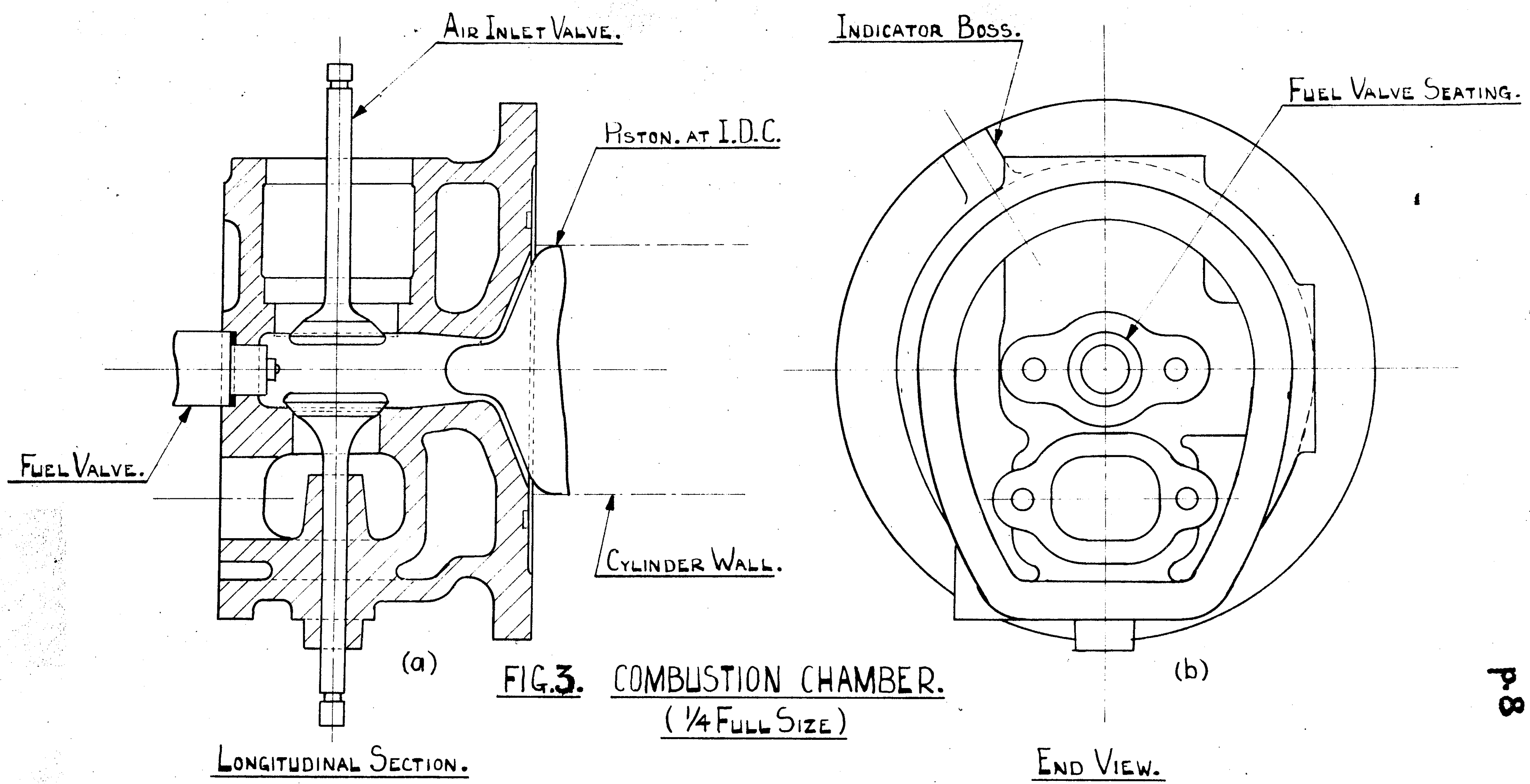
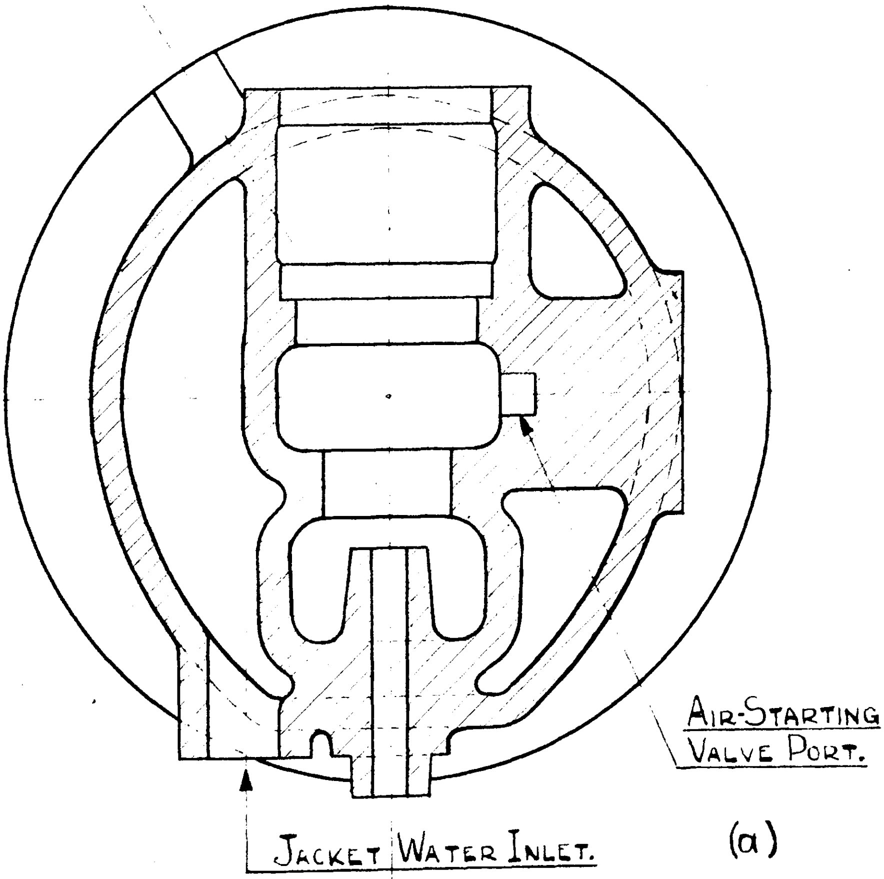
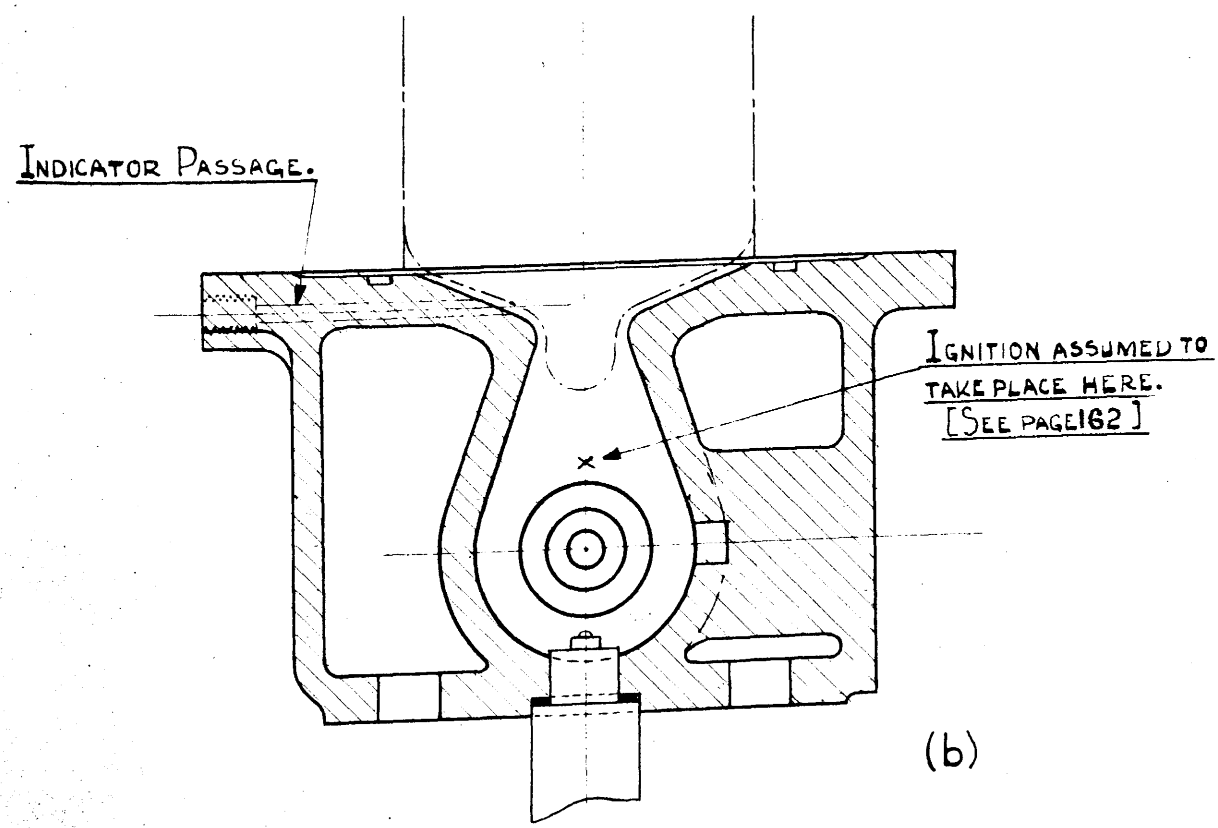


FIG.3. COMBUSTION CHAMBER.
(1/4 Full Size)



(a)

Transverse Section.



(b)

Sectional Plan.

FIG. 4 COMBUSTION CHAMBER.

(1/4 Full Size)

of the fuel valve. Bolted to the end of the cylinder head, but not shown in the sketch, is a circular end cover forming the outer end wall of the water jacket.

Fig.3(b) depicts an end view of the cylinder head with the end cover removed. The housings for the inlet, exhaust and air-starting valves are clearly shown in this view.

Fig.4(b) gives an impression of the shape of the combustion chamber into which the fan shaped spray of fuel is projected from the rectangular orifice of the fuel valve nozzle. The position also of the passage leading to the ordinary indicator is shown dotted.

In Fig.4(a) is a cross section of the head taken vertically through the centre line of the inlet, exhaust and air starting valves. Since this starting valve is of no direct interest to the present work, it was not withdrawn at any time and has not therefore been shown in section in any of the figures.

Fuel Pump.

The relative positions of the fuel pump and its operating cam are shown in Fig.5, p.11 and a horizontal section of the pump itself is shown in Fig.6, p.12 . The pump is brought into operation by turning the control handle into the position shown in Fig.6, thus allowing the roller to come into contact with the fuel pump cam, (see Fig.5). To cut off the fuel supply this control handle must be turned through approximately 180 degrees in the direction of the arrow, and the cam-shaped portion of the control handle spindle throws the roller clear of the fuel pump cam. The pump cylinder is in direct communication, except for a non-return ball valve, with the atomiser or fuel valve of the engine, and the timing of the pump cam, other things remaining constant, directly controls/

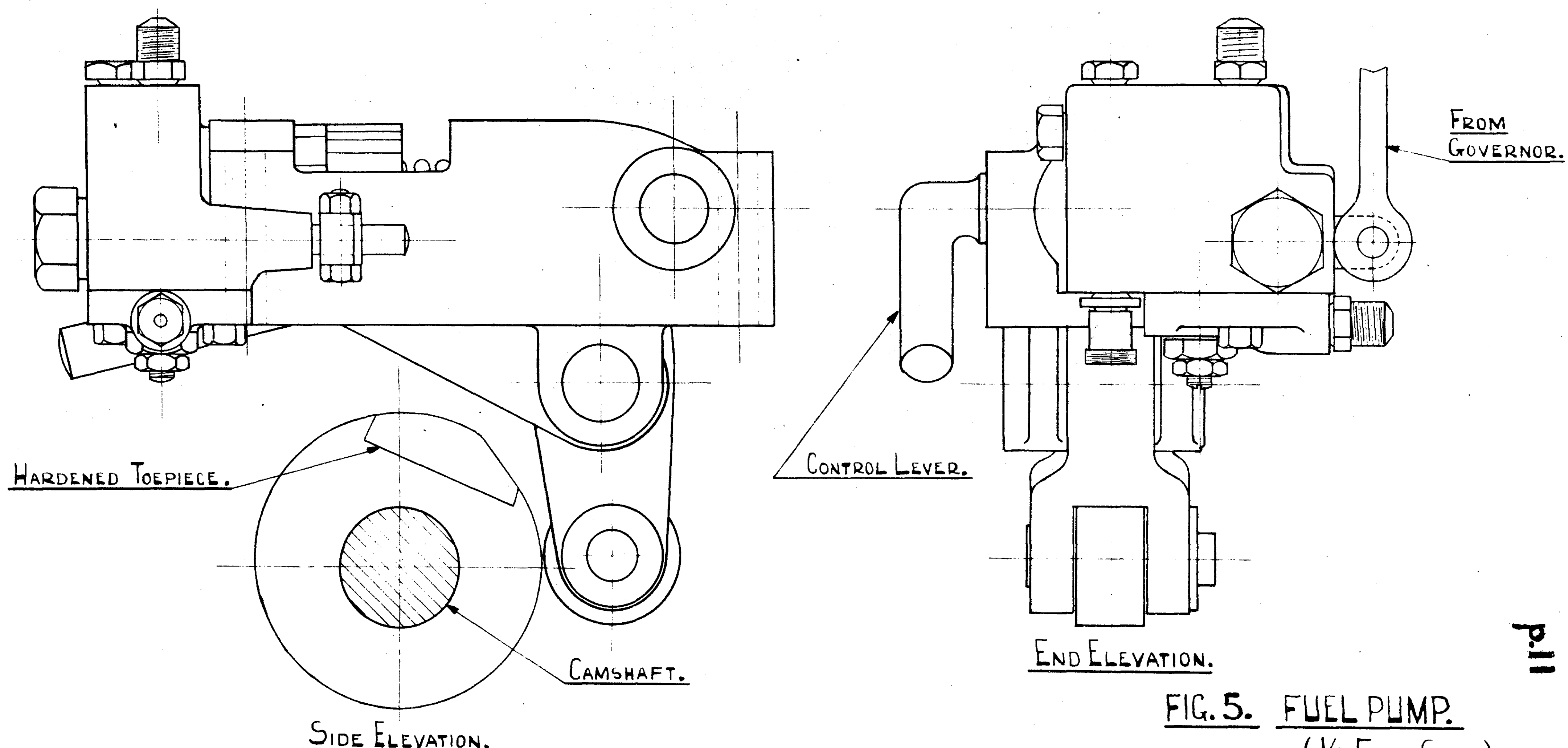


FIG. 5. FUEL PUMP.
(½ Full Size)

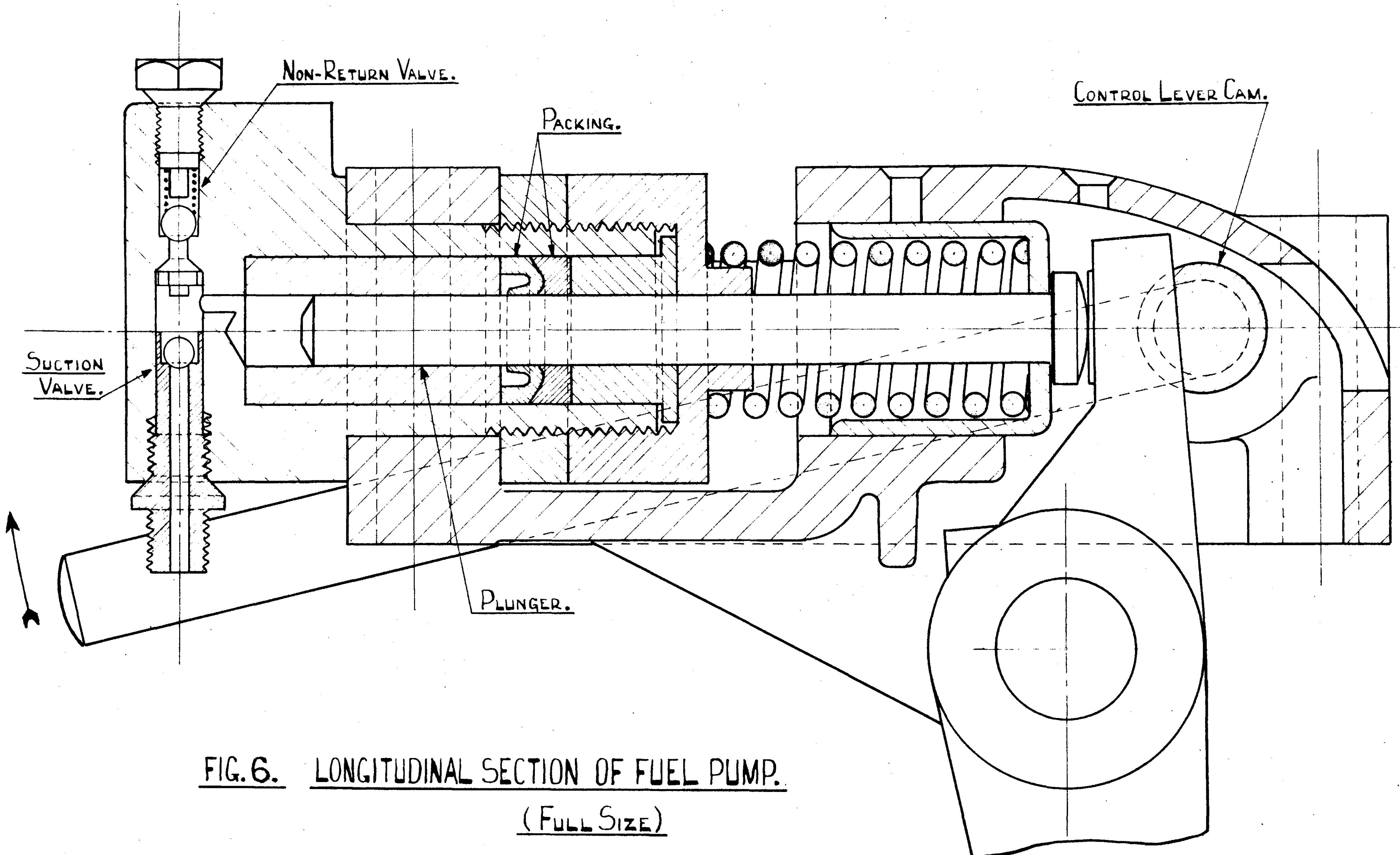


FIG. 6. LONGITUDINAL SECTION OF FUEL PUMP.

(Full Size)

controls the instant when injection of fuel into the combustion chamber commences.

The engine is governed by a spring loaded centrifugal governor which acts on a bye-pass valve, within the fuel pump. through a system of levers, the details of which are shown in Fig.7, p.14. The governor directly operates the cam 1. A small steel beam 2. is supported at one end by this cam, and at the other by a conically pointed screw 3. which can be adjusted if desired while the engine is running. Resting on the beam at the centre is a small push rod 4. on the top of which rests the bye-pass ball valve 5. This ball is held on to the top of the push rod by a light coil spring. The suction valve 6. is free, but the non-return valve 7. is (like 5) held down by a light coil spring . The small circle 8. represents the passage communicating with the pump cylinder. The connection for the delivery pipe leading to the fuel valve is situated immediately above the bye-pass valve.

While the engine is starting up, the cam 1. is in such a position that the push rod is allowed to fall clear of the bye-pass ball valve, and the maximum volume of fuel oil is injected directly into the combustion chamber. As the speed of the engine increases the governor rotates the cam and at a certain speed the ball valve is lifted from its seat by the push rod. The engine speed is kept constant by the slight influence of the governor on the amount of lift given to this ball valve, and can be varied within a limited range by means of the adjustable point 3. or alternatively by means of a convenient adjustment of the length of the rod connecting the governor to the arm at the end of the cam spindle as shown in Fig.(5) p.11. The overflow of oil through the bye-pass valve is forced back to the supply tank of the engine through a pipe leading from/

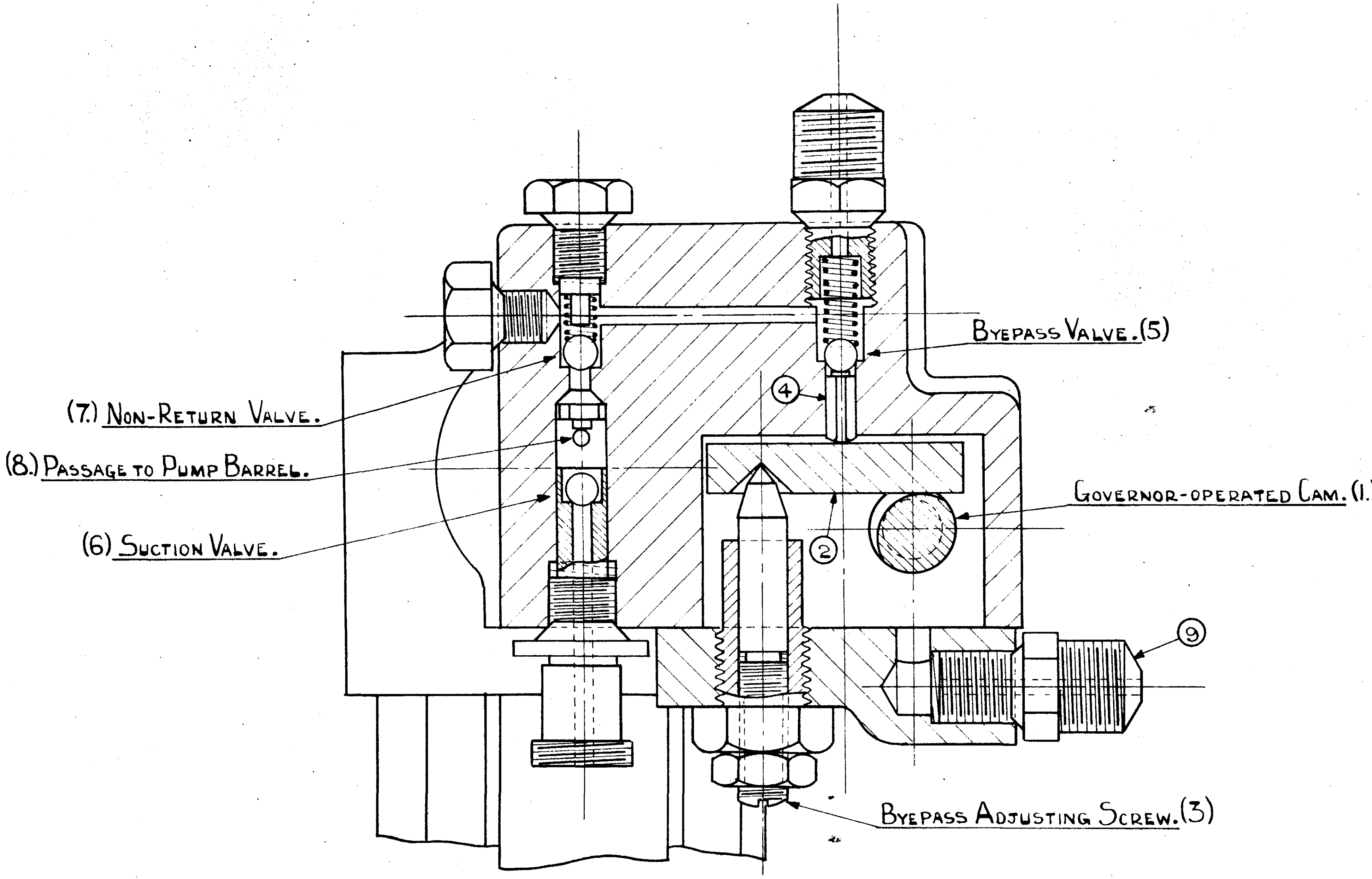


FIG. 7. BYE-PASS VALVE MECHANISM. (Full Size.)

from the connection 9.

Fuel Valve.

A longitudinal section of the fuel valve or atomiser is shown in Fig.8, p.16 . Under correct working conditions the entire space within the fuel system, namely the fuel valve, fuel pump and the connecting pipes between these and from the supply tank, should always be full of oil. During the injection the fuel valve is in direct communication with the pump, this communication being cut off during the remaining period by the non-return valve. (see Fig.7, p.14).

On the forward stroke of the pump plunger, the pressure in the body of the valve rises rapidly, and as this pressure, acting on the area of the valve spindle (minus the area of the valve seat) overcomes the load due to the valve spring, the spindle lifts off its seat and injection of fuel into the combustion chamber commences. As the fuel pressure falls, the spring forces the spindle back on to its seat and injection is cut off. The small gap, shown on the drawing, between the back end of the spindle and the corresponding end of the spindle barrel, represents fairly closely the total lift of the valve. By actual micrometer measurement this lift equals .0256".

A thermocouple and gland arrangement, as shown in Fig.13, p.27, was also used under running conditions on the normal fuel valve as depicted in Fig.8, p.16.

The extension to the valve spindle, shown in Fig.8, p.16, was used along with the spindle motion recorder described on pp.86-89.

- - - - - - - - - - - - - -

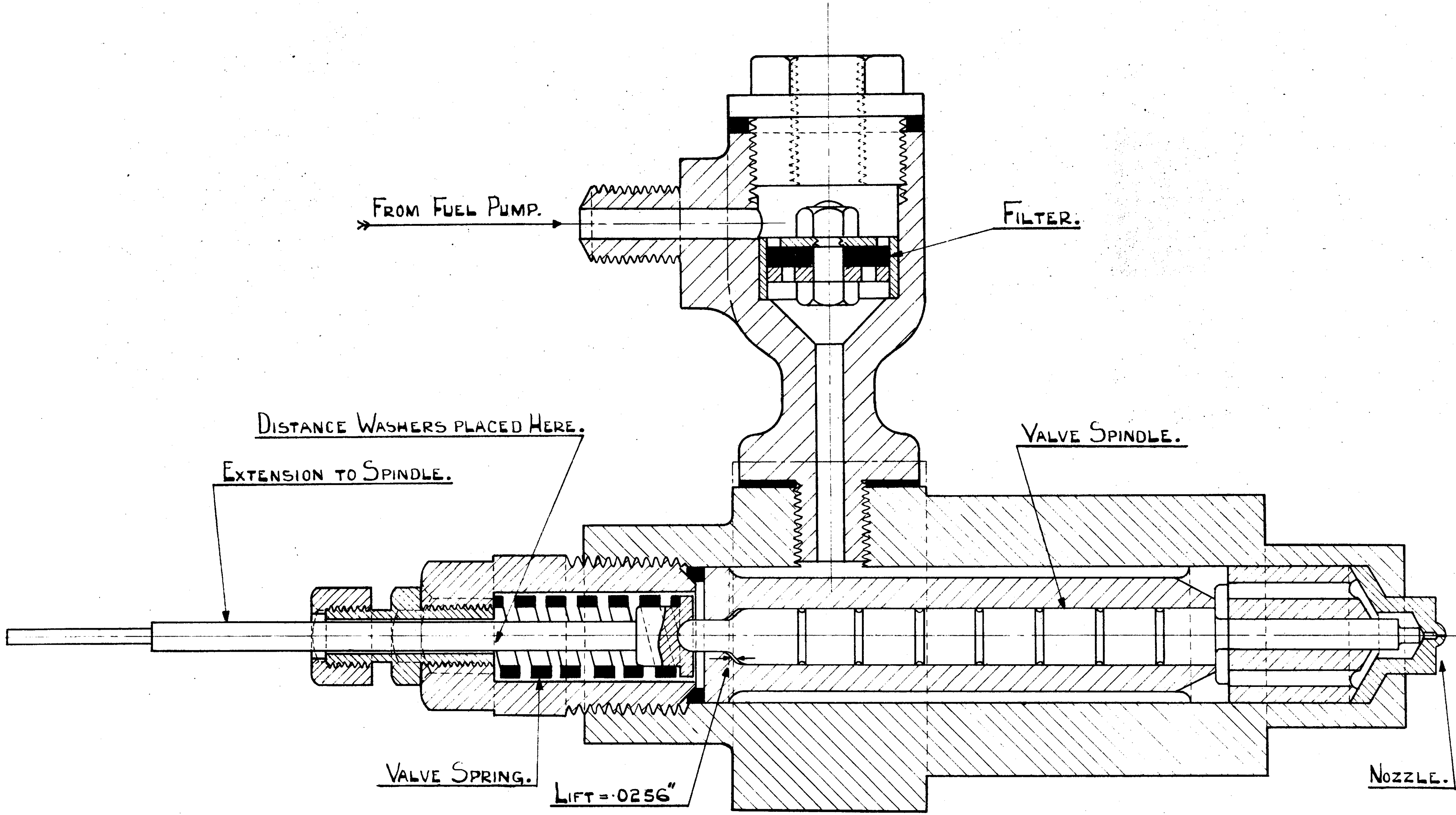


FIG.8 FUEL VALVE. (Full Size)

FUEL VALVE NOZZLE.

In Figs. 9 and 10, (pp. 18, 19.) are given the results of an investigation made in order to determine with reasonable accuracy the internal formation of the fuel valve nozzle. For this purpose a number of wax impressions were taken of the interior of the nozzle from the valve seat to the orifice. Various plastic substances were tried for this purpose but common candle wax was found to give the best results. A glance at the diminutive overall dimensions of the nozzle and orifice will suffice to show that some care and practice were required before a satisfactory impression could be obtained. The impression having apparently the true formation of the inside of the orifice was used to obtain the shape and dimensions of the orifice, and that with the apparently true formation of the valve seat was similarly used in relation to the valve seat. The dimensions of these impressions were obtained by a special form of microscope for use on an indexing machine. This microscope was moveable by means of accurately indexed micrometer screws, in two directions mutually at right angles. This enabled the coordinates of points on the contour of the wax impressions, and relative to a fixed pair of axes, to be obtained and plotted graphically on a large scale. With this instrument also were obtained the dimensions (<u>length</u> and <u>width</u>) of the projected rectangular slot of the orifice and those of the faces of the original fuel valve spindle and of the spindle made specially for the discharge experiments described later.

It was noticed that under the microscope the outside edges of the orifice were not uniformly sharp. Some parts had the appearance of having been chipped off during the process/

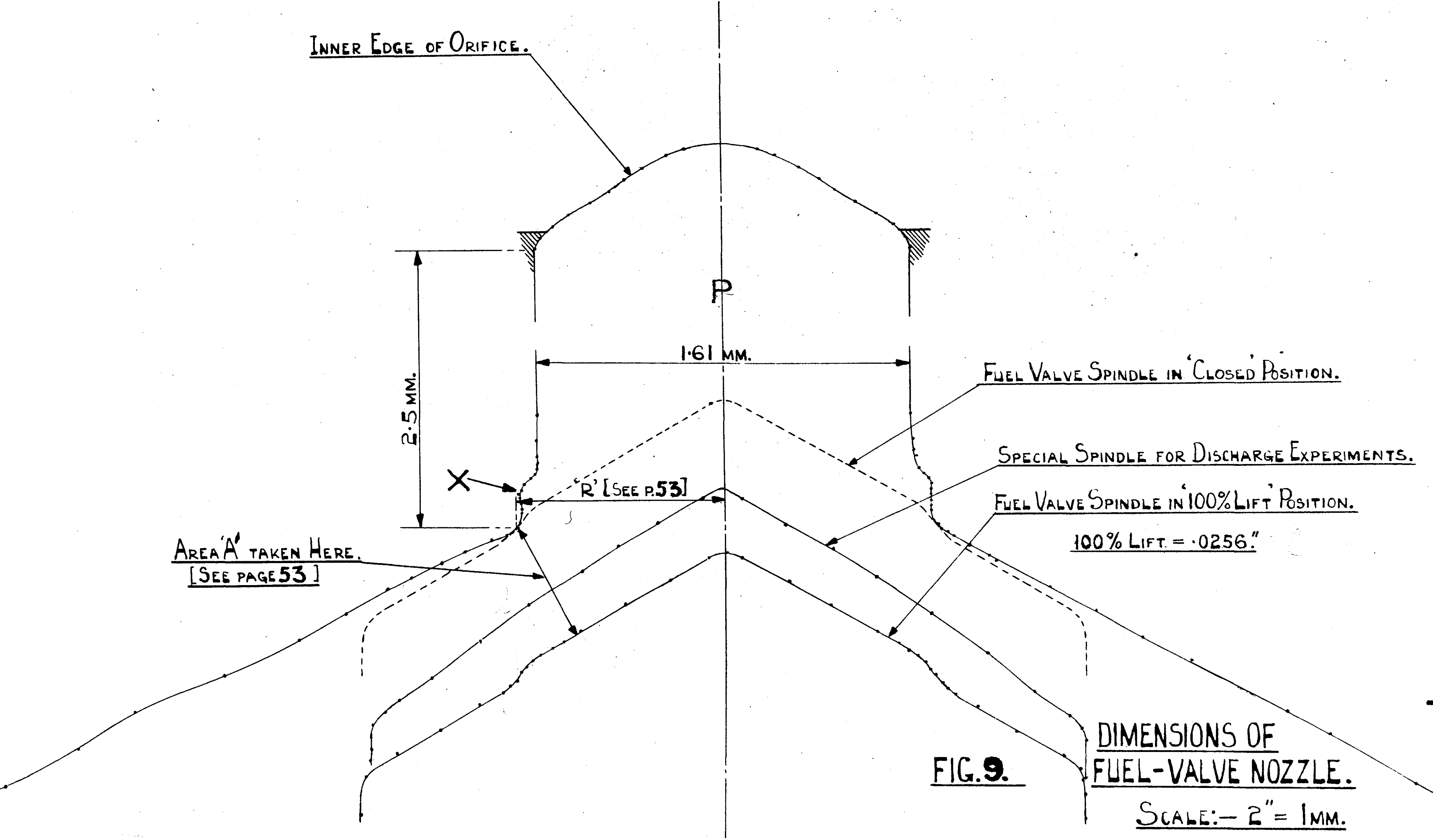


FIG. 9. DIMENSIONS OF FUEL-VALVE NOZZLE.

Scale:— 2" = 1mm.

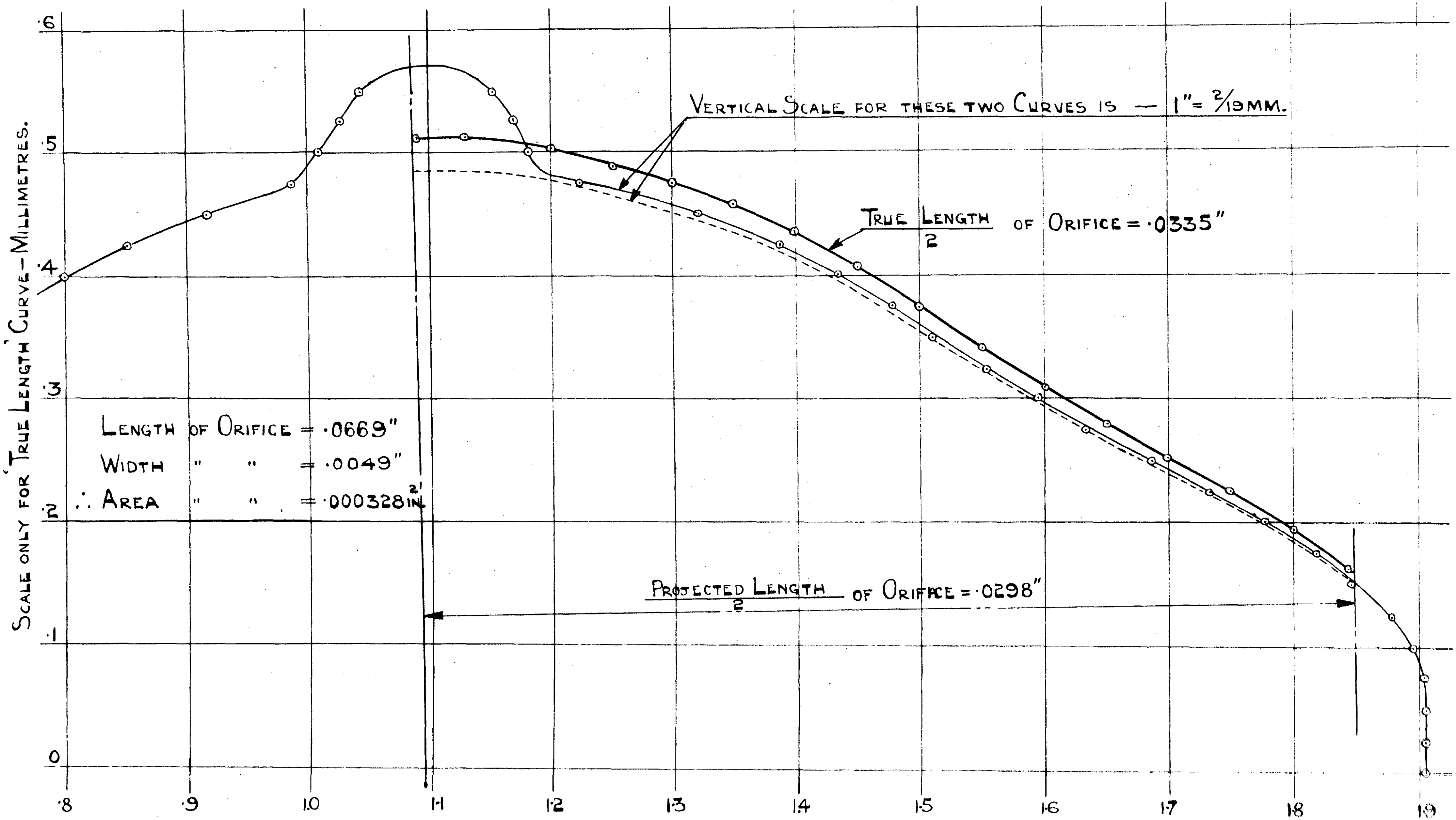


FIG. 10. FUEL-VALVE ORIFICE.

process of machining. It was found to be practically impossible to examine the state of the inside edges.

The 'length' of the orifice in the direction of flow was not determined, but was thought to be small at the middle and to increase towards each end of the rectangular slot. Except where otherwise specified, the word <u>length</u> is used to refer to the larger dimension of the orifice at right angles to the flow, and the word <u>width</u> to the smaller dimension at right angles to the length as defined.

During the course of the experiments on discharge, a steady increase in the readings under any given set of conditions was observed. The increase must either have been caused by an enlargement of the orifice or a rounding of the inside or outside edges. Possible alternative explanations for these are stated below. Frequently during the experiments the orifice became partially choked, and it was necessary, in order to avoid dismantling the complete fuel valve, to dislodge the foreign particle by means of a specially cut piece of steel. This operation repeated a number of times, although carried out with care, may possibly have slightly increased the effective discharge area. It is possible however that a more probable cause was the continuous abrasive action of minute particles of dust etc. present in the oil, especially as it is the inside edge and therefore the one least likely to be so affected by the cleaning process which principally determines the flow through the orifice. It must be clearly understood that the flow of liquid through the nozzle in these discharge tests was abnormal, as the amount of oil passed through the orifice in a given time was on the average about 30 times greater than that taking place on the engine under full load conditions.

Fig./

Fig. 10 shows the true length of the inner edge of the orifice. This will apparently be the effective length of the latter, on which a coefficient of discharge might be based. In this case clearly a determination of the coefficient cannot be made with any high degree of accuracy on account of the doubtful state of the inside of the orifice. Of the two other curves in Fig. 10, the full line represents the actual experimental readings and the dotted line merely a construction taking into account the fact that the inner edge did not lie exactly along the crest of the 'dome' shape but was displaced laterally by half the width of the orifice. The true length was obtained from this curve by redrawing it to equal vertical and horizontal scales. The boss-shaped portion on the middle of the 'dome' of the original curve was neglected.

Fig. 9, p. 18 shows definitely the effect on the valve face of the original spindle of the continued hammering on the valve seat. The fact that the 'step' on the face of the spindle does not mesh closely with the corresponding part of the seat might be due to various causes, such as for example the valve spindle and nozzle taking up slightly altered relative positions on reassembly; the wearing effect of an infinite number of injections or; inaccuracy of the wax impressions. None of these explanations, however, account for the very decided 'step' in the valve seat marked X in Fig. 9. It seems probable that this 'step' was formed in the course of manufacture.

The 'stepping' of the original valve face marks the only distinct difference between it and the face of the spindle used in the discharge experiments which will now be described.

- - - - - - - - - - - - -

DISCHARGE OF LIQUIDS THROUGH FUEL VALVE NOZZLE.

-oOo-

APPARATUS FOR DISCHARGE EXPERIMENTS.

Various arrangements of apparatus were used during the course of work on the discharge of liquids through the fuel valve nozzle. At first, only the discharge of water into air was investigated for various settings of spindle lift, and for these experiments the necessary apparatus used was of a simple character.

The valve spring (see Fig. 8, p. 16) was removed and an extension piece (similar to that shown in Fig. 8, p. 16) fitted, which had its outer end conically pointed to bear on the end of an adjusting screw recessed to suit. This adjusting screw (26 thds./inch) had fitted on the other end a circular indexed disc against the scale of which a stationary pointer was fixed. The fuel value and spindle adjusting apparatus were held vertically downwards on a projecting bracket over a receptacle. A pipe was led to the fuel valve from a convenient stop valve on a main from a hydraulic accumulator normally used for operating certain testing machines. A length of light copper tube, to the bottom of which were soldered sheets of wire gauze, was fitted over the end of the fuel valve in order to break the force of the spray and to direct it into the container below. The water discharged in a given time was weighed by means of a calibrated spring balance. Pressures were measured on a calibrated gauge screwed into the top of the fuel valve filter body.

In order that the discharge of liquids other than water might be investigated, two additions were made to the original apparatus. One of these consisted of a tall vertical cylinder (see Fig. 11, p. 25) made from a length of steam piping, to each end of which were screwed and welded heavy steel flanges. To these flanges were bolted

circular/

circular covers fitted with the necessary valves and pipe connections. The other addition was an electric heating coil which was wound round the pipe leading to the fuel valve. Both the cylinder and the heating coil formed part of the complete apparatus, by means of which all the results were finally obtained, and a more detailed description of them is given in another section. With the apparatus at this stage, readings of discharge were obtained by measuring the time taken for the flow of a definite weight of liquid. For this purpose an ordinary beam balance, with one slight modification, was found to be satisfactory. This modification consisted of the addition of a pair of long wire pointers attached, one to each tray of the balance, and formed so that they stood close together in a vertical position. The wires were bent over for a short distance at the top towards one another, and in such a manner that, with the beam of the balance horizontal, these ends were roughly at the same level. The object of these pointers was merely to indicate one definite attitude of the beam. During each test, when steady conditions of discharge into the receptacle were attained, as these pointers passed one another the stop watch was started, a known weight was added to the balance, and the time reading taken till the instant when the pointers again passed one another. Reasonably good results were obtained with this apparatus for 100% lift only of the valve spindle, and it was discovered that the fluid pressure acting on the end of the valve spindle (over the combined length of the valve spindle and the extension to the nut of the adjusting screw) caused so much compression on this total length as to upset all the other readings obtained. An attempt was made to allow for this/

this error by determining the correct zero reading on the adjusting screw scale for each hydraulic pressure employed. But even with this correction the results were not satisfactory, and, as it was the intention to investigate later the subject of the discharge of liquids into compressed air (a step which would involve a considerable alteration of the existing apparatus) the investigation was temporarily postponed.

The complete apparatus as finally used, is shown diagrammatically in Fig. 11, p. 25. By means of this arrangement all the readings that had already been obtained were repeated and an additional series of experiments carried out on the discharge of liquids into media, other than air at atmospheric pressure. The complete apparatus consisted of, fuel valve, spray chamber, oil cylinder, heating coil, air compressor and the various valves etc. as described in detail hereafter. A general view of the apparatus (except for the compressor) is shown in Fig. 12 p. 26.

__Fuel Valve__.

The original internal parts of the fuel valve shown complete in Fig. 8 p. 16 were replaced by those shown in Fig. 13 p. 27. The form of the valve face on the new spindle was made, as nearly as possible, an exact copy of the original. (See Fig. 9, p. 18). By means of the arrangement shown in Fig. 13, the possibility of error due to elasticity of the adjusting mechanism was reduced to a negligible amount. Fig. 9 shows that the slight difference in the sectional profiles of the valve faces may introduce a slight error, but this also has been assumed to be negligible. Normally the fuel oil on its way to the fuel valve passed through a very fine mesh filter screwed into the body of the valve (see Fig. 8, p. 16) on/

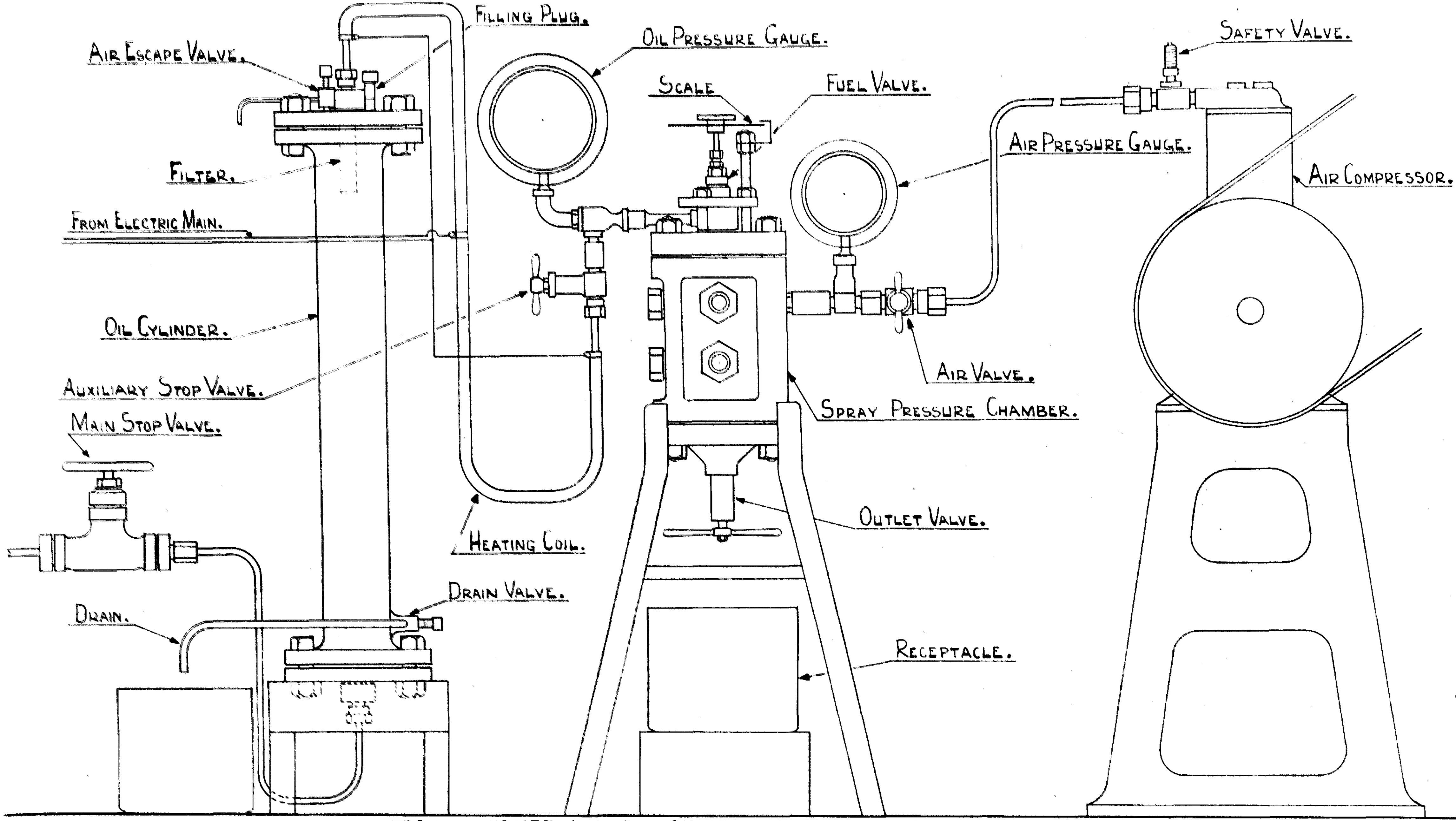
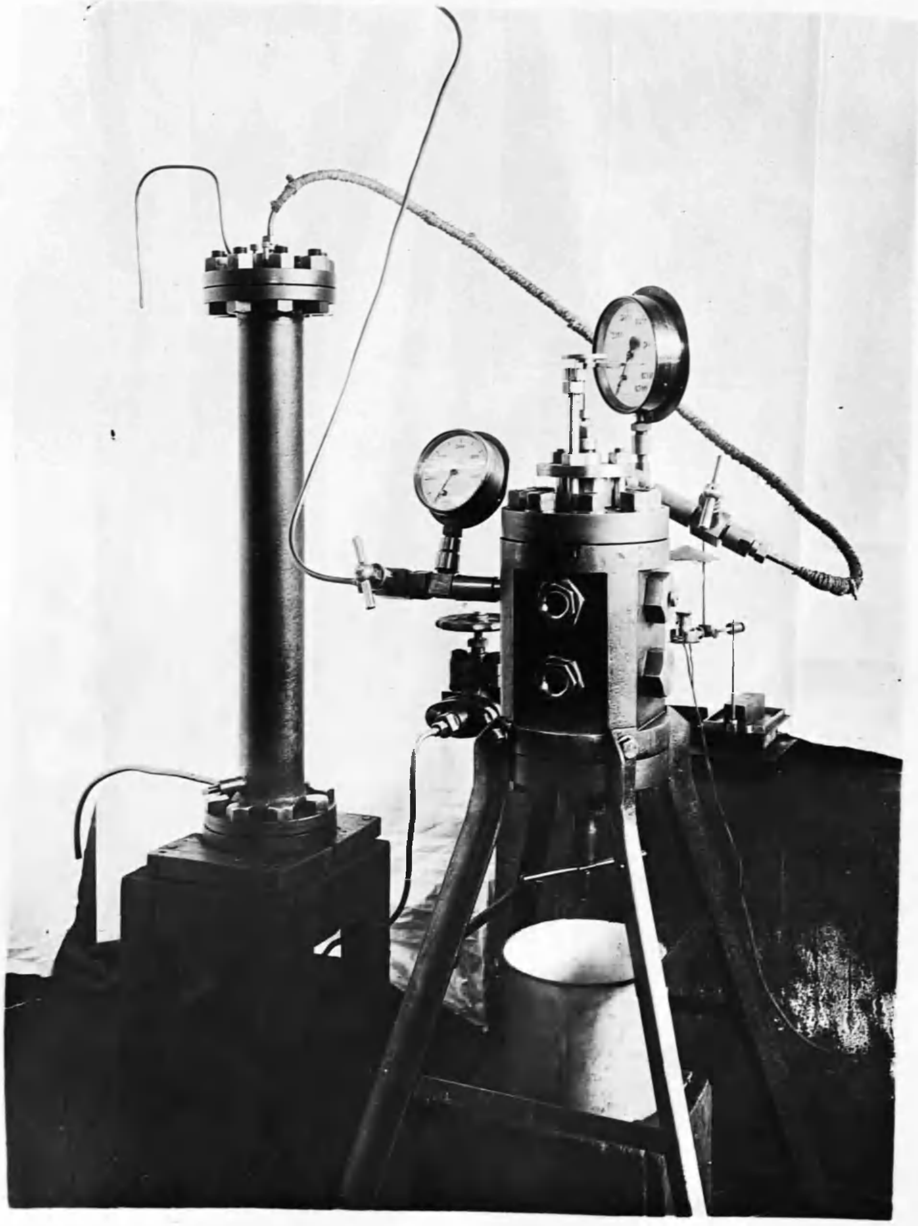


FIG.11. GENERAL ARRANGEMENT OF DISCHARGE APPARATUS. (1/8 FULL SIZE)



DISCHARGE APPARATUS.

FIG.12.

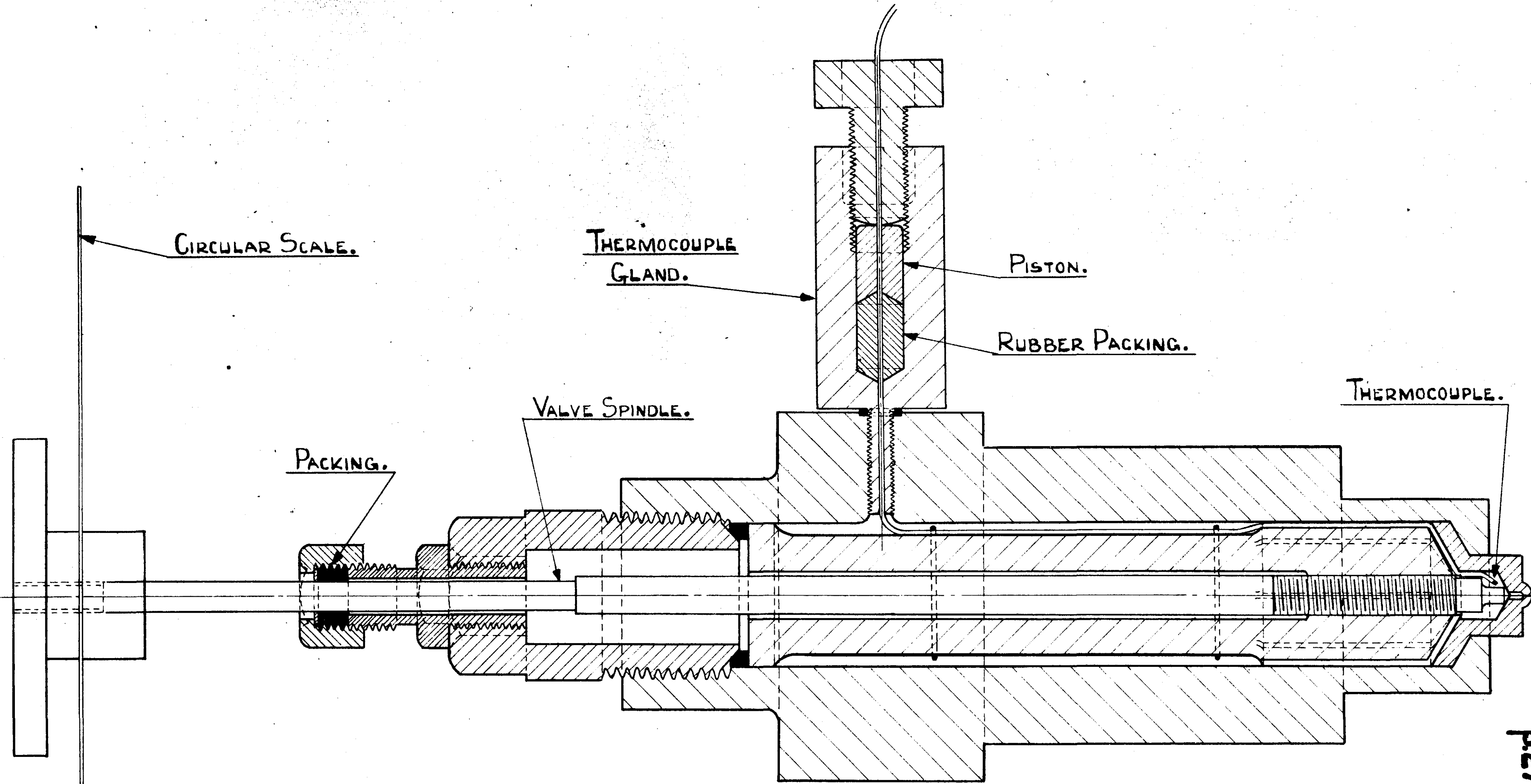


FIG. 13. FUEL-VALVE-SPINDLE LIFT ADJUSTMENT, AND THERMOCOUPLE. (FULL SIZE)

On the supply side of this filter was situated a pressure gauge for measuring the pressure in the liquid behind the nozzle; and in order that the pressure at the nozzle should be with certainty the same as at the pressure gauge, the gauzes were removed from this filter. It was afterwards found that these gauzes had no appreciable effect on the discharge and might profitably have been retained. For the experiments on the discharge of heated fuel oil, a thermocouple was inserted as shown in Fig. 13, p.27.

Spray Chamber.

Fig.14,p.29 shows a section of the apparatus by means of which the discharge of liquids into various media was investigated. The principal part of this chamber, the cylindrical wall, was made for both economical and practical reasons from a length of solid steel shafting. Two flats were machined on opposite sides of the body to accommodate the four windows shown, and a third flat at right angles to these to accommodate two large diameter plugged holes. In the remaining and curved side of the body, a hole was drilled and tapped to suit a connection for the pipe leading from the air compressor. The end covers were of cast iron and were held down on the machined ends of the cylindrical body by studs and nuts. The two joints were each made by means of a ring of round copper, softened and fitting neatly just inside the row of studs. Into the top cover was clamped the fuel valve. With the valve in position, the arrangement was such that the nozzle lay on a line between the two opposite top windows and in the same plane as the top plug. The idea here was, that with a bright light placed against one of the windows, the jet issuing from the/

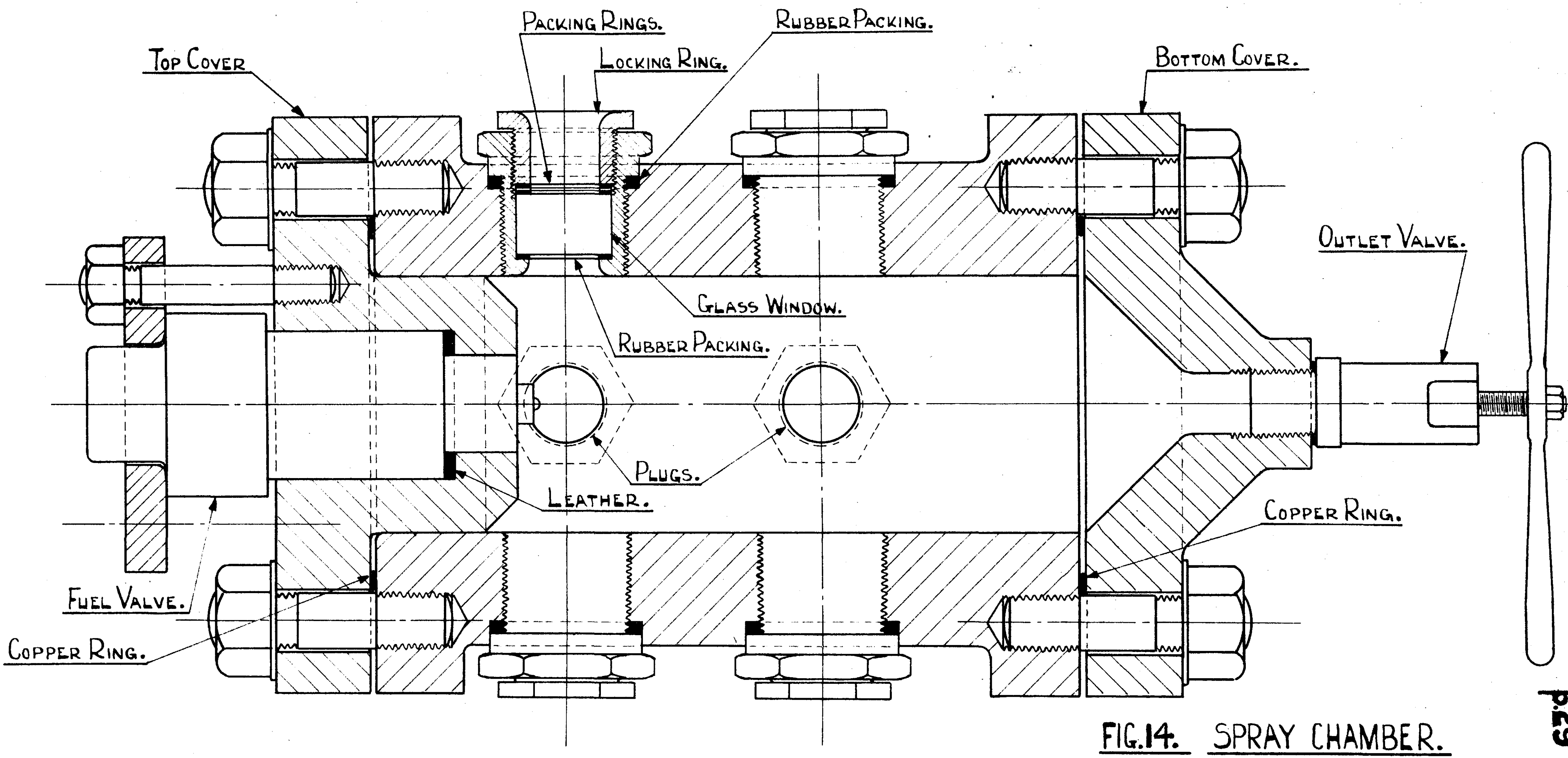


FIG. 14. SPRAY CHAMBER.
(½ FULL SIZE)

the nozzle could be examined through the other window, and if any foreign particle might become lodged in the orifice this would be noticed and removed by means of a specially shaped implement inserted through the hole at right angles when the plug had been removed. The windows were further intended to serve as a means of examining the jet when discharging into compressed air. Experience has suggested that it might have been preferable to place the windows so that the light was thrown on to the jet at right angles to the line of vision instead of directly against it.

These windows, if kept firmly screwed up, were apparently perfectly secure at 1500 lbs/in^2. The rubber or fibre packing,however, appeared to be compressed gradually, and if the locking ring was not occasionally tightened up,the rubber packing blew out and the glass split. Only twice did this happen, and on neither occasion did the window actually blow out.

To provide against such an occurrence a sheet steel guard was placed over the inspection windows during work with high pressure air. This shield had a narrow vertical slot over which was bolted a plate with a relatively small hole in the centre of it and in the line of the slot. On to the outside of this smaller plate was clamped a spare window in a machined brass housing. In the centre of the back of this brass piece was drilled a hole just sufficiently large to enable a clear view of the jet to be obtained. The plate carrying these fittings could be moved up or down in the slot in the guard and clamped so that the inspection hole came into line with either row of windows in the chamber. The bottom of the latter was shaped in the form of a hollow cone and in the centre of it was screwed/

screwed the outlet valve. This valve originally took the form of a weight-loaded relief valve, but it was found to be unsatisfactory and was replaced by the hand-operated valve shown in the sketch. A sheet tin shield was fitted round the body of this valve to divert downwards the streams of liquid issuing from the outlet passages.

Oil Cylinder.

The object of this cylinder (see Fig. 11, p. 25) was to make use of the water pressure from the accumulator as a means of maintaining steadily during discharge any desired pressure in the liquid under investigation (within the pressure limits of the accumulator). It was made from a length of 4" bore steam pipe and closed at either end by thick circular plates bolted to screwed and welded end flanges. (See Fig. 12). The joints between the end plates and the flanges were made with rings of round softened copper. The cylinder was arranged to sit vertically on a wooden stool. Through the centre of each cover plate was screwed a plug threaded at the outer end to take a pipe connection. To the lower of these was attached the pipe leading from the accumulator via the main stop valve. To the upper plug was attached the pipe leading to the fuel valve by way of a second stop valve. In the upper cover were also located a filling plug and a small screw-down valve. A drain valve was provided at the foot of the oil cylinder.

Oil was passed into the cylinder at the top by means of a hand pump and a long rubber connection with a specially designed end-coupling which could be screwed tightly into the filling hole when the plug had been removed. With the drain valve open, the oil could be forced/

forced in at the top of the cylinder, and the previously used water forced out at the bottom. If the cylinder were initially empty the small valve on the top cover had to be opened during filling to allow the air present to escape. The maximum useful capacity of the cylinder was obviously attained when the water surface was at the level of the drain valve.

On the opening of the main stop valve, the water under pressure from the accumulator acted on the oil above it in the cylinder and forced it out at the top along the pipe to the fuel valve. By throttling the flow of water past the main stop valve, any pressure from atmospheric to 2240 $lbs/in.^2$ could be obtained.

During the first attempts to use this system, the oil was sometimes used up and water inadvertently allowed to pass through the nozzle; this water not only spoiling the particular reading, but also fouling the inside of the fuel valve and the oil already discharged into the spray chamber. A thick emulsion of oil and water was left in the container along with the pure oil. When the latter was returned for use again to the oil cylinder the emulsion carried with it upset the readings of the following discharges.

Several attempts were made to obtain a float arrangement which would indicate the completion of the oil discharge. Floats of various materials were tried but were all found to be defective, due to the action of the water or oil under high pressure. What might have been a possible solution to this float problem, but which was not tried, would have been a very light hollow metal float filled with a very light fluid.

Finally after some practice it was found possible to judge from the duration of the experiment the probable oil level in the oil cylinder, and trouble from the foregoing cause/

cause was practically eliminated.

To the bottom of the plug in the top cover of the oil cylinder and therefore over the mouth of the pipe leading to the nozzle was fixed, in place of the above mentioned device, a large cylindrical filter made from 200 mesh copper gauze. This fine gauze was supported on the inside by a gauze of much coarser mesh. This was the only filter in the system during the experiments on discharge.

Heating Coil.

In order to investigate the effect of temperature on the rate of discharge of fuel oil, a heating coil was fitted to the pipe leading from the top of the oil cylinder to the fuel valve. For this purpose two lengths (each 44 ft.) of nickel chrome resistance wire were employed. The whole length of pipe was first wound carefully with one ply of asbestos string. The two lengths of resistance wire were then wound on in the opposite direction so that each length covered approximately half the total length of the pipe. The ends were clamped by small brass clips round the covered pipe. The whole length was finally wound with another layer of asbestos. The brass clips were also used as terminals, and the two coils were connected in parallel to a switch on the 250 volt main. With the object of keeping the temperature constant, a variable resistance was added to the circuit. This latter however, was found to be of no advantage, as its effect took too long to make any appreciable difference in the temperature of the oil, and it was dispensed with.

The temperature of the oil caused by the heating coil was measured by means of a eureka-copper thermocouple placed close up to the valve seat as shown in Fig.13,p.27. On the whole, this heating system did not prove to be satisfactory/

satisfactory since during any single test the oil temperature value at the nozzle varied considerably. The average temperature however was found to be similar to the actual working oil temperature recorded on the engine. (See tables 1 and 8 p.52 and p.107).

Air Compressor.

The compressed air necessary for the experiments was obtained from a small Reavell two-stage compressor designed for pressures of from 600 to 700 lbs./in^2. It was found to work satisfactorily at 900 lbs/in.^{2} or even more. It was driven by belt off the main line of shafting in the laboratory. For this purpose it had to be fitted with a specially heavy flywheel. In order to simplify the working of the complete apparatus and make it controllable by one person, the compressor was provided with a small safety valve loaded with an ordinary indicator spring which was readily adjustable to any desired pressure. When the air was compressed to the required amount, the spray chamber could be cut off by means of the air valve (Fig.11,p.25) and the safety valve prevented the pressure in the compressor and in the pipe leading to the chamber, from rising unduly. As soon as was possible after the valve at the chamber was closed, the load was taken off the compressor by releasing the safety valve completely until pressure was again required. The cylinder of the compressor was water cooled from a small circulating tank situated immediately above. The length of copper pipe between the compressor and the spray chamber provided ample cooling for the heated air. Fig.11,p.25 shows the arrangement of the compressor in relation to the spray chamber and auxiliary units.

- - - - - - - - - - - - -

DISCHARGE EXPERIMENTS.

Particularly essential to an investigation of the rate of fuel injection into the engine combustion chamber by means of practical curves of pressure variation and fuel valve spindle motion, are practical experiments on the rate of discharge of this fuel through the particular nozzle with which the former curves were obtained - unless the results of previous investigation are complete enough to allow this part of the work to be dispensed with. In order to attain a reasonable degree of accuracy, the range of certain conditions in the discharge experiments must agree as nearly as possible with the actual running conditions.

The possible conditions affecting discharge appear to be,

(a) The pressure in the oil behind the nozzle.
(b) The density and viscosity of the oil at the nozzle.
(c) The amount of lift of the valve spindle.
(d) The pressure, density and viscosity of the medium into which the oil discharges.

In order to bring out more clearly the relationship existing between these above conditions, various liquids were used in the following experiments, and for this reason the results may possibly be, apart from their immediate value in this work, of more general interest. On this score however, it is perhaps unfortunate that the orifice should have been of the 'slot' form instead of being circular.

The experimental determination of curves of discharge into the atmosphere proved to be a more difficult task than was at first anticipated. As already pointed out, during the course of these experiments the rate of discharge under any given set of conditions was found to increase steadily (as indicated on Figs. 15 and 16, p.39).

The/

The readings taken during the earlier set of experiments which gave increasing discharge were discarded and another series of a comparatively small number of individual readings, each occupying about the same period as before, was carried out. Over the duration of this series a relatively small discharge increase took place.

The discharge readings determined by this series of experiments are rather unsatisfactory, in the matter of consistency. One explanation of this condition may be that advanced by Dr. Telford Petrie in "An Investigation into the Phenomena of Discharge through short lengths of small diameter Nozzles." published in Vol 13 of the Journal of the Municipal College of Technology, Manchester. The results given therein show that for a small circular orifice there are two critical ratios of length to diameter, in the case cited between 2 and 4 and between 0 and 1, at which unstable conditions of flow exist. Under these unstable conditions the jet was found to vary considerably in its formation and the rate of discharge to vary by as much as ±6% from the mean value. These tests, it appears, were carried out with water, but the same phenomenon is likely to occur with low viscosity oils such as have been mainly used in the present experiments. It seems reasonable to suppose therefore, that due care should be exercised in fixing the dimensions of the orifice, or orifices, for an engine injection nozzle so that under running conditions it does not require to operate within either of these critical zones. It would be interesting to study the actual effect of this phenomenon on an engine under such conditions.

The partial choking of the orifice encountered in these/

these experiments would seem to be a particular disadvantage of the high pressure injection type of moderately low-powered oil engine. A particle of foreign matter need not be very large to lie across an orifice measuring say .004", and more especially if the axis of the fuel valve be vertical. A particle may be only sufficient to alter slightly the actual discharge through the orifice, and yet may considerably impair the formation of the spray. In order to minimise the possibility of this trouble it is suggested that, whatever shape of orifice is used, the edges should be kept sharp, thus reducing the coefficient of discharge and allowing of the use of a larger opening.

This trouble of choking clearly can only be partially eliminated by very efficient filtering, since even with a great amount of care, traces of grit may be allowed to remain in the actual body of the fuel valve after it has been dismantled for any reason.

Another interesting series of experiments bearing on this subject of discharge has been published under the title of "An Investigation of the Coefficient of Discharge of Liquids through small round Orifices" by W.F.Joachim, in the Eleventh Annual Report of the National Advisory Committee for Aeronautics 1925, in America. Owing however, to the much wider range of Oil Pressures employed in the above investigation, the different types of orifices and differences in the methods of stating the results, a comparison, other than in a very general sense, is almost impossible.

Discharge of Liquids through Fuel Valve Nozzle into Air at Atmospheric Pressure (with Valve Spindle at Maximum Lift.)

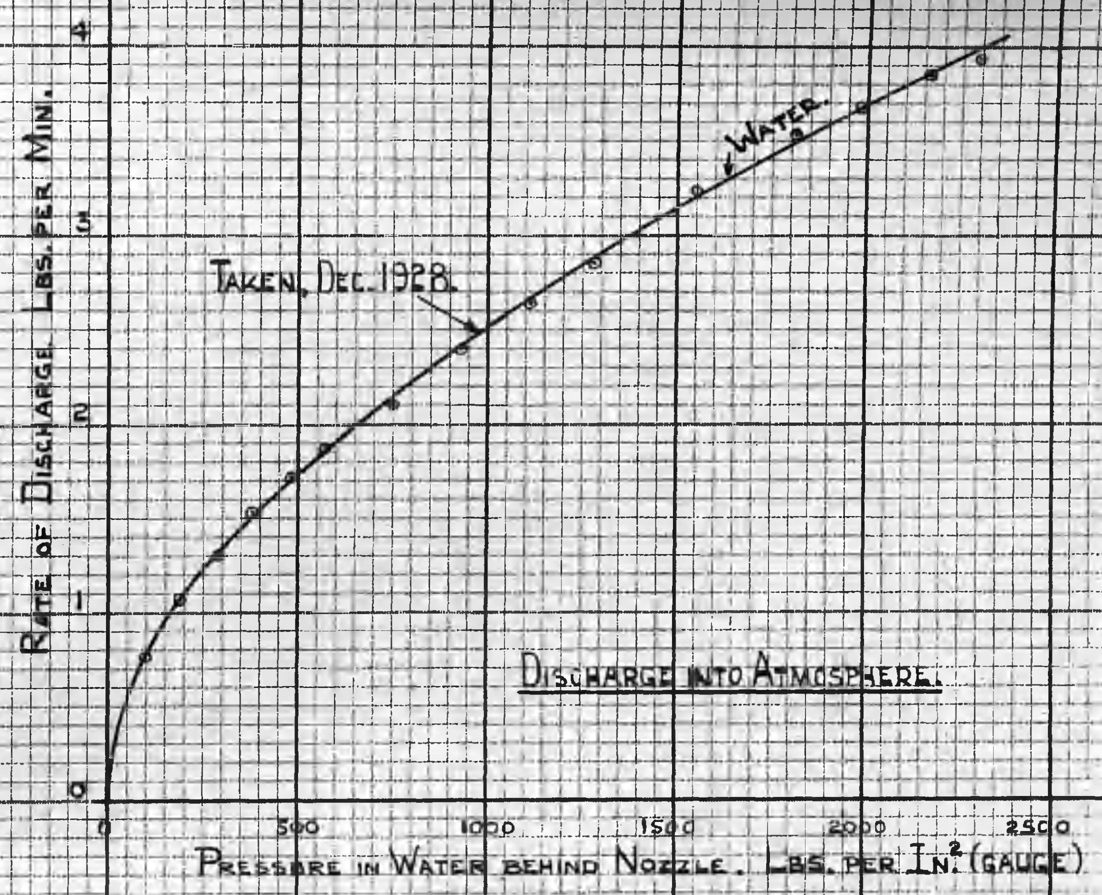
In the following series of experiments the valve spindle was maintained at its maximum lift and therefore, as/

as is shown in the next section of the paper, had no appreciable throttling effect on the discharge. Readings were taken of the discharge into atmosphere of water, paraffin, fuel oil at ordinary temperature, fuel oil heated to approximately the temperature prevailing under running conditions, and lubricating oil (Mobiloil BB) at ordinary temperature. The maximum range of pressures obtainable was from atmospheric to 2240 $lbs/in.^2$ gauge, a range of 2240 $lbs/in.^2$ At first, readings were taken at pressures below 500 $lbs/in.^2$ but as difficulty was experienced in keeping a steady reading on the pressure gauge at lower pressures, and also as it was desirable to reduce the total number of readings as much as possible, 500 $lbs/in.^2$ oil pressure was finally taken as the lower limit of the range.

The discharge results of these tests are shown in Fig.16, p.39 in lbs/min. plotted to a base of 'pressures behind the nozzle' and in Fig.17, p.40 in ins^3/sec. plotted to a base of 'inches head of liquid'. Fig.17, p.40 represents the correct basis for comparison between the discharge coefficients for the various liquids, the standard formula for discharge being Volume discharged / unit time $= Cd \times A \times \sqrt{2gh}$ where Cd = discharge coefficient, A = orifice area, and h = head of liquid. The average curve drawn in Fig.17, p.40, is of the parabolic form $y^2 = ax$, where $y = in.^3$/sec. and x = head of liquid in inches.

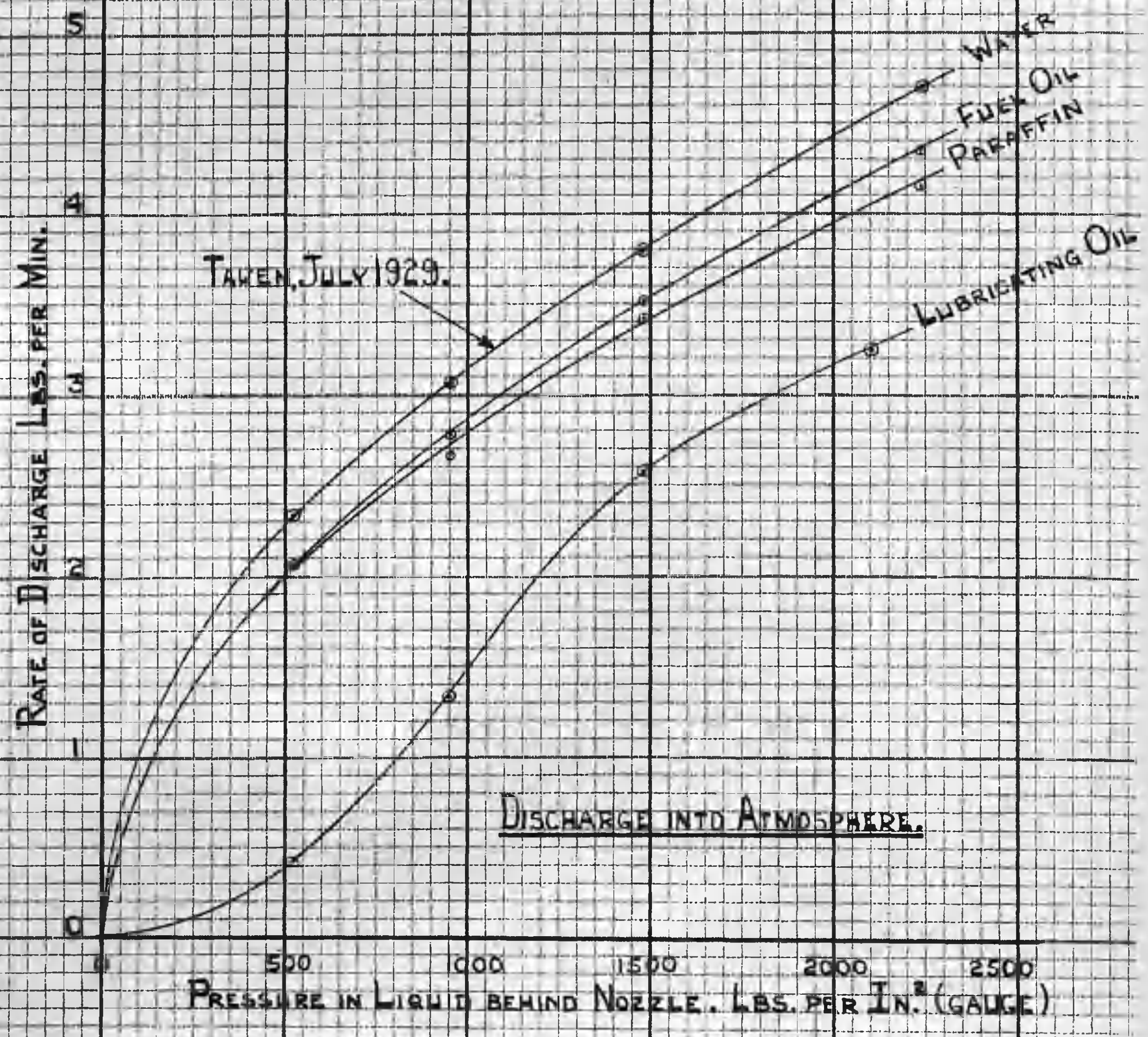
In table I, p.52 are given the physical characteristics of the liquids. On the assumption that the single curve (Fig.17, p.40.) represents with reasonable accuracy the discharge of the various liquids, it appears established that, over a fairly wide range, the viscosity of the liquid has very little effect on the coefficient of discharge.

The writer has been unable to ascertain whether or not this result has already been established. It is possible that the range may practically comprise all the values of viscosity met with in oil engine fuels. The coefficient/



Rate of Discharge Lbs. per Min.
0
1
2
3
4
Water.
Taken, Dec. 1928.
Discharge into Atmosphere.
0
500
1000
1500
2000
2500
Pressure in Water behind Nozzle. Lbs. per In.² (gauge)

FIG. 15.



Rate of Discharge Lbs. per Min.
0
1
2
3
4
5
Water
Fuel Oil
Paraffin
Lubricating Oil
Taken, July 1929.
Discharge into Atmosphere.
0
500
1000
1500
2000
2500
Pressure in Liquid behind Nozzle. Lbs. per In.² (gauge)

FIG. 16.

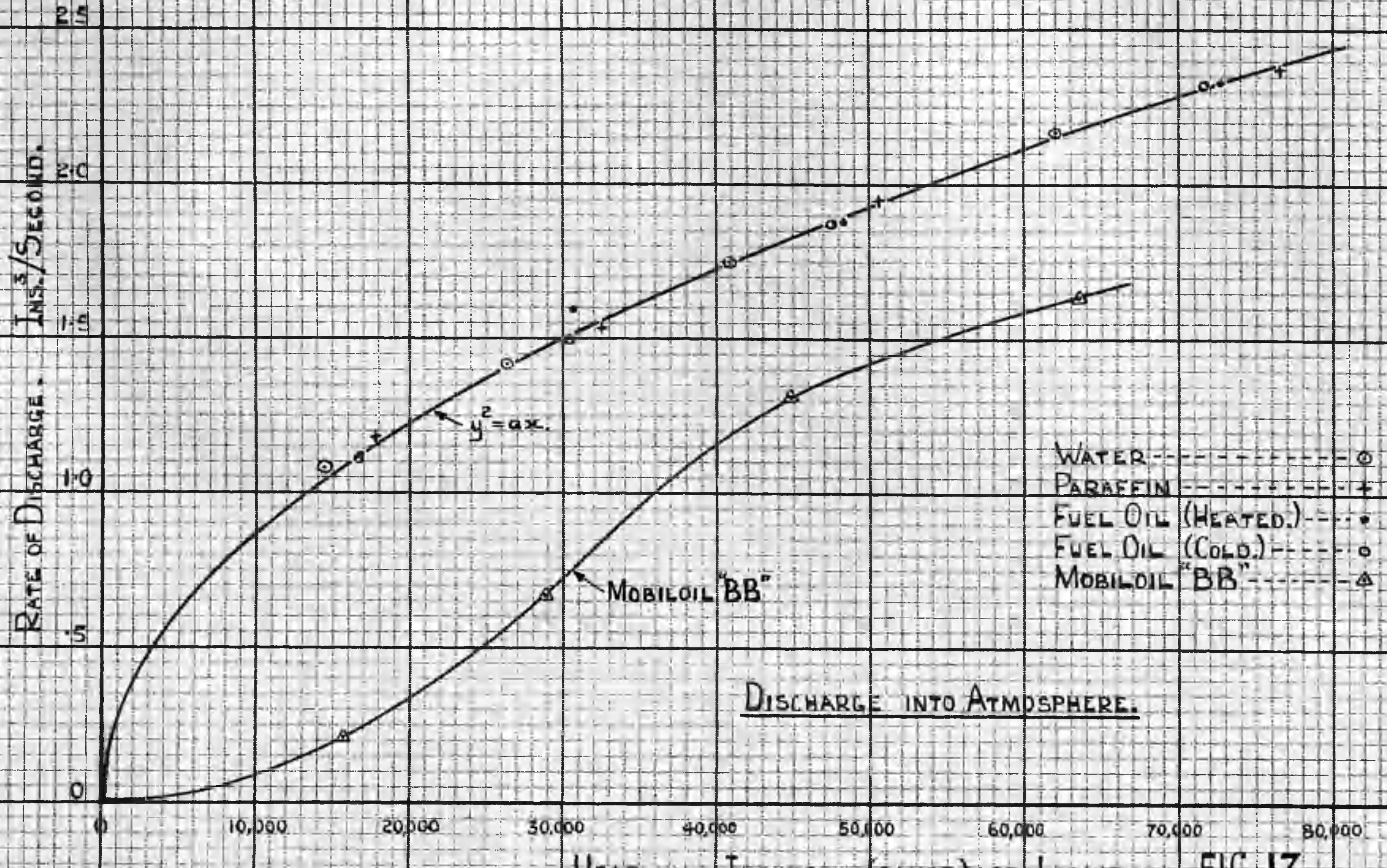

Rate of Discharge. Ins.³/Second.
2.5
2.0
1.5
1.0
.5
0
0
10,000
20,000
30,000
40,000
50,000
60,000
70,000
80,000
y² = ax.
Mobiloil "BB"
Water ⊙
Paraffin +
Fuel Oil (Heated) •
Fuel Oil (Cold) o
Mobiloil "BB" △
Discharge into Atmosphere.
Head in Inches, (gauge) of Liquid.

FIG. 17.

coefficient of discharge is (except for Mobiloil BB) practically the same for all liquids tested and is also practically constant over the range from 500 lbs/in.^{2} upwards. Within the range of the experiments the deviation from the above 'law' can be best seen in Fig.16, p.39, by taking the discharge values at 500 and 2000 lbs/in.^{2} gauge pressure in the liquid. For constant coefficient of discharge the discharge at 2000 lbs/in.^{2} should clearly be twice that at 500 lbs/in.^{2} For example, the water curve gives 2.27 lbs/min. at 500 lbs/in.^{2} and 4.43 lbs/min. at 2000 lbs/in.^{2} This last value should be 2.27 x 2 = 4.54 lbs/min. and the deviation from the 'law' is therefore $\frac{4.54-4.43}{4.54}$ or practically 2.5%. Paraffin gives a deviation of a smaller degree in the same direction but for the cold fuel oil it is 2.5% in the opposite direction, that is the discharge at 2000 lbs/in.^{2} is greater than it should be for constant discharge coefficient. For Mobiloil BB this is obviously the case also to a much greater extent. Strictly speaking therefore it appears that for low viscosities the coefficient decreases slightly as the pressure increases, but as the viscosity approaches that of fuel oil this decrease changes to a slight increase. When the viscosity becomes that of Mobiloil BB this increase becomes considerable (Fig.16, p.39). More intermediate viscosities would have been necessary to enable one to determine any definite relationships here.

Further it appears to be clear from Fig.17, p.40, that neither is density a determining factor on discharge. (It must of course be borne in mind that by discharge in these last remarks is essentially meant rate of volumetric discharge as plotted against a base of head which, strictly speaking is the only manner in which a true comparison can be made when dealing with various liquids). The justification for this statement regarding density is considerable on account of the similarity between the viscosities of water and paraffin, the marked difference between/

between their respective densities and the fact that their rates of discharge agree so closely.

Regarding temperature, except that it reduces viscosity, it seems reasonable to assume that it also has no effect on discharge.

As the possible degree of accuracy of the work did not appear to justify the additional complication, no account has been taken of the compressibility of the liquids. Consideration of the curve for Mobiloil BB seems to indicate that at the higher ranges of viscosity this property does very definitely reduce the coefficient of discharge, and especially so at the lower values of head. Here again various intermediate values of viscosity would be necessary to enable one to determine a relationship.

Effect of Fuel Valve Spindle Lift on Discharge through the Fuel Valve Nozzle into Air at Atmospheric Pressure.

All fuel valves whose spindles open either automatically by oil pressure or mechanically by a cam or some similar gear have the inherent fault that, no matter what the diameter of the valve seat may be, at the beginning of an injection into the combustion chamber the oil spray must grow from zero into a full spray and then, at the end of the injection, shrink from this back to zero again. This of course is quite apart from the idea of rate of spray growth as determined by the actual velocity of the jet front, which has been investigated and described in the Report on "Spray Penetration with a simple Fuel Injection Nozzle" in the Eleventh Annual Report of the National Advisory Committee for Aeronautics 1925 in America, and also in various other reports of the same committee. This growth or shrinkage under present discussion is due to the restricting influence on the flow of/

of fuel oil of the valve and valve seat. At normal speeds, with a well designed valve of which the opening and closing periods are extremely short, the effect may be hardly appreciable, but nevertheless it must exist and at higher speeds will become of increasing importance.

Generally speaking, except as regards simplicity no advantage is derived, but rather the reverse, by the elimination of the valve spindle. The periods then required for the growth and cut off of the spray will be greatly increased, depending as they do on the relatively slow rise and fall in pressure instead of the much more rapid opening and closing of the valve by the spindle. This will be readily seen from Figs. 48-52, pp113-120, which show the difference between a rise and fall of oil pressure, and the corresponding opening and closing of the valve.

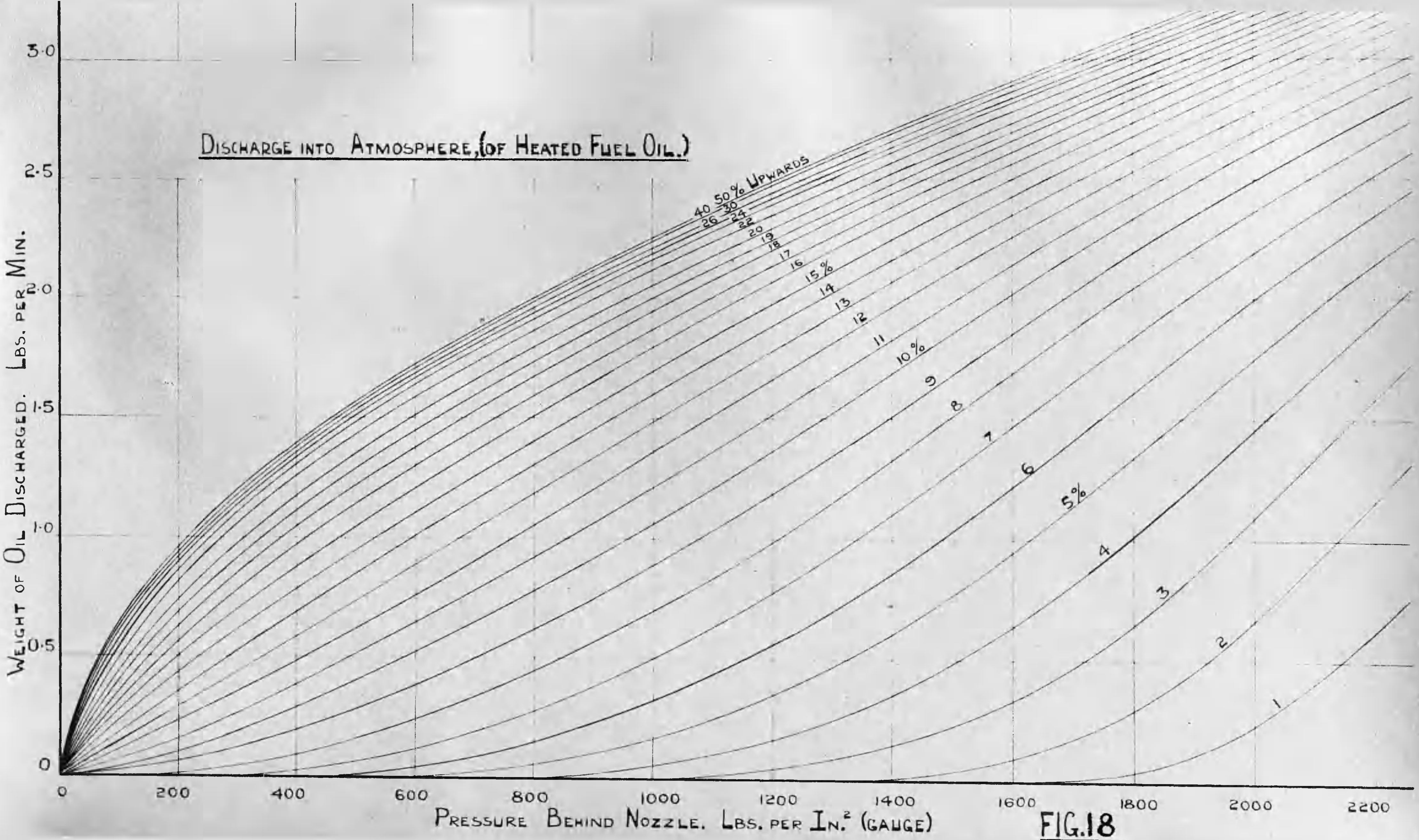
When a fuel pump has been devised which will raise the pressure to its maximum and then release it more rapidly than a well designed valve spindle could correspondingly open and shut, then clearly the elimination of the latter would be a distinct gain, both in simplicity and economy. In fact, such a number of fuel pumps and valves of every description have been recently put forward that it is possible that the design of such a pump may have been already accomplished.

The following experiments were carried out with the object of investigating the effect of valve spindle lift on the discharge of various liquids into air at atmospheric pressure. The effect on discharge into air at higher pressures is treated in the next section.

During the discharge experiments on valve spindle lift it was found, for all the liquids that during any single/

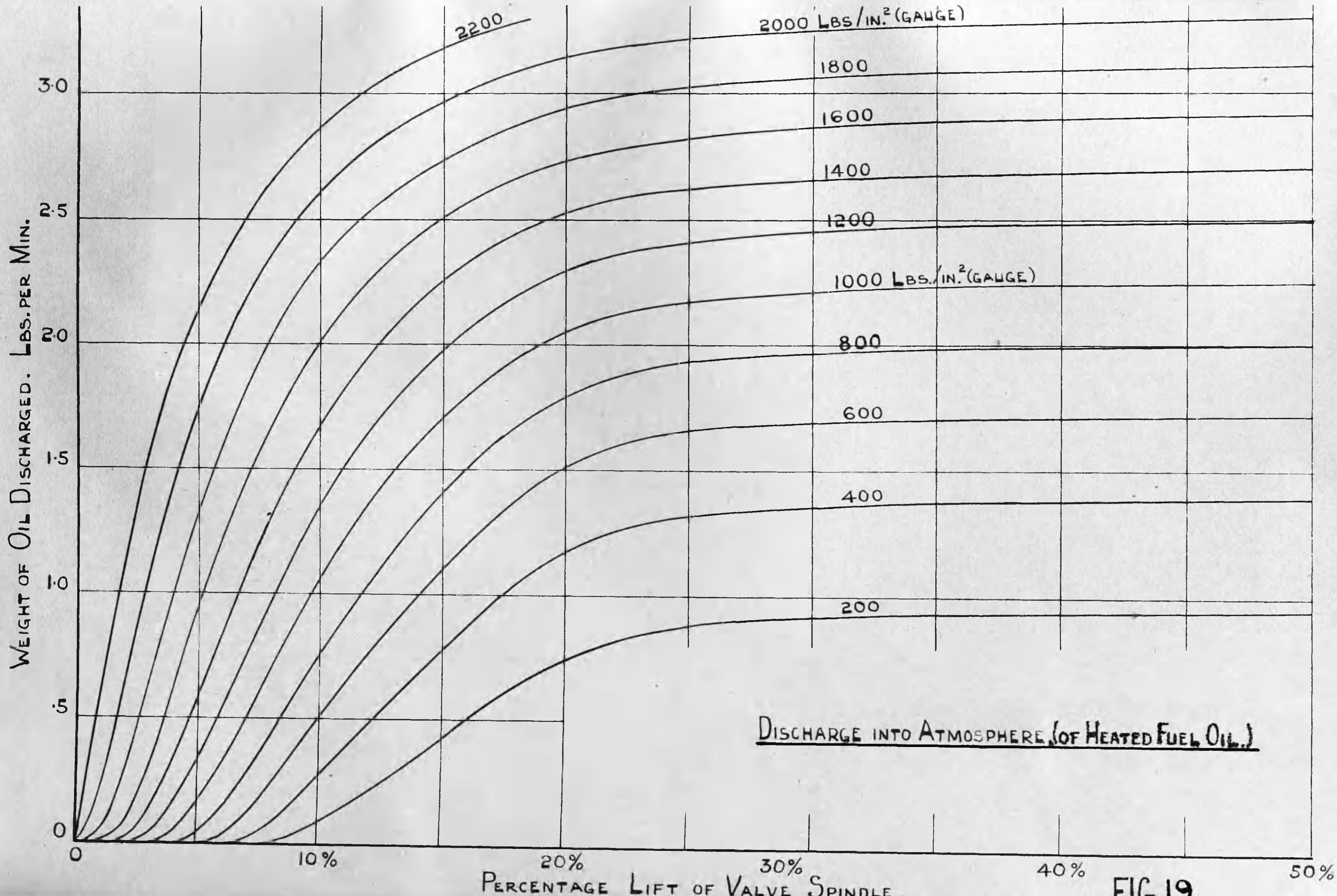
single test at comparatively small spindle lift the pressure behind the nozzle tended to rise and the rate of discharge to decrease simultaneously. A sharp movement of the spindle in either the closing or opening directions brought the pressure and discharge back again to their original states. If the spindle was left stationary the same again took place. Referring to the adjustment in Fig.13,p.27,the best way to overcome this difficulty was to keep the spindle in a continual but very slight state of oscillation throughout each test. After the phenomenon had been discovered the lift experiments were carried out with the spindle in continual motion, those previously carried out being repeated.

As has already been mentioned on page 23,considerable difficulty was encountered in obtaining a satisfactory lift adjustment. With the arrangement in which the adjusting screw was set to bear on the end of the extension to the valve spindle, on account of the compressive effect of the liquid pressure the readings obtained were practically valueless. A set of curves of discharge for heated oil obtained with this arrangement are given in Figs.(18) and (19) pp.45 and 46. These curves were built up from about half a dozen curves experimentally obtained. The effect of the pressure on the valve lift is very evident from these curves, the discharge being unduly high at the higher pressures and low at the lower pressures. In Figs.20-24pp.47-49 are the later curves of discharge at the various settings of the lift for the respective liquids. These results were obtained by the means of adjustment shown in Fig.13, p.27 and may be taken to be fairly reliable. Here also, as in the case of discharge under full lift, the effect of viscosity of the order met with in fuel oils is clearly not of great importance and the/



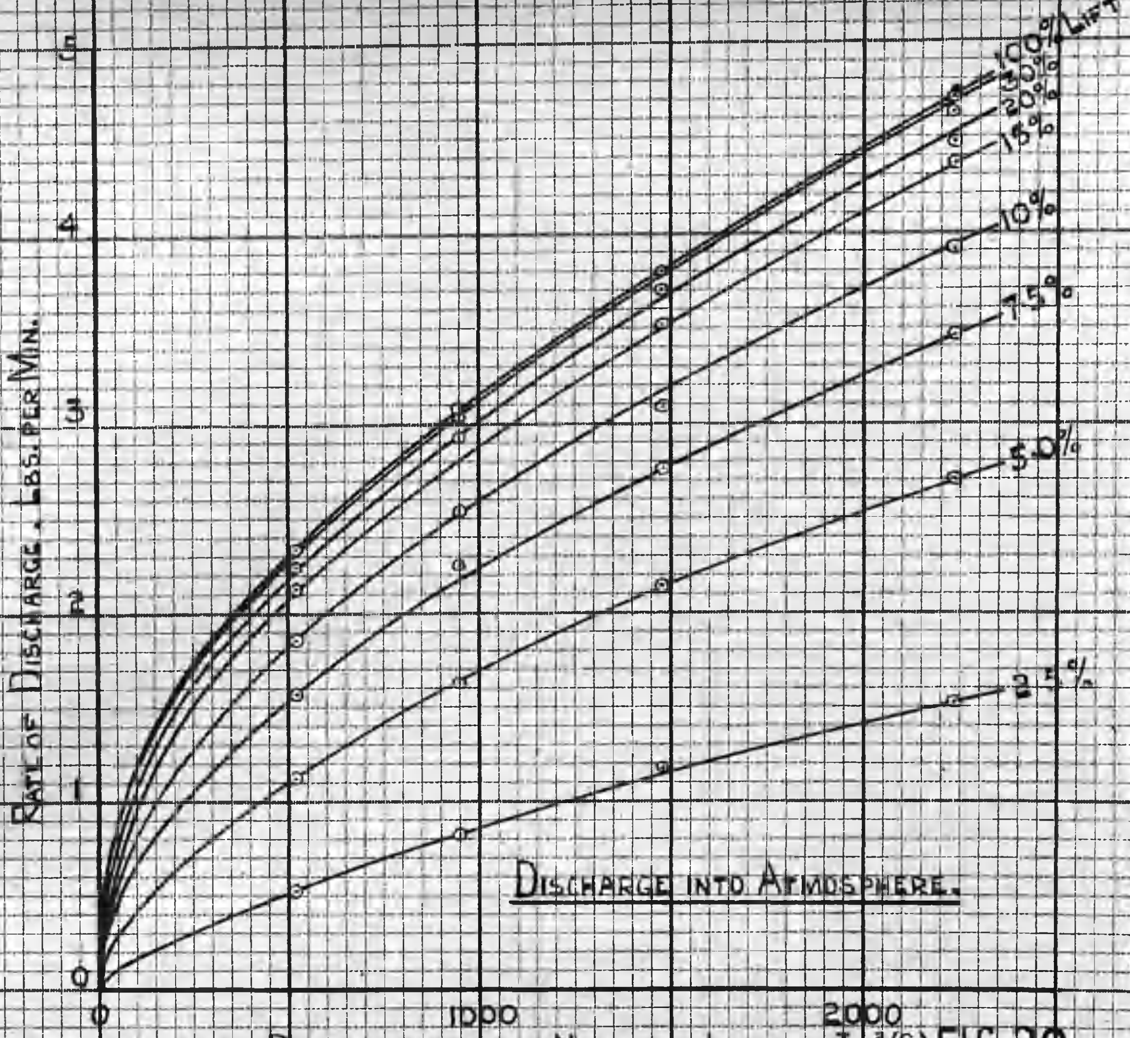
DISCHARGE INTO ATMOSPHERE, (OF HEATED FUEL OIL.)
WEIGHT OF OIL DISCHARGED. LBS. PER MIN.
PRESSURE BEHIND NOZZLE. LBS. PER IN.² (GAUGE)
50% UPWARDS
40
30
26
24
22
20
19
18
17
16
15%
14
13
12
11
10%
9
8
7
6
5%
4
3
2
1
0
0.5
1.0
1.5
2.0
2.5
3.0
200
400
600
800
1000
1200
1400
1600
1800
2000
2200

FIG. 18



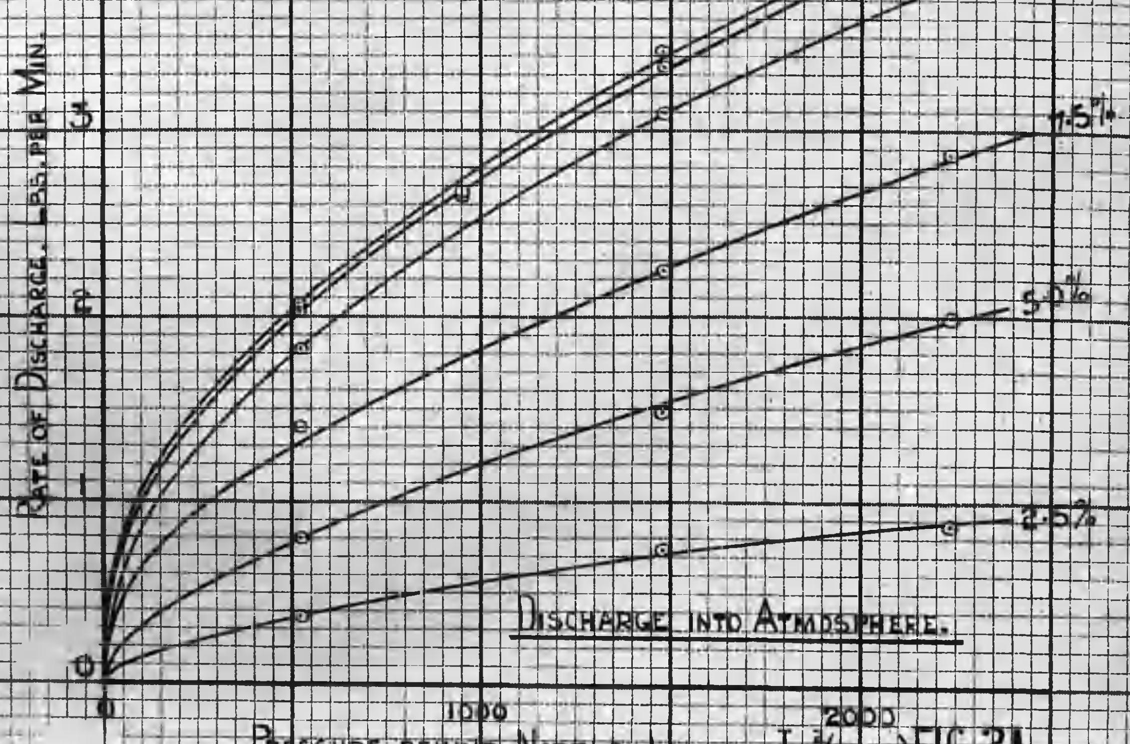
Weight of Oil Discharged. Lbs. per Min.
3·0
2·5
2·0
1·5
1·0
·5
0
0
10%
20%
30%
40%
50%
Percentage Lift of Valve Spindle.
2200
2000 Lbs/in.² (Gauge)
1800
1600
1400
1200
1000 Lbs./in.² (Gauge)
800
600
400
200
Discharge into Atmosphere (of Heated Fuel Oil.)

FIG. 19.



Pressure behind Nozzle. Lbs. per In.² (G) FIG. 20.

Discharge of Water into Atmosphere at Various Settings of Valve Lift.



Pressure behind Nozzle. Lbs. per In.² (gauge) FIG. 21.

Discharge of Paraffin into Atmosphere at Various Settings of Valve Lift.

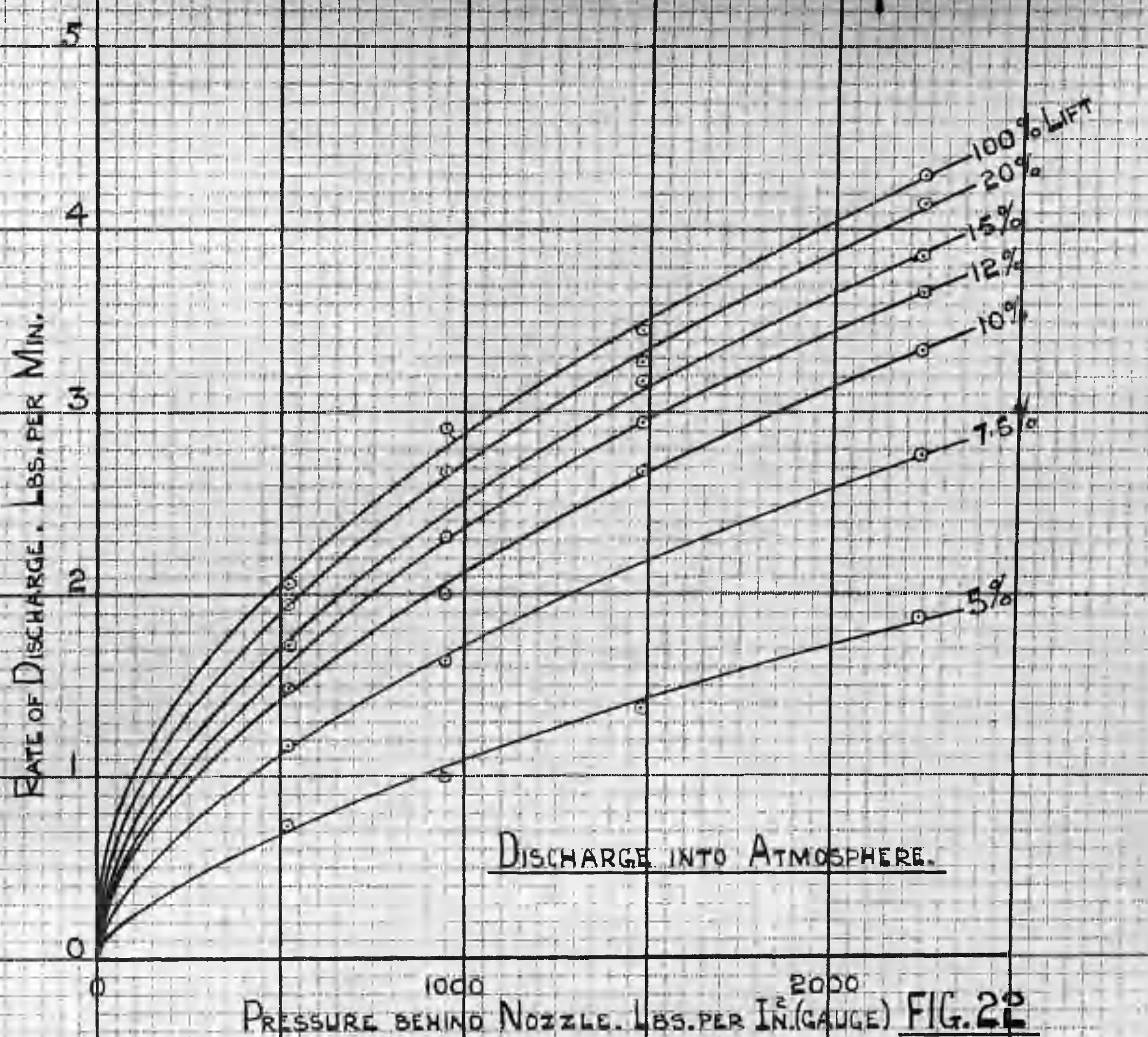


FIG. 22

Discharge of Fuel Oil (Heated) into Atmosphere at Various Settings of Valve Lift.

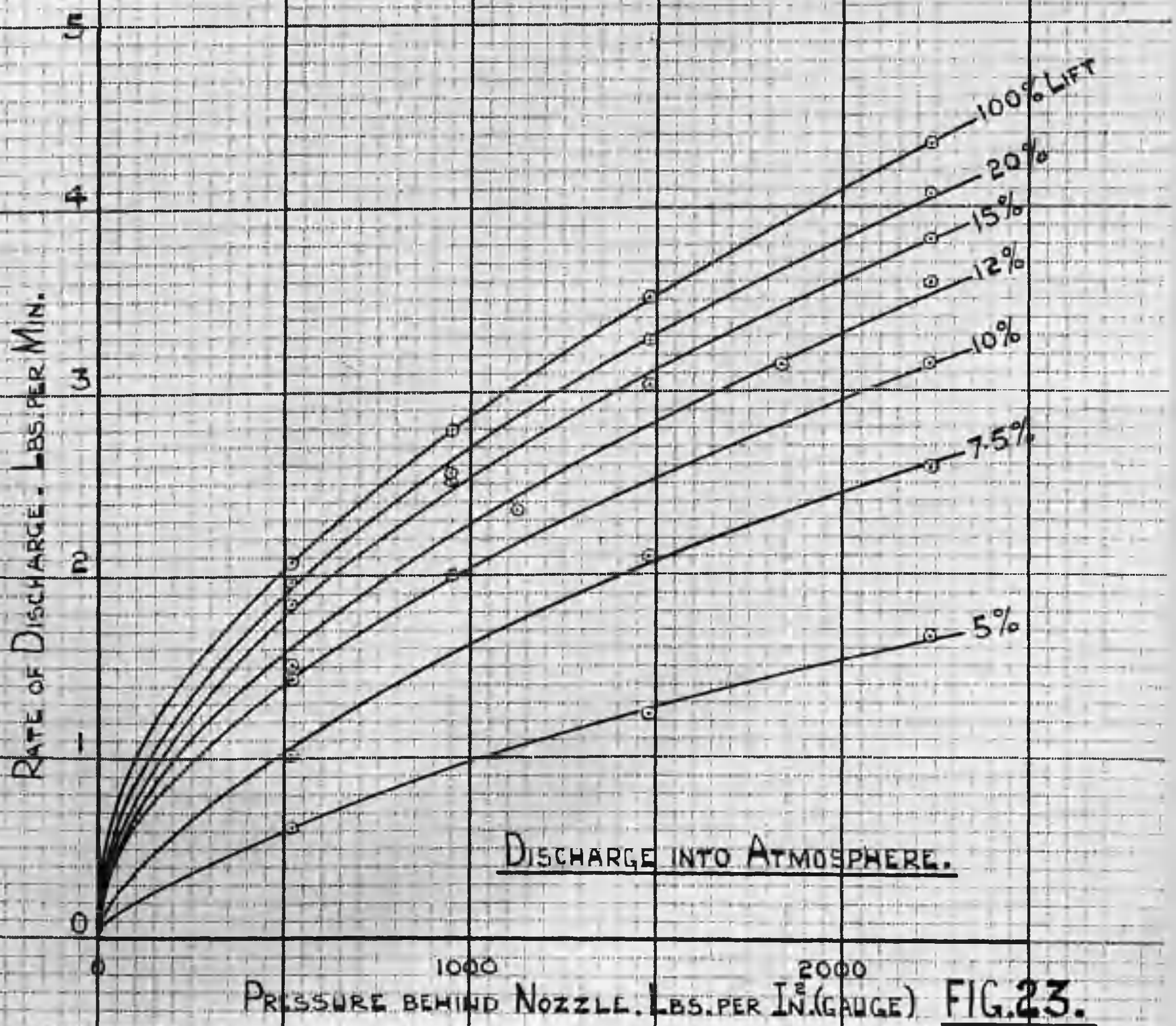


FIG. 23.

Discharge of Fuel Oil (Cold) into Atmosphere at Various Settings of Valve Lift.

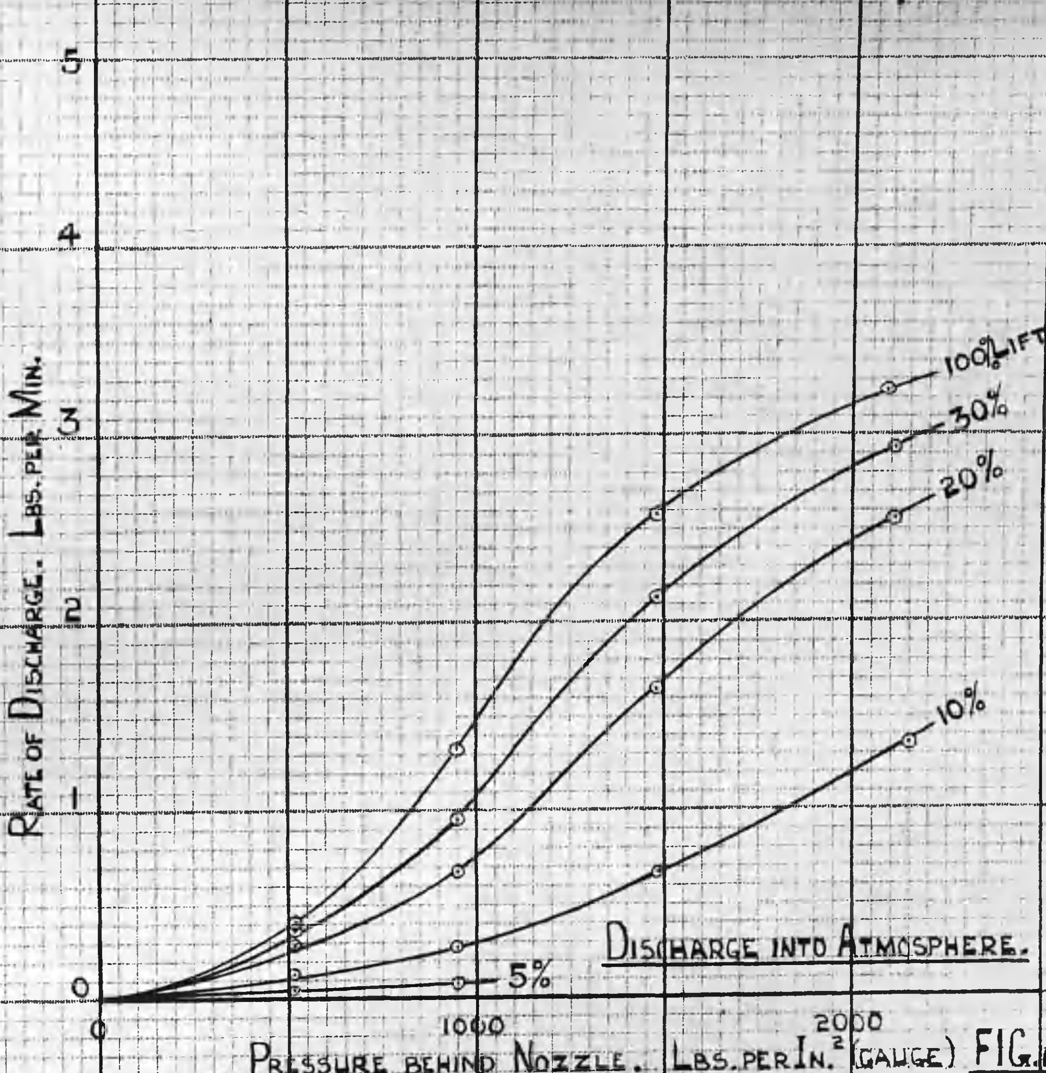


FIG. 24.

Discharge of Lubricating Oil into Atmosphere at Various Settings of Valve Lift.

the behaviour of the Mobiloil BB again proves that a relatively high order of viscosity has on the other hand a considerable effect. The viscosity effect is much more clearly brought out by the curves in Fig.(25), p.51 which were obtained by plotting mean values of discharge on a base of valve lift. The values of discharge averaged, were not absolute but were taken as a percentage of the discharge at the maximum lift for given conditions of pressure etc., or what is the same thing - a percentage of the maximum discharge within these conditions. The results necessary for these curves are given in tables 2-6, p.52. The final column of each table gives the mean of the four preceding columns and these mean values have been used to plot the curves in Fig.25, p.51. Although hardly permissible, on consideration of the wide variation in the individual readings, averages were taken for the Mobiloil BB also, in order that an approximate means of comparison might be obtained.

As regards the injection of fuel into the engine, it may be concluded from these results that a viscosity of the order of that of fuel oil has only a slight effect on the discharge under any condition of restriction between the valve spindle face and the valve seat; but the greater the viscosity the greater will be the reduction in discharge. Probably the curve for cold fuel oil gives a fair conception of the worst case, on account of the heating effect of the nozzle on the oil in actual practice.

The results of these experiments also show the probable effect of the restricting influence of the valve during the opening and closing periods. If the penetration and atomisation be correct when the valve spindle is at its maximum lift value, then at any lower lift these qualities/

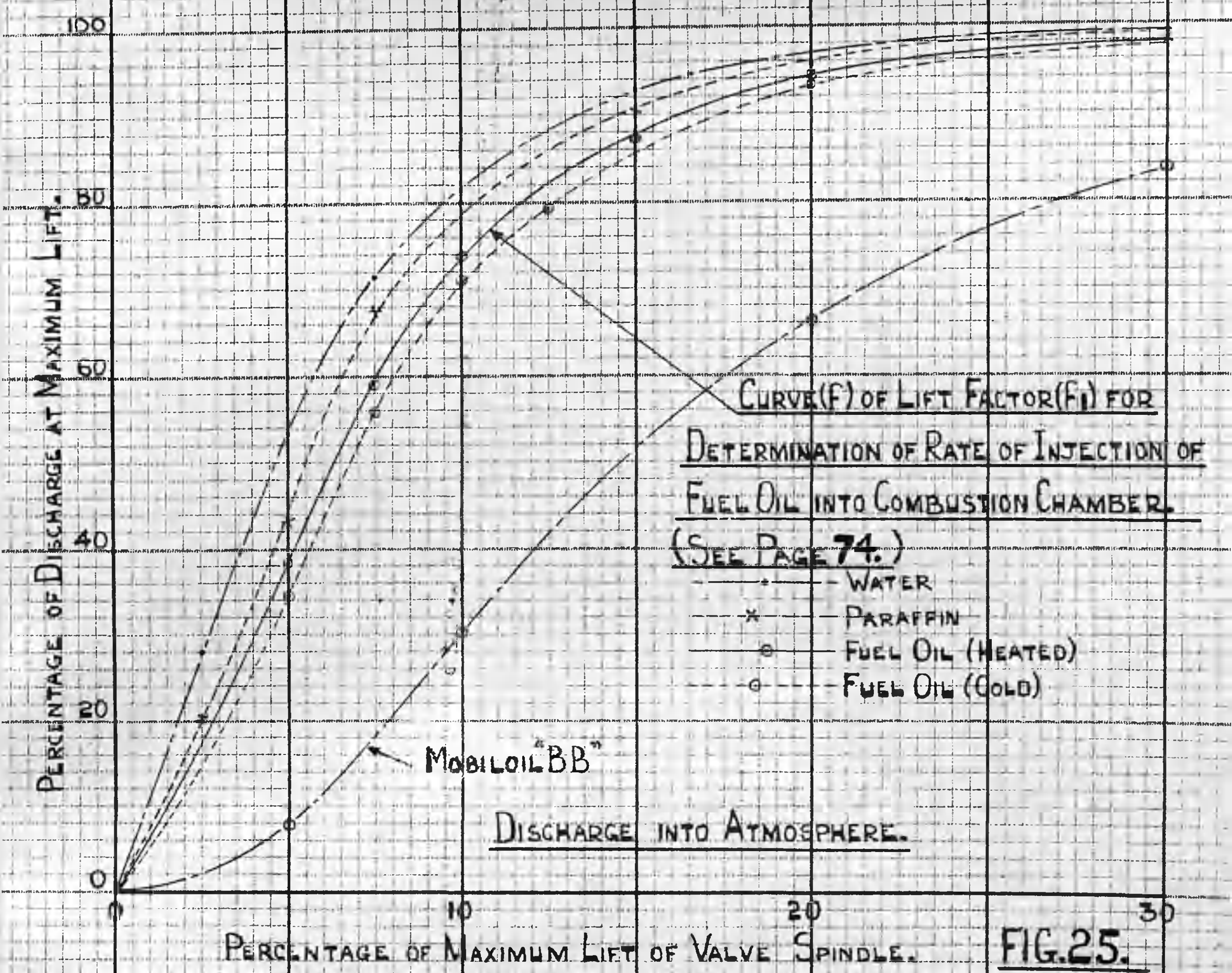

Percentage of Discharge at Maximum Lift.
100
80
60
40
20
0
0
10
20
30
Percentage of Maximum Lift of Valve Spindle.
Curve (F) of Lift Factor (F_1) for Determination of Rate of Injection of Fuel Oil into Combustion Chamber.
(See Page 74.)
Water
Paraffin
Fuel Oil (Heated)
Fuel Oil (Cold)
Mobiloil "BB"
Discharge into Atmosphere.

FIG. 25.

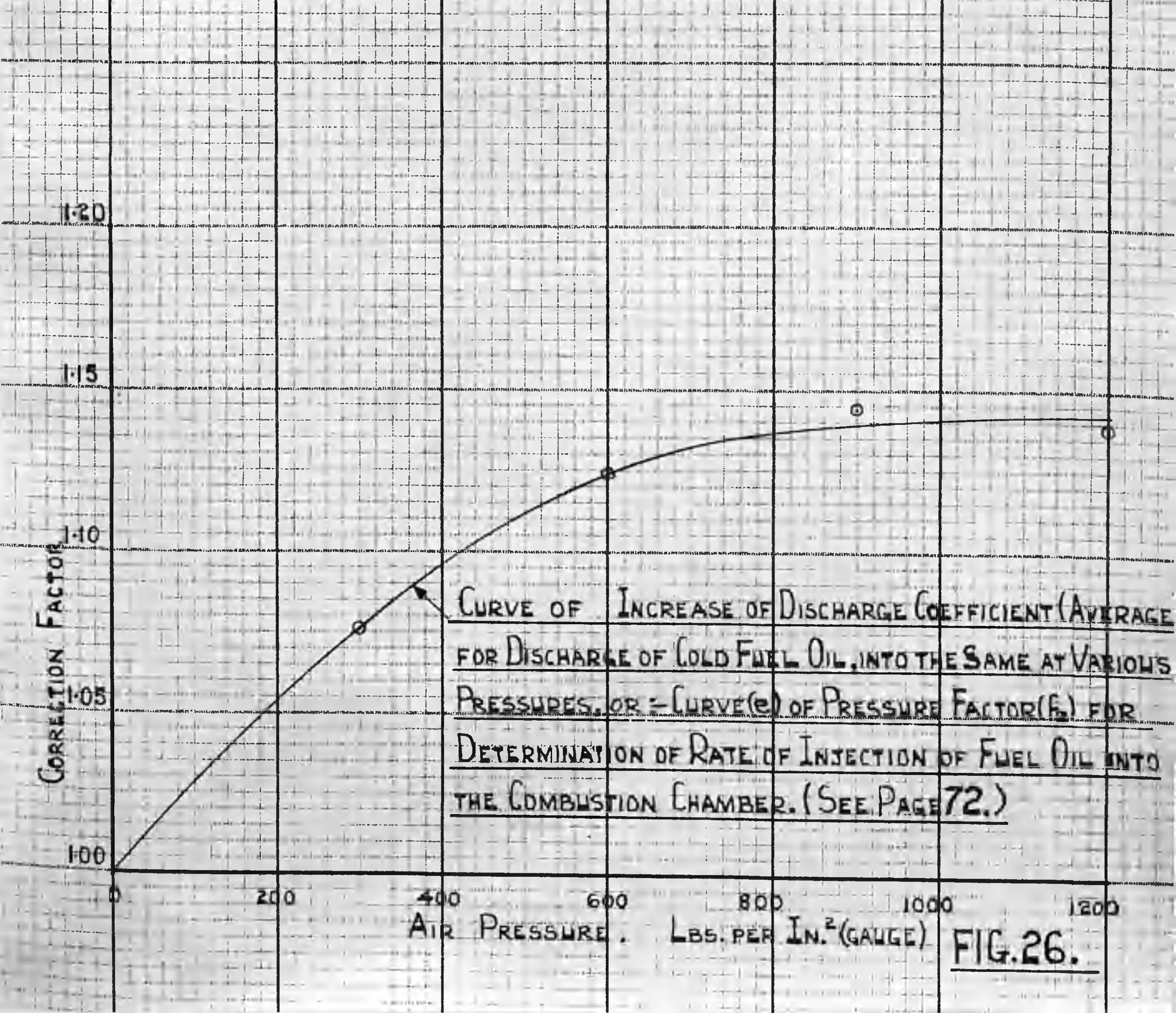

Correction Factor
1·20
1·15
1·10
1·05
1·00
0
200
400
600
800
1000
1200
Air Pressure. Lbs. per In.^{2} (gauge)
Curve of Increase of Discharge Coefficient (Average for Discharge of Cold Fuel Oil, into the Same at Various Pressures, or = Curve (e) of Pressure Factor (F_2) for Determination of Rate of Injection of Fuel Oil into the Combustion Chamber. (See Page 72.)

FIG. 26.

TABLE 1.	PARTICULARS OF LIQUIDS DISCHARGED.				
PRESSURE LBS./IN.²	HEAD IN INCHES.				
	WATER. 23°C.	PARAFFIN. 22°C.	FUEL OIL. 56°C.	FUEL OIL. 22·5°C	MOBILOIL BB. 17·5°C
520	14400	17800	16900	16700	15800
950	26300	32500	30900	30400	28900
1480	40900	50600	48200	47400	44900
2090	–	–	–	–	63500
2240	62000	76600	72800	71700	–
SPECIFIC GRAVITY	·999	·806	·846	·863	·910
DENSITY LBS./IN.³	·0361	·0292	·0307	·0312	·0329
VISCOSITY C.G.S.	·009		·39	·84	11·00

TABLE 2.	WATER INTO ATMOSPHERE. DISCHARGE FOR 100% LIFT = UNITY.				
PERCENTAGE LIFT	HYDRAULIC PRESSURE. LBS./IN.² G.				AVERAGE.
	520	950	1480	2240	
100	1·000	1·000	1·000	1·000	1·000
30	·995	·984	1·000	·980	·990
20	·960	·948	·972	·950	·958
15	·909	·948	·923	·927	·927
10	·797	·824	·812	·831	·816
7·5	·673	·730	·725	·735	·716
5·0	·475	·525	·563	·573	·534
2·5	·219	·263	·310	·323	·279

TABLE 3.	PARAFFIN INTO ATMOSPHERE. DISCHARGE FOR 100% LIFT = UNITY.				
PERCENTAGE LIFT	HYDRAULIC PRESSURE. LBS./IN.² G.				AVERAGE.
	520	950	1480	2240	
100	1·000	1·000	1·000	1·000	1·000
30	·980	·986	·975	·980	·980
15	·881	·921	·905	·915	·905
7·5	·675	·678	·657	·693	·676
5·0	·387	·432	·435	·481	·434
2·5	·179	·194	·219	·210	·201

TABLE 4.	FUEL OIL (HEATED) INTO ATMOS. DISCHARGE FOR 100% LIFT = UNITY.				
PERCENTAGE LIFT.	HYDRAULIC PRESSURE, LBS./IN.² G.				AVERAGE.
	520	950	1480	2240	
100	1·000	1·000	1·000	1·000	1·000
20	·945	·920	·946	·967	·944
15	·835	–	·890	·898	·874
12	–	·796	·852	·851	·833
10	·722	·686	·773	·775	·739
7·5	·568	·563	–	·643	·591
5·0	·354	·343	·398	·437	·383

TABLE 5.	FUEL OIL (COLD) INTO ATMOS. DISCHARGE FOR 100% LIFT = UNITY.				
PERCENTAGE LIFT.	HYDRAULIC PRESSURE. LBS./IN.² G.				AVERAGE.
	520	950	1480	2240	
100	1·000	1·000	1·000	1·000	1·000
20	·950	·916	·935	·935	·934
15	·892	·898	·862	·877	·882
12	·770	·788	·795	·825	·795
10	·695	·715	·711	·723	·711
7·5	·487	–	·597	·595	·559
5·0	·300	–	·351	·381	·334

TABLE 6.	MOBILOIL BB INTO ATMOS. DISCHARGE FOR 100% LIFT = UNITY.				
PERCENTAGE LIFT.	HYDRAULIC PRESSURE. LBS./IN.² G.				AVERAGE.
	520	950	1480	2000–2200	
100	1·000	1·000	1·000	1·000	1·000
30	·902	·714	·825	·904	·836
20	·706	·511	·639	·790	·662
10	·317	·211	·260	·419	·302
5·0	·097	·060	–	–	·079

qualities are likely to be impaired in a manner and proportion similar roughly to that shown in Fig.(25) p.51.

Coefficient of Discharge of Orifice.

In calculating the coefficient of discharge C_d into atmosphere for the orifice, (100% spindle lift) it is clear from Fig. 17, p.40, that any one point on this curve should give the approximate result for any of the low-viscosity liquids used in the tests - the coefficient being practically constant under all conditions of discharge into atmosphere, within the limits of the experiments (see p.41).

Taking therefore the case of water at 2240 lbs/in.^{2} gauge pressure discharging into atmosphere,

$$\text{Volume discharge} = C_d \times A\sqrt{2gh} \qquad [h = 62000 \text{ ins. from table I.}]$$

$$= C_d \times \cdot 000328 \times 6920 \text{ ins.}^3/\text{sec.} \qquad [A = \cdot 000328 \text{ in.}^2 \text{ Fig. 10.}]$$

$$= 2\cdot 27 \; C_d. \text{ ins.}^3/\text{sec.}$$

From Fig. 17, p.40 Actual volume discharge = 2.15 ins.^{3}/sec.

$$\therefore C_d = \frac{2\cdot 15}{2\cdot 27} = \underline{\cdot 95.}$$

This result can only be approximate on account of the difficulty of measuring the diminutive width of the orifice viz. .0049" with accuracy. In the case of discharge into a 'medium' at high pressure, as shown later, a further increase of 25% or more in the value of C_d can be obtained giving it then a value of 1.2 (approx.).

Effective Area at Valve Seat. (A')

It can be shown for a conical valve spindle and seat that the area A (see Fig. 9, p.18) for any value of lift L is given by

$$A' = \pi . L . \sin\theta \left[2R - \tfrac{L}{2} \cdot \sin 2\theta\right]$$

where θ = half the cone apex angle and R = half the diameter of the circle of contact of the valve spindle and valve seat.

In this case $\theta = 60^\circ$, and R = $\cdot 9$ mm.

$$\therefore A' = \frac{\sqrt{3}}{2} \cdot \pi \cdot L \left[\frac{1\cdot 8}{25\cdot 4} - \frac{\sqrt{3}}{4} L\right] \text{ in.}^2$$

Fig. 25, p.51, shows that for low-viscosity liquids the restricting influence (on the flow) of the valve seat practically disappears beyond a lift value of 30%.

From the above formula,

$$A' \text{ at } 100\% \text{ Lift} = \cdot 00416 \text{ in}^2$$
$$A' \text{ at } 30\% \text{ Lift} = \cdot 00141 \text{ in}^2$$

also from Fig.9 Area of connecting passage between valve seat and orifice = .00315 in^2 and area of orifice = .000328 in^2.

It appears then, that for maximum discharge under given pressure conditions the ratio $\frac{\text{Effective area at valve seat}}{\text{Effective area at orifice}}$ = should not be less than $\frac{\cdot 00141}{\cdot 000328}$ = practically 4.3. The effective area of the orifice has here been taken as the actual area, since the coefficient C_d appears to be practically unity, but where the orifice has a coefficient much less than this the effective area will then be that of the 'vena contracta'.

In the present case the velocity past the valve seat (at 100% lift) will be roughly $\frac{1}{12.7}$ of the velocity through the orifice. The results show that when the velocity at the valve seat exceeds $\frac{1}{4.3}$ of the velocity at the orifice (or 2.2 of that through the connecting passage) the loss in head caused by the restricting tendency of the valve seat becomes noticeable.

<u>Effect of Pressure and Density of Medium on Discharge.</u>

In the Diesel engine the fuel has to be injected against a considerable degree of back pressure due to the highly compressed air, and products of combustion. As the injection proceeds, the pressure in the combustion chamber may increase considerably, and the density also to a certain extent. The temperature of the gases is high when the injection begins and rises as this process continues/

continues, but it is probable that, as stated regarding fuel oil temperatures on page 42, temperature can only affect the rate of injection indirectly, that is in so far as it may cause a change in viscosity, pressure, or density. Its effect therefore is taken into account by those other conditions, both for the liquid being injected and for the medium into which the injection is taking place.

The following experiments were carried out in order that some knowledge might be obtained of the behaviour of the discharge through the fuel valve nozzle into media subjected to various conditions of both pressure and density. In addition, during these experiments readings were taken for various settings of fuel valve spindle lift. The liquids employed were the same as in the previous discharge tests, namely water, paraffin, fuel oil, and Mobiloil BB. The medium into which these liquids were first discharged was air at pressures varying from atmospheric to 1500 lbs/in.^{2} gauge. Such pressures are not normally present in the oil engine cycle, but, as it was found possible to attain this limit, it seemed worth while to investigate their effects. In order to find, then, the effect of the density and viscosity of the medium, these liquids were themselves used as media. That is, the water was discharged into water at pressures varying from atmospheric to 1500 lbs/in.^{2}, the paraffin into paraffin and so on. The effect of valve spindle lift was only investigated for discharge into compressed air.

For these experiments air pressures up to 900 lbs/in.^{2} were obtained directly by means of the air compressor. Under these circumstances the method of procedure was as follows, assuming all valves initially closed and the fuel valve spindle properly adjusted. The air valve (see Fig. 11, p. 25) was opened and the compressor safety valve screwed down. (The compressor was kept running continuously/

continuously during the experiments.) When the test pressure was obtained within the spray chamber the air valve was shut and the safety valve completely released. The auxiliary stop valve was then opened and immediately afterwards the main stop valve. After a wait period of perhaps a second or two, the oil gauge pointer jerked up to about the desired discharge pressure and the discharge time reading was taken from this instant. Simultaneously the outlet valve of the chamber was opened and throughout the discharge this valve together with the main stop valve (and occasionally the auxiliary valve) were operated in such a manner as to maintain the pressures within the chamber and behind the nozzle respectively at their required amounts, a continuous efflux of discharged liquid taking place from the outlet valve. When the period for the discharge had elapsed, the auxiliary valve was smartly closed (by a half turn) and the watch stopped simultaneously. The main valve was then closed but the chamber outlet valve left open until all the contained liquid had been blown out into the receptacle. The total liquid collected was weighed. The main valve was not used for the cut-off of the discharge, as its use gave rise to considerable 'after flow' through the nozzle on account of the elasticity of the whole system. Considerable practice was necessary for the manipulation of all the valves before satisfactory operation could be obtained.

To obtain for the experiments air pressures beyond 900 lbs/in.^{2} (viz. 1200 and 1500 lbs/in.^{2}) an indirect method was adopted, for which preliminary discharges were necessary. For the lower pressure the spray chamber was first filled with air in the usual manner to 600 lbs/in.^{2} and the air valve closed. The auxiliary valve was opened and the main valve adjusted to cause discharge into the chamber at an arbitrarily chosen liquid pressure of 2000 lbs/in.^{2} When

the/

the chamber pressure had risen to 1200 lbs/in.^{2} the auxiliary and then the main valves were closed. The liquid within the chamber was drained off and weighed. Several such readings were taken and a mean value obtained. (The individual readings agreed closely). The above procedure was repeated commencing at a chamber pressure of 700 lbs/in.^{2} and rising to 1500 lbs/in.^{2} The discharge experiments proper, into air at 1200 lbs/in.^{2} and 1500 lbs/in.^{2} were carried out by first obtaining the air pressures in the above manner and then following the method described for test air pressures of 900 lbs/in.^{2} downwards, except that when the chamber was drained at the end of a test and the total liquid weighed, the weight of liquid obtained by the preliminary discharge had to be subtracted to obtain the net discharge over the given period.

Higher air pressures could clearly have been obtained in the above manner, but it was considered that 1500 lbs/in.^{2} was about the safe limit for the packing rings of the windows, and in any case the maximum hydraulic pressure possible was 2240 lbs/in.^{2}, thus allowing as it was, a pressure difference of only 700 lbs/in.^{2} at the above limit.

It was at first intended to use the weighing method described on page 23, in which steady conditions of flow, hydraulic pressure and air pressure were first to be obtained. It was soon found, however, that this method was impracticable. Whether in the actual process of spraying or in the turbulence caused by the jet impinging on the walls and bottom of the chamber, the liquids dissolved, or absorbed, considerable quantities of air.

The effect was not lasting in the case of paraffin and almost negligible in the case of water; but the fuel oil and even more especially the Mobiloil BB were quite unsuitable/

unsuitable for further use for some time after being discharged into compressed air. The absorption became more pronounced at higher pressures both of liquid and air. Even when apparently badk to a normal state again, unless a considerable period of time was allowed to elapse, the fuel oil was found to give unsatisfactory results, and accordingly only a very few readings were taken, with considerable intervals intervening, which could be checked against those obtained for the discharge of fuel oil into fuel oil. For the same reason also, only a very few readings for Mobiloil BB into compressed air were taken. With the latter oil a longer interval had to be allowed for the recovery of its normal state. Owing to turbulence caused by the jet, air was frequently blown out through the outlet valve of the chamber instead of the liquid.

On account of the absorption and turbulence, the level of the liquid in the chamber did not remain steady and the method described on page 23 was therefore inapplicable.

For the experiments on discharge into liquid media, the spray chamber was removed from its stand and placed in a horizontal position. This was done to ensure that, even with a slight amount of air remaining in the chamber, the nozzle would be completely immersed. Readings in this case however were taken by the alternative method described on page 23.

Figs.27-32 pp.59-61, show the series of curves of discharge of water for various pressures of air, and lifts of the valve spindle. Similar sets of curves have been obtained for the discharge of paraffin and fuel oil, but these have not been added since they are practically the same/

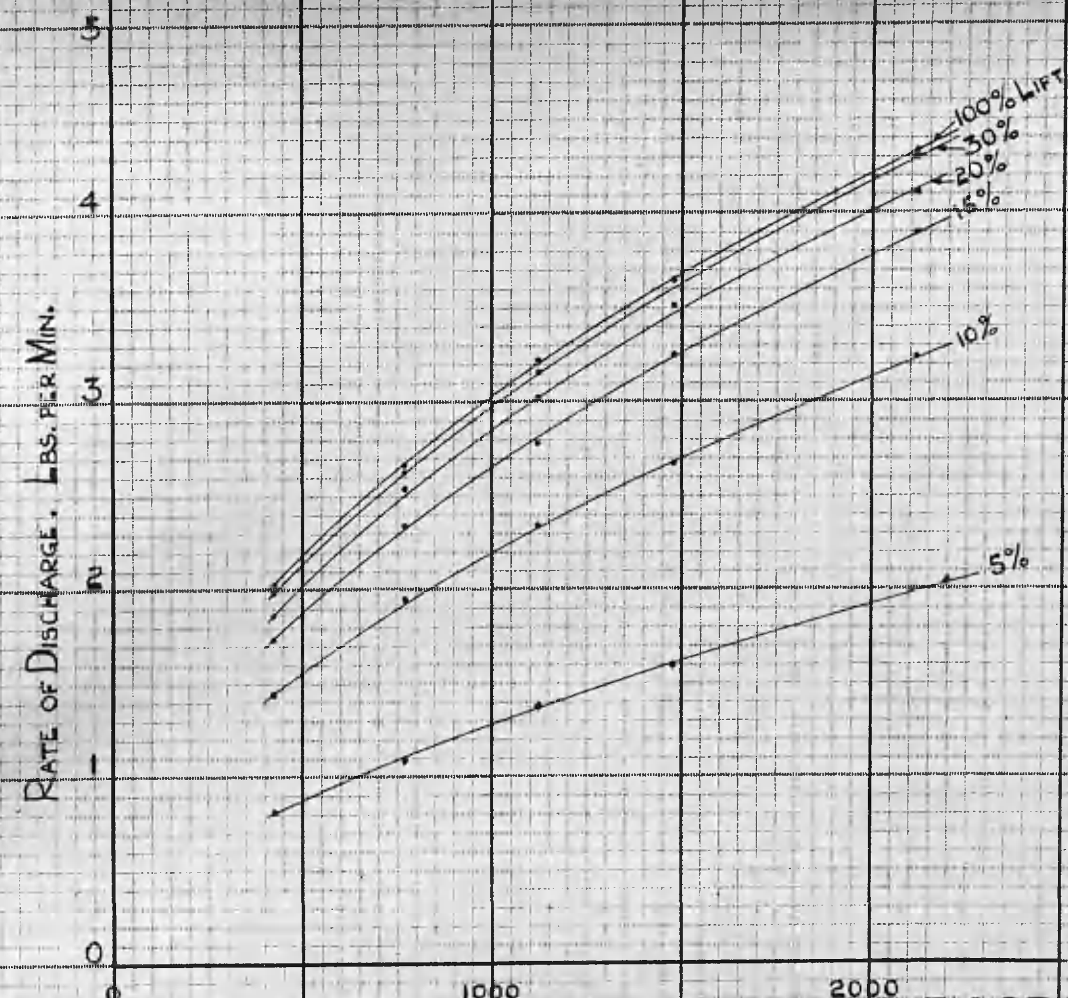


FIG.27.

Discharge of Water into Air at Atmospheric Pressure.

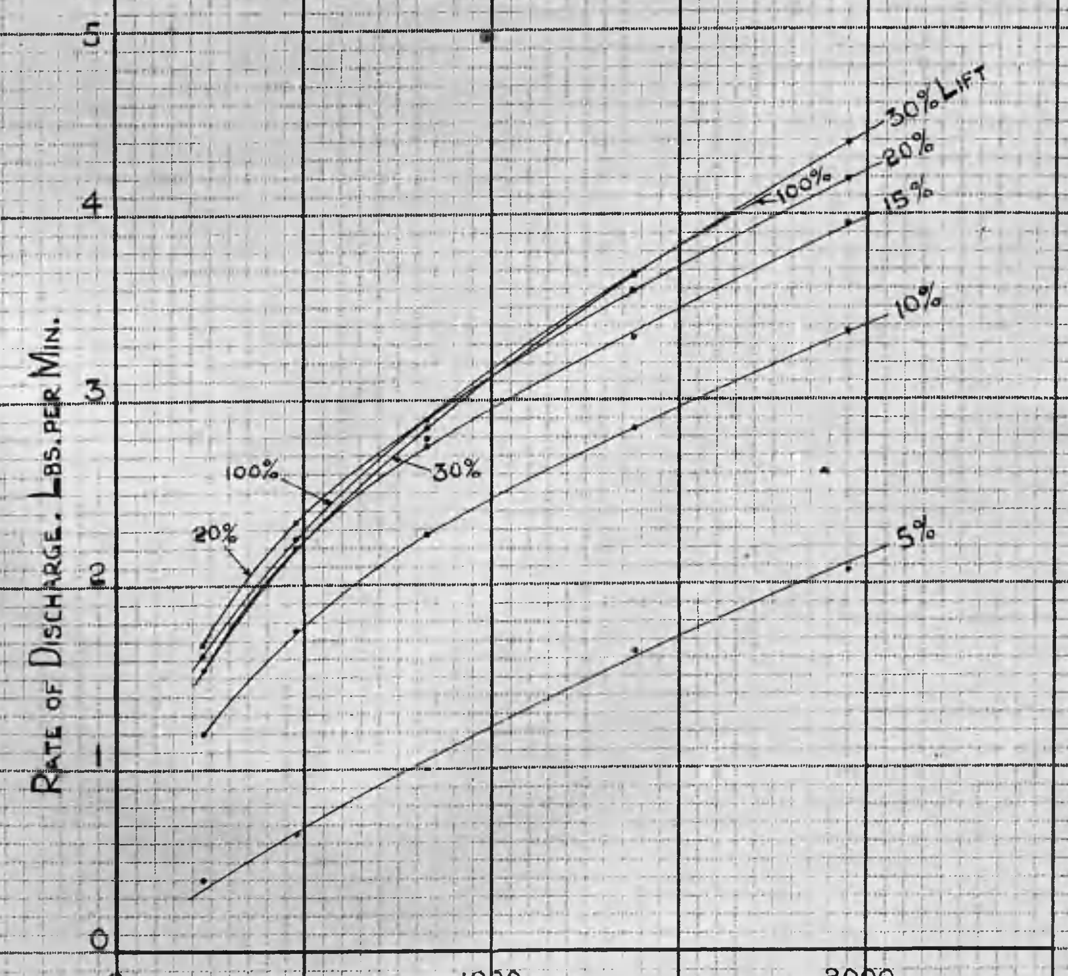


FIG.28.

Discharge of Water into Air at 300 Lbs. per In.^{2} Pressure.

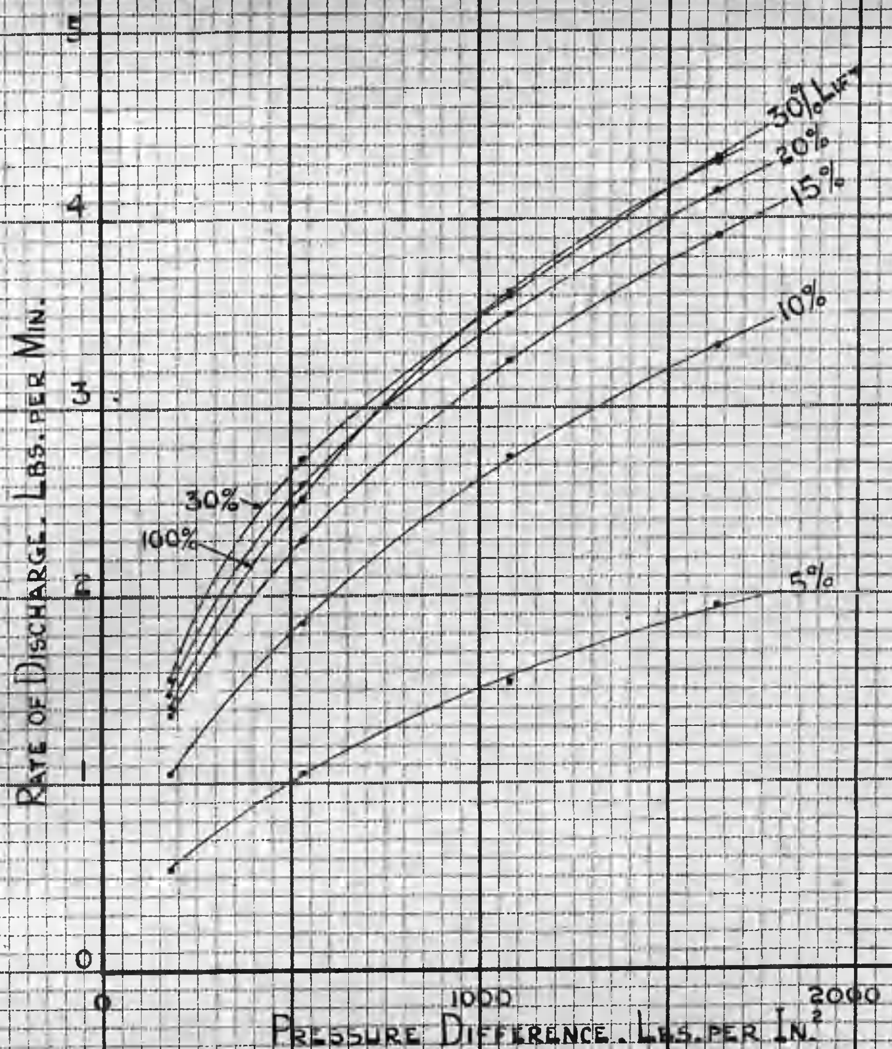


FIG. 29.

Discharge of Water into Air at 600 Lbs. per In.^{2} Pressure.

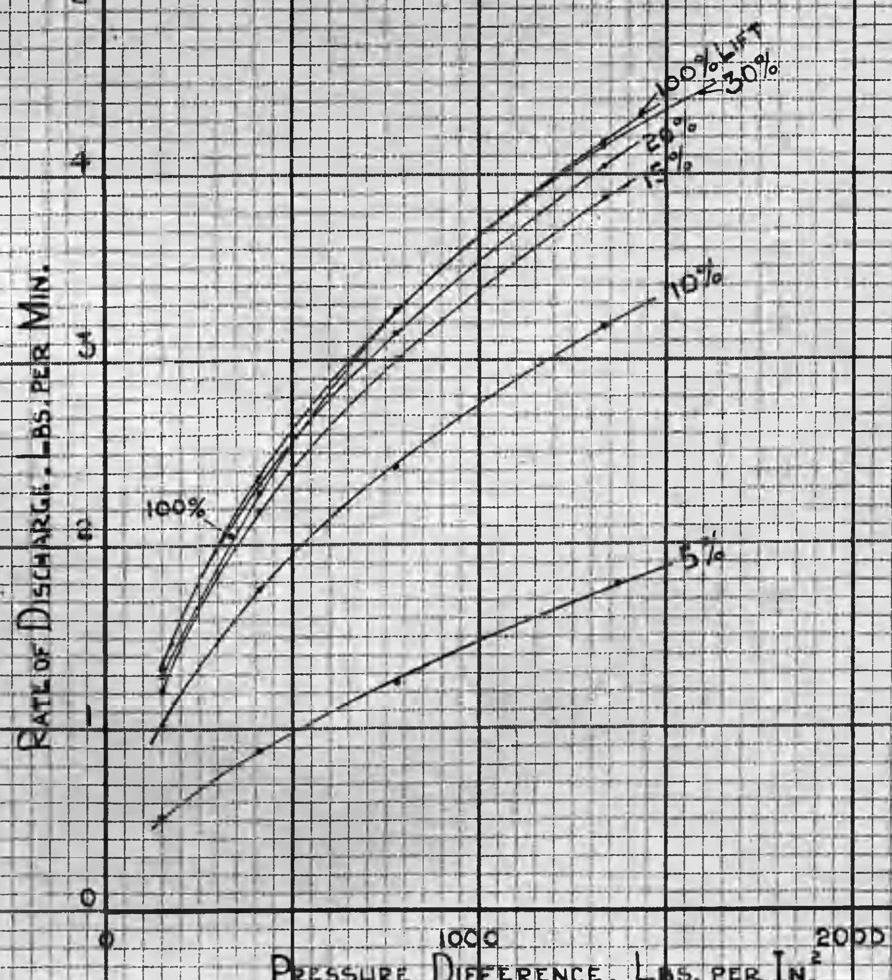


FIG. 30.

Discharge of Water into Air at 300 Lbs. per In.^{2} Pressure.

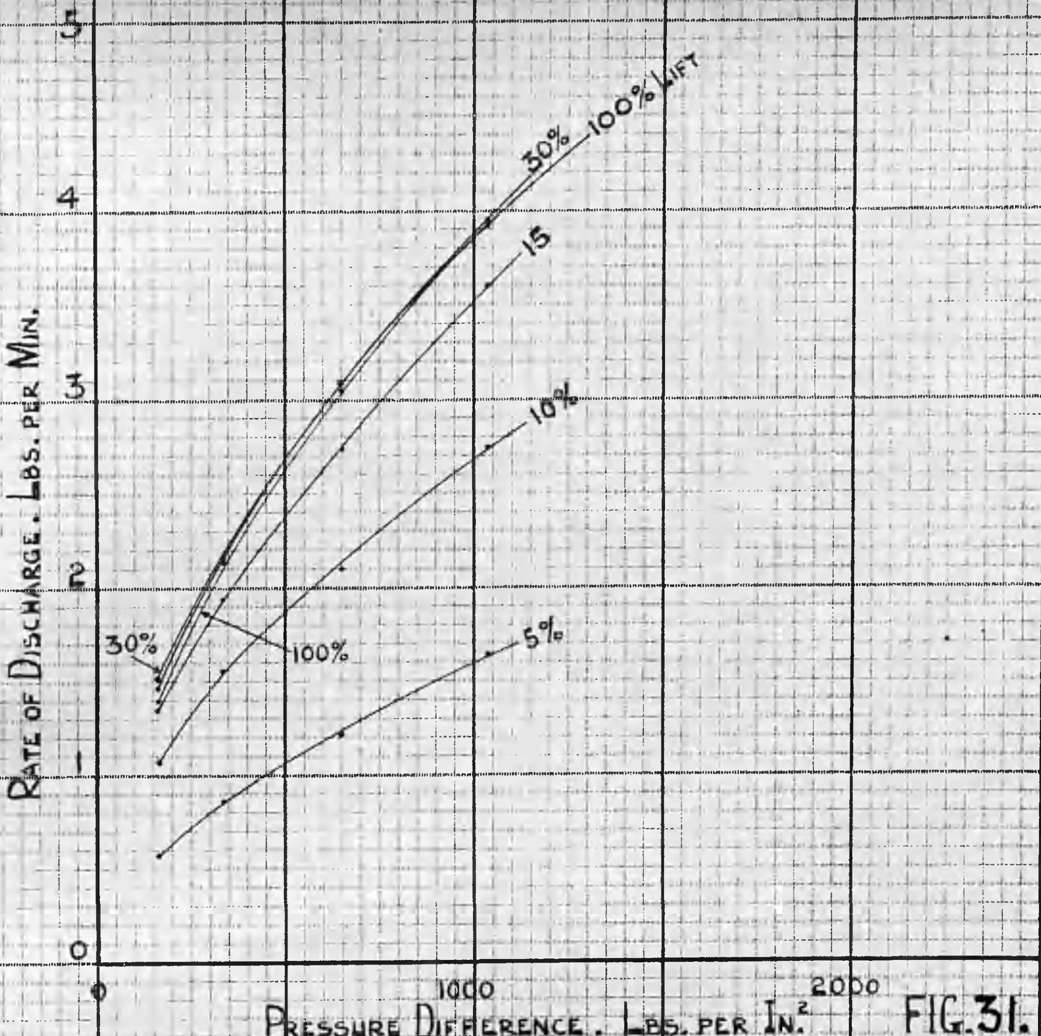


FIG. 31.

Discharge of Water into Air at 1200 Lbs. per In.² Pressure.

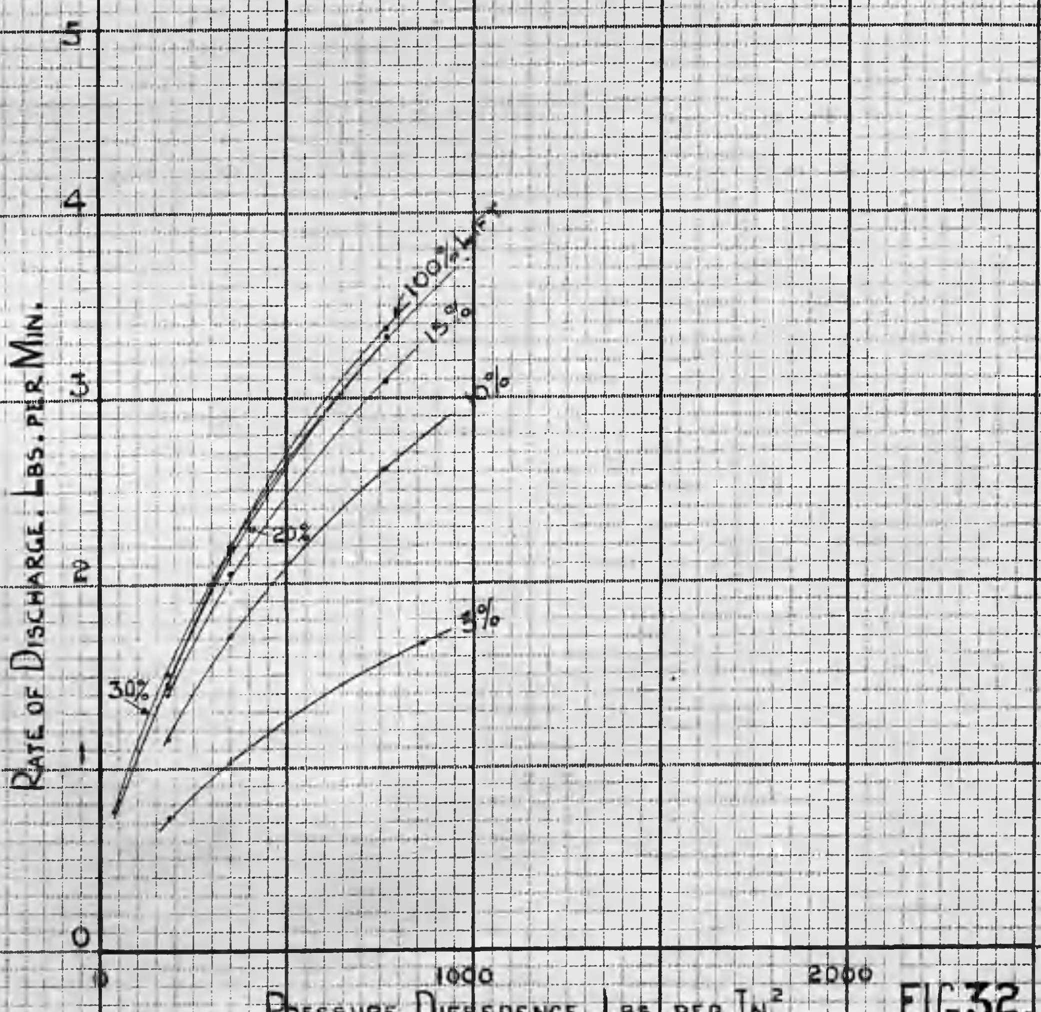


FIG. 32.

Discharge of Water into Air at 1500 Lbs. per In.² Pressure.

same as the series shown for water, and also, on account of the necessarily large number of readings for these tests, an increase in discharge was observed over their duration which was of sufficient magnitude to make any comparison between the liquids doubtful. The curves shown in Fig.27-32, pp.59-61, suggest that the effect of valve lift on discharge is similar for all air pressures but for low values of lift the discharge seems to increase slightly with increasing air pressure. Some of the lift curves are rather confused by what appears to be an instability-of-flow effect causing them to overlap.

The remaining tests were carried out at 100% lift and as already stated consisted of the discharge of water, paraffin, fuel oil and Mobiloil BB into air at various pressures, and then the discharge of the same liquids (excepting the Mobiloil BB) into media the same as the liquid being discharged. The results of these tests are plotted in Figs.33-38, pp.63-65.

Owing to the erratic nature of the readings of discharge, it has been found impossible to place them on any mathematical basis as was hoped for at one time, and it is only possible to remark on the general tendencies that are apparent.

In the experiments by Mr. W.F.Joachim already referred to on page 37, this subject of discharge of fuel oil into compressed air has been investigated. Hydraulic pressures up to 8000 lbs/in.^{2} and air pressures up to 1000 lbs/in.^{2} were used in these tests. One conclusion arrived at was that the coefficient of discharge of fuel oil "increases in the case of discharge into compressed air until the compressed air pressure equals approximately $\frac{3}{10}$ of the hydraulic pressure, beyond which pressure ratio it remains practically/

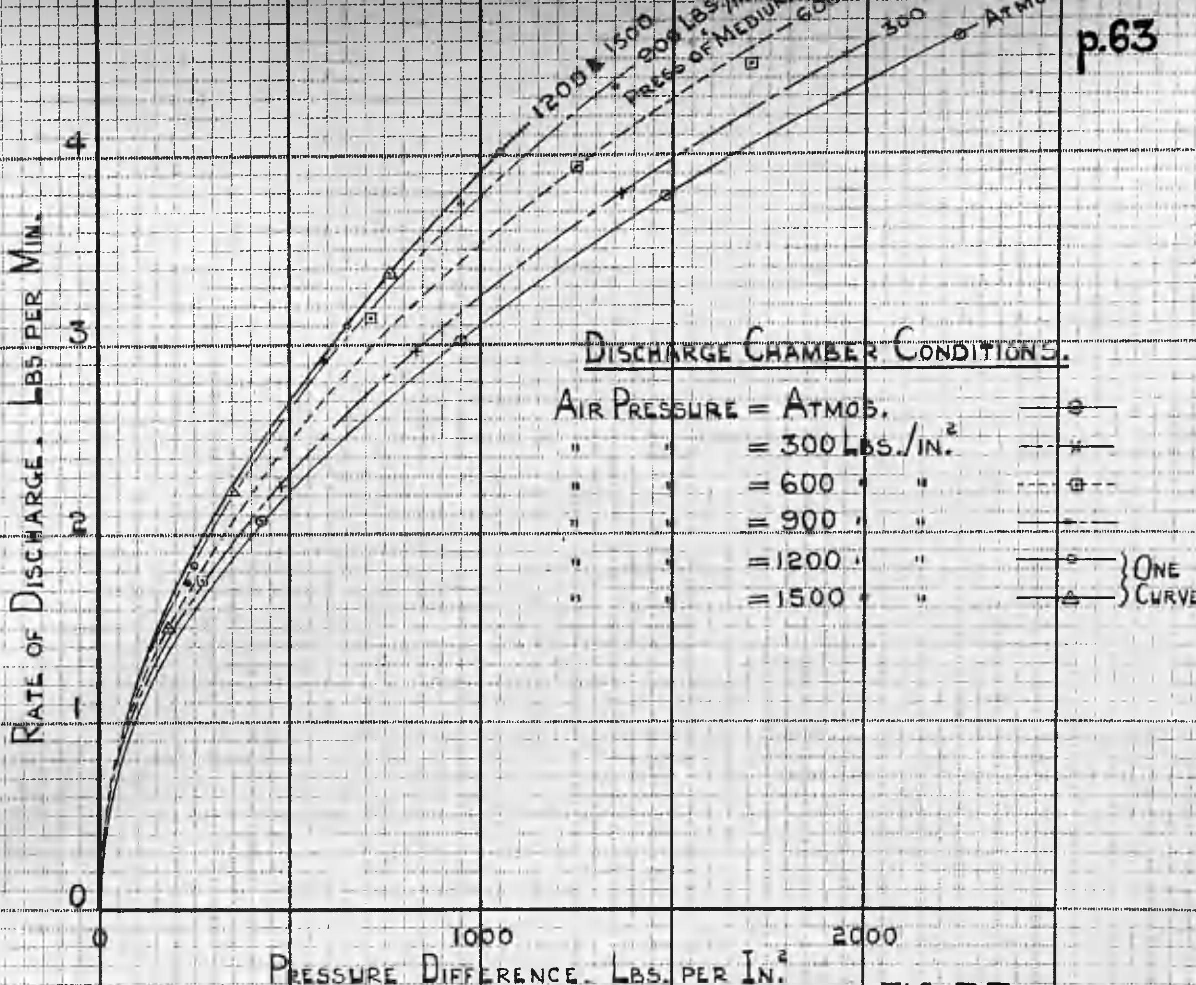


FIG. 33.

DISCHARGE OF WATER INTO AIR.

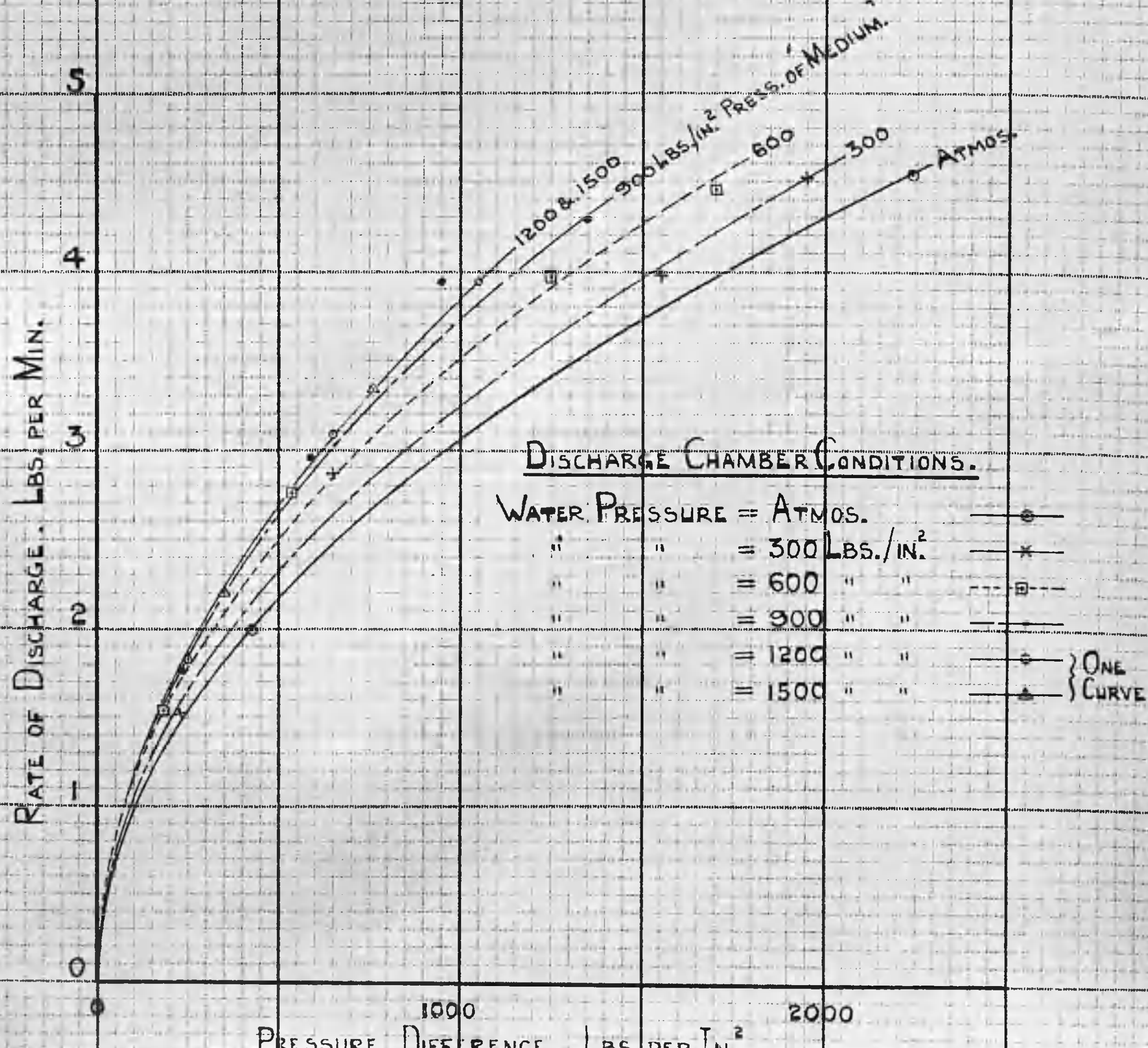
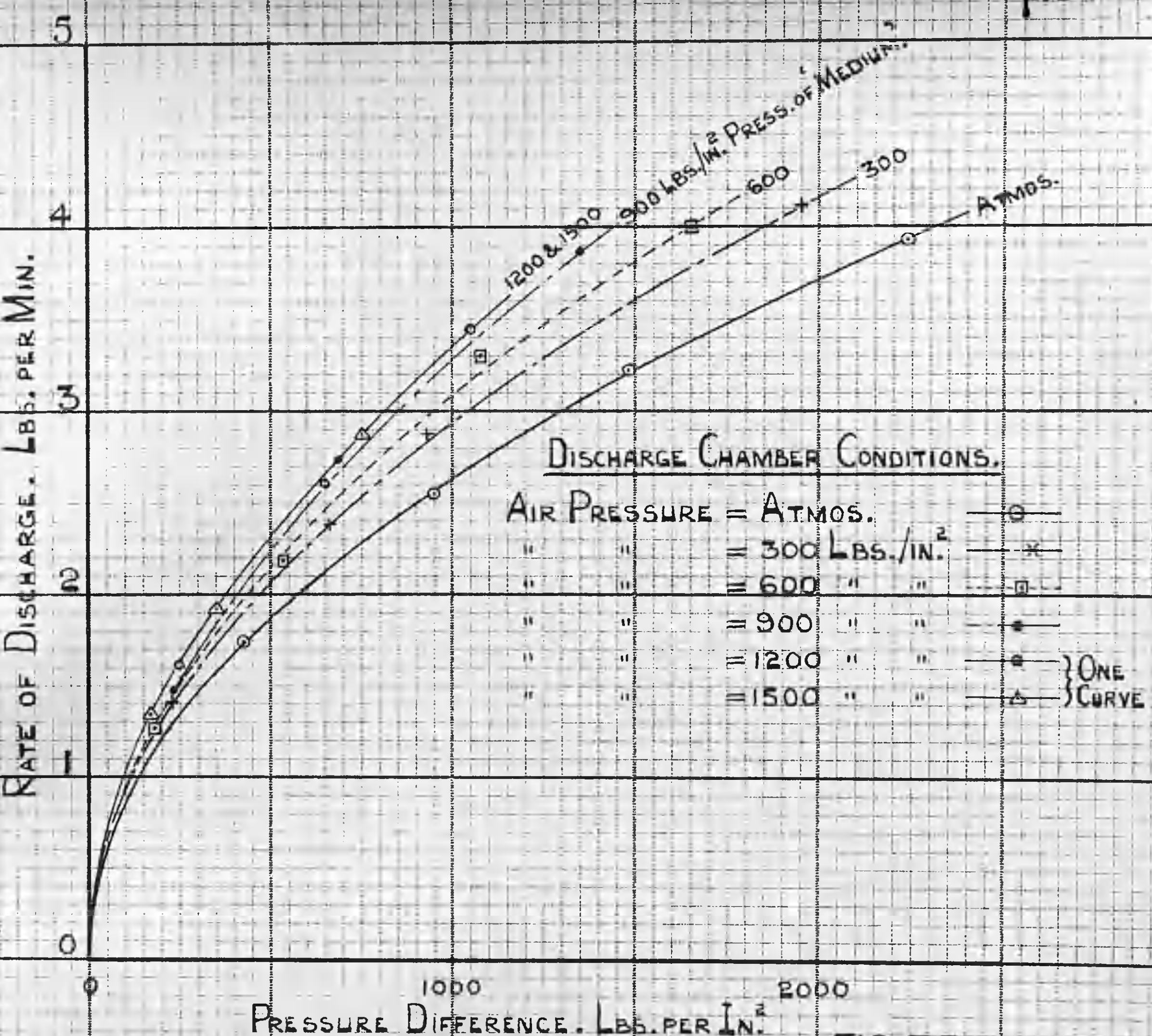


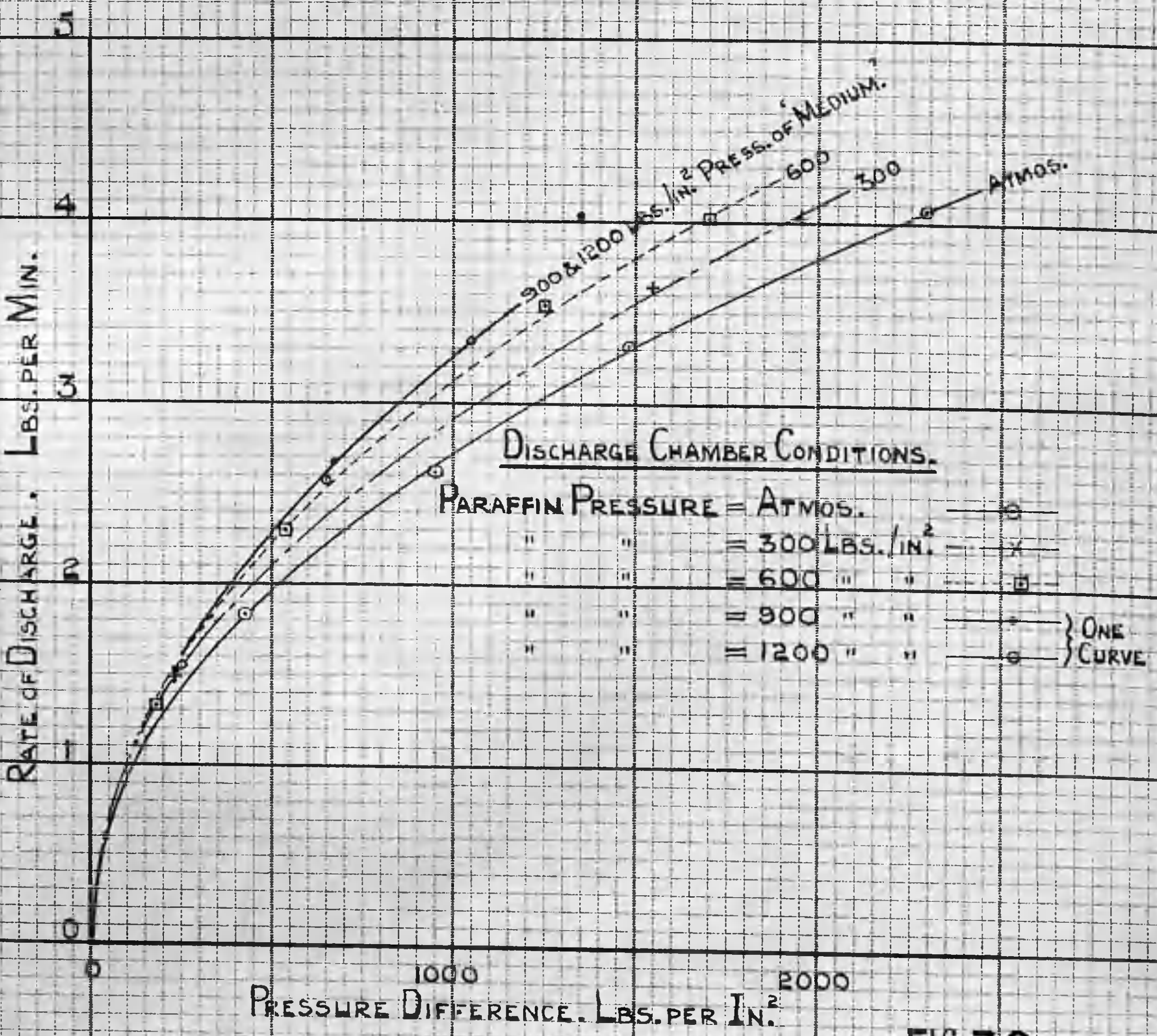
FIG. 34.

DISCHARGE OF WATER INTO WATER.



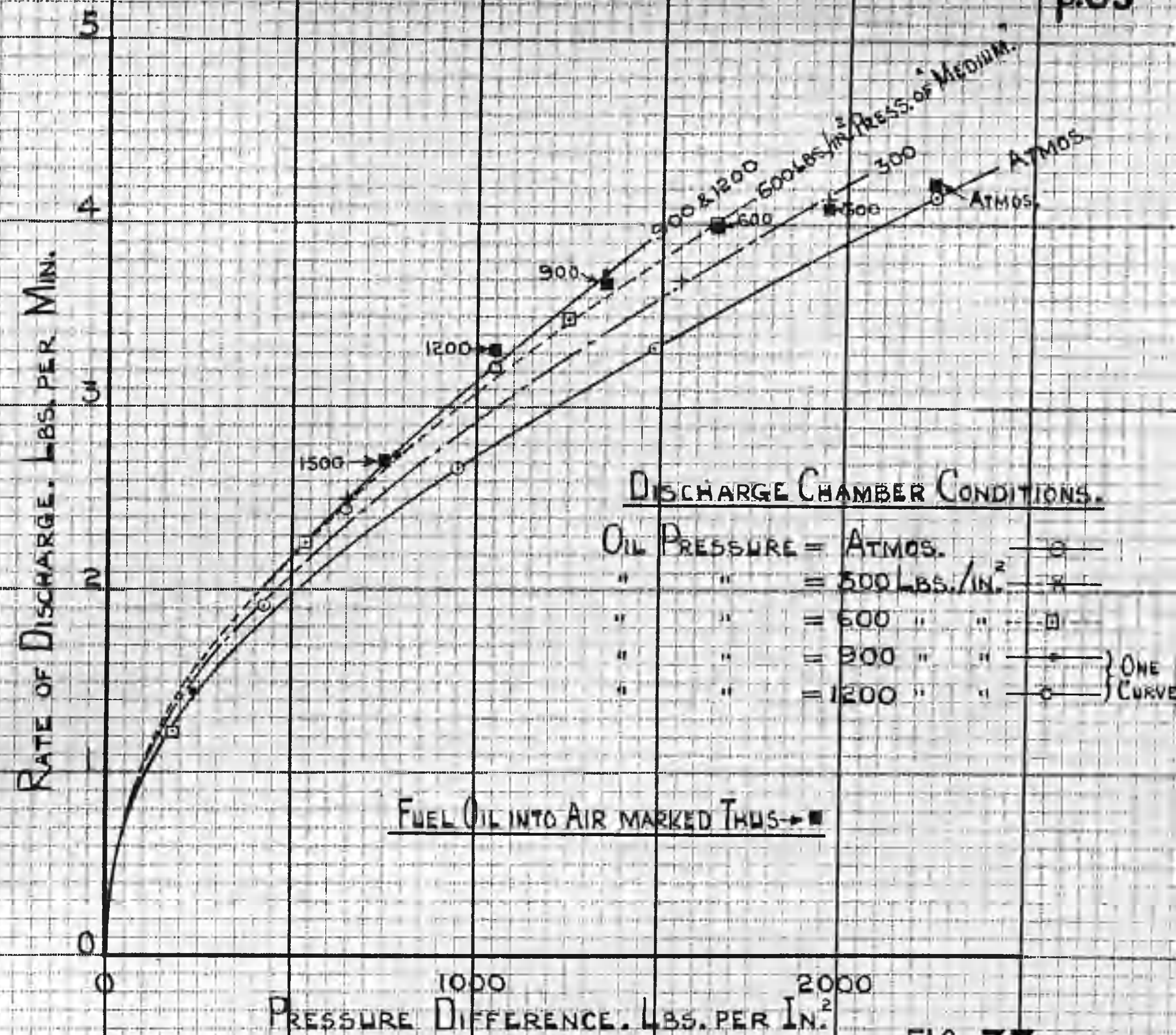
Discharge of Paraffin into Air.

FIG. 35.



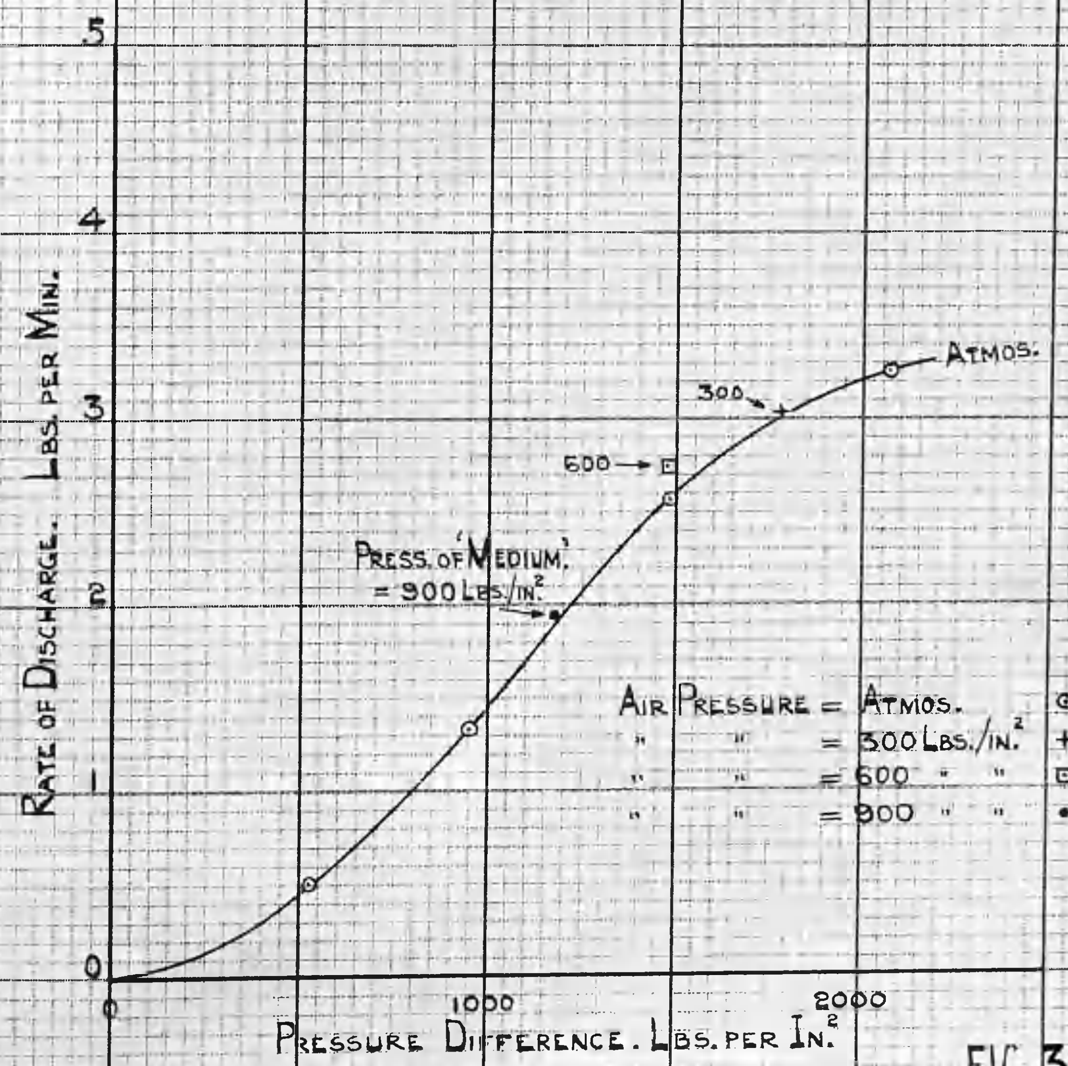
Discharge of Paraffin into Paraffin.

FIG. 36.



Discharge of Fuel Oil into Fuel Oil.

FIG. 37.



Discharge of Lubricating Oil into Air.

FIG. 38.

practically constant". The preceding curves of discharge are rather vague to allow a definite comparison but they appear to agree roughly with the foregoing theory; excepting perhaps the value '$\frac{3}{10}$'. In view of some of the results however, an alternative theory might be advanced, namely that for any liquid discharging, there is a definite limiting pressure in the medium beyond which the coefficient of discharge remains constant. To arrive at any really definite conclusions it is obvious that the extremely erratic nature of the experimental points would necessitate a great number of readings being taken and repeated.

It will be observed from the discharge curves (Figs.33-38 pp.63-65)that the increase of discharge coefficient for increase of pressure of the medium is much greater for water and paraffin than for fuel oil and least for lubricating oil. In fact for the latter it is almost impossible under the circumstances to state whether or not there is any increase. The above suggests that viscosity has some bearing on this discharge phenomenon.

The similarity to be observed between the discharge into compressed air and into compressed liquids would appear to justify the conclusion that the coefficient of discharge of a liquid into a medium is affected by the pressure of the medium but not by either the density or the viscosity of the medium. It will be evident that such an assumption simplifies greatly the application of these results to the actual injection of fuel into the engine. If this assumption together with the previous one regarding the viscosity of the discharging liquid be correct, they might possibly assist in the determination of/

of the theory of this discharge phenomenon. Such a theory of course may have been developed, although the writer is unaware of it.

The shape of the orifice might be another determining factor; and so also might be its dimensions. A fitting was made similar in outside appearance to the engine fuel valve, and designed to hold discs in which various forms of orifices could be pierced. The principal intention was to study the effect of the length of a circulat orifice having a diameter suitable to the resources of the pressure supply, on the coefficient of discharge into compressed air, but unfortunately time did not permit the carrying out of these tests. With a knowledge of the various possible types of flow at the 'vena contracta' as described in the paper referred to on page 36, the results if reasonably consistent, might have been of interest.

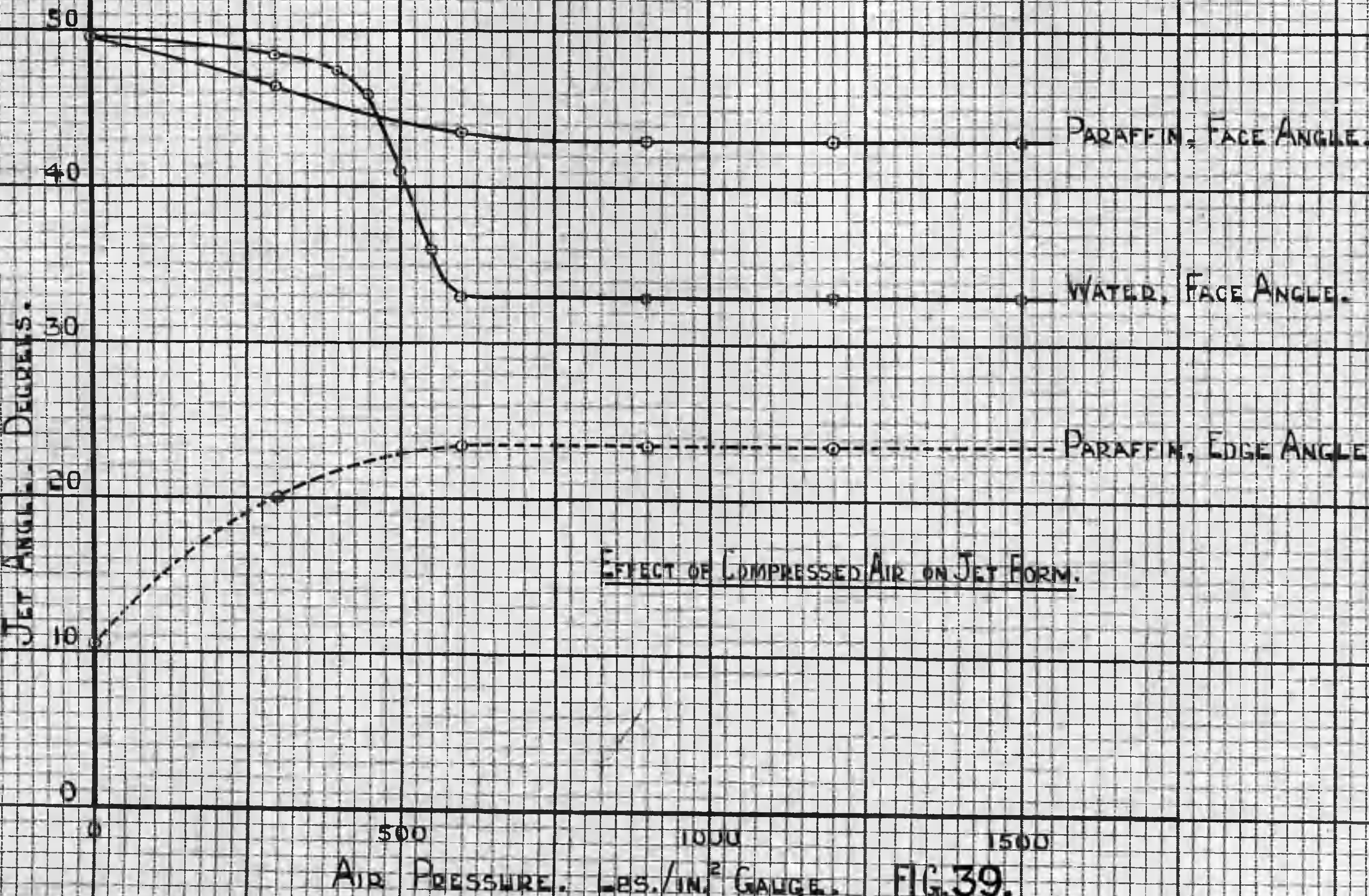
- - - - -oOo- - - -

EFFECT OF AIR PRESSURE ON JET FORM.

On account of the poor visibility obtained through the inspection windows of the spray chamber, only a few rough observations have been possible on the change of jet form caused by a change of air pressure. The air pressure may in this case have been only an indirect factor and air density the real cause of the change in jet form. Further elaboration of the experiments would have been necessary to separate the effects of these two properties of the air in the spray chamber.

The jet being of a simple fan shape, the readings were taken mainly of the angle contained by the two bounding edges of the plane of the jet. This for convenience has been termed the 'face angle' and the contained angle of the jet perpendicular to the plane of the jet has been termed the 'edge angle'. These angles were measured by means of a simple instrument consisting of two square pieces of celluloid placed against each other, and pinned together close to one edge in such a manner as to allow them to swivel about this point. A straight line was engraved on each piece and these lines could be adjusted to contain any angle between them. By placing the instrument against the outside of the window these two lines were set parallel to the edges of the jet and the jet angle was then obtained from them by means of a protractor.

Fig.39,p.69,shows the curves obtained by plotting the angles determined on a base of air pressures. These angles were measured for 100% valve lift, and at the most convenient hydraulic pressure from the point of view of visibility. Over a wide range, neither the valve lift nor the hydraulic pressure appeared to have any appreciable influence/



Effect of Compressed Air on Jet Form.

FIG. 39.

influence on the jet angles for either water or paraffin. It can be seen from the curves, that for paraffin the face angle decreased gradually as the air pressure increased; whereas the water held longer to the original angle and then dropped rapidly to about 67% of this. Within this region of rapid drop, inspection suggested an unstable condition of flow. From what could be seen of the jet of fuel oil through the windows, it appeared as though the angle varied only slightly, just as in the case of the paraffin.

The 'edge angle' of the jet was measured only for paraffin, it being the only liquid that could be seen reasonably well with the jet turned at right angles to the windows. Unlike the 'face angle', the 'edge angle' was found to increase with air pressure. The values obtained were also plotted in Fig.39. On inspection however, whereas the decrease in 'face angle' had the appearance of an actual change of flow of the liquids through the nozzle, the increase in 'edge angle' had that of a spreading of the jet after it was clear of the orifice, due to the resistance of the highly compressed air. Against this however, is the fact that, as is apparent from the curves, there is evidence of some relationship existing between the change of 'face angle' and the change of 'edge angle'. Further, there would almost appear to be a relationship between these angles, and the coefficient of discharge, except that the angles attain a constant value at about 600 lbs/in.^{2} air pressure whereas the coefficient increases to a value somewhat above this. More work would require to be carried out on this subject in conjunction with results of discharge before any definite conclusions could possibly be arrived at. For this purpose a much larger spray chamber would be/

be desirable.

Regarding the penetration of the jet into highly compressed air, it was rather interesting to find, during the experiments at an air pressure of 1500 lbs/in.^{2} that the energy with which the jet struck the surrounding vessel appeared to be very considerable, judging from the commotion observed within and the tendency on numerous occasions for the air to blow out of the valve at the bottom instead of the liquid being injected, even in the case of water which showed practically no tendency to absorb air under pressure.

APPLICATION OF DISCHARGE RESULTS TO THE DETERMINATION OF THE RATE OF INJECTION INTO THE COMBUSTION CHAMBER OF THE ENGINE.

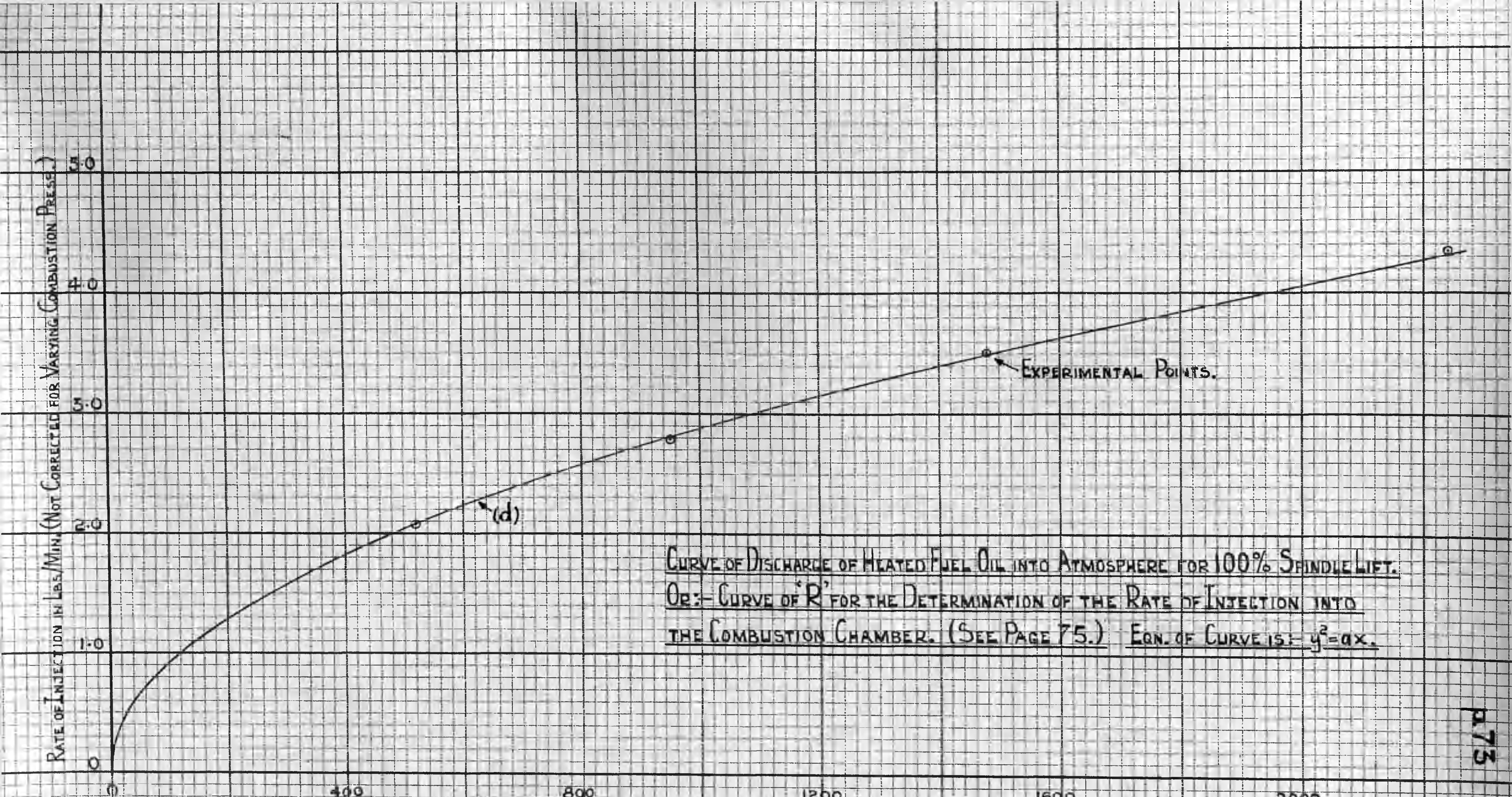
To simplify the application of the results of the foregoing experiments on discharge, certain assumptions were made.

The first assumption was that Fig.37, p.65, (showing discharge of cold fuel oil into fuel oil at various pressures) gives a correct comparison of the variation in discharge of heated fuel oil into compressed air or products of combustion. On consideration of the points that were actually taken of cold fuel oil into compressed air (Fig.37, p.65,) and the resemblance between Figs.33 and 34, and between Figs.35 and 36, p.64, the part of the assumption dealing with the medium appears to be justified. The part dealing with the fuel oil temperature was unlikely to involve more than a very slight error (caused, if any, by the effect of the difference in viscosities between heated and cold oil on the phenomenon of increase in discharge coefficient discussed on p.66).

The second assumption was, that at every value of pressure difference in Fig.37, p.65, the ratio

$$\frac{\text{Discharge into a medium at any given pressure}}{\text{Discharge into the same medium at atmospheric pressure}}$$

was a constant. This could obviously only be the case if all the curves were of the same mathematical form except for the constant factors. That is, if the equation for the discharge curve into atmosphere were $y^2 = ax$, then the other curves in Fig.37, p.65, would be correspondingly $y^2 = a_1 x$; $y^2 = a_2 x$; etc. Judging from the shape of the experimental curves this is probably not quite /



Rate of Injection in Lbs/Min. (Not Corrected for Varying Combustion Press.)
5.0
4.0
3.0
2.0
1.0
0
0
400
800
1200
1600
2000
Experimental Points.
(d)
Curve of Discharge of Heated Fuel Oil into Atmosphere for 100% Spindle Lift.
Or:- Curve of 'R' for the Determination of the Rate of Injection into the Combustion Chamber. (See Page 75.) Eqn. of Curve is:- $y^2=ax$.
Difference between Fuel and Combustion Chamber Pressures. Lbs. per In.^{2}

FIG. 40.

quite the case, but the error so involved is unlikely to be appreciable compared with the possible error due to the experimental points themselves.

The third assumption was that the variation of discharge with valve spindle lift as shown in Fig.22,p.48, was the same for discharge into any pressure of the medium. A glance at the diagrams for water into air pressures from atmospheric to 1500 lbs/in^2 on Figs.27-32, pp.59-61, show that this is not strictly correct.

The fourth assumption was (referring to Fig.22,p.48, again) that for every value of oil pressure, the ratio of

$$\frac{\text{Discharge at any given value of spindle lift}}{\text{Discharge at 100\% lift}}$$

was a constant. It was on this assumption that the curve of heated oil in Fig.25,p.51, was obtained from table 4, p.52. Any errors involved by these last two assumptions should be negligible as they only refer to a small fraction of the total oil injected; namely the quantity injected while the valve was opening and again while the valve was closing.

For application of the discharge results, based on the foregoing assumptions, to determine the rate of injection of fuel into the combustion chamber, the six following curves were used.

Curves obtained from engine test (see Figs.48-52,pp.113-120)

(a) Variation of fuel oil pressure behind the nozzle on a base of crank angle.

(b) Variation of combustion chamber or cylinder pressure on a base of crank angle.

(c) Variation of valve spindle lift on a base of crank angle.

Curves obtained from discharge experiments:-

(d) Discharge of heated fuel oil into atmosphere on a base of/

of gauge pressure behind nozzle. (Fig.40,p.73,). The base scale of this curve was termed 'pressure difference' as the curve was only intended for the present purpose of determining the rate of injection. The curve (d) (parabolic in form) formed the basic curve for this determination,and curves (e) and (f) (below) the auxiliary curves giving the correction factors for the modification of the basic value of 'rate of injection' R obtained from curve (d) by allowing for the varying conditions of pressure within the cylinder and lift of the valve spindle.

(e) Discharge of cold fuel oil for any condition of 'pressure difference' on a base of pressure of 'medium'-oil (discharge into atmospheric pressure for the same pressure difference being taken as unity) (Fig.26,p.51,). Curve (e) was built up from the curves in Fig.37,p.65, by taking, at various values of pressure difference, the ratio of discharge into each 'medium' pressure to the discharge into atmospheric pressure and averaging these ratios corresponding to each pressure of the medium. By the stated assumptions, curve (e) applies to the injection of heated fuel oil into the combustion chamber for any conditions of pressure and 'spindle lift'.

(f) Curve (f) (see Fig.25,p.51,) was obtained in the same manner from the curves in Fig.22,(see table 4, p.52) as that in which curve (e) (above) was obtained from the curves in Fig.37,p.65. The formation of curve (f) has been discussed earlier on p.50. By the assumptions, curve (f) in Fig.25,p.51, represents the variation with valve spindle lift, of discharge into the combustion chamber, under any pressure conditions (taking the discharge at 100% lift under the same pressure conditions as unity).

The/

The practical application (based on the foregoing assumptions and experimentally derived curves) to determine the rate of injection of fuel oil at any given value of crank angle during a particular test is as follows.

From the particular set of engine test curves (see Figs.48-52 pp.113-120) determine (1) Oil pressure (2) Valve spindle lift, and (3) Cylinder or combustion chamber pressure.

Subtracting the value of cylinder pressure from oil pressure gives the pressure difference, and with this value the basic value of discharge R lbs/min. is obtained from curve (d) in Fig.40,p.73 (see table 13, p.143).

Next apply the value (2) of spindle lift to curve (f) in Fig.25,p.51, and obtain the lift factor f_1 (see table 13, p.143).

Then apply the cylinder pressure value (3) to curve (e) in Fig.26,p.51, and obtain the cylinder pressure factor f_2 (see table 13, p.143).

The rate of injection $\frac{dw}{dt}$ at any instant is then given in lbs/min. by

$$\frac{dw}{dt} = R \times f_1 \times f_2 \quad \text{(see table 13, p.143).}$$

During the period when the valve lift exceeds about 30% the factor f_1 becomes practically unity and the expression becomes

$$\frac{dw}{dt} = R \times f_2$$

- - - - - - - - - - - - -

PROCESS OF INJECTION OF FUEL.

-oOo-

OIL PRESSURE INDICATOR.

Indicator.

The complete indicator outfit with which the variation of pressure in the oil behind the nozzle was obtained, consisted essentially of two parts; viz. (1) the indicator proper, a spring loaded piston with an electrical contact device, (2) a rotary electrical contact on the engine camshaft.

Fig. 41, p. 78, shows that part of the arrangement incorporating the spring loaded piston and contact gear. It was designed to be screwed vertically into the filter at the entrance to the fuel valve (see Fig. 2, p. 7), but obviously could function equally well in any other position. The body of the instrument is made up of two parts, the lower containing the cylinder and piston and the upper, the spring adjusting screw, contact etc. The action for which the indicator was designed was as follows.

A piston of about .1456" dia., whose axial motion was restricted to between 1.5 and 2 thousandths of an inch by an upper and a lower stop, was held down against the latter by a coil spring. The spring load could be adjusted to any desired amount by means of an adjusting screw (before use the instrument had to be calibrated). Assuming that the spring load was adjusted somewhat below the maximum load exerted by the oil, then at a certain instant during a rise in oil pressure the piston would rise against the upper stop. During a fall in oil pressure the reverse would take place. Electrical contacts were so arranged that at the lower and upper positions of the piston a circuit was closed, but with the piston in any intermediate position the circuit was broken. A rotary contact on the/

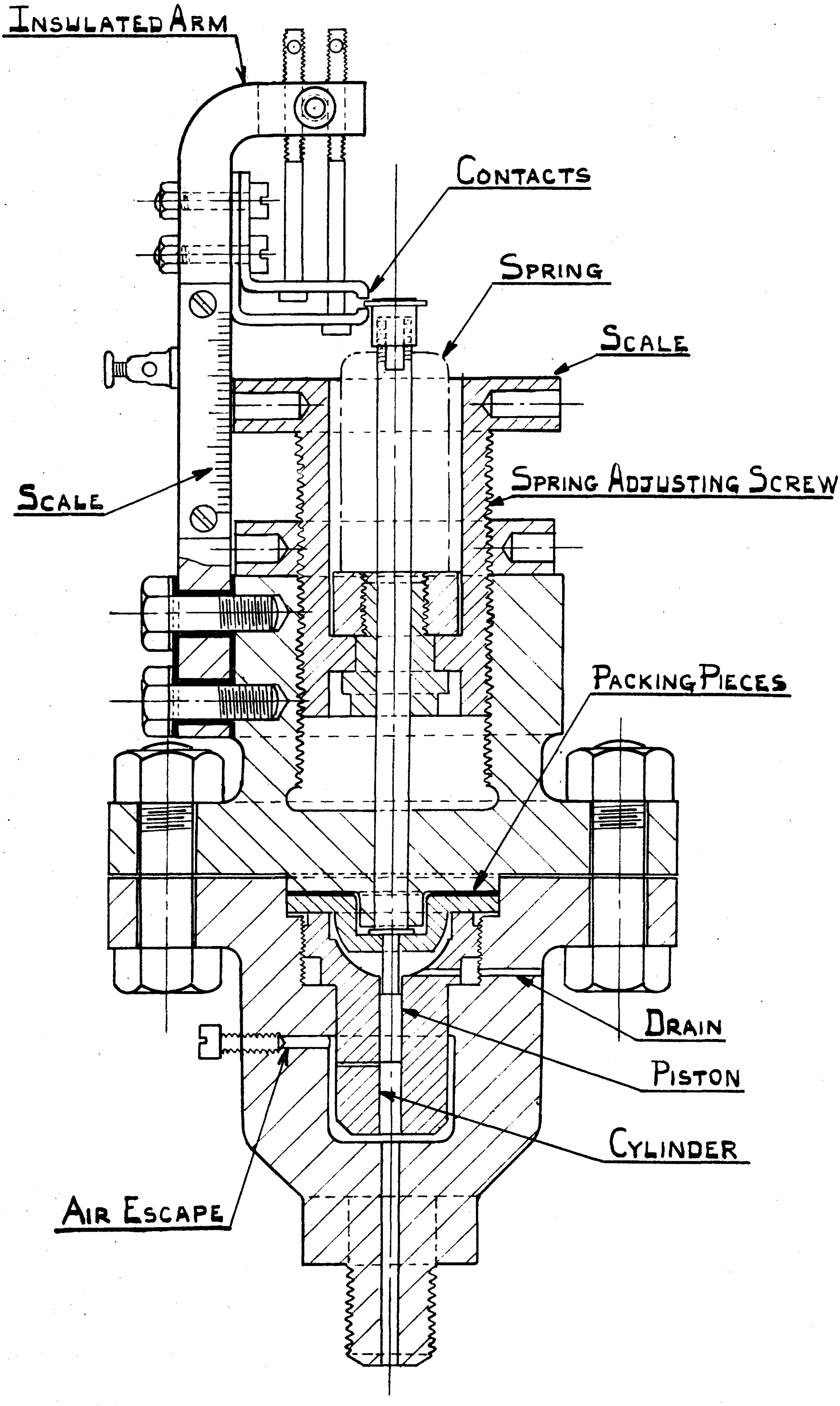


FIG.41. OIL PRESSURE INDICATOR.
(Full Size)

the camshaft (see Fig.42 p.84) which could be adjusted to act at any point during the complete engine cycle, was used to 'search' for these two breaks in the circuit. It was originally intended to use a millevoltmeter for the purpose of indicating the existence, or otherwise, of a current in the circuit, but it was found on trial that wireless earphones were more suitable and this method was accordingly adopted. By adjustments of the spring load on the piston, any number of values could be obtained which, when plotted, would give the desired curve of oil pressure.

The piston, cylinder, and spring were standard parts for a 'Maihak' engine indicator. The cylinder was screwed into the lower half of the body against an oil tight face, and to assist the thread to withstand the high pressures possibly exerted,a spigot on the upper half was bound down on the top of the cylinder by the main flange bolts. Held between the spigot and the top of the cylinder was a small cup-shaped component which acted as the lower stop for the piston. A projection on the underside of the upper half of the body acted as the upper stop. By a careful adjustment of the thin packing between the spigot of the upper half, and the cup-shaped part, the axial movement of the piston was limited to between 1.5 and 2 thousandths of an inch.

The spring was hooked to the upper end of the piston rod in the usual manner, and the lower end was held by a pin screwed into the lower end of the adjusting screw. Clearance was allowed between this pin and the latter, sufficient to allow the spring to remain stationary, if necessary, while the latter was being turned. The adjusting/

adjusting screw was provided with a knurled locking ring. The head of the screw was marked off at intervals of 5 degrees round the outer top edge, and this scale was read in conjunction with a brass scale fixed on the arm holding the contacts. This brass scale was calibrated to complete turns of the adjusting screw.

The fixed elements of the electrical contact were completely insulated from the rest of the instrument. A small standard spring cap which screwed on to the end of the piston rod formed the moving element (or 'earth') of the contact. Some difficulty was experienced in truing up the rim of this cap sufficiently, and in fact it was the inaccuracy of this rim which determined the 1.5 to 2 thousandths of an inch movement of the piston.

In order to obviate the formation of air locks on the underside of the piston, a small hole was drilled through the cylinder wall just below the piston and another through the body of the instrument. This second hole was stopped by means of a small screw which could be readily slackened back to allow any trapped air to escape.

Any oil leaking up past the piston was drained off automatically through the upper small hole in the side of the body, after flowing through a small hole out of the cup-shaped portion. This oil was thus prevented from rising sufficiently to foul the surfaces of the piston 'stops' immediately above. Indeed this was the reason for having this separate piece as the lower stop, rather than merely the top of the cylinder. It was thought that with such a small movement, the presence of a viscous oil between these faces might to some extent lower the efficiency of the instrument.

Since/

Since this indicator has been made, it was found that indicators, working on similar lines, and intended for high speed engine work, have been manufactured for some considerable time. Those referred to are of the diaphragm type, the motion of the latter being restricted by stops - one on each side of the diaphragm. Instead of the spring used in the present indicator, the load to balance the pressure being measured is obtained by a variable air pressure. Before this piston type already described had been decided upon, the practicability of the diaphragm was considered, but owing to the possibility of trouble due to the presence of oil between the large surfaces of the stops and of the diaphragm (a trouble which, of course, would not exist in the case of gases for which these indicators were intended) that type was considered unsuitable. The idea was to have a diaphragm reinforced in some manner by an additional adjustable spring.

As the writer had no idea of the rapidity of the pressure rise in the oil during injection it was thought that an ordinary type of engine indicator would probably be out of the question. Nevertheless, it appears that the latter has been used for this purpose. An example is to be found in an article describing a Richardson, Westgarth oil engine in "Engineering" March 8, 1929. The diagrams of oil pressure given therein show what might be taken to be a slight indicator vibration at the end of the sudden pressure rise, but for an indicator of this type (under such working conditions) the diagrams appear to be remarkably good.

It is clear however, that a diagram with a very rapid rise, even if free from vibrations, might easily be inaccurate if taken by an indicator having moving parts/

parts of considerable inertia, such as the common piston type, or perhaps to a much modified extent, the usual optical types.(piston or diaphragm). Apart from the vibrations and inertia of the piston type however, in order that a diagram of reasonable dimensions might be obtained, the displacement of the piston should probably be such as to affect appreciably the elasticity of the high pressure system.

The instrument as shown in Fig.41, p.78, worked satisfactorily except in one respect; namely that of static friction of the piston. The adjusting spring should not have been hooked on to the top end of the piston rod but by some means should have been attached to the latter as close to the piston as possible. The friction was clearly caused by the piston rod being pressed laterally at the top by a force exerted by the spring and caused by a lack of concentricity in the latter at the upper end. This appears to represent a similar fault in any indicator with the spring and piston so arranged. Theoretically the piston of the present indicator should have left the lower stop at a lower pressure than that at which it left the upper, owing to the difference in tensions of the spring at these positions of the piston rod. The friction on the average, was found to be such that the piston left the lower stop at an oil pressure practically the same as that at which it left the upper stop. Although the friction force was apparently not quite a constant quantity, the indicator was calibrated on the assumption of equal pressures, with what was thought to be sufficient accuracy.

Electrical Contact on Camshaft.

As has already been indicated, the instants at which/

which the circuit was broken at the contacts of the indicator piston had to be found by means of a contact on the camshaft. It was necessary for this purpose to have a means of obtaining a very short contact, and more especially one with a definite instant of commencement, for any point in the engine cycle. A spring brush rubbing on a rotating disc with a metal inset was first tried but it was found that the length of the arc of the inset had to be considerable before a proper contact was obtained. This therefore, did not give the desired definiteness to the contact. With the form of contact then devised to suit the requirements, the length of the period of contact could be reduced to almost zero without reducing its actual efficiency. And this was because the most definite part of the contact was at its point of commencement.

This contact is shown in Fig. 42 p. 84. It consisted mainly of (a) a frame, or bracket made in two pieces, and bolted to the bed plate of the engine, and insulated from the latter by strips of insulating fibre, and vulcanite bushes, (b) a solid brass ring clamped between these two brackets. This ring was marked off round the edge of one face in degrees of crank angle and could be rotated to, and clamped in, any position (the position being identified by the reading of a fixed pointer on the scale), (c) a hardened steel rocker pivoted on a hardened steel pin on the brass ring, and an adjustable stop for this rocker also mounted on the brass ring, and (d) a split steel ring clamped to the camshaft and carrying two steel hammers. The larger of these hammers was comphetely insulated from the ring, whereas/

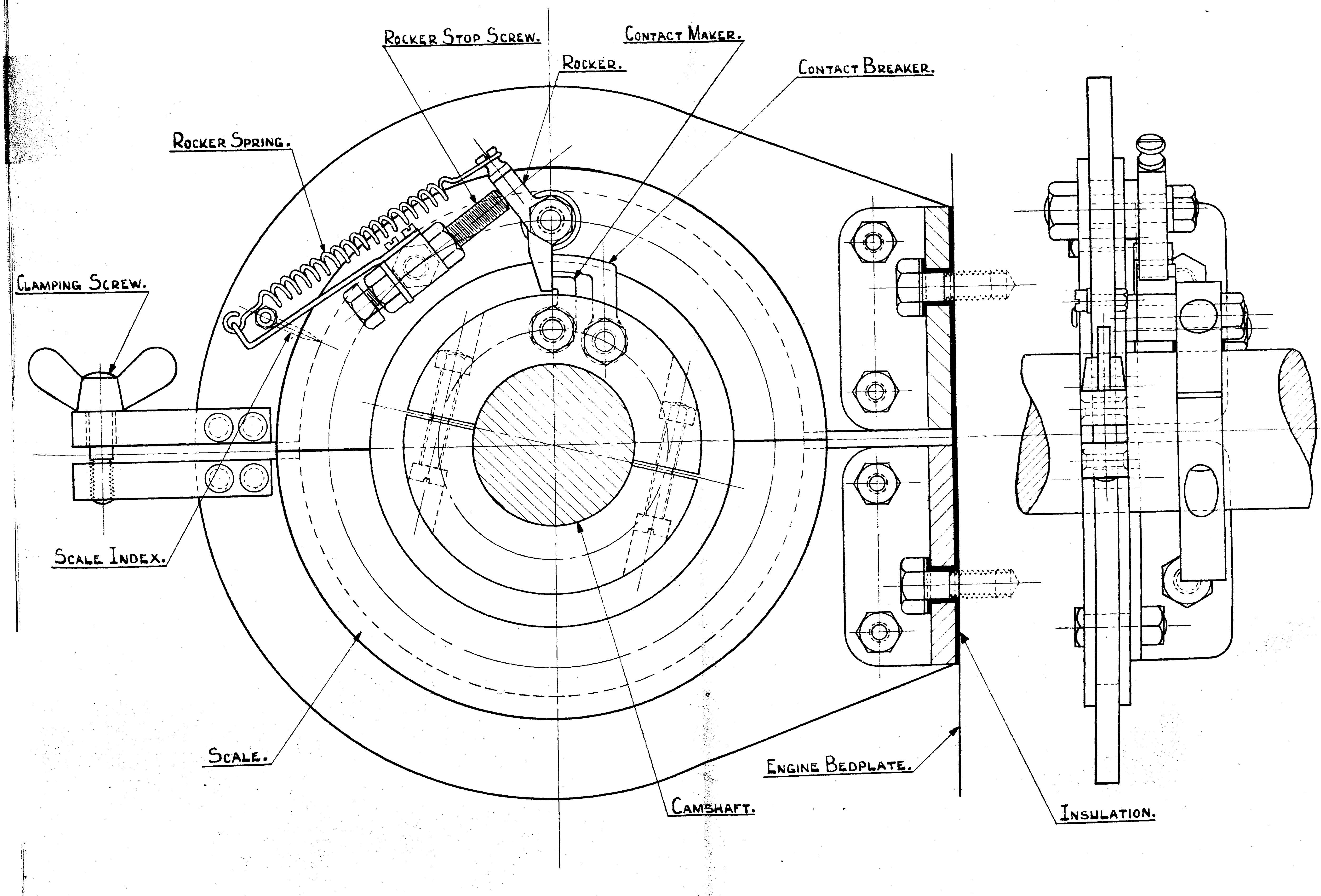


FIG. 42. CAMSHAFT ELECTRICAL CONTACT.
(Full Size)

whereas the smaller was not and formed the contact to 'earth'.

The steel rocker carried on the brass ring was held firmly against the adjustable stop by a long coil spring. The two hammers and the rocker were so adjusted that as the former rotated with the camshaft the smaller first touched the face of the rocker and made the contact and the larger immediately followed and sharply broke it. The period of contact therefore could be readily adjusted to any amount, no matter how small. The rigidity of the apparatus guaranteed a definite commencement of the contact period.

A pair of wireless earphones and some pocket lamp batteries completed the pressure indicating apparatus. For a preliminary adjustment of the contacts when the engine was not in motion, a resistance, for the benefit of the earphones was also desirable, but could afterwards be dispensed with.

Owing perhaps to a slight eccentricity of the frame of the above apparatus in relation to the camshaft or to a lack of uniformity of rotation of the camshaft relative to the engine crankshaft, it was found that the scale of the brass ring did not quite correspond with the existing scale of crank angle marked on the periphery of the flywheel. In order to eliminate this source of error the apparatus was calibrated with respect to the scale on the flywheel.

- - - - - - - - - - - - -

MOTION RECORDER OF FUEL VALVE SPINDLE.

In order to obtain complete records of the movement of the fuel valve spindle during the period of fuel injection, the following apparatus in the form of another electrical contact device was designed to operate in conjunction with the camshaft contact just described. Fig.43,p.87, shows the arrangement of this piece of apparatus. (See also Fig.2, p.7.)

A stout bracket made from flat steel bar was bolted by means of a projecting lug, to the end of a cast iron tray protruding from underneath the fuel valve. For additional rigidity a second piece of flat bar was used to connect the above bracket directly to the back of the fuel valve itself. One end of the bracket was slotted to accommodate a rectangular piece of steel which carried the contacts. This steel plate was rigidly supported at one end in an adjustable bearing. By means of this bearing the plate could be swivelled; and could also be moved endways if desired, by an adjusting screw and nut. The other end of this plate was clamped between two screws. One of these acted as a locking screw and the other was fitted with an indexed disc against which was placed a pointer, screwed to the main bracket. Adjustment of the position of the steel plate, in the direction of the axis of the valve spindle was obtained by means of this indexed screw. A small steel contact piece was clamped to the end of the extension of the valve spindle and a square hole cut in the steel plate sufficient to clear this. Bolted to and insulated from the steel plate on one side of this hole was a small brass plate. Bridging the hole was a strip of spring brass, bolted to and insulated from the steel/

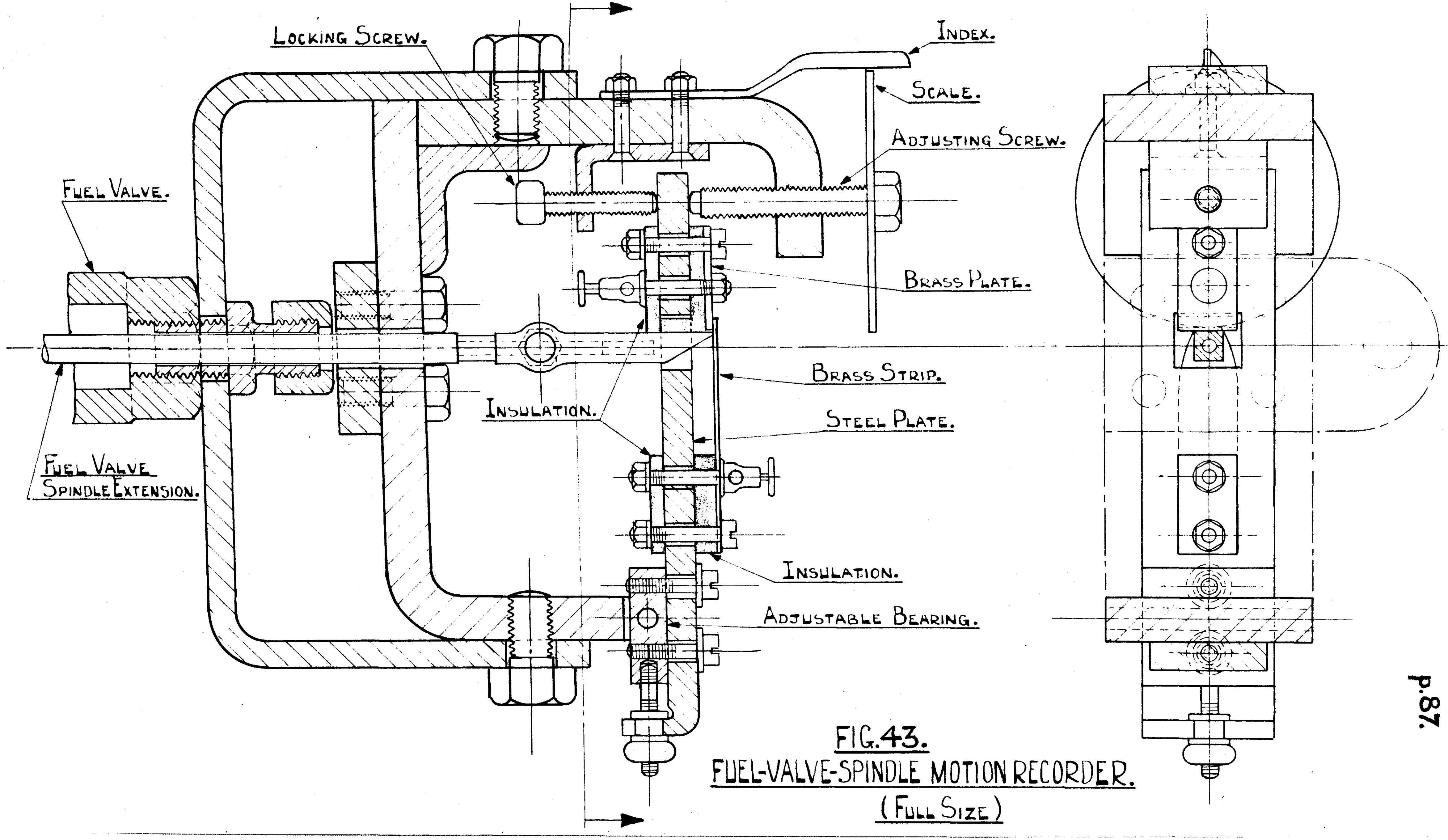


FIG. 43.
FUEL-VALVE-SPINDLE MOTION RECORDER.
(Full Size)

steel plate at one end and resting lightly against the brass plate at the other. Small blocks and bushes of vulcanite were used as insulation. The brass strip and the brass plate were each fitted with a terminal screw.

The anticipated action of the apparatus was as follows. As the spindle lifts, assuming everything properly adjusted, the contact piece strikes the brass strip, and as this is resting on the brass plate, an instantaneous electrical connection is formed between the latter and the 'earth' (i.e. the valve spindle). The same takes place on the return stroke of the spindle. If therefore, the brass plate be connected through the earphones and batteries to the camshaft contact, the instants of these two contacts with respect to crank angle should be obtained. By adjusting the position of the plate holding the contact strip, along the axis of the spindle be means of the indexed screw, it should be possible to obtain a record of the complete movement of the spindle from the moment of opening to that of closing.

Unfortunately the desired action, as described above, did not take place in practice. No suggestion of a contact could be heard in the phones, even with seven or eight batteries in the circuit. The reason for this must have been that the period of contact, which was probably almost instantaneous (and this if anything was the special feature of the apparatus) was really too short to allow the passage of sufficient current even to affect the earphones. Variations due to governing also may have had some influence. Two alternative simplifications were therefore adopted - (a) by connecting the brass strip to the circuit and detecting the period of contact between it and the end of the spindle/

spindle and (b) by connecting the brass plate to the circuit and the brass strip to 'earth' on the engine frame, and detecting the period when the circuit so formed was broken by the intervention of the valve spindle. It will be clear that these two methods used in conjunction, and checking with one another, are practically equivalent to the first method. They were found to check exactly and this fact indicates that the brass strip remained in contact with the end of the valve spindle over the required period.

- - - - - - - - - - - - -

DISPLACEMENT OF INDICATOR DRUM.

An ordinary 'Maihak' piston indicator, as designed to suit the requirements of the medium speed Diesel engine, was used for all cylinder pressure measurements necessary during this experimental work. This was done mainly to avoid additional complication of apparatus but for accurate work this form was found to be unsuitable. The indicator is shown in Fig. 2, p. 7.

The indicator was operated with the driving eccentric displaced out of phase, and in advance, by about 60 degrees of crank angle. It is generally agreed that this form of indicator is subject to errors caused by spring and piston vibration, (and also inertia) effects at speeds probably much less than 300 R.P.M., which was the normal speed of the engine under test. On this occasion however, although vibrations were generally apparent, this was not the principal source of trouble.

Theoretically, the obvious method of obtaining pressure readings on a base of crank angle from a displaced diagram, would be to turn the engine slowly round so many degrees at a time, and by means of the indicator pencil mark the intervals in the form of a scale on a card fixed to the drum. This method was found, directly at least, to be inapplicable on account of the fact that the length of travel of the pencil on the card while the engine was running at 300 R.P.M. exceeded the length of travel when the engine was slowly turned over, by about .088". This lack of agreement obviously could not be ignored. A good enough approximation to the solution of this problem might have been obtained by first drawing the length/

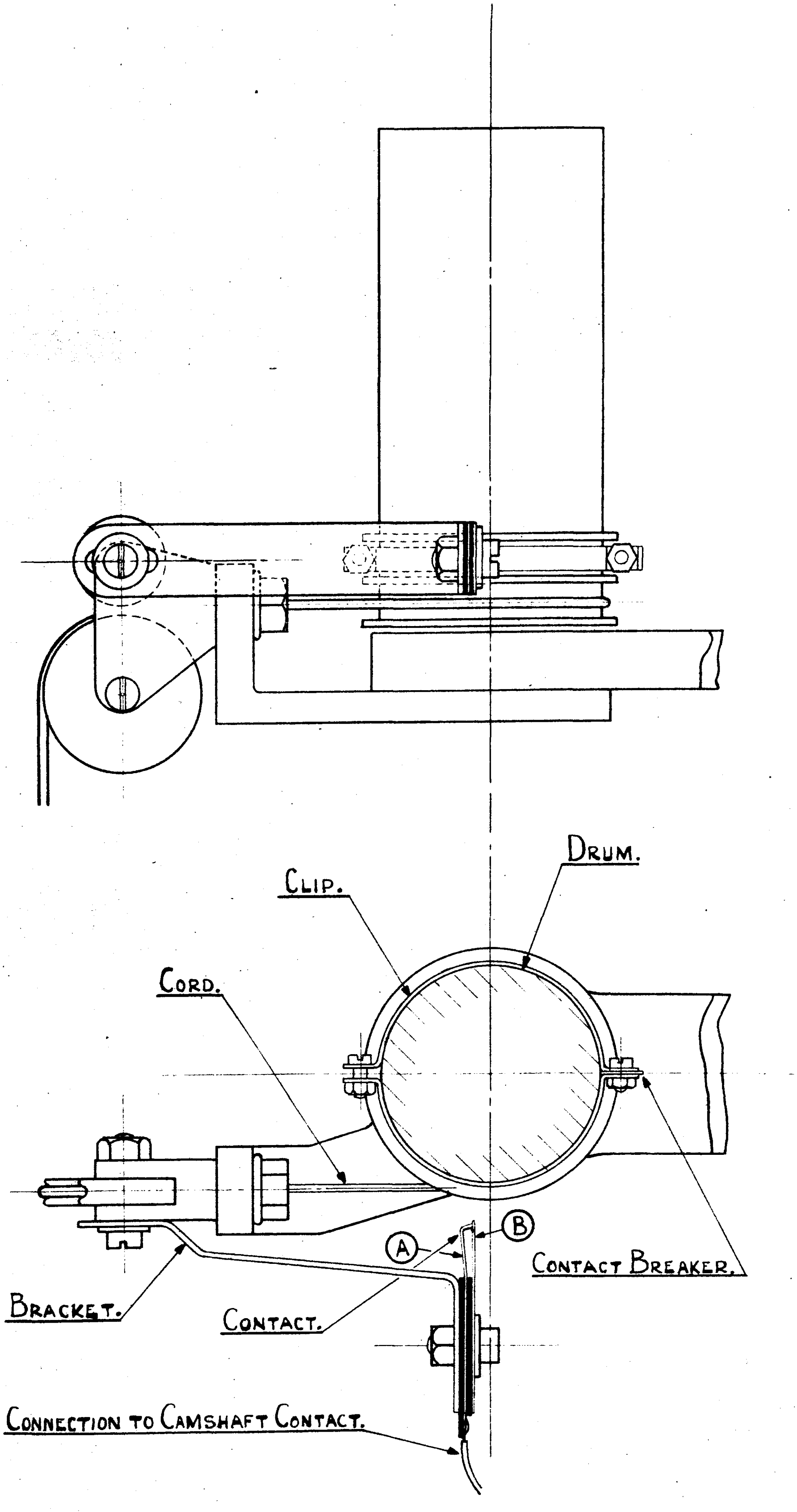


FIG.44. APPARATUS FOR INDICATOR DRUM DISPLACEMENT.
(Full Size)

length of travel on the card with the engine being turned over by hand, and then on the same card the length of travel with the engine running at 300 R.P.M. Such a procedure would have given the correct allocation of the total error to each end of the diagram, and the distribution of the error over the length of the diagram could have been obtained by assuming an 'acceleration curve' law of variation from one end to the other. The reason for this will be clear on examination of Fig.46,p.98. As this was not known at the time it was decided to investigate the actual motion of the drum at 300 R.P.M. and ascertain if possible the cause of the error.

The additional apparatus required for this is shown in Fig.44,p.91. The two halves of a light metal clip were fastened round the drum by two small nuts and bolts. Between one of these junctions was clamped a small piece of stiff steel. At the end of a rigid bracket bolted to a fixed part of the indicator, were clamped two strips of extremely light spring steel, insulated from each other and from the bracket by strips of red fibre. This was done for convenience since although the strip A bent sharply at the end and acting as a stop was insulated, the straight strip B was connected through to the bracket by means of the clamping nut and bolt. Strip A was connected to the camshaft contact(Fig.42,p.84,)as used in the apparatus already described and to a battery of about 24 volts and earphones. The action of the arrangement was as follows.

Normally a continuous contact was formed between the two strips A and B, but the instant that the small piece of steel fixed to the drum (rotating for example in/

in the plan view in Fig.44,p.91, in an anticlockwise direction) struck the projecting strip B, the electrical connection was broken for an instant. On the return stroke of the drum the contact breaker merely grazed the strips in passing.

Certain precautions had to be taken to make this piece of apparatus capable of rapid action.

(1) The steel strips had to be as light as possible in order not to affect the motion of the drum.

(2) They had to be so proportioned that when 'flicked' by the contact breaker they showed no tendency to vibrate,

The desired thickness and breadth were first decided upon and then some time was spent in obtaining the correct proportion of length to satisfy both the above conditions.

(3) As already mentioned,the bearing pressure between the extremeties of the strips was only sufficient to allow of a good electrical contact. One reason for this was that considerable bearing pressure would cause friction between the contact surfaces and possibly alter slightly (in an erratic manner) the relative positions of the strips when in contact. Another reason was that the success or otherwise of the experiment depended on the instant when the strip B left the strip A - not when the contact breaker touched the strip B. If therefore the strip A were in a state of elastic stress due to the pressure of strip B then, when this load was suddenly removed by the contact breaker on the drum, strip A would on account of its elasticity, tend possibly to follow the strip B for a greater distance than if the contact had been broken very slowly.

(4)/

(4) The strips were not set radially to the drum but after the manner shown in the sketch. This was done in order to obtain a definite break in the contact and at the same time a minimum impact on the return stroke. An appreciable break was thus obtained, and yet it almost appeared as though, on the return stroke, the contact breaker did not touch the strips at all. The sole reason for this type of spring contact was the necessity to allow for the return stroke of the drum.

(5) The drum clip had to be very light, in order not to alter appreciably the inertia of the moving parts. It was found to be rather heavier (3.3 gms.) than an indicator card (1.7 gms.) but this difference was so slight in comparison with the weights of the other moving parts as to be negligible.

The method of obtaining readings for this experiment was somewhat tedious, in that each reading involved starting and stopping the engine. With further elaboration of the apparatus this could possibly have been obviated. In each case the engine was started up and allowed to run for a few minutes until it had settled down to a steady speed on no load. By means of the earphones and camshaft contact, the instant when the contact at the indicator was broken was determined. The fuel was then cut off, and as the engine stopped the instant of breaking was again ascertained. The clip on the drum holding the contact breaker was then turned through a small angle and the same procedure repeated. One complete stroke of the drum was investigated in this manner and then an additional bracket was fitted so as to hold the original bracket and contact strips in exactly the reverse position. The return stroke of the drum/

drum was then similarly investigated.

The diagram length at 300 R.P.M. and zero R.P.M. (that is when turning the engine by hand or slowly on compressed air) were taken on the same card, and a scale of intervals of 10 degrees of crank angle marked off for one complete revolution at zero R.P.M. As in all other cases involving the use of the camshaft contact, the readings had to be corrected as explained on page .

All the readings (after correction) of this experiment, and the consequent lags, either positive or negative, are given in table 7, p. 96. If at a certain instant of crank angle during either stroke, the position of the drum at 300 R.P.M. is behind its position for the same instant of crank angle at zero R.P.M., the displacement has been called a positive lag; and if in front, a negative lag. In Fig. 46, p. 98, however, this rule had to be altered in order to obtain an unbroken curve throughout the complete cycle. The rule holds for the clockwise stroke (see plan view in Fig. 44, p. 91) but has to be reversed for the anticlockwise stroke.

In table 7, p. 96 taking column "X" as an example, 58 degrees (measured from the engine I.D.C.) is the reading at 300 R.P.M. and 60.5 degrees the reading at zero R.P.M. It is clear, therefore, that at 300 R.P.M. the drum has passed the particular position 2.5 degrees before it would have at zero R.P.M., and therefore the lag will be negative. In other words, (for 300 R.P.M.) at 58 degrees crank angle the drum is in the position which it would occupy at 60.5 degrees crank angle (for zero R.P.M.). From the curve of displacement for zero R.P.M. in Fig. 45 p. 97, the position of the drum at 60.5 degrees can be ascertained. This then is the position/

TABLE 7. Lags in Motion of Indicator Drum.

Crank Angle at Break. (zero RPM)	313·5	315·5	320·0	328·5	330·0	339·25	344·0	348·75	355·75	357·25	358·0	359·25	5·25	8·00	13·25
" " " " (300 RPM)	315·75	317·5	321·75	329·0	330·0	339·15	344·75	349·5	356·25	357·25	358·25	359·25	5·00	8·00	12·75
Lag in Degree of Crank Angle	+2·25	+2·00	+1·75	+·50	0·00	+·50	+·75	+·75	+·50	0·00	+·25	0·00	-·25	0·00	-·50
Lag in Inches	+·019	+·020	+·021	+·007	0	+·010	+·016	+·019	+·012	0	+·005	0	-·006	0	-·011

"X"

Crank Angle at Break (zero RPM)	20·5	22·50	28·50	36·00	38·5	44·75	49·25	54·0	58·00	60·5	74·00	79·5	88·25	87·5	104·25
" " " " (300 RPM)	19·5	21·75	28·25	36·75	39·0	45·50	48·75	52·5	55·25	58·0	67·75	72·5	82·25	86·0	100·50
Lag in Degrees of Crank Angle	-1·00	-·75	-·25	+·75	+·5	+·75	-·50	-1·5	-2·75	-2·5	-6·25	-7·00	-6·0	-1·5	-3·75
Lag in Inches	-·023	-·018	-·006	+·021	+·013	+·016	-·011	-·030	-·055	-·047	-·097	-·098	-·064	·015	-·020

Cr.∠. at Break (zero RPM)	105·75	107·00	115·00	126·0	131·5	136·5	144·75	145·5	156·75	166·5	176·00	179·5	185·5	192·00	194·5	197·75
" " " " (300 RPM)	103·00	103·75	106·00	135·25	138·0	142·0	148·25	149·0	158·75	168·0	177·75	181·0	187·0	193·75	196·0	198·75
Lag in Degrees of Cr.∠.	-2·75	-3·25	-9·0	+9·25	+6·5	+5·5	+3·5	+3·5	+2·0	+1·5	+1·75	+1·5	+1·5	+1·75	+1·5	+1·0
Lag in Inches	-·012	-·014	-·024	+·039	+·040	+·043	+·036	+·036	+·027	+·025	+·033	+·030	+·030	+·041	+·032	+·023

Crank Angle at Break (zero RPM)	207·25	209·50	215·25	224·25	226·75	234·75	239·0	243·5	250·25	258·75	266·75	275·5	287·75	291·25	297·5
" " " " (300 RPM)	206·50	210·25	216·00	225·00	227·25	235·00	239·5	244·0	250·25	258·25	265·5	274·0	285·50	286·50	288·0
Lag in Degrees of Crank Angle	-·75	+·75	+·75	+·75	+·50	+·25	+·50	+·50	0·00	-·50	-1·25	-1·50	-2·25	-4·75	-9·5
Lag in Inches	-·018	+·018	+·021	+·021	+·013	+·006	+·011	+·011	0	-·009	-·020	-·018	-·013	-·022	-·026

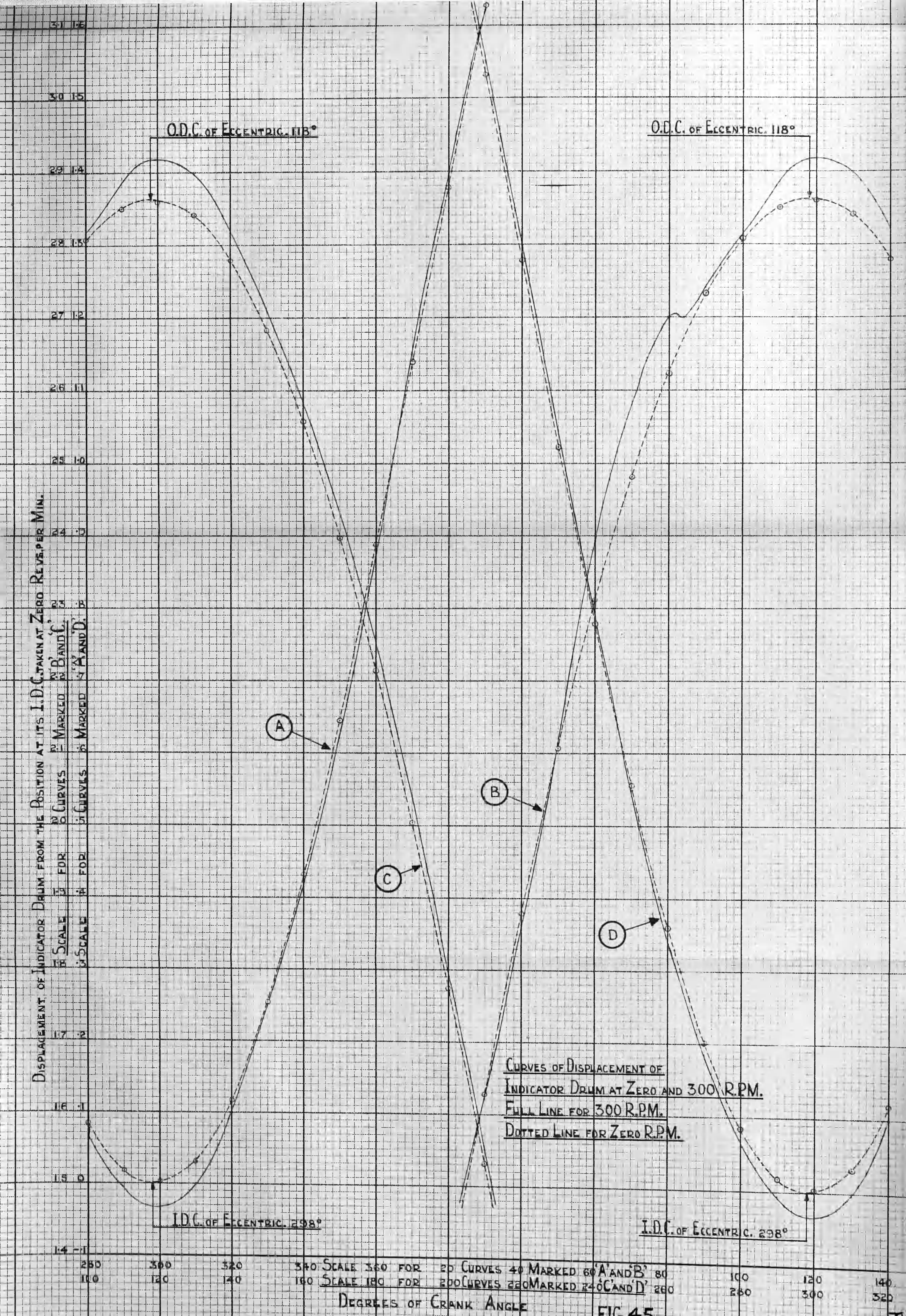
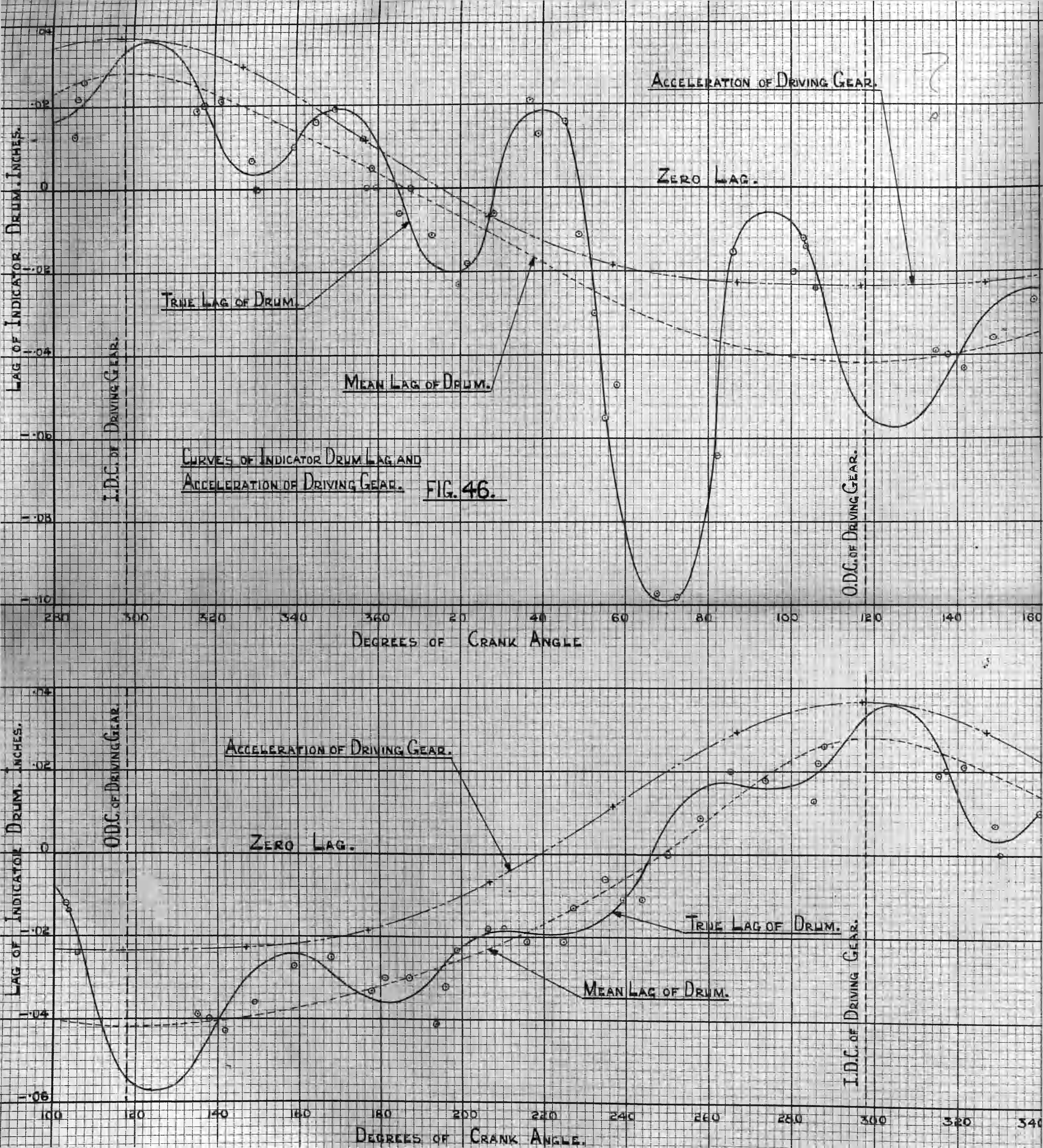

O.D.C. of Eccentric. 118°
O.D.C. of Eccentric. 118°
I.D.C. of Eccentric. 298°
I.D.C. of Eccentric. 298°
A
B
C
D
Curves of Displacement of
Indicator Drum at Zero and 300 R.P.M.
Full Line for 300 R.P.M.
Dotted Line for Zero R.P.M.
Displacement of Indicator Drum from the Position at its I.D.C. taken at Zero Revs. per Min.
Scale for Curves Marked 'B' and 'C'
Scale for Curves Marked 'A' and 'D'
Scale 360 for 20 Curves 40 Marked 60 'A' and 'B' 80
Scale 180 for 200 Curves 220 Marked 240 'C' and 'D' 260
Degrees of Crank Angle

FIG. 45.


Curves of Indicator Drum Lag and Acceleration of Driving Gear.
FIG. 46.
Acceleration of Driving Gear.
Zero Lag.
True Lag of Drum.
Mean Lag of Drum.
I.D.C. of Driving Gear.
O.D.C. of Driving Gear.
Lag of Indicator Drum. Inches.
Degrees of Crank Angle
·04
·02
0
-·02
-·04
-·06
-·08
-·10
280
300
320
340
360
20
40
60
80
100
120
140
160
Acceleration of Driving Gear.
Zero Lag.
True Lag of Drum.
Mean Lag of Drum.
O.D.C. of Driving Gear.
I.D.C. of Driving Gear.
Lag of Indicator Drum. Inches.
Degrees of Crank Angle.
·04
·02
0
-·02
-·04
-·06
100
120
140
160
180
200
220
240
260
280
300
320
340

position of the drum at 58 degrees for 300 R.P.M. From this curve of displacement can also be ascertained the position of the drum at 58 degrees for zero R.P.M. and obviously the difference between these two readings will give the lag (negative in this case) at 58 degrees under running conditions.

All the pairs of readings were treated in this manner and the lags plotted (with the alteration in sign where necessary) in Fig.46,p.98, the base representing the motion of the drum at zero R.P.M. Owing to the nature of the apparatus and the methods employed, points in the vicinity of the dead centres of the drum stroke could not be obtained. For example, if the clip on the drum were arranged so that contact was just broken close to the dead centre position, and in that direction, for 300 R.P.M., then the contact would not necessarily be broken at all with the drum oscillating very slowly.

It cannot be claimed that the lags plotted in Fig.46,p.98, are entirely satisfactory, but after all, a few thousandths of an inch is a relatively small quantity when dealing with a cord about $\frac{1}{16}''$ in diameter running over two 1" diameter pulleys and then round a drum about 1½" in diameter. When some of the readings at 300 R.P.M. were being taken, it was found that the reading sometimes altered quite considerably during the space of a minute or so, showing possibly that the cord took some time to settle down to a final working state; while at other times the reading was observed to vary slowly backwards and forwards. It was for this reason found advisable to allow time for conditions to settle down at 300 R.P.M., take the reading, and then shut down and take the corresponding reading at zero R.P.M./

zero R.P.M. Variations however, such as mentioned above, would be unlikely to have any appreciable effect during the time elapsing in the normal taking of an indicator diagram.

The formation of the curve in Fig.46,p.98 in the region of the dead centres of the drum stroke was definitely fixed by the values obtained from the differences in the lengths of the 300 R.P.M. diagram and the zero R.P.M. diagram. These differences did not directly give the lag at the dead centres (this would be the case only if a maximum or minimum value of the vibration coincided with the dead centre position) and the curves in Figs.45,p.97, and 46,p.98, had to be experimented with around these points until the desired difference in lengths between the 300 R.P.M diagram and the zero R.P.M. diagram were obtained for both ends. The curve of actual drum displacements at 300 R.P.M. was readily obtained from a combination of the curve of displacements at zero R.P.M. and the vibration curve in Fig.46,p.98 .

The mean curve in Fig.46,p. 98,was fixed by eye to represent approximately the 'forced' element of the vibration. The actual vibration curve taken with respect to this mean curve represents the 'natural' element of the vibration of the system. An inspection of the curve makes it apparent that this mean curve is not a simple function of the acceleration; in other words, of the two maximum turning values of acceleration the greater is at the inner dead centre of the drum stroke, while of the two maximum turning values of the mean curve the greater is at the outer dead centre position of the drum. A possible explanation for this might lie/

lie in the fact that the modulus of elasticity of the indicator cord (as obtained by direct loading and measurement of extensions) was not a constant within the elastic limit, but increased considerably with load.

A continuous diagram of horizontal lines was taken with the engine slowing down from 300 R.P.M. to stop. This diagram showed, as might now be expected, the general tendency of the diagram length to decrease more or less uniformly from its 300 R.P.M. value to its zero R.P.M. value. In addition to this general tendency however, the extremities of the horizontal lines did not appear to form a smooth curve. For agreement with the vibration results already given, these extermities should have conformed roughly to a periodic curve, but their rather vague nature did not allow such a detailed examination to be made. The fact that the extermities did not form a smooth curve suggests at least that the vibration may possibly exist at speeds much lower than 300 R.P.M. - perhaps with a reduced amplitude. At lower ranges of speed it may disappear entirely, leaving only the 'forced' element of the vibration, that is, the vibration in synchronism with the acceleration forces of the system. In this region therefore, the extremities of the horizontal lines of the last mentioned diagram should form a smooth curve if drawn with the correct spacing between them. To accomplish this satisfactorily however, some mechanical contrivance would be required to operate the 'feed' (parallel to the drum axis) of the indicator pencil.

In order to investigate the possibility of errors caused by pencil friction, a card was fixed to the drum and the instants of breaking contact examined for two arbitrarily/

arbitrarily chosen positions of the drum, first with the pencil bearing sufficiently heavily on the card to obtain a good diagram, and second with the pencil clear of the card. No change of position of the drum at either of these instants could be detected, indicating that in this case pencil friction was of a negligible order.

The mean length of the indicator cord was about 20.5". The driving gear and eccentric were all of a reasonably rigid nature and unlikely to be the cause of any vibrations such as have been discussed. The indicator hook used was not of the type supplied for the purpose, but was made as light as possible from a short length of comparatively thin wire.

The results of the experiments show that, with a cord actuated indicator, it cannot be assumed that the differences between the diagram taken at speed and taken at zero R.P.M. as shown at either end of the diagram represent the maximum possible horizontal error of the diagram. Fig.46,p.98, shows an error, almost twice the greater 'end error', at 70 degrees crank angle. Neglecting this possibility, an approximate solution to the problem of the distribution of error over the diagram would be to assume a curve of acceleration,displaced if necessary to suit the inequality of the errors at the ends of the diagram. Fig.46,p.98, indicates that for the present investigation and also in fact for a normal 'in phase' diagram, it would have been more satisfactory, on account of its small amplitude of 'natural' vibration, to have used the anticlockwise stroke for the combustion diagram of the engine, that is the stroke during which the drum spring, rather than the eccentric, is doing the direct pulling/

pulling. Possibly the amplitude of both the 'forced' and the 'natural' vibration could have been somewhat reduced by increasing the tension of the drum spring and so increasing the average 'modulus of elasticity' of the cord.

Practically all the diagrams that have been taken with this indicator (see compression card, Fig. 54, p. 141.) show an irregularity which has been found to correspond fairly closely with the wave crest shown at about 95 degrees crank angle in Fig. 46, p. 98, (the effect of this wave being brought out clearly in Fig. 45, p. 97). It is improbable that this irregularity was due to piston vibration as it was present on the compression cards which otherwise showed no evidence of piston vibrations.

A further complication in this problem was due to the fact that with a similar indicator on a gas engine running normally at about 200 R.P.M. the diagrams obtained were shorter than those at zero R.P.M. Here the case was slightly different because the length of the cord was less than on the oil engine, but otherwise the entire gears were practically identical. It is possible that at a certain speed the 'natural' vibrations, if any, would be so placed as to cause a contraction of the diagram. If time had permitted this would have been possibly another interesting case to investigate. It will be evident from Fig. 45, p. 97, that for a normal 'in phase' diagram taken on the oil engine, the period of maximum pressure will appear more prolonged, and give the diagram the appearance of greater constant pressure combustion than actually is the case.

The/

The object of these tests was essentially to obtain a means of accurate calibration of the horizontal axis of the indicator cards, but they also serve as an example of the possible inertia effect on the behaviour of any indicator drum running at speeds of the order of 300 R.P.M.

Shortly after these tests had been completed the results of previous experimental work were examined and showed that indicator drums do not always behave in the above manner. Dr. Brown of the Royal Techincal College, Glasgow, in a paper entitled "Measurement of Power" and published in the Trans. of the N.E.Coast Inst. of Engineers and Shipbuilders (1927-28) presents a different aspect of the subject. He shows that, in the case of another Maihak indicator drum the predominant factor was spindle friction, and that this latter caused a definite contraction in the length of the diagrams obtained, owing to the card, at each reversal of motion, having to be stretched sufficiently to overcome the static friction of the drum.. He further shows that up to a speed equivalent to 60 R.P.M. the diagram length is not altered. These results offer a possible explanation for the case of the gas engine which has been already mentioned, if it were not for the fact that its speed was considerably greater. Regarding this question of drum friction, it is perhaps interesting to note that, judging from the formation of the curve in Fig.45,p.97, of drum displacement at zero R.P.M. in the region of the dead centres of the drum stroke, the "stationary period" discussed by Dr. Brown is practically non-existent. Despite this however, when the drum spring was balanced by a load applied to the end of the cord it remained at rest over quite a considerable range of positions/

positions in a similar manner to Dr. Brown's example. The cord used throughout was a long-used piece of ordinary plaited cotton.

In a paper entitled "Notes on Indicator Cards" by Mr. S.Ashworth and published in the Trans.of the Inst. of Marine Engineers, the author apparemtly encountered a similar vibration phenomenon by the use of piano wire as a driving agent for an indicator drum. The effect of this, he points out, was to cause distinct 'ripples' on the compression curve of the card. These vibrations apparemtly must have been of considerable magnitude, if one can judge from the fact that the only wave of Fig.46,p.98, that could readily be detected on a compression card was that already mentioned as occurring at about 95 degrees of crank angle.

- - - - - - - - - - - - - - - - - -

EFFECT OF JACKET TEMPERATURE AND BRAKE LOAD ON THE TEMPERATURE OF THE OIL AT THE FUEL VALVE NOZZLE.

The following results of oil temperature under running conditions were obtained by means of a thermo-couple as shown in Fig. 13, p. 27 . In these tests, within the limits of the governor, the speed of the engine was maintained at roughly a little over 300 R.P.M. and a series of temperature readings were taken for various loads on the brake and various jacket water outlet temperatures. On each occasion of an increase in load the oil temperature was at first observed to fall appreciably and then rise very gradually to a value somewhat higher than that before the alteration in load. This initial drop was evidently caused by the increased flow of oil through the nozzle and the following rise by the greater heat of combustion of this increased flow. The readings obtained were not very consistent, but they give a general idea of the conditions prevailing. Table 8, p. 107, gives the principal results of these tests. The values of revs./min. and fuel/B.H.P. hr. in this table show that the engine was running rather irregularly on this occasion.

The general inferences from the table of results are (1) that for a constant value of B.H.P., a rise of 1°C of jacket outlet temperature caused a rise of about $\frac{1}{3}$°C in the fuel oil temperature at the nozzle; and (2) that for a constant value of jacket outlet temperature an increase of 1 B.H.P. caused a rise of about 1°C in the fuel oil temperature at the nozzle.

As the jacket water temperature and net brake load were set roughly at 35° and 110 lbs. respectively during/

TABLE 8. Oil Temperatures at Nozzle.						
Jacket Outlet Temp. °C	Net Load on Brake Lbs.	R.P.M.	B.H.P.	Oil/hr. Lbs.	Oil/BHP.hr Lbs.	Temp. of Oil at Nozzle °C.
30	110·2	309	13·2	7·39	·560	55·5
	80·2	306	9·6	5·41	·563	52·5
	62·2	304	7·32	4·05	·553	50·7
	52·2	304	6·15	2·99	·486	49·7
40	110·7	318	13·2	6·71	·508	58·6
	95·7	308	11·44	6·43	·562	57·6
	79·2	301	9·22	5·37	·582	55·3
	61·2	314	7·33	4·35	·593	52·4
	42·2	304	4·97	3·43	·689	49·7
50	114·7	300	13·34	7·53	·564	61·8
	98·7	305	11·70	6·23	·532	58·2
	79·7	312	9·64	5·47	·567	56·6
	61·7	304	7·15	4·05	·566	55·2
	42·2	310	5·06	3·49	·689	53·3
60	114·7	–	–	–	–	66·0

during the engine tests involving the 'rate of injection of fuel' and described later on pp.129etc.it is seen from table 8, p.107,that the desirable temperature for the heated oil in the discharge experiments (pp.35etc.) should lie between 55.5°C and 58.6°C. This heated oil temperature, averaged over the discharge tests, was actually 56°C (see table 1, p.52).

- - - - - - - - - -

DISPLACEMENT OF PUMP PLUNGER.

For comparison with the results given in the next section of fuel oil pressure and fuel valve spindle lift, a knowledge of the displacement of the fuel pump plunger relative to the engine cycle was desirable. The pump cam was unaltered throughout all the following engine tests and the procedure described below was carried out with the pump cam at the particular setting adopted.

The apparatus used to determine the displacement of the pump plunger was simple. The discharge pipe was detached from the top of the fuel pump and in its place was fitted vertically, by means of a brass adapter and gland, a straight length of glass tube of about $\frac{3}{16}''$ bore. A scale of inches was attached to this tube. The engine was rotated through a known number of degrees of crank angle, and the corresponding displacements of the oil level in the tube were noted. As there was a tendency for the oil level to fall slowly on account of some leakage in the system, in order to avoid the cumulative error arising from this, the displacement readings were all taken from the point at which the plunger stroke commenced, and each was taken as rapidly as possible.

The weight of water necessary to fill a given length of the glass tube was carefully measured, and from this calibration the displacement of the plunger could be fairly accurately converted to lbs. of fuel oil.

The curve obtained by the above method has been plotted for comparison with the curves of oil pressure and valve spindle motion in Figs.48-52pp.113-120, given in the next section.

- - - - - - - - - - - - -

OIL PRESSURE, VALVE SPINDLE MOTION AND RATE OF INJECTION.

With some of the apparatus that has already been described and the results of the experiments on discharge of liquids through the fuel valve nozzle, it was now possible to obtain definite information regarding the principal features of the process of injection of fuel oil into the combustion chamber. The rate at which the fuel was injected at any instant during the process could now be determined with a reasonable degree of accuracy. The degree of accuracy was measured by checking the integration of the 'rate' curve against the actual oil delivered per stroke by the pump (as obtained from the reading of total consumption during the test). For three cases treated in this way the arithmetical mean error was found to be 2.8%, and this average value of error for the aggregate results of total fuel injected suggests that the degree of accuracy obtained for each individual curve (see Figs.50-52, pp.118-120) relating to the process was fairly good.

Turning to the curves of oil pressure (Figs.48-52, pp.113-120), some irregularity in the experimental points will be observed, especially in the flatter parts of the curves after the fuel valve has closed. This is accounted for by the fact that, owing to the governing of the engine, these points fluctuated considerably during the readings. The other parts of the curves are clearly not affected to the same extent by the governing, and therefore could be much more readily determined.

Three series of tests were run with conditions of fuel valve spring and brake load varied in such a manner as to bring out clearly the general character of/

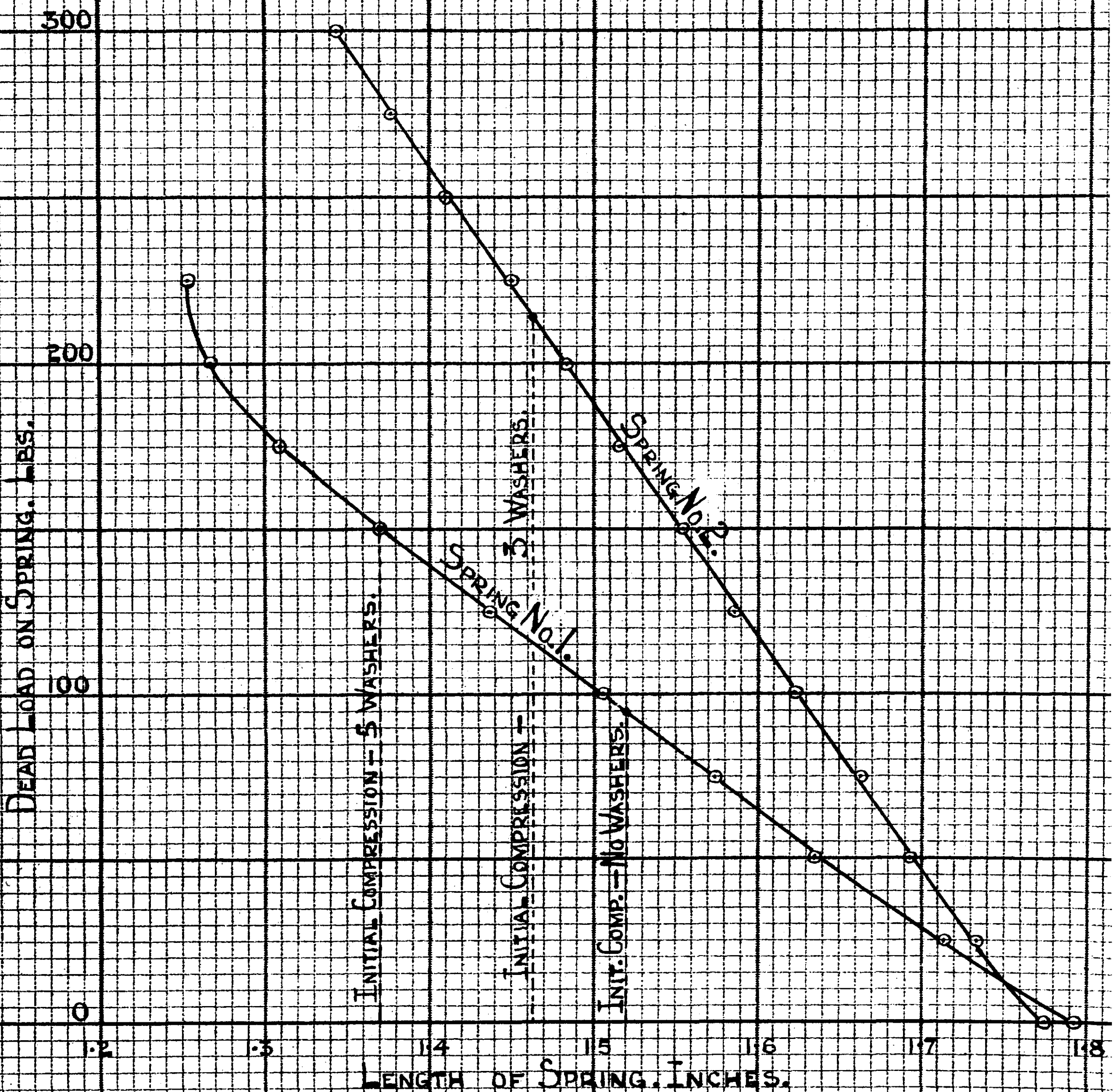
of the injection system.

The first series consisted of five tests at different brake loads, in which the valve spring No.1 for which the calibration curve is shown in Fig.47, p.112, operated under normal conditions. Curves for this series, of oil pressure, valve spindle motion, cylinder pressure, and pump plunger displacement are given in Fig.48, p.113.

The second series consisted of five other tests at various brake loads, in which the same spring No.1 was used, but this time having a much higher initial compression. This was obtained by inserting behind it five circular washers each .03" thick. For this series a set of curves, similar to those drawn for the first series, is given in Fig.49, p.114.

In the third series (see Figs.50-52, pp.118-120) the brake load was kept constant at a little under full load throughout, and only the valve spring was altered. In the first test A, spring No.1 was used as in the first series. In the second test B, spring No.1 was used as in the second series. In the third test C, a heavier spring No.2 (see calibration curve in Fig.47, p.112,) was employed, and the initial compression increased by means of three washers each .03" thick. This third series has been more fully investigated than the first two and the tests designated A, B, and C respectively, for reference purposes.

Take to begin with the first and second series. As was desired, the effect of brake load, through the action of the governor, on the oil pressure and valve motion diagrams is pronounced in both cases. The evident relation between these curves, and that of pump displacement is perhaps worthy of note. The rapidity/



Calibration Curves of Fuel-Valve Springs. FIG. 47.

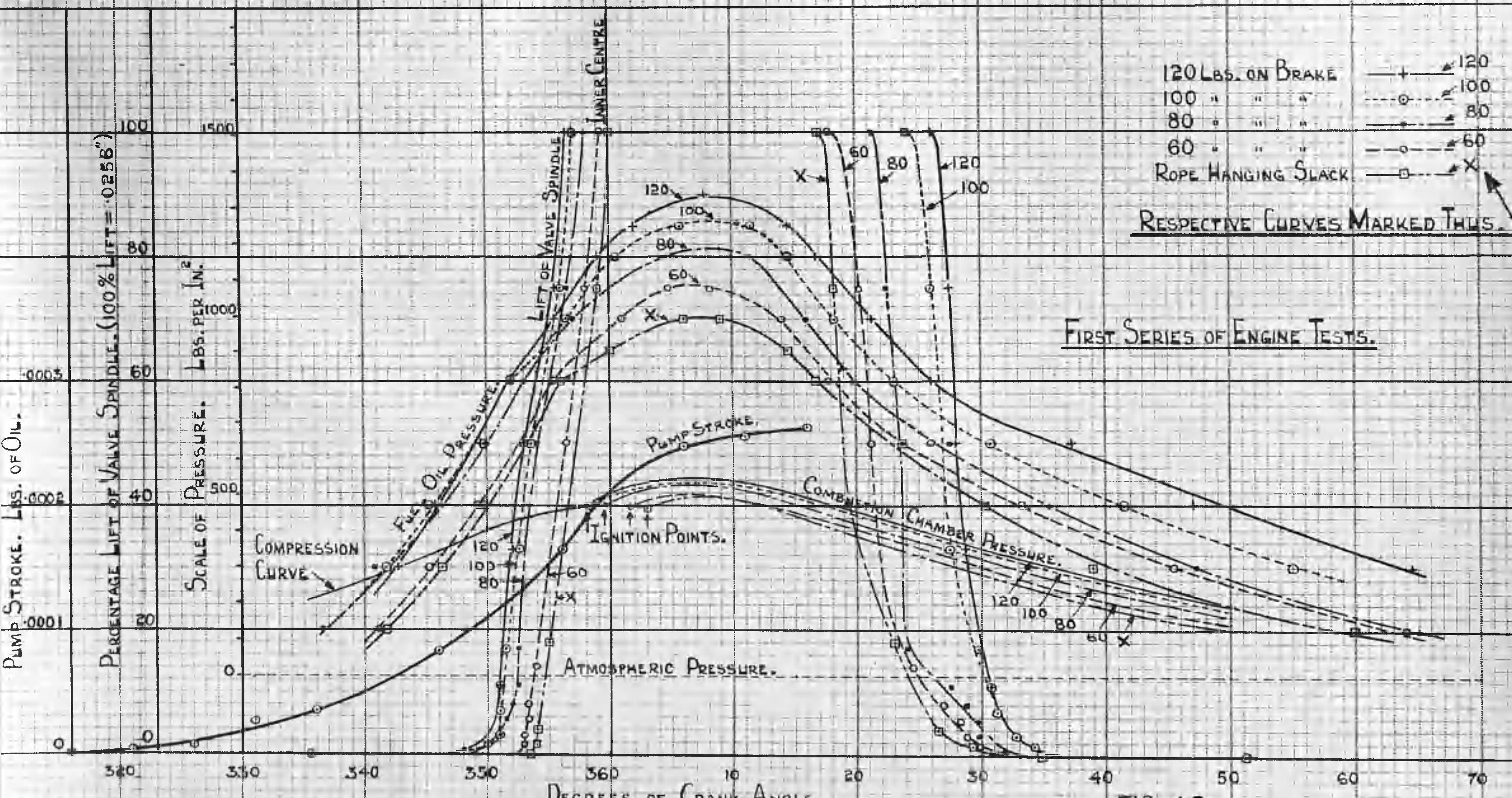

120 Lbs. on Brake
100 " " "
80 " " "
60 " " "
Rope Hanging Slack
Respective Curves Marked Thus.
First Series of Engine Tests.
Pump Stroke. Lbs. of Oil.
Percentage Lift of Valve Spindle. (100% Lift = ·0256")
Scale of Pressure. Lbs. per In.²
Inner Centre
Lift of Valve Spindle
Fuel Oil Pressure
Pump Stroke.
Compression Curve
Ignition Points.
Combustion Chamber Pressure.
Atmospheric Pressure.
Degrees of Crank Angle.

FIG. 48.

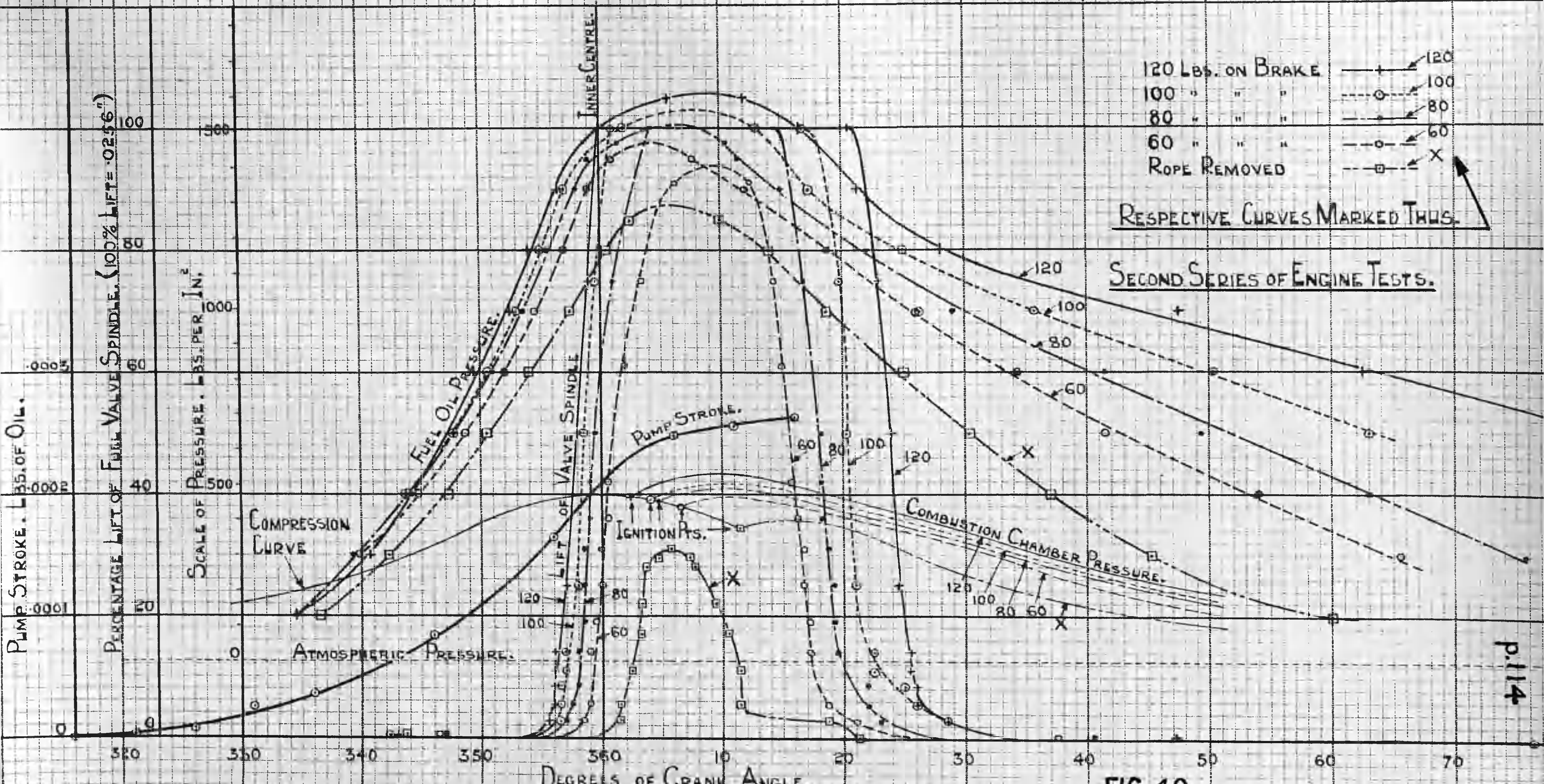

120 Lbs. on Brake
100 " " "
80 " " "
60 " " "
Rope Removed
120
100
80
60
X
Respective Curves Marked Thus.
Second Series of Engine Tests.
Inner Centre.
Fuel Oil Pressure.
Lift of Valve Spindle.
Pump Stroke.
Compression Curve
Ignition Pts.
Combustion Chamber Pressure.
Atmospheric Pressure.
Pump Stroke. Lbs. of Oil.
Percentage Lift of Fuel Valve Spindle. (100% Lift = ·0256")
Scale of Pressure. Lbs. per In.²
Degrees of Crank Angle.
0
·0001
·0002
·0003
20
40
60
80
100
500
1000
1500
320
330
340
350
360
10
20
30
40
50
60
70

FIG. 49.

rapidity with which the valve opens will be dependent to a large extent on the gradient of the plunger displacement curve, and at the same time adversely affected by a reduction of brake load. The fact that all the curves in Fig.48,p.113 are practically parallel shows that as the brake load is reduced this reduction is compensated by a later opening of the valve, and therefore at a steeper portion of the pump displacement curve. The effect is different in Fig.49,p.114 . Here, as the load is reduced, the valve spindle lifts later, but in doing so, instead of gaining the advantage of greater plunger velocity as in the former case, it is handicapped by a rapid reduction in this velocity, as indicated by the plunger displacement curves. The 'lift' curve of the case with the brake rope removed, that is with the engine running against no load save its own friction, represents what must be practically a limiting condition. Here the valve spindle has only time to lift about 30% (or roughly .008") when the spring pressure appears to overcome the oil pressure and the inertia of the spindle, and forces it back on to its seat again. This last diagram however, may give rather a false impression of the actual quantity of fuel injected, unless it is borne in mind that at 30% of the maximum lift,the rate of injection has practically attained its maximum value for the pressure conditions at the instant. (See Fig.25,p.51). Considering that the resultant motive force acting on the spindle remains positive during the lifting process (this can be shown by means of the calibration curve, Fig.47,p.112) it is interesting to note that the majority of lift curves approximate to straight lines. This must be a frictional effect, and in addition, friction/

friction is seemingly the only explanation for the irregular nature of some of the valve closing curves of the diagrams. Near the end of the closing period in some cases there appears almost to be a suggestion of the spindle actually sticking - especially in the case of 'no load' already cited when the spindle velocity at closing was probably relatively small.

The question of spindle friction has been investigated in detail for the spindle lifting in test A of the third series, yet to be described. (See p.126)

It is noteworthy in the above diagrams that the points of ignition (as determined by the beginning of the pressure rise on the indicator diagrams) follow closely the sequence of the other curves; that is, there is approximately the same spacing between the points of ignition as between the curves of rising oil pressure, and those of spindle lift. This evidently serves as a rough check on the reliability of the curves and their interrelationship.

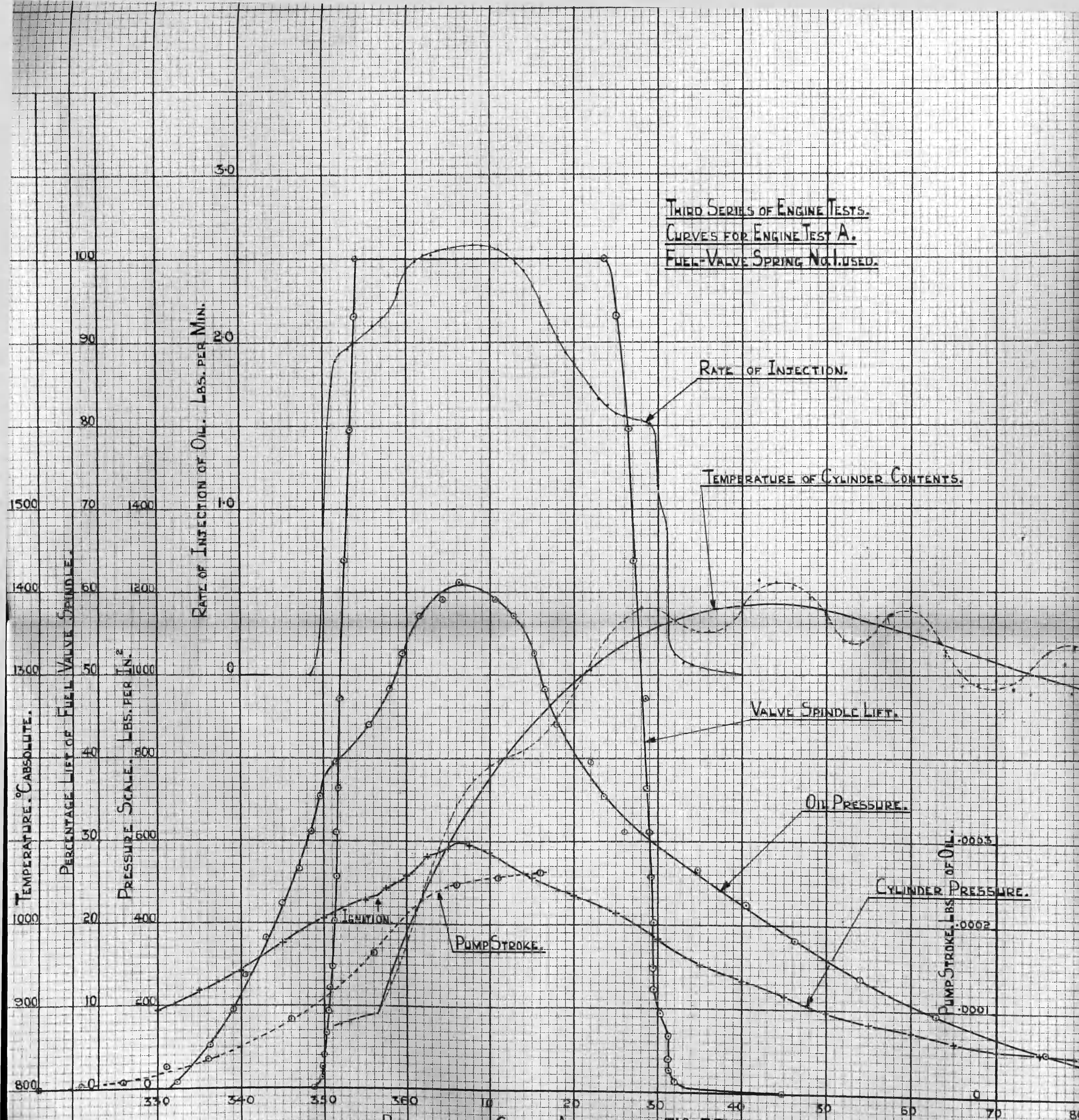
Shortly after the two series of tests just recorded were carried out, the author obtained a paper by Mr. D.H.Alexander, M.Sc., entitled "Airless Injection and Combustion of Fuel in the High Compression Heavy Oil Engine" which was published in the Trans. of the Inst. of Marine Engineers, August 1927. The subject in this paper is treated up to a certain point in a manner rather similar to that described here, and is further referred to in page 122.

Turning now to the third series of tests, an examination of Figs.50-52,pp.118-120, each of which gives the curves of oil pressure, valve spindle lift, cylinder pressure, rate of injection and temperature of cylinder contents for one test, shows the same characteristic/

characteristic features as shown for the first two series.

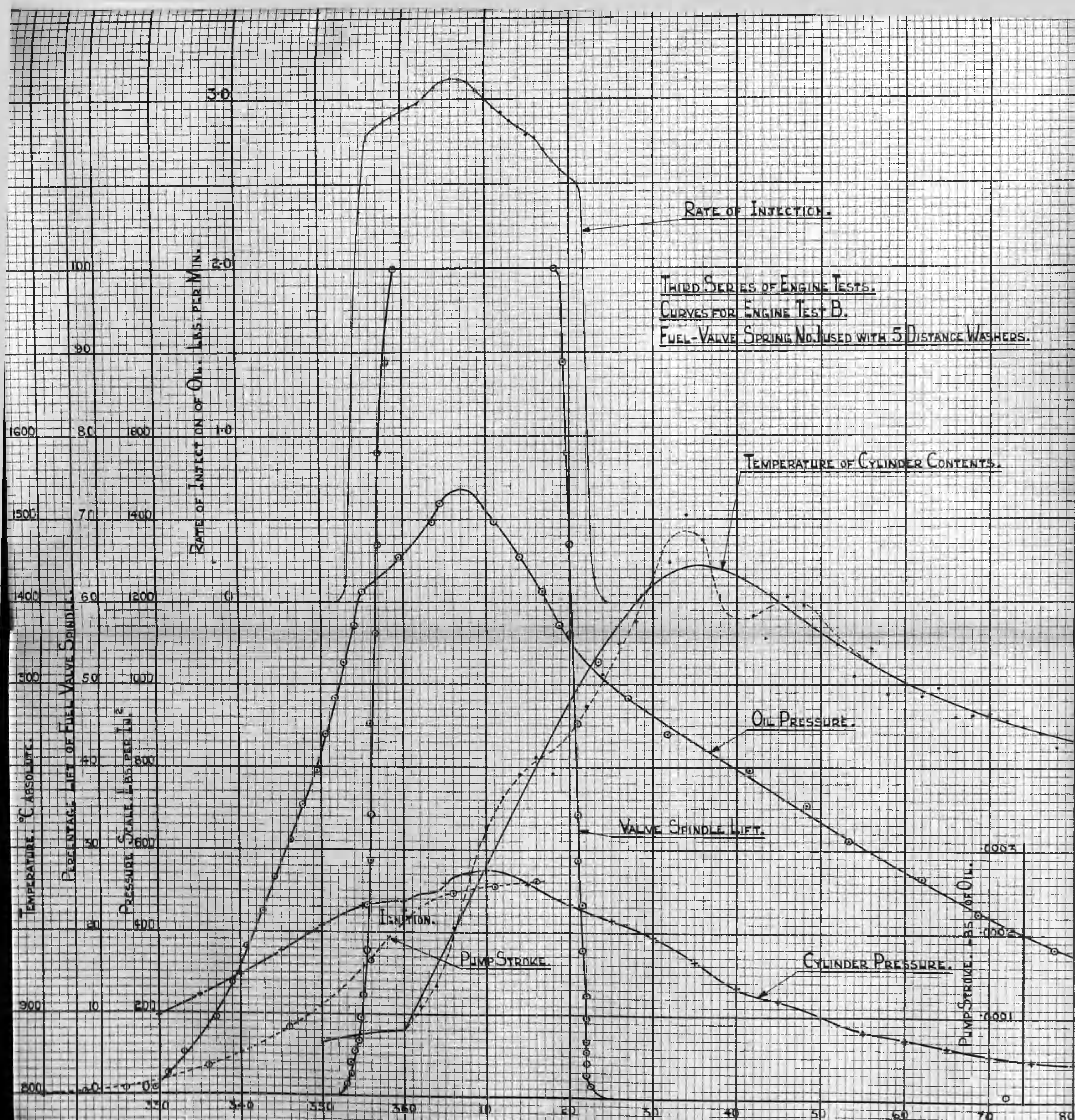
In a manner similar to that caused by an increment of brake load, an increase in valve spring load causes the governor to reduce the amount of opening of the bye-pass valve. This is obviously necessary if the engine is to maintain steadily the same output of work. A higher spring load causes higher oil pressures, and therefore a tendency to a greater flow of fuel per cycle through the bye-pass. This must therefore be counteracted by a partial closing of the latter. The partial closing is not however, just sufficient to maintain the same total flow through the bye-pass. At the higher spring loads the flow is considerably less than this, and the reduction, neglecting leakage, is an approximate measure of the additional elastic strain in the system caused by the higher pressures prevailing. This feature was brought out very clearly when an attempt was made to run the engine with the valve spring, No.2, initially compressed by means of 5 distance washers already mentioned. Even with the bye-pass valve completely shut, as is always the case when the engine is being started, the valve spindle would not lift at all, and naturally no fuel could be injected.

By tracing the three curves of rising oil pressure in Figs.50-52, pp.118-120, on to one sheet, it can be shown that they are roughly parallel. This can be explained by the nature of the pump displacement curve (shown along with the oil pressure curves etc.) in a manner similar to that discussed in connection with the parallel lift curves in Fig.48,p.113, and Fig.49,p.114 . If the velocity of the plunger were constant, then the earlier/



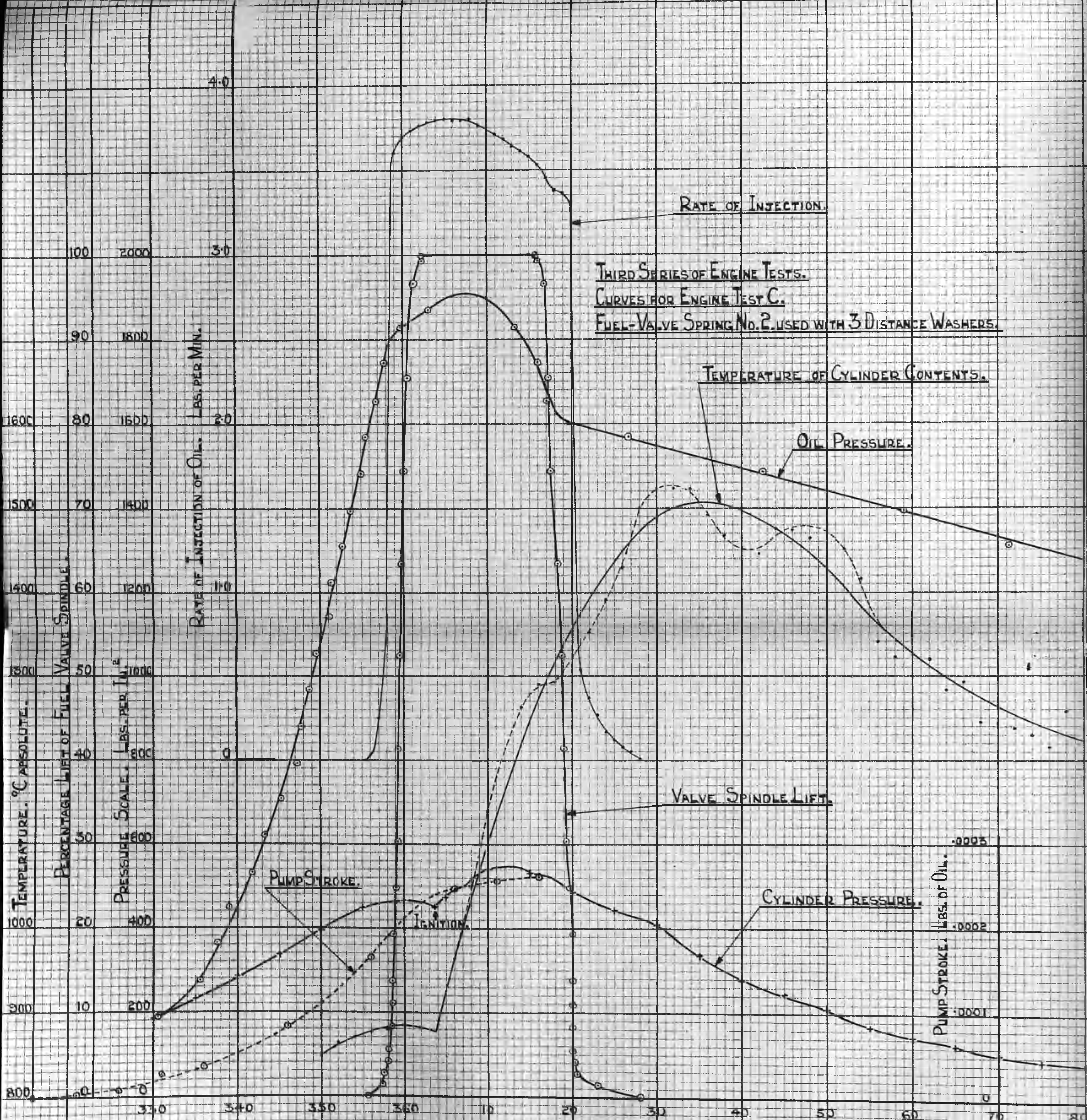
Third Series of Engine Tests.
Curves for Engine Test A.
Fuel-Valve Spring No. 1. used.
Rate of Injection.
Temperature of Cylinder Contents.
Valve Spindle Lift.
Oil Pressure.
Cylinder Pressure.
Pump Stroke.
Ignition.
Temperature. °C absolute.
Percentage Lift of Fuel Valve Spindle.
Pressure Scale. Lbs. per In.²
Rate of Injection of Oil. Lbs. per Min.
Pump Stroke. Lbs. of Oil.
Degrees of Crank Angle.

FIG. 50.



Rate of Injection.
Third Series of Engine Tests.
Curves for Engine Test B.
Fuel-Valve Spring No. 1 used with 5 Distance Washers.
Temperature of Cylinder Contents.
Oil Pressure.
Valve Spindle Lift.
Ignition.
Pump Stroke.
Cylinder Pressure.
Rate of Injection of Oil. Lbs. per Min.
Temperature. °C Absolute.
Percentage Lift of Fuel Valve Spindle.
Pressure Scale. Lbs. per In.²
Pump Stroke. Lbs. of Oil.
Degrees of Crank Angle.

FIG. 51.



RATE OF INJECTION.
THIRD SERIES OF ENGINE TESTS.
CURVES FOR ENGINE TEST C.
FUEL-VALVE SPRING NO. 2. USED WITH 3 DISTANCE WASHERS.
TEMPERATURE OF CYLINDER CONTENTS.
OIL PRESSURE.
VALVE SPINDLE LIFT.
PUMP STROKE.
IGNITION.
CYLINDER PRESSURE.
TEMPERATURE. °C ABSOLUTE.
PERCENTAGE LIFT OF FUEL VALVE SPINDLE.
PRESSURE SCALE. LBS. PER IN.²
RATE OF INJECTION OF OIL. LBS. PER MIN.
PUMP STROKE. LBS. OF OIL.
DEGREES OF CRANK ANGLE.

FIG. 52.

earlier the pressure rose , the more rapidly would it do so, since an earlier rise means a smaller bye-pass opening - provided that the pump timing is unaltered. In the case under consideration, however, the earlier the pressure rise took place the lower was the mean velocity of the plunger, and as will be apparent the combined effect is that the curves are practically parallel.

During the valve spindle lifting process the pressure gradient decreases considerably and then tends to increase again on account of the further movement of the pump plunger. This movement, as can be seen from the diagrams, falls off fairly rapidly in the region of 5 degrees of crank angle, with the result that the pressure curve passes its maximum turning point and then also begins to fall rapidly. As the valve closes, the pressure curve is once more deflected into its final phase, and this represents the curve of dissipation of elastic energy of the system through the bye-pass, and also of course through any minor leak. If the bye-pass is practically closed, as in test C (Fig.52,p.120) the pressure may not fall to atmospheric before the next forward stroke of the pump.

The relation between 'valve opening' and 'valve closing' pressures is decided by the relative effects of the ratio of $\dfrac{\text{Total spindle area}}{\text{Total area} - \text{valve seat area}}$, and the ratio $\dfrac{\text{Spring load (valve open)}}{\text{Spring load (valve closed)}}$. The greater the first ratio the more does the opening pressure exceed the closing pressure and the greater the second ratio the more does the closing pressure exceed the opening pressure. If the load-extension curve of the valve spring be a straight line, then for any initial compression of the spring the effect of the second ratio will/

will be constant. On the other hand, the effect of the first ratio will increase as the initial compression of the spring and therefore the average oil pressure increase. From a consideration of the combination of these effects with the curves of oil pressure it can be shown that in the present cases (A, B and C) the closing pressures will exceed the opening pressures. This is clearly the case in Fig. 50-52, pp. 118-120. On account of the lack of certainty regarding the instants of opening and closing of the valve, and the steepness of the pressure curves at these instants, no real purpose is served by attempting to verify these opening and closing pressures. It is interesting to refer to the pressure curves obtained by Mr. Alexander in the aforementioned paper (p. 116). The valve spindle that he used had a very much larger ratio $\frac{\text{Total spindle area}}{\text{Total spindle area} - \text{valve seat area}}$ and the effect of this is clearly shown by the fact that the opening pressures are considerably higher than the closing pressures. An interesting feature of his curves, which does not appear at all in the present cases, is a distinct secondary rise in oil pressure caused by the continuance of the pump stroke after the valve has shut.

As a check on the total lift obtained by means of the motion recorder (Fig. 43, p. 87) it is of interest to refer to the micrometer measurement taken of this quantity during the discharge experiments. It was found then to be .0256". On the recorder the equivalent reading was about 370 degrees of the circular adjusting scale (see table 12, p. 142). From the working drawing of the recording apparatus the ratio of

$$\frac{\text{Movement at centre line of spindle}}{\text{Movement of adjusting screw}} = \frac{2.35''}{3.60''}$$

and therefore the total lift of the spindle as given by the recorder is:-

$$\frac{1}{26} \times \frac{2.35}{3.60} \times \frac{370}{360} = \underline{.0258''} \text{ (Adjusting screw} = 26 \text{ thds./in.)}$$

This appears to be a satisfactory check.

When considering the actual efficiency of the valve spindle motion, it must be borne in mind that approximately 30% only of the maximum lift is necessary for maximum discharge. (See Fig.25,p.51). Any narrowing therefore of the areas enclosed by the lift curves, above this value does not sensibly reduce the working efficiency of the injection. The curves of rate of injection show that the reduction of efficiency of the combustion of the oil injected, due to the throttling effect at the valve seat during opening and closing of the valve, is hardly appreciable. The value of $\frac{\text{Total spindle area}}{\text{Total spindle area} - \text{valve seat area}}$ was not more than 1.073. The results indicate that a much larger ratio than this is unnecessary. Other things remaining constant, the larger this ratio the greater will be the difference between 'opening pressure' and 'closing pressure'.

A maximum lift, considerably greater than that necessary for maximum discharge, is desirable, since it enables the spindle to attain higher velocity as it approaches the valve seat. The diagrams indicate also that a valve of this type could be readily adapted to suit much higher speeds if desirable. At higher speeds also, an increase of opening and closing velocities is to be expected on account of the more rapid rise and fall of pressure. This gradient of pressure is undoubtedly one of the principal factors governing the efficient motion of the spindle, since it is roughly the mean value of the increase in pressure above the lifting value which determines the effective lifting force. The gradient of the falling pressure would seem/

seem to depend entirely on the flow through the nozzle and through the bye-pass valve (neglecting leakage). It follows that the pump setting should be such that the velocity of the pump plunger is at its maximum during the opening period, and should fall to zero before the closing period commences. It is here, probably, that the injection system fitted with a mechanically operated release has its principal advantage over the simpler bye-pass type; the advantage being that a much more rapid fall in pressure is made possible Another recognised advantage of course, is that the beginning of injection remains unchanged for all loads on the engine.

By way of comparison with the present experiments, is the corresponding performance of a mechanically operated solid injection system given in a series of results in 'Engineering' Dec.3, 1920 by Professor C.J.Hawkes. He shows that under certain conditions, only the first part of the motion of this particular valve is in fact mechanically operated, since the valve spindle and lever, owing to the unbalanced oil pressure, on the valve face (and possibly to inertia), leave the cam altogether, and complete the injection as a very inefficient automatic valve; - inefficient on account of the fact that it has to close, not against a rapidly falling pressure as in the case of a true automatic valve, but against a high pressure which tends to become constant as the valve closes. This pressure may even tend to rise as the valve is closing, if a number of fuel pumps are steadily feeding the constant pressure 'rail' in a multi-cylinder engine. Even in the case where the valve mechanism follows precisely the profile of a cam with rapid opening and closing/

closing, this high pressure at closing would appear to be an inherent disadvantage of the 'rail' or 'mechanical' system of solid injection.

Reference to an article published in the 'Motorship' for July and August 1929 shows the automatic fuel injection valve in a different light. The author of the article here gives diagrams of oil pressure and spindle motion - the latter being obtained by connecting the valve spindle to the piston rod of an indicator. These diagrams show violent pulsations in the fuel oil pressure and oscillations of the valve spindle during the entire period of injection and form a rather interesting comparison with the more normal behaviour shown in the present experiments.

The lifting efficiency of a fuel valve could be stated as the ratio of fuel injected under conditions free from the restricting influence at the valve seat to the total fuel injected. On the average, for the tests A,B and C this works out roughly at about 95%. In this respect, when it is considered that a large portion of the remaining 5% is not severely throttled, the valve appears to compare favourably with the ideal. This however, is not the whole problem. An ideal valve is one which opens and closes instantaneously and therefore in such a manner that the later part of the oil injected leaves the orifice at approximately the same velocity as the major part of the injection. The rate of injection curves in Figs. 50-52, pp. 118-120, show clearly that such cannot be the case with this particular fuel value. It would be difficult to say whether it actually suffers from the 'after drip' effect or not, as this will depend to an extent on the turbulence around the nozzle at the closing of the valve. On the

fuel/

fuel valve being removed, it was generally found to be coated liberally with carbon, but this could be explained by the fact that the engine was usually oiled to excess. It may be added that, for the purpose of overcoming this difficulty, some engines are so designed that a jet of hot gases is made to impinge on the nozzle at the end of the injection period.

Comparison between Actual and Theoretical Rates of lifing of the Fuel Valve Spindle.

The following investigation was made in order to show the effect of friction on the rate of lifting of the fuel valve spindle. This was accomplished by determining, for one particular case, the theoretical motion of the spindle for the actual pressure conditions prevailing at the nozzle and comparing this motion with that actually obtained during the test. The particular test adopted for this purpose was test A of the third series, the curves for which test are shown in Fig.50, p.118.

In order to determine the theoretical lifting velocity from a given curve of oil pressure such as is shown in Fig.50 the following data are necessary:-

l = free length of spring, (1.775")
L = length of spring, valve closed (1.52")
P_{S_1} = spring load, valve closed (95lbs)
p = oil pressure at any instant (Fig.50, p.118.)
A = effective area of spindle (in this case taken as total area, .15in.^{2})
S = lift of spindle at any instant.
M = mass of moving parts + half the mass of valve spring (.34lbs)
P_S = spring load at any instant.

$$\text{Acceleration } \frac{dv}{dt} = (p.A - P_S)\cdot g/M$$

$$= \left[p.A - \left\{1 + \left(\frac{S}{l-L}\right)\right\}\cdot P_{S_1}\right]\cdot g/M$$

$$= \frac{p.A.g}{M} - \frac{P_{S_1}.g}{M}\left\{1 + \left(\frac{S}{l-L}\right)\right\}, \text{ but } S = \frac{t^2}{2}\cdot\frac{dv}{dt}$$

$$\therefore \frac{dv}{dt} = \frac{p.A.g}{M} - \frac{P_{S_1}.g}{M} - \frac{P_{S_1}.g}{M(l-L)}\cdot\frac{t^2}{2}\cdot\frac{dv}{dt}$$

$$\therefore \frac{dv}{dt} = \frac{(p.A - P_{s_1}).g/M}{1 + \frac{P_{s_1}}{(1-L)}.\frac{g}{M}.\frac{t^2}{2}}$$

$$= \frac{(p.A - P_{s_1})}{\frac{M}{g} + \frac{P_{s_1}}{(1-L)}.\frac{t^2}{2}}$$

$$s = \int_0^t \left[\int_0^t \frac{dv}{dt}.dt \right].dt$$

In the case dealt with (Fig.50,p.118,) the valve was taken to lift at 348.5 degrees crank angle. (i.e. $t = 0$, $\theta = 348.5^o$). The engine was running at 306 R.P.M. giving the time for 1^o as $.545 \times 10^{-3}$ seconds.

The following values of $\frac{dv}{dt}$ were calculated from the foregoing formula and plotted to a base of time (or crank angle) in Fig.53,p.128.

Crank Angle degrees.	Time t seconds $\times 10^{-3}$	$\frac{dv}{dt}$ ft/sec^2
348.5	0	0
349	.273	460
349.5	.545	880
350	.820	1485
351	1.36	1485
352	1.91	1330
353	2.46	1160
354	3.00	1005

The curve thus obtained was twice integrated graphically giving first the curve of spindle velocity $(\frac{ds}{dt})$ and then that of actual lift (s).

On the same Figure (Fig.53) is shown the actual curve of spindle motion for test A; the difference between this curve and theoretical curve (s) being due (neglecting errors) to spindle friction.

It is clear from the diagram that in this particular/

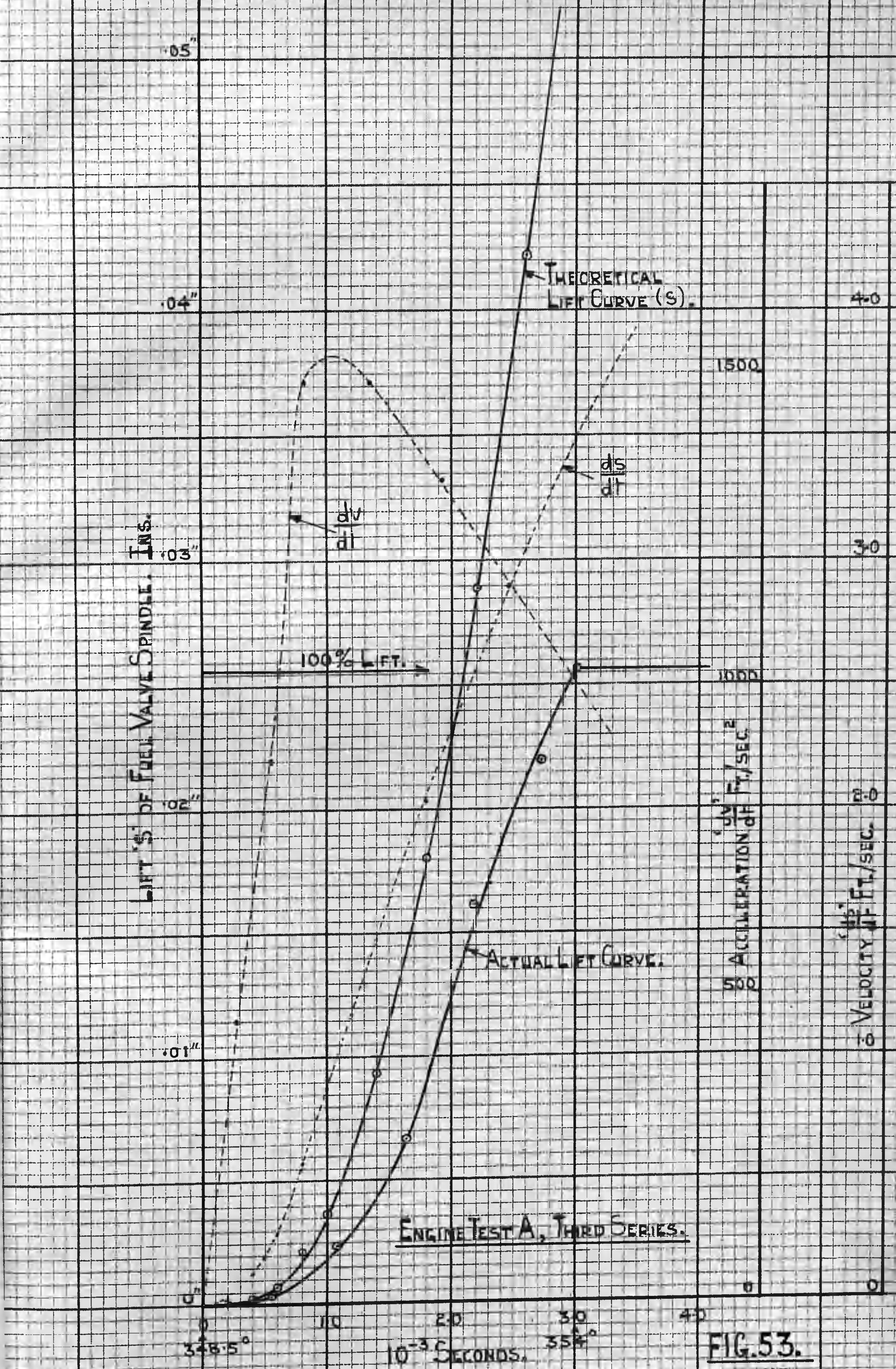


FIG. 53.

COMPARISON BETWEEN ACTUAL AND THEORETICAL SPINDLE-LIFT CURVES.

particular case the spindle, on account of the frictional effect, takes almost 50% in excess of the time theoretically necessary for lifting.

<u>Rate of Injection</u>.

The main point of note regarding the rate-of-injection curves shown in Figs.50-52,pp.118-120, appears to be their varying degrees of uniformity - the rate being more uniform the higher the average oil pressure. In case A the rate is about 80% of the maximum at the beginning of the injection and about 60% at the end. The rate in test C is clearly much more uniform than this. Whether or not the rate is uniform naturally depends to some extent on the duration of the injection period, (the shorter the period the less probably will be the total variation in conditions affecting discharge) but in addition to this it is evident that the greater the oil pressure the smaller will be the increase or decrease in the rate of injection caused by a given increase or decrease of oil pressure.

Whether the rate should be uniform for efficient combustion conditions, and if not, how it should vary, appears to be an extremely complicated question; and the complication is rendered greater by the close interconnection between the rate of injection and the initial velocity of the jet leaving the orifice. From this follows the consideration of penetration, atomisation, dispersion, shape of the combustion chamber (whether separate or a part of the working cylinder), turbulence, etc. After a study of much of the recent work done on the subject of the internal combustion engine, it becomes obvious that a complete and accurate knowledge of the mechanics, physics, and chemistry of the engine cycle would appear to be still wanting/

wanting, and that the main reason for this defect is the lack of adequate means of measurement. As is mentioned later, it would appear that some means of accurate and instantaneous sampling of cylinder contents would be of great value in this connection.

It is at present possible to measure the penetration and growth of a spray of oil into various conditions of gases, either static or in certain states of turbulence; it is possible to determine the atomisation of an oil jet injected into gas under various conditions; it is possible to determined the effect of varying degrees of turbulence, gas pressure, temperature etc. on the ignition of jets, and to measure the rate of heat loss through the cylinder walls, the rate at which the fuel enters the combustion chamber and numerous other quantities; but the one thing that cannot be determined is the exact history of what takes place when all these aforementioned processes combine simultaneously to form the complete working cycle of an oil engine.

It is obviously impossible therefore to state what the variation of the rate of fuel injection should be to obtain the best possible conditions for combustion of the fuel spray in the present cases. Apart from the above consideration however, a knowledge of the actual rate of injection is clearly of considerable interest, especially for purposes of comparison with actual combustion, since it indicates what is probably to be expected under similar conditions, and may perhaps pave the way for possible improvement.

An attempt has been made in the next section to analyse the indicator diagrams obtained for test A, B and C (for which the curves are given in Figs.50-52, pp.118-120, in the hope that some approximate idea of the relation existing/

existing between the rate of injection and the rate of combustion might be arrived at.

- - - - - - - - - - - -

FUEL INJECTION AND COMBUSTION.

-oOo-

RATE OF INJECTION AND COMBUSTION. (INTRODUCTORY)

During the early course of this research, the intention was to attempt later to draw off gas samples from the combustion chamber or cylinder during the combustion period, and a possible type of valve for the purpose was designed. It was afterwards learned that this form of experiment had already been carried out - for example by Dr.T.W.F.Brown at the Royal Technical College of Glasgow using an adapted Young's Pressure Recorder, and by Mr. Alexander (see p.116.) using a valve of his own design and having a mechanical instead of electrical spindle-lifting device.

Results of gas sampling experiments indicate that the mass of gas within the working cylinder is by no means homogeneous. Mr. Alexander shows that in his solid injection engine the CO_2 content obtained by his sampling valve at the end of the working stroke was considerably less than that of the exhaust gases.

The disadvantage of the gas sampling valve situated flush with the cylinder wall is clearly that it deals too exclusively with the gases circulating close to the wall. It is just possible that the gases, owing to their proximity to the walls, do not become thoroughly mixed with the main turbulent body of the cylinder contents, and consequently do not find an opportunity of uniting with their share of the fuel, so that although samples drawn from them definitely indicate a lack of homogeneiety, the main body of the gases may be, to all intents and purposes, homogeneous. In this way, even a number of similar sampling valves distributed over the cylinder walls might give an erroneous conception of/

of the state of the inner gases. Obviously, as was mentioned in a previous section, what is required is a means of picking out instantaneous samples from any point within the cylinder volume for any point of the stroke, and the difficulty of accomplishing this will be equally obvious. Nevertheless, until this or something equivalent has been done successfully, one cannot but feel that this problen unsolved represents an extensive gap in the knowledge of the subject.

It is clear therefore, that as a means of estimating the average rate of combustion of fuel throughout the cylinder, withdrawing samples close to the cylinder walls is not altogether reliable. No other possible method appears to be so far available. In the present work, therefore, this phase in the sequence of the cycle was passed over and only the rate of heat reception by the gases meantime dealt with.

For this purpose the temperature-entropy method of analysis of the indicator diagrams of the working cycle has been attempted. This method however cannot be altogether satisfactory owing to the difficulties attending the use of the indicator diagram and can only be expected at best to give an approximate solution.

The main sources of error were:-

(1) The difficulty of obtaining accurate average pressure readings from the indicator diagrams on account of the action of the governor (see Fig.54,p.141.)

(2) The difficulty of marking off on the diagrams the correct ordinates at the various points of the stroke. This has been fully dealt with under an earlier heading (see page 90.).

(3) The impossibility of estimating the variation in composition of the gases during combustion.

(4) The variation in average density of the gases, caused by the addition of the fuel mass.

(5)/

(5) What is probably of appreciable account only as regards the point of ingition - the lag between the pressure variations in the combustion chamber or cylinder, and their corresponding effect at the indicator piston situated at some distance from the combustion chamber.

By means of an efficient form of indicator situated right on the combustion chamber wall and capable of giving an accurate diagram on a base of crank angle (1), (2), and (5) could have been possibly eliminated (see p.90).

Item (3) of the above list of errors was probably the most difficult to cope with, apparently the only possible method of attack being accurate gas sampling as already discussed on p.133. The T.ϕ. diagram, being at present mainly concerned with the combustion process was drawn for the average composition of the contents during combustion. In the three tests under consideration the weight of fuel injected per stroke was 50% of the weight that could possibly be burned in the air in the cylinder (theoretically); that is the engine ran with 100% excess air. The average composition during combustion therefore was roughly that of the products of combustion of 25% of the possible fuel that could be burned (theoretically), mixed with excess air, and the T.ϕ. diagram was drawn for this average composition.

Item (4), namely the increase in density of the contents during combustion, has been allowed for approximately by the assumption of a fixed lag period after the injection of the fuel and then a reduction of the volume of the lb.mol. (see tables 17, and 18, pp.147, 148) This was done in order that each T.ϕ. curve should represent a definite weight of gas, instead of one increasing. The assumption is probably not correct, but/

but is likely to obtain a better result than if this factor were ignored completely.

Then for the study of combustion conditions the cylinder contents were assumed to consist of the contents at the end of suction plus half the total fuel injected.

For the conditions at the end of expansion the cylinder contents were assumed to consist of the contents at the end of suction plus the total fuel injected.

For the compression conditions the cylinder contents were assumed to consist only of the contents at the end of suction.

The assumption of the average composition during during combustion on which the T.ϕ. diagram was based, has probably caused the rate of heating obtained to be comparatively high at the commencement of combustion, and low towards the finish.

Item (5) is dealt with separately in a later section.

In the section commencing on p.140, the T.ϕ. method (etc.) has been worked out in detail for case A of the third series of engine tests, preceded by all the necessary steps for obtaining the curve of oil pressure, rate of injection etc. Only test A has been shown in detail, since the three tests were all dealt with in the same manner.

- - - - - - - - - - - - - - - - - -

MEASUREMENT OF AIR FLOW TO ENGINE.

For the analysis of the indicator cards in tests A, B, and C, readings of air flow to the engine were obtained by means of a pair of air boxes (see Fig.1, p.6.) in series, each of a capacity of 5ft. x 2ft. x 2ft., and a sharp edged orifice of 2" diameter; - these being part of the normal experimental equipment of the engine. Another part of this equipment is a large gasometer of about 63ft. capacity. The pipes leading from the latter and the air boxes are joined by a T piece with large stopcocks, so that, if desired, the engine suction can be switched over from one to the other, while the engine is running. The 'bell' of the gasometer is so counterbalanced that, if open to atmosphere, it slowly descends into the outer tank. Between the T piece junction and the engine is fitted a cylindrical apparatus made in the form of a 'bellows' with a collapsable leather wall and circular wooden ends; the normal length being about 3ft.3ins. and the mean diameter about 1ft.5ins. The purpose of this fitment is to damp the intermittent nature of the engine suction. It is suspended vertically, being supported at the top by the pipe connecting it with the main pipe leading from the air boxes and gasometer to the engine, and is kept in a certain state of tension by means of four lengths of stout rubber fastened to the circular bottom, and held to the floor by means of a weight. A general view of the complete arrangement is shown in Fig.1, p.6.

With accurate readings of temperature, volume and pressure the gasometer would clearly give an absolute reading of air flow over a given period, and a number of engine tests were run to determine the accuracy of the air boxes with a 2" dia. orifice. It was assumed here/

TABLE 9.	Measurement of Air Flow.			
Tests with 2" dia. Orifice	B.H.P.	Air per Minute in Ft.^{3}		Percentage Error
		Air Boxes	Gasometer	
1	13·95	3·180	3·215	−1·12
2	13·65	3·220	3·271	−1·56
3	12·74	3·216	3·278	−1·89
4	12·46	3·324	3·355	−0·87
5	12·44	3·431	3·359	+2·14
6	12·54	3·392	3·359	+0·98
7	12·32	3·370	3·378	−0·24
Mean	—	3·305	3·316	−0·33
Test with 1½" dia. Orifice	B.H.P.	Air per Minute in Ft.^{3}		Percentage Error
		Air Boxes	Gasometer	
8	12·46	3·313	3·411	−2·88

TABLE 10.	Indicator Card Readings.		Engine Test A.						
Crank Angle	320	325	330	335	340	345	350	355	357·5
Ordinates. Ins.	·297	·379	·471	·595	·707	·881	1·030	1·145	1·210
Lbs./in.^{2} Gauge	119	152	188	238	283	352	412	458	484
Crank Angle	360	2·5	5		7·5		10	15	20
Ordinates. Ins.	1·294	1·405	1·452		1·475		1·431	1·283	1·174
Lbs./in.^{2} Gauge	518	562	581		590		572	513	470
Crank Angle	25	30	35	40	45	50	55	60	65
Ordinates. Ins.	1·066	·912	·757	·666	·580	·486	·417	·370	·311
Lbs./in.^{2} Gauge	426	365	303	266	232	194	167	148	124
Crank Angle	70	75	80	85	90	95	100	120	180
Ordinates. Ins.	·275	·255	·238	·229	·199	·174	·145	·100	·070
Lbs./in.^{2} Gauge	110	102	95	92	80	70	58	40	28

here that the conditions at the end of suction were the same whether the engine was drawing from the air boxes or the gasometer, and this assumption was apparently justified by the fact that the light spring cards (1"=10 lbs/in^2) for the two cases coincided exactly. Various sizes of orifice could be used, but that of 2" dia. was found to be the most convenient for giving monometer readings which could be properly dealt with by the correction factors available.

The coincidence of the light springcards shows that the air boxes, if suitably dimensioned have no appreciable effect on the conditions of suction of the engine. In an article on this subject in 'Engineering', April, 13 and 20, 1923, by Mr. R.O.King, from which all the necessary constants and correction factors have been taken, is given the formula for the total capacity of the boxes (or box) as

$$Q = K \times \frac{\text{H P measured}}{\text{No. of Cyls.} \times \text{R.P.M.}} \text{ ft}^3.$$

Rough values for K are 600 and 1000 for high speed and slow speed engines respectively.

The gasometer readings as stated already, have been assumed accurate, but this can only be the case when the average of a number of readings is taken (except in the case of an exceedingly large gasometer) since each individual reading of flow is susceptible to acertain personal error. With the oil engine running at 300 R.P.M. the maximum duration of a gaometer reading was about one minute.

Table 9, p.137, shows the results of the tests and the accuracy obtainable by means of the air boxes used. The mean percentage error was calculated on the mean values of the air box and gasometer readings.

Table/

Table 9, p.137, also shows the result of a test in which a $1\frac{1}{2}$" dia. orifice was used. The large error obtained in this case, being negative, possibly shows the effect of undue throttling of the air flow - although the manometer did not indicate, what was thought to be excessively low pressure within the boxes. It is evident, however, that given an accurate manometer, the smaller this reading of difference between the pressure within the box and atmospheric, the greater will be the accuracy obtainable - that is, of course, accuracy with respect to normal conditions of engine suction either from the gasometer or directly from the atmosphere.

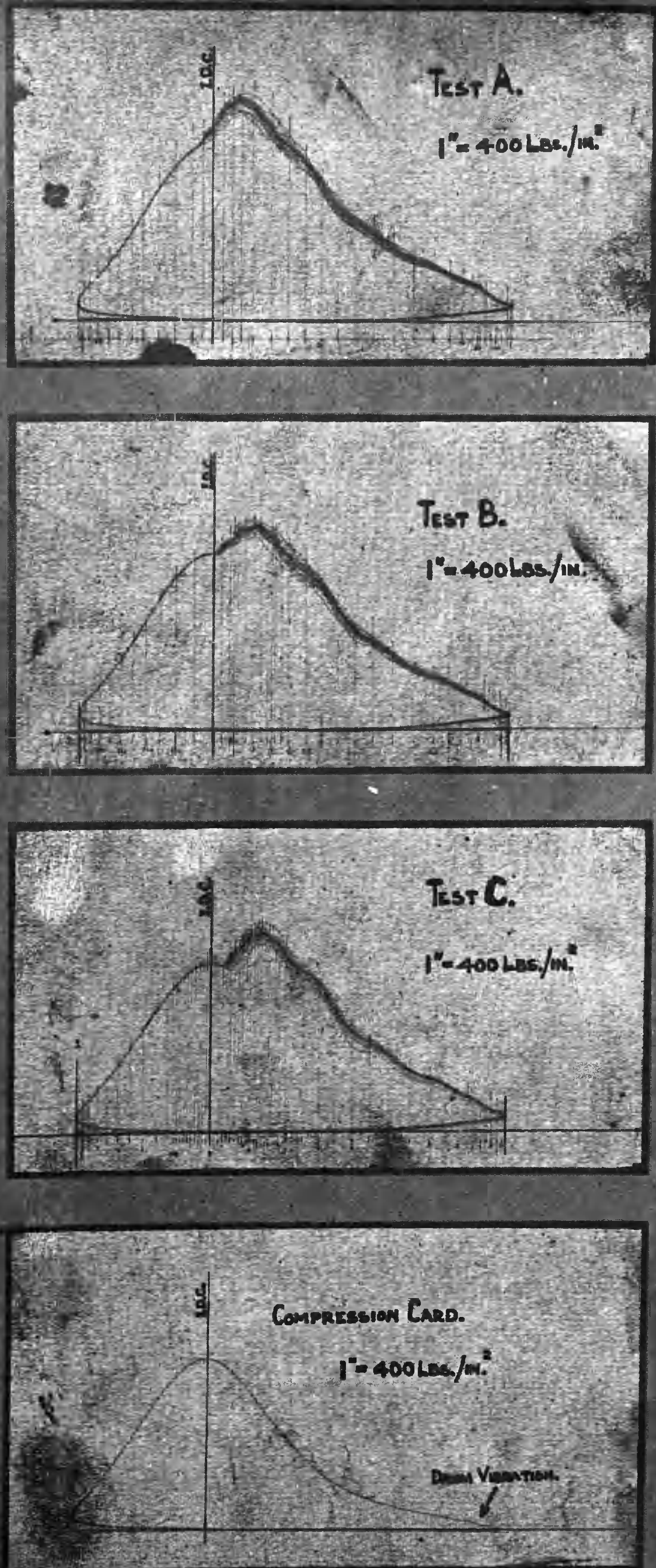
The 2"dia.orifice was used in all the work following these experiments.

- - - - - - - - - - - -

DETAILS OF ENGINE TEST A.

The displaced indicator card shown on Fig. 54, p.141, was one representing about 50 cycles. On account of the irregularity of the governing, the combustion line of this diagram was of considerable breadth and neccéssitated a mean line being drawn to represent the average stroke. The card was marked off horizontally at inter--vals of crank angle by means of the displacement curve for 300 R.P.M. in Fig. 45, p.97 . The pressure values(table 10) are plotted in Fig. 50, p.118, the scale of the diagram being 1" = 400 lbs/in^2. Tables 11 and 12, p.142, show the actual readings and derived values of oil pressure and valve spindle lift. These two sets of values are also plotted in Fig. 50, p. 118 .

The three curves obtained, together with those in Figs. 40, p. 73, and 25, 26, p.51, now enabled the rate of injection of fuel to be calculated. In table 13, p.143 are the remaining figures required for this purpose, the columns of oil and cylinder pressure and valve lift being obtained from the three curves already plotted, and the lift factors f_1 and air pressure factors f_2 from Figs. 25 and 26, p. 51, respectively. The values of C.H.U. per second were readily calculable from these of Lbs. of oil per minute and a knowledge of the calorific value of the fuel. This calorific value (higher) was obtained by means of a bomb calorimeter, the average of three such tests working out at <u>10,680 CHU/lb.</u> To allow for the error obtained in the results of rate of injection a correction of the values of CHU/second was made by the ratio $\dfrac{\text{Actual total fuel injected per stroke}}{\text{Calculated total fuel injected per stroke}}$.
The curve A" obtained from these corrected values is drawn in fig. 56, p.157 . In table 16, p.144, the values in column 10 of table 13, p. 143, were converted to total heat injected up to any instant, in C.H.U. These total/



Actual Indicator Diagrams for Third Series of Engine Tests.

FIG. 54.

TABLE 11.	Fuel Oil Pressure.				Engine Test A.		
Indicator Scale	Oil Press. Lbs./In.² G.	Bottom Contact			Top Contact		
		Cam Shaft Scale	Cr. Angle Uncorrected	Cr. Angle Corrected	Cam Shaft Scale	Cr. Angle Uncorrected	Cr. Angle Corrected
26·25	17	271·75	331·75	332·25	36·0	96·0	93·5
26·0	104	276·0	336·0	336·25	18·0	78·0	75·75
25·75	190	278·75	338·75	339·0	5·0	65·0	63·0
25·5	276	280·25	340·25	340·5	356·0	56·0	54·0
25·25	363	282·75	342·75	343·0	348·0	48·0	46·25
25·0	449	284·75	344·75	345·0	342·0	42·0	40·5
24·75	534	286·75	346·75	347·0	336·0	36·0	34·75
24·5	622	288·25	348·25	348·5	327·0	27·0	26·0
24·25	708	289·5	349·5	349·5	324·5	24·5	23·5
24·0	793	291·5	351·5	351·5	323·0	23·0	22·0
23·75	880	295·5	355·5	355·5	318·75	18·75	18·0
23·5	960	298·0	358·0	358·0	317·25	17·25	16·5
23·25	1053	299·5	359·5	359·5	315·75	15·75	15·25
23·0	1141	301·5	361·5	361·5	313·25	13·25	12·75
22·875	1181	304·5	4·5	4·5	311·0	11·0	10·5
22·75	1225	306·5	6·5	6·5	—	—	—

TABLE 12.		Motion of Valve Spindle.			Engine Test A.			
Adjusting Screw Scale	Difference	Percentage Lift	Duration of Contact between Spindle and Brass Strip					
			Cam Shaft Scale		Cr. Angle Uncorrect.		Cr. Angle Corrected.	
			From	To	From	To	From	To
325	0	0	288·75	346·25	348·75	46·25	348·75	44·75
330	5	1·34	289·75	333·25	349·75	33·25	349·75	32·0
335	10	2·70	289·75	332·5	349·75	32·5	349·75	31·25
340	15	4·05	290·0	332·5	350·0	32·5	350·0	31·25
350	25	6·75	290·25	332·5	350·25	32·5	350·25	31·25
360	35	9·45	290·5	331·5	350·5	31·5	350·5	30·25
370	45	12·16	290·75	330·75	350·75	30·75	350·75	29·5
20	55	14·85	291·0	330·75	351·0	30·75	351·0	29·5
40	75	20·3	291·25	330·5	351·25	30·75	351·25	29·5
60	95	25·7	291·5	330·0	351·5	30·5	351·5	29·25
80	115	31·1	291·5	329·75	351·5	30·0	351·5	29·0
100	135	36·4	291·75	329·5	351·75	29·75	351·75	28·75
140	175	47·2	292·0	328·0	352	29·5	352·0	28·5
200	235	63·4	292·5	328·0	352·5	28·0	352·5	27·0
260	295	79·6	293·25	327·5	353·25	27·5	353·25	26·5
310	345	93·1	293·75	326·0	353·75	26·0	353·75	25·0
335	370	100	294	324·5	354	24·5	354	23·5

TABLE 13. Engine Test A. Rate of Fuel Injection.										
Crank Angle Degrees	Oil Press. Lbs/in² G.	Cylinder Press. Lbs/in² G.	Press. Difference Lbs/in²	R Lbs./min.	Percent Lift	Lift Factor f_1	Press. Factor f_2	Rate of Injection Lbs./min.	Sum of Rate Ordinates	Heat in Oil C.H.U./sec.
349	653	405	250	1·44	0·6	·04	1·096	0·06	0·06	10
350	750	410	340	1·68	3·7	·39	1·097	0·72	0·78	128
351	780	425	355	1·72	14·1	·92	1·100	1·74	2·52	310
352	802	430	370	1·75	51·0	1·00	1·101	1·93	4·45	343
353	825	440	385	1·78	–	"	1·102	1·96	6·41	348
354	847	450	400	1·82	–	"	1·104	2·00	8·41	357
355	870	455	415	1·85	–	"	1·105	2·05	10·45	365
356	900	465	435	1·90	–	"	1·106	2·10	12·56	373
357	925	475	450	1·93	–	"	1·108	2·14	14·70	382
358	965	495	470	1·97	–	"	1·111	2·19	16·80	390
359	1015	505	510	2·06	–	"	1·112	2·29	19·18	408
360	1085	515	570	2·18	–	"	1·114	2·43	21·61	433
1	1130	535	595	2·22	–	"	1·116	2·48	24·09	442
2	1160	550	610	2·25	–	"	1·118	2·52	26·61	448
3	1180	570	610	2·25	–	"	1·121	2·52	29·13	448
4	1195	575	620	2·27	–	"	1·122	2·55	31·68	453
5	1210	585	625	2·28	–	"	1·123	2·56	34·24	455
6	1215	590	625	2·28	–	"	1·123	2·56	36·80	455
7	1220	590	630	2·29	–	"	1·123	2·57	39·37	457
8	1220	585	635	2·30	–	"	1·123	2·58	41·95	460
9	1210	580	630	2·29	–	"	1·122	2·57	44·52	457
10	1200	570	630	2·29	–	"	1·121	2·57	47·09	457
11	1185	560	625	2·28	–	"	1·119	2·55	49·64	453
12	1160	550	610	2·25	–	"	1·118	2·52	52·16	448
13	1130	540	590	2·21	–	"	1·117	2·47	54·63	440
14	1095	525	570	2·18	–	"	1·115	2·43	57·06	433
15	1050	515	535	2·10	–	"	1·114	2·34	59·40	417
16	995	505	490	2·02	–	"	1·112	2·24	61·64	398
17	940	495	445	1·92	–	"	1·111	2·13	63·77	378
18	890	485	405	1·83	–	"	1·109	2·03	65·80	362
19	850	480	370	1·75	–	"	1·108	1·94	67·74	345
20	810	470	340	1·68	–	"	1·107	1·86	69·60	332
21	780	460	320	1·63	–	"	1·106	1·80	71·40	320
22	750	450	300	1·57	–	"	1·104	1·73	73·13	308
23	720	445	275	1·50	–	"	1·103	1·66	74·79	295
24	695	435	260	1·47	–	"	1·102	1·62	76·41	288
25	675	430	245	1·42	–	"	1·101	1·57	77·98	280
26	660	415	245	1·42	–	"	1·098	1·56	79·54	277
27	640	405	235	1·40	65·0	1·00	1·096	1·54	81·08	275
28	625	390	235	1·40	45·0	1·00	1·094	1·53	82·61	272
29	610	375	235	1·40	31·0	·99	1·091	1·51	84·12	268
30	595	365	230	1·38	10·4	·76	1·089	1·14	85·26	203
31	580	350	230	1·38	7·3	·59	1·086	·88	86·14	157
32	565	340	225	1·37	1·2	·08	1·084	·12	86·26	21
33	550	330	220	1·35	·8	·05	1·082	·07	86·33	12
34	535	315	220	1·35	·6	·04	1·079	·06	86·39	10
35	520	305	215	1·34	·5	·03	1·077	·04	86·43	7
36	505	295	210	1·32	·4	·02	1·074	·03	86·46	5
37	490	290	200	1·28	·3	·02	1·074	·03	86·49	5
38	480	280	200	1·28	·2	·01	1·072	·02	86·51	3
39	465	270	195	1·27	·1	·01	1·070	·01	86·52	2
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)

TABLE 14. Oil per Stroke. (Calculated and Actual)			
Test	Oil per Stroke in Lbs.		Percentage Error.
	Calculated.	Actual.	
A	·000785	·000753	4·25
B	·000732	·000749	2·27
C	·000767	·000754	1·81

TABLE 15. Test Readings. Engine Test A.	
B.H.P.	12·88
R.P.M.	306·0
Exhaust Temperature. °C.	358
Oil per Min. in Lbs.	·1152
Air per Min. in Lbs.	3·3612
Oil per B.H.P. hour in Lbs.	·5365
Air per Lb. of Oil in Lbs.	29·17
Barometer Pressure. Lbs./in.^{2}	14·58
Pressure at End of Suction. Lbs./in.^{2} gauge.	0·07
Pressure at End of Exhaust. Lbs./in.^{2} gauge.	1·84

TABLE 16. Heat in Oil. Engine Test A.			
Crank Angle	Sum of Ordinates from Table	Heat in Oil/Stroke in C.H.U. (H.C.V.)	Heat in Oil/Stroke in C.H.U. (L.C.V.)
348·5	0	0	0
350	·78	·07	·07
352	4·45	·41	·39
354	8·41	·78	·73
356	12·56	1·17	1·10
358	16·89	1·57	1·48
360	21·61	2·01	1·89
2	26·61	2·48	2·33
4	31·68	2·95	2·78
6	36·80	3·43	3·23
8	41·95	3·91	3·68
10	47·09	4·38	4·12
12	52·16	4·86	4·57
14	57·06	5·32	5·01
16	61·64	5·73	5·39
18	65·80	6·13	5·76
20	69·60	6·47	6·09
22	73·13	6·80	6·39
24	76·41	7·11	6·69
26	79·54	7·40	6·96
28	82·61	7·68	7·22
30	85·61	7·93	7·46
32	86·26	8·03	7·55
34	86·39	8·04	7·56
36	86·46	8·05	7·57
38	84·51	8·05	7·57
40	86·52	8·05	7·57

total heat values were corrected by the same ratio of $\frac{\text{Actual fuel}}{\text{Calculated fuel}}$ and have been plotted, some in Fig. 57, p. 158, and some in Fig. 59, p. 166 .

The principal test readings in a final condition are given in table 15, p. 144 . In table 14, p. 144, are given, for the three tests A, B and C, both the actual oil/stroke and that obtained by the integration of the rate-of-injection curves.

Piston displacements were calculated at intervals of 2 degrees of crank angle from 350° to 80°, and at intervals of 10 degrees from 360° to 180° from the formula $x = r(1 - \cos\theta) + r(n - \sqrt{n^2 - \sin^2\theta})$ where r = crank length and $n = \frac{\text{Length of connecting rod}}{r}$

From the first mentioned series of piston displacements were calculated the actual cylinder volumes (in $ft.^3$) shown in column 2 of table 18, p. 148. The second series of values was used to construct the 'in phase' pressure-volume diagrams in Fig. 58, p. 159.

By means of the calculation which is given as a sample on pp. 149 and 150, the volume of the lb. molecule for the conditions at the end of suction were obtained and from this value, by simple proportion with respect to the actual cylinder volumes, values of $\bar{V}$ were calculated for a number of points in the stroke. The values of $\bar{V}$ thus obtained, although correct for the compression stroke, were considerably in excess of the truth for the working stroke on account of the increase in density of the cylinder contents during the processes of injection and combustion.

Table 17, p. 147, shows the attempt made to correct for the increase in density of the cylinder contents. The method employed was simple and consisted of calculating the weight of oil in the cylinder at any instant, adding this to the weight of air and products at the end/

end of suction, and then assuming this to be the correct total weight in the cylinder, not at the particular instant referred to but later by 8 degrees of crank angle. This was merely assumption, but it was based to some extent on the fact that there was an appreciable lag between the instant of commencement of injection and the commencement of combustion (See figs. 48-52, . pp. 113-120,) a matter of about 5 degrees at least and possibly even more at a later stage in the stroke. A few degrees either way, however, would not introduce any appreciable error. The last column of table 17, p.147 , gives the crank angle at which the corrections were actually made. In column (4) of this same table are the divisors by which the values of the lb. molecule volume $\bar{V}$ had to be divided for correction. Table 18, p.148, shows the process of correction.

From the values of pressure (absolute) and $\bar{V}$ (corrected) the corresponding temperatures in ^{o}C absolute were deduced from the formula. $T = \frac{144 p \bar{V}}{2780}$.
and are given in column (7) of table 18, p. 148 . These temperatures values are obviously very erratic, and were accordingly 'faired' before being used on the $T\phi$ field. This 'fairing process' has been shown in Fig. 50, p.118, and it is rather interesting to observe the wave effect of the uncorrected values of temperature, apparently due to indicator vibrations. The actual values required for transfer to the $T\phi$ field were those in columns (5) and (8) of table 18 , p. 148 .

TABLE 17. Corrections to $\bar{V}$. Engine Test A.				
Crank Angle	Sum of Ordinates from Table 13 Column 10.	Total Fuel in Lbs. (Corrected)	Correction Divisor for $\bar{V}$	Crank Angle at which Correction is to be applied.
348	0	·000000	1·000	356
350	·78	·000007	1·000	358
352	4·45	·000039	1·002	360
354	8·41	·000074	1·003	2
356	12·56	·000109	1·005	4
358	16·89	·000147	1·006	6
360	21·61	·000180	1·008	8
2	26·61	·000232	1·010	10
4	31·68	·000276	1·012	12
6	36·80	·000321	1·014	14
8	41·91	·000366	1·016	16
10	47·09	·000410	1·018	18
12	52·16	·000455	1·020	20
14	57·06	·000497	1·022	22
16	61·64	·000536	1·023	24
18	65·80	·000573	1·025	26
20	69·60	·000606	1·026	28
22	73·13	·000637	1·028	30
24	76·41	·000666	1·029	32
26	79·54	·000692	1·030	34
28	82·61	·000719	1·031	36
30	85·26	·000742	1·032	38
32	86·26	·000751	1·032	40
34	86·39	·000752	1·033	42
36	86·46	·000753	1·033	44
38	86·51	·000753	1·033	46
40	86·52	·000753	1·033	48
(1)	(2)	(3)	(4)	(5)

TABLE 18. Values for Tφ Diagram. Engine Test A.

Crank Angle	Actual Volume Ft.^{3}	$\bar{V}$ Ft^3/lb.mol (Uncorrected)	Correction Divisor	$\bar{V}$ Ft^3/lb.mol (Corrected)	Pressure Lbs./in.^{2} abs.	Temp. °C abs. (Uncorrected)	Temp. °C abs. (Corrected)
350	·0316	39·7	1·000	39·7	425	869	870
352	·0303	38·1	"	38·1	445	878	879
354	·0292	36·7	"	36·7	465	884	886
356	·0285	35·8	"	35·8	480	890	890
358	·0283	35·6	"	35·6	510	939	934
360	·0282	35·4	1·002	35·3	530	968	988
2	·0283	35·6	1·003	35·5	565	1040	1034
4	·0285	35·8	1·005	35·6	590	1088	1075
6	·0292	36·7	1·006	36·5	605	1144	1114
8	·0303	38·1	1·008	37·8	600	1175	1146
10	·0316	39·7	1·010	39·3	585	1191	1176
12	·0330	41·5	1·012	41·0	565	1201	1203
14	·0348	43·7	1·014	43·1	540	1206	1228
16	·0368	46·3	1·016	45·5	520	1225	1250
18	·0380	47·7	1·018	46·8	500	1213	1270
20	·0414	52·0	1·020	51·0	485	1282	1289
22	·0441	55·4	1·022	54·2	465	1305	1306
24	·0470	59·1	1·023	57·7	450	1344	1322
26	·0500	62·8	1·025	61·3	430	1364	1335
28	·0537	67·5	1·026	65·8	405	1381	1347
30	·0572	71·9	1·028	70·0	380	1378	1356
32	·0608	76·4	1·029	74·2	355	1365	1364
34	·0649	81·6	1·030	79·2	330	1354	1371
36	·0692	86·9	1·031	84·2	310	1351	1376
38	·0736	92·4	1·032	89·5	295	1353	1380
40	·0783	98·3	1·032	95·2	280	1381	1382
42	·0831	104·3	1·033	101·1	270	1414	1383
44	·0879	110·5	"	107·0	255	1411	1383
46	·0931	117·0	"	113·2	240	1405	1382
48	·0984	123·6	"	119·6	225	1392	1380
50	·1039	130·6	"	126·2	210	1371	1375
52	·1092	137·2	"	132·8	195	1340	1370
54	·1148	144·2	"	139·5	185	1336	1365
56	·1209	152·0	"	147·0	180	1370	1359
58	·1266	159·1	"	154·0	170	1357	1354
60	·1329	167·0	"	161·6	165	1380	1347
62	·1388	174·4	"	168·8	155	1354	1340
64	·1449	182·1	"	176·2	145	1322	1334
66	·1513	190·2	"	184·1	135	1287	1327
68	·1575	197·9	"	191·6	130	1289	1321
70	·1640	206·8	"	200·1	125	1293	1314
72	·1700	213·7	"	206·5	120	1282	1307
74	·1764	221·9	"	214·5	115	1278	1301
76	·1830	230·0	"	222·5	115	1326	1295
78	·1893	238·0	"	230·3	110	1312	1288
80	·1957	246·0	"	238·0	110	1355	1281
100	·2582	324·0	"	314·0	72	–	1170
120	·3112	391·0	"	378·0	55	–	1082
180	·3861	485·0	"	469·0	43	1050	1050
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)

CALCULATIONS FOR SUCTION CONDITIONS. (Engine Test A)

Percentage Composition of Fuel:-

Element	%	Equivalent wt. of O_2/lb. fuel
Carbon	85	2.2667 lbs.
Hydrogen	12.7	1.0192 lbs.
Sulphur	1.48.	.0148 lbs.
Oxygen) Nitrogen)	0.8	-
	Total	= 3.301 lbs.
	O_2 in fuel	= .004
	∴ O_2 in combustion air	= 3.297 lbs.

N_2 in this air $= \frac{3.297 \times 76.8}{23.2} = 10.914$ lbs.

N_2 in fuel = .004 lbs.

∴ Total N_2 = 10.914 + .004 = 10.918 lbs.

Combustion air reqd./lb. of fuel = 10.914 + 3.297 = 14.211 lbs.

Actual air/lb. of oil = 29.17 lbs. (See table 15, p.144)

∴ Excess air = 29.170 − 14.211 = 14.959 lbs.

∴ N_2 in excess air = 14.959 x .768 = 11.490

Total N_2 in products = 10.918 + 11.490 = 22.408 Lbs.

Total O_2 in products = O_2 in excess air = 3.469 lbs.

Per lb. of fuel:-

Element	Weight lbs.	Derived gas	Vol. of gas at 100°C & 14.7 lbs/in² V ft³.	Mol. wt. of gas m.	mV
C	.85	CO_2	34.730	44	1528.30
H_2	.1274	H_2O	31.225	18	562.07
N_2	22.408	N_2	392.370	28	10986.36
O_2	3.469	O_2	53.149	32	1700.77
S	.0148	SO_2	.227	64	14.53
		$\Sigma V =$	511.701	$\Sigma mV =$	14792.03

$$\therefore \text{ Mol. wt. of products}, m_p = \frac{\Sigma mV}{\Sigma V} = \frac{14792}{511.7} = 28.907.$$

Exhaust temperature, te = 358°C. (See table 15, p.144.)
$\therefore$ Te = 631°C. abs., and

Exhaust pressure = 1.84 lbs/in^2. (gauge) or 16.42 lbs/in^2 abs.

$$144\, p_e \bar{v}_e = \bar{R} \cdot Te$$

$$\therefore \bar{v}_e = \frac{2780 \times 631}{144 \times 16.42} = 742.52 \text{ ft}^3/\text{lb.mol. of exhaust gases.}$$

Clearance volume = .02816 ft^3. (See page 5.)

$\therefore$ Weight of exhaust gases in clearance $= \dfrac{.02816 \times 28.907}{742.52}$

= .0010965 lbs.

Total charge in cylinder at end of suction =(.021969 + .0010965) lbs.

= .023066 lbs.

Gas constant for air = 96.3.

Gas constant for products $= \dfrac{\bar{R}}{m_p} = \dfrac{2780}{28.91} = 96.16.$

$\therefore$ Gas constant for mixture $= \dfrac{W_a R_a + W_p R_p}{W_a + W_p} = 96.27.$

For mixture at end of suction $144\, p_1 V_1 = WRT_1$.

$$\therefore T_1 = \frac{144 \times .38609 \times 14.65}{96.27 \times .023066}$$

$$= \underline{368\ °C.\ abs.}$$

or t_1 = $\underline{95°C.}$

$\therefore$ Conditions at end of Suction are:-

p_1 = 14.65 lbs/in^2 abs.

V_1 = .38609 ft^3.

T_1 = 368°C. abs.

$$\therefore \bar{v}_1 = \frac{RT_1}{144\, p_1} = \underline{485.29 \text{ ft}^3/\text{lb.mol.}}$$

TEMPERATURE ENTROPY DIAGRAMS (Tests A,B andC)

The $T\phi$ diagram curves were drawn for the average composition of the cylinder contents during combustion as stated on p.134. The specific heat values used were those of Partington and Shilling for the quadratic law of variation of specific heat with temperature. For this average mixture the equation became

$Cv = 5.0316 + .1 \times 10^{-3} T + .3275 \times 10^{-6} T^2$.

where Cv = Sp. heat at constant volume in CHU./lb. mol. and T = $^{\circ}$C absolute.

The basic equation for the $T\phi$ diagram construction is derived as follows :-

The equation of heat addition to a gas is $dH = Cv \cdot dT + \frac{P.dv}{J}$

Assuming the quadratic law of specific heat variation with temperature,

$$dH = (a + sT + uT^2)\,dT + \frac{P.dv}{J}$$

or

$$\frac{dH}{T} = d\phi = (a + sT + uT^2)\frac{dT}{T} + \frac{\bar{R}}{J}\cdot\frac{dv}{v}$$

For a change in entropy from ϕ_1 to ϕ_2,

$$\phi_2 - \phi_1 = \int_{T_1}^{T_2}(a + sT + uT^2)\frac{dT}{T} + \int_{V_1}^{V_2}\frac{\bar{R}}{J}\cdot\frac{dv}{v}$$

$$a \cdot \log_e \frac{T_2}{T_1} + s(T_2 - T_1) + \frac{u}{2}(T_2^2 - T_1^2) + \frac{\bar{R}}{J}\cdot\log_e\frac{\bar{V}_2}{\bar{V}_1} \quad \text{.......(1)}$$

For heating at constant volume,

$$\phi_2 - \phi_1 = a \cdot \log_e \frac{T_2}{T_1} + s(T_2 - T_1) + \frac{u}{2}(T_2^2 - T_1^2) \quad \text{.......(2)}$$

Equation (2) is that of the constant volume heating curve of the $T\phi$ diagram (See table 19, p.152.)

The $T\phi$ diagram is similar to the form originated by Professor Goudie and thoroughly treated by him in a recent paper delivered to the Inst. of Engineers and Shipbuilders in Scotland, one difference being that what Professor Goudie terms the portable volume scale has been drawn for the present purposes in the form of the curve

$$\phi_2 - \phi_1 = \frac{\bar{R}}{J}\cdot\log_e\frac{\bar{v}_2}{\bar{v}_1}$$ (See table 20, p.152.)

which/

TABLE 19. VALUES FOR Tφ CONSTANT VOLUME CURVE.

TEMP. °C. ABS. T_2	T_2/T_1	$a \log_e T_2/T_1$	$s(T_2-T_1)$	$u/2(T_2^2-T_1^2)$	CONST. VOL. CURVE $\phi_2-\phi_1$
200	0	0	0	0	0
300	1·5	2·04	·01	·01	2·06
400	2·0	3·49	·02	·02	3·53
500	2·5	4·62	·03	·03	4·68
600	3·0	5·53	·04	·05	5·62
700	3·5	6·31	·05	·07	6·43
800	4·0	6·98	·06	·10	7·14
900	4·5	7·57	·07	·13	7·77
1000	5·0	8·10	·08	·16	8·34
1100	5·5	8·58	·09	·19	8·86
1200	6·0	9·02	·10	·23	9·35
1300	6·5	9·43	·11	·27	9·81
1400	7·0	9·79	·12	·31	10·22
1500	7·5	10·16	·13	·36	10·65
1600	8·0	10·48	·14	·41	11·03
1700	8·5	10·78	·15	·47	11·40
1800	9·0	11·08	·16	·52	11·76
2000	10·0	11·60	·18	·65	12·43
2200	11·0	12·76	·20	·79	13·75
2400	12·0	13·91	·22	·94	15·17

TABLE 20. VALUES FOR PORTABLE VOLUME SCALE.

$\bar{V}_2/\bar{V}_1$	1	2	3	4	5	6	7	8
$4{\cdot}574 \log_e \bar{V}_2/\bar{V}_1$	0	1·377	2·182	2·754	3·197	3·559	3·865	4·130
$\bar{V}_2/\bar{V}_1$	10	11	12	13	14	15	16	17
$4{\cdot}574 \log_e \bar{V}_2/\bar{V}_1$	4·574	4·761	4·937	5·094	5·242	5·380	5·507	5·626
$\bar{V}_2/\bar{V}_1$	18	19	20	21	22	23	24	25
$4{\cdot}574 \log_e \bar{V}_2/\bar{V}_1$	5·740	5·851	5·950	6·046	6·138	6·229	6·313	6·396
$\bar{V}_2/\bar{V}_1$	30	32	36	39	46	48	–	–
$4{\cdot}574 \log_e \bar{V}_2/\bar{V}_1$	6·756	6·884	7·117	7·279	7·607	7·690	–	–

TABLE 21. VALUES OF INTERNAL ENERGY.

TEMP. T_2 °C. ABS.	aT_2	$s/2 \cdot T_2^2$	$u/3\ T_2^3$	I_2 C.H.U./LB. MOL.
0	0	0	0	0
200	1006	2	1	1009
400	2013	8	7	2028
600	3019	18	24	3061
800	4025	32	56	4113
1000	5032	50	109	5191
1200	6038	72	189	6299
1400	7044	98	300	7442
1600	8051	128	448	8627

which gives the change in entropy for any change in volume of the gas. For convenience this curve has been broken into lengths, and to obviate the necessity for an additional vertical scale of $\frac{V_2}{V_1}$, this latter has been marked at intervals of 10 units along each length of curve.

The curve of Internal Energy was obtained from the equation, $I_2 - I_1 = a(T_2 - T_1) + \frac{s}{2}(T_2^2 - T_1^2) + \frac{u}{3}(T_2^3 - T_1^3)$

This latter, like that for entropy change is a standard equation derived as follows:-

$$dI = C_v . dT$$
$$= (a + sT + uT^2) . dT$$
$$\therefore I_2 - I_1 = \int_{T_1}^{T_2} . dI = \int_{T_1}^{T_2} (a + sT + uT^2) . dT$$
$$= a(T_2 - T_1) + \frac{s}{2}(T_2^2 - T_1^2) + \frac{u}{3}(T_2^3 - T_1^3)$$

(See table 21., p.152.)

By means of this curve, the internal energy can be determined at any desired temperature level. In this case it was used to determine approximately the quantity of heat carried away in the exhaust gases.

On the entropy field, which is drawn in Fig. 55, p.155, the diagrams for the three tests A, B and C are shown. To avoid confusion they have not been superimposed, but have been separated from each other by 1 unit of entropy. The distance between each pair of points along any one curve represents 2 degrees of crank angle, and it is obvious that the heating is very rapid towards the beginning of the stroke, where the points are wide apart and slowly decreases as combustion continues. The point for 180° crank angle was obtained by producing the expansion curve of the 'in phase' diagram to the end of the stroke, thus eliminating the slight drop in pressure in this region due to the opening of the exhaust. Towards the end of the stroke, the readings of pressure were not very reliable/

reliable and consequently the temperature values had to be liberally faired up. The 180° points, however, were not altered in any way and appear to be quite reasonable.

In each case the areas under the combustion and compression curves were integrated and the internal energy values determined both at the end of the combustion curve, that is at 180°, and at the beginning of the compression curve. These results being in CHU/lb. molecule have been converted to CHU/cycle - the conversion being readily obtained from the formula

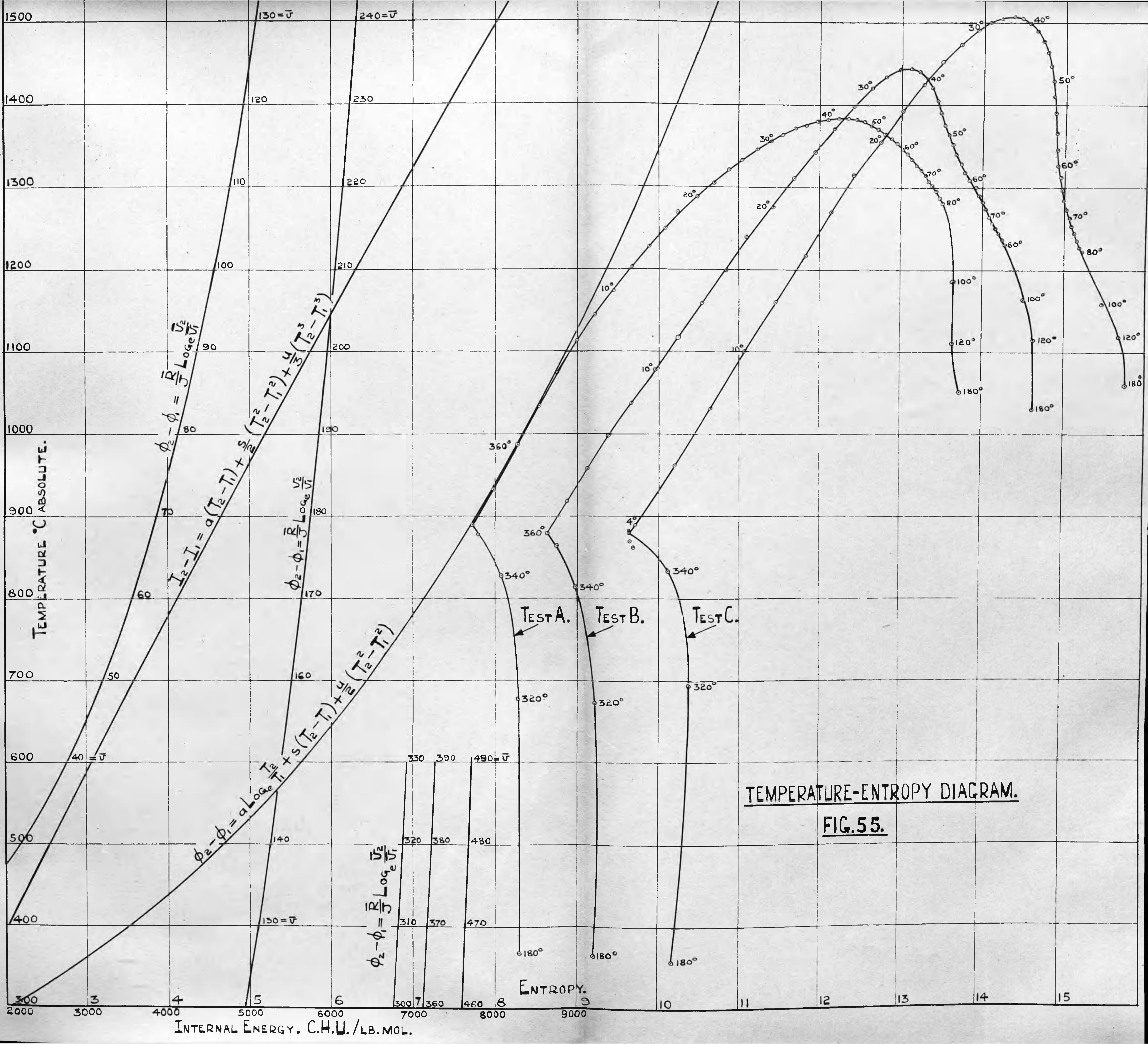
$$\text{C.H.U./cycle} = \text{C.H.U./lb.mol.} \times \frac{W}{m}$$

where W is the weight of the contents and m the molecular weight of same. These two quantities must be appropriately chosen with reference to the point in the cycle under consideration. (See p.135.) For instance in converting the heat under the combustion curve of the Tϕ diagram, W and m were taken as the mean weights of the contents in the cylinder before and after injection. Table 22, p.156, shows this process of conversion for test A.

The rates of heat reception (curves a'', b'' and c'') are plotted with the corresponding rates of fuel heat, calculated on a higher calorific value basis, injected (curves A'', B'' and C'') in Fig. 56, p.157; and the curves of actual heat reception (a, b and c) with those of actual fuel heat (H.C.V. basis) injected (curves A, B and C) in Fig. 57, p.158. The points for the curves of rate of heat reception were obtained from the 'faired up' curves in Fig. 57, p.158, and not from experimental points. The commencement of each of the rate of heat reception curves has been drawn as a straight vertical line. This is unlikely to be strictly correct but the indicator cards did not permit of any greater detail.

Fig. 58, p.159, shows the same curves as Fig. 57, p.158, plotted/

p.155

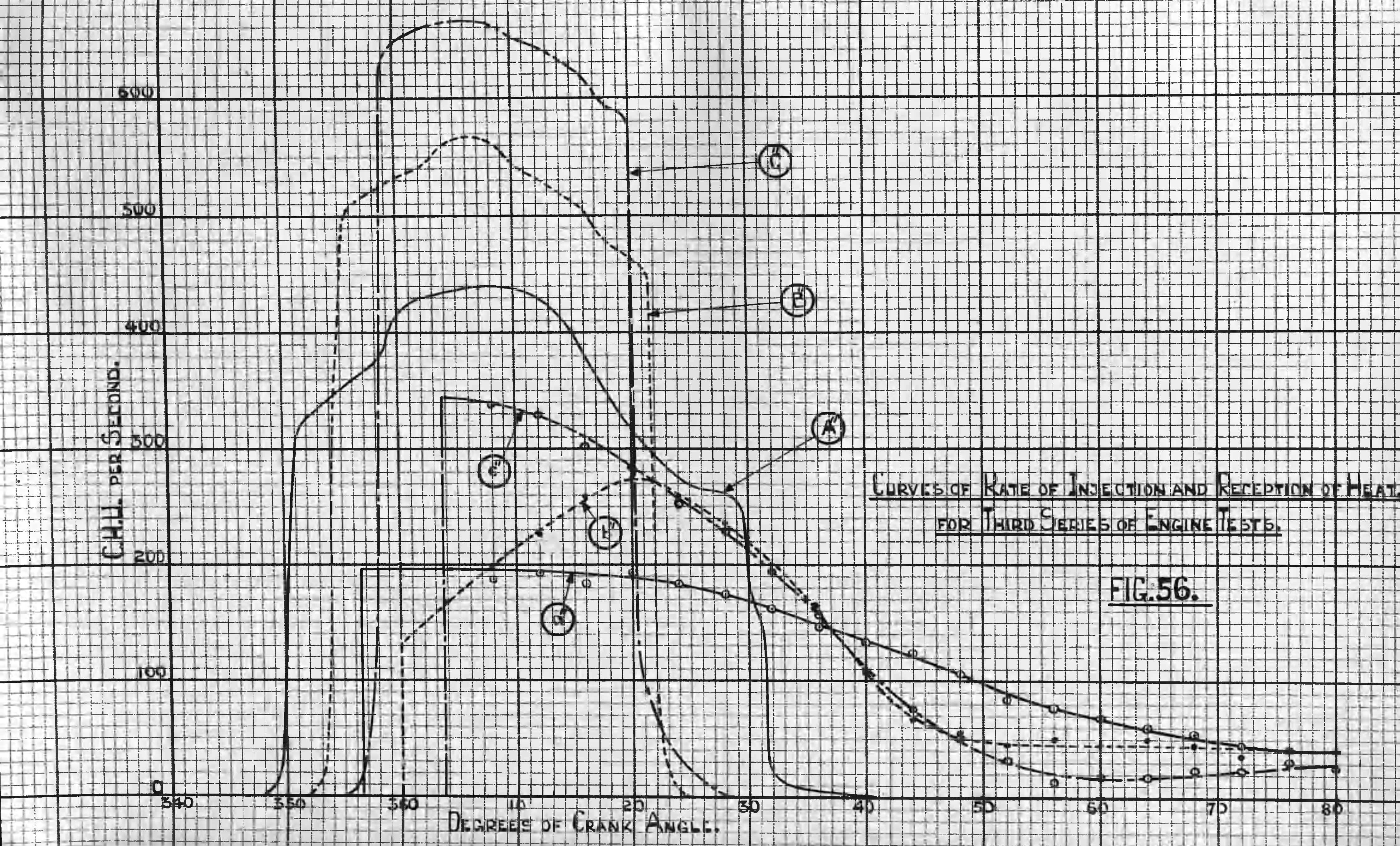


TEMPERATURE-ENTROPY DIAGRAM.

FIG. 55.

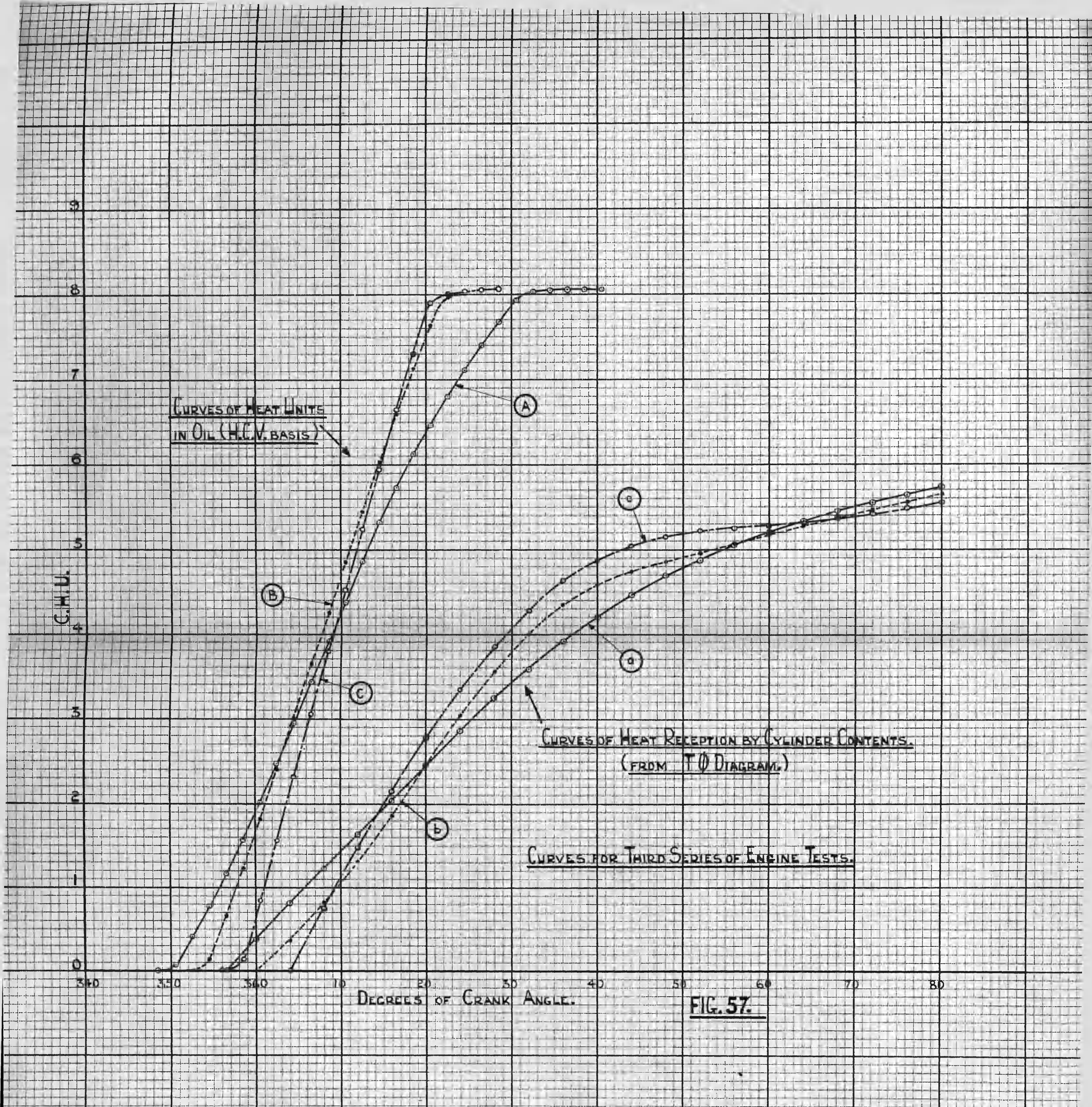
TABLE 22. Heat in Combustion Curve of TΦ Diagram. Engine Test A.						
Crank Angle	C.H.U/lb.mol Increments.	C.H.U./stroke Increments	C.H.U/stroke Sum	C.H.U/stroke Sum Corrected	C.H.U/stroke Increments Corrected	C.H.U/second Corrected
356	0	0	0	0	0	–
360	516	·412	·412	·38	·43	198
4	496	·396	·808	·81	·43	198
8	501	·400	1·208	1·22	·41	188
12	528	·422	1·630	1·64	·42	193
16	527	·421	2·051	2·04	·40	184
20	508	·406	2·457	2·46	·42	193
24	509	·407	2·864	2·86	·40	184
28	481	·384	3·248	3·24	·38	175
32	420	·336	3·584	3·59	·35	161
36	397	·317	3·901	3·91	·32	147
40	372	·297	4·198	4·20	·29	133
44	346	·277	4·475	4·47	·27	124
48	277	·221	4·696	4·70	·23	106
52	234	·187	4·883	4·88	·18	83
56	219	·175	5·058	5·05	·17	78
60	176	·141	5·199	5·20	·15	69
64	161	·129	5·328	5·33	·13	60
68	146	·117	5·445	5·45	·12	55
72	118	·093	5·538	5·55	·10	46
76	130	·104	5·642	5·64	·09	41
80	103	·082	5·724	5·73	·09	41
180	235	·191	5·915	5·92	·19	–

TABLE 23. Heat Loss During Combustion. Engine Test A.						
Crank Angle	Wall Area A in.^{2}	$T_G - T_J$ (mean) °C	$A(T_G - T_J)$ $\times 10^{-6}$	Area under Curve (incrts.)	Heat Loss per Stroke C.H.U.	True Combust. C.H.U./Stroke
360	150	690	72	248	·01	·39
10	152	880	119	940	·04	1·48
20	162	990	159	1380	·10	2·56
30	177	1060	198	1780	·16	3·58
40	196	1080	230	2120	·24	4·44
50	220	1080	258	2420	·33	5·12
60	246	1050	271	2640	·43	5·63
70	275	1010	281	2760	·54	6·04
80	305	980	293	2860	·65	6·38



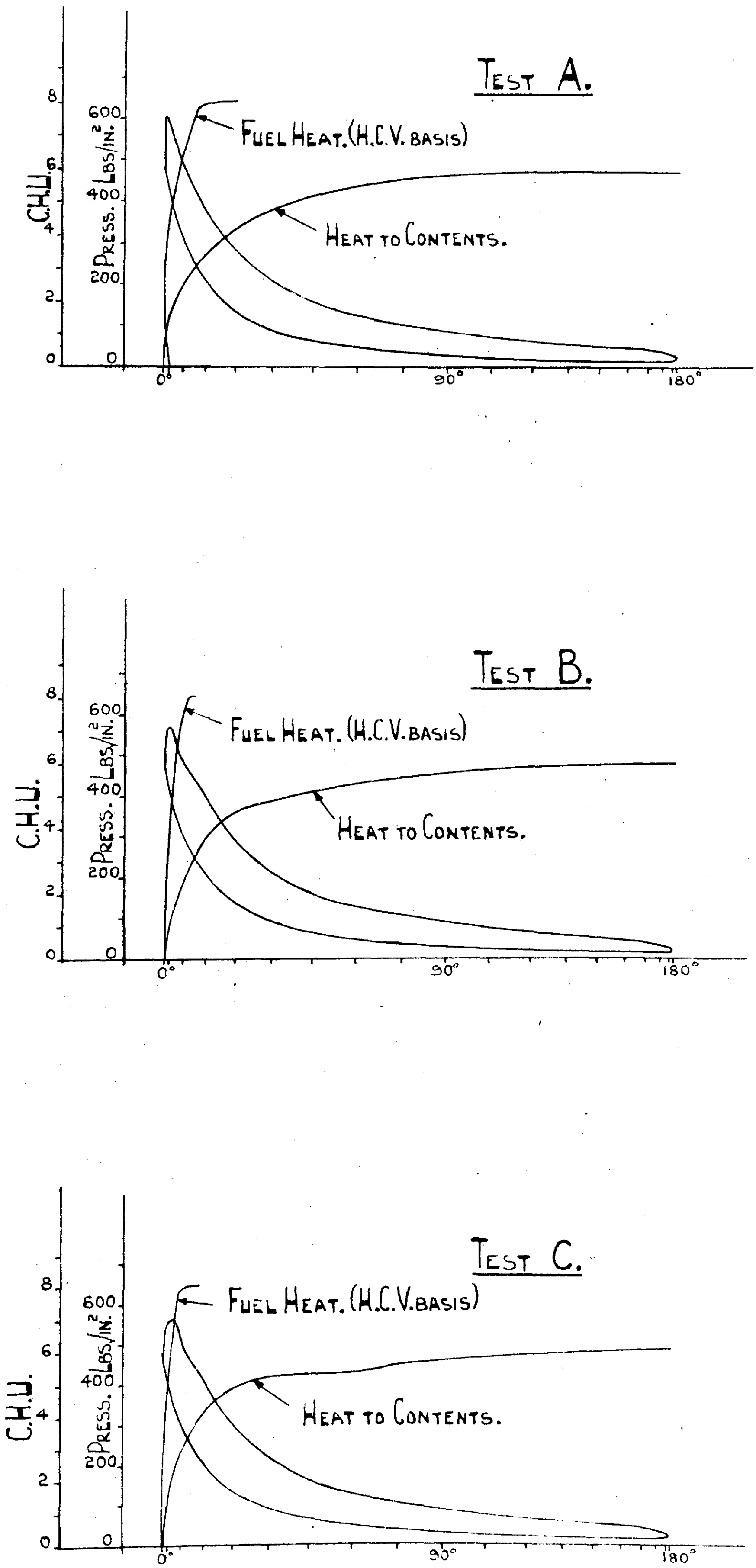
CURVES OF RATE OF INJECTION AND RECEPTION OF HEAT, FOR THIRD SERIES OF ENGINE TESTS.

FIG. 56.



Curves of Heat Units in Oil (H.C.V. basis)
A
B
C
a
b
c
Curves of Heat Reception by Cylinder Contents.
(from TΦ Diagram.)
Curves for Third Series of Engine Tests.
C.H.U.
0
1
2
3
4
5
6
7
8
9
340
350
360
10
20
30
40
50
60
70
80
Degrees of Crank Angle.

FIG. 57.



'IN PHASE' INDICATOR DIAGRAMS FOR THIRD SERIES OF ENGINE TESTS.

FIG. 58.

plotted on a base of engine stroke along with the respective 'in phase' indicator diagrams of the tests. It might have been preferable to have plotted the curves of Fig. 59, p. 166, but the difference anyway would be imperceptible on such a scale. Fig. 58, p. 159 gives a better impression of injection and combustion conditions relative to the actual stroke of the engine.

A possible error which has not been taken into account, is that due to the time lag which seemingly must occur between any chamge of pressure state in the combustion chamber and the corresponding effect of this change at the piston of the indicator. Since any allowance for this lag is, under the circumstances, likely to be only approximate, the results so far have been treated without reference to it. Later in the stroke, the introduction of various other sources of error is clearly liable to swamp this lag effect, but it must be accounted for, if any true estimate is desired of the ignition lag; that is the period of time which elapses between the first entrance of fuel into the combustion chamber and the instant of ignition of this fuel. Obviously the first difficult question to be answered is:- Where in the combustion chamber will the first particle, or group of particles of fuel, ignite? Clearly some assumption must be made. It seems probable that the first impulse of pressure will travel as a wave from the source to the indicator piston, and if this is the case, the speed of the wave motion must be considered.

The speed of sound is given in Watson's Physics as

$$V = 33060 + 60{\cdot}5\,t \text{ cms./sec.} \quad \text{.........} \quad (1) \qquad (t = {}^{\circ}C)$$

apparently for moderate pressures. Even if this were assumed to hold approximately for pressures such as prevailed at the end of compression in the engine, one cannot assume that the temperature in the air inside the long passage ($\frac{3}{8}''$ dia.) leading to the indicator is/

is the same as that calculated for the gases in the combustion chamber. (See figs. 50-52, pp.118-120) Clearly it must be considerably less than this owing to the much higher ratio of 'surface area' to 'volume enclosed'. The irregular path that the wave has to travel (See fig.4.(b)p. 9.) is also likely to add to the complication, as also must possible gas velocities and the degree of turbulence.

Further, as stated in "Flame and Combustion in Gases" by Bone and Townend, experiments were carried out by Vieille in 1899 on the velocity of what he termed 'shock waves' in various gases. He obtained the waves by the bursting of a celluloid diphram under pressure or by the explosion of a small charge of Fulminate of Mercury at one end of the glass tube, and showed that the waves were propagated at velocities greater than the speed of sound under like conditions, at least for some distance from the origin. It is further stated that on continued propagation the shock wave gradually loses its extra energy and eventually becomes an ordinary sound wave. Vieille found that his values satisfied the following equation,

$$V = \sqrt{p_0/d_0 \left(\gamma + \frac{\gamma+1}{2} \times \frac{p_1 - p_0}{p_0} \right)} \quad \text{- - - - - - - - - -} \quad (2)$$

where p_0 and d_0 = initial pressure and density in the medium and p_1 = pressure in the wave; γ = Ratio of sp. heats. These experiments were apparently carried out at relatively low pressure.

The theory of 'shock waves' clearly suggests a further possible treatment of the lag problem, but even on the assumption that the above equation might give the desired result it is clearly inapplicable on account of the p_1 term, which has no corresponding function in the case under consideration.

Owing to a lack of the neccessary data, therefore, it is only possible to obtain from the above equations rough estimates of the time that the propagation of the/

the pressure impulse is likely to require in travelling along the given path, say 17". (See Fig.4(b)p.9.). Taking first equation (1) and assuming an air temperature of half that in the combustion chamber, and therefore about 300°C, the equation becomes

$$v = 33060 + 300 \times 60.5 = 51,210 \text{ cms/sec.}$$

or practically 20,000 ins/sec.

$$\text{Time required to travel } 17'' = \frac{17}{20,000} \text{ sec.}$$

$$= .00085 \text{ sec.}$$

$$\text{At 300 R.P.M. 1 degree of crank angle} = \frac{1}{5 \times 360} \text{ sec.}$$

$$= .00055 \text{ sec.}$$

$$\therefore \text{Time lag} = \frac{8.5}{5.5} = \underline{1.5 \text{ degrees of crank angle.}}$$

Taking now equation (2) and assuming $p_1 = p_o$ and the same temperature of 300°C. ($p_o = 485$ lbs./in.^{2})

$$d_o = \frac{144 \times 485}{96 \times 573 \times 32} = .0396 \frac{\text{lbs/ft}^3}{g}$$

$$v = \sqrt{\frac{p_o}{d_o} \cdot \gamma} = \sqrt{\frac{485 \times 1 \cdot 392 \times 144}{\cdot 0396}}$$

$$= 1,570 \text{ ft./sec. or } 18,840 \text{ ins./sec.}$$

$$\therefore \text{Time Lag in this case} = \underline{1 \cdot 59 \text{ degrees of crank angle.}}$$

It is clear from the second result that the assumption $p_1 = p_o$ does not give a fair comparison, since the value of velocity given by equation (2) should at least exceed that given by (1).

The 17 inches of travel assumes that the first ignition of fuel takes place at a point well within the volume of the combustion chamber, but that it allows accurately for additional effects such as constrictions, corners, turbulence etc it would be impossible to claim. After the first pressure impulse has reached the indicator, further effects such as wave interference must, it seems, add to the complication.

It/

It is assumed here that the 1.5 degrees of lag is sufficiently near the truth for the first pressure impulse, and also for simplicity that this lag is a constant quantity throughout the combustion period of the fuel. The correction for this lag has been applied to the results of actual combustion discussed in the next section.

Actual Rate of Combustion.

The analysis of the T.ϕ. diagrams was advanced a stage further in an attempt to determine the actual rate of combustion for tests A, B and C. For this purpose additional assumptions were necessary.

In his interesting series of articles recently published in 'Engineering', entitled "The Transfer of Heat in Reciprocating Engines", Dr. Nagel has treated on various attempts that have been made to obtain a mathematical solution to the problem of heat loss from the gases to the cylinder walls etc. during the working cycle of the internal combustion engine.

One solution, due to Professor Eugen Meyer of Berlin in 1899 appeared to be of especial interest, probably on account of the fact that it was obtained by means of indicator card analysis. He assumed that the rate of heat loss per unit area of wall was dependent on some power of the temperature difference between the gas and the walls, and proved, by the analysis of indicator cards and a knowledge of the total heat loss, that this power was practically equal to 2. By this means he obtained the equation

$$dQ = c.F\,(T - T_w)^2.\,dz.$$

where Q = heat quantity, c = constant, T = gas temperature, T_w = wall temperature, z = time interval and F = wall area. The/

The assumption on which this was based, has been shown by more recent work to be not strictly correct. Dr. Nagel states that the coefficient of heat transfer increases with the temperature difference and to a less extent with density.

A careful study of Dr. Nagel's articles shows that the subject is one of immense complication, and since it was impossible to enter into this fresh investigation, it was decided to make use of the formula devised by Professor Meyer, on the understanding that it was only likely to give a rough estimate of the actual rate of heat loss.

The total wall area enclosing the cylinder contents was therefore calculated approximately for a number of positions of the piston. These areas were multiplied by the corresponding values of $(T-T_w)^2$. (T_w was taken as the mean jacket water temperature). The values then obtained on a base of crank angle gave a curve of rate of heat loss multiplied by a constant, and by integration, the total heat loss multiplied by a constant.

By equating this complete integral to the difference between the total heat in the fuel per stroke (on a basis of mean lower calorific value) and the total heat under the combustion curve of the $T.\phi$.diagram, the heat loss could be distributed over the combustion period. The latent heat of the steam formed in combustion will not have been available during the working stroke. The foregoing, of course, assumes complete combustion at 180 degrees of crank angle. The addition of this distribution of heat to that of the $T.\phi$. combustion curve then gave the actual heating value of the combustion process. (See table 23, p.156)

Curves a', b' and c' representing this actual combustion are shown in Fig. 59, p. 166. The three remaining/

remaining curves A', B' and C' on the same diagram were drawn to represent total heat in the fuel on a basis of lower calorific value.

The significance of each of the four sets of curves in Fig.57, p.158, and 59, p.166, will now be apparent. Curves A, B and C with respect to the other three groups do not truly represent the maximum possible heating capacity of the fuel, but rather the latter together with the latent heat in the steam produced. The curves really represent potential heat units. Curves A', B' and C' on the other hand represent approximately the true heating capacity of the fuel under conditions of ideal combustion, while by a, b and c is represented the heating ofthe cylinder contents as determined by the $T.\phi$ combustion curves (neglecting indicator lag). The remaining curves a', b' and c' represent as nearly as could be determined the actual combustion conditions (in terms of heat units) and are therefore directly comparable with the corresponding group consisting of A', B' and C'.

The twelve curves in Figs.57, p.158, and 59, p.166 embody the product of this series of experiments. They suggest two points that appear to be of interest regarding (1) Ignition lag and (2) Combustion lag. These points are confined to Fig. 59, p. 166 .

(1) It is apparent that the ignition lag is not appreciably affected by the initial pressure of the fuel oil as the valve is opening. This lag seems to be approximately 5 degrees of crank angle or about .0028 seconds. Regarding this lag or 'wait period' between the first entry of fuel and its ignition, Mr. Bird of Cambridge University, in his paper on "Oil Jets/

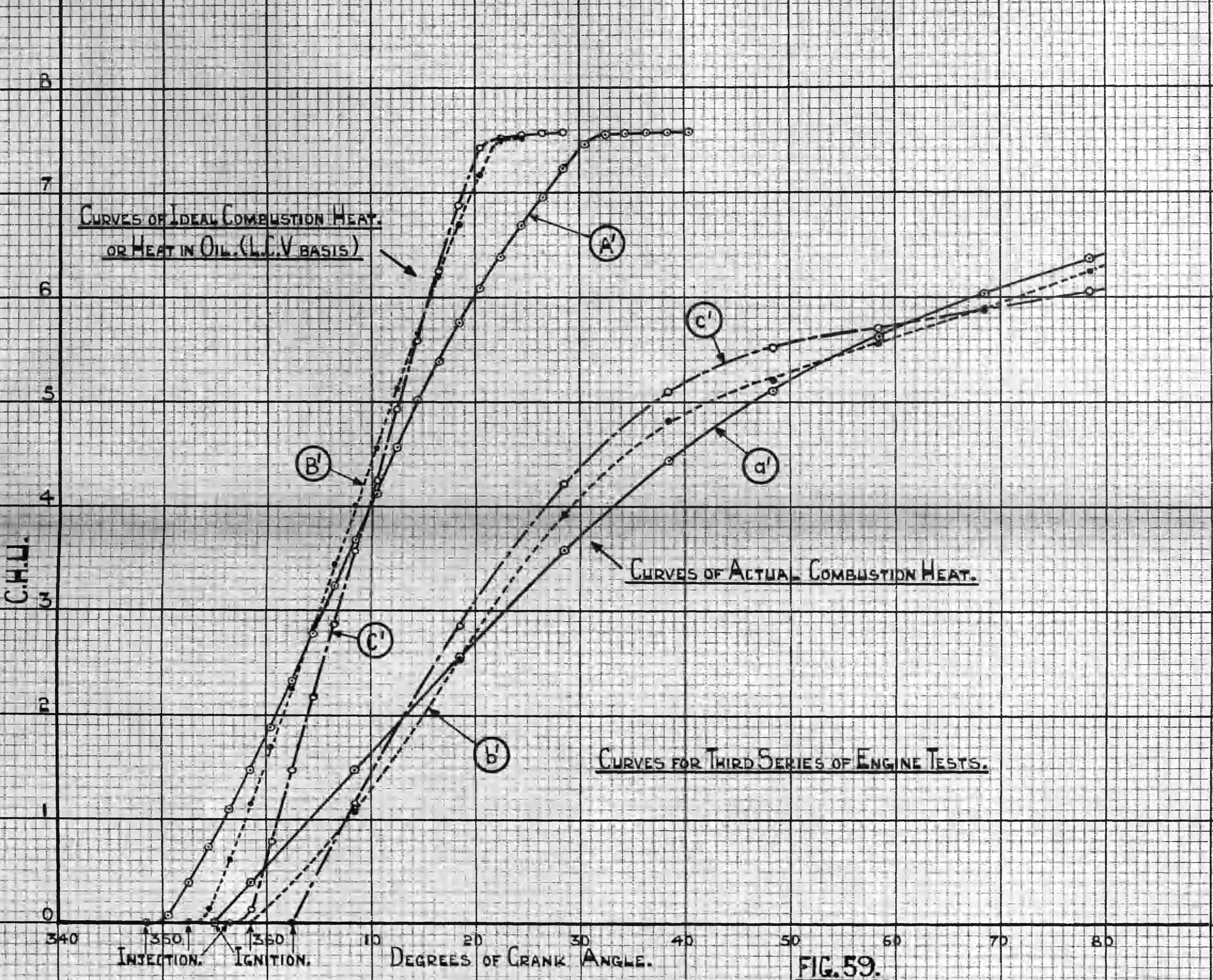

Curves of Ideal Combustion Heat.
or Heat in Oil. (L.C.V basis)
Curves of Actual Combustion Heat.
Curves for Third Series of Engine Tests.
A'
B'
C'
a'
b'
c'
C.H.U.
Degrees of Crank Angle.
Injection.
Ignition.
0
1
2
3
4
5
6
7
8
340
350
360
10
20
30
40
50
60
70
80

FIG. 59.

Jets and their Ignition" gives a 'surface' obtained from his experiments, showing the values of this 'wait period' for injections into a combustion chamber containing air at various conditions of temperature and density. Unfortunately his range of conditions does not cover these present in the Diesel engine. All his values of 'wait period' considerably exceed the value of .0028 seconds, but it could be imagined that the 'surface' produced to the oil engine conditions would attain a value of the same order. Considering that the results were obtained in the static conditions of an experimental combustion chamber, even such a tendency is rather enlightening; the indication being that turbulence does not assist much in eliminating this initial lag. This will certainly be the case where the turbulence causes the oil particles to be thrown into contact with the cool walls surrounding the combustion chamber.

(2) The curves in Fig. 59, p. 166, show that the rate of combustion is, as regards the main body of the fuel, definitely dependent on the rate of injection, and in addition, that the rate of combustion could probably be further increased by an increase of the rate of injection. Widely different conditions of atomisation and penetration would naturally be expected to exist between the three cases and to have considerable effect on the rate of combustion, but the simple process of getting the fuel into the combustion chamber seems, at least in this series of tests, to be practically the whole matter. Fig. 59, p. 166 indicates that for $\frac{3}{4}$ total wt. of fuel, the rate of combustion is roughly half the rate of injection. The term 'rate of combustion' is here used/

used in a very general way, meaning more strictly the 'rate of generation of heat by the combustion'. Clearly each constituent of the fuel may burn at a different rate, depending partly on its affinity for oxygen and partly on the supply of the latter available, and no one rate, therefore, can actually cover all the complex processes taking place. Only through reliable sampling could this more detailed study of combustion be approached.

The curves of rate of heating in Fig. 56, p. 157, suggest that in general the rate of combustion starts at about its maximum value and gradually but with increasing rapidity falls to a much lower value. By the time it has reached this lower value most of the fuel has been consumed. After this point the curves show, as might be expected, a certain amount of 'after-burning', and are therefore in agreement with the recognised failing of fuel combustion in general.

Table 24, p. 169 has been drawn up in the form of a heat balance, and shows the principal features obtained from the T ϕ diagrams and test readings. The values of 'possible heat to work' should check with 'heat equivalent if I.H.P.'. The error in test C is rather high but the average error (3.4% of the total heat input)/

TABLE 24. Heat Quantities per Cycle.			
Item.	C.H.U. / Cycle.		
	Test A	Test B	Test C
Total Heat in Fuel. (H.C.V. basis)	8·05	8·00	8·06
L.C.V. of Fuel.	7·57	7·52	7·58
Heat lost to Exhaust (except Latent Heat in Steam).	2·87	2·84	2·93
Heat lost in Compression.	·38	·39	·41
Heat accounted for by Combustion Curve			
of TΦ Diagram.	5·92	5·98	6·05
Possible Heat to Work.	2·67	2·75	2·71
Heat Equivalent of I.H.P.	2·89	2·99	3·08
Heat Equivalent of B.H.P.	1·98	2·03	1·98
Mechanical Efficiency %.	68·7	67·9	64·3
Indicated Thermal Efficiency % (H.C.V. basis).	35·9	37·3	38·2
Brake Thermal Efficiency % (H.C.V. basis).	24·6	25·4	24·5

input) for all three cases is not more than might be expected. The results in this table indicate what appears to be a disadvantage of a cycle approaching (as in test C) constant pressure combustion, viz - an exceedingly low mechanical efficiency. This may have been due to the higher pressures and possibly the higher temperatures prevailing during the earlier part of the piston travel. (See Figs.50-52,pp.118-120) It is interesting to note that as the overall thermal efficiency on an I.H.P. basis increases from A to C, the mechanical efficiency decreases. On combining these two factors,case B proves to be the best with an overall brake thermal efficiency of 25.4%

The table also brings out notably what an extremely inefficient form of heat-to-work converter an oil engine actually is. It is supplied with 8.05 C.H.U. per cycle (taking case A). During the working stroke it loses 1.65 of these, mainly to the cooling water. It then loses 2.87 (plus .48 of latent heat) to the exhaust. Over and above this it dissipates .91 C.H.U. in internal friction (of which about half is generally supposed to be due to the piston) and another .38 C.H.U. during the compression stroke (mainly to the jacket water). In the end it is left with 1.98 of the 8 C.H.U. to perform its useful work. It is evident that the exhaust heat is the greatest individual loss; although if the jacket heat had been measured by the conditions of circulating water, and the exhaust heat by means of a calorimeter, the former would quite possibly have appeared as great as, or even greater than the latter. The jacket heat as measured in this manner, however, not only represents the heat lost by the gases during combustion but in addition includes/

includes a certain amount of the exhaust heat transferred during the exhaust stroke, possibly a considerable amount of the piston friction losses and also the heat given up by the gases during compression. The exhaust heat, on the other hand, measured by the calorimeter, represents not the true heat lost to the exhaust, but that quantity diminished by the amount transferred to the jacket and otherwise dissipated during the exhaust stroke before the gases have reached the calorimeter.

- - - - - - - - - - - - - - - -

EFFECT OF INERT GASES ON IGNITION AND COMBUSTION.

It has been shown that under normal conditions there is a lag of about 5 degrees crank angle between the beginning of injection of fuel and the beginning of its combustion. This however only applies to the first portion of fuel entering the combustion chamber and therefore impinging on a mass of practically pure air. The case of the last portion of the fuel to enter is likely to be rather different, when the oxygen content of the combustion chamber has been considerably depleted and the inert-gas content (products of combustion and nitrogen) correspondingly enlarged.

The following experiment was carried out in order to investigate the effect of the presence of varying proportions of inert gases, on the ignition and combustion of the fuel. For this purpose the gasometer described on p.136 was utilised. It was first filled with a certain quantity of air drawn from the atmosphere and then smaller quantities of nitrogen from a cylinder were added as a diluent.

The method of taking readings, all in the form of displaced indicator cards, was as follows. The engine was run steadily under normal conditions, drawing its air supply from the air boxes (see p.136). The desired mixture of air and nitrogen was meantime obtained in the gasometer. By means of the valves at the T piece situated between the gasometer and the air boxes, the engine was suddenly switched over to the gasometer, a period of 2 to 3 cycles allowed to elapse to clear the connecting pipe of air, then an indicator diagram of 3 or 4 cycles was taken and the engine switched back to/

to the air boxes again. In one or two cases of low percentage O_2 content during the above process the speed dropped as much as 10 R.P.M., but generally the slowing was imperceptible.

It will be clear that any slowing down of the engine before taking an indicator card would have the effect of advancing the point of injection (in the same way as an increase of brake load) and therefore the results of increased lag which have been obtained, if affected at all, will have been minimised rather than exaggerated.

The suction damping device described on p.136 had to be detached for these experiments to enable an estimate to be made of the volume of air to be cleared from the suction pipe, and this had the effect, when drawing from the gasometer, of reducing slightly the compression pressure. Before each test the pipe from the gasometer to the valves at the T piece junction was cleared of gases of doubtful composition by switching the engine over to the gasometer for a few strokes.

Nitrogen was used as the diluent in these tests in order that as nearly as possible only the effect of the dilution of the oxygen might enter the problem. Considering that air is 80% pure nitrogen, the alteration to the density of the working substance was clearly negligible under the circumstances. A gas such as CO_2 might have been used in place of N_2 but in doing so the effects of density and possibly temperature would have added unnecessary complication to the problem.

Nitrogen was added to the gasometer until the fuel injected practically failed to ignite.

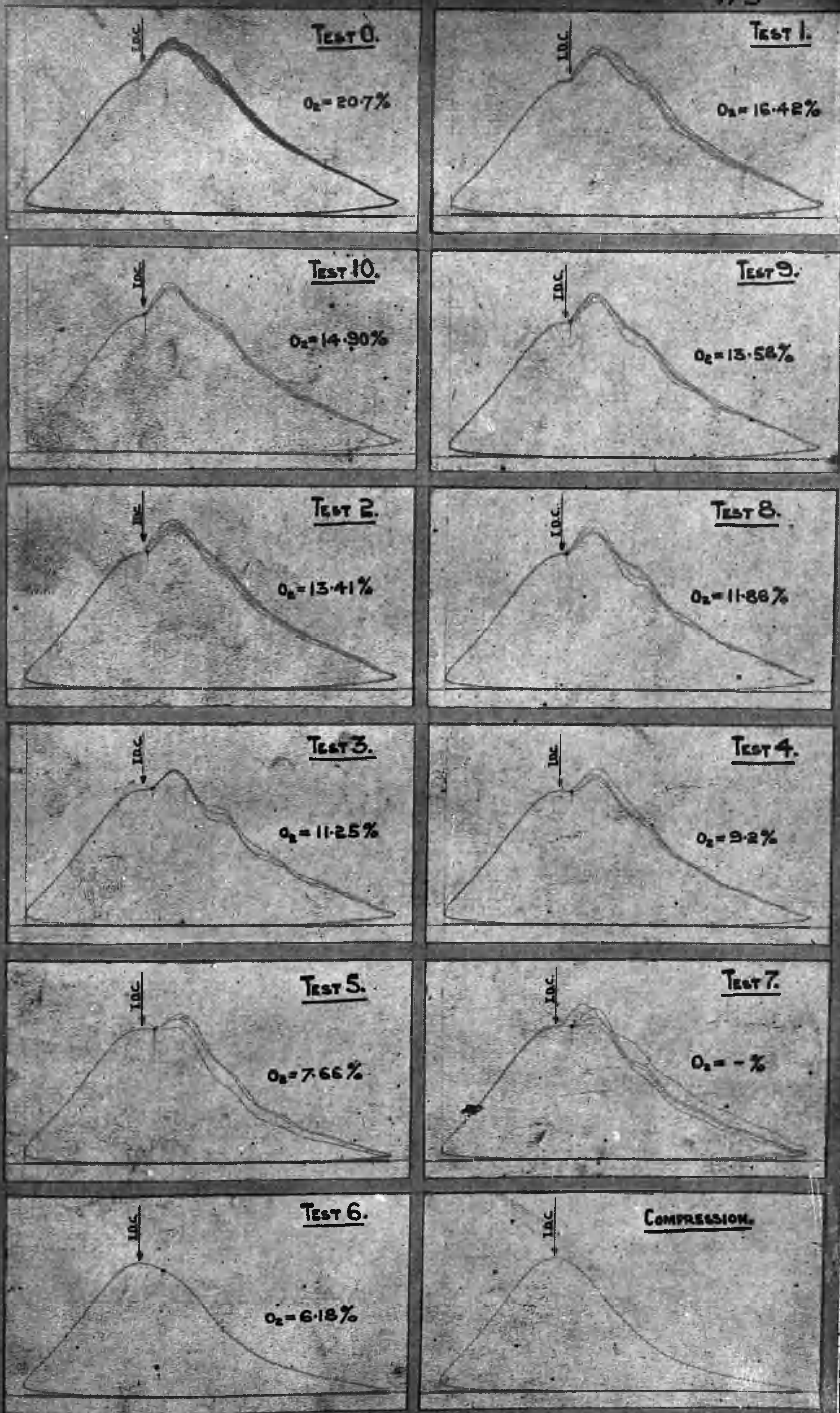
Fig.60,p.175 shows the actual indicator cards obtained

In/

In order to bring out more clearly the effects on combustion, the diagrams were 'faired up' and superimposed. This combined diagram is shown in Fig.61, p.176. On Fig.62,p.176 are shown the results of ignition lag plotted against the corresponding values of percentage oxygen content. In each case the distance of the ignition point along the diagram was measured and the lag calculated on the assumption of a lag of 5 degrees for 'pure air' conditions. Strictly speaking, a slightly greater lag than 5 degrees should have been added since, with the engine drawing from the gasometer, the compression pressure was slightly reduced. The addition of only 5 degrees, however, corrects for this compression effect, and brings the results into line with normal running conditions.

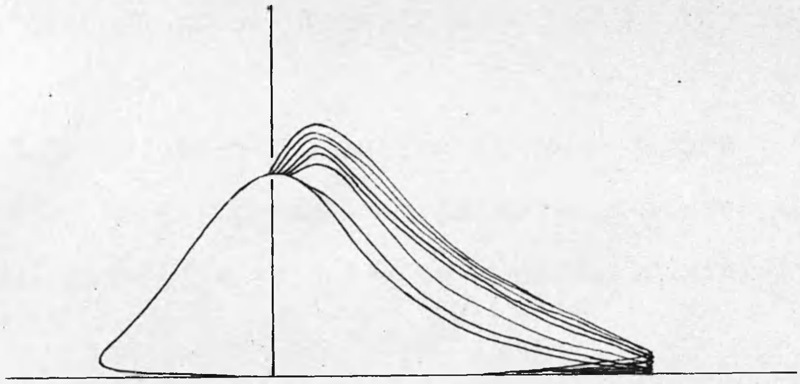
Fig.62,p.176,shows that as the O_2 percentage content within the combustion chamber is reduced, the ignition lag steadily increases to a certain - apparently critical - point. Beyond this point, with further reduction of the O_2 content, the lag at first only slightly increases but very rapidly after this it becomes indefinite and probably increases to infinity. This suggests a second critical point. The indicator diagrams show that around these critical points the combustion itself appears to be rapidly snuffed out. The full line curve in Fig.62,p.176 has been modified in the region of the critical points by the dotted curve - the form of the latter being the more probable, as suggested by the results.

It is interesting to note that at the first critical point the O_2 percentage content is about 10.5% or practically half that of pure air, and that the lag is almost twice that for pure air (neglecting



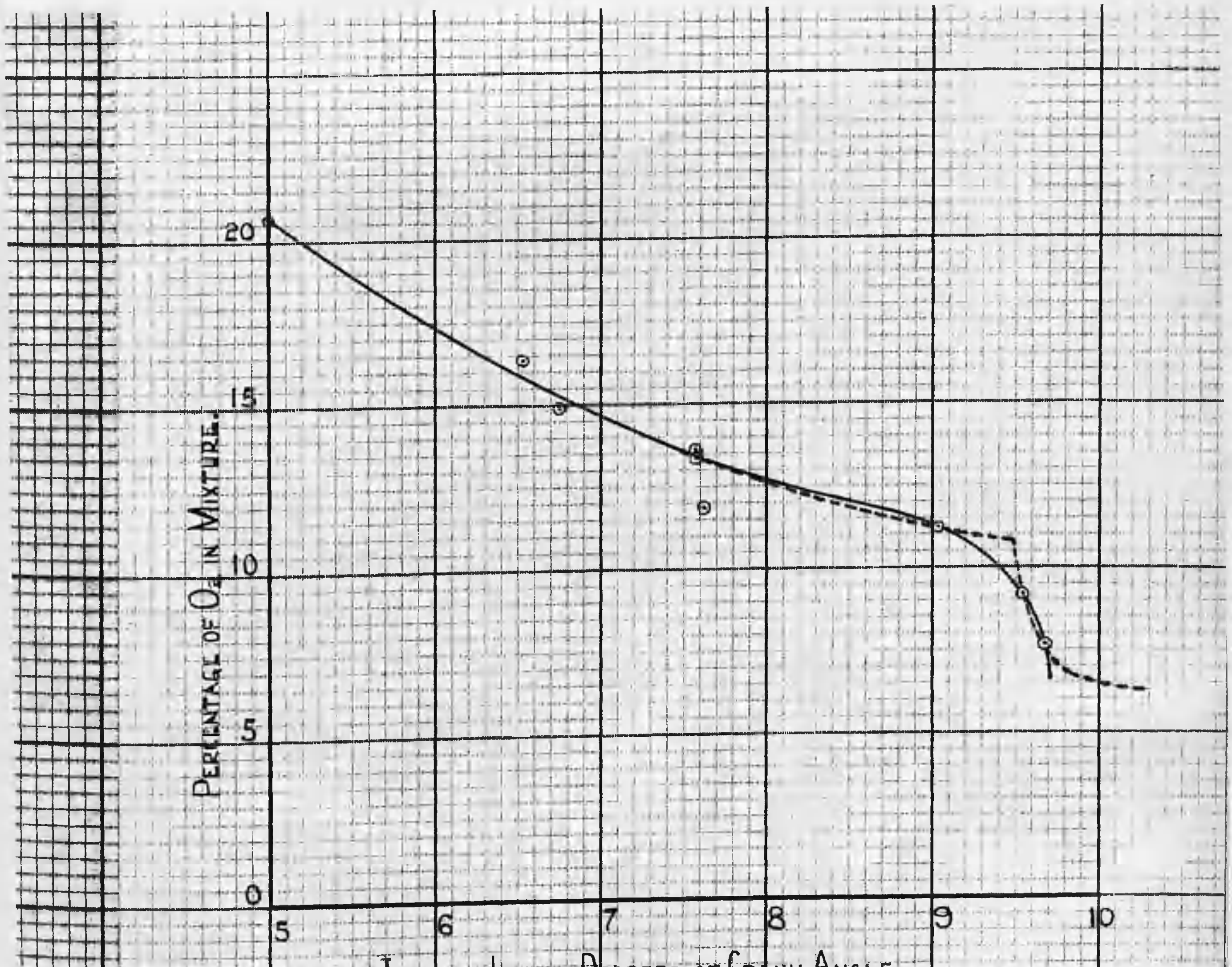
Effect of Inert Gases on Ignition and Combustion.

FIG. 60.



INDICATOR DIAGRAMS FOR NITROGEN EXPERIMENTS.

FIG.61.



EFFECT OF INERT GASES ON IGNITION LAG.

FIG.62.

the products of combustion present under normal running conditions).

From 10.5% to about 7% the lag remains almost constant but by 6% it has again considerably increased and is becoming indefinite owing to combustion rapidly dying out.

Clearly a fuller treatment of this subject and analyses of the combustion process might yield some interesting results. The fact that in the composite diagram in Fig.61,p.176, none of the combustion curves for reduced O_2 content overlap that for pure air suggests that combustion, possibly just complete for pure air, becomes more and more incomplete as the O_2 content is reduced, on account of the rate of combustion diminishing.

The point stressed by these experiments therefore is the necessity for efficient spraying of fuel, and that the rapidity of ignition and combustion will depend very largely on the purity of the air into which the fuel is injected. The results show that an effect, similar to what is called 'after-burning' might be brought about by the last portion of the injection impinging on gases already considerably depleated, or devoid of their O_2 content.

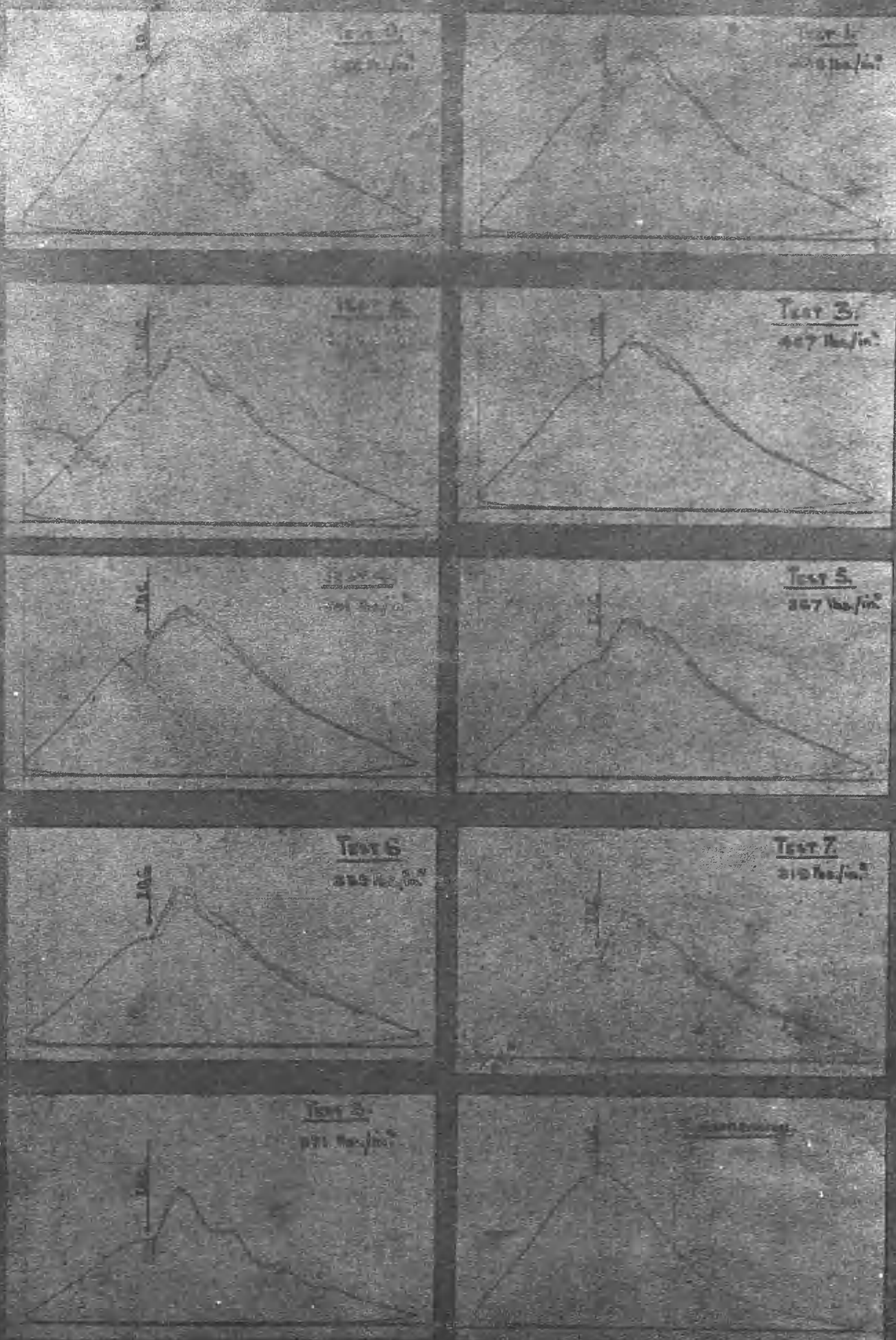
Under normal conditions, as injection and combustion proceed, the temperature of the gases increases, and the density decreases (on account of the outward movement of the piston). Mr. Bird, in the paper already referred to on p.165 shows that while an increase of temperature reduces the ignition lag, a decrease of density increases it and it is possible that these mutually opposing factors may modify the effect on lag caused by the reduction of O_2 content already discussed. It might be/

be emphasised, however, that whether or not the lag does actually increase during the process of injection and combustion, the results of the foregoing experiments show that it will nevertheless tend to do so and conditions should be such that the inert gas effect is reduced to a minimum and advantage taken as far as possible of the counteracting temperature effect.

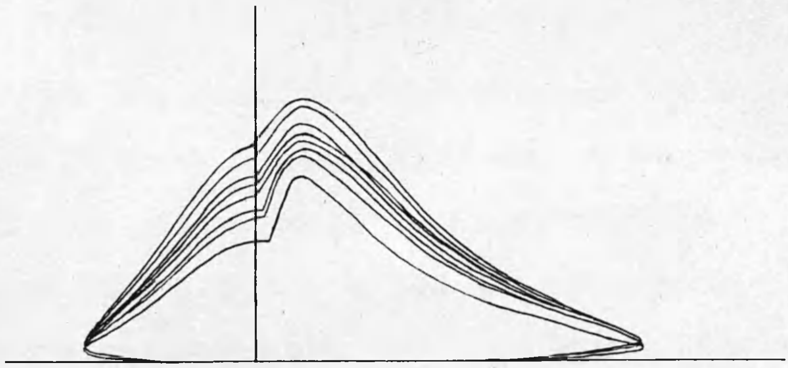
In order to obtain an approximate idea of the extent to which the ignition lag might increase from causes other than reduction of the O_2 content, another series of tests was carried out. The gasometer was not used in this case. The engine, drawing from the air boxes, was run under normal conditions as before and then suddenly throttled by partly closing the air box stop cock at the T piece junction. A few of the following cycles were recorded by means of the indicator and then the stop cock switched back to its normal position (full open). The reading during each test occupied such a short space of time that, except in the case when the suction was completely closed, no slowing of the engine could be detected.

The throttling by the stop cock clearly must have had a cooling effect on the suction air, and whether or not the temperature at the end of compression was constant throughout these tests would depend largely on whether or not the cooled air again attained atmospheric temperature (through contact with the walls of the suction pipe) before entering the combustion chamber.

On account of this uncertainty regarding temperature and the fact that both pressure and density were varied simultaneously, the results can only be in the form of a supplement/



INDICATOR DIAGRAMS FOR SUCTION-THROTTLING TESTS.



INDICATOR DIAGRAMS FOR THROTTLING EXPERIMENTS.

FIG. 64.

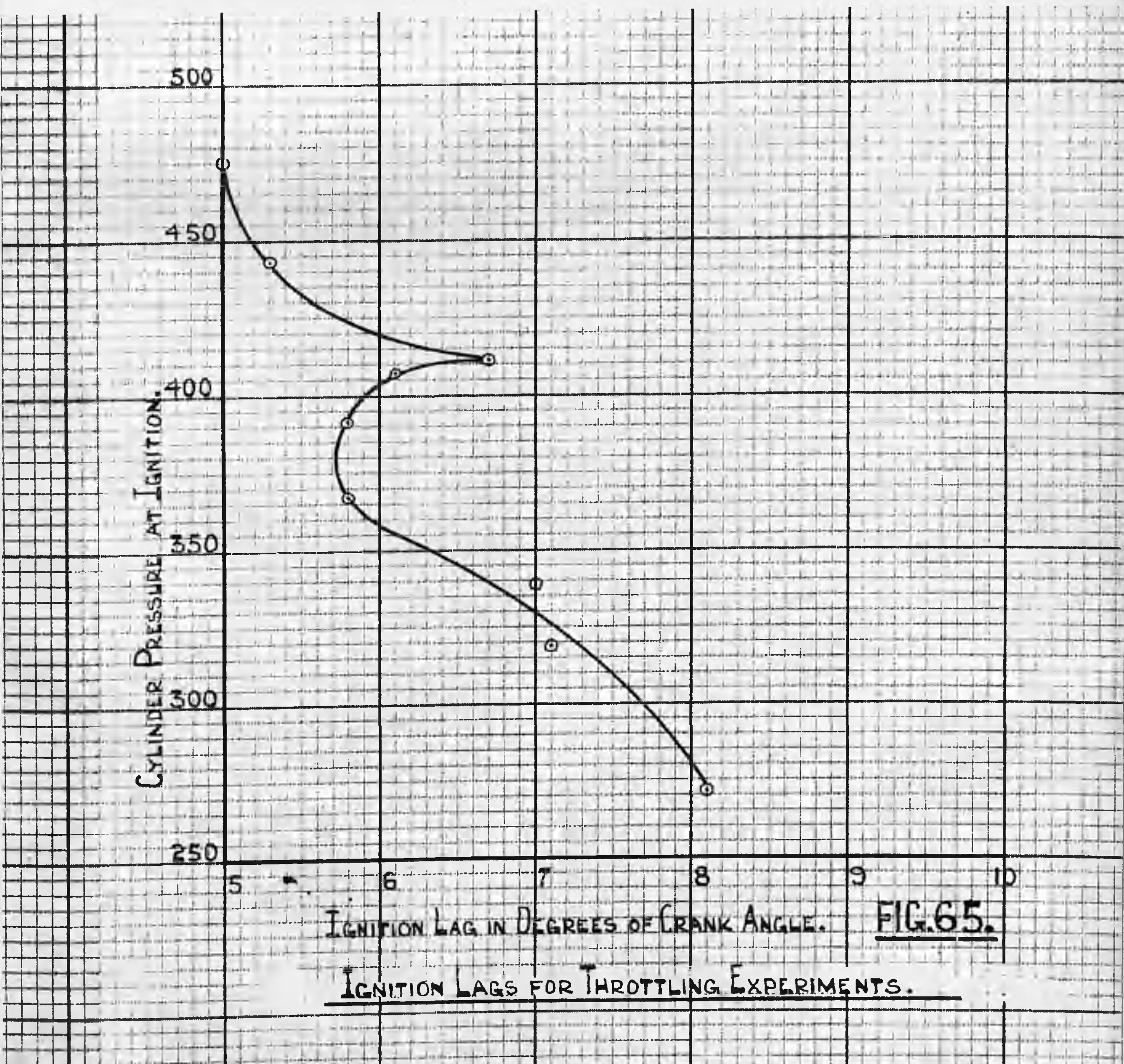


FIG. 65.

IGNITION LAGS FOR THROTTLING EXPERIMENTS.

supplement to those of the preceding tests.

The actual indicator diagrams obtained are shown in Fig.63,p.179, These diagrams were also superimposed, and in this form are shown in Fig.64,p.180. The ignition lags were obtained by the method desdribed for the previous experiments and they have been plotted in Fig.65 against the corresponding values of pressure at the end of compression.

The author desires to express his gratitude to Professor Goudie, D.Sc., and Mr. James Small, B.Sc., for their unfailing courtesy, their willing help and the invaluable suggestions afforded him during the course of his investigations. He also wishes to thank the others of the engineering staff for the assistance they so obligingly gave him in preparation of the various pieces of apparatus required.

oOo