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"FACTORS AFFECTING COMBUSTION IN A HIGH SPEED COMPRESSION IGNITION OIL ENGINE".

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FOREWORD.

This research on combustion in a high speed Diesel engine, concerning in particular air swirl, delay period and Diesel knock, was carried out in the Mechanical Engineering Department, Birmingham University, under the supervision of Professor S. Lees, M.Sc., M.A., whose help and criticism were of great value, especially when deciding the manner in which the research should be conducted.

The author would like to thank Professor S. Lees and Mr. S.J. Ellis, M.Sc., for their assistance and advice, which is greatly appreciated.

The author also wishes to thank all the Mechanical Engineering Technical Staff, for their help in making a good deal of apparatus used for the research.

C O N T E N T S.

	Page.
OBJECT OF RESEARCH.	4
PART I.	
"Air Swirl and Its Effect on Combustion".	6
Introduction.	6
Process of Combustion.	8
Description of Apparatus.	14
SECTION "A".	
Effect of Variation in the Extension and Orientation of the Inlet Valve Shrouding on Combustion.	21
Test Results.	27
Discussion of Results.	32
SECTION "B".	
Variation in the Lift of the Inlet Valve and its effect on Combustion.	43
Apparatus.	43
Test Results.	46
Discussion of Results.	52

C O N T E N T S - CONTINUED.

	Page.
PART II.	
"Delay Period and Diesel Knock".	62
Introduction.	62
Description of Additional Apparatus.	70
SECTION "A".	
Effect of Speed on Delay Period.	
Test Results.	84
I. Variable Volumetric Efficiency.	84
II. Constant Volumetric Efficiency.	87
Discussion of Results.	91
SECTION "B".	
Pilot Injection.	99
Test Results.	100
Eight-Stroking.	109
Discussion of Results.	110

C O N T E N T S - CONTINUED.

	Page.
SECTION "C".	
The Behaviour of Different Fuels and a Rating Suggestion.	115
Test Results and Discussion.	118
CONCLUSION.	122
Appendix I (Specification of Fuels).	124
Appendix II (Calculation of Volume of Air and Volumetric Efficiency).	126
Publication.	
Bibliography.	

SUMMARY.

The success of the high speed Diesel engine depends greatly upon the combustion process. Rotational movement of the air is probably the only successful means for uniform distribution of the liquid fuel inside the combustion chamber; thus successful and more or less complete combustion is achieved by bringing the air in contact with the fuel, at the right moment. Different methods have been employed to produce this rotational movement inside the combustion chamber, but up to the present there has not been published a great deal of information on the principles of its production, in so far as the poppet valve type engine is concerned. From the experiments carried out by the author, with an open type combustion chamber, one draws the conclusion that an optimum air swirl may exist with a shrouded inlet valve. The velocity of the air, through the inlet valve, was varied at first by altering the extent of shroud, and later on by varying the lift of the inlet valve. The jet of the air and its direction on entering the cylinder may

influence the fuel consumption. Under all conditions, the rotational movement of the air resulted in a much better fuel consumption than the indiscriminate movement produced when an unshrouded inlet valve was used.

The delay period as measured by the cathode ray oscillograph high speed engine indicator and a neon tube arrangement for measuring the time of injection, seems to be variable in time and decreases, in most of the cases, as the engine speed increases.

The "Diesel knock" which may be due to the rate of pressure rise, is probably caused by the accumulation of the fuel entering the combustion chamber prior to ignition. This seems to be eliminated somewhat by the introduction of a minute quantity of the fuel before the top dead centre (Pilot Injection).

The effect of injecting Parrafin (Karosene) and Petrols have been studied, the delay period and Torque was measured when timing of injection was varied over a big range. It is probable that the spontaneous ignition angle

(S.I.A.) may bear a relation to the delay Period and Diesel knock.

The S.I.A. is defined as the critical angle between the retarded position of injection, at which the fuel ceases firing or fires, and top-dead-centre.

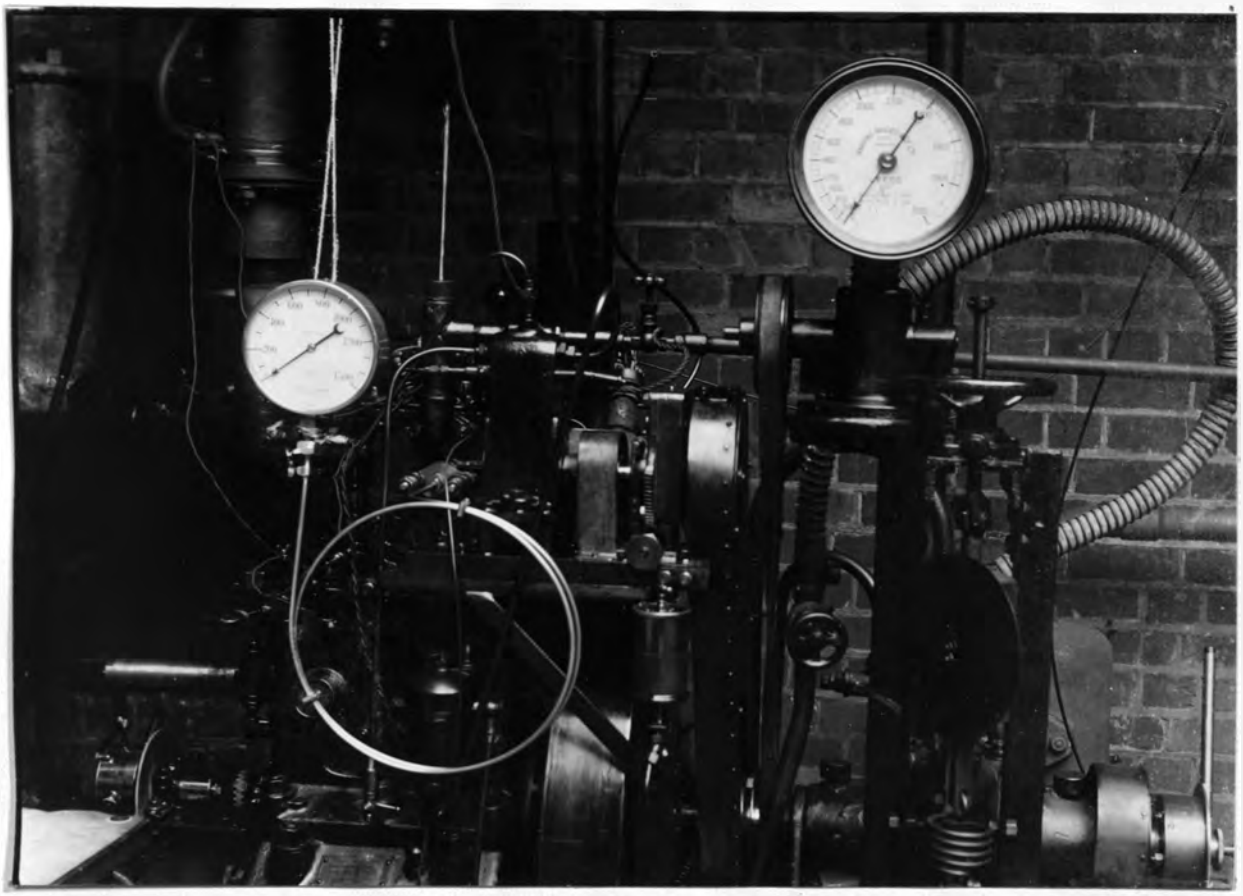
OBJECT OF RESEARCH.

It was felt that a certain amount of work should be carried out on the process of combustion in a high speed compression ignition oil engine; in particular the effect of the air inlet velocity and its direction on the fuel consumption. Later on, the research was mainly concerned with the delay period and Diesel knock.

It was arranged at first that the author should continue the work which was started by Mr. H. Faridany, M.Sc., namely the variation of the speed of the inlet air and its direction, by using a number of shrouded valves with different

openings, The effect of these on the fuel consumption, and therefore the efficiency of the engine, was studied.

Another method for varying the velocity of the inlet air had to be devised, by the author, so that the jet effect when the different valves, which were used to start with, could be very nearly eliminated. The avoidance as far as possible of this jet effect was considered important, and was achieved by using the same valve but varying the lift while the engine was running.



Tangye Engine (Top Portion).

PART I.

"AIR SWIRL AND ITS EFFECT ON COMBUSTION".

INTRODUCTION.

Satisfactory combustion in a high speed compression ignition oil engine is a matter of importance for the development of this type of engine, particularly for aviation and road use. To make full use of the many advantages possessed by this type of engine over the petrol engine, the fuel must be completely burnt in the short time available for combustion. The rate at which this fuel burns is also of great importance to ensure smoother running of the engine, and at the same time longer life.

A smoky exhaust is usually a sign of incomplete combustion of the fuel oil, and with such an exhaust carbon will be deposited on the combustion chamber, exhaust valve, atomizer holes, and round the exhaust pipe; these faults may cause the engine to shut down.

Generally speaking, smooth running is essential to this type of engine if it is ever to replace the petrol engine. The "Diesel knock" is objectionable to those who

have been accustomed to the smooth running of the six-cylinder petrol engine.

In this connection a summary of the advantages of the compression ignition oil engine over the petrol engine may be of interest.

1. Utilisation of cheaper fuels, with high flash point. Reduction in fire risks (particularly with aircraft).
2. Reduction in fuel consumption.
3. The elimination of carburetter and electric system.
4. Higher torque at low speeds, which reduces the amount of gear changing and better pulling power on hills.
5. Cooler running, and therefore less cooling water.
6. More suitable for supercharging. The possibility of this is obvious, especially with the 2-stroke type.

"PROCESS OF COMBUSTION".

In a petrol engine, a more or less homogenous mixture exists in the combustion chamber towards the end of the compression stroke, i.e. at the start of the combustion process. Magneto or coil is usually used to cause the sparking of a plug or two placed inside the combustion chamber.

In the compression ignition oil engine, air is compressed alone, and the fuel oil undergoes injection into this highly compressed air a few degress before the top dead centre is reached. The temperature of the highly compressed air causes the ignition of fuel.

To obtain satisfactory combustion within the small time available for burning the fuel completely, thorough mixing of the air and the fuel is necessary, there being two practical ways in which this can be carried out.

First. The Directed Jet Method.

In this method the fuel reaches the combustion chamber in such a manner that it will be uniformly dis-

tributed, but this of course necessitates a very elaborate design of fuel nozzle, to stand a pressure of from five to eight thousand pounds per square inch. As the liquid fuel oil can only take the straight line direction on leaving the nozzle holes, to allow it to mix thoroughly with the air, and ensure uniformity of distribution, the nozzle must possess a number of very small holes, varying in size, so that the injected fuel oil reaches different distances in the combustion chamber at the same time. Apart from the difficulty of maintenance of the nozzle holes, the rate of pressure rise, when the fuel is ignited, becomes quite uncontrollable, owing to the presence of a large quantity of fuel inside the combustion chamber, at the end of the delay period.

The question of fuel compressibility also arises, which makes the running of the engine, at variable speeds, unsatisfactory.

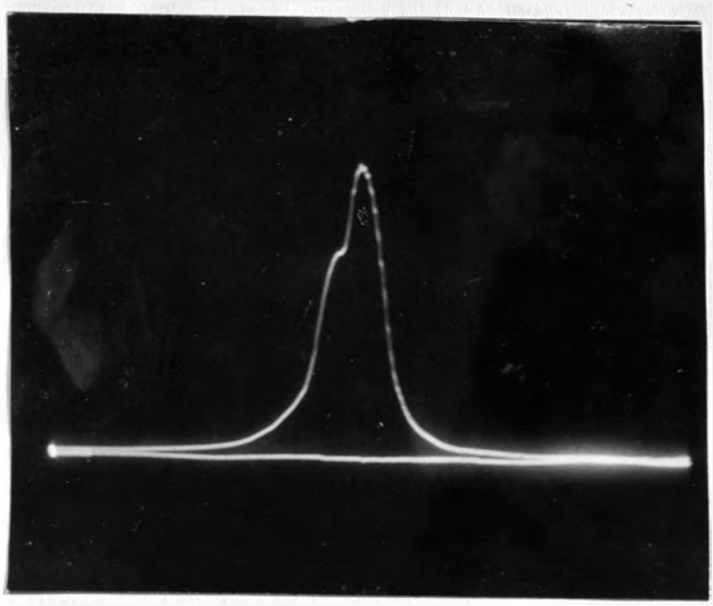


Fig.(I) Actual Out-of-Phase Indicator Diagram.

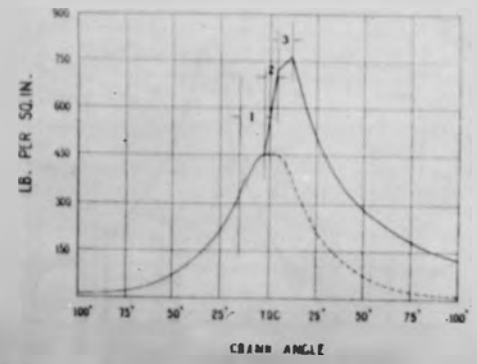


Fig. 2.—Diagram showing three phases of combustion process in compression ignition engine.

Secondly. The Directed Air Method.

In this method the air can be directed to meet the fuel in a much simpler manner. The fuel can be injected at a much lower pressure of the order of one to two thousand pounds per square inch, and for this a nozzle with one hole, or one with many holes of the same size, can be used.

The rate of pressure rise, in this method is more controllable than that of the directed jet method. This ensures a smoother running of the engine.

The air takes a rotational movement inside the combustion chamber, which causes it to be in contact with the fuel oil and supplies this with oxygen throughout the combustion period.

Mr. Ricardo, in a Paper read before the Institution of Automobile Engineering, March 1930, divided the process of combustion in a compression ignition solid injection type oil engine into three phases. In this connection Fig. (1) shows an actual out-of-phase diagram

taken at the University of Birmingham (Mechanical Engineering Department) by means of a Cathode Ray Oscillograph, designed by Dr. Dodds, and manufactured by the Metropolitan-Vickers Company Limited, of Manchester.

Fig. (2) is the theoretical out-of-phase diagram as suggested by Mr. Ricardo.

PHASE I. The Delay Period, during which time fuel is admitted, but either ignition does not take place or is restricted to some very localised nuclei. It seems probable that some minute quantity of the fuel, which has had time to mix with sufficient Oxygen, while in a vaporised condition, forms a readily combustible mixture; from these nuclei the flame spreads at first in almost the same manner as it would in a Petrol engine (presumably with a number of sparking plugs).

Ricardo regards this period as constant in time and only depending on the nature of the fuel, the temperature and pressure after compression and fineness of pulverisation.

PHASE II. The Spread of the Flame. This, like that of the petrol engine depends on turbulence. Fortunately, this flame spread is appreciably constant in terms of crank angle, but not in terms of time. The actual pressure attained, after ignition has started, depends on the quantity of the fuel present in the cylinder, the rate of injection, the temperature, and pressure of the highly compressed air.

PHASE III. At the end of the second phase, of the process of combustion, the temperature and pressure are high enough to burn the fuel as it enters the cylinder. The burning of the fuel at this phase, is under direct Mechanical control. The pressure variation depends upon the rate of injection and turbulence..

Pye, in his book entitled "The Internal Combustion Engine" (Vol. I), states that as soon as the delay period, which is constant in time, is over, the rate of combustion is directly dependent upon turbulence. Without turbulence the rise of pressure would be so slow, that the combustion would not be completed in the available time.

Ricardo's and Alcock's experiments (1) (2) on a sleeve valve engine with a specially constructed cylinder head (so that there was a rotational movement of the air, which was measured by a vane type instrument), showed that a rate of air swirl of about ten times the R.P.M. gave the most satisfactory fuel consumption.

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(1) Proceedings I.A.E. March 1930.

(2) Proceedings I.Mech.Eng. 1935.

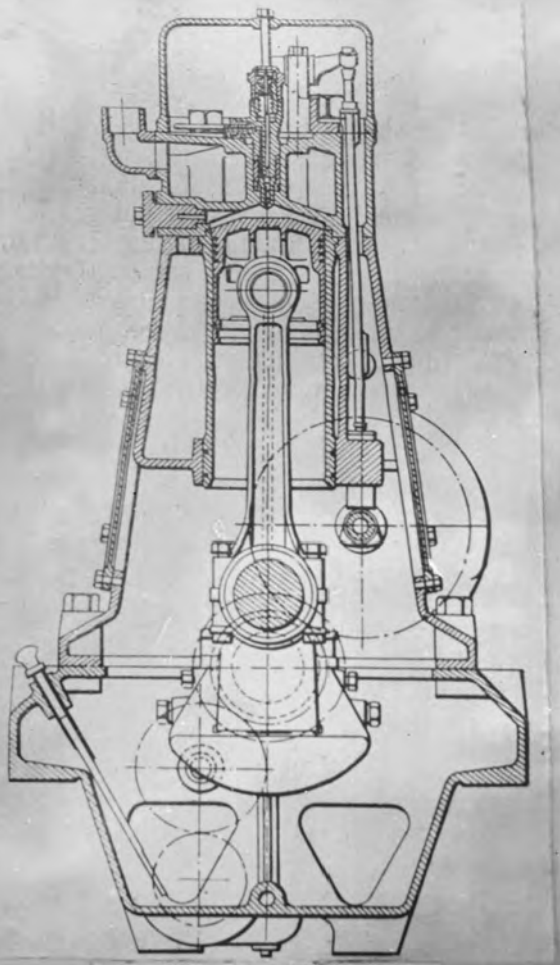
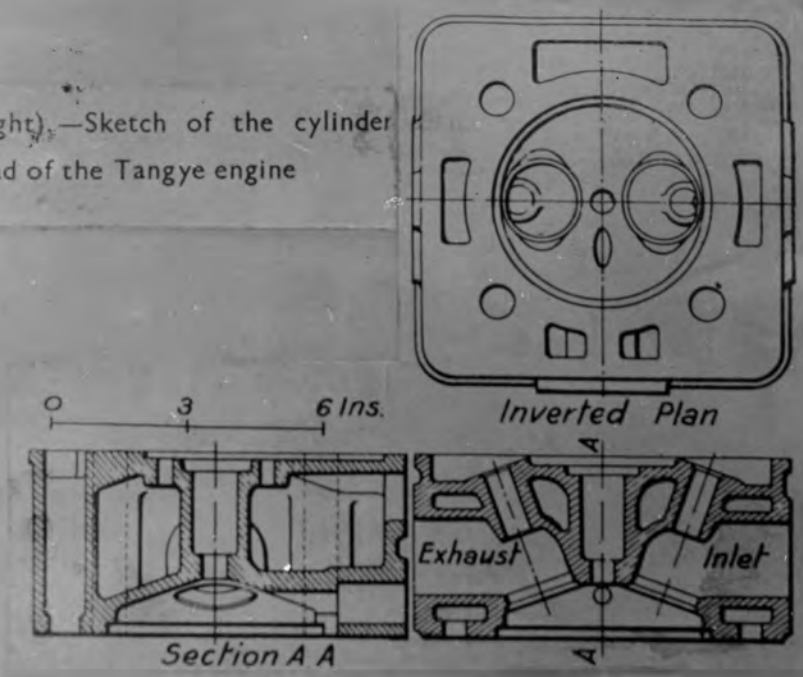


Fig. 3 (~~left~~). — Transverse section through the Tangye engine

Fig. 4 (right).—Sketch of the cylinder head of the Tangye engine



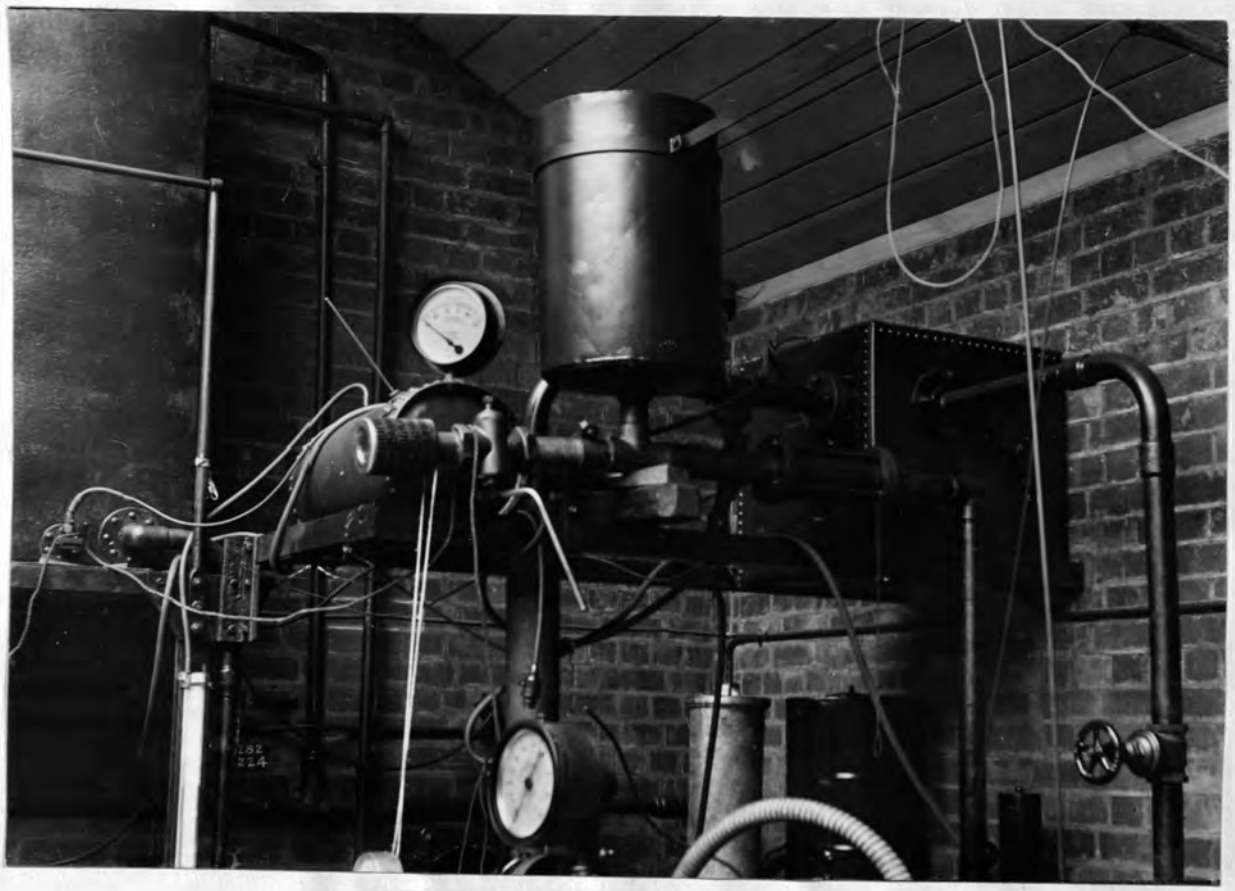
DESCRIPTION OF APPARATUS.

ENGINE DATA.

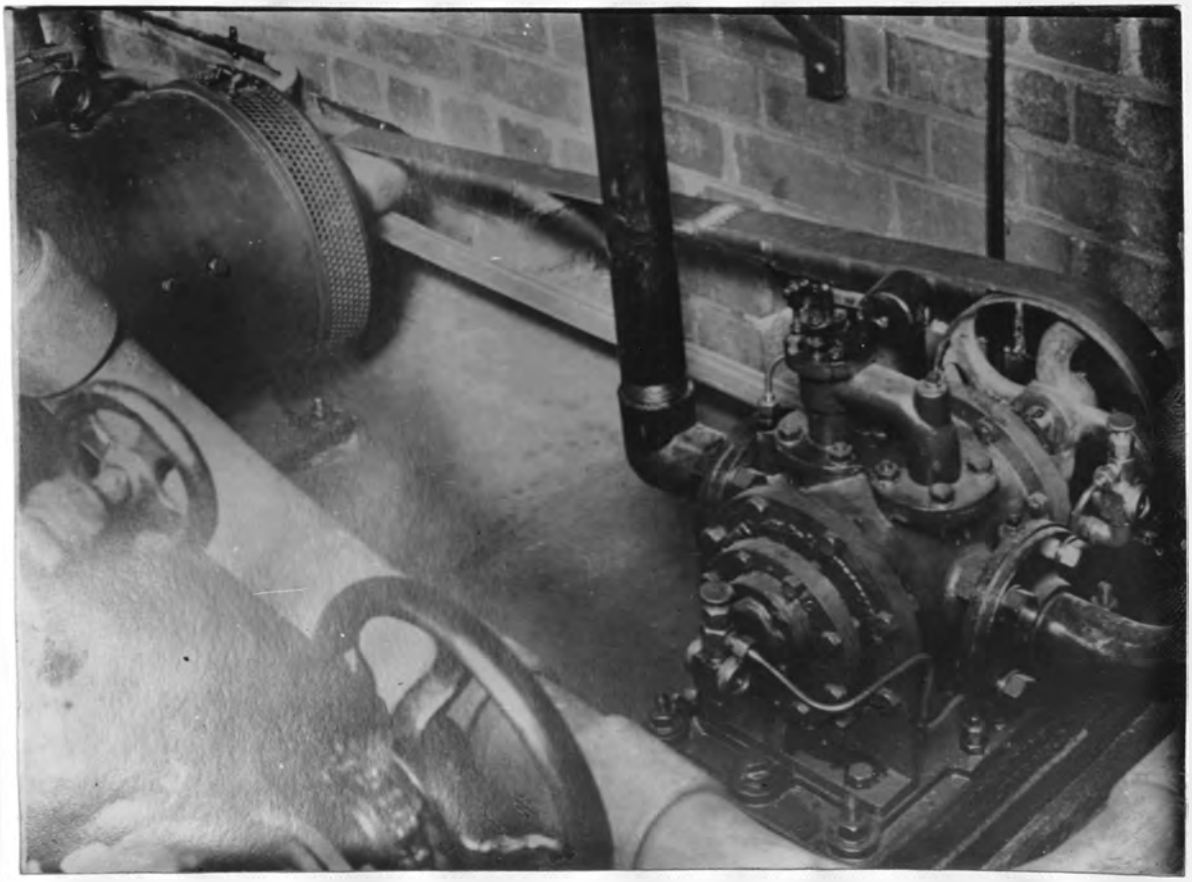
The experiments were made on a Tangye four-stroke solid injection compression ignition engine, capable of giving 10 B.H.P. at 1200 R.P.M. The cylinder diameter was $4\frac{3}{8}$ inches, and the stroke $5\frac{3}{4}$ inches. The normal running speed is 1200 R.P.M., but the engine was fitted with a special cast steel flywheel which could be run up to 2000 R.P.M. Fig. (3) is a cross-sectional elevation of the engine.

The atomizer was fitted in the centre of the cylinder head, and had four small holes. The fuel pump is of the Bosch type and is driven by the cam shaft. The combustion chamber is of the open type, the piston is slightly domed (see Fig. 4.)

The Governor was disconnected from the pump, and the engine was coupled directly to the brake.



Air Supply Pipes and Tanks.



View of Supercharger for Tangye Engine.

AIR SUPPLY.

The air supplied to the engine was passed through a suitable air vessel fitted with a nozzle on its suction side. An inclined water gauge was used for the measurement of the pressure drop across the orifice, so that the air supplied to the engine could be regulated and calculated.

Two more vessels were also fitted on the suction side, between the above tank and the engine, to act as dampers, and prevent big fluctuations in the pressure of the air on the induction side.

A supercharger was also fitted between the engine and the air vessel. The supercharger was supplied by Messrs. Bryan Donkin & Co. Ltd., of Chesterfield, and its capacity was 4,000 cubic feet per hour against 5 lbs. pressure.

A pressure gauge and a mercury manometer were also fitted for the measurement of the air pressure on the induction side.

Three valves were also fitted on the induction side. The first was a quick action one, to connect the suction side of the engine directly to the atmosphere.

The second was introduced between the engine and the air vessel, so that the air used by the engine could be regulated by throttling. The third valve was between the air vessel and the supercharger. See Fig. (5).

AIR INLET TEMPERATURE.

This was measured in two places, (1) in the air vessel and very near to the nozzle, (2) in the induction pipe and very near to the inlet valve of the engine. The first measurement was made use of when calculating the quantity of air used by the engine, and the second one was useful in controlling the temperature of the air as it enters the cylinder.

Provided the volumetric efficiency is kept constant the air inlet temperature has a big effect on the temperature at the end of compression, and also has a direct effect on the velocity of the air through the inlet valve. The temperature of the air was kept constant (20 ^{o}C. spprox.) except when otherwise stated, so that the results of tests carried out could be compared, especially when air swirl was considered, either when using a number of shrouded inlet

DIAGRAMMATIC SKETCH OF AIR INTAKE.

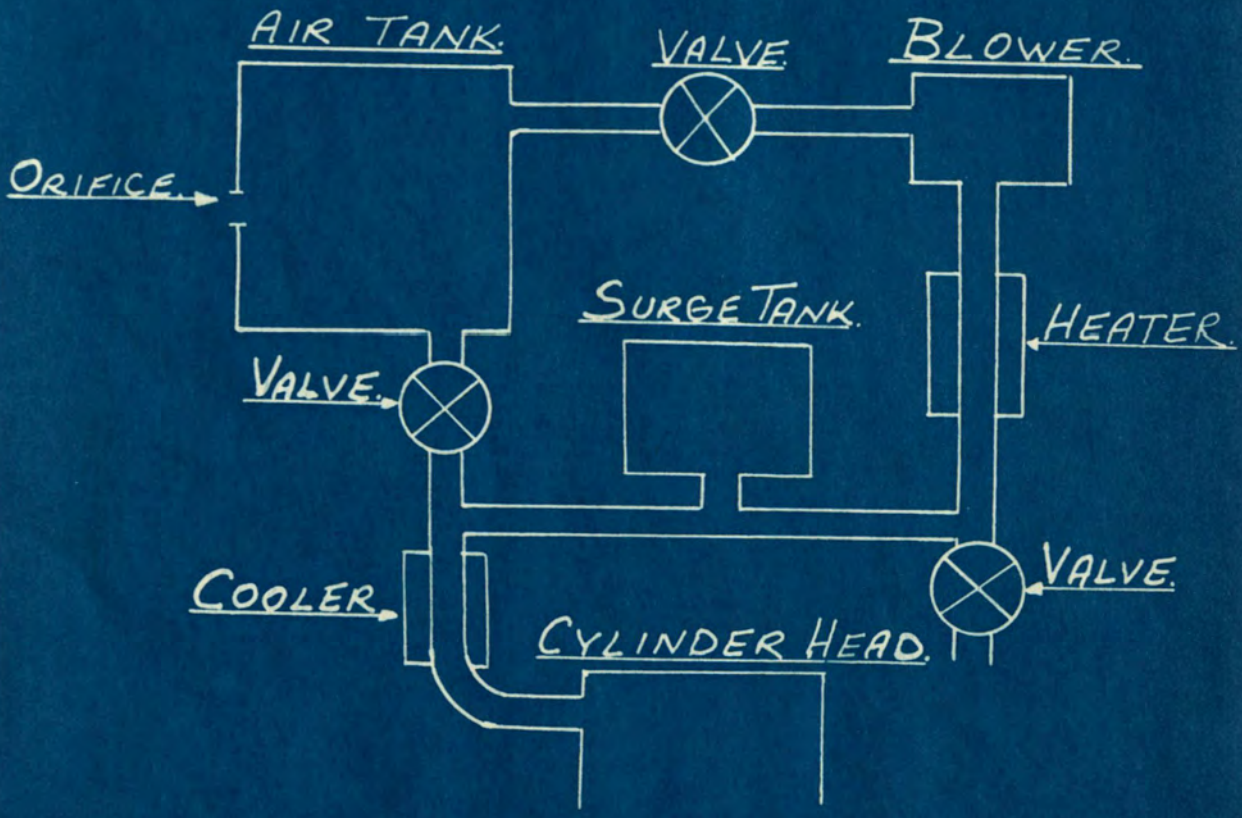


FIG. 5.

valves with different openings, or when varying the lift of the inlet valve. The temperature of the inlet air was regulated by two means.

First, An Electric Heater. This was installed in the induction side for heating the air before entering the cylinder. It consisted of a resistance wire (Glowray) coiled inside a quartz tube, round the circumference. A conical baffle was also fitted at the entrance of the tube, so that the air could pass very near the resistance wire. The quartz tube was also insulated by a thick layer of asbestos from an outer steel tube. This heater was fastened between two flanges by means of four bolts, so that the whole device could be easily removed and inspected.

Second. An Air Cooler. When the supercharger was made use of, the air (due to the compression and the rotating blades) warmed up, and a temperature of the order of 56 ^{o}C. was recorded. An air cooler was installed in the induction system. This consisted of a steel tube of 4 inch inside diameter. A great number of copper tubes were fitted longitudinally inside the steel pipe. The cooling water circulated outside the copper tubes, whilst

the air passed through the inside of the tubes.

On the exhaust side, the temperature was noted by means of a Platinum wire thermometer.

Brake Horse Power.

This was measured by means of a Froude Hydraulic Dynamometer, directly coupled to the engine.

Speed.

The measurement of this was made by means of two instruments. The first was a belt driven tachometer used for the purpose of running the engine at a given approximate speed. The second was a clutch operated, chain-driven, revolution counter, for the accurate measurement of the engine speed.

When these experiments were carried out, the speed of the engine was regulated at 1200 R.P.M. approximately, so that comparison of the test results could be made, without the introduction of many errors due to delay period, friction-losses, and injection difficulties.

Cooling Water.

The cooling water was supplied to the engine, from a fairly large tank situated a few feet above the cylinder

head level. A constant head of water was maintained throughout the experiments. A mercury thermometer was kept in contact with the water inside the tank for recording the water inlet temperature. Another mercury thermometer was inserted in the cooling water outlet pipe, very near the cylinder head, for the purpose of measuring the cooling water outlet temperature. As this outlet temperature has an effect on the running conditions of the engine, it was felt desirable to keep it constant throughout the tests at approximately 60 ^{o}C.

Fuel Oil and its Measurement.

The fuel oil used was of the Diesel type supplied by Messrs. Shell Mex and B.P. Ltd., and has been the same throughout the tests. For analysis and specification, see Appendix.

The tank containing the Diesel Oil was of a fairly large size and a calibrated glass bulb was installed in the fuel supply pipes for the fairly accurate measurement of the fuel oil used by the engine.

Injection Timing.

To eliminate the "Diesel Knock" at high loads, the injection timing was retarded throughout the tests on air swirl; while in the second part of this paper, as will be discussed later, the timing of the injection was varied.

After this preliminary description of the apparatus used, it is convenient to describe the work (on air swirl) to be discussed in this part (Part I) of the thesis, under two headings, labelled respectively Sections A and B.

S E C T I O N "A".

EFFECT OF VARIATION IN THE EXTENSION AND ORIENTATION OF THE INLET VALVE SHROUDING ON COMBUSTION.

It will be of some interest at this stage to outline a brief statement of the research work carried out by Mr. H. Faridany, M.Sc., at Birmingham University. In his thesis entitled "Studies of Combustion in a High Speed Diesel Engine", it is mentioned that the obvious method of varying the inlet velocity of the air is to alter the extent of shrouding, and this work was done by using a number of different inlet valves. To start with he neglected the effect of variation of the volumetric efficiency, and discovered that the indicator plug hole had upset the symmetry of the combustion chamber, and on fitting a smoothing plug symmetry was established. In the latter part of his work a super-charger was installed for the purpose of keeping the volumetric efficiency constant, but time did not permit of much work in this connection.

INLET VALVES USED.

The inlet valve supplied by the makers had 120° of shrouding round its circumference, leaving 240° of arc of valve opening. This valve will be referred to as valve No. I.

Three more inlet valves were used.

Valve No. II 240° of shrouding, 120° of opening.
" " III 280° " " 80° " "
" " IV 300° " " 60° " "

An unshrouded valve 0° of shrouding, 360° of opening, was also used.

Method Used for Varying the Direction of the Inlet Air.

Arrangements were made for fixing the inlet valve in eight different positions, at 45° intervals. Special caps were used in this connection. The top of each cap was slotted for the rocker arm, and a 7/64" pin was fitted at the bottom for locating the slot in the end of the inlet valve stem. The angle between the pin and the slot was different in each cap, so that the eight positions of the inlet valve could be obtained, by using four caps. The eight different positions are shown in Fig. (6).

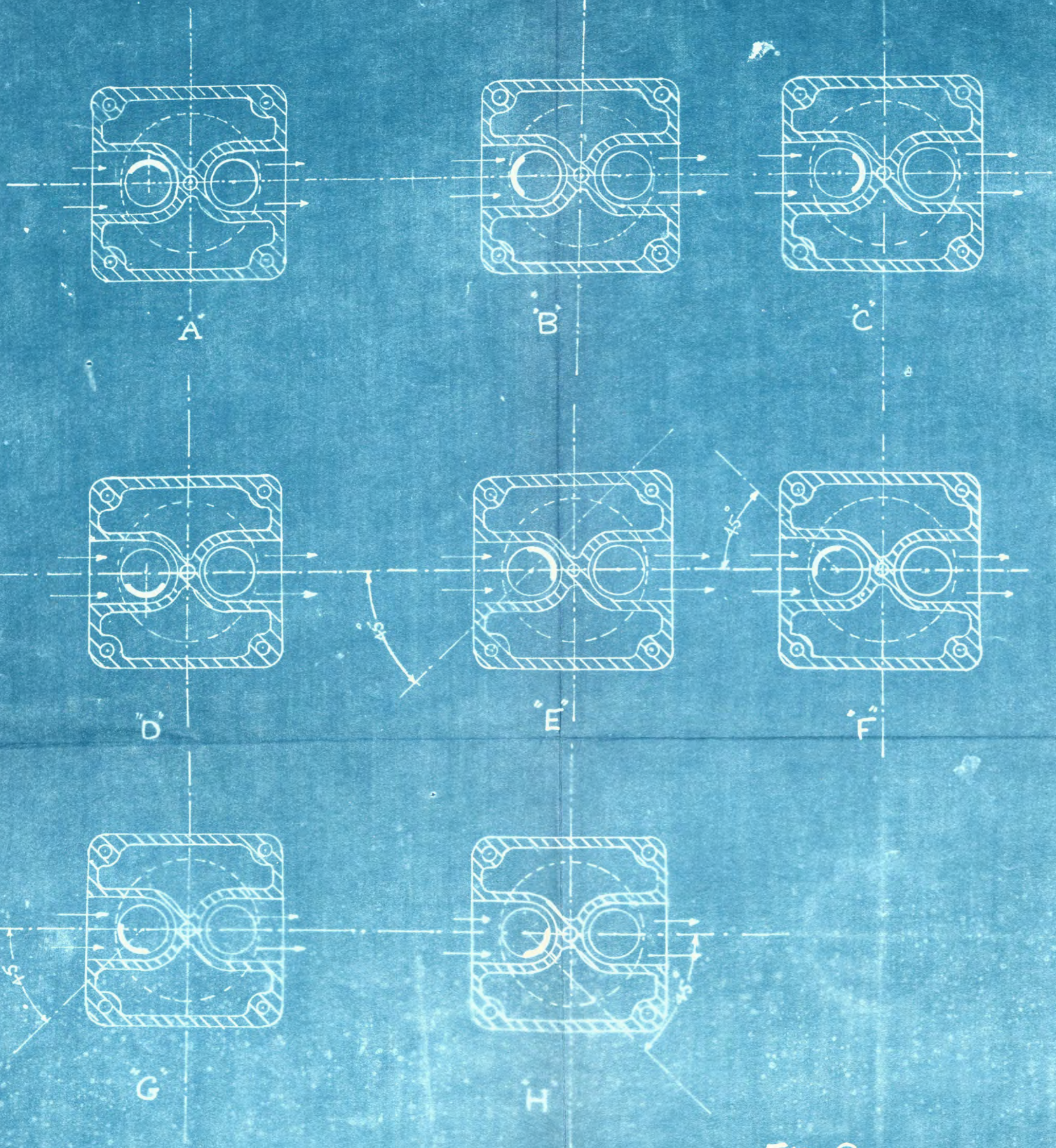


FIG. 6.

TANGYE EXPERIMENTAL ENGINE

POSITIONS OF SHROUDING ON INLET VALVES

Assuming the coefficient of discharge of all the inlet valves is constant, and that the volumetric efficiency, temperature of the inlet air as it passes through the valve, and speed of the engine was kept constant, it follows that the velocity of the inlet air through the inlet valve will be proportional to the opening of the valve.

Question of Volumetric Efficiency.

To compare the test results for a number of inlet valves with different angles of arc of shrouding, so far as the inlet velocity of air is concerned, the Volumetric efficiency should preferably be kept constant. A test was carried out to study the effect of variation of the volumetric efficiency on the fuel consumption of the engine. By throttling down on the suction side of the engine, the volumetric efficiency was varied, while valve No. I (120^{o} of shroud, 240^{o} opening) in Position "A" was used. The fuel consumption, quantity of air, B.H.P., water outlet temperature, speed (approx. 1200 R.P.M.), air inlet temperature (almost constant), and exhaust temperature, were recorded.

Fig. (7) shews the relation between fuel consumption and volume of air used by the engine, in cubic feet per minute; it also shews the exhaust temperature. As can be seen from the curves in these diagrams the fuel consumption per B.H.P. hour varied between 0.472 lbs. to 0.544 lbs., when the air drawn into the cylinder varied from 30 cubic feet to 15 cubic feet per minute. The exhaust temperatures corresponding to these valves were 292° C and 507° C, respectively.

It was thought, therefore, that some means of fixing the volumetric efficiency must be devised. The blower which was installed heated up the air considerably.

To study the effect of the air inlet temperature on fuel consumption, the electric heater was made use of. The air inlet temperature was varied and recorded; the load, speed of the engine, and the volumetric efficiency, were kept constant. The fuel consumption and the exhaust temperature were also recorded. With Valve No. IV (300° shrouding, 60° opening) as the inlet valve in Position "A", Fig. (8) shews the results obtained.

VALVE NO 1
POSITION "A"

TEMPERATURE

FUEL

R.P.M. = 1100 APPROX.
B.H.P. = 5.2 " "
TIME FULLY RETARDED
WATER OUTLET TEMP
= 60° C

FIG. 7.

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FUEL IN GMS. PER B.H.P. HOUR

VOLUME OF AIR IN CU. FT. PER MIN. (600 R.P.M.)

TEMPERATURE (EXHAUST)

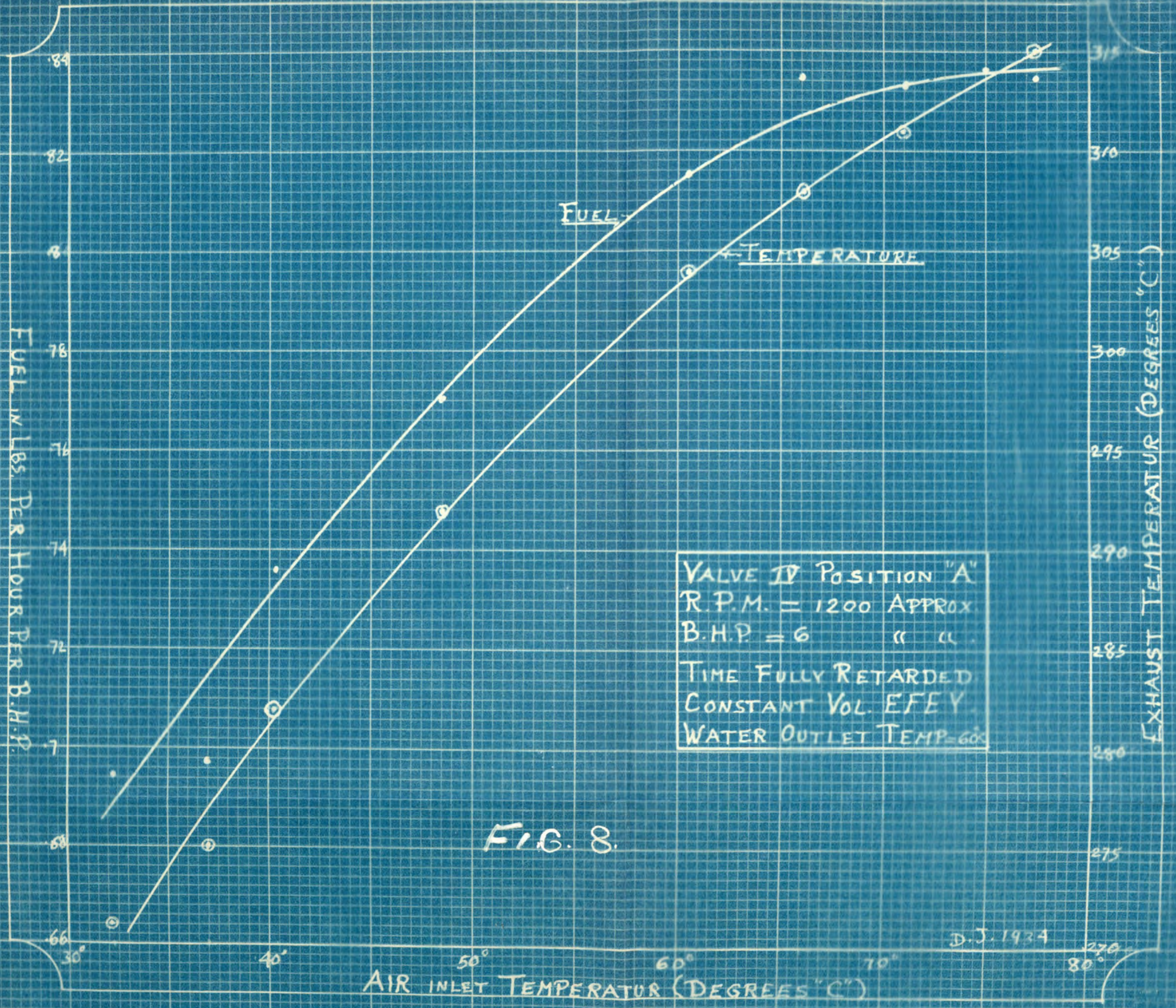

FUEL
TEMPERATURE
VALVE IV POSITION "A"
R.P.M. = 1200 APPROX
B.H.P = 6 " "
TIME FULLY RETARDED
CONSTANT VOL. EFEY
WATER OUTLET TEMP=60°
FUEL IN LBS. PER HOUR PER B.H.P.
EXHAUST TEMPERATUR (DEGREES "C")
AIR INLET TEMPERATUR (DEGREES "C")
84
82
8
78
76
74
72
7
68
66
315
310
305
300
295
290
285
280
275
270
30°
40°
50°
60°
70°
80°
D.J. 1934

FIG. 8.

As can be seen from the curves, the fuel consumption at 40°C. is 0.73 lbs. per B.H.P. per hour, as compared with 0.83 lbs. per B.H.P. per hour at 70°C. A slight increase in the exhaust temperature was also observed. It was anticipated that as the inlet temperature increased, fuel consumption would decrease, as the fuel could burn more readily and combustion would be more completed in the available time. The reason suggested for the higher fuel consumption with this valve (60° opening only) in position "A", is that the velocity of the air is much too high, even without heating. With heating, owing to the increased volume of the air taken in, a still higher velocity is attained, which seems to make combustion unsatisfactory.

Prior to the installation of an air cooler, it was decided not to make use of the supercharger, owing to the temperature difficulty. The principle was used of throttling down on the induction side, when valves with big openings (small shroudings) were employed, in order that the volumetric efficiency might be kept constant when a number of inlet valves with different openings were to be tested. In this way, the results could be compared,

(although with a low volumetric efficiency). In this method the author has eliminated the effect of variation of the inlet temperature on the air velocity, through the shrouded inlet valves.

Although, as stated, the volumetric efficiency was low, it was kept constant throughout the experiments, and it was felt that this method was fairly accurate for studying the effect of turbulence on the fuel consumption, when using different shrouded valves. The smoothing plug was used with these experiments. The author suggests that this method may be of some interest for aircraft compression ignition oil engines with a high compression ratio. Throttling down, while at low altitudes may be used. At high altitudes, normal run, full throttle may be made use of. It is realised, of course, that in an aircraft engine, at different altitudes, the exhaust pressure varies.

TEST RESULTS.

VALVE I. (120° SHROUD, 240° OPENING).

This valve was tested for five different positions, namely A, E, F, B and C, see Fig. (6). The fuel consumption and B.H.P. were recorded, while speed, volumetric efficiency, and air inlet temperature, were kept constant. Fig. (9) shows the relationship between B.H.P. and fuel consumption per hour. The position of the valve is marked on each curve.

At 7 B.H.P. position E gives a fuel consumption of 3.5 lbs. per hour, while for positions B and C 6.2 lbs. per hour of fuel consumption was recorded for the same B.H.P. The fuel consumption for position F was 3.85 lbs. per hour at 7 B.H.P. It may be of some interest to note that the unshrouded valve gave a fuel consumption of the order of 5.85 lbs. per hour at 7 B.H.P.

These results shew very clearly that for the same valve the fuel consumption is almost doubled for positions B and C, as compared with position E. Thus by letting the air enter the cylinder in a direction so as to ensure a circular motion, and hence come in contact with

UNSHROUDED VALVE

C

B

A

F

A

E

B

C

A

R.P.M. = 1200 APPROX.
COOLING WATER
OUTLET TEMP = 60C
TIMING FULLY RETARDED.
VOL. EFF. IS CONSTANT.

TEST RESULTS.
VALVE No. 1

FUEL IN LBS PER HOUR

B.H.P.

D.J. 1915

FIG. 9.

the fuel almost at right angles, the combustion is greatly improved; while positions B and C, which in effect do not allow of the air having such motion, raised the fuel consumption considerably for constant values of the volumetric efficiency; inlet temperature of the air, load, outlet temperature of the cooling water, and speed of the engine.

VALVE II (240° Shrouding, 120° OPENING).

The curves showing the relationship between fuel consumption and B.H.P. for different positions of the inlet valve shrouding are plotted in Fig. (10). At 7 B.H.P. the fuel consumptions for the different positions of the shrouding are as follows:-

Position.	Fuel in lbs. per hour.
F	3.53 lbs. per hour.
E	3.82 " " "
A	4.03 " " "
B	6.1 " " "
C	7.2 " " "

These results still indicate a high variation in fuel consumption per hour, when the direction of the air is varied through the shrouded inlet valve.

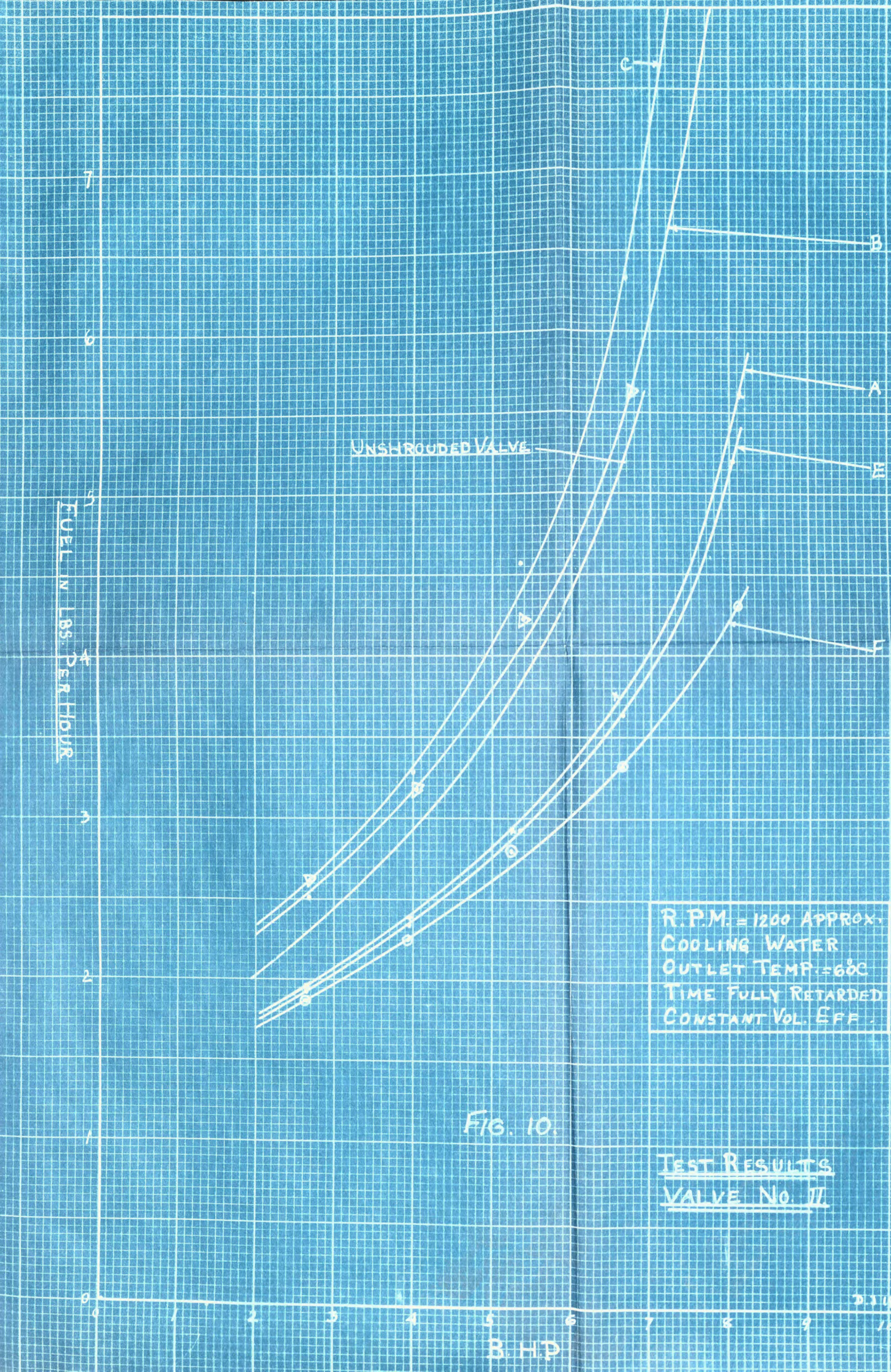

C
B
A
UNSHROUDED VALVE
E
F
D
7
6
5
4
3
2
1
0
FUEL IN LBS. PER HOUR
0
1
2
3
4
5
6
7
8
9
10
B.H.P
R.P.M. = 1200 APPROX.
COOLING WATER
OUTLET TEMP. = 60C.
TIME FULLY RETARDED
CONSTANT VOL. EFF.
TEST RESULTS.
VALVE No. II.

FIG. 10.

VALVE NO. III (280° SHROUD, 80° OPENING).

The five curves in Fig. (11) are test results for the fuel consumption, as against B.H.P. All other conditions are being kept very nearly constant.

The fuel consumptions for the five different positions of the shrouded inlet valve at 7 B.H.P. are as follows:-

Position.	Fuel in Lbs. per hour.
E	3.75 lbs. per hour.
F	3.9 " " "
A	4.45 " " "
B	5.45 " " "
C	6.6 " " "

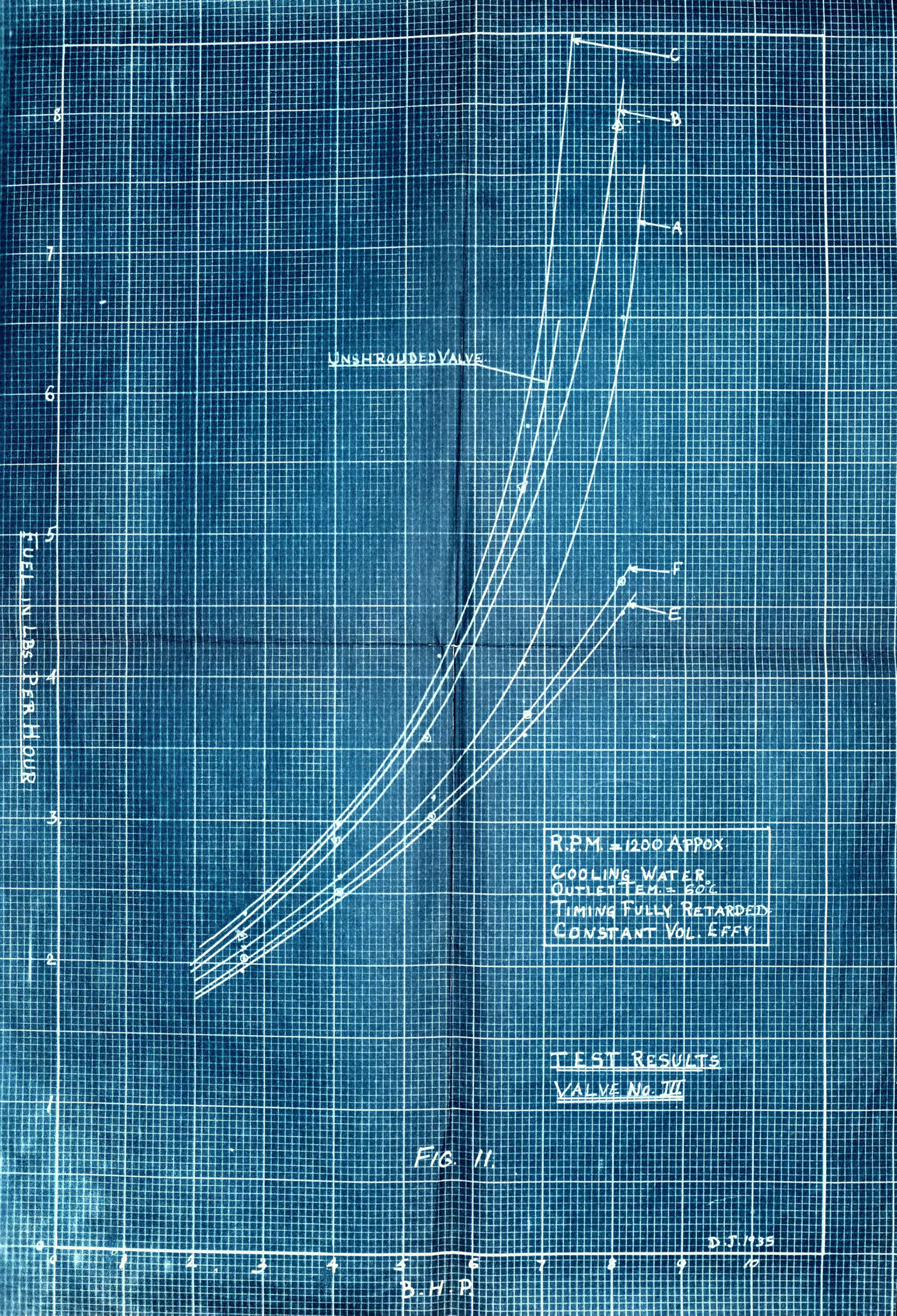

C
B
A
UNSHROUDED VALVE
F
E
FUEL IN LBS. PER HOUR
B.H.P.
R.P.M. = 1200 APPOX.
COOLING WATER
OUTLET TEM. = 60°C.
TIMING FULLY RETARDED
CONSTANT VOL. EFFY
TEST RESULTS
VALVE NO. III
D.J. 1935

FIG. 11.

VALVE IV (30° SHROUD, 60° OPENING).

Fig. (12) contains test results which shew the fuel consumption for this valve, as against B.H.P., for the positions marked on the curves.

At 7 B.H.P. The fuel consumptions of the different positions given below:-

Position.	Consumption in lbs. per hour.
E	3.65 lbs. per hour.
F	4.05 " " "
A	4.5 " " "

The results for the four shrouded valves shew a remarkable vairation in fuel consumption, when comparisons of the different valve orientations are made.

FIG. 12.

Test Results
Val. Ve. No. 12

R.P.M. = 1200 Approx
Cooling Water
Outlet Temp. = 60°
Timed Live Rate Taxed
Const Vol. Eff.

Fuel Lbs.

Un Shrouded Val Ves

B.H.P.

DISCUSSION OF RESULTS OF SECTION A.

THE IMPORTANCE OF THE DIRECTION OF THE INLET AIR ON THE PROCESS OF COMBUSTION.

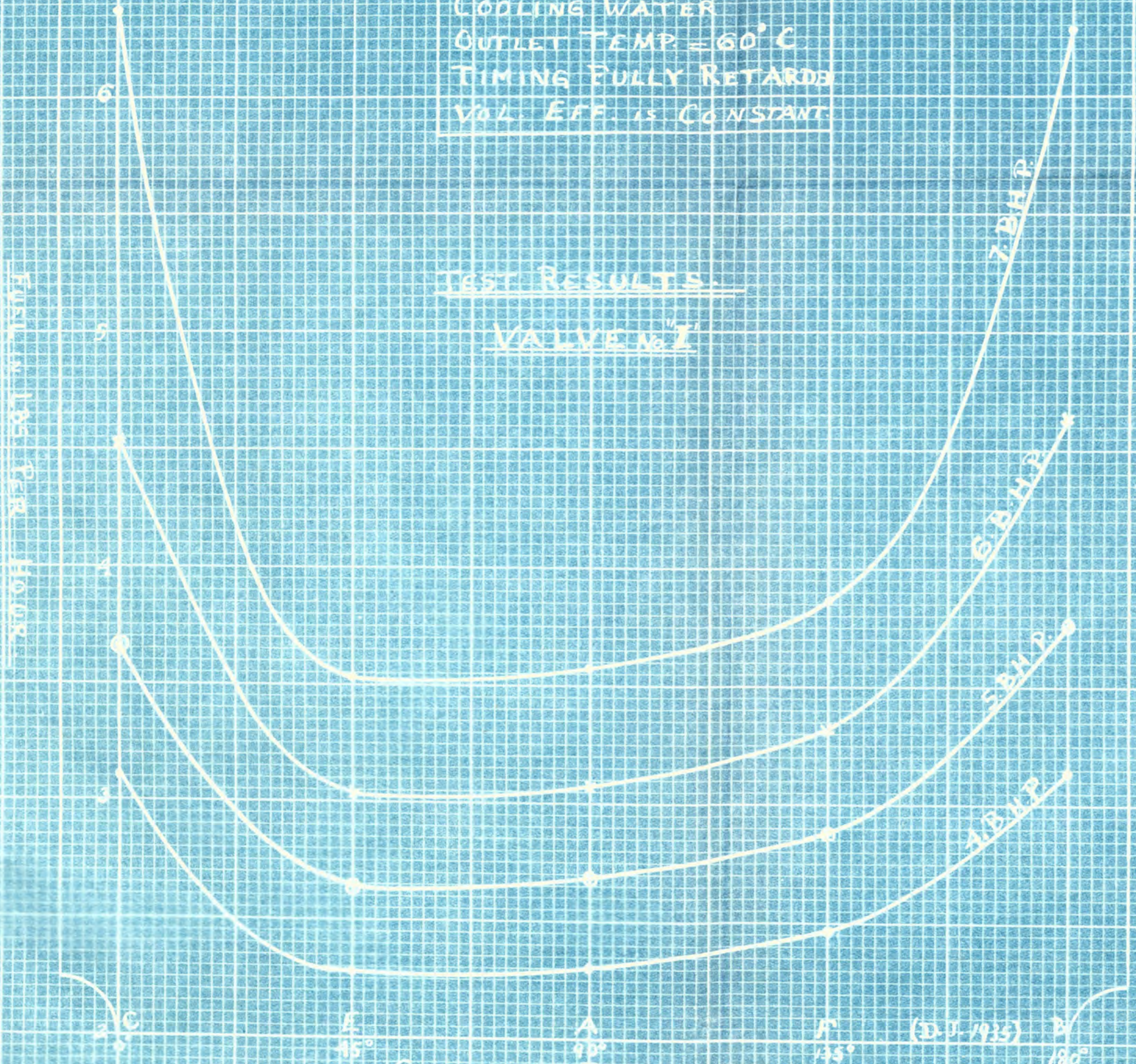
In Fig. (6), as mentioned in the earlier part of this thesis, the inlet valve could be placed in eight different positions. Owing to the existence of symmetry, these could only be treated as five different positions (so far as the effects of the direction of the inlet air are concerned), namely Positions A, B, E, F and C. Thus by placing the inlet valve in any of these positions, the air could be given five different directions on entering the cylinder. For the study of the effect of the direction of the inlet air on the fuel consumption, Figs. (9), (10) and (11), have been replotted on a different basis shewing the relationshop between the fuel consumption per hour, as against position of shrouding in degrees, for a few different values of the B.H.P.

VALVE I (120^{o} Shroud, 240^{o} Opening). Fig. (9), has been replotted as shewn in Fig. (13). The curves shew the relation between fuel consumption against shrouding

R.P.M. = 1200 APPROX.
COOLING WATER
OUTLET TEMP. = 60° C.
TIMING FULLY RETARDED
VOL. EFF. IS CONSTANT.

TEST RESULTS

VALVE No. "I"



SHROUDING POSITION

FIG. 13

position (in degrees) for 7, 6, 5 and 4 B.H.P.

It may be of interest to note that all the four curves start at very high fuel consumptions (Position C) and slope down rapidly until position E is reached (45^{o}), then the fuel consumption is nearly steady until Position F is reached (135^{o}), when all the four curves slope up rapidly shewing a marked increase in fuel consumption. In the case of this valve Position E (45^{o}) gives the best results at all loads.

VALVE NO. II (240^{o} Shroud, 120^{o} Opening).

Curves in Fig. (10) were replotted in the same way as those of Valve No. I. Fig. (14) shews relation between fuel consumption in lbs. per hour against shrouding position, for 7, 6, 5, and 4 B.H.P. All the four curves shew a very high fuel consumption at position C (0^{o}), and then slope down rapidly towards position E (45^{o}). There is then a very slight increase to position A (90^{o}) followed by a fall in fuel consumption at position F (135^{o}), which in the case of this valve gave the best fuel consumption. The curves slope upwards again, and there is a marked increase

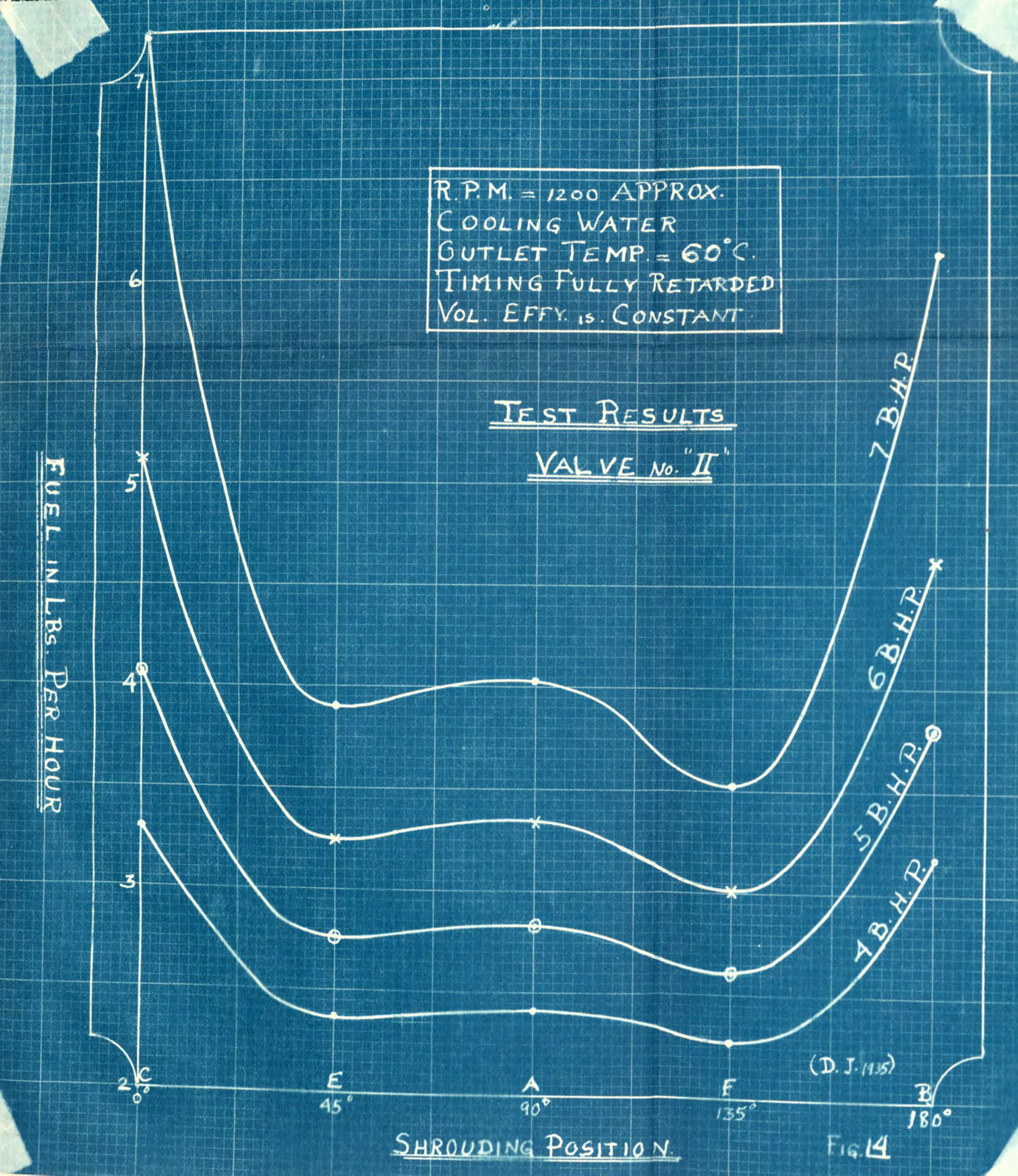

R.P.M. = 1200 APPROX.
COOLING WATER
OUTLET TEMP. = 60°C.
TIMING FULLY RETARDED
VOL. EFFY. IS CONSTANT
TEST RESULTS
VALVE No. "II"
7 B.H.P.
6 B.H.P.
5 B.H.P.
4 B.H.P.
FUEL IN LBS. PER HOUR
7
6
5
4
3
2
C
E
A
F
B
45°
90°
135°
180°
SHROUDING POSITION.
(D. J. 1935)

FIG. 14

in fuel consumption at position B (180^{o}).

VALVE NO. III (280^{o} Shroud, 80^{o} Opening). In the same way Fig. (11) was replotted as in Fig. (15). In the case of this valve the shapes of the curves are similar to those of the previous valve.

Positions E (45^{o}) and F (135^{o}) gave almost the same fuel consumptions, and Position A gave a higher fuel consumption than that in Fig.14.

The results for these three valves also shew that when valves with big shroudings, i.e. small openings were used, or in other words when the air was let into the cylinder at high velocities, position A, which imparts almost circular motion to the air, was no longer the suitable position for a satisfactory combustion. This will be explained later.

R.P.M. = 1200 APPROX.
COOLING WATER
OUTLET TEMP. = 60°C.
TIMING FULLY RETARDED
VOL. EFFY. IS CONSTANT

TEST RESULTS

VALVE NO. "III"

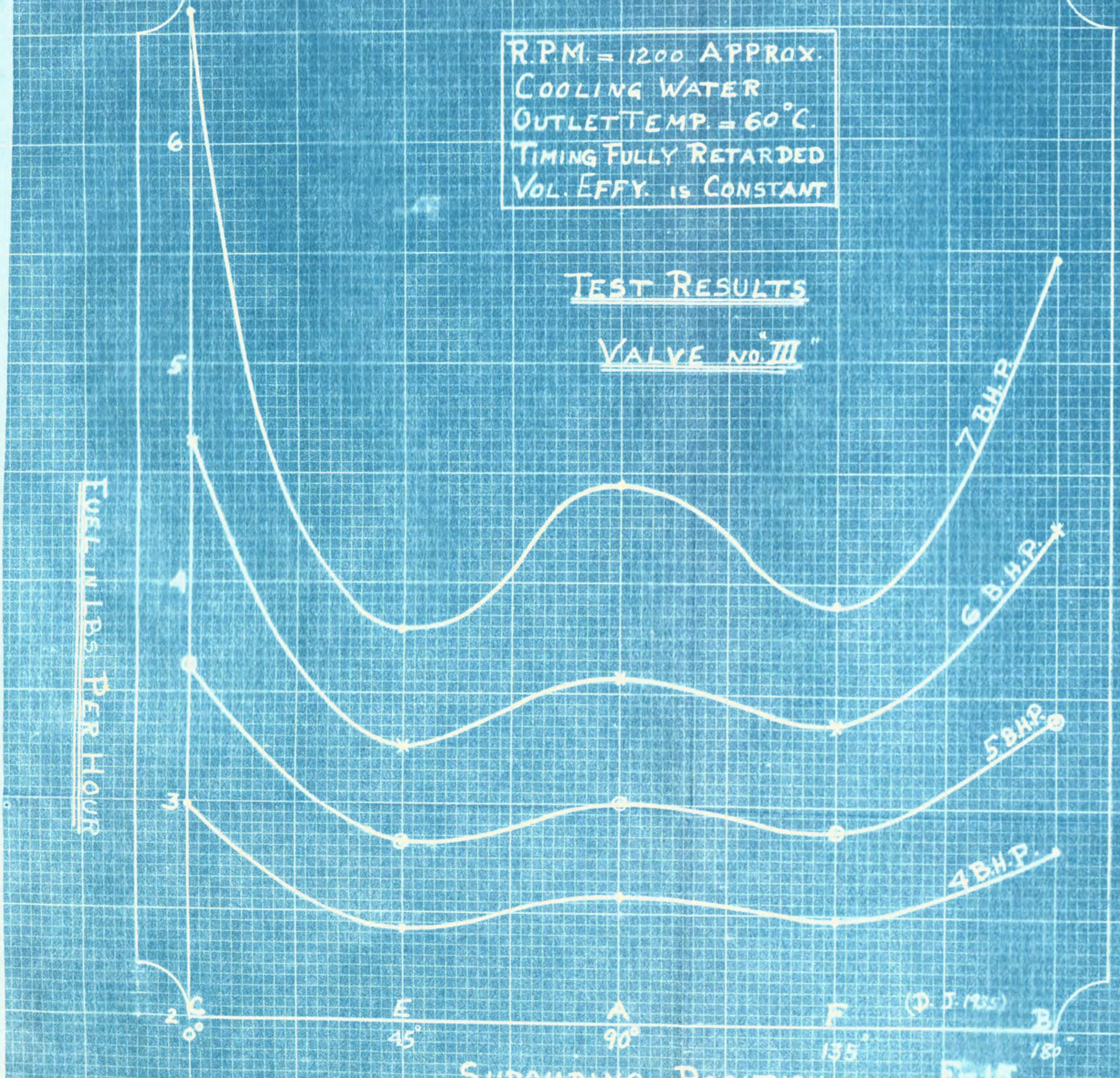


Fig 15

DISCUSSION OF RESULTS OF SECTION A - CONTINUED.

EFFECT OF AIR VELOCITY ON COMBUSTION.

Under this heading the author wishes to discuss the importance of turbulence and how it effects the process of combustion in general. It has been suggested, as mentioned earlier in this paper, when experiments by Messrs. Ricardo & Alcock were carried on a sleeve valve engine, that there seems to exist a definite air swirl which gives the minimum fuel consumption.

It has also been pointed out, that in the second and third ignition oil engine (solid injection), turbulence is very essential to attain a satisfactory combustion. The reason for the necessity of turbulence advanced by the present author is one which extends the views given by Mr. Ricardo. As soon as the fuel rushes into the cylinder, in a pulverized condition, the minute droplets of the fuel find themselves in an atmosphere of air rich

in oxygen.[1] When the delay period is over and a number of these droplets have already been ignited, the flame then starts spreading. At the same time, the pressure commences to rise, which marks the beginning of the second phase.

The rate of pressure rise depends of course on the rate of fuel combustion, which in turn depends upon the quantity of fuel present inside the combustion chamber, and the rate at which this pulverized fuel comes into contact with the oxygen of the highly compressed air (which is at high temperature).

At the start, the few droplets of the fuel, inside the combustion chamber, find no difficulty in burning, when plenty of oxygen surrounds them. As soon as a minute fraction of a second passes, these droplets, together with the fresh pulverised fuel entering the cylinder, will be surrounded by zones where oxygen is rare, and unless there

---

(1) The chemical changes during combustion in a Diesel Engine have received careful attention at the hands of Professor Egerton, F.R.S., [illegible] Smith, D.Phil., and A.N. Ubblohde, M.A., in a Philosophical Transaction of the Royal Society of London.

is relative motion between the air and fuel particles inside the combustion chamber (in order to supply the fuel with the required quantity of Oxygen), the combustion process will be too slow to be completed in the short time available. This suggests, of course, that the higher the velocity of approach between the air and the fuel, the quicker the combustion takes place, and there that combustion will be hastened, and fuel consumption will be improved. The above statement, however, has been proved by experiments to be incorrect, as it will be shewn later that optimum swirl may exist in this (open Tangye) type of combustion chamber.

It has already been mentioned that tests carried out by Messrs. Ricardo and Alcock on a sleeve valve engine with directed air swirl, gave a value for optimum swirl of about 10 times the crank R.P.M.

The results obtained in Figs. (9), (10), (11) and (12) were plotted again for each position, and the curves obtained shew the relation between the fuel consumption and opening in valve shrouding for a few different B.H.Ps.

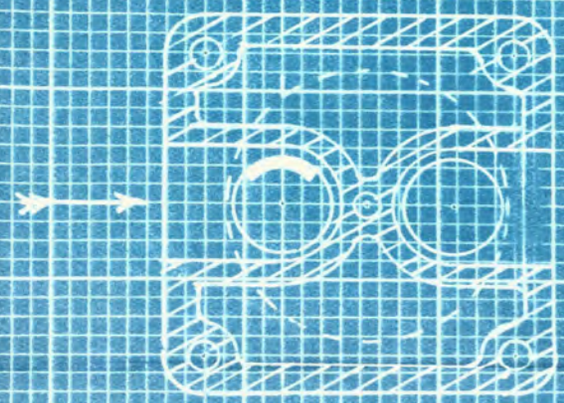
POSITION "A". Fig. (16) shews clearly, for this position, the best fuel consumptions for the open Tangye type of conbustion chamber (for the same volumetric efficiency, cooling water outlet temperature and speed), for the following B.H.Ps.

B.H.P.	Fuel Consumption in lbs. per hour.
5	2.5 lbs. per hour.
6	3.0 " " "
7	3.55 " " "
7.5	3.85 " " "

It appears from the curves that the optimum value of arc of shrouding is of the order of 150^{o} (i.e. 210^{o} of opening). This gives approximately the best fuel consumption, in all the cases shown. With position (A) at 7.5 B.H.P. the fuel consumption varied from 3.85 lbs. per hour up to 7.1 lbs. per hour. The curves for the four different B.H.Ps. slope down from 60^{o} valve opening to a minimum at 210^{o} valve opening, then slope up to a remarkably high fuel consumption.

R.P.M. = 1200 APPROX.
TIMING FULLY RETARDED.
CONSTANT VOL. EFF'Y.

POSITION "A"



7.5 B.H.P.

7 B.H.P.

6 B.H.P.

5 B.H.P.

B.M.E.P. IN POUNDS PER SQUARE INCH

7

6

5

4

3

2

60° 80° 120° 140° 300°

OPENING IN VALVE SHROUDING (DEGREES)

FIG. 16.

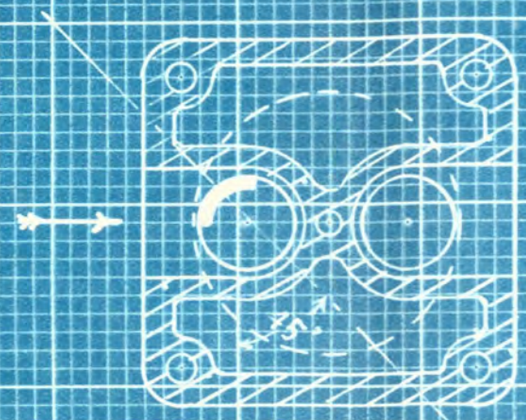
D. J. 1935

POSITION "F". In Fig. (17) the best fuel consumptions in this particular combustion chamber, and for the same conditions of running for the few different B.H.P., are as follows:-

B.H.P.	Fuel Consumption in lbs. per hour.
5	2.5 lbs. per hour.
6	3.0 " " "
7	3.42 " " "
7.5	3.7 " " "

The optimum opening for this position appears to be of the order 160^{o}, i.e. (200^{o} of shroud), roughly independent of load. At 7.5 B.H.P. the fuel consumption varied between 3.7 lbs. per hour up to 7.2 lbs. per hour. The four curves in the last mentioned figure slope down from 60^{o} opening to reach the minimum fuel consumption at approximately 160^{o} opening and rise again to a high value of fuel consumption at 360^{o} opening.

R.P.M. = 1200 APPROX.
TIMING FULLY RETARDED
VOL. EFFY. IS CONSTANT



POSITION "F"

7.5 B.H.P.

7 B.H.P.

6 B.H.P.

5 B.H.P.

FUEL IN LBS. PER HOUR

7
6
5
4
3
2

60° 80° 120° 240° 360°

OPENING IN VALVE SHROUDING (DEGREES)

D.J. 1935

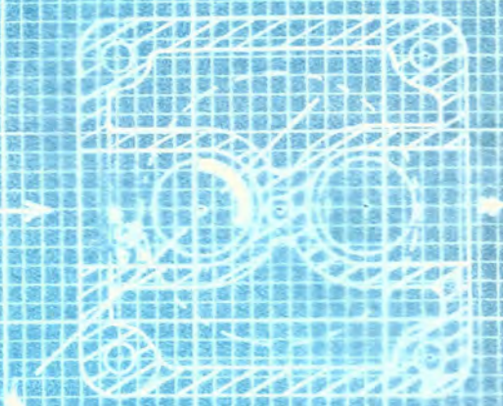
FIG. 17.

POSITION "E". This position gave curves of peculiar shapes, which may be due to the rather complicated paths taken by the air. Starting from 360° opening the curves in Fig. (8) slope down to the best positions at about 140° of shroud, i.e. (220° opening), when they curve up and then come down again to a certain extent when they reach 60° opening. At the best opening the fuel consumptions are as follows:-

B.H.P.	Fuel Consumption in lbs. per hour.
5	2.56 lbs. per hour.
6	3.0 " " "
7	3.5 " " "
7.5	3.75 " " "

Also at 7.5 B.H.P. the variation in fuel consumption was from 3.75 lbs. per hour up to 7 lbs. per hour. The four curves start with fairly low fuel consumption at 60° opening and curve upwards and then downwards again. They indicate minimum consumptions at about 220° and the fuel consumption

R.P.M. = 1200 APPROX.
TIMING FULLY RETARDED
VOL. EFFY. IS CONSTANT



Position "E"

7½ B.H.P.

7 B.H.P.

6 B.H.P.

5 B.H.P.

FUEL IN LITRES PER HOUR

6

5

3

60° 90° 120° 240° 340°

OPENING IN VALVE SHROUDING (DEGREES)

FIG. 18

D.J. 1935

increases up to the greatest at 360° opening (unshrouded valve).

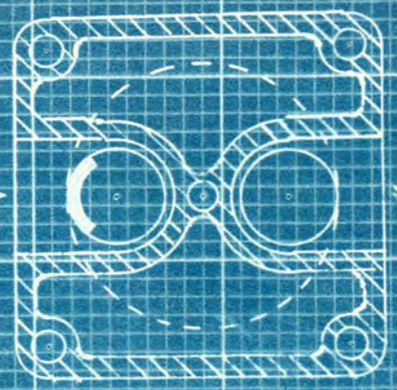
POSITION "B". It may be of interest to note that in this position the best fuel consumptions were obtained when the inlet valve with the smallest opening was used; the next best consumption was given when the unshrouded inlet valve was used. The curves in Fig. (19) also shew maximum fuel consumptions, which were at approximately 180° of opening in the inlet valve (180° of shroud), are given below.

B.H.P.	Fuel Consumption in lbs. per hour.
5	3.86 lbs. per hour.
6	4.75 " " "
7	6.35 " " "

The three curves in Fig. (19) start at fairly low fuel consumption at 60° opening, and slope upward to reach a maximum at about 180° of opening. They then slope down until they reach 360° of opening.

POSITION "C". In this position the fuel consumption was very high. The three curves in Fig. (20) are almost the same shape as those of position B shewn in Fig. (19), but

POSITION "B"



7 H.P.

6 H.P.

5 H.P.

R.P.M. = 1200 APPROX.
TIMING FULLY RETARDED
CONSTANT VOL. EFFY.

FUEL IN LBS PER HOUR

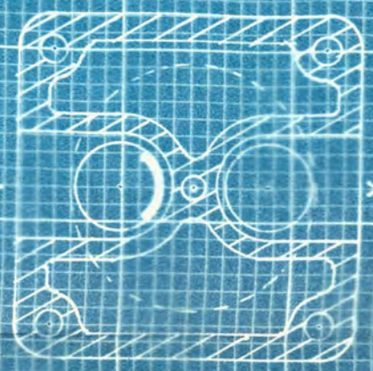
7
6
5
4
3
2

60° 80° 120° 240° 360°

OPENING IN VALVE SHROUDING (DEGREES)

FIG. 19.

D. J. 1935



POSITION "C"

7 B.H.P.

6 B.H.P.

5. B.H.P.

R.P.M. = 1200 APPROX.
TIMING FULLY RETARDED
CONSTANT VOL. EFF'Y.

FUEL IN LBS. PER HOUR

7
6
5
4
3
2

60° 80° 120° 240° 360°

OPENING IN VALVE SHROUDING (DEGREES)

D.J. 1935

FIG 20

the fuel consumptions are higher.

It appears that an inlet valve with 140° opening (220° of shroud) would give a high figure of fuel consumption, which is shewn as follows:-

B.H.P.	Consumption in lbs. per hour.
5	3.86 lbs. per hour.
6	4.75 " " "
7	6.35 " " "

These results, obtained by the author, may indicate the existence of an optimum inlet velocity of the air, and it might be argued from this (with certain valve positions) optimum swirl. These results not only indicate the existence of optimum air inlet velocity alone, but may show that the value of this is effected by the shrouding position, i.e. by the direction in which the air enters the cylinder through the inlet valve. In other words, there is a definite valve opening for every shrouding position, which gives the minimum fuel consumption for that position.

SECTION "B".

VARIATION IN THE LIFT OF THE INLET VALVE AND ITS EFFECT ON COMBUSTION.

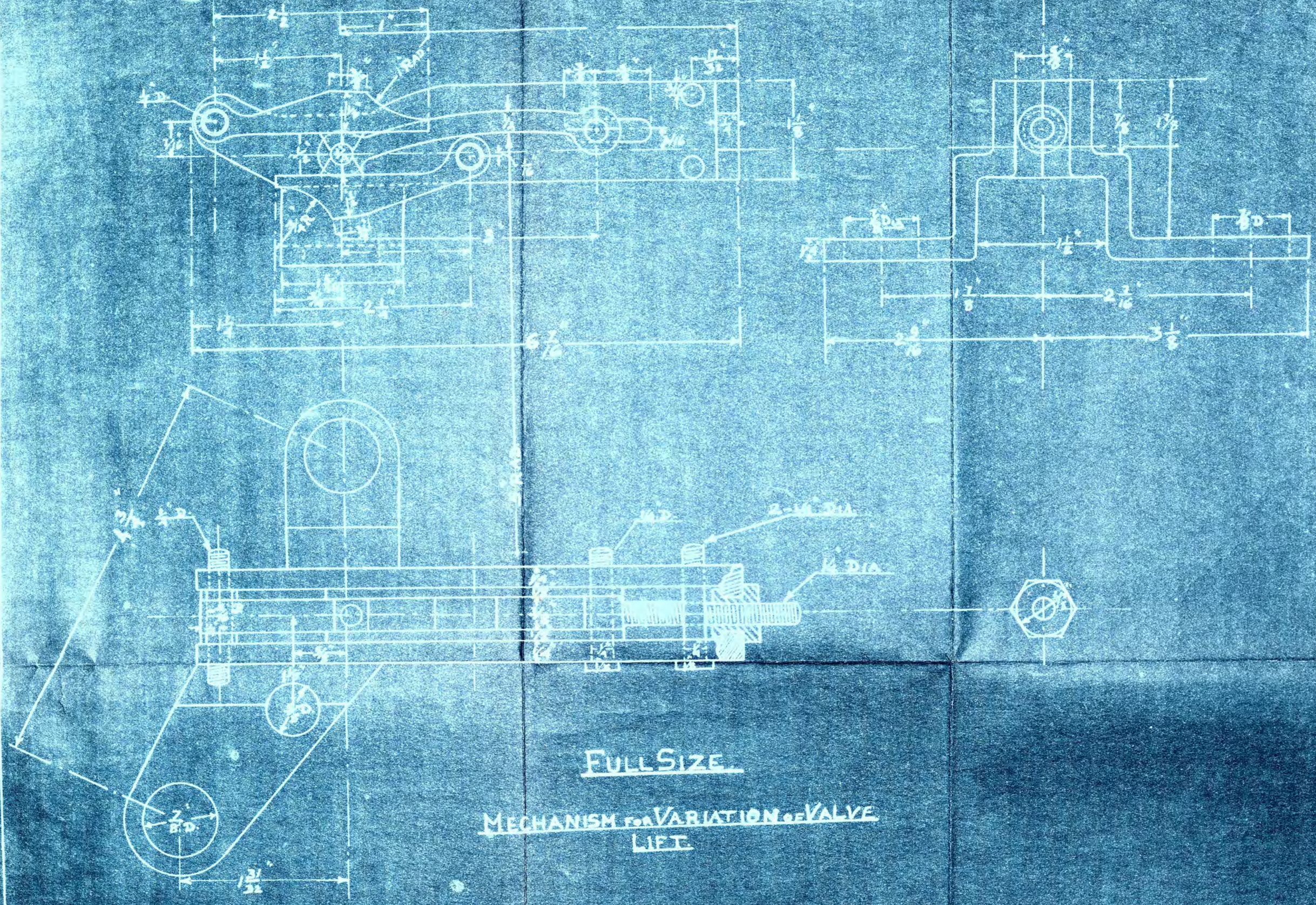
It was mentioned in the early part of this paper that two methods were used by the author in varying the air inlet velocity. The first (as was described earlier) is by varying the extent of the shrouding. The second is by varying the lift of the inlet valve.

DESCRIPTION OF APPARATUS.

The engine and most of the apparatus described in Section "A" were used in the present experiments. In addition, the air cooler,which has been mentioned earlier, was installed. There were also the following:-

Mechanism for Varying the Valve Lift.

Fig. (21) shows the mechanism which was designed by the author for the variation in the inlet valve lift. It is of a very simple type, consisting of two levers pivoted at their ends with a sliding pin in between them, held by two rockers. The lift of the valve could be varied by varying the position of the sliding pin. This could be done even when the engine is

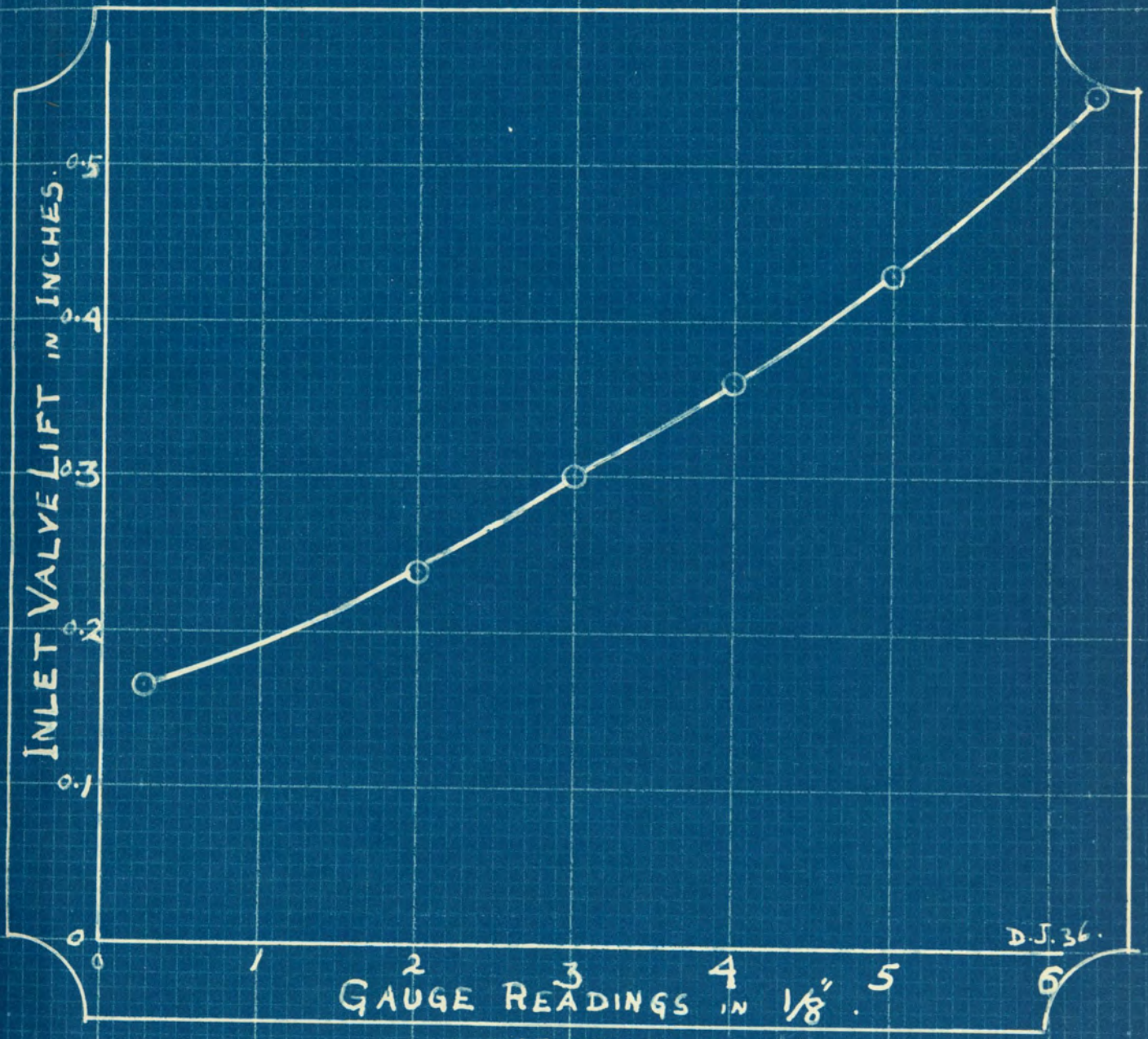


FULL SIZE.

MECHANISM FOR VARIATION OF VALVE LIFT.

FIG. 21.

D. J. 5/3/35.


INLET VALVE LIFT IN INCHES.
0
0.1
0.2
0.3
0.4
0.5
0
1
2
3
4
5
6
GAUGE READINGS IN 1/8".
D.J.36.

FIG(22)

running. A small scale at the side enabled the operator to know the value of the lift of the valve after a calibration had been carried out.

Fig. (22) shows a calibration curve giving the lift of the valve in inches in relation to the side scale of the mechanism.

INLET VALVES. Two inlet valves were used. The first had 120^{o} arc of shrouding (240^{o} opening), the second 150^{o} shrouding (210^{o} opening). These shrouded inlet valves were specially made, having a shrouding which was $\frac{5}{8}$" in height, so that the air could be properly directed, even at high values of inlet valve lifts. Also, the thickness of the shrouding was reduced to compensate for the addition of the weight due to height of shrouding.

Method Used for Carrying Out the Experiments.

The two valves mentioned were set in three different positions, namely A, E and F (see Fig.(6)). Three loads for each position were tried.

The speed of the engine was regulated so as to register approximately 1,200 R.P.M. The air inlet temperature, with the aid of the cooler, was kept at about 20°C, while the cooling water outlet temperature was kept at 60°C. Most of these tests were carried out while the volumetric efficiency was constant, the method used to achieve this being to throttle down, when big inlet valve lifts were employed, and supercharge the engine when small inlet valve lifts were used. A few of the tests were carried out, while the throttle was fully opened and in direct communication with the atmosphere. These few tests were carried out for the purpose of studying the effect of the lift of the inlet valve on the fuel consumption for the few loads. In each of these tests a curve was obtained, when the engine was steady. The load was kept constant for each curve, as also was the speed of the engine, the only variables being the lift of the valve and the fuel consumption. Only in a few of the curves was the volumetric efficiency altered.

TEST RESULTS.

VALVE WITH 120° OF SHROUD (240° OPENING).

Position A.

In Fig. (23) there are three curves which shew the relation between the valve lift, which varies from .166" to 0.454", and fuel consumption. The volumetric efficiency was kept constant. The three curves are plotted with constant loads, namely 10, 7, 8 and 6 B.H.Ps. The fuel consumption varies considerably with the valve lift. With a load of 10 B.H.P. the variation was between 46 to 4.93 lbs. per hour. On the 8 B.H.P. load the variation was from 3.67 to 3.94 lbs. per hour. On 6 B.H.P. load the consumption varied from 2.91 to 3.175 lbs. per hour.

Starting from the very small values of valve lift, the curves slope down to a minimum fuel consumption, then rise slightly. There seems to be an optimum valve lift of the value of 0.430", where the fuel consumption is a minimum.

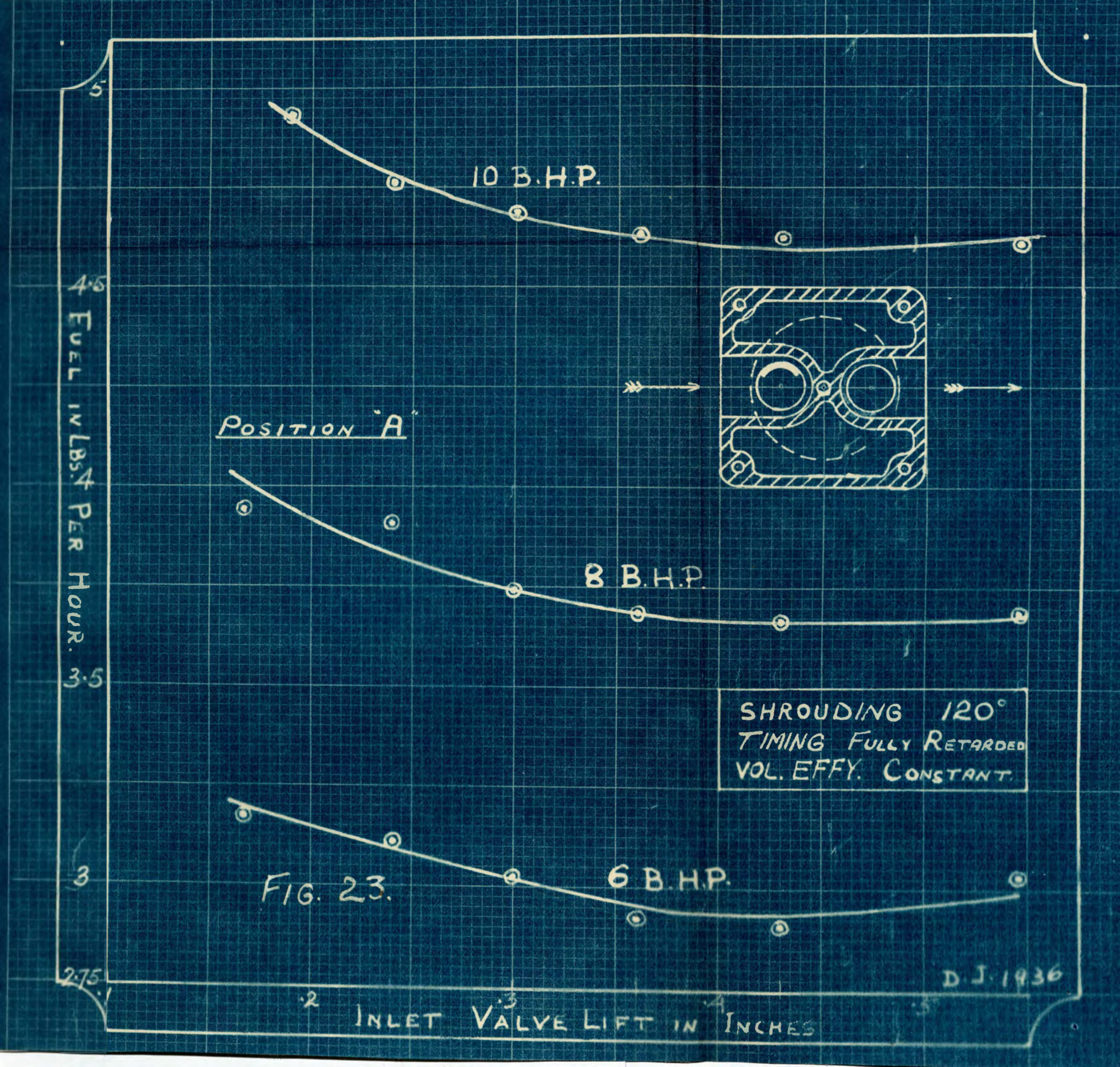

10 B.H.P.
POSITION "A"
8 B.H.P.
6 B.H.P.
SHROUDING 120°
TIMING FULLY RETARDED
VOL. EFFY. CONSTANT
5
4·5
4
3·5
3
2·75
FUEL IN LBS. PER HOUR.
·2
·3
·4
·5
INLET VALVE LIFT IN INCHES
D.J. 1936

FIG. 23.

Position E.

The three curves shewing the relation between valve lift and fuel consumption are plotted in Fig. (24). For three different loads in these tests, the fuel consumptions varied as follows:-

B.H.P.	Consumption at 0.166" lift.	At 0.43" lift.
10	4.95 lbs. per hour.	4.58 lbs. per hour.
8	4.06 " " "	3.84 " " "
6	3.14 " " "	2.92 " " "

The fuel consumption with each curve is very slightly increased at the maximum inlet valve lift. The optimum value of the valve lift is somewhere about 0.43".

Position F.

In this position the tests in Fig. (25) give a much higher fuel consumption at high loads, as compared with the two previous positions, this being shown by:-

B.H.P.	Consumption at 0.166" lift.	At 0.43" lift.
10	5.9 lbs. per hour.	4.78 lbs. per hour.
8	4.23 " " "	3.83 " " "
6	3.1 " " "	2.9 " " "

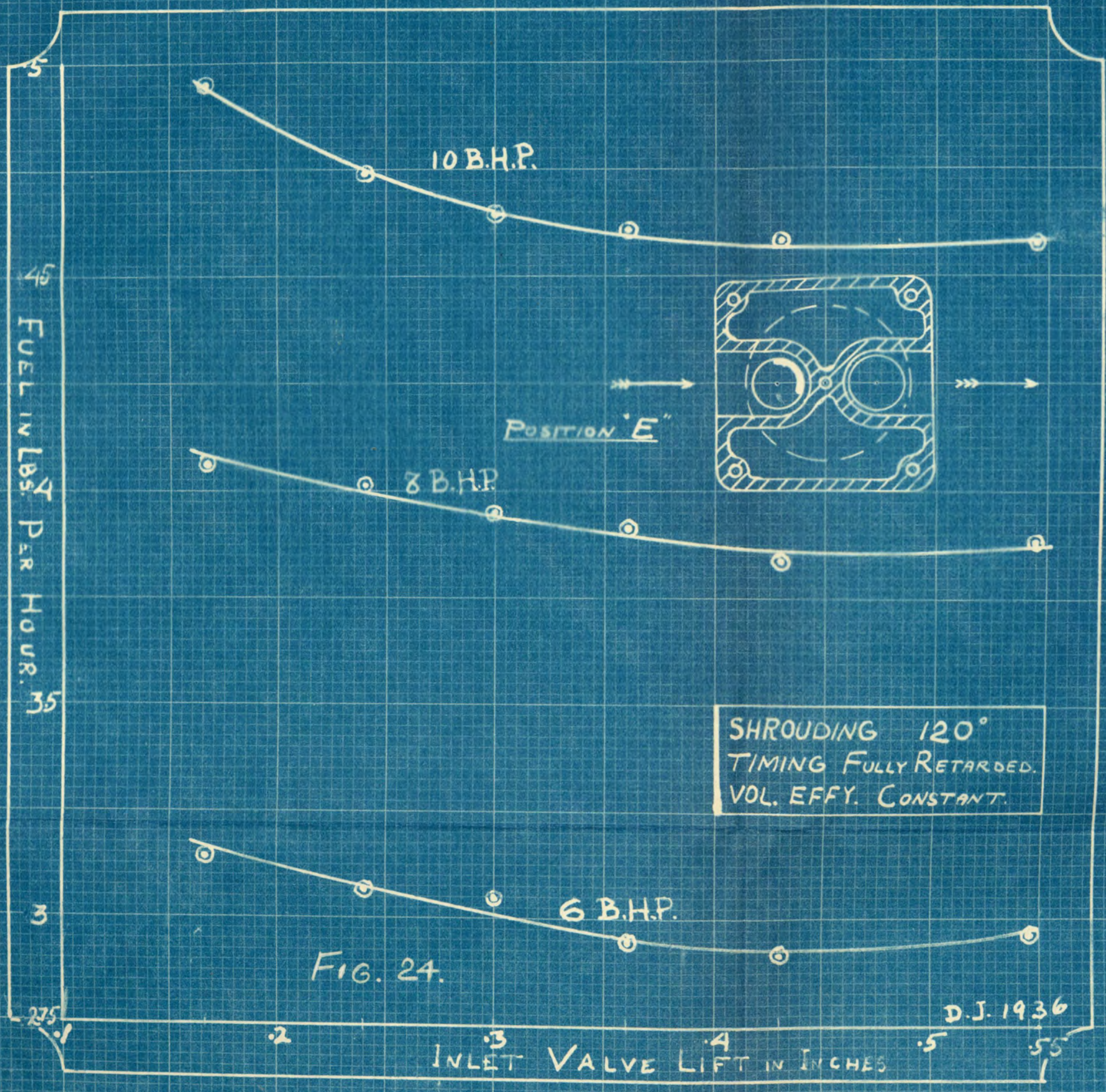
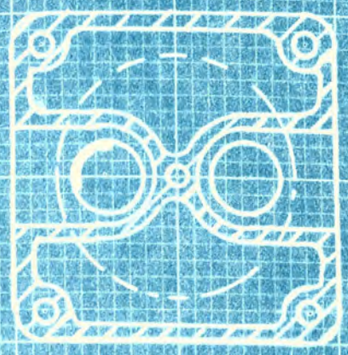

10 B.H.P.
8 B.H.P.
6 B.H.P.
POSITION "E"
SHROUDING 120°
TIMING FULLY RETARDED.
VOL. EFFY. CONSTANT.
FUEL IN LBS. PER HOUR.
5
4.5
3.5
3
2.75
.1
.2
.3
.4
.5
.55
INLET VALVE LIFT IN INCHES
D.J. 1936

FIG. 24.

10 B.H.P.

POSITION "F"



SHROUDING 120°
TIMING FULLY RETARDED
VOL. EFFY. CONSTANT

8 B.H.P.

6 B.H.P.

FUEL IN LBS. PER I.H.P. HOUR

6
5.5
5
4.5
4
3.5
3
2.5

.1 .2 .3 .4 .5

INLET VALVE LIFT IN INCHES

FIG. 25

D J. 1936

The optimum valve lift which gives the minimum fuel consumption is 0.43" approximately for this position, and it is more marked than in the two previous positions.

Fig. (26) contains two curves for the same valve and the same position, namely the valve with 120° of shroud and position "F". These two curves are test results for two loads which shew the variation in fuel consumption relative to valve lift, while the volumetric efficiency was not constant. The supercharger was not used for these two curves. The fuel consumption was fairly low at high values of the valve lift, but increased considerably with minimum valve lift. At 6 B.H.P. the fuel consumption was 2.86 lbs. per hour at 0.43" valve lift, and increased to 3.62 lbs. per hour at 0.166" valve lift. At 10 B.H.P. the fuel consumption was 4.9 lbs. per hour at 0.43" valve lift, and increased enormously at small lifts, until it was not possible to reach the point where the lift of the valve is 0.166". This is undoubtedly due to the fact that there is not sufficient air to burn the large quantity of the fuel introduced into the cylinder head. In other words,

10 B.H.P.

POSITION "F"



6 B.H.P.

SHROUDING 120°
TIMING FULLY RETARDED

FUEL IN LBS PER HOUR.

5.5
5.0
4.75
3.5
3.25
3.0
2.75
2.5
2.25

·2 ·3 ·4 ·5

INLET VALVE LIFT IN INCHES.

D.J. 1930

FIG. 26.

the air was the scarce fluid in this case.

The curves in Fig. (26) were plotted to shew the effect of variation in the lift of the inlet valve, while the suction side of the engine was in direct communication with the atmosphere and with no restriction in the suction side whatever.

VALVE WITH 150° SHROUD (210° OPENING).

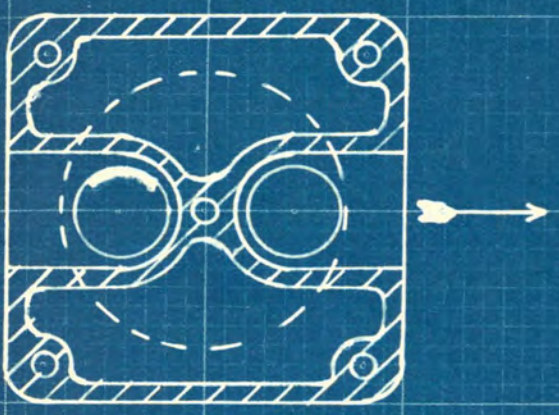
Position "A".

In Fig. (27) the three curves are test results which shew the relation between fuel consumption in lbs. per hour and valve lift, the volumetric efficiency being constant. The curves are plotted for three B.H.Ps., namely 10, 8 and 6. The variation in fuel consumption is shown as follows:-

B.H.P.	Consumption at 0.166" lift.	At 0.545" lift.
10	5.06 lbs. per hour.	4.59 lbs. per hour.
8	4.13 " " "	3.65 " " "
6	3.377 " " "	2.92 " " "

10 B.H.P.

POSITION "A"



8 B.H.P.

6 B.H.P.

SHROUDING 150°
TIMING FULLY RETARDED.
VOL. EFFY. CONSTANT.

FUEL IN LBS PER H

Hour.

5
4.5
4
3.5
3
2.5
2.25

·2 ·3 ·4 ·5

INLET VALVE LIFT IN INCHES.

D. J. 1936

FIG. 27.

The curves also shew in the case of this valve that the least fuel consumption was obtained, when the inlet valve lift was at 0.545".

Position "E".

In this position the test results are plotted in Fig. (28) for the three different loads, the fuel consumptions in this position being:-

B.H.P.	Consumption at 0.166" lift.	At 0.545" lift.
10	5.34 lbs. per hour.	4.83 lbs. per hour.
8	4.06 " " "	3.73 " " "
6	3.24 " " "	2.95 " " "

The least fuel consumptions were at the maximum lift, i.e. 0.545" lift.

Position "F".

The fuel consumption in this position is much higher than in the two previous positions. The three curves shewn in Fig. (29) are test results, and give the following values:-

10 B.H.P.

8 B.H.P.

6 B.H.P.

POSITION "E"



SHROUDING 150°
TIMING FULLY RETARDED.
VOL. EFFY. CONSTANT.

FUEL IN LBS PER B.H.P. HOUR.

5
4.5
4
3.5
3
2.75

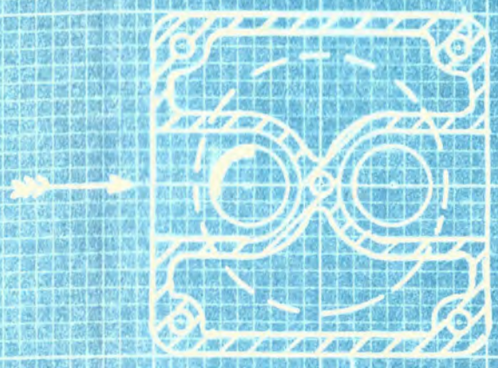
.2 .3 .4 .5

INLET VALVE LIFT IN INCHES.

D.J. 1936

FIG. 28.

10 B.H.P.



POSITION "F"

SHROUDING 150°
TIMING FULLY RETARDED
VOL. EFFY. CONSTANT

8 B.H.P.

6 B.H.P

FUEL IN LBS. PER H.P. HOUR

4.5
4.0
3.5
3.0
2.5

.1 .2 .3 .4 .5

INLET VALVE LIFT IN INCHES

D.J 1936

FIG. 29.

B.H.P.	Consumption at 0.166" lift.	At 0.545" lift.
10	6.1 lbs. per hour.	4.9 lbs. per hour.
8	4.66 " " "	3.88 " " "
6	3.63 " " "	3.04 " " "

The least fuel consumptions were also at about 0.545" lift.

Fig. (30 contains three curves which are test results for 10, 8 and 6 B.H.P. The valve and position were as before, but volumetric efficiency was variable and the engine was in direct communication with the atmosphere. At 8 and 10 B.H.P. when the valve lift was at its minimum there was insufficient air in the cylinder to burn the fuel introduced, therefore the load could not be maintained; while at 6 B.H.P. the fuel consumption was about 0.9 lbs. per B.H.P. hour. For the maximum valve lift, the consumptions were 4.83, 3.94, and 3.14 lbs. per hour, corresponding to 10, 8 and 6 B.H.Ps. respectively.

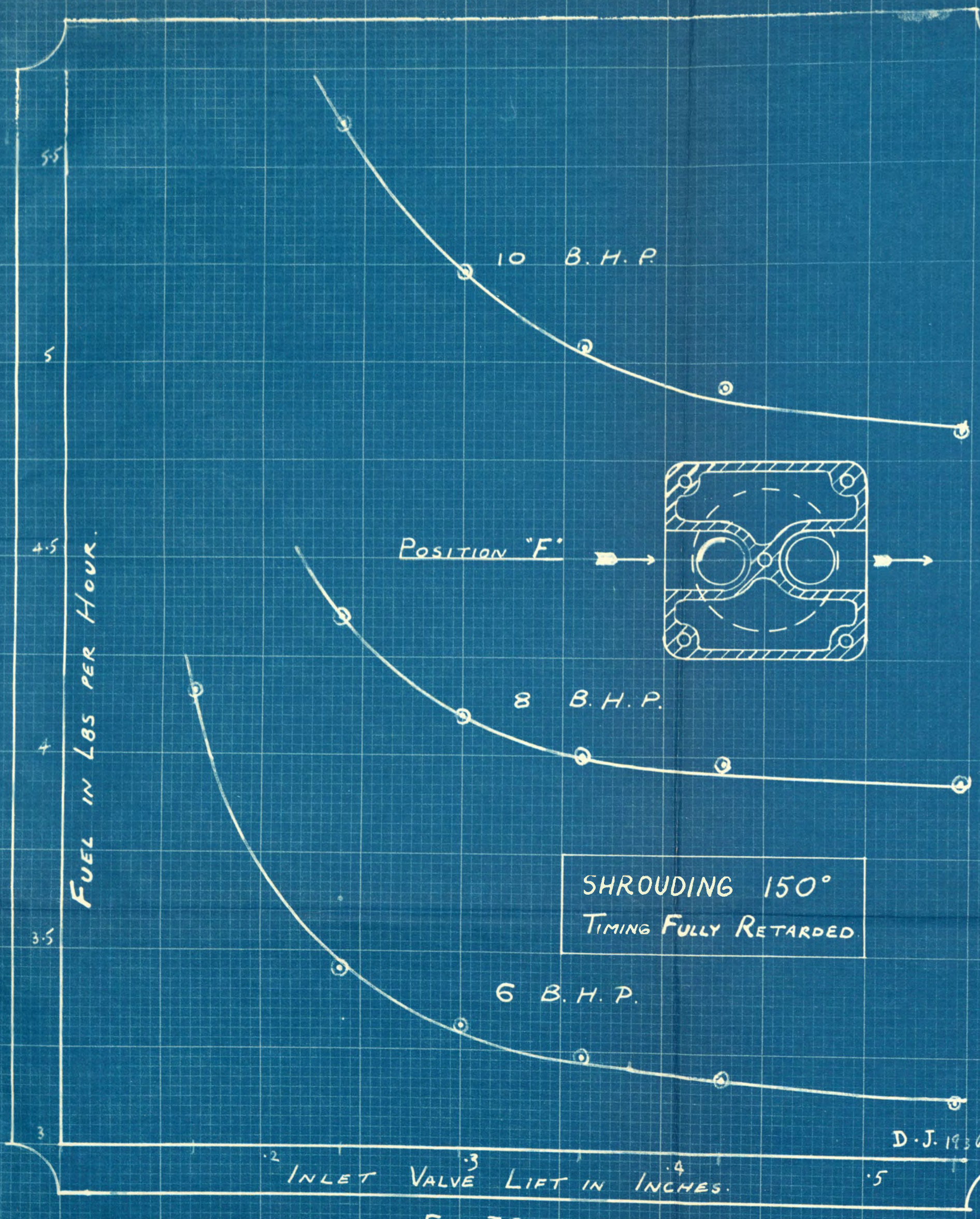


FIG. 30.

BRITISH MADE

DISCUSSION OF RESULTS.

A general survey of the experiments with valve lift variation carried out on the Tangye Oil Engine may give some idea of the important part the air velocity plays in the combustion process. It seems that certain valve lifts give best fuel consumptions, while an increase or decrease in the value of these lifts raises the fuel consumption to a considerable value in some cases.

To study the effect of the air inlet velocity on the combustion process, i.e. fuel consumption, the curves shewing the relation between the fuel consumption and inlet valve lift were replotted on different lines. The air inlet velocity was calculated roughly. The assumption made was that the inlet valve opened instantly to the maximum value at the beginning of the suction stroke, and closed instantly at the end of the suction stroke. It is realised, however, that this is not the actual fact, but for the sake of comparison this method of calculating the average speed of the air through the inlet valve was used.

VALVE WITH 120° SHROUDING (240° OPENING).

Position A.

Fig. (23) is replotted in Fig. (31). The curves shew the relation between the fuel consumption in lbs. per hour against air inlet speed (average) in feet per second. The consumptions of the fuel per hour expressed against B.H.P. at the speed of (147) ft. per second are as follows:-

B.H.P.	Fuel Consumption. at 147 ft. per sec.	Fuel Consumption at 381 ft. per sec.
10	4.62	4.93
8	3.67	3.94
6	2.91	3.175

It may also be of some interest to note that the optimum air speed was 147 ft. per second approximately.

This speed could also be put in terms of the crank revolutions, making the following assumption. We assume (1) that the air on leaving the inlet valve takes the almost circular motion round the centre of the combustion chamber, and (2) that the mean radius of the swirling air is taken at the centre of inlet valve to the centre of the combustion chamber.

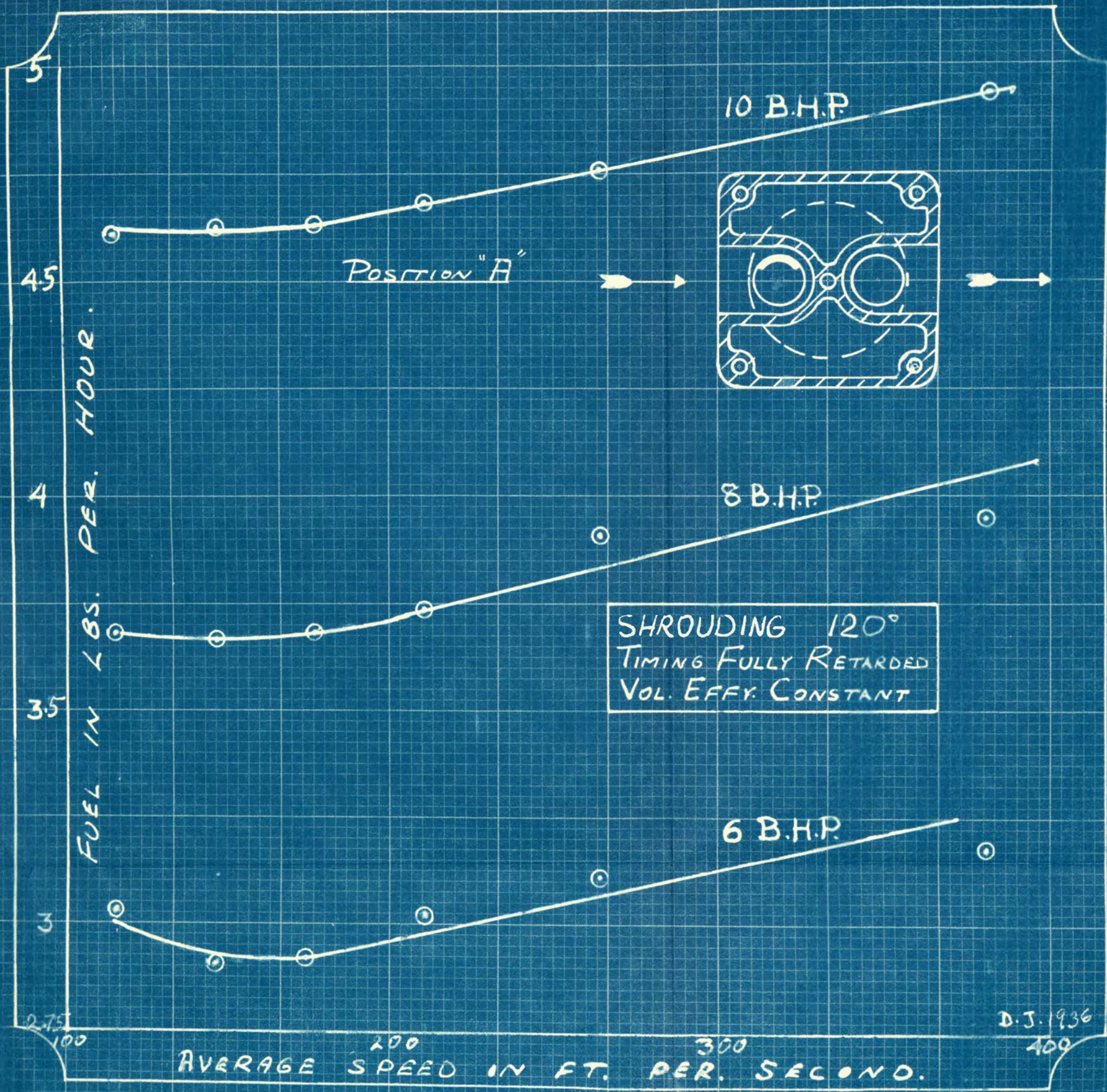

10 B.H.P.
POSITION "A"
8 B.H.P.
6 B.H.P.
SHROUDING 120°
TIMING FULLY RETARDED
VOL. EFFY. CONSTANT
FUEL IN LBS. PER. HOUR.
5
4.5
4
3.5
3
2.75
100
200
300
400
AVERAGE SPEED IN FT. PER. SECOND.
D.J. 1936

FIG. 31.

It is appreciated, however, that this radius varies with the shrouding orientation; as the problem of the motion of the air inside the combustion chamber is much too complicated, it was decided on the second assumption for the sake of comparison only, and as a rough method of expressing air swirl in terms of crank revolutions.

Assuming the average velocity of the inlet air v ft. per second, the

Air swirl $\frac{v}{2\pi r}$ Revs./sec., r being the distance between the centre of the combustion chamber and the centre of the inlet valve.

$\therefore$ Air swirl in terms of the crank revolution

$= \frac{60\ v}{2\pi \times n}$ approximately where n is the crank revolution per minute. In the case of this position of the valve, therefore, the optimum swirl in terms of crank revolutions is of the order of 11.70 : 1 approximately.

The curves in Fig. (31) start at a fairly high fuel consumption at an air inlet speed of the value 381.0 ft. per second, and slope down to a minimum value of fuel consumption at the speed of the inlet air of 147.0 ft. per second. They then seem to curve very slightly upwards, to

show a slight increase in the fuel used by the engine at the low air speed of the order 116.0 ft. per second.

Position "E".

Fig. (32) shews the test results which were plotted in Fig. (24). The ordinates give the fuel consumption in lbs. per hour, while the abscissae represent the average inlet air velocity in ft. per second. The three curves represent as before 10, 8 and 6 B.H.P. The values of the fuel used are as follows:-

B.H.P.	Consumption at 147 ft. per sec.	Consumption at 381 ft. per sec.
10	4.58 lbs.per hr.	4.95 lbs. per hr.
8	3.84 " " "	4.06 " " "
6	2.92 " " "	3.14 " " "

From the curves the best average air inlet velocity to give minimum consumption is of the order of 147.0 ft. per second, approximately, which corresponds to an air swirl of about 11.7 times the crank revolutions. The curves are almost the shape of those in the previous position.

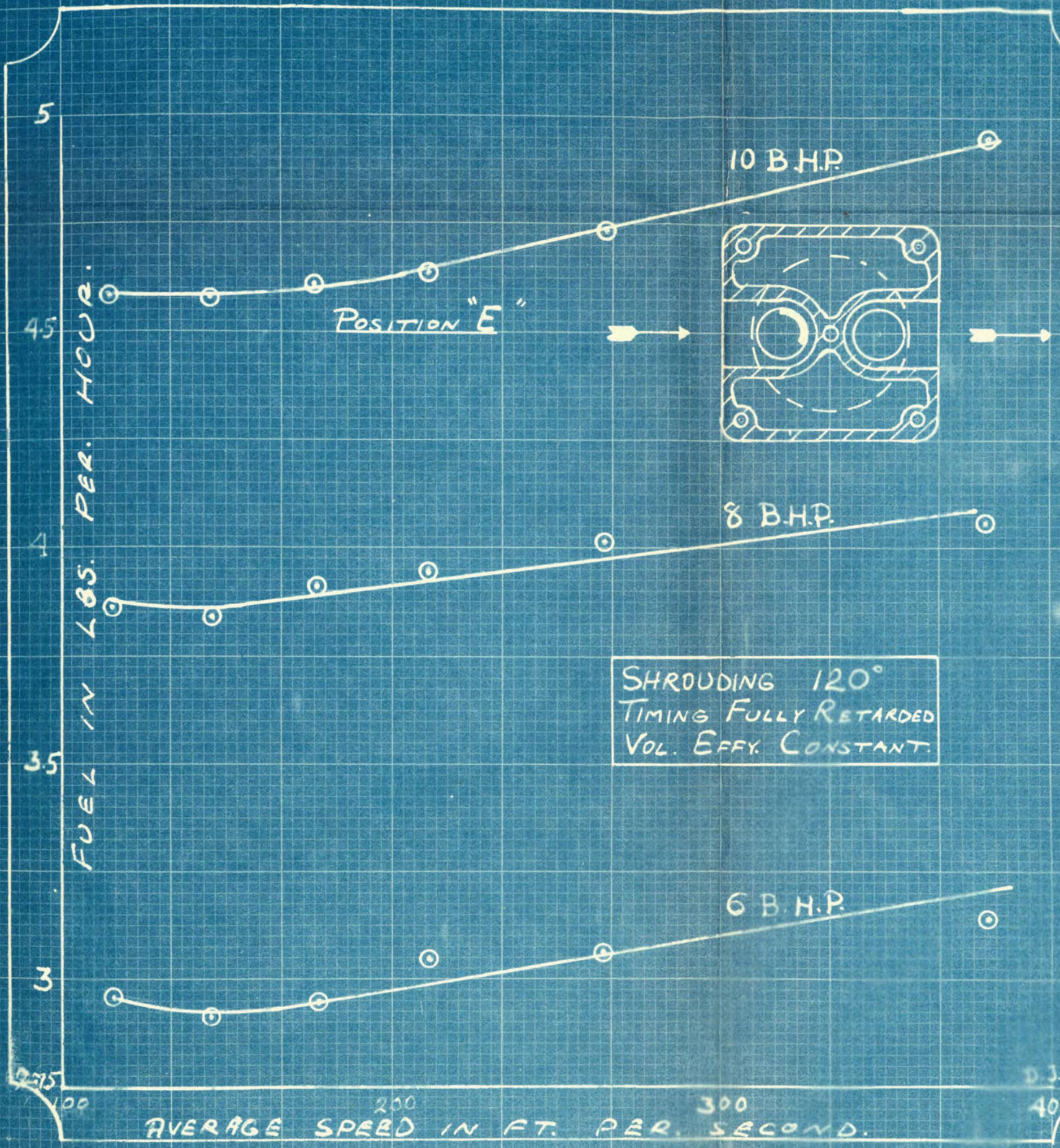

5
4.5
4
3.5
3
2.5
LBS. PER. HOUR.
FUEL IN
10 B.H.P.
POSITION "E"
8 B.H.P.
6 B.H.P.
SHROUDING 120°
TIMING FULLY RETARDED
VOL. EFFY. CONSTANT.
D.J.1936
100
200
300
400
AVERAGE SPEED IN FT. PER. SECOND.

FIG. 32.

Position "E".

Fig. (33) is also a reproduction of Fig. (25) and the fuel consumptions at the three loads are as follows:-

B.H.P.	Fuel Consumption at 147.0 ft. /sec.	Fuel Consumption at 381.0 ft./sec.
10	4.78 lbs. per hour.	5.91 lbs. per hour.
8	3.83 " " "	4.23 " " "
6	2.90 " " "	3.10 " " "

From the three curves the average air inlet velocity at which the engine is most economical at this position is of the order 147 ft. per second approximately. The corresponding air swirl in terms of crank revolutions is 11.70 approximately. The three curves shew a higher fuel consumption at high air inlet velocities as compared with the two previous positions.

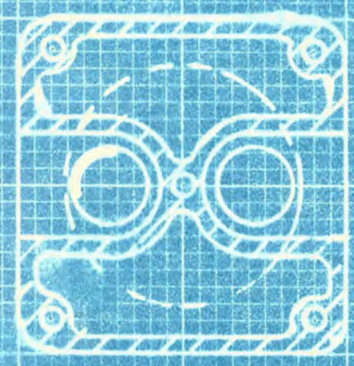
BRITISH MADE

FUEL IN LBS. PER B.H.P. PER HOUR.

6
5.5
5
4.5
4
3.5
3
2.75

10 B.H.P.

POSITION "E"



8 B.H.P.

SHROUDING 120°
TIMING FULLY RETARDED
VOL. EFFY. CONSTANT

6 B.H.P.

D.J. 1932

100 200 300 400

AVERAGE SPEED IN FT. PER. SECOND.

FIG. 33.

VALVE WITH 150° SHROUD (210° OPENING).

Position "A".

Fig. (27) was replotted with the average speed of the air through the inlet valve shown against fuel consumption (see Fig. (34)). The fuel used by the engine in this position was as follows:-

B.H.P.	Fuel Consumption at 133.0 ft./sec. Average Speed.	Fuel Consumption at 435 ft./sec. Average Speed.
10	4.59 lbs. per hour.	5.06 lbs. per hour.
8	3.65 " " "	4.13 " " "
6	2.92 " " "	3.377 " " "

The three curves indicate clearly that the average air inlet velocity of 133 ft. per second gave the best fuel consumption; the corresponding air swirl in terms of the crank angle was of the order 10.60 approximately. The curves also seem to slope downwards as the speed of average inlet velocity decreases.

Position "E".

In Fig. (35) are test results plotted from Fig. (28). The fuel consumptions for this position were:-

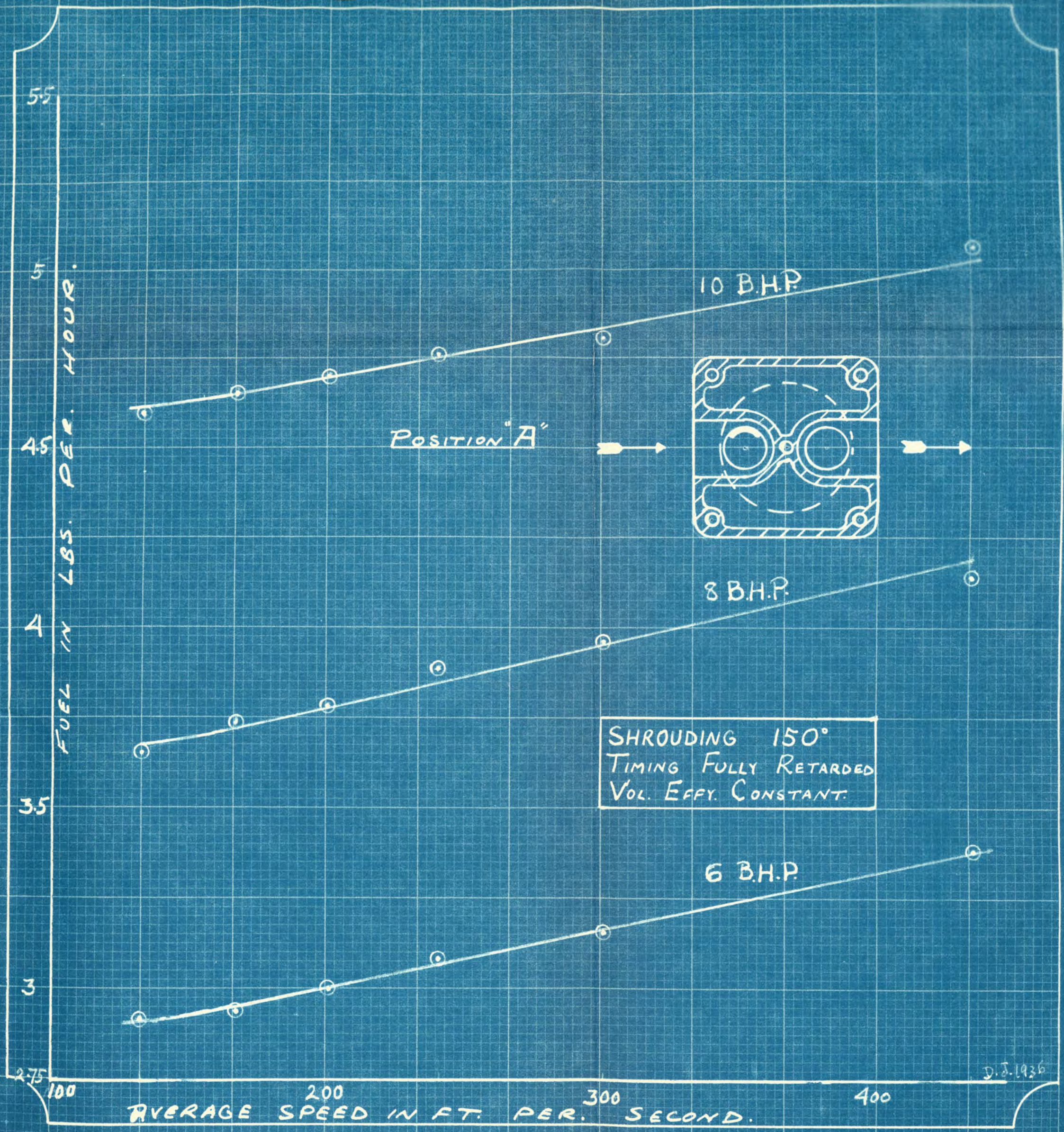

5.5
5
4.5
4
3.5
3
2.75
FUEL IN LBS. PER. HOUR.
10 B.H.P.
8 B.H.P.
6 B.H.P.
POSITION "A"
SHROUDING 150°
TIMING FULLY RETARDED
VOL. EFFY. CONSTANT.
D.J. 1936
100
200
300
400
AVERAGE SPEED IN FT. PER. SECOND.

FIG. 34.

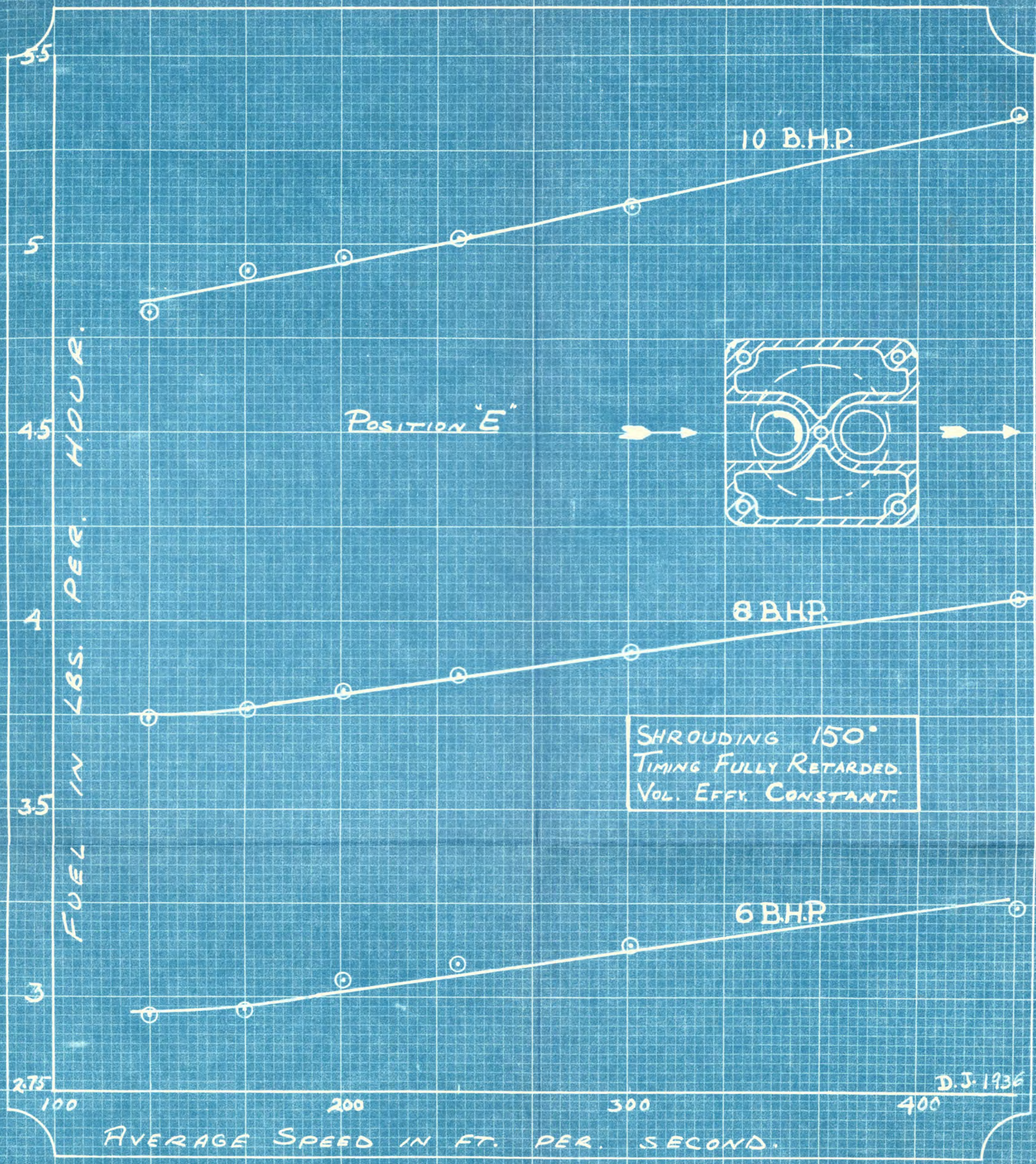

10 B.H.P.
8 B.H.P.
6 B.H.P.
POSITION "E"
SHROUDING 150°
TIMING FULLY RETARDED.
VOL. EFFY. CONSTANT.
FUEL IN LBS. PER. HOUR.
5.5
5
4.5
4
3.5
3
2.75
100
200
300
400
D.J. 1936
AVERAGE SPEED IN FT. PER. SECOND.

FIG. 35.

B.H.P.	Fuel Consumption at 133 ft./sec.	Fuel Consumption at 435 ft./sec.
10	4.82 lbs. per hour.	5.34 lbs. per hour.
8	3.73 " " "	4.06 " " "
6	2.95 " " "	3.24 " " "

The three curves also shew the least fuel consumption occurs when the air velocity through the inlet valve was 133 ft. per second. This corresponds to an optimum swirl of 10.60 (approx.) times the crank revolutions.

Position "F".

In this position the fuel consumptions against average air inlet velocity are plotted in Fig. (36), from test results from Fig. (29). Some of the results are as follows:-

B.H.P.	Fuel used at 133 ft./sec.	Fuel used at 435 ft./sec.
10	4.90 lbs. per hr.	6.1 lbs. per hr.
8	3.88 " " "	4.66 " " "
6	3.04 " " "	3.63 " " "

The optimum swirl in terms of crank revolutions is 10.60 approximately. The average speed at the minimum fuel consumption was about 133 ft./sec. The three curves

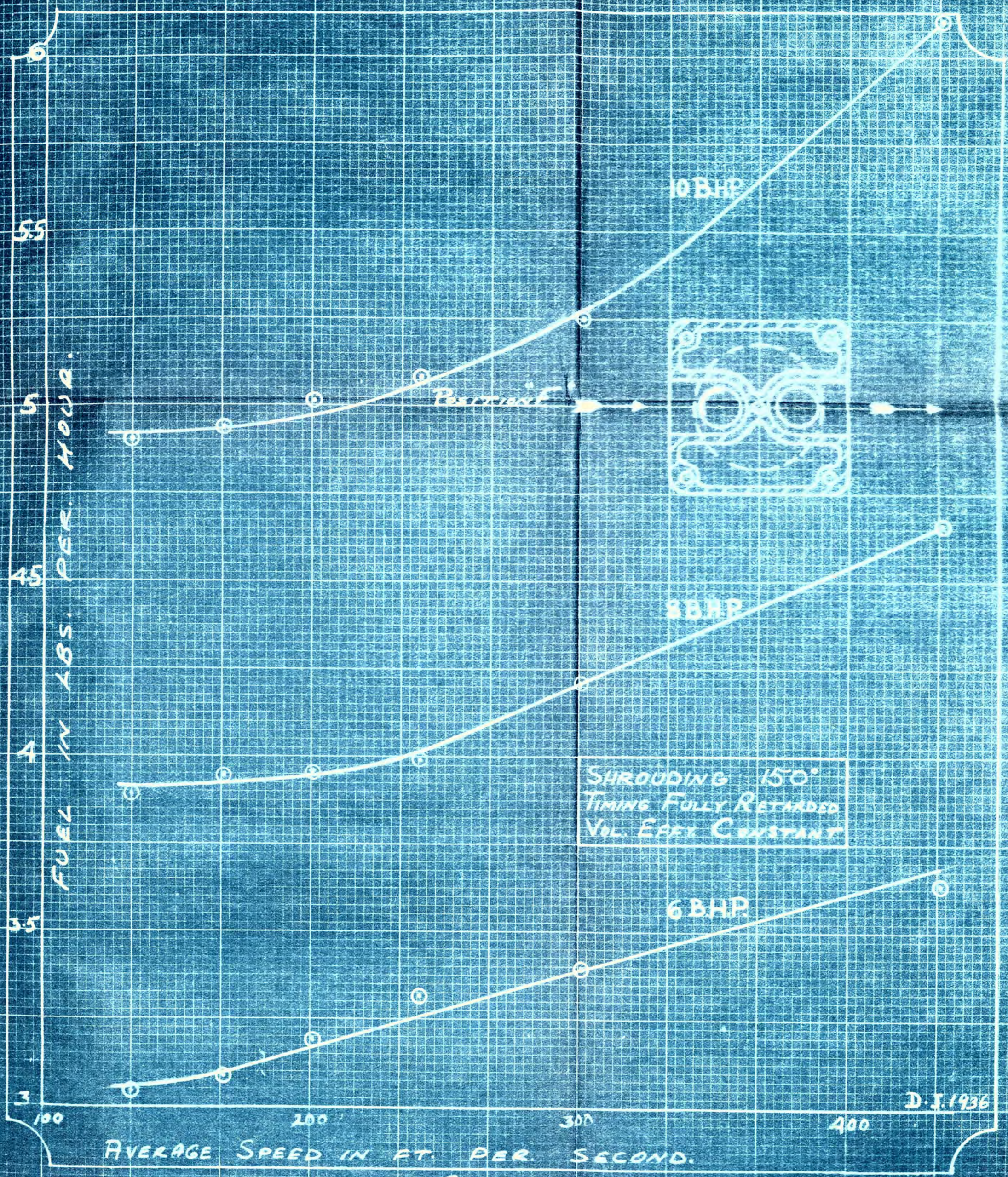

FUEL IN LBS. PER H.P.
5.5
5
4.5
4
3.5
3
10 B.H.P.
8 B.H.P.
6 B.H.P.
Barometer
SHROUDING 150°
TIMING FULLY RETARDED
VOL. EFFCY. CONSTANT
100
200
300
400
AVERAGE SPEED IN FT. PER SECOND.
D.I.1936

FIG. 36

also shew a slight increase in fuel consumed at high loads, as compared with the two previous positions.

From the above results, it seems probable that an optimum swirl does exist in the case of the open combustion chamber (Tangye) type. It is certain that by using very small lifts with the inlet valve (and it may be argued from this at very high inlet velocities of the air) the combustion process is effected and a considerable amount of the fuel is wasted.

A proper investigation of the results on the Tangye Engine, together with the evidence of the experiments carried out by Messrs. Ricardo and Alcock, reveals that there is a certain air swirl inside the combustion chamber, which would result in the most satisfactory combustion. In exceeding this value, the fuel consumption seems to increase. The author suggests the following reasons for this behaviour:-

(1) With excessive air swirl, the air may perhaps blow out the flame or delay its movement.

(2) A high swirl movement given to the air may perhaps cause the latter to drag the fuel with it. This of course will result in a slow burning due to the fact that the fuel is carried in zones where Oxygen is rare, and therefore the fuel will burn in a very slow manner.

(3) Another effect of the excessive air swirl may be to drag the fuel and force it on to the surface of the combustion chamber, resulting in a slow and unsatisfactory combustion. The surface of the combustion chamber is comparatively cool, and when the fuel is in contact with a cool surface and is not completely surrounded by air, it naturally burns very slowly. This in effect may account for the higher fuel consumptions.

(4) The excessive air swirl may cause the concentration of hot burnt gases in the middle of the combustion chamber. These burnt gases may surround the freshly injected fuel, in this type of combustion chamber where the nozzle is in the middle and delaying the burning process at the end of the second phase and during the third phase of the combustion process.

(5) Heat losses to the cylinder walls may be very appreciable when the air swirl exceeds a certain limit.

Any of the above reasons may be correct. It may be that the whole process may be effected for all these reasons, or may be due to one only or some combination of the reasons.

PART II.

"DELAY PERIOD AND DIESEL KNOCK".

INTRODUCTION.

C.B. Dicksee[3] states that the question of noise in a Diesel engine is important, not only from a public point of view, but because noise, which is sometimes accepted as inevitable in the compression ignition oil engine, is in reality detrimental to the engine itself. The Diesel knock, like detonation in a petrol engine, may result in high stresses, so that the relative roughness of the compression ignition engine is a serious handicap to its speedy and general adoption..

Heldt[1], in his book "High Speed Diesel Engines", states that in general the efficiency of the Diesel engine can be increased by advancing the injection, but unfortunately this improvement is usually accompanied by roughness in operation.

The general adoption, and the improvement in combustion efficiency of the Compression Ignition solid injection type seems to be dependent on methods of tackling the Diesel knock. Although the Diesel engine has many

advantages over the petrol engine, the noise problem, probably one of the main disadvantages, seems to be a big obstacle in the path of its success, especially in the motor car industry. For the same reason, the injection of the fuel is retarded, although this method of smoothing the operation of the engine, lowers its thermal efficiency.

So far as the petrol engine is concerned, there have been many theories suggested for the causes of knock. The author does not feel justified in describing all of these in this thesis. For detail classification see e.g. book by A.W. Nash and A. Howes. (36)

The following are the important theories advanced at present for petrol engine knock.

First:- The Spontaneous Ignition Theory of Knocking. (26)

Lloyd Withrow and Geral M. Rassweiler, with the aid of a speed camera and a quartz window, arrived at the following conclusion:- The knock in a petrol engine (the petrol having an octave number of 48, and the spark advance 25° degrees), started when auto-ignition occurred near the end of the combustion space, at a point well separated from the normal flame front. According to

this theory, the knocking properties of the fuel are determined by the temperature at which it will ignite on compression by the travelling wave.

Second:- The Peroxide Theory of Knocking.

This involves chemical changes inside the combustion chamber. A sampling mechanism has been used, gas being collected at different parts of the cycle and analysed. For details see paper by Professor Egerton and others.(18)

In a compression ignition oil engine, knock occurs at the early part of the combustion process. The fuel is injected and there is a time interval before it burns and the pressure rises; this time is called the delay period. During the delay period a quantity of fuel enters the cylinder; some of this fuel evaporates (with possible chemical change), and the rest is still in a liquid form (very fine droplets) at the end of this period. Once the ignition starts, the combustible mixture burns quickly. The knock occurs if a considerable amount of this fuel burns very rapidly at the end of this period.

C.B. Dicksee,(3) states that "the delay period

or time lag, which exists between the moment at which injection commences and the appearance of pressure-rise due to combustion, plays a vitally important part in the successful operation of the engine. It is during this period that knock originates; in fact it may be said without fear of contradiction that it is the delay period which produces the knock".

From the evidence given above, and the absence of published information concerning the chemical changes involved, it seems that the knock in a compression ignition oil engine may be caused by the existence of a large quantity of fuel, inside the combustion chamber, ready to unite after the delay period with the oxygen it has already mixed with.

Taking these views into account, the author decided to attempt to eliminate a great deal of the knock often observed in Diesel engines (and so ensure a smoother running of the engine and a longer life). Two methods suggest themselves.

First:- The Shortening of the Delay Period.

In this method, the delay period must be made as short as possible, by doping the fuel or increasing the compression temperature. With regard to doping, some liquid fuels have been tried out with some success. Thus, e.g. Mr. Ricardo,(10) has found that a small percentage of Amyl-nitrate has the effect of decreasing the ignition lag.

G.D. Boerlage and J.J. Broeze(23) have found that when some fuels cause knocking they could be made non-knocking by supercharging. Non-knocking fuels, however, could be made to knock on throttling down on the suction side of the engine. How far these effects are due to temperature, i.e. whether density of air charge has itself an effect, it is difficult to say. This method sometimes requires very high pressures to prevent rough fuels from knocking, and ensures a smoother running of the engine.

Secondly:- The Use of Pilot Injection for Anti-Knock Purposes.

The author here suggests the introduction of a small quantity of fuel to start ignition, this "pilot injection" to be followed by the main charge within a few

degrees after the delay period is over, i.e. after the flame has established itself in the combustion chamber. The author is not aware of any published results dealing with this method for reducing the Diesel knock.

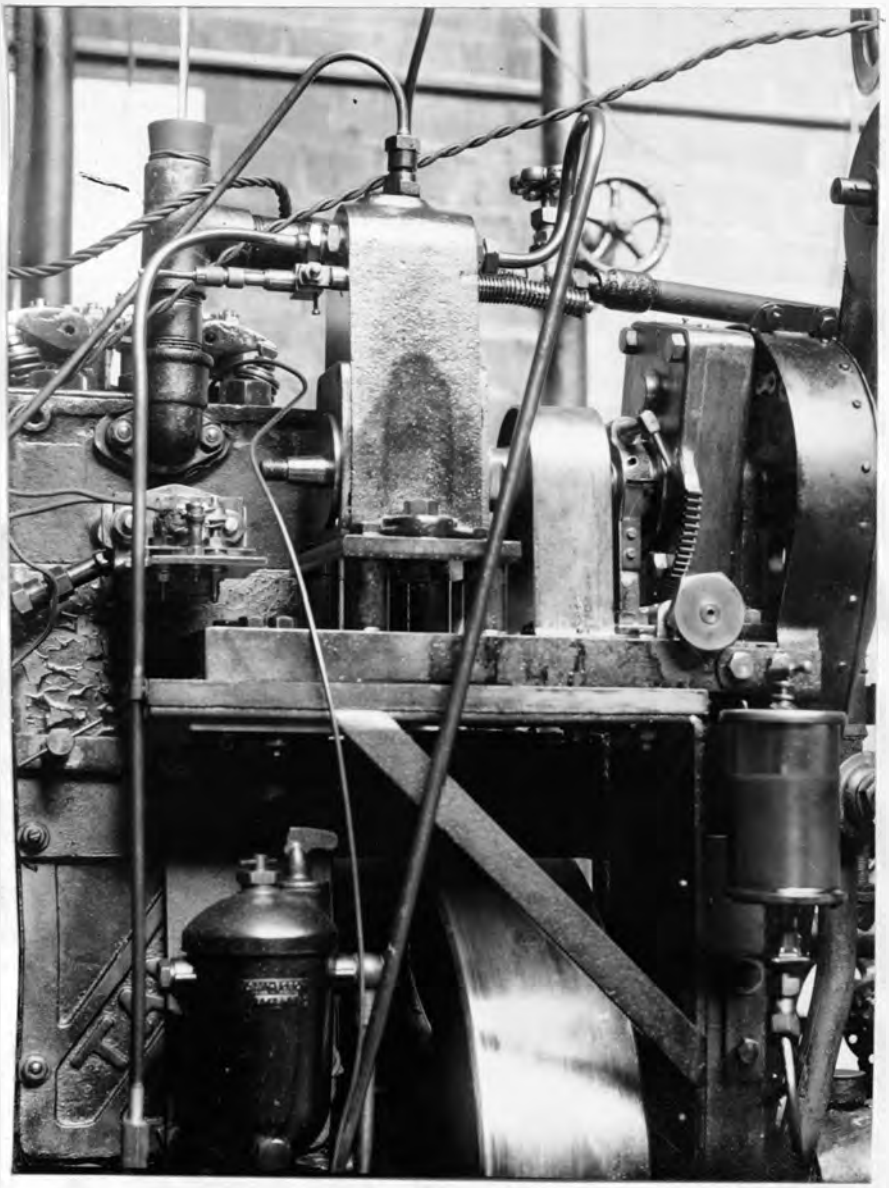
This method is an attempt to overcome the difficulty of the presence in the cylinder of a large quantity of the fuel charge after the delay period, and should ensure a quieter running of the engine. This method, as will be seen later, has proved to be fairly successful. To test the author's suggestion, a new Bosch pump and nozzle had to be installed. This will be referred to as the pilot injection pump and nozzle. Details of the installation and tests will be discussed later.

The author also suggests the possibility of a third method, which involves a device acting as a pressure relief mechanism. To keep the pressure constant after ignition some mechanism must be made to increase the volume. One arrangement would be the mounting of the crown of the piston on a spring, so that as the pressure inside the combustion chamber increases, the crown of the piston compresses the spring, and relieves the pressure.

In this method there is very little loss of efficiency due to making the cycle more of a constant pressure one. The advantage of it is on the one hand that only a little movement of the piston (of the order of $\frac{1}{8}$") is needed to increase the volume of the combustion chamber; on the other hand, at the end of the exhaust stroke, the piston clears out a larger volume of the exhaust gases. But on calculation it was found that the beam type spring, which is the best type for a rigid construction of the piston, would compress very little, and the utility of the device is probably negligible.

Another mechanism that suggested itself consisting of an auxiliary detachable chamber, may be of interest. This mechanism is a simple closed chamber, the volume of which is about one-third of that of the combustion chamber, with two mushroom type valves. One valve opens into the combustion chamber, and is held on its seat by means of a lift spring. The other opens into the auxiliary chamber and is held by a stiff spring to allow it to lift only when the pressure inside the combustion chamber exceeds a

certain limit. For simplicity these two valves could be made concentric. The thermodynamic analysis of the cycle has not been attempted. Actually the device has not been tried. It is in effect a relief valve, the escape from which is usefully employed later.



Pilot Injection System.

DESCRIPTION OF ADDITIONAL APPARATUS.

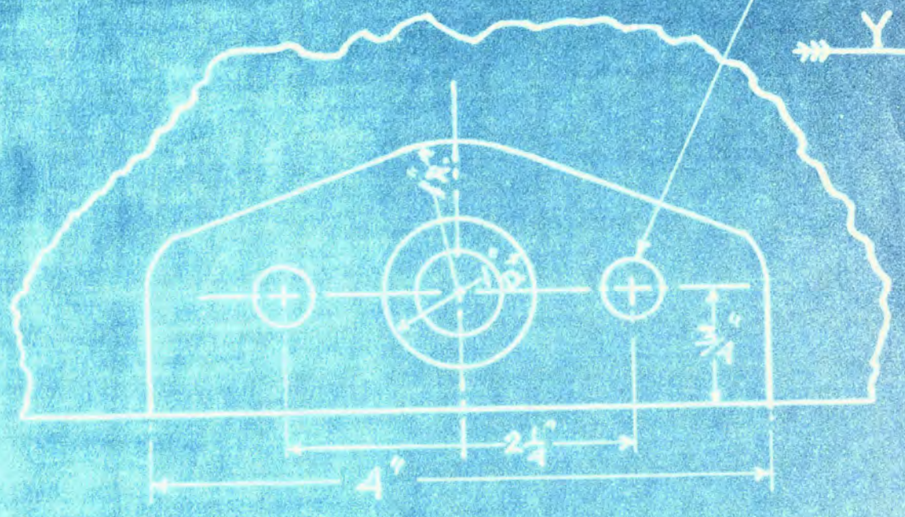
The Cylinder Head.

When the work on the delay period and "Diesel Knock" was decided upon, a new cylinder head had to be made with an additional hole for indicating purposes, while the original indicator hole was used for a second nozzle. Fig. (37) shows the new cylinder head designed by the author, which is a modification of the old one. The indicator hole and the new nozzle hole, are in a straight line in opposition to each other inside the combustion chamber.

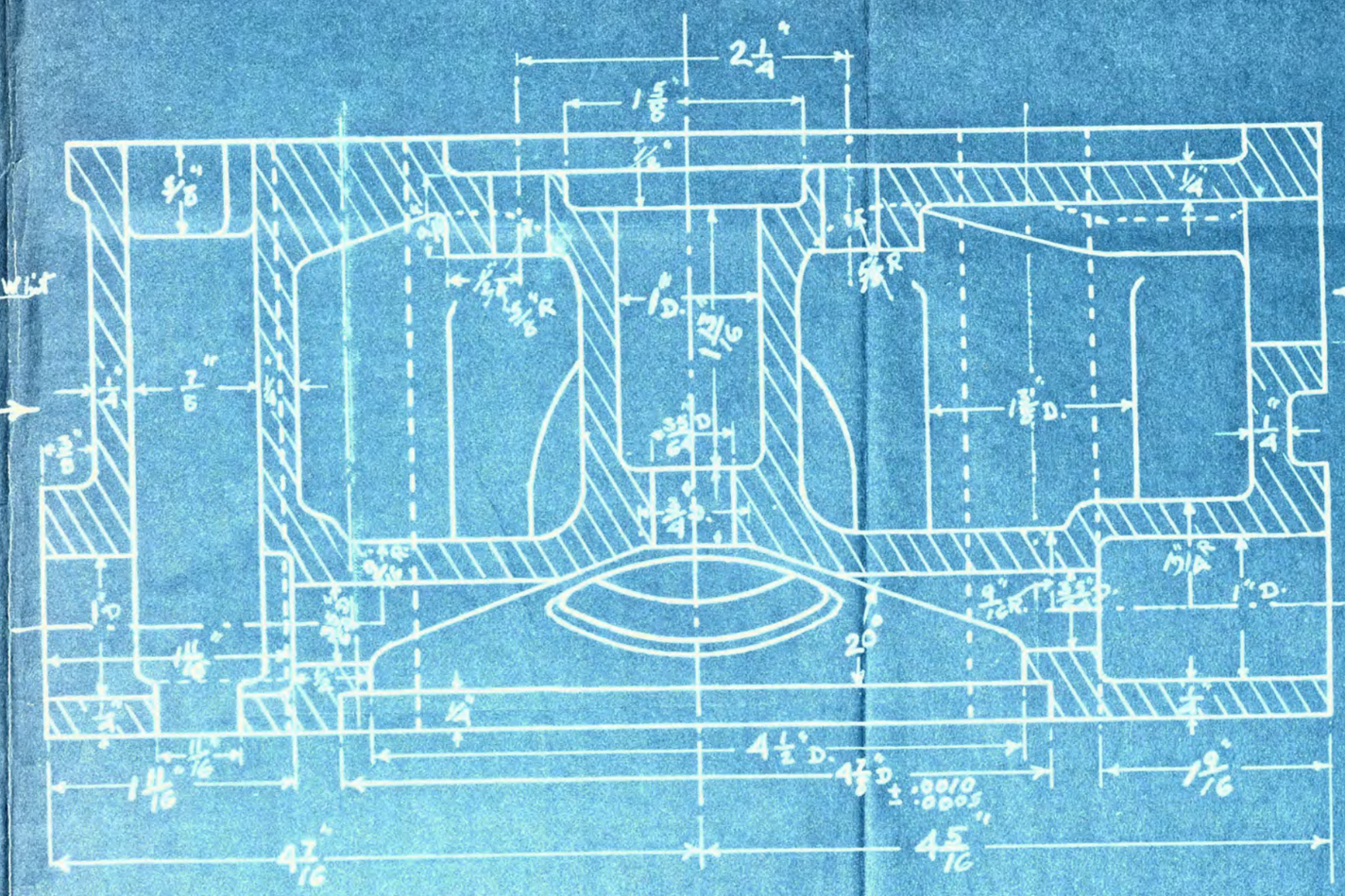
The pressure element of the indicator, which is 1 1/16" diameter had to be screwed into the cylinder head from between the two push rods. To allow of this being done the push rods had to be reduced in diameter near the pressure element of the indicator.

Pilot Injection System.

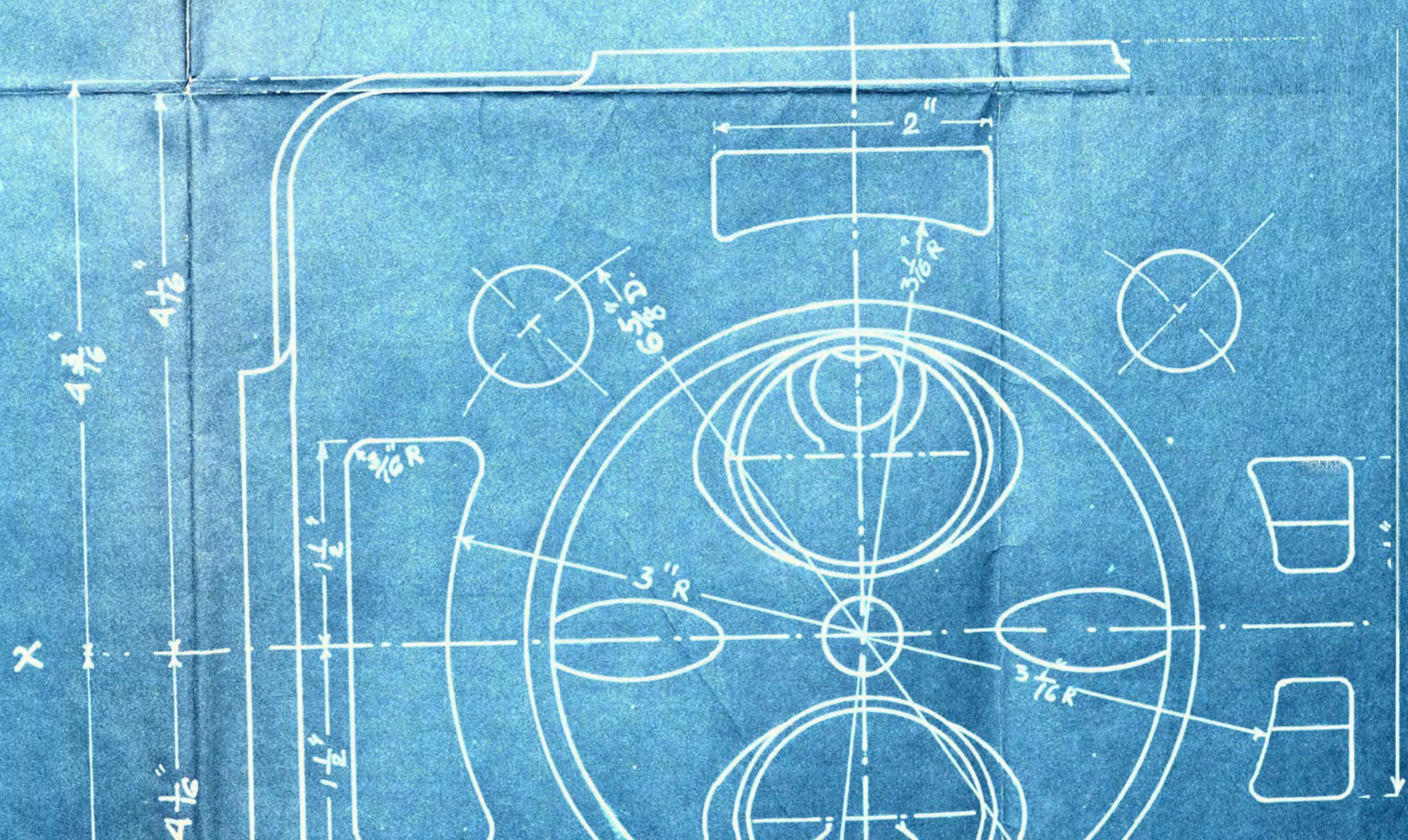
An additional Bosch pump was installed for the purpose of supplying the engine with a pilot charge. A nozzle, with one small hole, was fitted in the cylinder

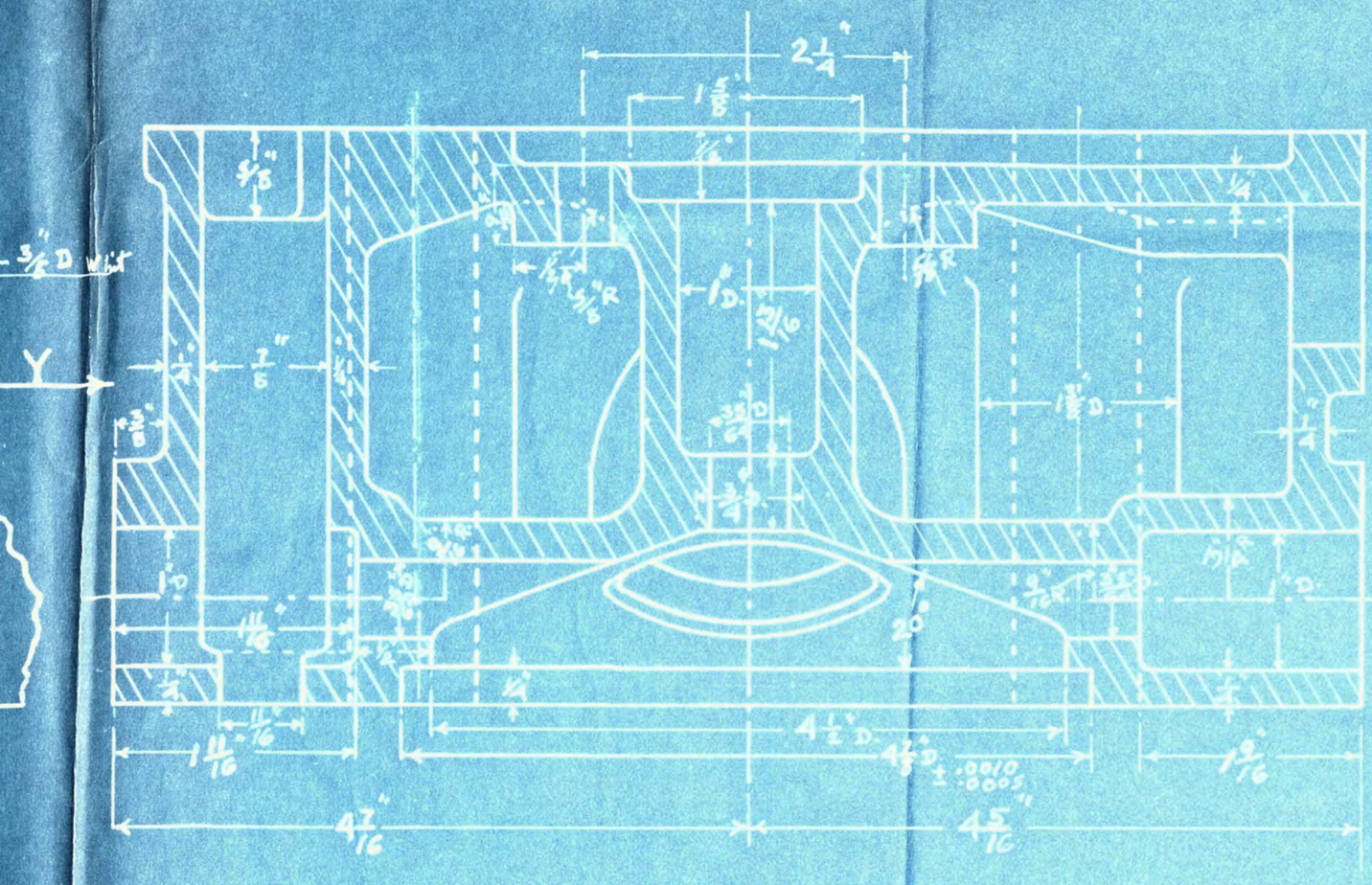


VIEW IN DIRECTION OF ARROW Y.

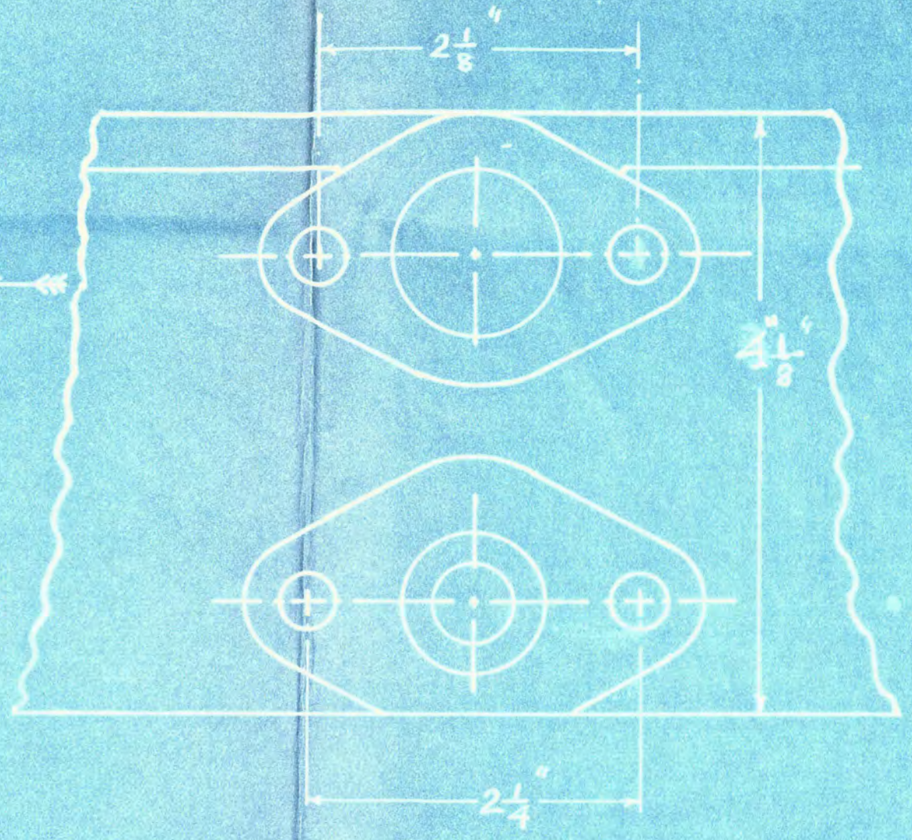


SECTION XX





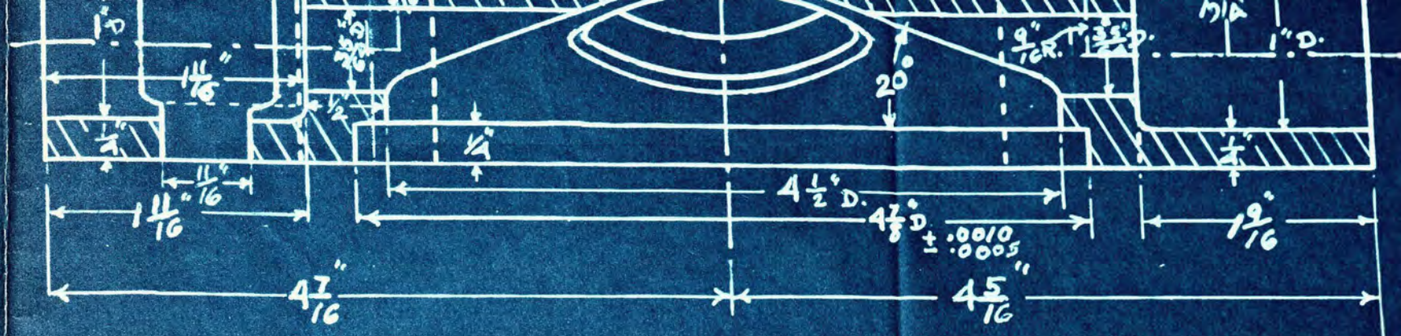
SECTION XX



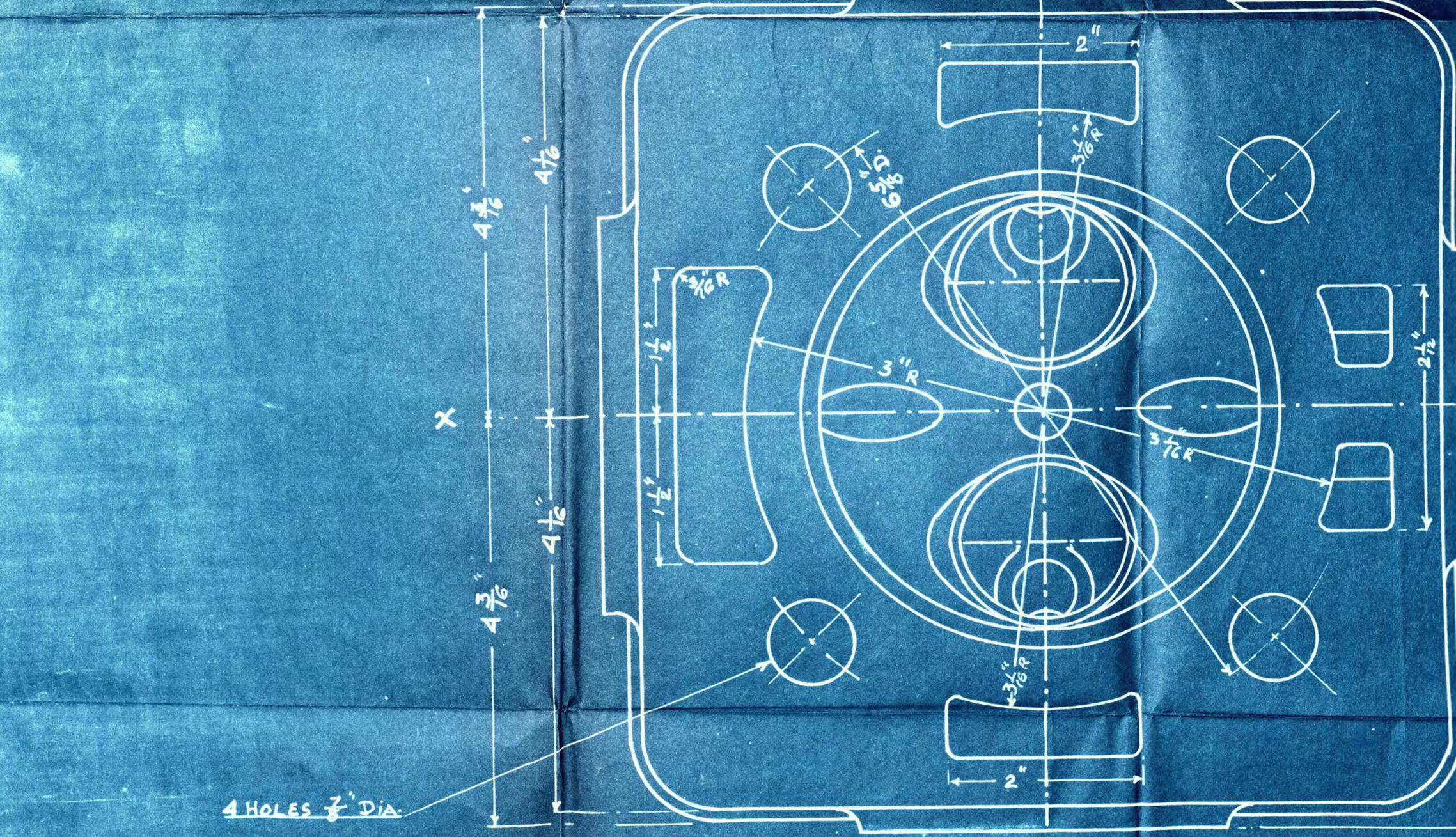
VIEW IN DIRECTION OF ARROW Z.



VIEW IN DIRECTION OF ARROW Y.

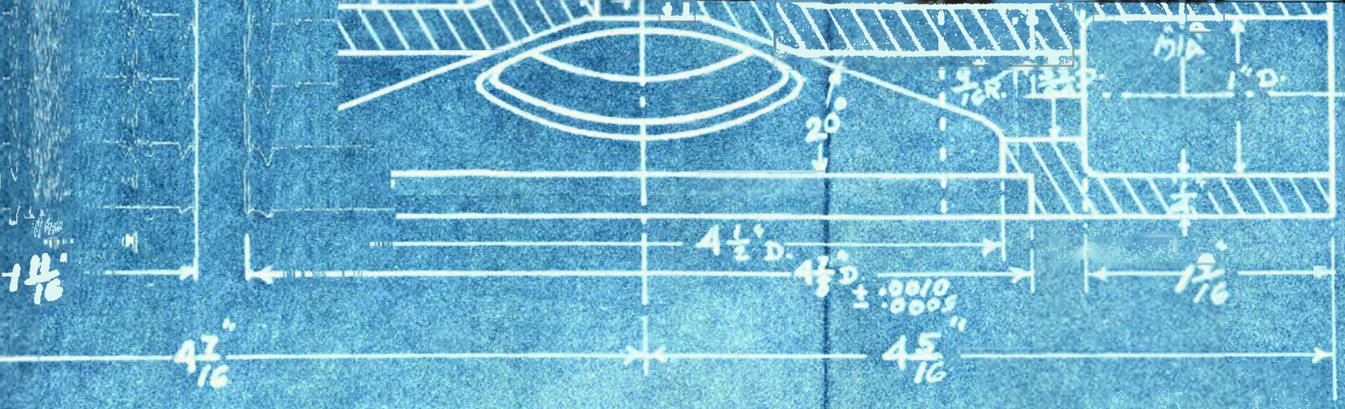


SECTION XX

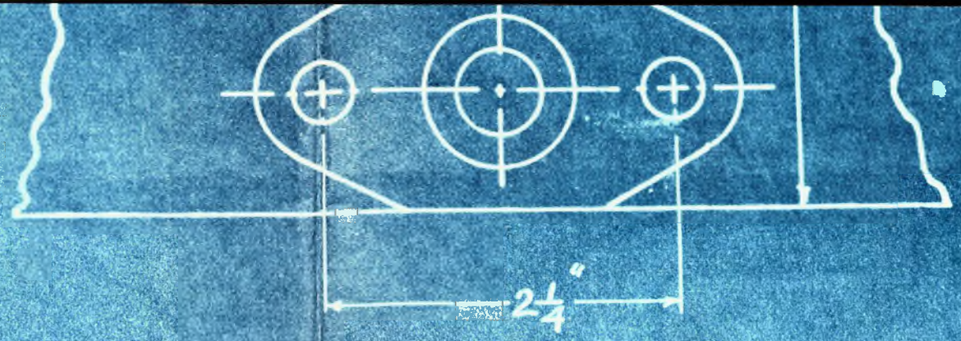


INVERTED PLAN.

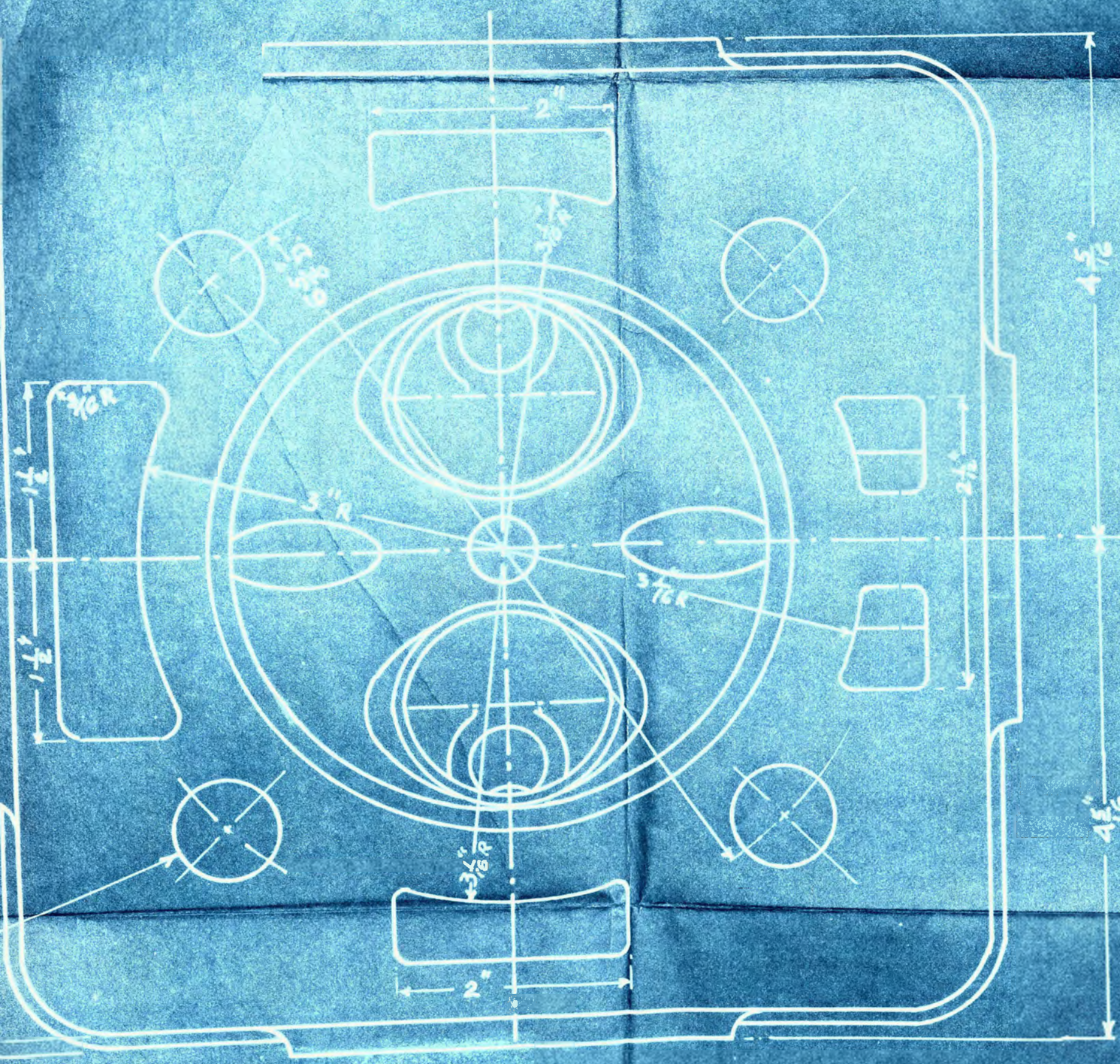
CYLINDER



SECTION XX



VIEW IN DIRECTION OF ARROW Z.



INVERTED PLAN.

FIG.(37)

CYLINDER HEAD HIGH SPEED DIESEL ENGINE

SCALE: FULL SIZE. D. JAFAR 1935.

head horizontally. The fuel was injected from this nozzle in a direction inclined to the normal to the cylinder, in fact in a direction having opposite direction to the air movement inside the combustion chamber.

The pump was chain driven by the engine. A gear wheel mechanism reduced the speed of the pump to half the engine speed, and could be operated by hand in order to vary the timing of injection. See photograph.

Mechanism for Varying the Injection Timing.

The mechanism was designed by the author for the purpose of obtaining a big range of variation in the time of injection of the inlet charge. Fig. (38) shows the mechanism designed.

The gear box was so arranged that its cover was extended and formed a circular segment (rather more than 180^{o} in arc), with slant teeth cut on the circular edge. A hand operated worm wheel, which was mounted on two small bearings directly under the gear box, engaged the slant teeth. By turning the worm wheel by hand, the whole gear box could be made to rotate round two bearings which carried the box; the axes of these bearings were in

1½" DIA GEAR WHEEL. 24 TEETH
16 PITCH.

2 - 2½ DIA. GEAR WHEEL

(+⅛" FOR OUTSIDE DIA. IN ALL THE GEAR WHEELS)

3" DIA GEAR WHEEL.

5/16 RAD.

3/8

2¼"

2 7/8" R.

3/8 SQUARE

X

¼ D.

FULL SIZE.

ELEVATION IN DIRECTION OF ARROW "N"

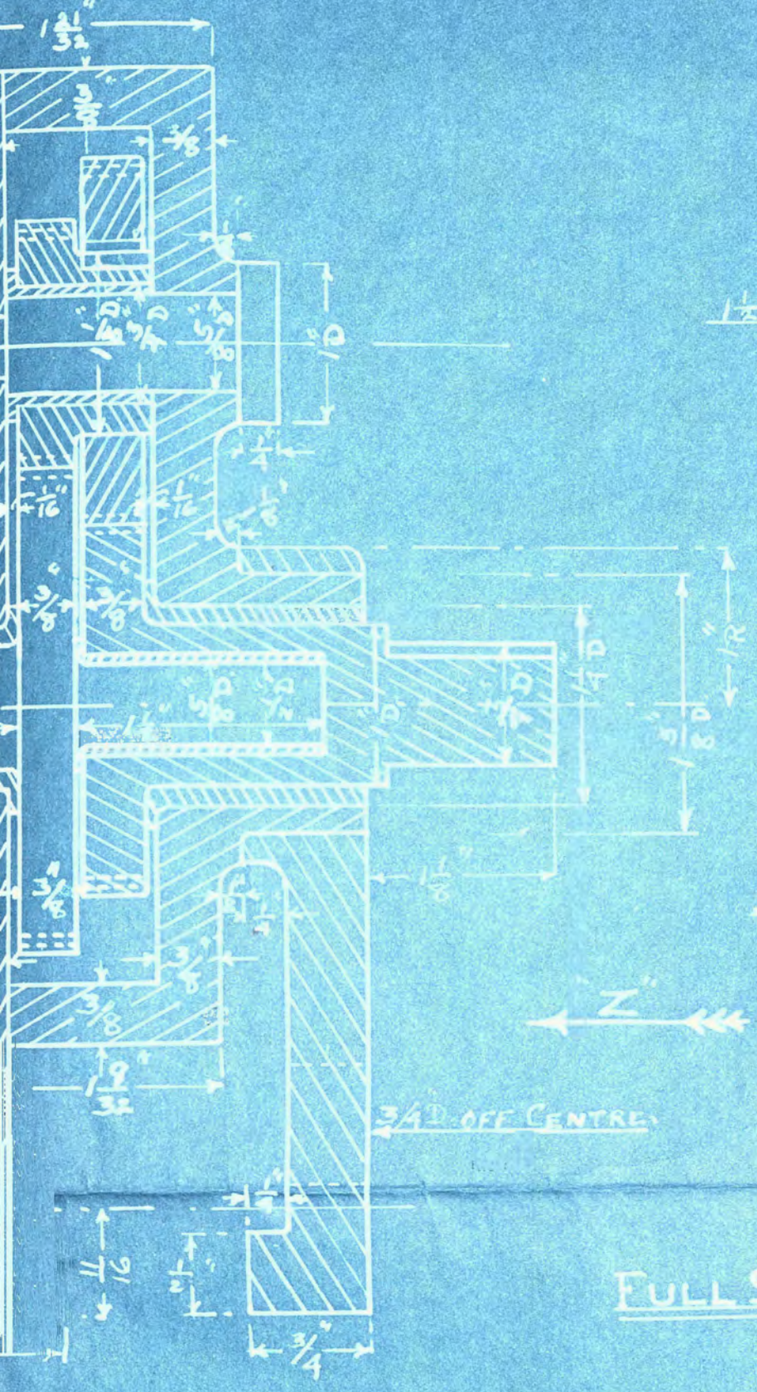
CROSS-SECTION Y-Y.

ELEVATION IN DIRECTION OF ARROW "Z"

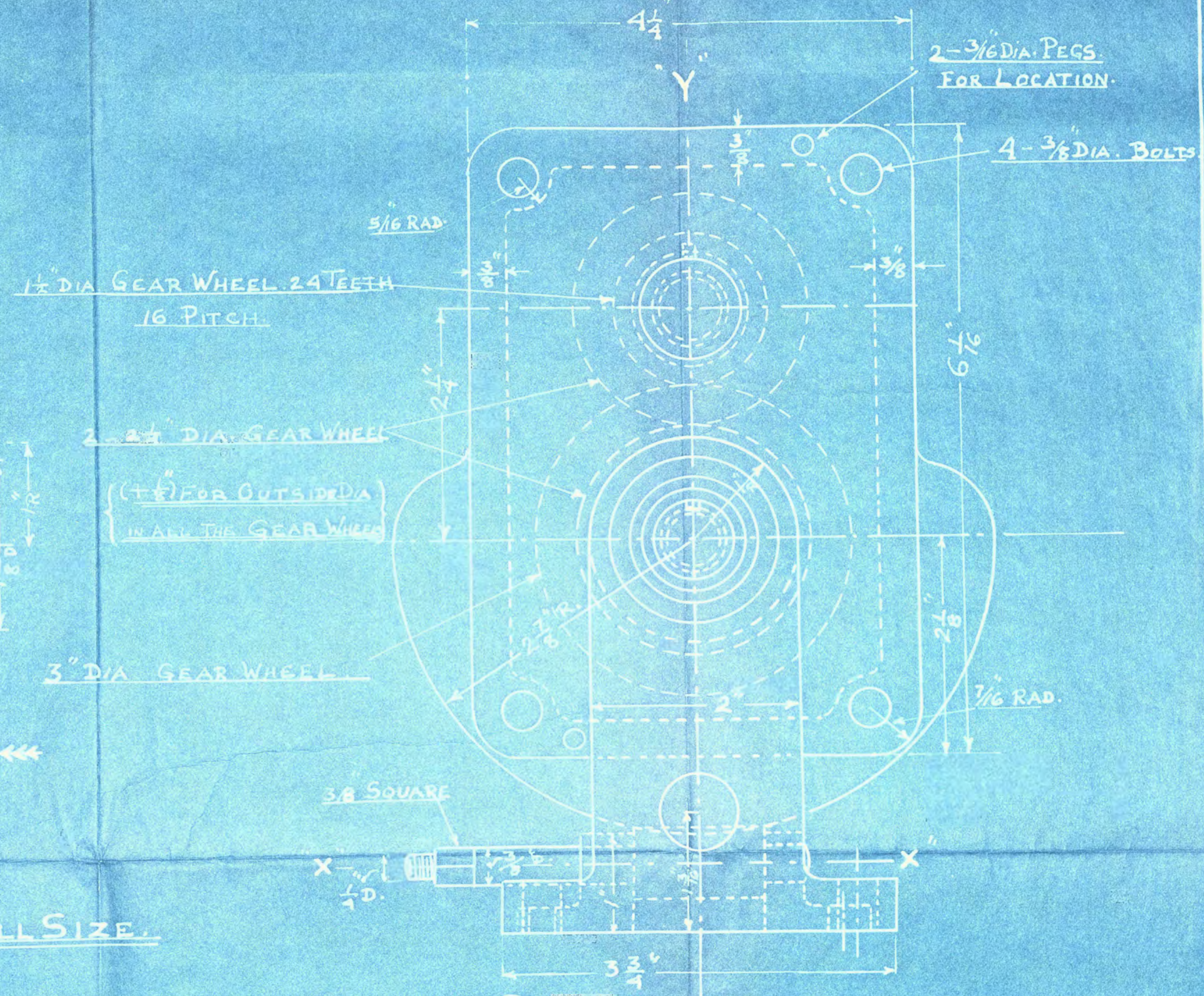
4-½ DIA. HOLES.

FIG. 38.

FUEL ADVANCING ARRANGEMENT FOR TANGYE OIL ENGINE.



CROSS-SECTION Y-Y.



FULL SIZE.

ELEVATION IN DIRECTION OF "Y" ARROW "Z"

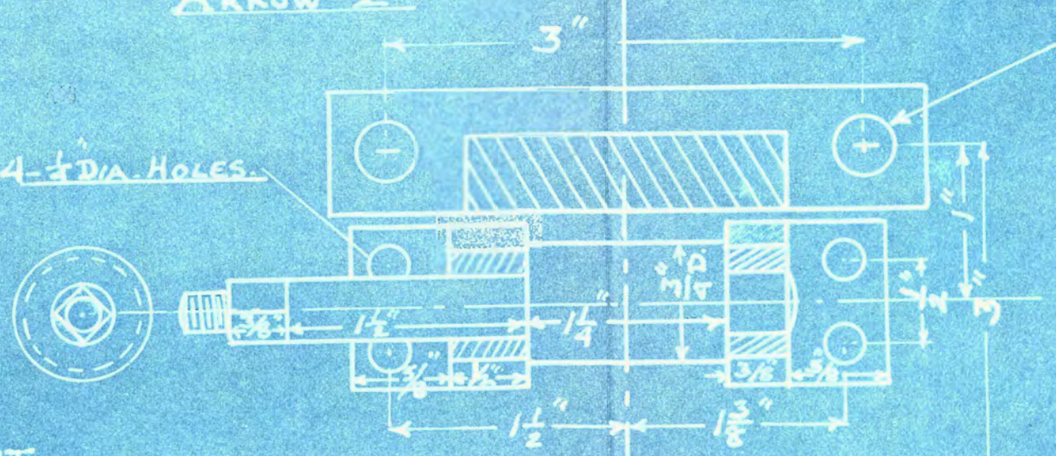


FIG. 38.

FUEL ADVANCING ARRANGEMENT FOR TANGYE OIL ENGINE.

B'HAM UNIVERSITY D.J. 1935

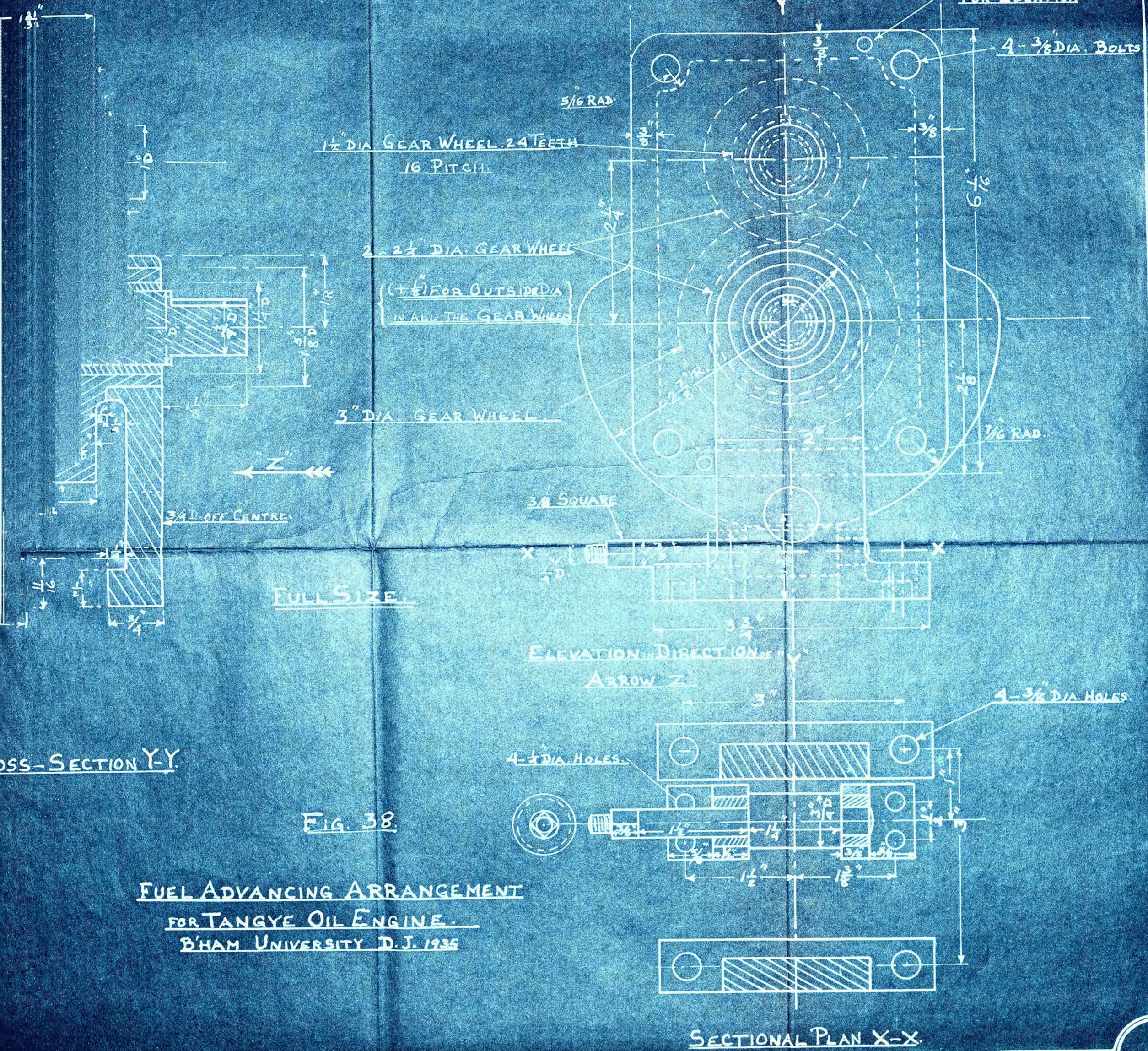
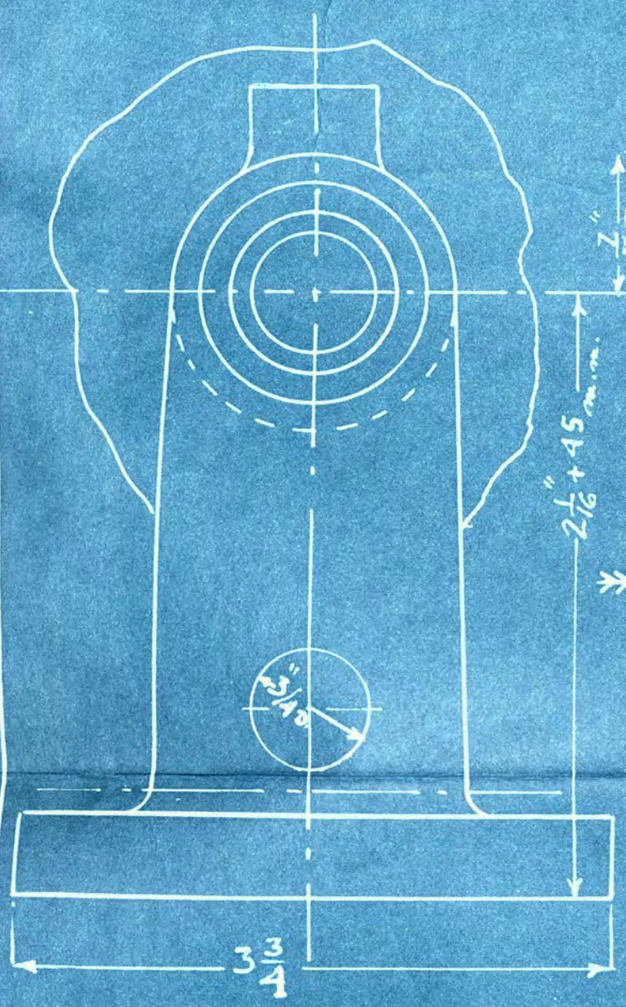


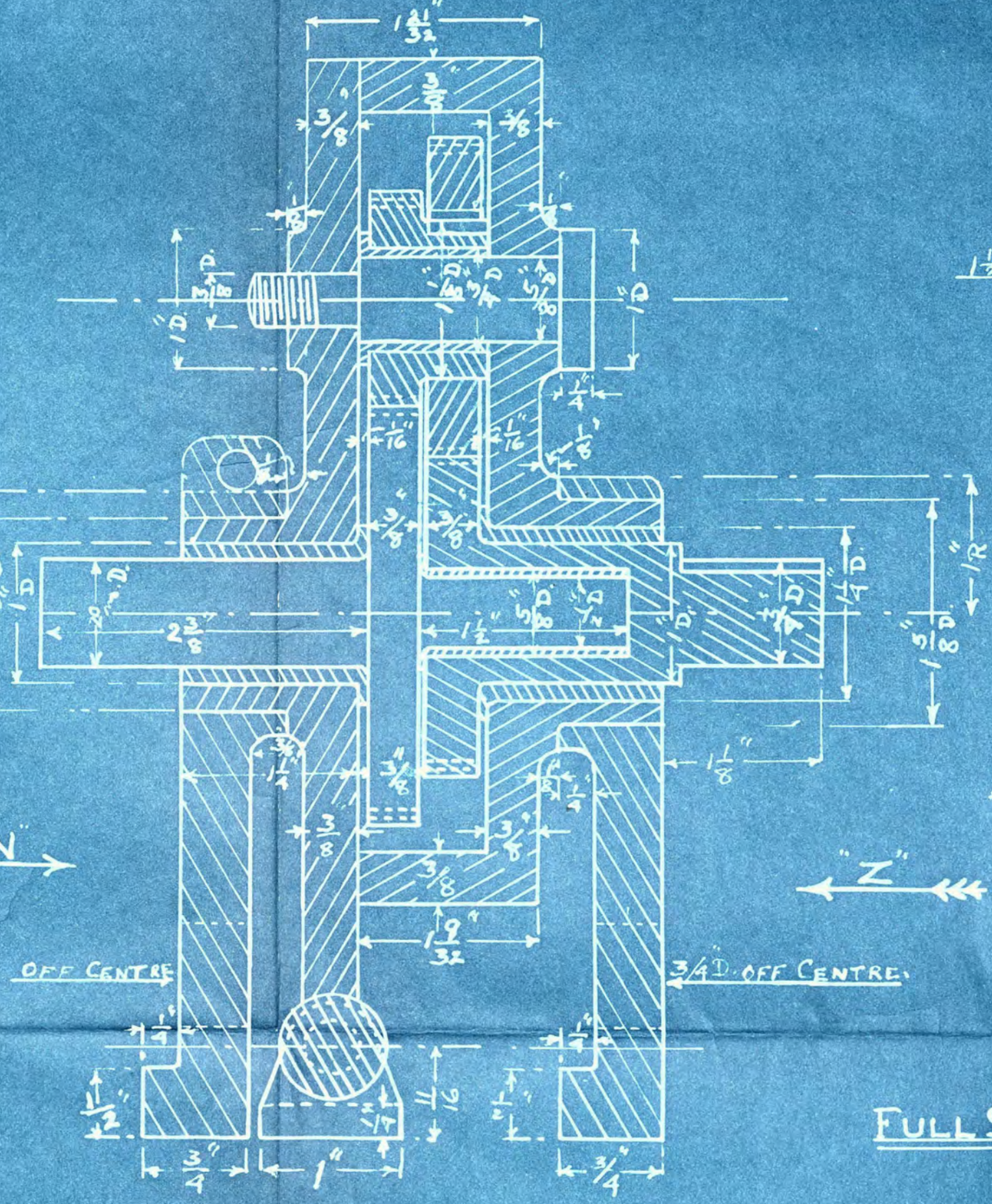
FIG. 38.

FUEL ADVANCING ARRANGEMENT FOR TANGYE OIL ENGINE.

B'HAM UNIVERSITY D.J. 1935



ELEVATION IN DIRECTION OF ARROW "N"



CROSS-SECTION Y-Y.

1 1/2" DIA. GEAR WHEEL. 24 TEETH
16 PITCH.

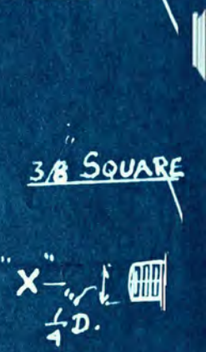
2 - 2 1/4" DIA. GEAR WHEEL

{(+1/8") FOR OUTSIDE DIA IN ALL THE GEAR WHEELS

3" DIA. GEAR WHEEL.

FULL SIZE.





ELEVATION
ARRO



FIG. 38.

FUEL ADVANCING ARRANGEMENT FOR TANGYE OIL ENGINE.

B'HAM UNIVERSITY D.J. 1935

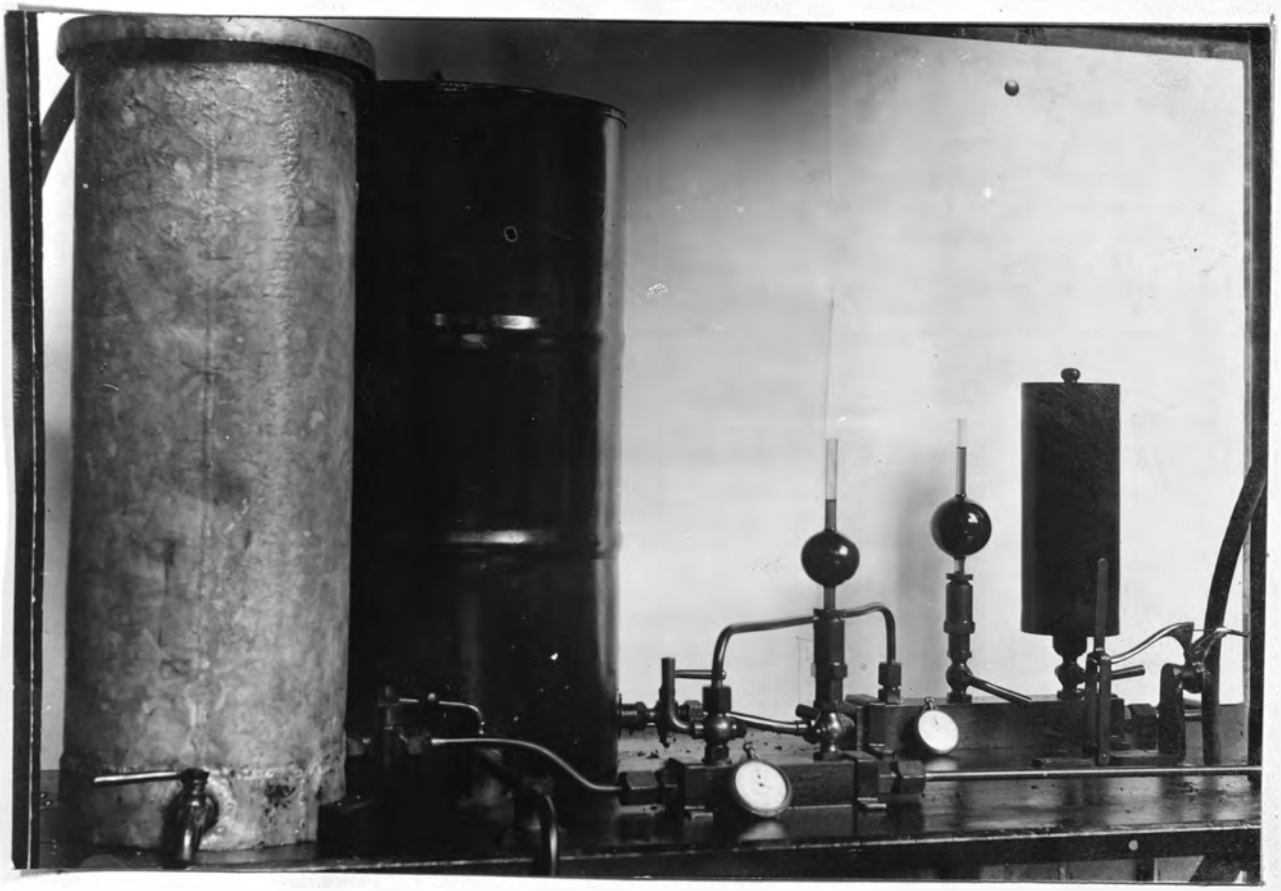
line with those of the two spindles, as can be easily seen from Fig. (38). In this way the timing of injection could be varied while the engine was running. A circular brass index, marked in degrees, was attached to the gear box cover, concentric with the centre of rotation of the gear box.

One of the two bearings, supporting the gear box, had a locking device. The gear box was made oil tight, and the lubricating oil was introduced from a small hole at the top.

Devices for Setting the Two Bosch Pumps.

The makers supplied the engine with a special hand operated lever and a ratchet arrangement. The governor and the fuel pump were connected to this lever. It was not a suitable mechanism for accurate setting of the fuel pump.

The author disconnected the lever from the pump and attached a special micrometer (made in the Mechanical Engineering Department), directly on the fuel pump ratchet only. It was arranged that this micrometer was mounted in such a way that its spindle could be reduced quickly in



Fuel Tanks and Fuel Measuring Bulbs.

the case of emergency. A locking device was also provided.

The other end of the ratchet of the fuel pump was spring loaded to take up the back-lash.

The governor had no control over the engine with the above arrangement.

Another micrometer of a similar type was fixed to the pilot Bosch fuel pump.

Fuel Supply to the Pilot Fuel Pump.

An additional fuel tank, small in size, was installed for the purpose of supplying the pilot injection system with different fuels. A calibrated glass bulb, small in size, was fitted (in the suction line to the pilot fuel pump), for fuel measurement purposes. A tube connection between the two fuel supply systems, with a quick action valve was installed. With this connection the pilot injection system could be supplied with fuel from the original tank and vice-versa.

A small valve was also fitted very near the pilot fuel pump, for the purpose of emptying the fuel pipes as well as the Bosch fuel filter, in short time. See photograph.

Injection Timing Indicator.

The two fuel atomizers (nozzles) used were of the Bosch type with a spring loaded valve. The valve lifts, when the pressure of the fuel reaches a certain value and allows the fuel to force its way through the hole or holes, in the nozzles, into the engine cylinder.

It is not much wrong to assume that the times at which the nozzle valve lifts and the fuel enters the cylinder of the engine coincide, i.e. the two phenomena are almost instantaneous. The author assumed that the time at which the atomizer valve lifted is the commencement of the fuel injection. Another method has been used extensively by many authors. This method included the pressure diagram of the fuel injected. The commencement of the pressure rise is taken as the timing of injection.

This method, in the author's opinion, is not very accurate. The fact is that the pressure rise of the fuel does not indicate the commencement of injection. An appreciable time may pass in order that the pressure may build up and force the nozzle valve off its seat. Also the time for the pressure wave to reach the nozzle

may introduce an error, which varies with the speed of the engine.

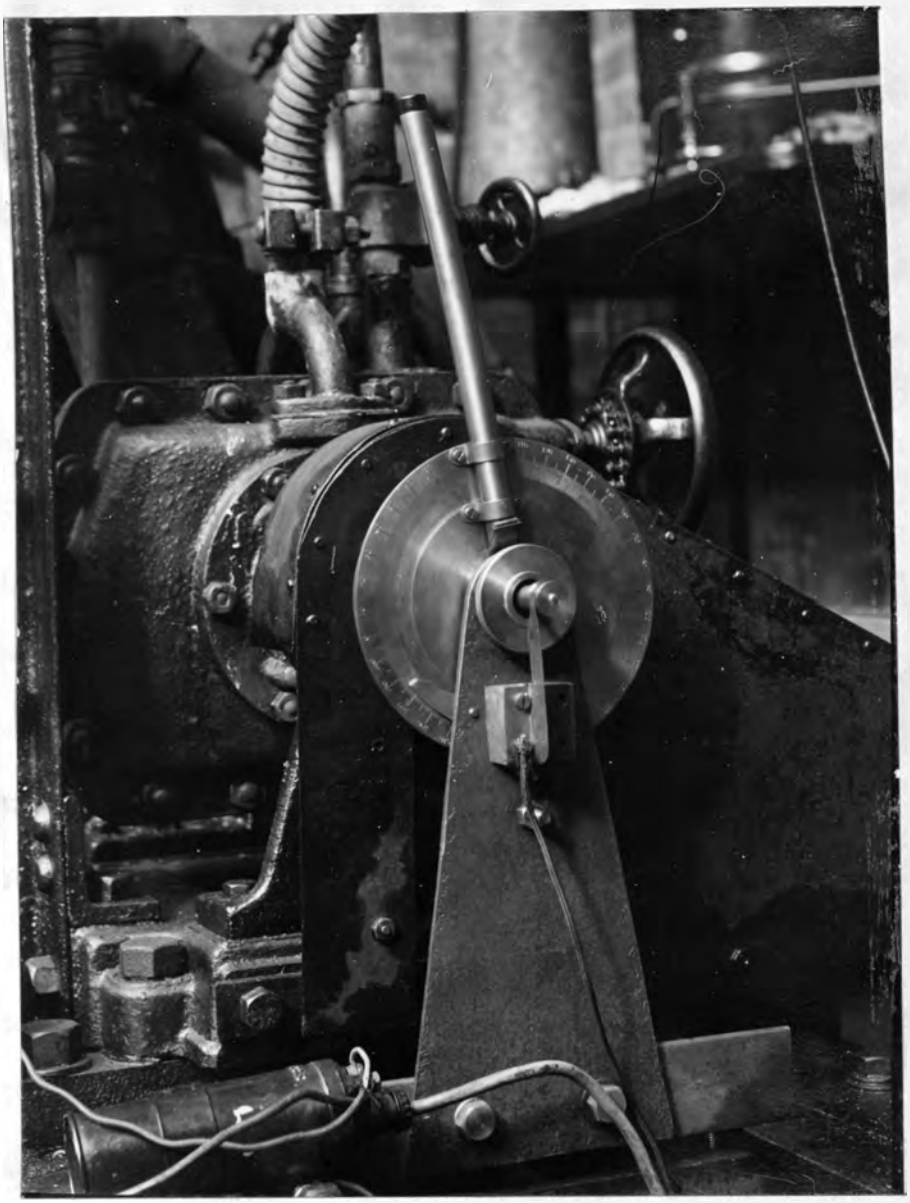
A petrol engine contact breaker was fitted on top of each nozzle. In each case a light needle was used, one end of it was in contact with the top end of the nozzle valve, and the other end was in contact with the insulated centre of the light rocker of the contact breaker. The contact points were adjustable.

The adjustment was made so that as soon as the nozzle valve lifted, the needle pushed the roller and broke the contact. The roller of the contact breaker was spring loaded and very light.

To start with, the primary circuit of a sparking coil was connected to the contact breaker. The secondary side (high tension) was connected to an insulated steel rod ending in a cone with a very small (ball shaped) end situated near a brass drum. This brass drum was about $5\frac{1}{2}$" outside diameter and was mounted on the end of the hydraulic brake shaft. The drum rotated at engine speed. A thin piece of paper was on the drum. The drum formed the earth side of the high tension circuit. A spark pierced the

paper at the time the nozzle valve lifted. For the top dead centre mark, the contact breaker was operated while the engine was at top dead centre. This method, however, was not accurate enough, due to the fact that the spark did not travel in a straight line, and the result was the spark pierced the paper in a number of places over a fairly wide range.

Professor Egerton and others,(18) have described a method for the measurement of the spark timing in a petrol engine. The author, on Professor Lees' advice, adopted the same principles. The primary circuit was connected as before. A small neon lamp was inserted inside an insulated material with a slot in it (1/32" in width). This neon lamp was fixed to the brake shaft inside the brass drum, and rotated with it at engine speed. A graduated disc, marked in degrees, with its centre concentric with that of the shaft, ~~This disc~~ formed a cover for the drum, and could be rotated by hand or locked at any position. Another slot was machined in this disc at the same distance from the centre as that of the insulated material containing the neon lamp.



Neon Tube Injection Timing Indicator.

A periscope was fixed to the disc, with a mirror fixed at 45° near the thin slot of the disc.

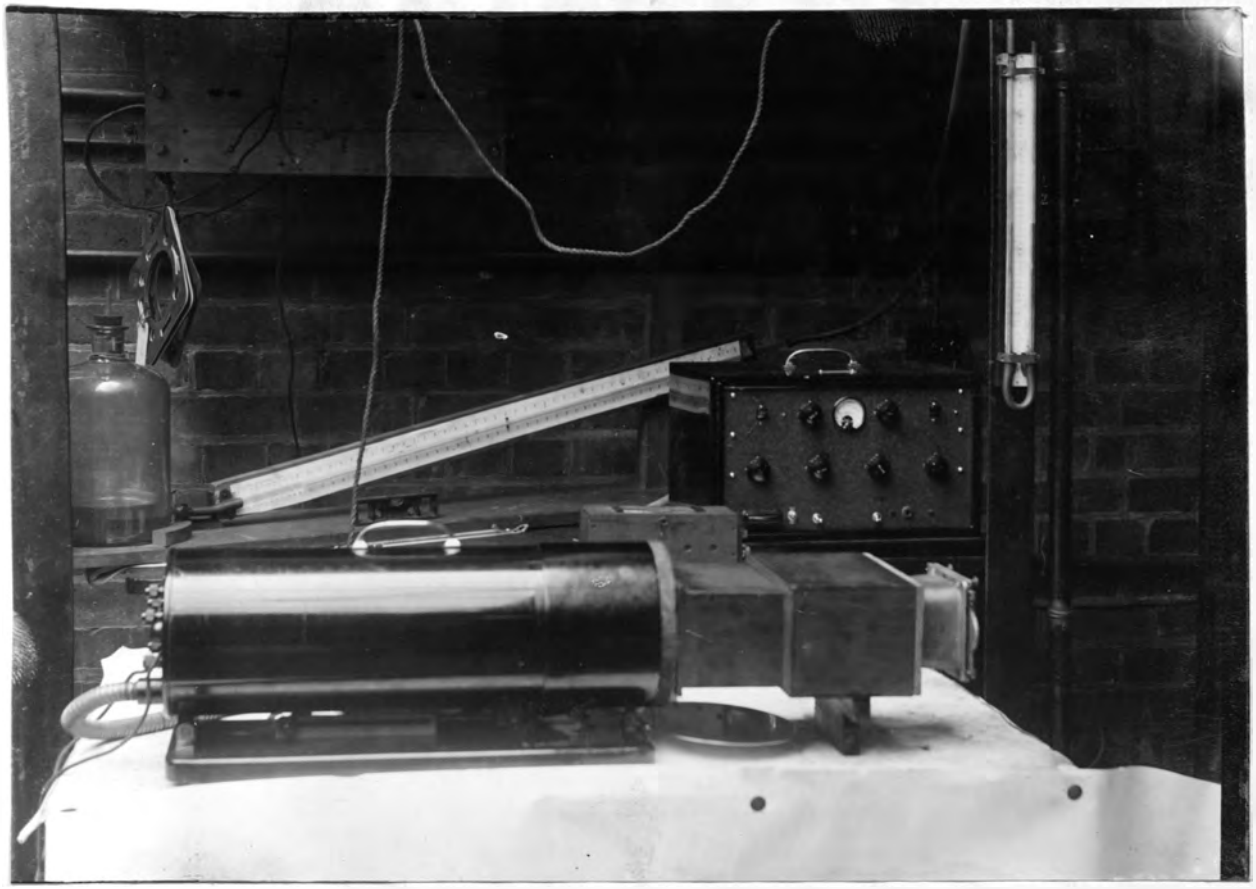
The secondary side (high tension) of the sparking coil was connected to the neon lamp. Every time the nozzle valve lifted, a flash passed through the neon lamp and was seen through the periscope. See photograph.

While the engine was running, the periscope with the disc was rotated by hand until the beginning of the flash appeared and the angle was then noted. The top dead centre was marked, when the engine was stationary, and the contact breaker was operated by hand. By this method, the timing of injection relative to top dead centre was noted.

This mechanism was synchronised with the contact breaker of the engine indicator (which is described later), and was found to be fairly accurate.

High Speed Engine Indicator.

The Engine Indicator used was a Metrovick-Dodds high speed indicator, supplied by Messrs. Metropolitan-Vickers Electrical Co. Ltd., of Manchester. The Indicator



Metro-Vick-Dodds Engine Indicator.

consisted of five separate units, namely, the exciter unit, cathode ray tube with a magnetic deflecting system, contact breaker, pressure element, and a pressure gauge for recording maximum pressures. The pressure element was made up of a pile of resistance material (composition unknown), the resistance of which varied with the pressure. The change of this resistance varied the current, though the magnetic deflecting coils, mounted on the cathode ray tube, produced a deflection in the ordinate direction as the pressure varied inside the cylinder. The pressure element consisted of a stainless steel body with a circular flat diaphragm 27/1000" thick integral with it. The whole element was water cooled.

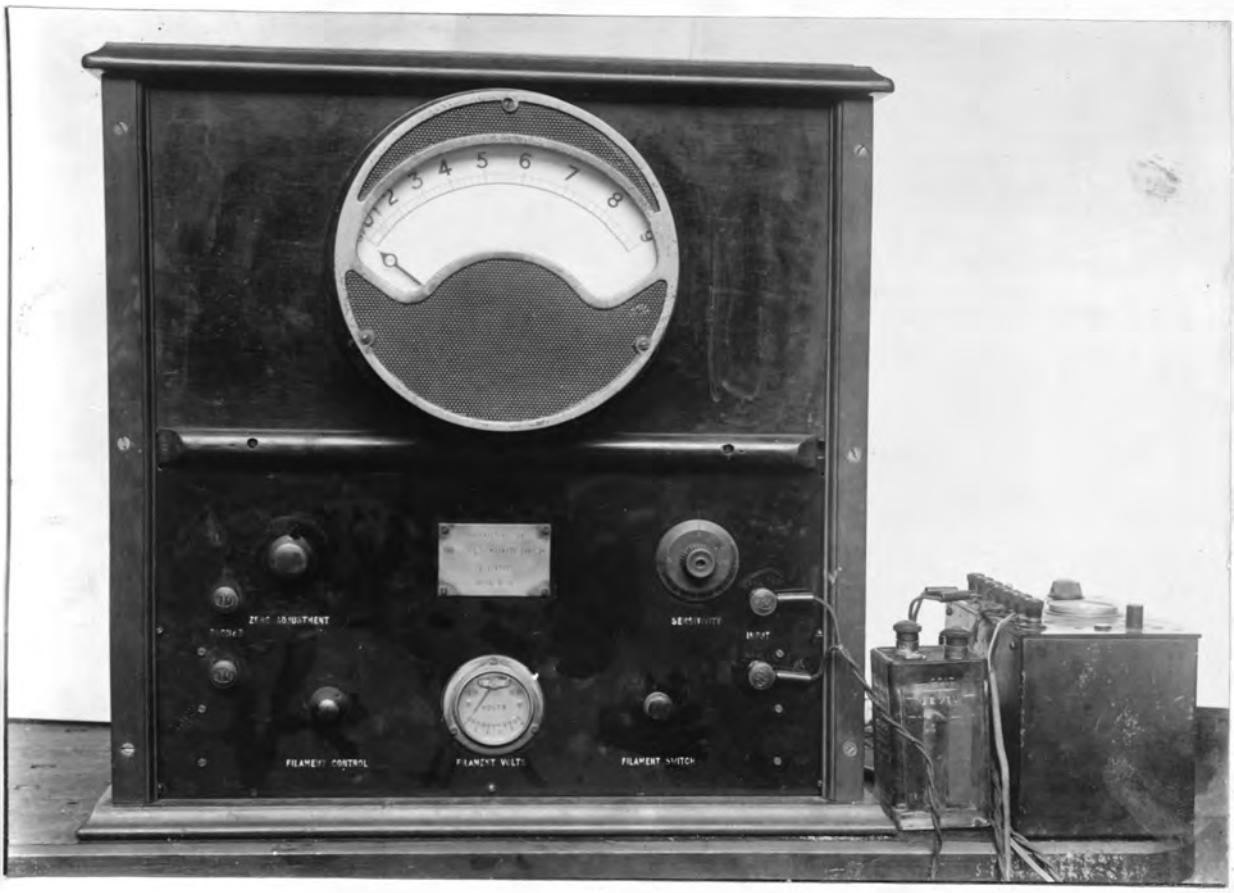
The base line or the horizontal sweep of the indicator diagram, which was formed at the screen of the cathode ray tube, was obtained electrostatically by charging up a variable condenser through a high resistance, and suddenly discharging this condenser by short circuiting it through a contact breaker mechanism driven from the end of the crank shaft of the Tangye engine. The base line of the indicator diagram represented crank angles. See photograph of the indicator.

A graduated disc, in degrees, was also part of the equipment of the contact breaker, for the purpose of measuring and locating different points on the indicator diagram.

Noise Measuring Instrument.

The photograph shews the instrument used for measuring the engine noise. It was a M.L. Noise Comparator, which was kindly loaned by the Institution of Automobile Engineers (Research Department). In order to obtain a measurement of the voltage generated by the sound waves, the voltage is amplified by means of a third, or rectifying valve, which is polarised by a grid bias battery, so as to work on the bottom bend of the anode characteristic curve. A milliammeter is inserted in the plate circuit of this last valve, and normally reads zero.

When an alternating voltage is impressed on the amplifier the resultant alternation in grid potential on the rectifying valve permits a current to flow in the plate circuit, and the reading on the milliammeter is a measure of the amplitude of the impressed voltage, and hence the



View of M.L. Noise Comparator.

amplitude of the noise is picked up by the receiver. The photograph shews the instrument with the microphone transformer. The microphone was placed very near the cylinder head. It was actually attached to the air inlet pipe (which in turn was connected to the engine by a rubber tube).

PART II.

SECTION A.

EFFECT OF SPEED ON DELAY PERIOD.

It was felt that some light might be thrown on the subject of the delay period and its behaviour when the speed of the engine was varied. Ricardo in the Proceedings of the I.A.E. 1930,(10) stated that "under any given conditions as to fuel, mixture, strength, compression pressure and temperature, the delay period occupies a substantially constant period of time irrespective of engine speed, and therefore, as the speed increases the injection timing must be advanced to compensate for the delay. Davies and Giffen, I.A.E. 1931,(8) found that the ignition lag or delay period varied in time with the speed. Dicksee has also given results which are not in complete agreement with those just mentioned.

In the author's opinion, the instruments used by many researchers, were not good enough, for the measurement of the delay period at high speeds. To clear up this point, with the aid of the instruments already described, some tests (to be described later), were carried out.

At first a number of tests were carried out for studying the effect of speed on the delay period, with the main fuel pump in operation only, and the smoothing plug fitted in the place of the pilot nozzle. The volumetric efficiency was not kept constant. A test was also carried out to find the effect of variation of volumetric efficiency on the delay period, with the engine speed kept constant. At the end of these tests a number of further tests were on the delay period, but with the volumetric efficiency kept constant at different speeds.

Method Used for Carrying out the Tests.

The engine speed was varied from about 1000 R.P.M. up to about 1800 R.P.M. The engine "eight-stroked" (when the timing of injection was fully retarded) at about 900 R.P.M. and therefore it was not desirable to go below 1000 R.P.M. A reason will be suggested for "eight-stroking" later on.

---

† At about 11° after T.D.C.

It was also felt that for the sake of comparison the quantity of the fuel injected per cycle must be kept constant. This was done by means of the micrometer on the fuel injection pump. The volumetric efficiency was regulated by means of the inclined water gauge, (for calculation see Appendix). Air inlet temperature was regulated by means of the air cooler, and was kept at about 20°, approximately. The water outlet temperature was kept at about 60°C. The brake was adjusted to give the required speed. Timing of injection was noted by means of the neon lamp indicator. The fuel consumption measurement was carried out using the revolution counter, so that the fuel per cycle could be calculated accurately.

Timing of Combustion Measurement.

The contact breaker of the engine indicator was rotated by hand, until the beginning of the portion representing the commencement of combustion coincided with the starting point of the indicator diagram, or, with any given datum line on the tube screen. The first method was preferred as the starting point, and is always constant. The angle on the graduated disc was noted. Top-dead-centre

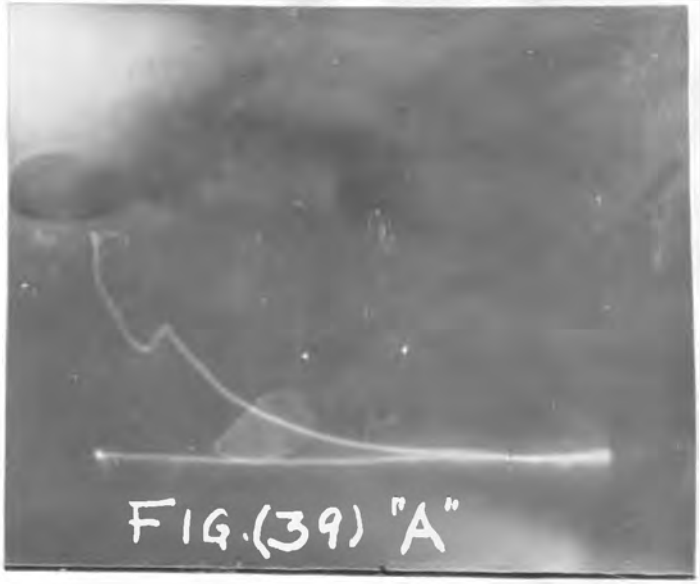


FIG.(39) "A"

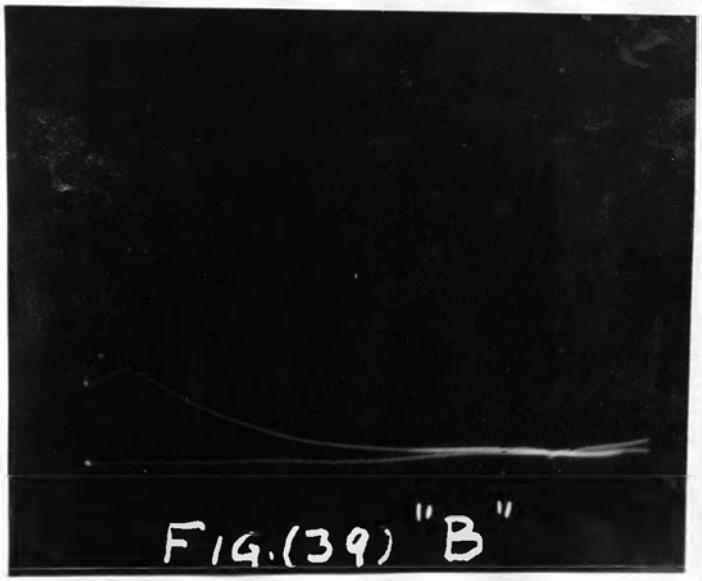


FIG.(39) "B"

was checked by means of the neon tube indicator, and therefore the timing of combustion in relation to top dead centre was used on the contact breaker disc, hence the delay period. Fig. (39) shews two photographs, A and B. The hump of combustion very near the commencement is shown by A. B shows the beginning of combustion at the starting point of the diagram. Injection timing was retarded.

TEST RESULTS.

I. Variable Volumetric Efficiency.

The timing of injection of the main pump was fixed in three different positions.

(i) Timing of Injection Retarded.

Fig. (40) shews a diagram when the timing of injection was at 8.75^{o} after top-dead-centre (average). Three curves are plotted in this Fig. (a) Delay period in seconds x 10^{-3} plotted against engine speed in revolutions per minute. (b) Volumetric efficiency against revolutions per minute, and (c) torque in lb.ft. plotted against revolutions per minute.

The delay period decreased from 1.2 x 10^{-3}

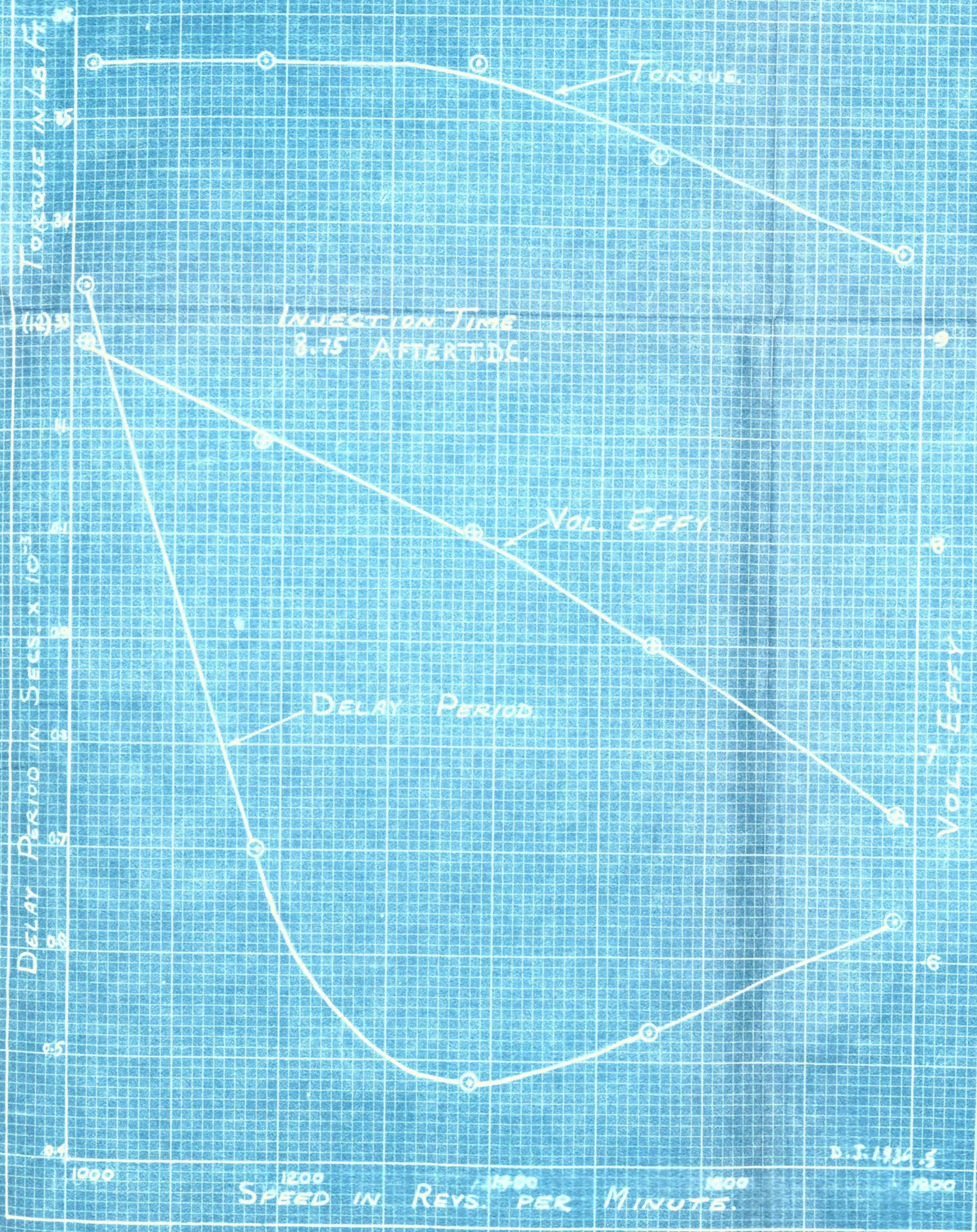

Torque.
Injection Time
8.75 After T.D.C.
Vol. Effy.
Delay Period.
Delay Period in Secs. x
Torque in Lbs. Ft.
Vol. Effy.
1000
1200
1400
1600
1800
Speed in Revs. per Minute.

Fig. 40.

to 0.48 x 10^{-3} seconds, as the speed of the engine increased from nearly 1015 R.P.M. to 1380 R.P.M. After this, the delay period increased as the engine speed increased; at 1780 R.P.M. the delay period was 0.64 x 10^{-3} seconds. This increase, however, in the delay period, as will be explained later, may be due to a decrease in volumetric efficiency at high speeds. The curve for the volumetric efficiency shews a steady drop in the value of this, as the speed of the engine increases. The volumetric efficiency at 1015 R.P.M. was 0.895, while at 1780 R.P.M. it fell to 0.67.

The torque curve shews a steady drop from 35.6 lb.ft. at 1380 R.P.M. to 33.8 lb.ft. at 1780 R.P.M. This drop must be largely due to drop in volumetric efficiency.

I. (ii) Injection Timing near Top-Dead-Centre.

Fig. (41) shews the results when the injection timing was at 0.87° after top-dead-centre (average). The curves in this fig.,as before, shew delay period in seconds x 10^{-3}, volumetric efficiency and torque in lb.ft. against speed in R.P.M.

The delay period, as before, decreased as the

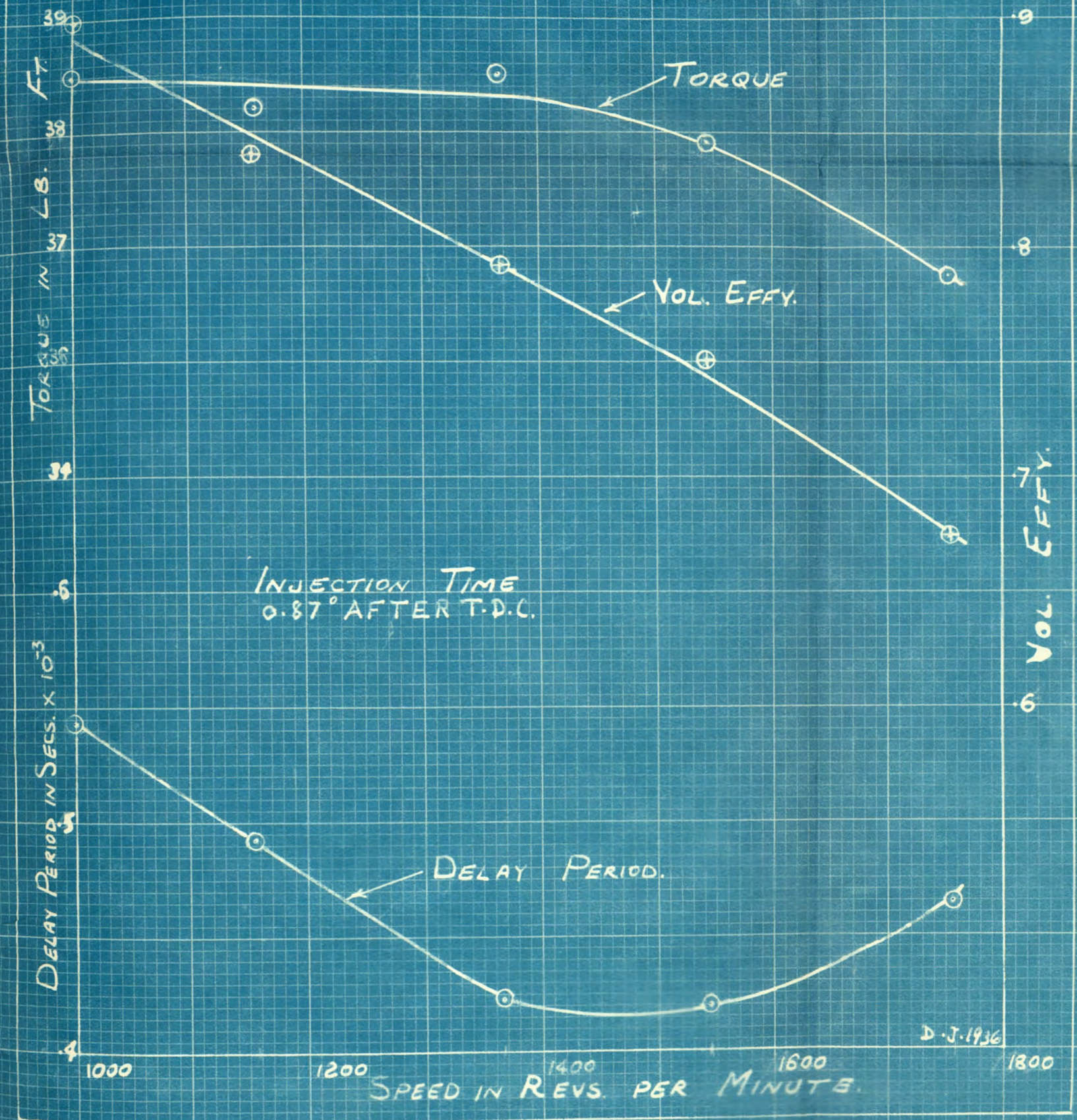

TORQUE IN LB. FT.
39
38
37
36
35
TORQUE
VOL. EFFY.
INJECTION TIME
0.87° AFTER T.D.C.
DELAY PERIOD IN SECS. X 10⁻³
.5
.4
DELAY PERIOD.
.9
.8
.7
.6
VOL. EFFY.
D.J.1936
1000
1200
1400
1600
1800
SPEED IN REVS. PER MINUTE.

FIG. 41.

speed of the engine increased to about 1400 R.P.M., and after this increased as the engine speed increased. The volumetric efficiency decreased throughout as the speed increased. The torque started dropping when the engine speed increased above 1368 R.P.M. A few of the results are given below:-

R.P.M.	D.P. Seconds x 10^{-3}	Vol. Eff.	Torque in lbs.ft.
1000	0.542	0.9	40.8
1368	0.422	0.795	40.8
1760	0.466	0.675	35.6

D.P. represents delay period.

I. (iii) Injection-Timing Advanced.

In Fig. (42) the injection timing for the results plotted was 5.75^{o} (average) before top-dead-centre. The fig. shows three curves. The abscissae represent speed of the engine in R.P.M., the Ordinates represent the delay period in seconds x 10^{-3}, volumetric efficiency and torque in lb.ft. The curve for the delay period starts from the highest point at 1000 R.P.M. and slopes down rapidly to a minimum at about 1600 R.P.M. Then it increases at the higher speeds. The volumetric efficiency (as before)

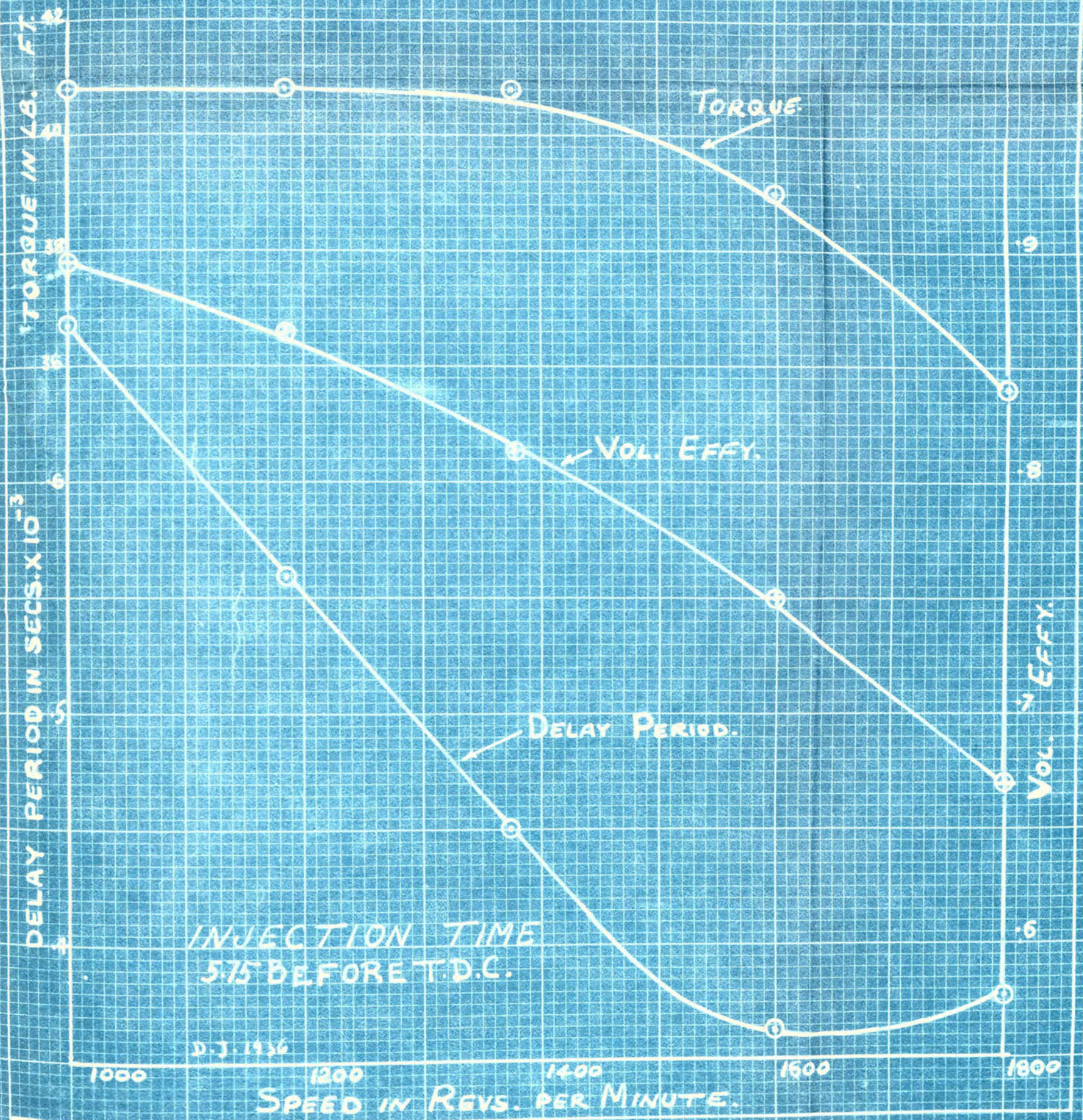

TORQUE IN LB. FT.
DELAY PERIOD IN SECS. X 10⁻³
VOL. EFFY.
TORQUE
VOL. EFFY.
DELAY PERIOD.
INJECTION TIME
5.75° BEFORE T.D.C.
D.J. 1936
1000
1200
1400
1600
1800
SPEED IN REVS. PER MINUTE.

FIG. 42.

dropped steadily as the engine speed increased, while the torque only started decreasing when the speed increased above 1378 R.P.M. The following are a few of the test results:-

R.P.M.	D.P. in Secs. x 10^{-3}.	Vol. Eff.	Torque in lb.ft.
1000	0.666	0.895	38.5
1600	0.362	0.75	37.9
1800	0.376	0.67	36.7

II. Effect of Speed on Delay Period with Constant Volumetric Efficiency.

To study the effect of speed on delay period only, other variables must be kept very nearly constant. Volumetric efficiency has a big influence on the temperature and pressure of the compressed air inside the engine cylinder. It also affects the speed of the inlet air, which may in turn affect the swirl at the time of combustion. A test was carried out to discover the effect of variation of volumetric efficiency on delay period. The speed of the engine was kept constant at about 1200 R.P.M. The quantity of fuel injected per cycle was kept constant by means of the micrometer on the fuel pump.

The brake was so adjusted as to keep the speed of the engine constant. The temperature of the inlet air was kept at about 20 ^{o}C,, while the outlet temperature of the cooling water was about 60 ^{o}C. The volumetric efficiency was varied by throttling down (when low values were required) and supernharging (at high values of volumetric efficiency). Injection timing was kept at 2.5 ^{o} after top-dead centre.

Fig. (43) shews test results. There are two curves shewing the relationship between delay period in secs. x 10^{-3} and torque in lb.ft. as against volumetric efficiency. The curve for the torque starts from 38.2 lb.ft. (this being the highest at 1.025 volumetric efficiency) and slopes down gradually until it reaches 36.7 lb.ft. at 0.663 volumetric efficiency. Then it slopes rapidly to the value of 31.45 lb.ft. at 0.538 volumetric efficiency. After this point it was very difficult to maintain the speed.

The delay period curve starts from the value of 0.833 x 10^{-3} seconds at 0.538 volumetric efficiency,

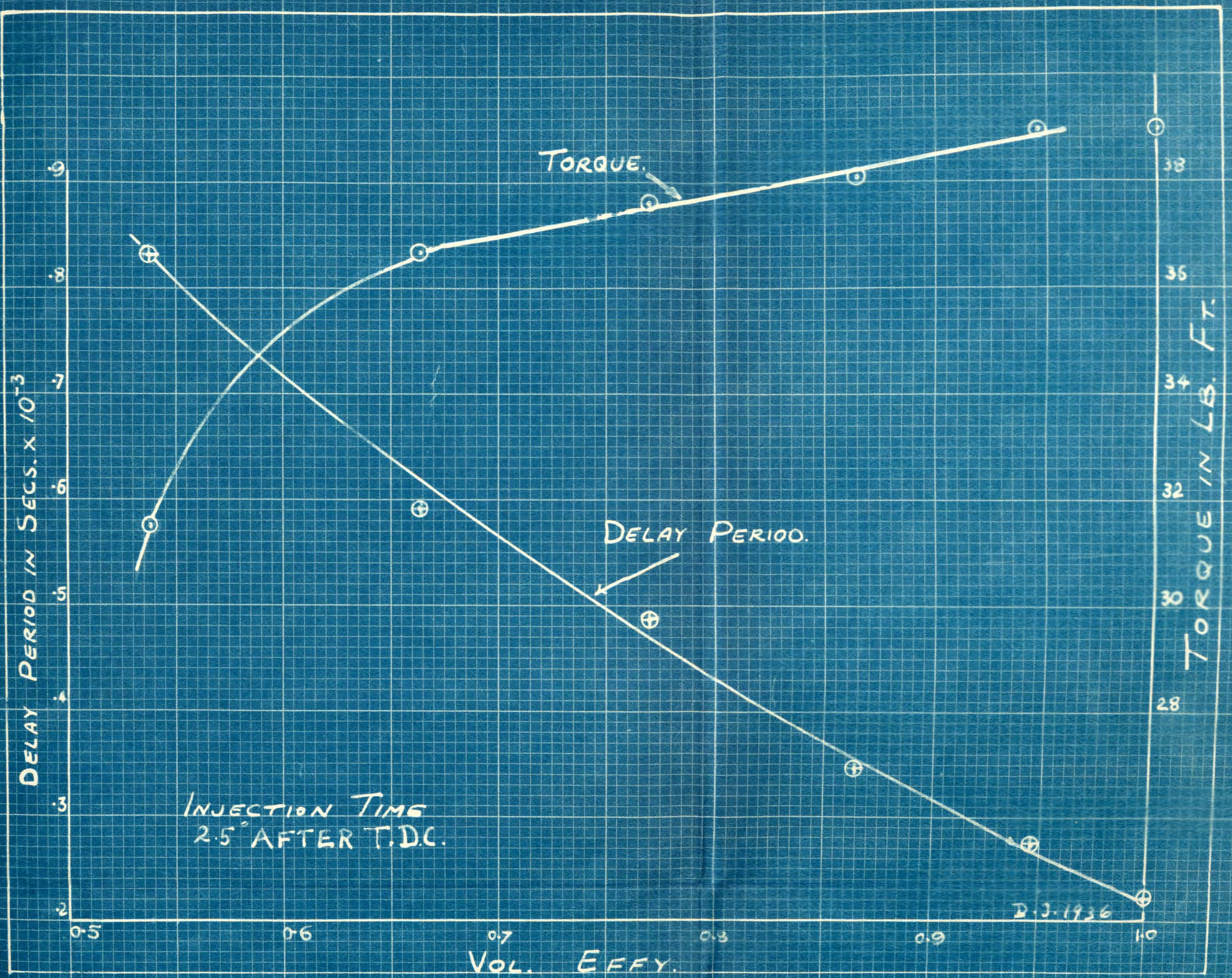

TORQUE.
DELAY PERIOD.
INJECTION TIME
2.5° AFTER T.D.C.
DELAY PERIOD IN SECS. X 10^{-3}
TORQUE IN LBS. FT.
·9
·8
·7
·6
·5
·4
·3
·2
38
36
34
32
30
28
0·5
0·6
0·7
0·8
0·9
1·0
VOL. EFFY.
D.J. 1936

FIG. 43.

and drops down steadily until it reaches the value of 0.208×10^{-3} seconds at 1.025 volumetric efficiency. This curve, however, as will be discussed later, may explain the reason for the slight increase in the value of the delay period at very high speeds in tests already described.

It was therefore decided to supercharge the engine at high speeds in order that the volume of air used by the engine per cycle could be kept very nearly constant. The inclined water gauge was used for regulating the quantity of air. The inclination of the original water gauge was 8 : 1. This had to be replaced by one of 5 : 1 inclination, as the volume of air down at 1800 R.P.M. could not be measured by the original water gauge. For calculation and graphs see Appendix. The air inlet temperature was kept about 20^{o} C. by means of the air cooler. The speed was regulated from the brake. The fuel was set to give constant quantity per cycle. The delay period was measured as described before. The outlet temperature of the cooling water was 60^{o} C.

TEST RESULTS.

The tests were carried out for three different values of injection timing. Fig. (44) contains three curves shewing the relationship between engine speed (R.P.M.) and delay period in seconds x 10^{-3} for the three different injection timings, viz., 10.25^{o}, 0.5^{o} after top-dead-centre, and 5.25^{o} before top-dead-centre; at constant volumetric efficiency. As can be seen from the figure the three curves shew a decrease in the value of delay in secs. x 10^{-3}, as the engine speed increases. The curve for 10.25^{o} after top-dead-centre forms almost a straight line, while the other two curves tend to bend upwards at very high speeds. But in all cases they decrease as the speed increases, and the variation in value of the delay period is very marked. The following are some of the results:-

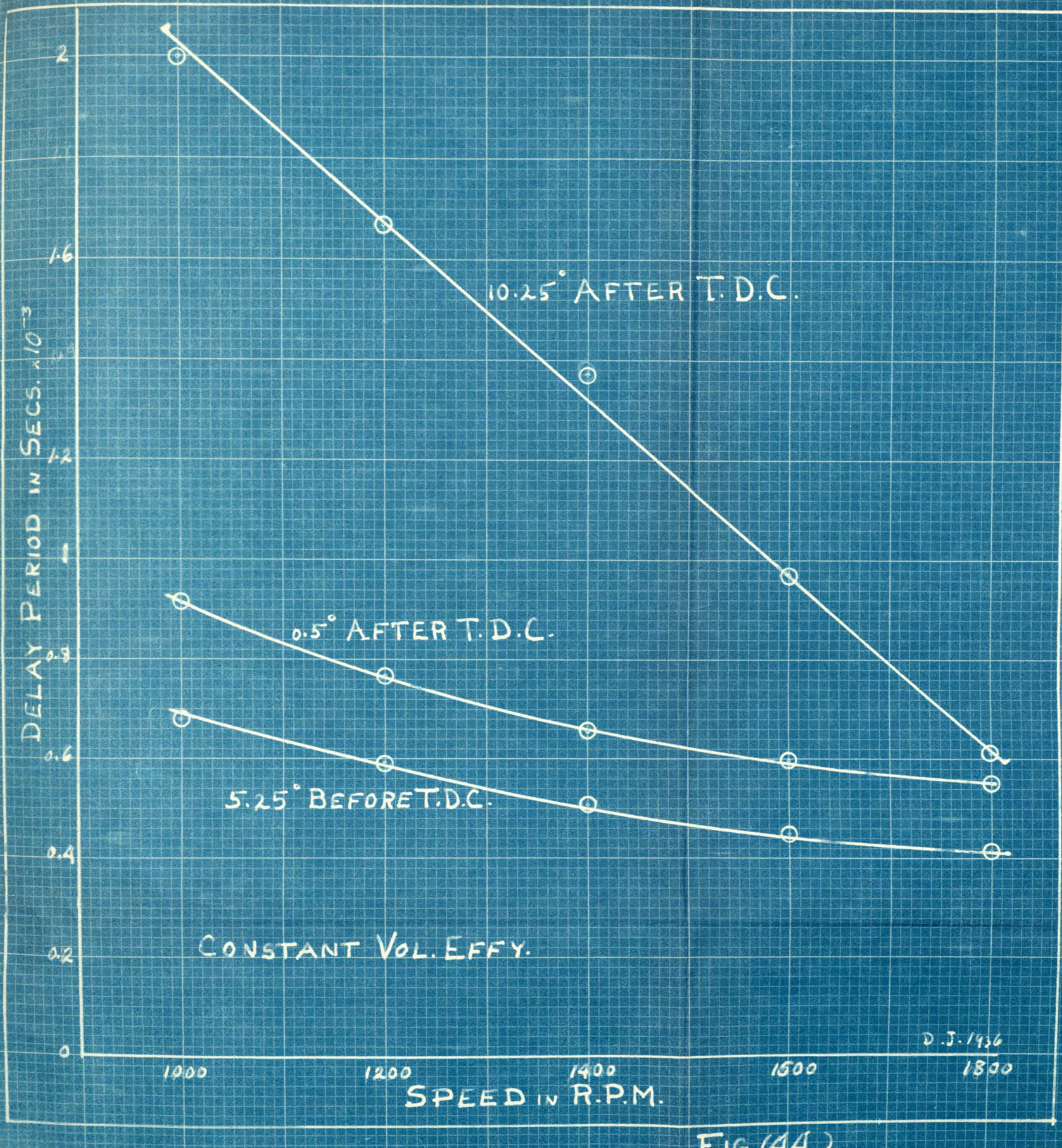

DELAY PERIOD IN SECS. x 10⁻³
2
1.6
1.2
1
0.8
0.6
0.4
0.2
0
10.25° AFTER T.D.C.
0.5° AFTER T.D.C.
5.25° BEFORE T.D.C.
CONSTANT VOL. EFFY.
D.J. 1936
1000
1200
1400
1500
1800
SPEED IN R.P.M.

FIG. (44)

Injection Timing.	Speed.	Delay Period in Secs.
10.25^{o} After T.D.C.	1000	2×10^{-3}
" " "	1800	0.605×10^{-3}
0.5^{o} " "	1000	0.915×10^{-3}
" " "	1800	0.55×10^{-3}
55.25^{o} before T.D.C.	1000	0.680×10^{-3}
" " "	1800	0.43×10^{-3}

DISCUSSION OF RESULTS.

A general survey of the test results described in Section "A", Part II of this thesis, indicates that the delay period is not constant in time at different engine speeds. When the volumetric efficiency was allowed to vary, there was a slight increase in the delay period at very high speeds. See Figs. (40, 41, 42). This increase was no doubt due in a great measure to the fall in volumetric efficiency. The test described, in which the volumetric efficiency was varied, while speed and quantity of fuel injected per cycle were constant, supports the above statement. Test results shown in Fig. (44), in which the volumetric efficiency was constant and the speed

varied, give another demonstration that the delay period is not constant in time, for the delay period decreased as the engine speed increased, even at very high speeds.

The author submits the following views on the subject:-

For given conditions of temperature, pressure, and fuel, the delay period depends (amongst other things) on the relative velocity of approach between the particles of fuel and the swirling air. The delay period, it is suggested, is the time required for the heat from the air (which is at high temperature) to be transferred to some of the fuel oil droplets, so raising its temperature to the ignition point that it ignites it under the given conditions. Now the swirl and the relative velocity of approach is almost constant in terms of the crank revolutions, and therefore vary as the engine speed varies, provided the volumetric efficiency is kept constant. It follows that this process of heat transfer varies with the speed, amongst other things.

According to the principles of dimensional theory, and considering a droplet of fuel stationary, while the air

moves round it. Heat transfer between the air and the fuel per unit time is a function of:-

1. Densities of air and fuel. (ρ).
2. Viscosities of air " " . (μ).
3. Radius of the fuel droplets.(a)
4. Heat conductivities of fuel and air K .
5. Temperature difference between the air and fuel droplets. (θ)
6. Specific heats of fuel. K_p.
7. Relative velocity of approach between the air and the fuel droplets. (v). (depending on engine speed and injection speed).
8. Surface tension of the fuel droplets.
9. Latent Heat. (L_1).

It will be noticed that chemical factors are here omitted. No doubt such factors are of primary importance, but the object of the present investigation is to endeavour to interpret the facts observed in terms of the physical factors, as it seemed possible that some progress might be made in this direction.

g is ignored, and also the surface tension.

$a = L.$

$\theta = \theta$

$K = MLT^{-3}\theta^{-1}$

$\rho = ML^{-3}$

$L_1 = L^2T^{-2}$

$v = LT^{-1}$

time $= t.$

$\mu = ML^{-1}T^{-1}$

$K_p = L^2T^{-2}\theta^{-1}$

Consider a droplet. Temperature difference θ at surface of radius(a)at time t is:

Take $\theta = a^{\alpha}\theta^{\beta}K^{\gamma}\rho^{\delta}L_1^{\epsilon}v^{\zeta}K_p^{\eta}t^{i}\mu^{\lambda}$

By substituting the values of each as above.

$$\therefore \theta = L^{\alpha}\theta^{\beta}(MLT^{-3}\theta^{-1})^{\gamma}(ML^{-3})^{\delta}(L^2T^{-2})^{\epsilon}(LT^{-1})^{\zeta}T^{i}(ML^{-1}T^{-1})^{\lambda}(L^2T^{-2}\theta^{-1})^{\eta}$$

From above the following equations are derived:-

For θ 1 $= \beta - \gamma - \eta$ ..... (1)

" L 0 $= \alpha + \gamma - 3\delta + 2\epsilon + \zeta - \lambda + 2\eta$ .....(2)

" M 0 $= \gamma + \delta + \lambda$ .....(3)

" T 0 $= -3\gamma - 2\epsilon - \zeta + i - \lambda - 2\eta$.....(4)

These are four equations for nine unknowns.

Take independent variables as $\beta, \gamma, \delta, i, \zeta$.

$$\therefore \quad \eta = \beta - (\gamma+1) \qquad \text{from (1)}$$

$$\lambda = -(\gamma+\delta) \qquad \text{" (3)}$$

$$\text{and} \quad 2\epsilon - i = -3\gamma - \zeta - \lambda - 2\eta \quad \text{" (4)}$$

$$= -3\gamma - \zeta + (\gamma+\delta) - 2\eta$$

$$= -2\gamma - \zeta + \delta - 2\eta$$

$$= -2\gamma - \zeta + \delta - 2(\beta - \gamma - 1)$$

$$= -\zeta - 2\beta + \delta + 2$$

$$\therefore \quad 2\epsilon = i - \zeta - 2\beta + \delta + 2$$

and hence $\alpha = -\gamma + 3\delta - 2\epsilon - \zeta + \lambda - 2\eta$ from (2)

Substituting $= -\gamma + 3\delta - i + \zeta + 2\beta - \delta - 2 - \zeta - (\gamma+\delta) - 2(\beta - \gamma - 1)$

$= \delta - i$

$\therefore$ Substituting

$$\theta = a^{\delta - i}\, \Theta^{\beta}\, K^{+\gamma}\, \rho^{\delta}\, L_1^{\frac{i - \zeta - 2\beta + \delta + 2}{2}}\, v^{\zeta}\, k_p^{\beta - \gamma - 1}\, t^{i}\, \mu^{-(\gamma+\delta)}$$

$$\text{or } \theta = L_1 k_p^{-1} \left[\Theta L_1^{-1} k_p\right]^{\beta} \left[K k_p^{-1} \mu^{-1}\right]^{\gamma} \left[a \rho L_1^{\frac{1}{2}} \mu^{-1}\right]^{\delta}$$

$$\left[a^{-1} L_1^{\frac{1}{2}} t\right]^{i} \left[v L_1^{-\frac{1}{2}}\right]^{\zeta}$$

$$\text{or } \theta = L_1 K_p^{-1} f\left[\frac{\theta k_p}{L_1}, \frac{K}{\mu k_p}, \frac{a \rho L_1^{\frac{1}{2}}}{\mu}, \frac{L_1^{\frac{1}{2}} t}{a}, \frac{v}{L_1^{\frac{1}{2}}}\right]$$

Now for a given air $\frac{K}{\mu k_p}$ is constant, and by dividing the two sides by $L_1 k_p^{-1}$

$$\therefore \frac{\theta k_p}{L_1} = f\left[\frac{\theta k_p}{L_1}, \frac{a \rho L_1^{\frac{1}{2}}}{\mu}, \frac{L_1^{\frac{1}{2}} t}{a}, \frac{v}{L_1^{\frac{1}{2}}}\right] \quad \ldots\ldots (5)$$

$$\text{or alternatively } = f_1\left[\frac{\theta k_p}{L_1}, \frac{a \rho v}{\mu}, \frac{L_1^{\frac{1}{2}} t}{a}, \frac{v}{L_1^{\frac{1}{2}}}\right]$$

There is also the possibility of:-

$$\frac{k_p}{k_p \text{oil}}, \frac{\rho \text{air}}{\rho \text{ oil}}, \frac{K \text{ air}}{K \text{ oil}}, \text{ and } \frac{v \text{ injection}}{v \text{ swirl}}$$

coming in as well.

Similarly at time t if radius a becomes radius r:

$$\frac{r}{a} = f_2\left[\frac{\theta k_p}{L_1}, \frac{a \rho v}{\mu}, \frac{L_1^{\frac{1}{2}} t}{a}, \frac{v}{L_1^{\frac{1}{2}}}\right]$$

Now from (5) $\left(\frac{\theta k_p}{L_1}\right)$ and $\left(\frac{a \rho L_1^{\frac{1}{2}}}{\mu}\right)$ are approximately constants for a given air and fuel, when the only variable is speed.

$$\therefore \quad \frac{\Theta k_p}{L_1} = f_3\left[\frac{L_1^{\frac{1}{2}} t}{a}, \quad \frac{v}{L_1^{\frac{1}{2}}}\right]$$

Θ being the difference of temperature between oil and air.

If ignition temperature (difference) Θ is fixed (on the theory that the combustion effectively commences, i.e. the delay period ends, when the temperature at the surface of a drop reaches a certain value.) Note that this value will depend on the temperature and pressure of the air, which are both here taken as constant.

$$\therefore \quad \frac{L_1^{\frac{1}{2}} t}{a} = \phi\left[\frac{\Theta k_p}{L_1}, \frac{v}{L_1^{\frac{1}{2}}}\right] = \phi_1\left[\frac{v}{L_1 t}\right]$$ where t is now the delay period in time.

for $\frac{\Theta k_p}{L_1}$ is taken as constant for end of delay.

Period. There is however the probability of the ratio of velocity of injection to rate of swirl coming in; in which case

$\therefore \quad \frac{L_1^{\frac{1}{2}} t}{a}$ may have to be written a function of

$$\left[\frac{v}{L_1^{\frac{1}{2}}}, \frac{v}{V}\right] \quad \ldots\ldots\ldots (6)$$

where v is taken to be a mean velocity of swirl.

V is taken to be the velocity of injection.

It appears that the delay period t is proportional to the original radius a of the drop, provided v and V are constant (as well as the other factors referred to).

(6) may also be written

$$\frac{L_1^{\frac{1}{2}} t}{a} = f_4\left[\frac{v}{V}, \frac{V}{L_1^{\frac{1}{2}}}\right]$$

$$= f_4\left[\frac{v}{V}\right]$$

$$t = \frac{a}{L_1^{\frac{1}{2}}} f_4\left[\frac{v}{V}\right] \quad \ldots\ldots\ldots\ldots\ldots (7)$$

Strictly L_1 may vary slightly with conditions. With the type of injection gear used (with spring controlled injection valve) it is likely that the variation of injection speed V with speed of engine (and therefore with v) is small.

The writer has to express his acknowledgments to Professor Lees for several discussions on dimensional theory applied to this problem, but the above theory must not be considered as necessarily giving Professor Lees' final opinions on the subject.

PART II.

SECTION B.

"PILOT INJECTION".

It has already been mentioned that in order to get smooth running of the engine, the delay period should either be reduced or a very small quantity of fuel be introduced in the cylinder at first (Pilot injection). (See Introduction, Part II).

Method Used for Carrying Out the Tests.

A number of tests were carried out to study the effect of pilot injection. Naturally a few different quantities of the fuel injected into the cylinder (by the pilot pump), have been tried. Also the timing of injection of the pilot jet was varied over a wide range for three values of timing of injection of the main pump.

For comparison purposes, the speed of the engine was kept constant, at 1200 R.P.M. For the same reason the quantity of fuel injected per cycle by the main pump was kept constant.

The load was adjusted to keep the speed constant. The air inlet temperature and the cooling water outlet temperature were kept constant at about 20°C. and 60°C. respectively. Timing of injection and of combustion. (i.e. end of delay period) of the main pump were recorded. Time of injection of the pump, load, fuel injected per cycle, noise and delay period were all recorded at 1200 R.P.M.

TEST RESULTS.

(I) MAIN INJECTION TIMING AT T.D.C.

(i) Quantity Injected by Main Pump = 0.812×10^{-4} lbs./cycle.

" " " Pilot = 0.137×10^{-4} " " "

Fig. (45) shews the results obtained. The main pump was injecting at T.D.C. The three curves in this Fig. shew the relation between Torque in lb.ft., delay period in secs. $\times 10^{-3}$ and noise (in Decibels)† against pilot injection timing, with T.D.C. taken as 0° degrees.

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† The noise readings are comparative only and have no relation to the threshold of hearing. The readings are really reductions below the greatest noise, the highest reading representing the quietest condition.

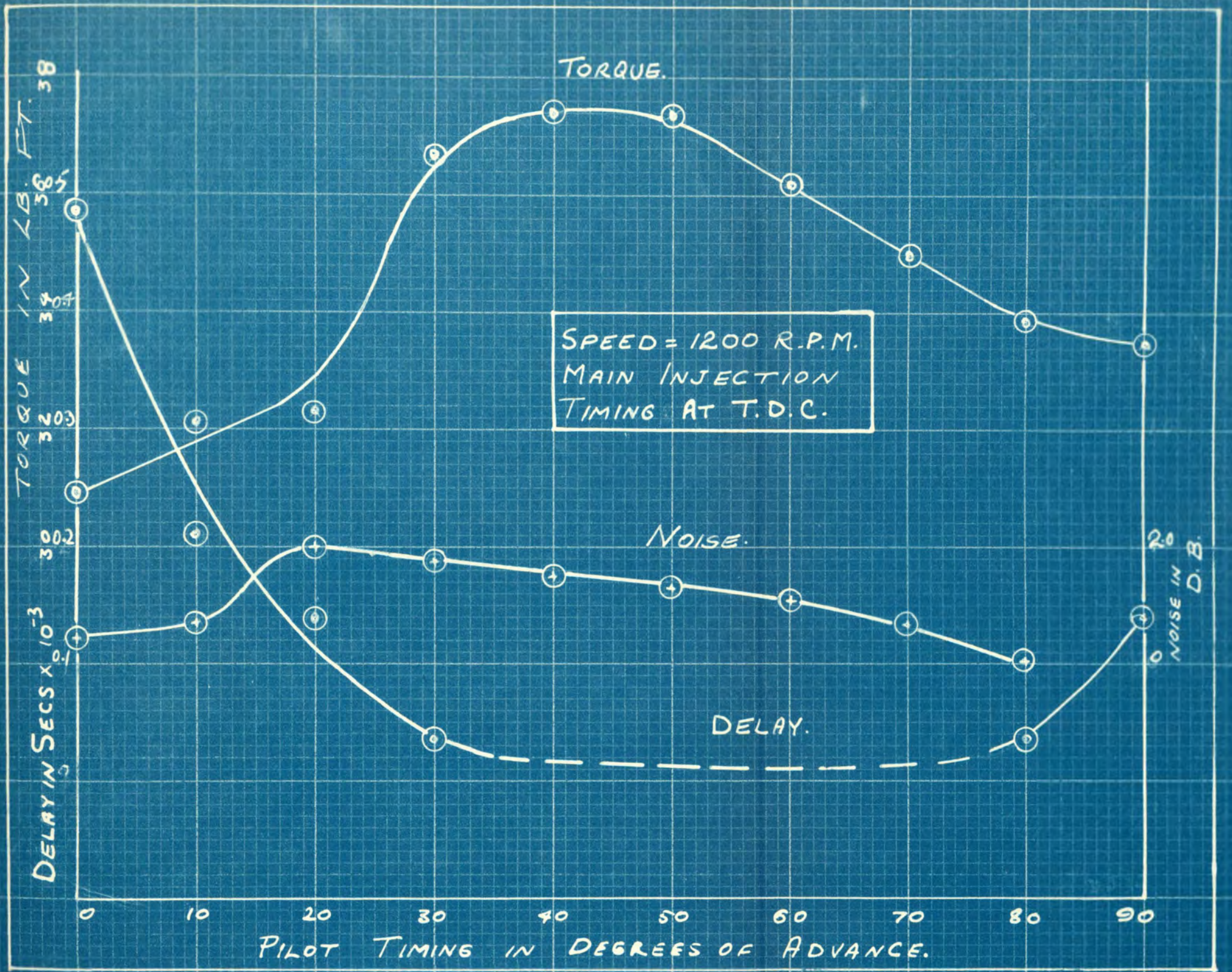

TORQUE.
SPEED = 1200 R.P.M.
MAIN INJECTION
TIMING AT T.D.C.
NOISE.
DELAY.
TORQUE IN LB. FT.
38
36.5
35
33.5
32
30.5
30
DELAY IN SECS x 10⁻³
0.2
0.1
0
NOISE IN D.B.
20
0
0
10
20
30
40
50
60
70
80
90
PILOT TIMING IN DEGREES OF ADVANCE.

FIG. 45

The curves shew a decrease in delay period as the pilot injection advances to 30° before T.D.C. After this, the indicator diagrams were very smooth and the measurement of timing of combustion could not be carried out accurately. When the pilot injection was advanced to 80° before T.D.C., a small hump indicating combustion appeared again. After this point the delay period increased. The results (with illustrations) will be discussed later.

With regard to the noise measurements, it was found that the M.L. Instrument, was not sensitive to Diesel knock at all. A heavy knock could be heard sometimes, while this instrument recorded nothing appreciably different to the reading with smoother running.

At T.D.C. there was a fairly heavy knock, and as the pilot charge was advanced, the engine ran more smoothly until 40° before T.D.C. was reached for the pilot injection timing. Before 40° and 60° advance for this timing there was no appreciable change in the noise. When the pilot jet was advanced beyond 60° before T.D.C., the audible knock was found to increase with the advance.

The torque, as the curve in Fig. (45) shews, increased to a maximum at minimum delay period.

(I) (ii)

Quantity Injected by Main Pump $= 0.814 \times 10^{-4}$ lbs.per cycle

" " " Pilot " $= 0.186 \times 10^{-4}$ lbs. " "

Fig. (46) shews test results with the pilot pump injecting a little more fuel than in the previous test. The delay period, in this case, decreased as the pilot charge injection was advanced up to 20^{o} before T.D.C. The indicator diagram was fairly smooth after this, and the timing of commencement of combustion could not be measured with any accuracy. A small hump only appeared when the injection advance of the pilot pump was 90^{o} before T.D.C. It will be noticed, however, that the indicator diagram was smooth over a longer range (in fact for 20^{o} more) than in the previous test.

In this case the range of smooth indicator diagrams was from 20^{o} to 90^{o}, while in the previous case the range was from 30^{o} to 80^{o}. This may be due to a larger quantity delivered by the pilot pump.

There was also a slight increase in the torque as the pilot injection was advanced.

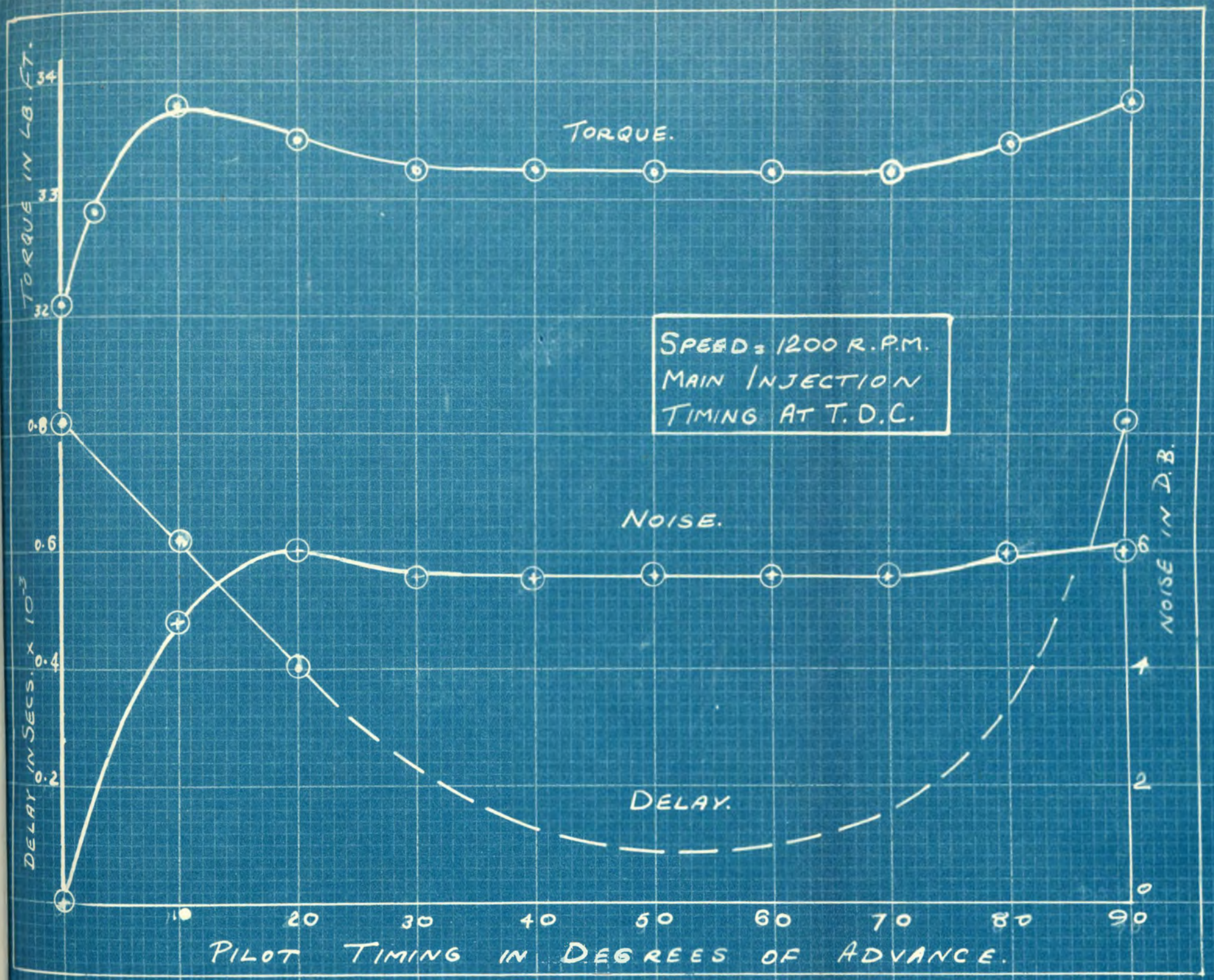

TORQUE IN LB. FT.
34
33
32
TORQUE.
SPEED = 1200 R.P.M.
MAIN INJECTION
TIMING AT T.D.C.
0.8
0.6
0.4
0.2
DELAY IN SECS. x 10⁻³
NOISE.
DELAY.
NOISE IN D.B.
6
2
0
10
20
30
40
50
60
70
80
90
PILOT TIMING IN DEGREES OF ADVANCE.

FIG. 46.

At T.D.C., there was a noticeable knock which faded away as the pilot injection timing was advanced, and appeared again at about 60^{o} injection advance, beyond which it increased as the injection advanced.

(I) (iii)

Main Pump Supply = 0.813×10^{-4} lbs. per cycle.

Pilot " " = 0.2656×10^{-4} lbs. per cycle.

The three curves in Fig. (47) shew the test results. The delay period, as before, could only be measured up to 20^{o} advance of the pilot injection. The indicator diagrams were smoothed out over a range of injection advance of nearly 80^{o}. There was a noticeably small combustion hump on the diagram due to the large quantity of pilot fuel charge being injected. This only appeared when the pilot injection was advanced to between 40^{o} and 60^{o} before T.D.C.

The knock was marked with pilot injection at T.D.C. and after advancing the pilot charge there was a light knock up to 50^{o} advance. Beyond this, the knock appeared again. This may be due to the larger quantity being rejected.

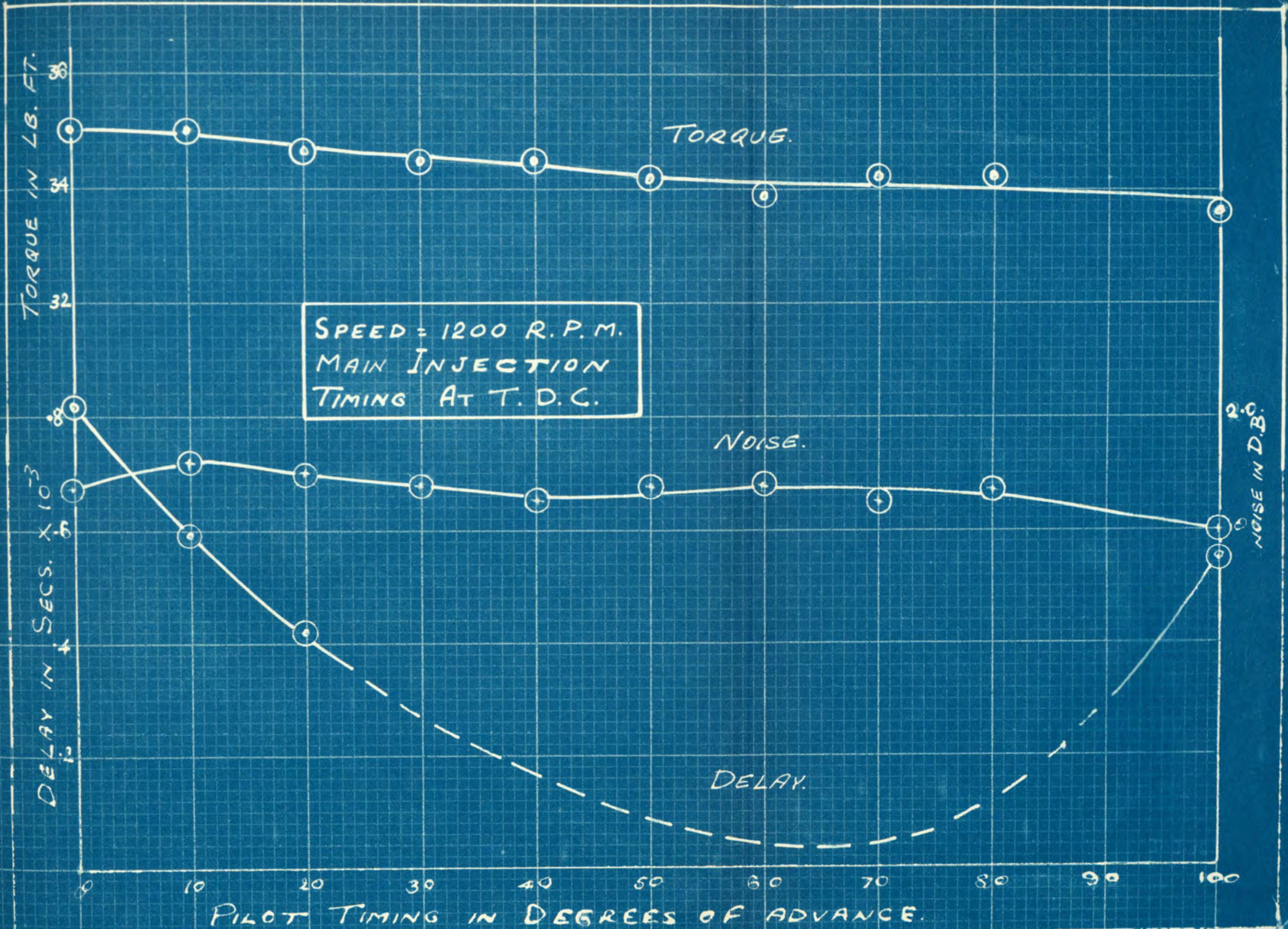

TORQUE IN LB. FT.
38
34
32
TORQUE.
SPEED = 1200 R.P.M.
MAIN INJECTION
TIMING AT T.D.C.
NOISE.
2.0
NOISE IN D.B.
DELAY IN SECS. X 10⁻³
.8
.6
.4
.2
DELAY.
0
10
20
30
40
50
60
70
80
90
100
PILOT TIMING IN DEGREES OF ADVANCE.

FIG. 47.

The torque was very erratic throughout this test.

(II) MAIN INJECTION TIMING 11.5 AFTER T.D.C.

(i) Main Pump Supply = 0.828×10^{-4} lbs. per cycle.

Pilot " " = 0.1366×10^{-4} " " ".

Fig. (48) shews the test results for the quantities delivered per cycle as mentioned above.

The delay period is in seconds $\times 10^{-3}$. The speed of the engine was kept constant at 1200 R.P.M. This was checked by means of a calibrated tachometer.

The delay period decreased, as the pilot injection was advanced. The big hump on the indicator diagram smoothed out also with pilot injection advance. The torque curve shows a minimum value near 30^{o} advance. The fairly heavy knock, which could be heard at a retarded position, disappeared gradually as the pilot injection was advanced.

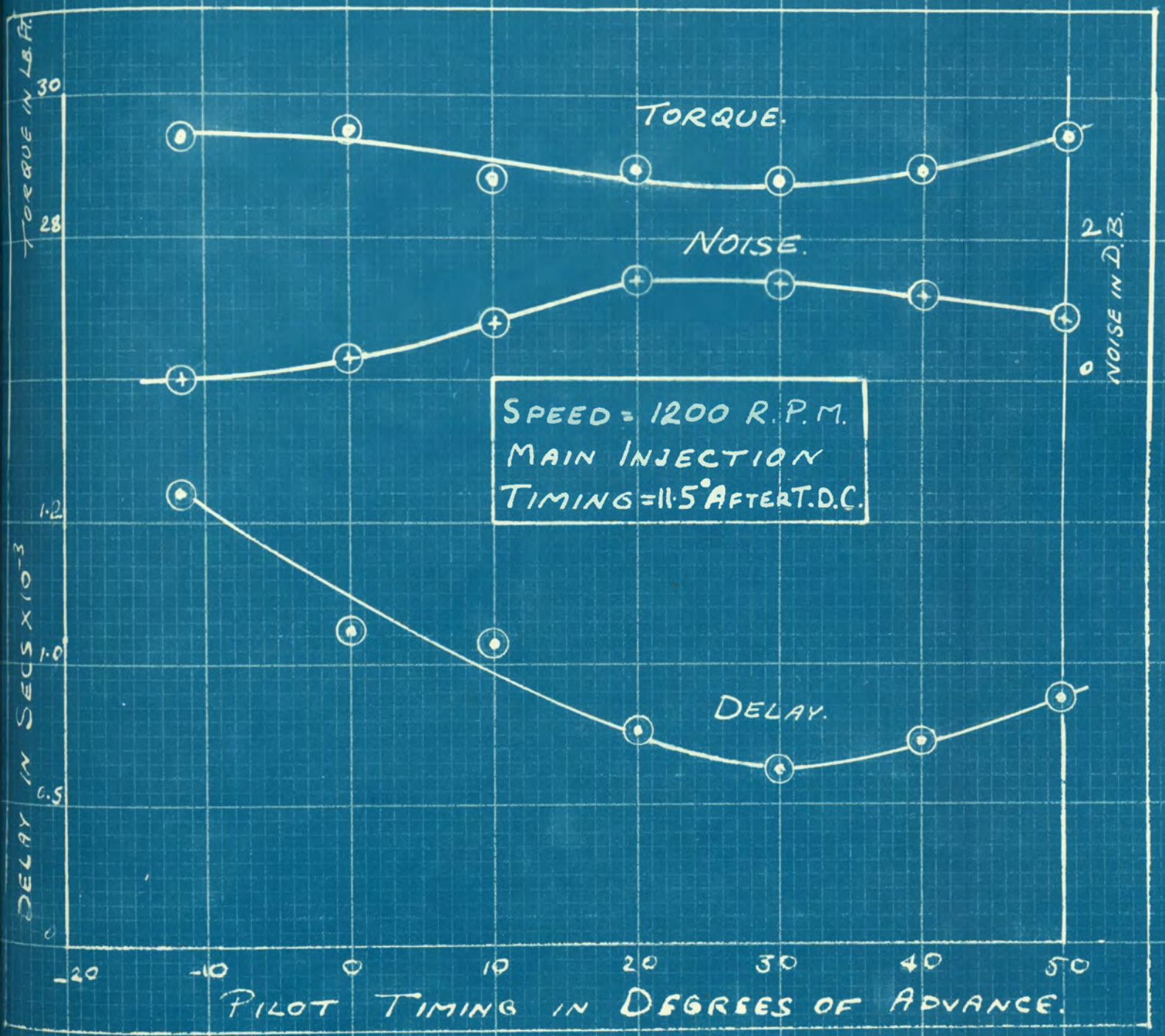

TORQUE IN LB. FT.
30
28
TORQUE.
NOISE.
2
0
NOISE IN D.B.
SPEED = 1200 R.P.M.
MAIN INJECTION
TIMING = 11.5° AFTER T.D.C.
1.2
1.0
0.5
0
DELAY IN SECS X 10⁻³
DELAY.
-20
-10
0
10
20
30
40
50
PILOT TIMING IN DEGREES OF ADVANCE.

FIG. 48

(II) (ii)

Main Pump Supply = 0.818×10^{-4} lbs. per cycle.

Pilot " " = 0.1785×10^{-4} " " "

The three curves in Fig. (49) shew, as before, delay period, torque and noise against timing of pilot injection.

The delay period decreased as the position of pilot injection changed from 11.5 after T.D.C, to 20^{o} before T.D.C. Beyond this last position, the indicator diagram was very smooth, and it was not easy to notice the commencement of the bump accurately. The dotted line is only a suggestion of what the delay period would be.

There was an increase in load to a maximum value as the injection timing advanced.

The heavy knock accompanied by a big hump on the indicator diagram soon disappeared when the timing was advanced up to 20^{o} before T.D.C. A slight knock appeared after 40^{o} advance.

(II) (iii)

Main Pump Supply = 0.814×10^{-4} lbs. per cycle.

Pilot " " = 0.233×10^{-4} " " "

Fig. (50) shews the tests results. As before, the quantity delivered by the pilot charge was slightly increased.

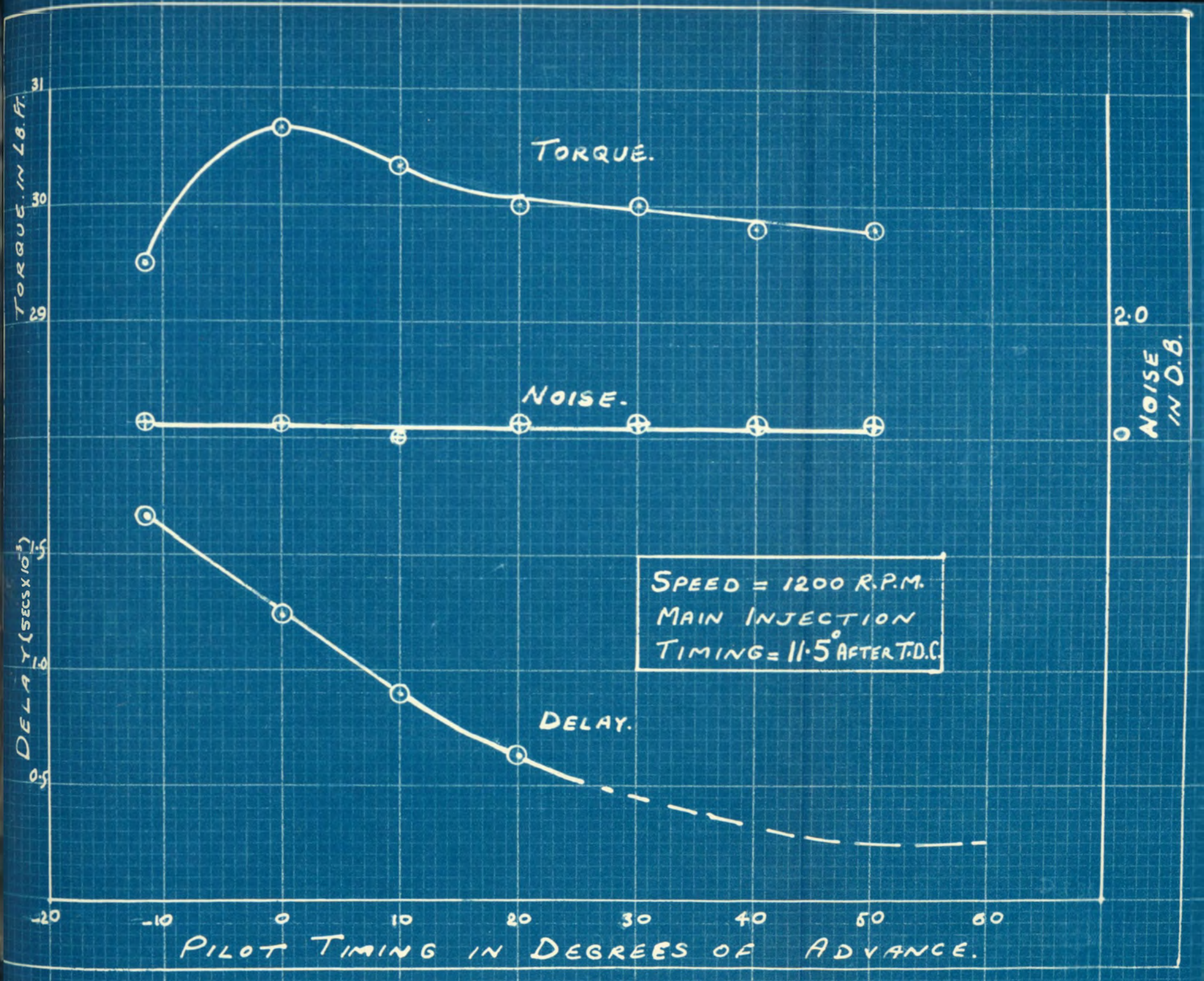

TORQUE. IN LB. FT.
31
30
29
TORQUE.
NOISE.
2.0
0
NOISE IN D.B.
DELAY (SECS X 10⁻³)
1.5
1.0
0.5
DELAY.
SPEED = 1200 R.P.M.
MAIN INJECTION
TIMING = 11·5° AFTER T.D.C.
-20
-10
0
10
20
30
40
50
60
PILOT TIMING IN DEGREES OF ADVANCE.

FIG. 49.

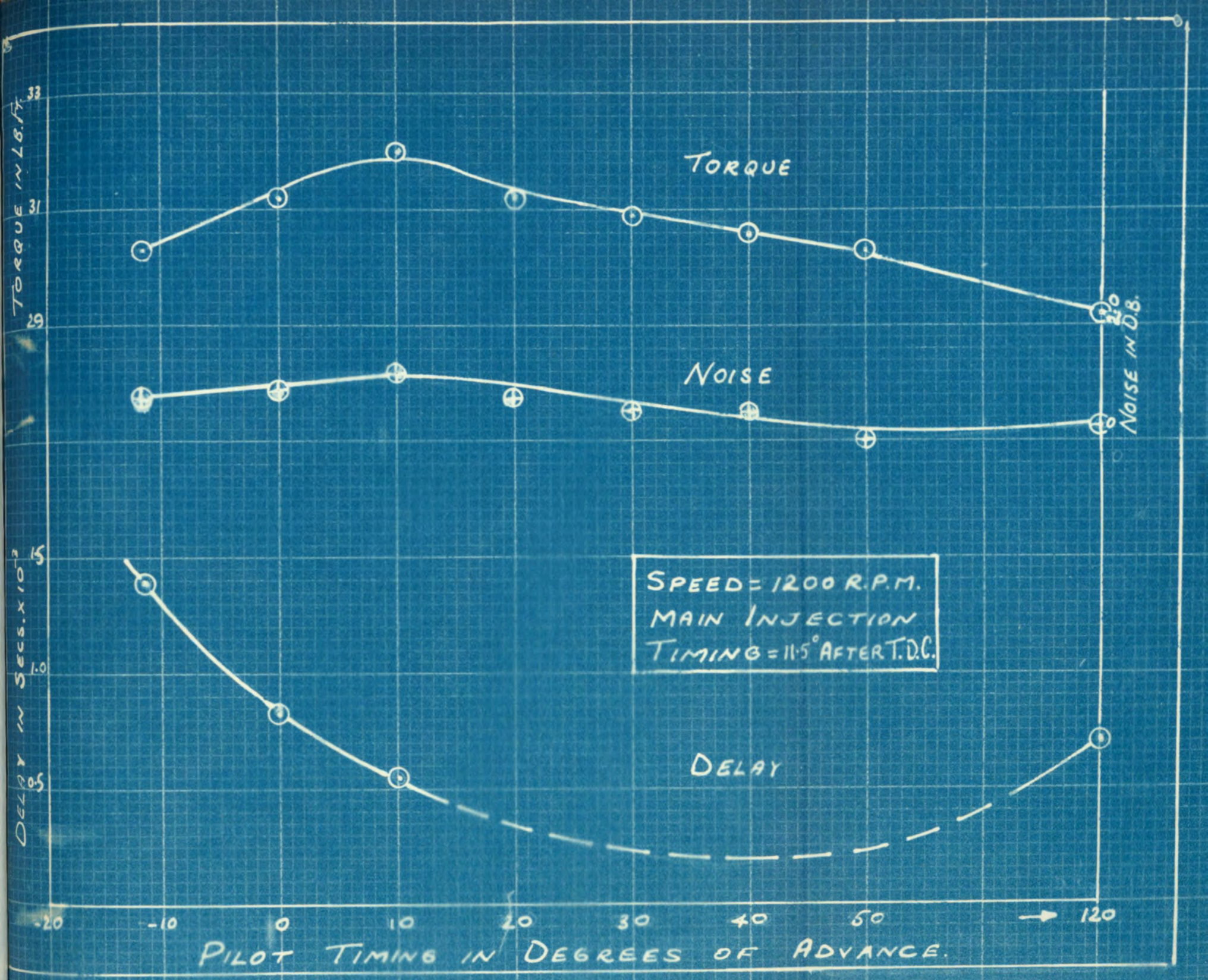

TORQUE IN LB. FT.
33
31
29
TORQUE
NOISE
NOISE IN D.B.
120
110
DELAY IN SECS. X 10⁻³
1.5
1.0
0.5
SPEED = 1200 R.P.M.
MAIN INJECTION
TIMING = 11.5° AFTER T.D.C.
DELAY
-20
-10
0
10
20
30
40
50
120
PILOT TIMING IN DEGREES OF ADVANCE.

FIG. 50

The torque curve shews an increase to a maximum at about 10^{o} before T.D.C.

The delay period, however, decreased up to 10^{o} advance. After this the delay period was not easy to detect. The hump on the diagram appeared only at 120^{o} advance. The knock at a retarded position was very heavy indeed. It decreased a little and rose again at about 50^{o} advance. This may again be due to the large quantity of fuel injected by the pilot pump. The "smooth range" of indicator diagrams was about 5^{o}.

(III) MAIN INJECTION TIMING 4^{o} BEFORE T.D.C.

(i) Main Pump Supply = 0.814×10^{-4} lbs. per cycle.

Pilot " " = 0.1336×10^{-4} " " "

Test results are plotted in Fig. (51) which shew, as before, the delay period, torque and noise plotted against pilot injection angle of advance. The quantities delivered by each pump are mentioned above.

The torque curve shews a slight increase to a maximum at about 20^{o} pilot injection advance.

The delay period was decreasing as the injection advanced to 30^{o} before T.D.C. After this it could not be measured accurately, as the indicator diagram was very smooth. A slight hump on the diagram appeared at 110^{o} advance.

The knock, occurring at the T.D.C. position of pilot ignition, disappeared as the injection advanced. It only appeared again at an injection timing of about 90^{o} before T.D.C.

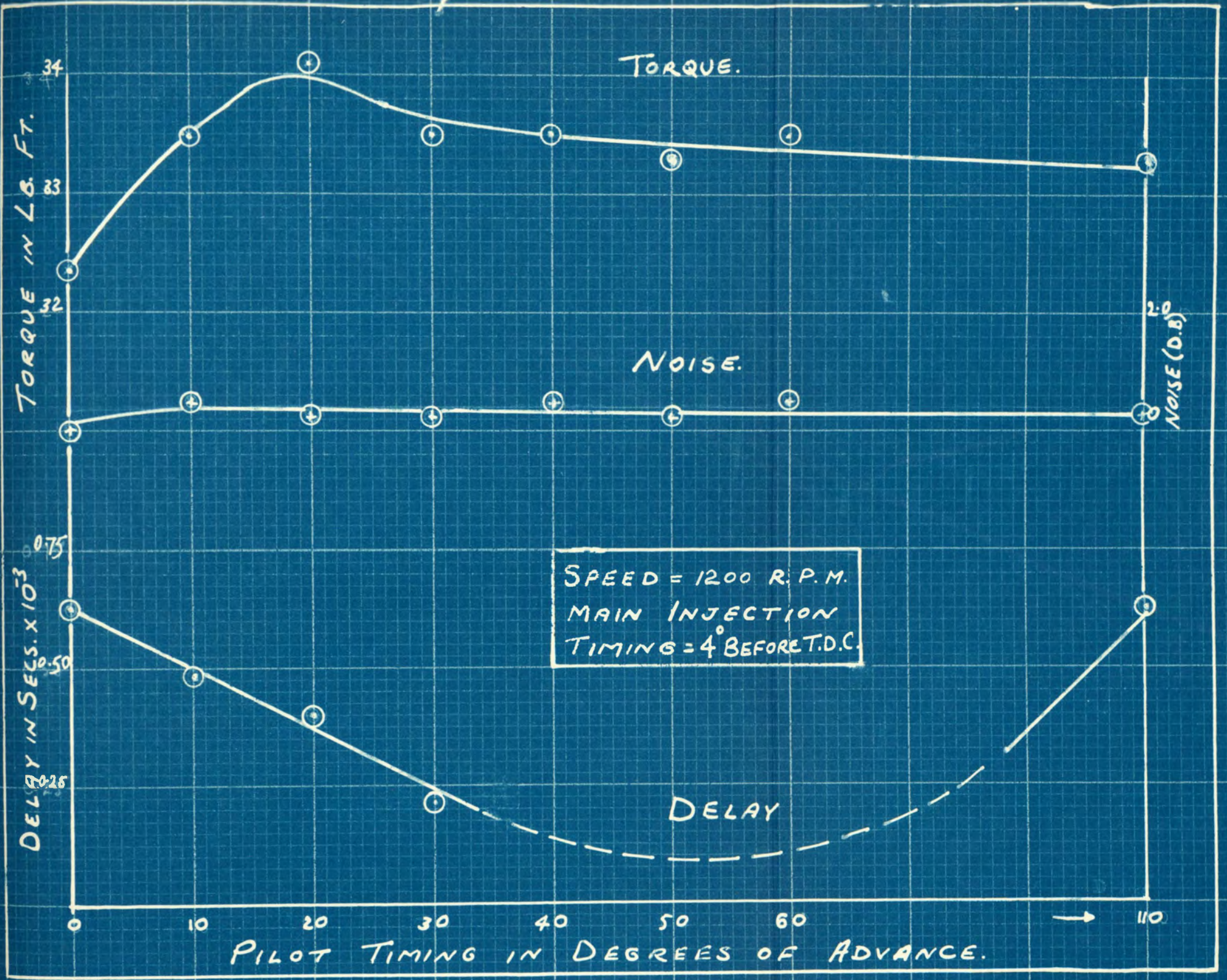

TORQUE.
NOISE.
DELAY
TORQUE IN LB. FT.
34
33
32
NOISE (D.B.)
2.0
0
DELAY IN SECS. X 10⁻³
0.75
0.50
0.25
SPEED = 1200 R.P.M.
MAIN INJECTION
TIMING = 4° BEFORE T.D.C.
0
10
20
30
40
50
60
110
PILOT TIMING IN DEGREES OF ADVANCE.

FIG. 51.

(III) (ii)

Main Pump Supply = 0.813×10^{-4} lbs. per cycle.

Pilot " " = 0.1768×10^{-4} " " "

Fig. (52) shews the test results.

The torque curve shows a maximum at about 30^{o} advance of pilot injection. The delay period decreased from injection at T.D.C. to about 30^{o} advance; beyond this advance the indicator diagram was smooth for a range of 90^{o}. After this a small hump on the diagram appeared. A heavy knock could be heard with injection at T.D.C.; this decreased to a slight knock as the angle of advance increased. The heavy knock appeared again at 50^{o} Pilot injection timing before T.D.C.

(III) (iii)

Main Pump Supply = 0.813×10^{-4} lbs. per cycle.

Pilot " " = 0.220×10^{-4} " " "

The test results plotted in Fig. (53) shew very clearly that the delay period decreased again as the pilot injection timing was advanced from 0^{o} to about 30^{o} before T.D.C. The hump on the indicator diagram disappeared until nearly 140^{o} advance was reached. The indicator diagram was fairly

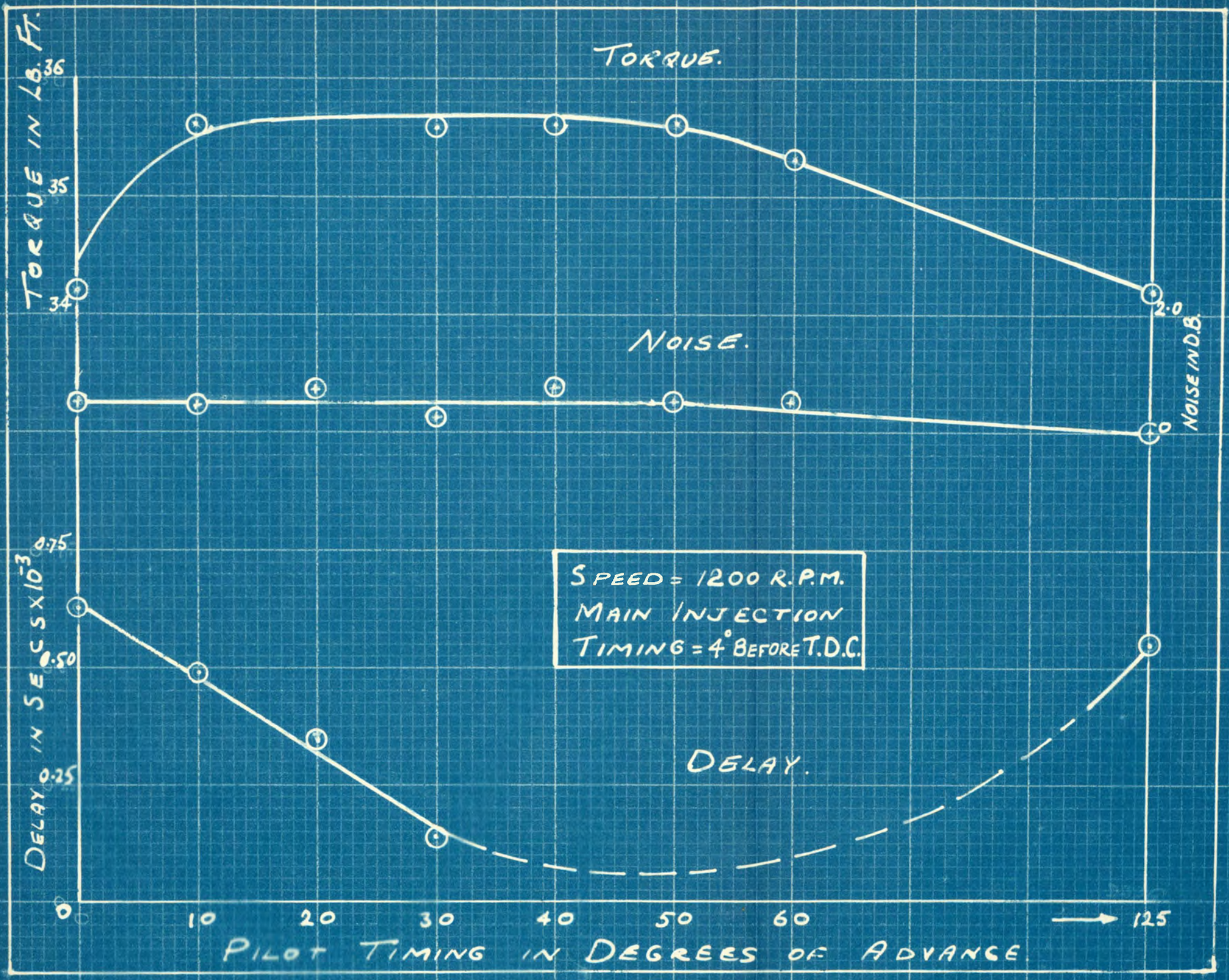

TORQUE.
TORQUE IN LB. FT.
36
35
34
NOISE.
2·0
0
NOISE IN D.B.
SPEED = 1200 R.P.M.
MAIN INJECTION
TIMING = 4° BEFORE T.D.C.
DELAY.
DELAY IN SECS X 10⁻³
0·75
0·50
0·25
0
10
20
30
40
50
60
125
PILOT TIMING IN DEGREES OF ADVANCE

FIG. 52.

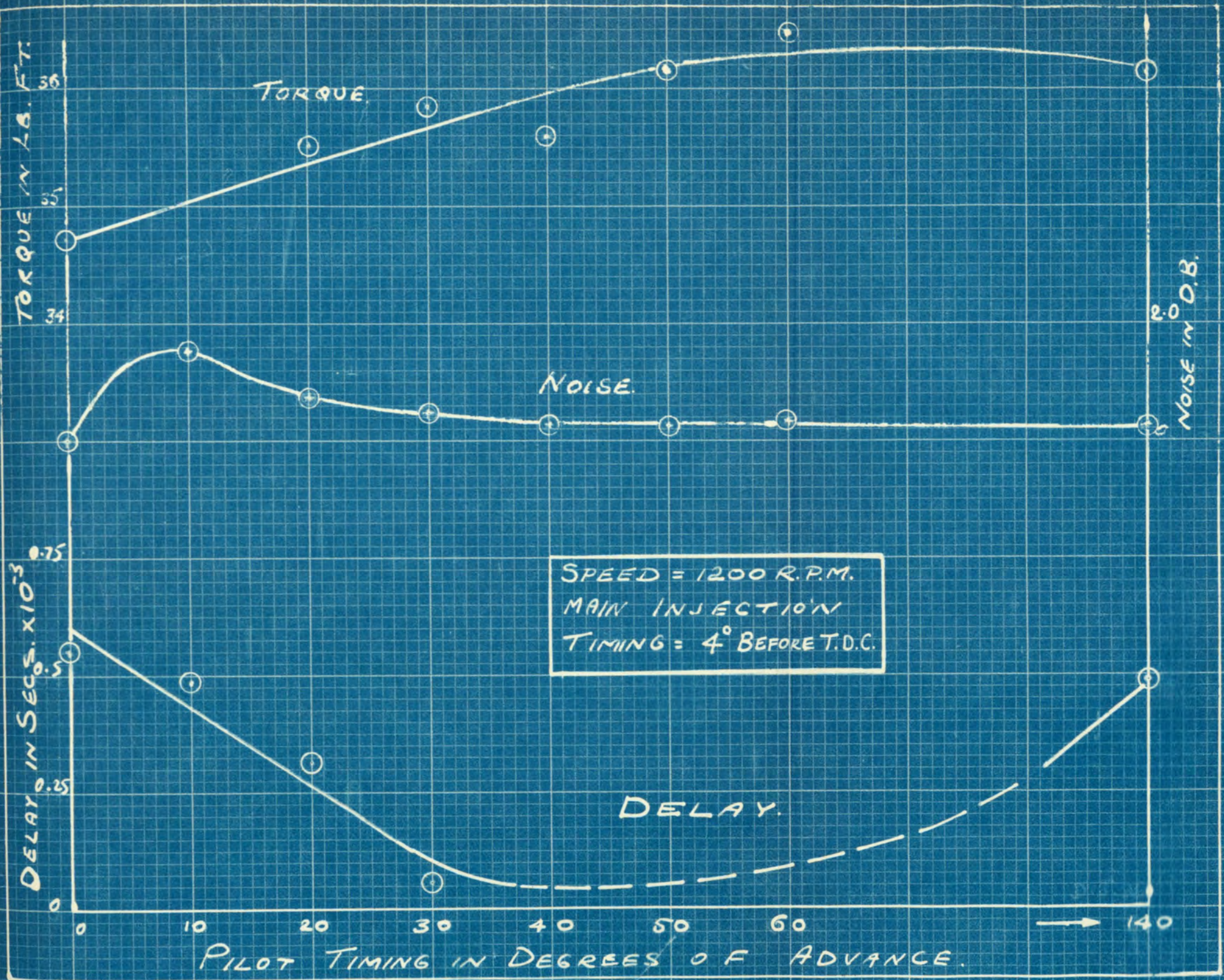

TORQUE.
NOISE.
DELAY.
TORQUE IN LB. FT.
36
34
NOISE IN D.B.
2.0
DELAY IN SECS. X10⁻³
0.75
0.5
0.25
0
SPEED = 1200 R.P.M.
MAIN INJECTION
TIMING = 4° BEFORE T.D.C.
0
10
20
30
40
50
60
140
PILOT TIMING IN DEGREES OF ADVANCE.

FIG. 53.

smooth, with the exception of a small hump due to the pilot charge (a large quantity of which was delivered) for a range of 105°.

The torque increased slightly and was very erratic. The knock, with injection near T.D.C., was heavy indeed; it was smoothed somewhat as the injection advanced. Then it appeared again at about 50° advance.

EIGHT -STROKING.

It was mentioned before, that the engine eight-stroked regularly at about 900 R.P.M., when the injection was retarded with the main pump alone supplying the engine with the fuel.

The running was very rough, and the engine shut down after about half-an-hour's run. In this case, a heavy deposit of carbon was found on the exhaust valve stem. The reasons suggested for these phenomena are discussed below.

Max. Pressure 690 lb./in²

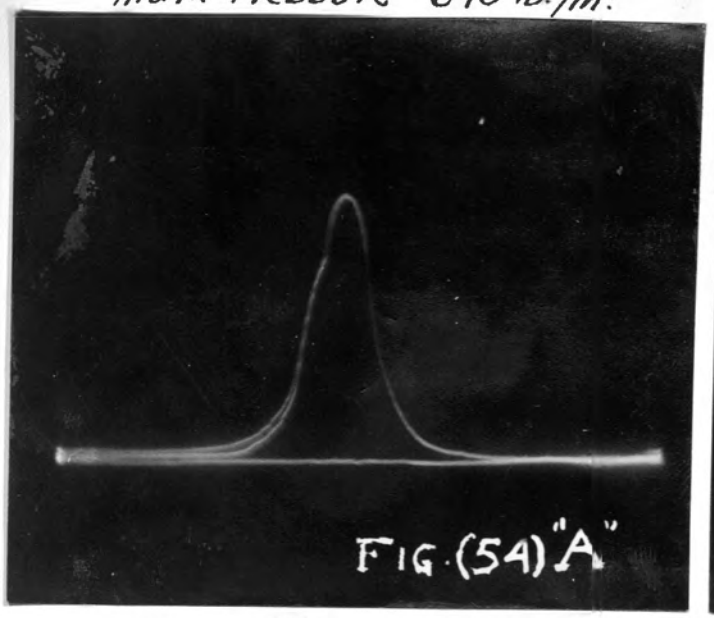


FIG. (54) "A"

Max. Pressure 710 lb./in²

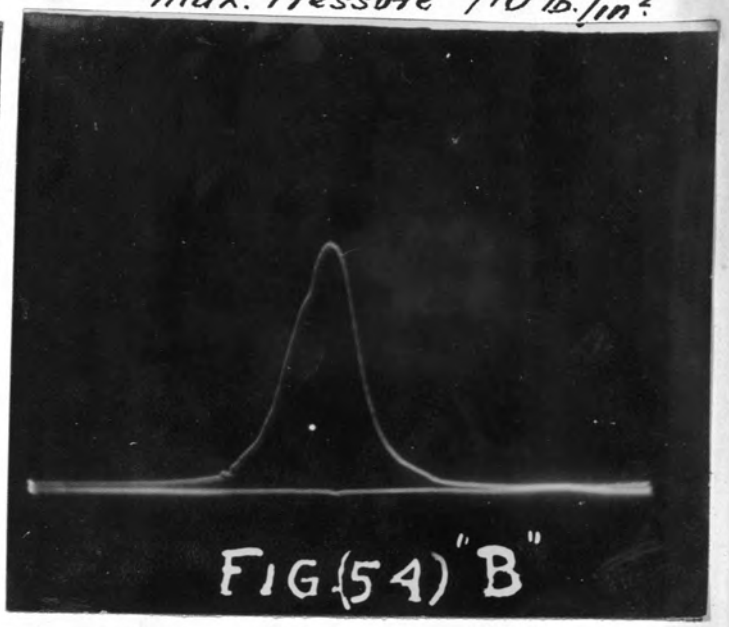


FIG. (54) "B"

Max. Pressure 730 lb./in²

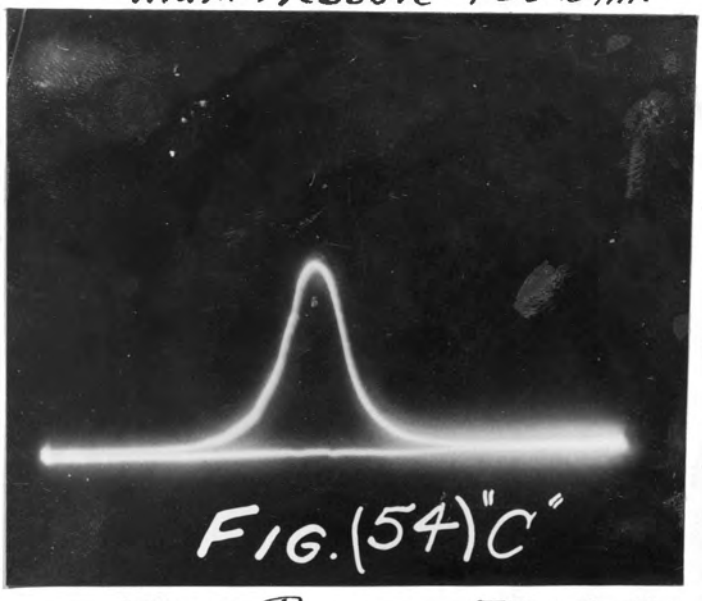


FIG. (54) "C"

Max. Pressure 740 lb./in²

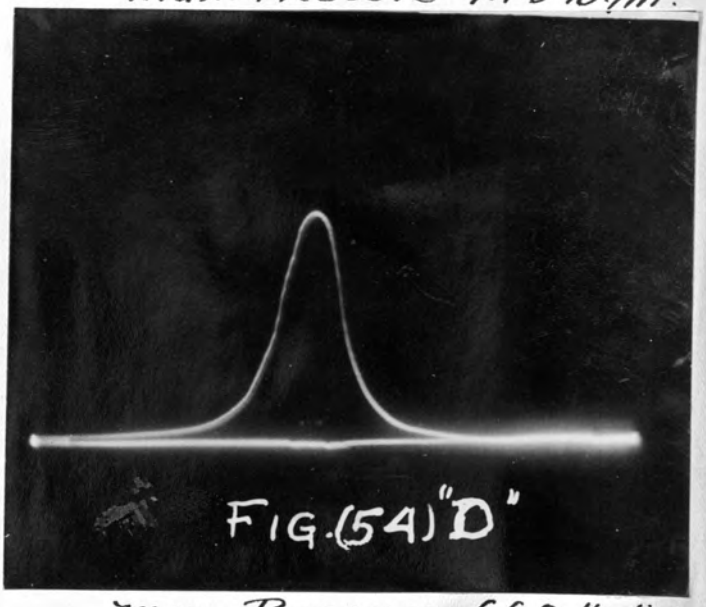


FIG. (54) "D"

Max. Pressure 730 lb./in²

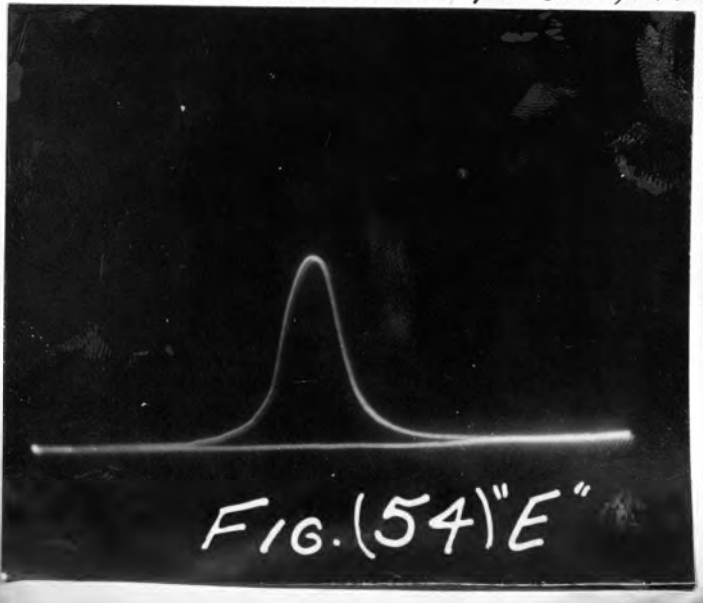


FIG. (54) "E"

Max. Pressure 660 lb./in²

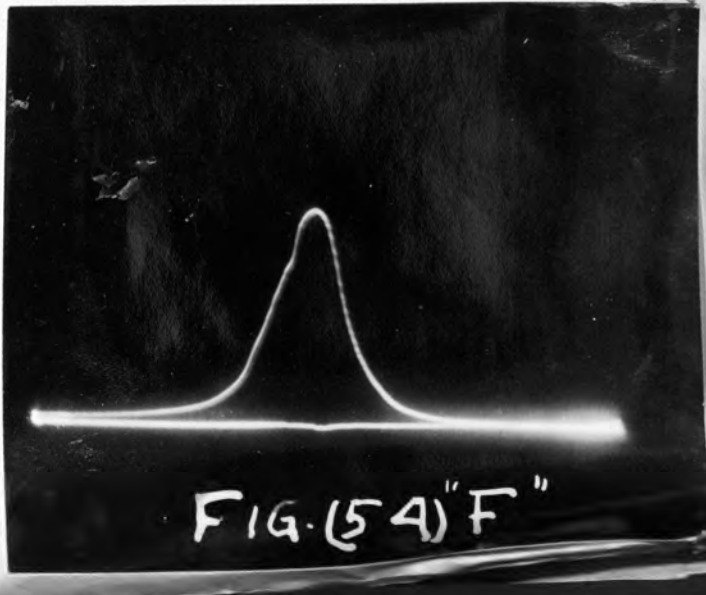


FIG. (54) "F"

DISCUSSION OF RESULTS OF SECTION B.

Tests shown in Figs. (45) to Fig. (54) all reveal that there is, in some cases, a marked improvement in the engine running, so far as the knock and torque questions are concerned. In all cases, with certain angles of pilot injection advance, the delay period of the main pump was nearly decreased to nothing. The best quantity delivered by the pilot pump, so far as the smoothness of running is concerned, seems to be in the region of 0.177×10^{-4} lb. per cycle, or for a main injection quantity of 0.813×10^{-4} lb., in the region of about little over % 21.75 of the main charge.

The best timing of pilot injection appears to be between 40^{o} and 50^{o} before T.D.C. To illustrate the description of the indicator diagram, and the changes taking place, when the pilot injection was advanced, Fig. (54) contains six photographs of indicator diagrams (out-of-phase) on a time (or angle) base. In this Fig. The timing of injection of the main pump was advanced by 4 degrees. The quantity delivered per cycle is 0.813×10^{-4} lb. main pump and 0.1768 lb. pilot pump.

Fig. (54) (A) is the indicator diagram with the pilot injecting at T.D.C. It is clear that combustion is fairly rapid. When the pilot injection was advanced, this rapid pressure rise smoothed out gradually as shewn in (B) and (C) with pilot injection timing 10^{o} and 20^{o} before T.D.C. When the timing of injection was advanced further, the diagrams smoothed out completely. This is illustrated in Fig. (54) (D) & (E), where the pilot injection timing was 40^{o} and 60^{o} before T.D.C. The hump appeared very distinct at 130^{o} angle of advance, as shewn in Fig. (54)(F).

It is suggested that the appearance of the hump was due to the fact that by admitting the pilot injection at excessive advance angles, some of the fuel clings to the cylinder walls, and very little of it burns, before the main charge is admitted. Thus the temperature is not high enough to burn the main charge as soon as it enters the cylinder. It is important to note that this explanation also suggests itself by the fact that the larger the quantity of charge, the larger the deviation of smoothness of the diagram in terms of angle of advance. This

Maximum Pressures. lb./in².

530.

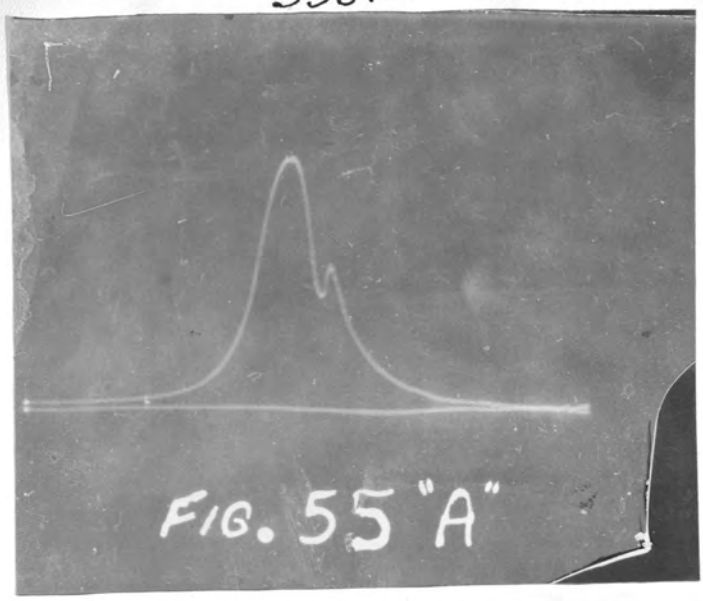


FIG. 55 "A"

650.

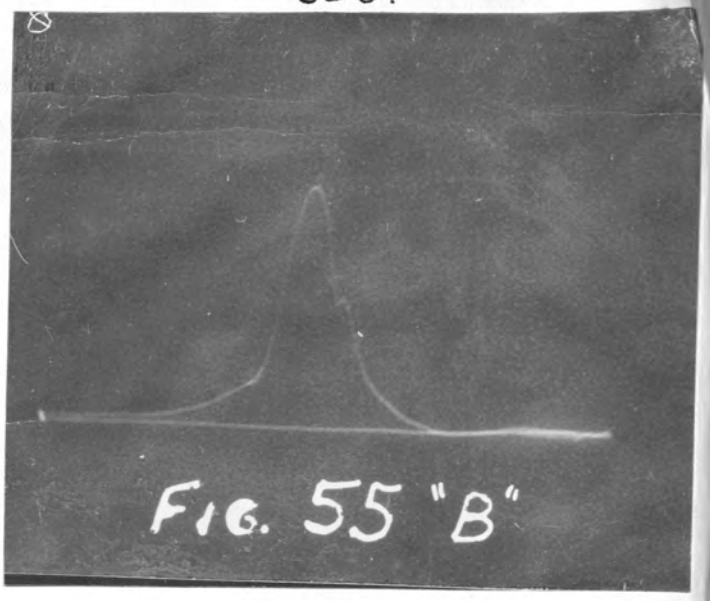


FIG. 55 "B"

670.

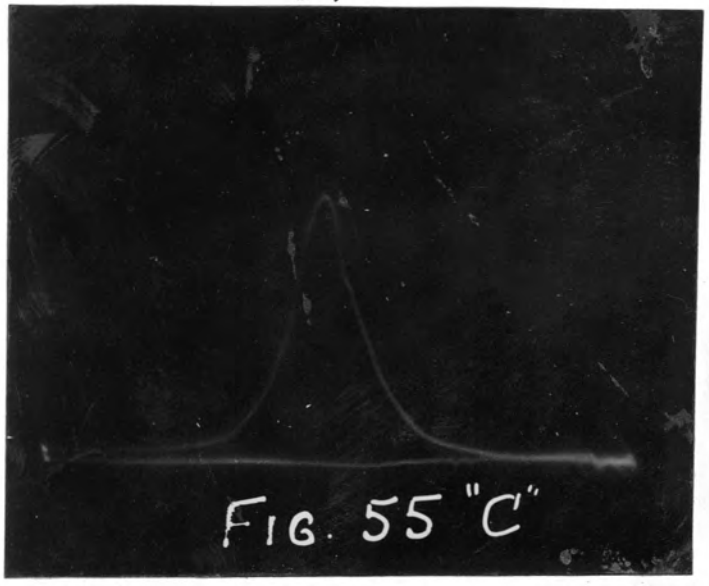


FIG. 55 "C"

670.

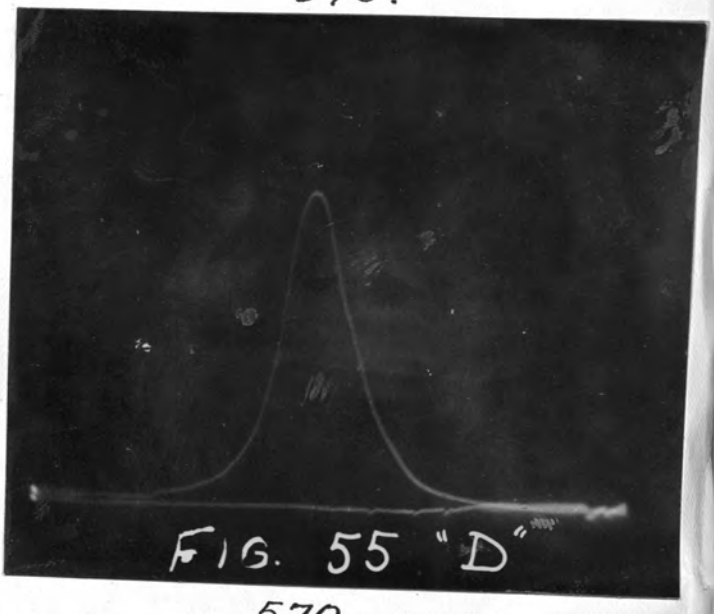


FIG. 55 "D"

650.

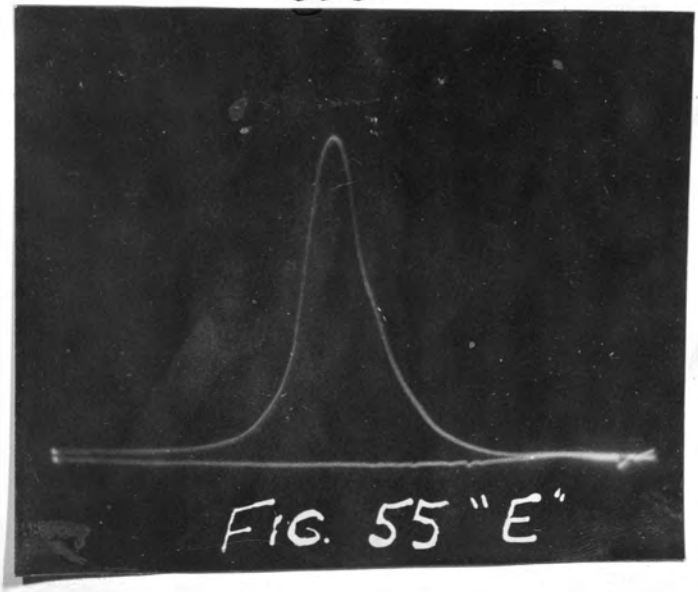


FIG. 55 "E"

570.

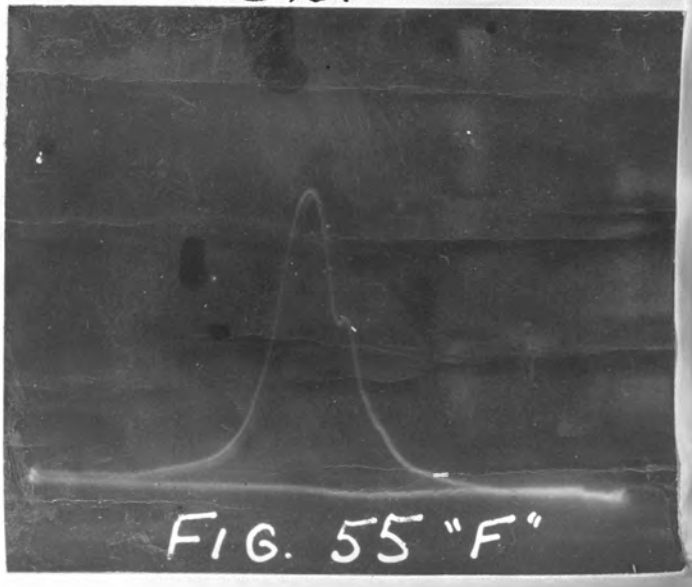


FIG. 55 "F"

behaviour (already mentioned) was consistent for all the tests mentioned.

Fig. (55) shews six indicator diagrams as follows:-

	Injection Timing of Pilot Charge.
A	11.5° After T.D.C.
B	0° at T.D.C.
C	20° before " "
D	40° " " "
E	60° " " "
F	95° " " "

The fuel injected was 0.818×10^{-4} lb. per cycle, and 0.1785 lb. per cycle from the main and pilot pumps respectively. The main fuel was injected at 11.5° after T.D.C. A and B shew a distinct hump on the diagram. The hump appears small at C. At D and E, the diagrams, as can be seen, are quite smooth. At F a small hump appears again.

These indicator diagrams vary in height. This, however, may be due to the variation in voltage of the pressure element and must not be considered as necessarily indicating any variation in the combustion pressure. The shapes of the diagrams were fairly consistent.

Maximum Pressures. lb./in.²

665.

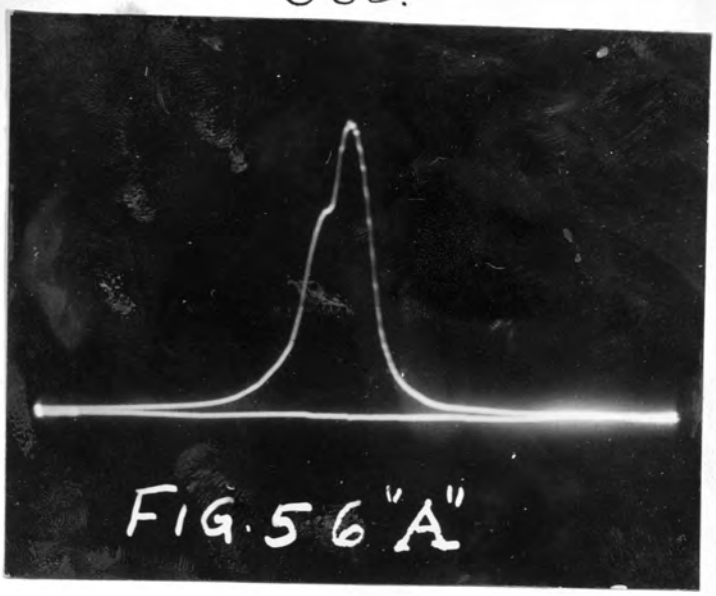


FIG. 56 "A"

720

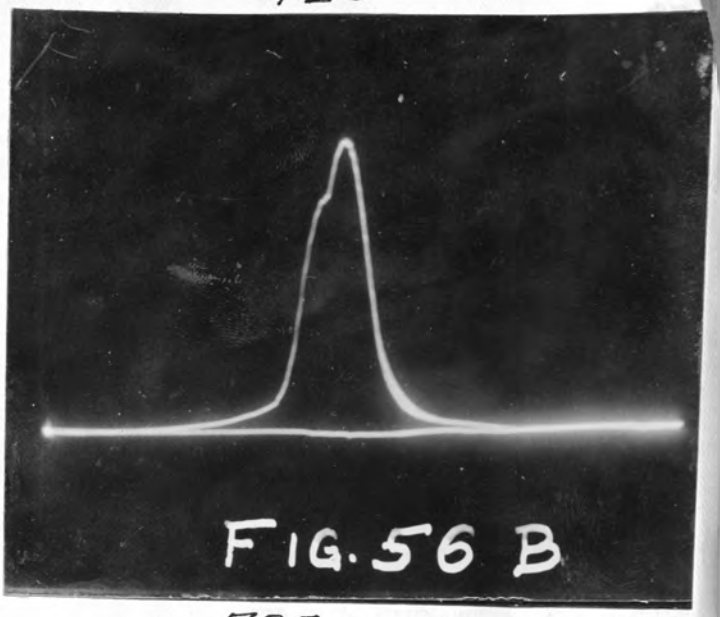


FIG. 56 B

720.

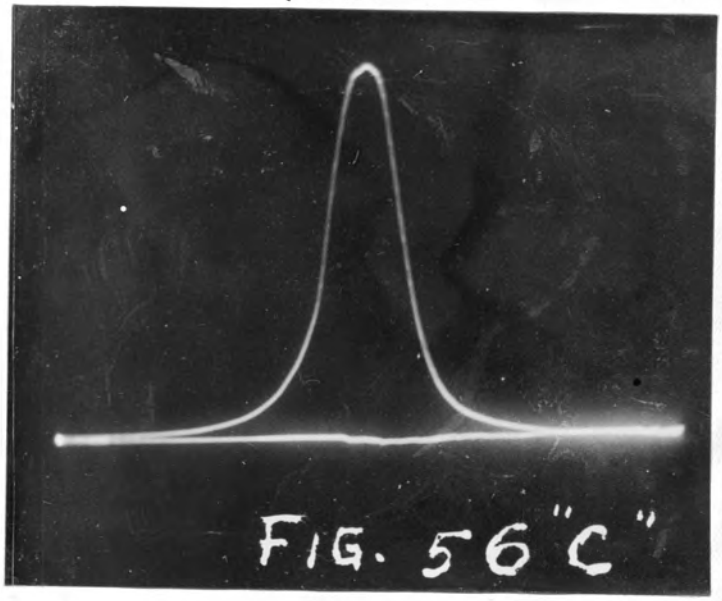


FIG. 56 "C"

725.

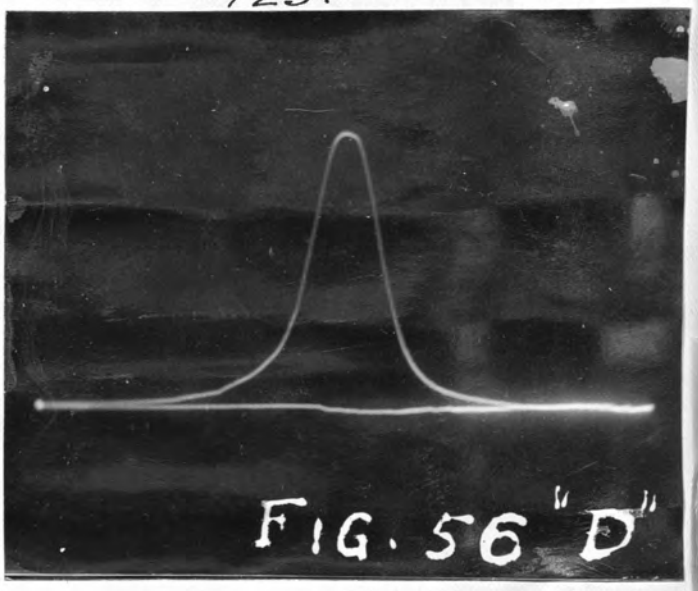


FIG. 56 "D"

670.

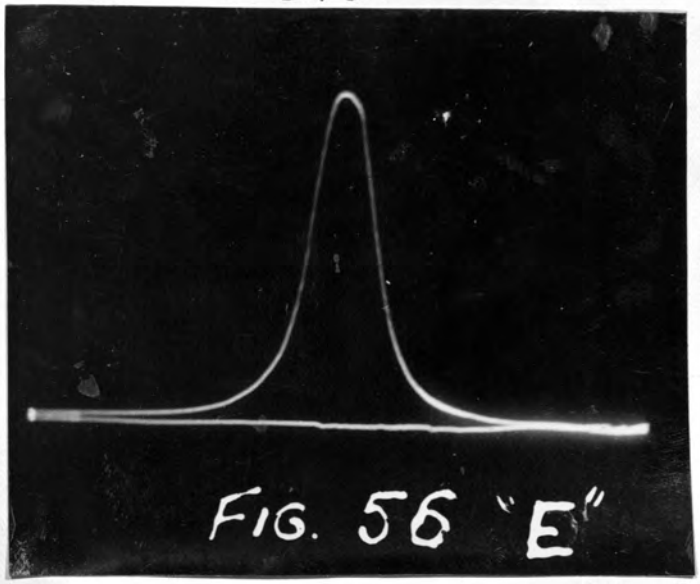


FIG. 56 "E"

600.

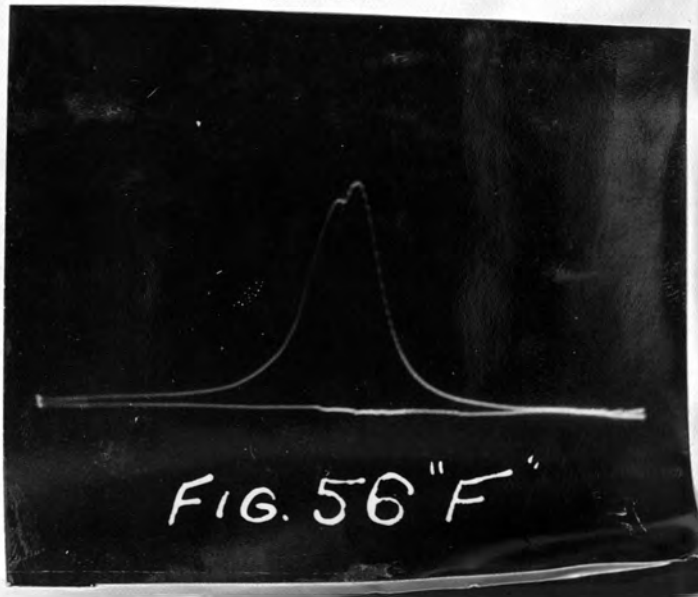


FIG. 56 "F"

In Fig. (56) the six diagrams are:-

	Injection Timing of Pilot.
A	0^{o} T.D.C.
B	10^{o} before T.D.C.
C	20^{o} " " " "
D	40^{o} " " " "
E	60^{o} " " " "
F	130^{o} " " " "

The quantity of fuel delivered by the main charge was 0.813×10^{-4} lb. per cycle, while that delivered by the pilot was 0.7168×10^{-4} lb. per cycle. A and B show a striking appearance of the pressure rise. The main pump was delivering fuel at T.D.C.

C appeared to be much smoother than A, while D and E are quite smooth. A small hump appears in F.

It appears from these diagrams that the pilot charge certainly smoothed out all the rapid pressure rise when pilot injection was about 40^{o} - 50^{o} before T.D.C.

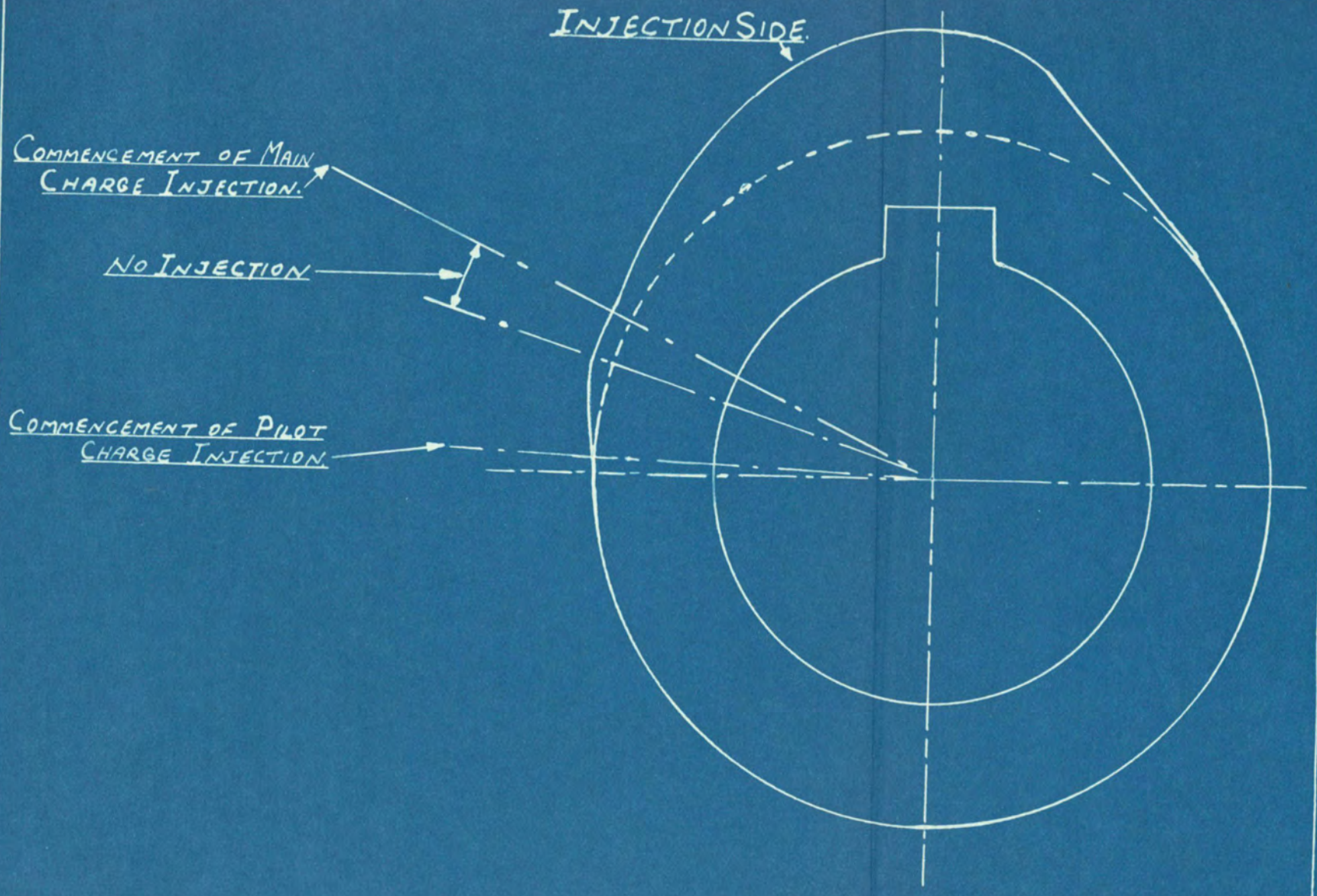
The reason suggested for eight-stroking when main (and only) injection was fully retarded (at about 12^{o} after T.D.C.) and the speed was at the region of 900 R.P.M. is

as follows:-

The temperature at the time of injection, due to late injection and low speeds, is not high enough to ignite the fuel. When the piston clears the oil charge out on the exhaust stroke, some of the charge is blown through the exhaust valve, and a little of it clings on its stem. The remainder, a small quantity, is left inside the cylinder. This acts as a pilot charge and ignites the fuel injected in the following cycle. The fuel which burns at the exhaust valve stem, leaves a carbon deposit on it, and after a while the exhaust valve keeps off its seat and causes the engine to shut down.

The author also suggests the following practical procedure for utilising the results of these tests:-

The cam in Fig. (57) is now suggested. This cam may operate the fuel pump. A small quantity is injected at first (about 20% of the maximum). This should be followed by the remainder of the charge 40^{o} later. The injection may commence about 45^{o} before T.D.C. For smooth running purposes, this cam saves the installation of two pumps, which are expensive items.



D. J. 1936

FIG. 57.

PART II.

SECTION "C".

THE BEHAVIOUR OF DIFFERENT FUELS AND A RATING SUGGESTION.

So far as the petrol engine is concerned, and according to the spontaneous ignition theory, the tendency to knock is dependent on the spontaneous ignition temperature (S.I.T.).

When the Diesel engine is considered, the above theory may hold in the reverse way. Thus knocking occurs more readily with aromatic than with paraffinic hydrocarbons, since the ignition temperatures of the former are higher; more readily with paraffins of low molecular weight than with higher paraffins for the same reason; also perhaps the flame temperature may slightly increase with the molecular weight,

It is customary to define the S.I.T. of a substance as the temperature at which that substance, under certain specified conditions, takes fire or explodes spontaneously without the application of spark or flame.

The S.I.Ts. of liquid fuels have been studied by a number of investigators, mainly in connection with

anti-knock properties of petrol and Diesel engine fuels. The investigators have used a variety of methods. The work done has not established definite S.I.Ts. for various substances, as the variables involved are very complicated.

The most common method used is to allow a small drop of fuel into a heated pot, often supplied with a stream of oxygen. The lowest temperature at which the drop ignites is taken as the S.I.T. Other workers have used adiabatic compression.

In reviewing what has been done in the direction of rating fuels for a Diesel engine, it would seem that not a great deal has been done in this direction. The S.I.T. found statically may not prove very useful, as the behaviour of a fuel sometimes differs when used in an engine, and when used under other conditions.

To compare different fuels for a Diesel Engine the following procedure was suggested, for some standard engine:-

1. The running conditions of the engine must be very nearly the same, particularly speed, fuel quantity injected inside the cylinder, volumetric efficiency, air inlet

temperature and water outlet temperature.

2. The injection timing should be varied over a wide range, before and after T.D.C., and the delay period, knock and torque should be noted.

3. The injection timing should be retarded until the critical angle is reached. At this angle the fuel ceases to fire regularly. This angle will be referred to as the Spontaneous Ignition Angle (S.I.A.), which is defined as the angle between the retarded position of the injection at which the fuel fires or ceases to fire, and the T.D.C. This angle, it was thought, may bear a relation to the quality of fuel used, inside a Diesel engine, so far as the delay period and Diesel knock are concerned. The author is not aware of any published results relating to this method of comparing and rating fuels.

TEST RESULTS AND DISCUSSION.

It was decided to do some further tests with the engine, using only one fuel pump. To start with, the pilot pump and nozzle were chosen so that a big range of injection timing could be tested. It was found that by admitting all the necessary fuel from the pilot nozzle, the exhaust was black in colour, and the engine was not developing more than 5 B.H.P. at 1200 R.P.M. This injection timing was about 10^{o} before T.D.C. On advancing the injection beyond 10^{o}, the engine was developing more power, but the knock was very heavy. On investigating the problem, the following points suggested themselves:-

The pilot nozzle contains one small hole and when the fuel is injected into the cylinder it only accumulates in one corner. This is due to the fact, as was mentioned before, that the small hole in the nozzle was drilled at an angle to its axis, so that the fuel takes the same direction as that of the hole, i.e. at an angle to the axis of the cylinder head.

The air inside the combustion chamber, although in motion, does not seem to reach the fuel and burn it

in the short time available. This would appear to follow from the fact that by advancing the injection the load increases considerably, but the engine knocked badly.

To overcome the difficulty mentioned, the pilot pump was connected to the main nozzle, which contained four holes. This nozzle proved most satisfactory. From what has been mentioned, it seems that to ensure a satisfactory combustion, the fuel should be distributed evenly inside the combustion chamber, even when directed air swirl is employed.

To test the behaviour of different fuels with the single pump, when the injection timing was varied over a wide range, the following fuels were selected:-

(1) Diesel Oil (for specification, see Appendix.)

(2) Kerosene (" " " ")

(3) Dominion Petrol (Summer Grade) with an Octane number of 69. †

(4) Esso Ethyl (Summer Grade) with an Octane number of 89.

---

† Figures kindly supplied by Mr. T.G. Hunter of the Oil Mining Department of the University of Birmingham.

The speed of the engine was kept constant throughout the tests. Volumetric efficiency, air inlet temperature, quantity of fuel injected per cycle, and water outlet temperatures, were kept constant as before.

The torque, delay period and injection timing, were recorded. The critical angle at which the fuel ceased firing was particularly noted, and was repeated fairly accurately.

Fig. (58) and Fig. (59) shew the test results. Fig. (58) shews the delay period in relation to time of injection for the different fuels. Fig. (59) shews torque against injection timing.

The most satisfactory fuel of the fuels tested, so far as knock and power are concerned, seems to be the Diesel fuel, with Kerosene next. The petrols behaved most unsatisfactorily; the engine was hardly pulling any load, and combustion was irregular. The knock was terrific at advanced injection. Fig. (59)(A) shews two out-of-phase diagrams which may give some idea of the roughness of this fuel.

It may be of interest to mention that it was noticed the bigger the S.I.A. the smoother the running of

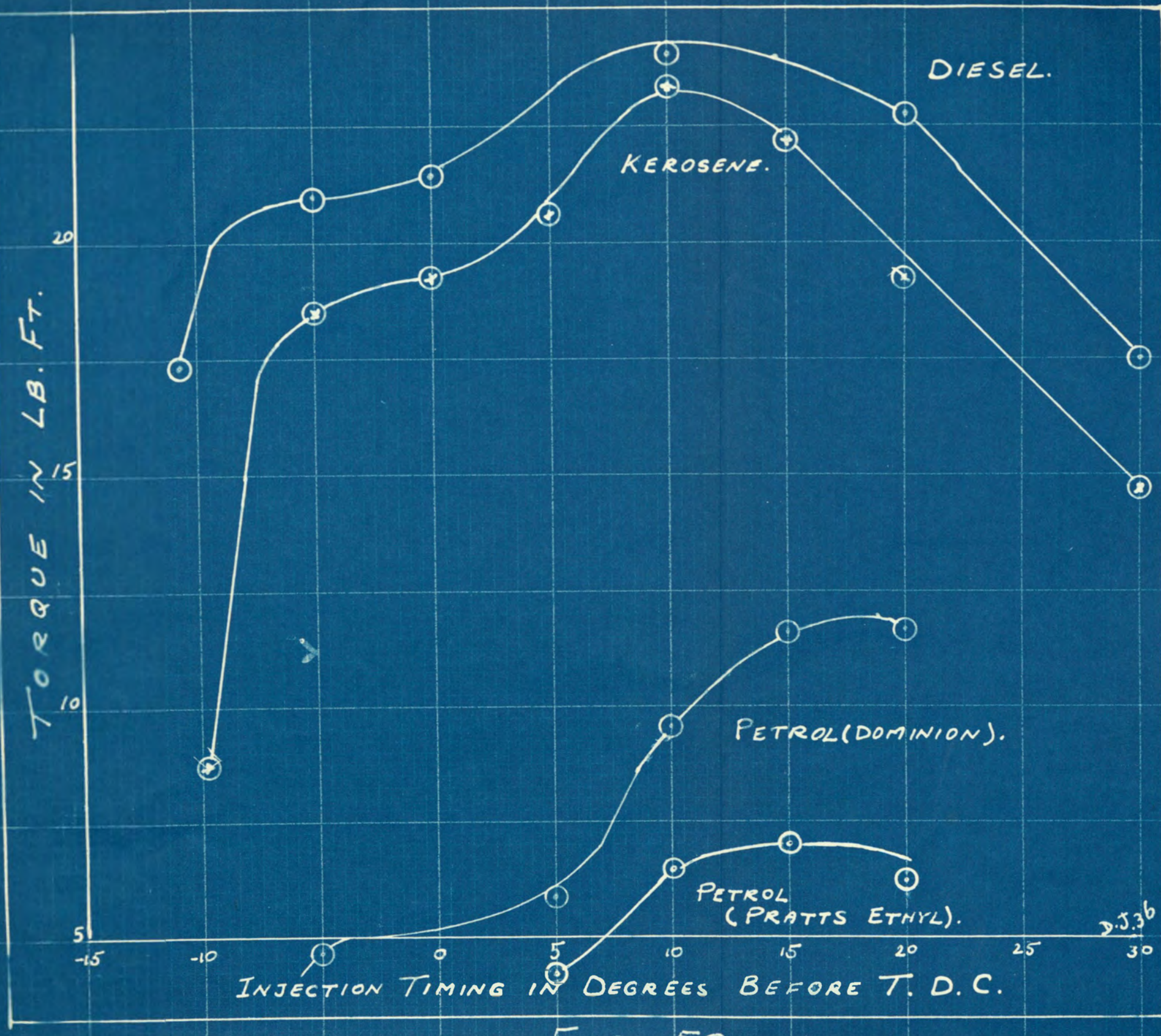

DIESEL.
KEROSENE.
PETROL (DOMINION).
PETROL (PRATTS ETHYL).
TORQUE IN LB. FT.
20
15
10
5
-15
-10
0
5
10
15
20
25
30
INJECTION TIMING IN DEGREES BEFORE T. D. C.
D.J.36

FIG. 59.

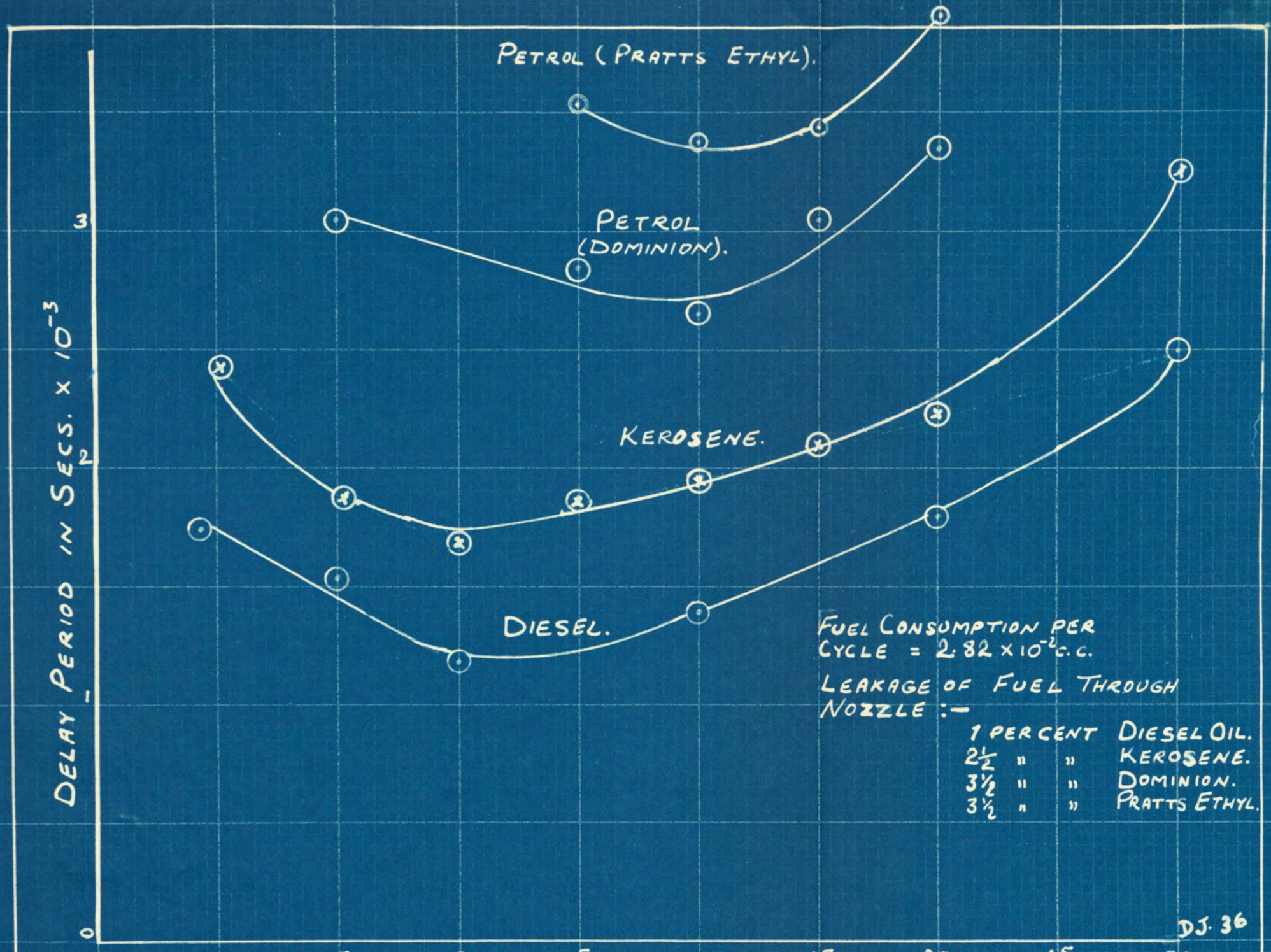

PETROL (PRATTS ETHYL).
PETROL (DOMINION).
KEROSENE.
DIESEL.
DELAY PERIOD IN SECS. x 10^{-3}
0
1
2
3
-15
-10
-5
0
5
10
15
20
25
30
INJECTION TIMING IN DEGREES BEFORE T.D.C.
FUEL CONSUMPTION PER CYCLE = 2.82×10^{-2} C.C.
LEAKAGE OF FUEL THROUGH NOZZLE :-
1 PER CENT DIESEL OIL.
2½ " " KEROSENE.
3½ " " DOMINION.
3½ " " PRATTS ETHYL.
DJ. 36

FIG. 58.

the engine, so far as the ear could tell. The writer does not wish to stress this point, as further evidence is necessary.†

The following are S.I.As. as measured:

	S.I.A.
Diesel Oil.	12.5°
Kerosene.	10.5°
Dominion Petrol.	6.0°
Esso Ethyl.	5.5°

---

† This research will probably be continued in the Mechanical Engineering Department.

It may be of interest to note that on examining the two indicator diagrams in Fig. (59 "A") and some of the previous diagrams, where knock was severe, a wave is noticed during combustion. This wave continues along the expansion side of the indicator diagram. The amplitude of this wave is small.

In a petrol engine, Lloyd Withrow and Gerald M. Rassweiler,(38) observed a shock wave when the engine knocked severely. By comparing the results of the flame and pressure records for the same engine explosion, they found that the frequency of shock corresponded well in the measurements. By the frequency of shock wave is meant the number of times per second that the pulse travels across the combustion chamber and back.

The above statement may throw some light on the nature of knock in a Diesel engine. It seems probable that knock in a Diesel engine is of a similar nature to that of a petrol engine; Possibly it occurs when a wave is set up inside the combustion chamber.

The frequency of this wave, as indicated in Fig. (59A)I seems to be about 2800 cycles per second.

The diameter of the combustion chamber=$4\frac{3}{8}$".

∴ the velocity of this wave =

$$\frac{2800 \times 4\frac{3}{8} \times 2}{12} \text{ ft. per second.}$$

$$= \begin{matrix}(2044)\\ \cancel{2044}\end{matrix} \text{ ft. per second.}$$

This is approximately equal numerically to the velocity of sound inside the combustion chamber during combustion.

C O N C L U S I O N.

Finally, the author wishes to conclude with some of the factors affecting combustion in a Diesel engine. It seems that for a satisfactory combustion, the air inside the combustion chamber should be kept in motion, preferably as directed swirl. An optimum value of this in all probability exists. The fuel, however, must also be directed so that it is more or less evenly distributed inside the combustion chamber, particularly with the open type in question.

Eight-stroking is a dangerous phenomenon to the running conditions of an engine, particularly of the poppet valve type, and it ultimately shuts the engine down in a short period.

The delay period in time varied with volumetric efficiency, while other conditions were kept nearly constant. It also varied with the speed of the engine. It is exceedingly probable that the delay period controls the Diesel knock.

The introduction of a small quantity of fuel (Pilot injection) at about 45^{o} before T.D.C., to be followed by the main charge 35^{o} - 40^{o} later seems to improve the running conditions of the engine. The knock was smoothed somewhat with the above arrangement. It was also noticed that the engine started much easier with the pilot charge in operation, particularly when the main charge was in a retarded position. †

The spontaneous ignition angle may bear a relation to Diesel knock, and it seems to offer a simple method for rating fuels. With this method the delay period may play an important part.

---

† S.I.A. is defined as the angle between the retarded position of the injection at which the fuel ceases firing, and the T.D.C.

A P P E N D I X I.

SPECIFICATION OF FUELS.

(1) The Diesel Fuel.

For the Diesel fuel used, the test data were supplied by Messrs. Shell-Mex and B.P. Ltd.

Approximate Analysis and Specification:-

Carbon	86.0%
Hydrogen	13.0%
Sulphur	0.7%
Undetermined Oxygen, Nitrogen, etc.	0.3%
	100.0%

Specific gravity at 60°F. = 0.84

Flash point (closed) = 200°F.

Higher Calorific Value. B.Th.Us. per lb. = 19600.

Carbon residue (Conradson) = 0.01%

Asphalt, Ash, Water and Sediment contents are negligible.

Viscosity Redwood No. 1. at 100°F. 35 secs.

Distillation.

Initial Boiling Point	220°C.
Final " "	360°C.
Distilling to 300°C.	70°C.

(2) Kerosene.

The Kerosene used was supplied by Messrs. Mathews' Oils Ltd., and the following data were supplied by them.

Specification.

Specific gravity at 60°C	0.805
Initial Boiling Point at	171°C
1% "	175°C
15½% "	200°C
53½% "	225°C
81½% "	250°C
95% "	275°C
Final Boiling Point at	294°C
Recovery	99%
Residue	1%
Loss	Nil.
Closed Flash Point	130°F.

A P P E N D I X II.

CALCULATION OF VOLUME OF AIR AND VOLUMETRIC EFFICIENCY.

A rounded orifice was employed on the suction side of the engine for calculating the volume of air used. The orifice was 1 " diameter. The pressure drop across the orifice was measured to start with by an inclined gauge of 8 : 1 inclination. Later on this gauge was replaced by one having an inclination of 5 : 1.

For calculating the quantity of air (Q) used in cubic feet per minute, the following method was employed:-

Let: A = Area of orifice in sq.ft.

K = Coefficient of discharge through orifice, (taken as 0.604).

H = Pressure drop across the orifice.

V = Velocity through the orifice in feet per second.

$H_1 - H_2$ = Difference in pressure between two sides of orifice.

$$\therefore \quad H = \frac{H_1 - H_2}{\text{Density}} = \frac{H_1 - H_2}{P} \times RT.$$

Assuming the compressibility is negligible

$$\therefore \quad v^2 = 2gH$$

or on substituting,

$$\therefore \quad v = \frac{eg(H_1 - H_2)}{P} RT.$$

One inch pressure drop = 5.2 lbs. per square foot.

Let L = pressure drop across the orifice in inches water.

$\therefore$ 5.2 L = pressure in lbs. per square foot.

$$\therefore \quad v = 2gA\ 5.2n \frac{RT}{P}$$

Now $Q = 60\ KAV$

$$= 60KA\ \ 2g \times 5{,}2\ n \frac{RT}{P}$$

substituting

$$= 4.15 \frac{n.T}{P} \quad . \text{ Taking } T = 293^{o} \text{ abs.}$$

and P = 14.55 lbs./sq.ft.

$$\therefore \quad Q = 4.13\ n_1$$

For inclination of 8 : 1

h_1 in cms. of water on the inclined gauge.

$$= 5.22\ h_1$$

" " " " " " " " "

Fig. (60) shews the relation between Volume of air used and readings on the inclined gauge.

For calculations of the volumetric efficiency all the values are for the air drawn into the cylinder, taken as a ratio (when the air is at 20 ^{o} C. and the atmospheric pressure as 14.55 lbs. per square foot) , to that of the volume swept by the piston. This is not the volumetric efficiency reckoned on a charge at N.T.P.

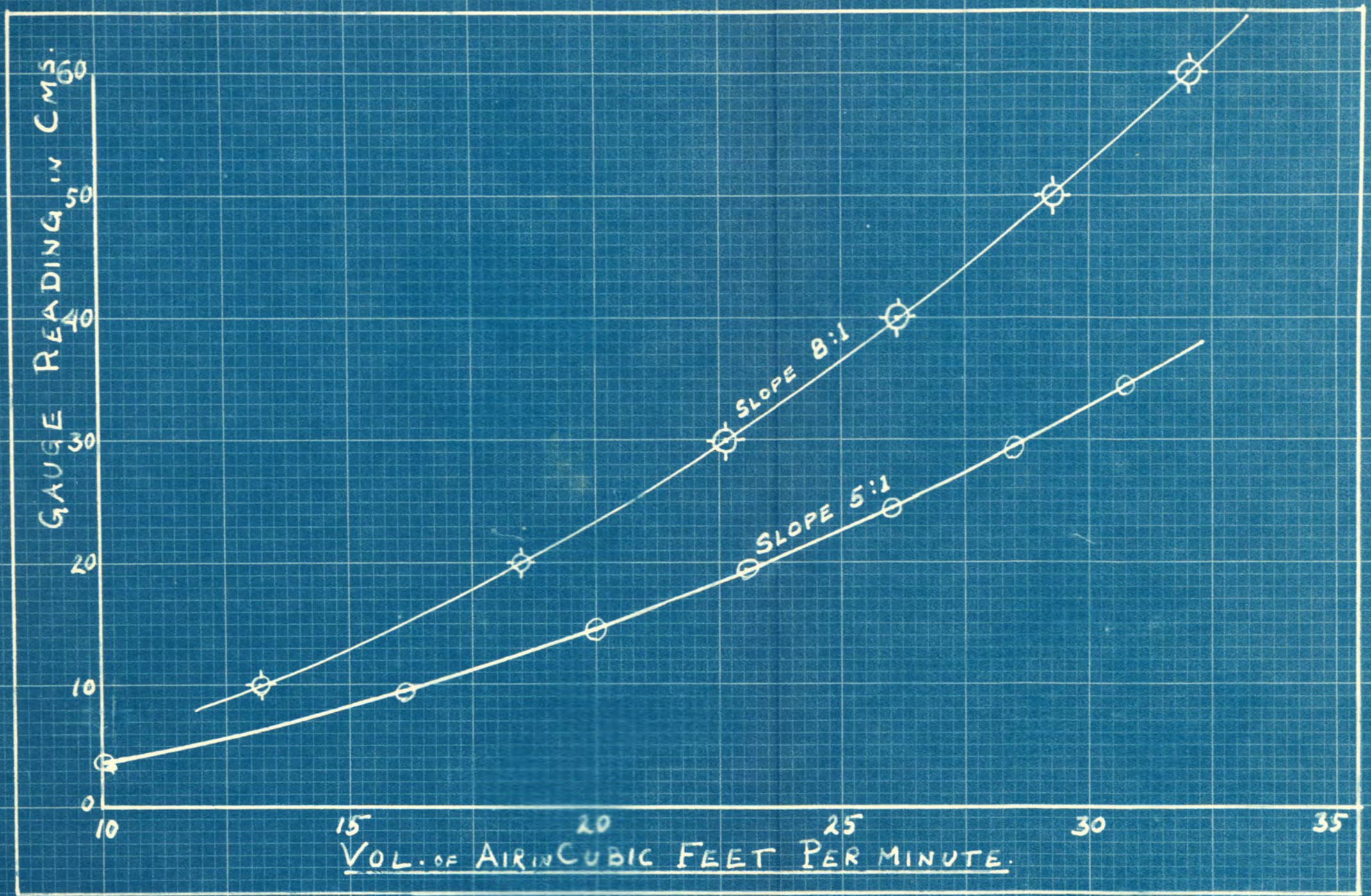

GAUGE READING IN CMS.
60
50
40
30
20
10
0
SLOPE 8:1
SLOPE 5:1
10
15
20
25
30
35
VOL. OF AIR IN CUBIC FEET PER MINUTE.

FIG 60

B I B L I O G R A P H Y.

	Title.	Author.
1.	"High Speed Diesel Engines.	Heldt.
2.	"Fuel Injection System for High Speed Oil Engines".	H.O. Farmer, M.Sc., B.Sc., A.M.I.Mech.E. & J.F. Alcock, B.A., A.M.I.Mech.E., Proceedings I.Mech.E. 1931.
3.	"Some Problems Connected with High Speed Compression Ignition Engines."	C.B. Dicksee. Proceedings I.A.E. 1931-32.
4.	"The Internal Combustion Engine". Vol. I.	H.R. Ricardo, B.A., F.R.S.
5.	"The Compression Ignition Automobile Engine".	S.W. Nixon, Graduate I.A.E. Proceedings I.A.E. 1930 - 31.
6.	"High Speed Diesel Engines".	A.W. Judge.

BIBLIOGRAPHY - CONTINUED.

	Title.	Author.
7.	"Some Notes and Observations on Petrol and Diesel Engines".	H.R. Ricardo, B.A., F.R.S. Proceedings I.A.E. March 1933.
8.	"Injection, Ignition and Combustion in High Speed Heavy Oil Engines".	Dr. S.J. Davies and E. Giffen. Proceedings I.A.E. March 1931.
9.	"The Internal Combustion Engine". Vol. I.	D.R. Pye.
10.	"Combustion in Diesel Engines".	H.R. Ricardo, B.A., F.R.S. Proceedings I.A.E., March 1930.
11.	"Flame Movement and Pressure Development in Gasolene Engines".	Clarke. C. Minter. S.A.E. Journal, March 1935.
12.	"High Speed Heavy Oil Engines, their Characteristics and the Trend of their Development".	S.I. Davies, Ph.D., M.Sc., B.Sc.(Eng)., M.I.Mech.E. Proceedings I.Mech.E. 1932.

BIBLIOGRAPHY - CONTINUED.

Title.	Author.
13. "The Progress of the Internal Combustion Engine and its Fuel".	H.R. Ricardo, F.R.S. "The Engineer", October 25, 1935.
14. "Penetration of Oil Sprays in Dense Air".	P.H. Schweitzer. "Gas and Oil Power", Nov. 1935.
15. "The Heavy Oil Engine on Road and Rail".	A.E.L. Chorlton. Proceedings I.A.E. 1929.
16. "The High-Speed, Compression-Ignition, Heavy-Oil Engine for Aero-Engines".	F.R. Banks. Proceedings I.A.E. November 1929.
17. "The Minimum Regular Discharge of Jerk-Pump Fuel Injection System for Diesel Engines.	Theodore B. Hetzel. "Gas and Oil Power". November 1935.
18. "Estimation of Combustion Products from the Cylinder of Petrol Engines and its Relation to Knock."	A. Egerton, F.R.S. F.Ll. Smith, D.Phil., and A.R. Ubbelohde, M.A. Philosphical Transactions of the Royal Society of London, 30th July 1935.

BIBLIOGRAPHY - CONTINUED.

	Title.	Author.
19.	"High-Speed Diesel Engines".	H.R. Ricardo, M.I.A.E. Proceedings I.A.E., 1928 - 29.
20.	"Piston Temperature in a Solid-Injection Oil-Engine".	G.F. Mucklow, D.Sc., A.I.Mech.E., Proceedings I.Mech.E., Dec. 1932. Vol. 123.
21.	"Air Swirl in Oil Engines".	J.F. Alcock, B.A., A.M.I.Mech.E. Proceedings I.Mech.E., 1934. Vol. 128.
22.	"Combustion in a High Speed Diesel Engine".	H. Faridany, B.Sc., M.Sc., Thesis submitted for the Degree of M.Sc., at Birmingham University.
23.	"Some Points in Combustion Efficiency of Diesel Engines".	Irs. G.D. Boerlage and J.J. Broeze. "Gas & Oil Power", Jan.1935.

BIBLIOGRAPHY - CONTINUED.

	Title.	Author.
24.	"The Correlation of Tests on the Ignition Quality of Diesel Fuels".	Ir. G.D. Boerlage, Ir. J.J. Broeze, L.J. Le Mesuria, and R. Stansfield. Journal I.P.T. July 1936.
25.	"The Diesel Engine in Road Transport".	W.H. Goddard. Proceedings I.A.E. 1929-30.
26.	"Flame Propagation. Slow-motion Pictures of knocking and non-knocking Explosions Recorded with a Speed Camera".	Lloyd Withrow and Geral M. Rassweiler. The Automobile Engineer, August 1936.
27.	"High Speed Diesel Engines for Marine Service".	H.R. Ricardo, B.A., F.R.S., M.I.Mech.E., Proceedings I.Mech.E., 1933.
28.	"Spark Photography of Fuel Sprays, and High-Speed Motion Picture Study of Combustion in a Compression Ignition Engine".	R. Stanfield. Abstract of four Papers published by N.A.C. for Aeronautics, U.S.A. Journal I.P.T., May 1936.
29.	"Supercharging Marine Oil Engines".	Sterry B. Freeman, C.B.E., M.Eng., M.I.Mech.E. Proc. I.Mech.E., 1935.

BIBLIOGRAPHY - CONTINUED.

Title.	Author.
30. "Some Tests on a Two-Stroke Cycle Oil Engine.	E.A. Allcut, Professor of Mechanical Engineering University of Toronto Bulletin, No. 6, 1926. Section No. 13.
31. "A few Points Against Supercharging".	H.R. Ricardo, B.A., F.R.S., M.I.Mech.E., Proceedings I.Mech.E., 1935. (Vol. 129).
32. "The Ignition Quality of Fuels in Compression-Ignition Engines".	G.D. Boerlage and J.J. Broeze. "Engineering", Nov.13, Dec. 4, and 18, 1931.
33. "Experiments on Oil Jets and Their Ingition".	A.L. Bird, M.A., Lecturer in Engineering, University of Cambridge. Proceedings I.Mech.E., 1926, Vol. II.
34. "Different Methods of Assessing the Ignition Quality of Diesel Fuels.	F.H. Garner, M.Sc., Ph.D., F.I.C. Journal I.P.T., July 1936.

BIBLIOGRAPHY - CONTINUED.

Title.	Author.
35. "Cathode-Ray Oscillograph Engine Indicator.	"Engineering", August 23, 1935.
36. "The Principles of Motor Fuel Preparation and Application". Vol. II. Pages 736-800.	Prof. Alfred Nash, M.Sc., M.I.Mech.E., F.C.S., F.Inst. Fuel., and Donald A. Howes, B.Sc. Ph.D., A.M.I.P.T.
37. "Some Notes and Observations on Petrol and Diesel Engines".	H.R. Ricardo, F.R.S. Proceedings I.A.E. March 1933.
38. "Engine Knock." By L. Withrow, and G.M. Rassweiler.	"The Automobile Engineer." Aug. 1934.

Reprinted from GAS AND OIL POWER April, 1936

Experiments on Combustion in a High-speed Airless-injection Oil Engine

These researches were carried out at Birmingham University on a Tangye open combustion chamber engine and have particular reference to air swirl and the influence of valve shrouding upon combustion efficiency

By H. FARIDANY, M.Sc., and D. JAFAR, B.Sc.

This research on the combustion in a high speed Tangye oil engine has been carried out in the Mechanical Engineering Department, Birmingham University, under the supervision of Professor S. Lees, M.A. His help and criticism were much appreciated when deciding the line along which the research was to be conducted. The authors would like to express their thanks to Professor Lees and Mr. S. J. Ellis, M.Sc., for their valuable advice and assistance throughout the work.

SUMMARY

THE success of the heavy-oil engine of the airless-injection type depends a great deal on the combustion process. Rotational movement of the air is possibly the only successful means for a uniform distribution of the fuel inside the combustion chamber. Different ways have been used to produce this rotational movement, but up to now there has not been published a great deal of information on the principles of its production so far as the poppet-valve-type engine is concerned.

From the experiments carried out by the authors, one can come to the conclusion that an optimum air swirl may exist with an open-type combustion chamber and a shrouded inlet valve. The jet of the air and its direction, on entering the cylinder, may influence the fuel consumption. Any pocket inside this type of combustion chamber may upset the process of combustion. Under all conditions the rotational movement of the air, which gives the oxygen of the air an easier access to the fuel, resulted in a fuel consumption much better than the indiscriminate movement produced when an unshrouded inlet valve was used.

Process of combustion

Satisfactory combustion in a high-speed diesel engine is a matter of particular importance both for aviation and road use. Generally speaking, a smoky exhaust is a sign of incomplete combustion of the fuel. Naturally, with this, carbon will be deposited on the combustion chamber, exhaust valve, atomiser holes, and around the exhaust pipe ; these faults may even cause the engine to cease functioning.

In a petrol engine, a homogeneous mixture exists in the combustion chamber at the end of the compression stroke, while in a diesel engine air is compressed alone, and injection only starts a few degrees before top dead centre is reached. To obtain satisfactory combustion within the short space of time available for burning the fuel completely, thorough mixing of the air and the fuel is essential ; this can be done in two ways :—

(1) The directed jet method, which necessitates a very elaborate design of the fuel nozzle to operate at a pressure of 5,000 to 8,000 lb. per sq. in.

The question of fuel compressibility also arises, which makes the work of the engine at variable speeds unsatisfactory. As the liquid fuel can only take the straight line direction on leaving the nozzles, to allow it to mix with the air thoroughly, and to ensure uniform distribution, a number of very small holes varying in size must be used, so that the injected fuel reaches different distances in the combustion chamber at the same time. Apart from the difficulty of maintenance of the nozzle holes, the rate of pressure rise becomes quite uncontrollable, due to the presence of a large quantity of fuel at the end of the delay period.

(2) The directed air method, which has two advantages over the first method : (*a*) It is easier to direct the air ; (*b*) fuel is injected at much lower pressure, of the order of 1,000 to 2,000 lb. per sq. in.

Ricardo, in a paper read before the Institution of Automobile Engineers in March, 1930, divided the process of combustion in a compression-ignition engine into three phases. In this connection, Fig. 1 is an actual out-of-phase diagram taken* by means of an electrical indicator. Fig. 2 is the theoretical diagram as suggested by Ricardo.

Phase 1 is the delay period, during which fuel is admitted but either no ignition takes

* At Birmingham University on the Tangye high-speed oil engine.

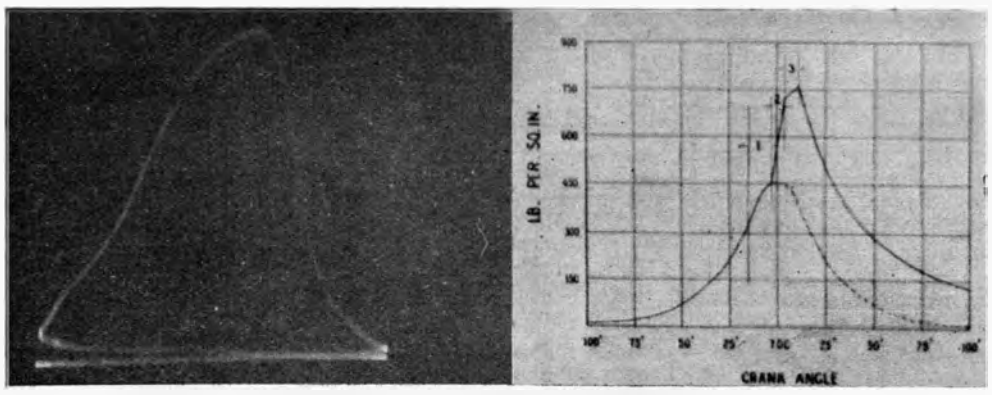


Fig. 1

An actual out-of-phase diagram taken on the Tangye engine

Fig. 2

The theoretical diagram as suggested by Mr. H. R. Ricardo

place or the ignition is restricted to some very localised nuclei. It seems probable that a minute quantity of the fuel, which has had time to mix with sufficient oxygen, while in a vaporised condition, forms a readily combustible mixture. From these nuclei the flame spreads at first in almost the same manner as it would in a petrol engine with a number of sparking plugs. Ricardo regards this delay period as constant in time, and as only depending on the nature of the fuel, the temperature and pressure after compression, and fineness of pulverisation.

Phase 2 is the spread of flame which, as in the petrol engine, depends on turbulence ; fortunately this is appreciably constant in terms of crank angle but not of time. The actual pressure attained after ignition has started depends on the quantity of the fuel present in the cylinder, the rate of injection, and the temperature and pressure of the highly compressed air.

Phase 3.—At the end of the second period, the temperature and pressure are high enough to burn the fuel as it enters the cylinder. The burning is under direct mechanical control, and the pressure variation depends upon the ratio of injection and turbulence.

Pye, in his book " The Internal Combustion Engine," states that as soon as the delay period—which is constant in time—is over, the rate of combustion is directly dependent upon turbulence. Without turbulence the rise of pressure would be so slow that combustion could not be completed in the available time. Ricardo(1) and Alcock's(2) experiments on a sleeve-valve engine showed that a rate of swirl of about ten times the r.p.m. gave the most satisfactory fuel consumption.

Researches made

It was felt that a certain amount of light would be thrown on the subject of combustion in a compression ignition engine by varying the speed of the inlet air and its direction. A number of shrouded valves were used to meet this requirement. It was anticipated that various points might arise in the course of investigation which would be worth pursuing, and this was actually found to be the case.

Method used for varying direction and velocity of air.—An experimental Tangye high-speed engine, as supplied by the makers, had at first an inlet valve fixed in certain positions with 120° of shroud, thus giving 240° of unshrouded opening. The authors proposed to use ultimately two methods of varying the inlet velocity of the air (for the poppet valve type engine) :

(1) To vary the extent of shroud ;
(2) to vary the lift of the valve.

Only the first method is described in the present paper, and the following shrouded valves were used :—

Valve No.	I.	120° of shroud.	240° of opening.
,,	II.	240° ,,	120° ,,
,,	III.	280° ,,	80° ,,
,,	IV.	300° ,,	60° ,,
Unshrouded valve		0° ,,	360° ,,

Arrangements were also made for fixing the shrouded valve in eight different positions relative to the engine crankshaft centre line. Fig. 3 shows the eight different positions.

Experimental procedure

The engine used was a four stroke cycle, direct-injection Tangye oil engine, having a single cylinder $4\frac{3}{8}$ in. in diameter by $5\frac{3}{4}$ in. stroke and running at a normal speed of 1,200 r.p.m. Fig. 4 is a cross-sectional elevation of the engine.

The fuel pump and atomiser are of the Bosch type, and a separate lever was fitted to vary the speed of the engine. The combustion chamber is of the open type, but the piston and head are slightly domed ; the central injector has four holes. Fig. 5 shows clearly the arrangement of the combustion chamber and the position of the valves and atomiser.

Measurement of air supply

The air supplied to the engine was passed through a suitable metal air vessel fitted with a nozzle on its suction side. An inclined water gauge was used for measuring the pressure drop across the nozzle and the air supplied was calculated from the pressure drop. The air inlet temperature was measured in two places : (1) very near to the nozzle, and (2) very near to the inlet valve of the engine

Provided the volumetric efficiency is kept constant, the air inlet temperature has a big effect on the temperature at the end of compression, and also has a direct effect on the velocity of the air through the inlet valve. The temperature of the air was kept at about 20° C., except where otherwise stated. A valve was also fitted between the air vessel and the engine, for regulating the quantity of air used.

The B.H.P. was measured by means of a Heenan & Froude hydraulic dynamometer directly coupled to the engine. Unfortunately, a really reliable high-speed engine indicator was not available in time for the earlier experiments, and deductions had to be drawn from the B.H.P. and fuel consumption. Crankshaft speed was measured by a clutch-operated chain-driven revolution counter ; an extra tachometer was also fitted.

The delay period being practically independent of the engine speed, it follows that as the speeds increase, the injection must be advanced to establish the same conditions. To avoid this difficulty, the speed of the engine was kept constant throughout the tests at 1,200 r.p.m.

Cooling Water.—The outlet temperature

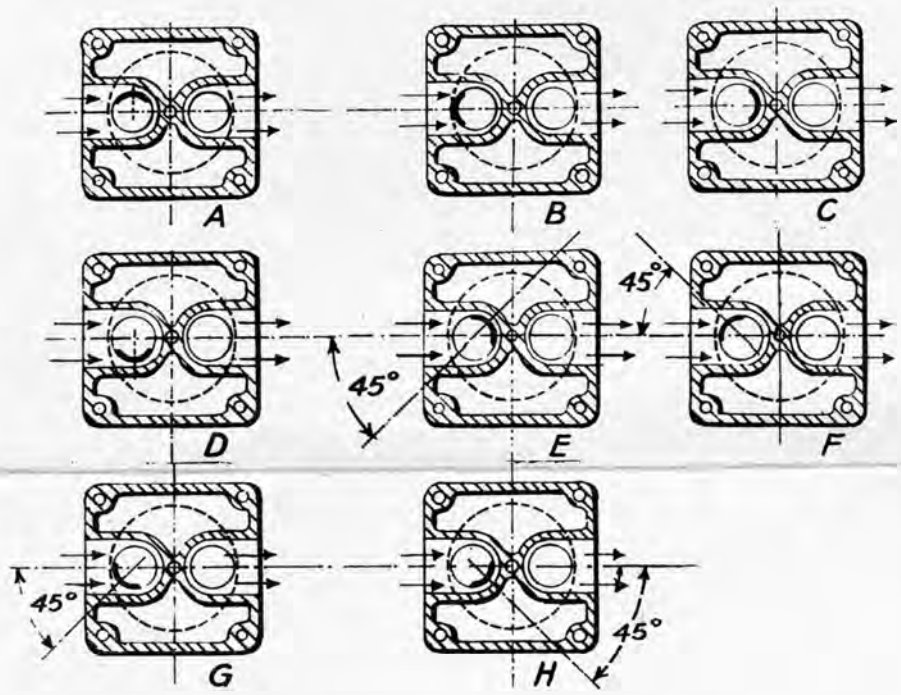


Fig. 3

The eight arrangements of shrouded inlet valve tested by the authors

Position A is the standard arrangement, but the best all-round results were obtained with the location shown in F

(1) Proceedings I.A.E., March 1930.
2) Proceedings I.Mech.E., 1934.

of the cooling water was kept at 60° C. throughout the experiments.

Injection timing

Heldt in his book entitled " High-Speed Diesel Engine " states that in general the efficiency of the diesel engine can be increased by advancing the injection, but unfortunately, this improvement is usually accompanied by roughness of operation (diesel knock). Under light loads there is an improvement without marked roughness. To eliminate the diesel knock at high loads the authors decided to retard the ignition throughout the tests.

The following diagrams show the effect of advancing the injection. The diagrams were taken when the B.H.P. was about 7, and the speed of the engine 1,200 r.p.m. A piezo-electric indicator* was made use of and a hydraulic calibration was carried out immediately after the diagrams were taken. Also, these diagrams were repeated and found to be consistent and very steady indeed. In any case, they are fairly reasonable for comparison purposes.

It will be seen from Fig. 3 that certain positions of the inlet valve are symmetrical, and it would be anticipated that such positions would give similar results as regards engine performance. The following tests were carried out to see if such was the case, the symmetrical positions being (A & D), (G & F), (E & H).

* Originally designed in the Mechanical Engineering Department, Birmingham University, by Dr. F. Postlethwaite, and further developed by Mr. H. L. Martyn.

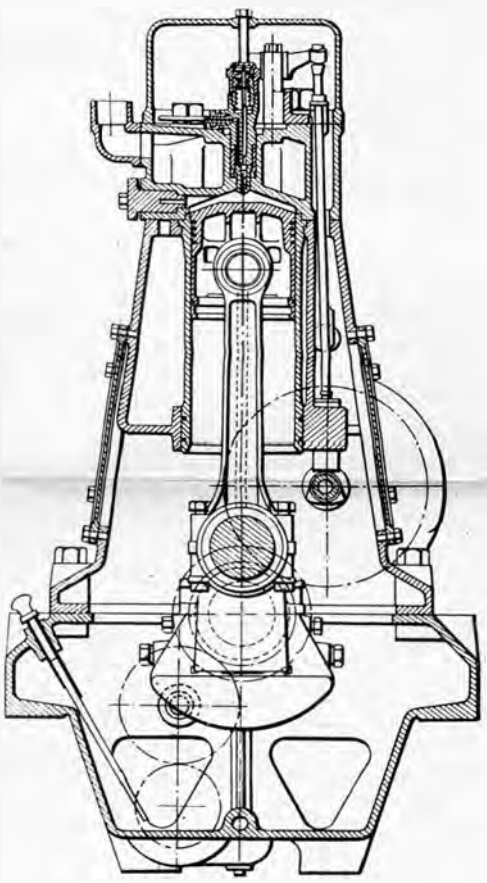


Fig. 4 (left).—Transverse section through the Tangye engine

Best valve position

Test Results. Valve No. 1 (120° of Shroud).—Tests were carried out at first with valve No. 1 in different positions. Results are shown plotted in Fig. 6. The curves show the relation between B.H.P. and fuel consumption.

Positions G and H are the two superimposed curves which give the best fuel consumption.

D is better than positions F and A, A being the standard position given by the makers, while C and B gave the worst fuel consumptions.

A saving of 10 per cent. in fuel consumption is noticeable at 12 B.H.P. when the valve is in position G and H, as compared with position A. The saving is much less at lower loads. The exhaust colour at the two positions G and H was very clear up to 12 B.H.P., while position A gave a clear exhaust up to 8 B.H.P. only.

It is interesting to note from these curves that symmetrical positions gave different results. It can also be seen from these curves that there is over 75 per cent. increase in the fuel consumption at 10 B.H.P. between positions G or H and B. This difference must be due to the variation of direction of air.

Test Results. Valve No. 2 (240° of Shroud).—When this test was carried out, a fall in volumetric efficiency was noted in comparison with valve No. 1. Fig. 7 shows that symmetrical positions again gave different results. Position F on the whole gave the best results, especially at high loads. E and G on the whole gave better results than A, while C gave the worst of all.

Valve 3 (280° of Shroud).—Fig. 8 shows test results for different positions. The fall of volumetric efficiency was very noticeable in this case. Positions E and A gave the best results, while C on the whole gave the worst fuel consumption. As can be seen, the lack of symmetry still existed.

Smoothing plug

Investigations were carried out to find the cause of lack of symmetry in combustion. The nozzle with its four holes was tested and found to be quite in order. Consideration of the cylinder head drawing (Fig. 5) showed that the only thing causing the lack of symmetry was the indicator plug hole. A special smoothing plug was therefore fitted so that symmetry in the combustion chamber was established. The smoothing plug was designed to fill up exactly the indicator hole. To eliminate hot spots, the smoothing plug was water-cooled. Valve No. 1 was tested again for the different positions and the results were plotted, Fig. 9. The effects of combustion symmetry were now found to hold. Comparing these curves with the curves obtained without the smoothing plug, Fig. 6 show an improvement in the fuel consumption of the order of 15 per cent. at 10 B.H.P.

On fitting the smoothing plug, the compression ratio was increased from 16·3 to 17·4 to 1, and accordingly the theoretical efficiency was increased by less than 1 per cent. There was also a slight improvement in the mechanical efficiency of the engine, due to the running in, of the order 2 to 3 per cent. The total improvement due to increase of compression ratio and mechanical efficiency cannot exceed 4 per cent., and it was clear, therefore, that the fitting of the smoothing plug had contributed a great deal towards the increase in the efficiency. The smoothing plug certainly removed the pocket inside the combustion chamber, which must have been an obstacle in the path of the swirling air at the end of the compression stroke. The air can now swirl at higher speeds, which in the case of this valve gave a better fuel consumption. Also, it must not be forgotten that part of the fuel enters the pocket (without the smoothing plug) and thus makes a bad contact with the moving air.

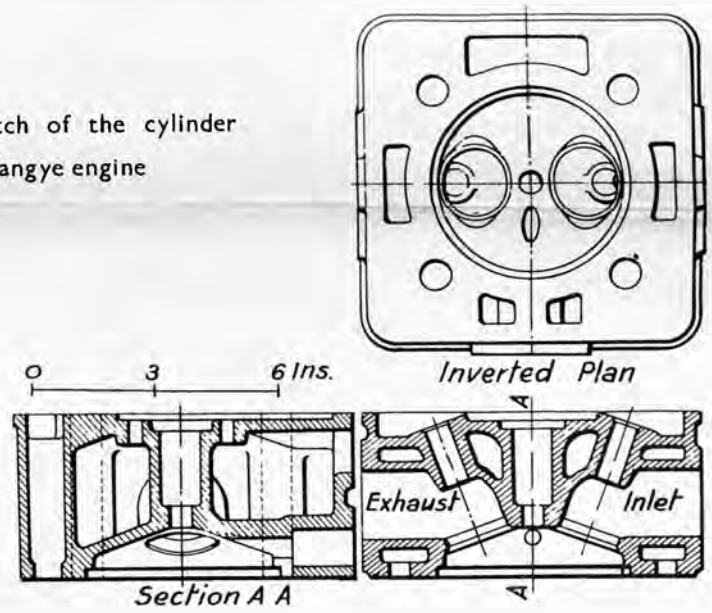


Fig. 5 (right).—Sketch of the cylinder head of the Tangye engine

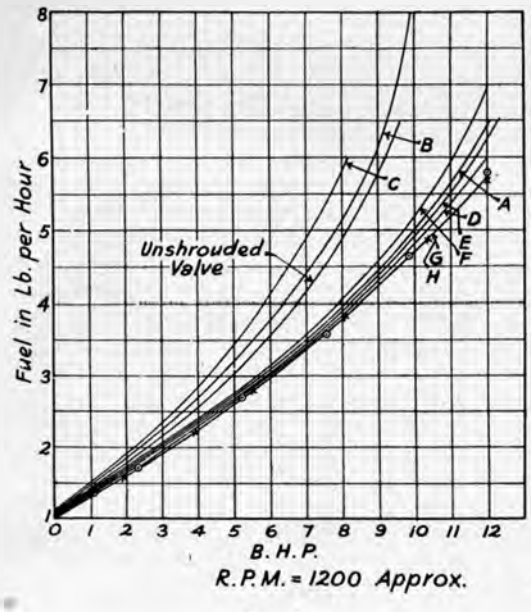


Fig. 6

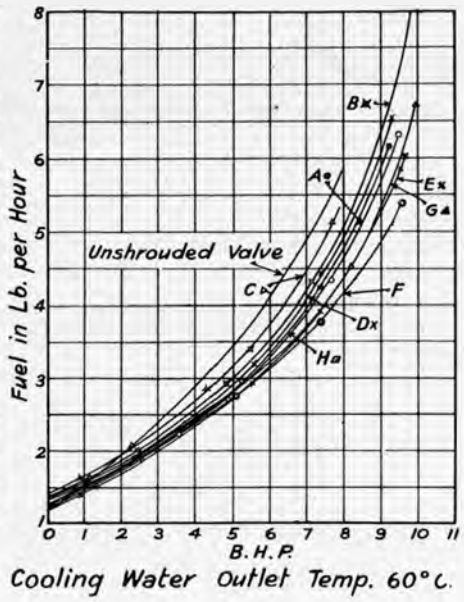


Fig. 7

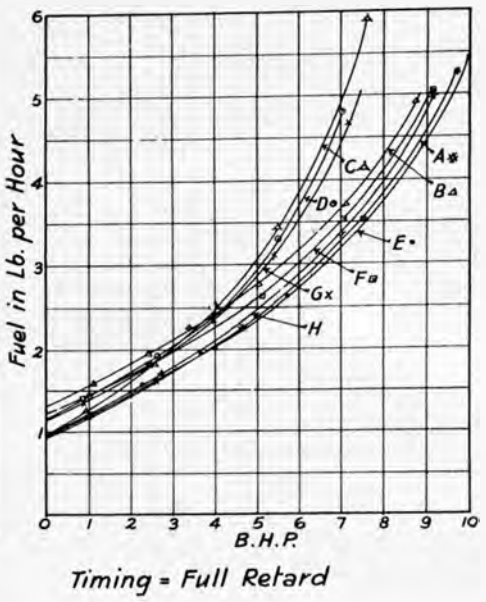


Fig. 8

It may be of interest at this stage to note that the fuel consumption at 9 B.H.P. is only 3·7 lb. per hour, or 0·411 lb. per B.H.P. per hour (position A, valve No. 1), which corresponds to a brake thermal efficiency of 31·5 per cent. Test results for valve No. 11 in Fig. 10 are still further proof of the establishment of combuston symmetry. Fig. 11 shows results for valve No. 3.

The smoothing plug made the fuel consumption higher for these two valves. In the best position for valve No. 2 the fuel consumption at 7 B.H.P. is shown as follows :—Without the smoothing plug 3·6 lb. per hour, and with the plug 4·1 lb. per hour.

For valve No. 3 the consumption without thesmoothing plug was apparently 3·2 lb. per hour, and with the plug 3·8 lb. per hour, both at 7 B.H.P.

The obvious explanation which can be offered in the case of these two valves, is the existence of an optimum swirl. (See later.) It should be added, however, that during the tests shown in Fig. 10, the tachometer drive was unsatisfactory, and so the thermal efficiencies are not guaranteed. The main point arising is that symmetrical positions gave similar results. The smoothing plug presumably helped the air to move about at higher velocity at the time of combustion, and this made affairs unsatisfactory for the combustion process.

(To be continued)

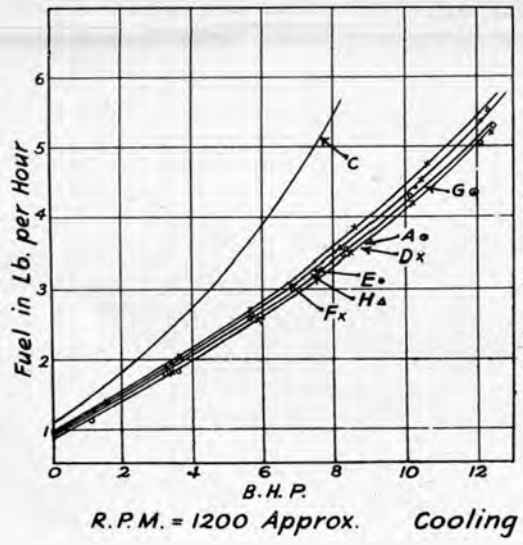


Fig. 9

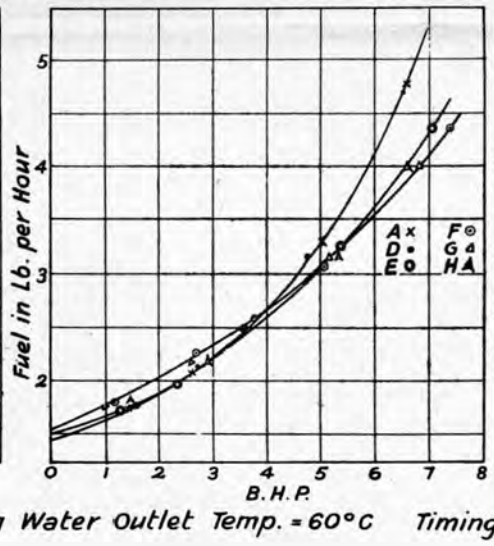


Fig. 10

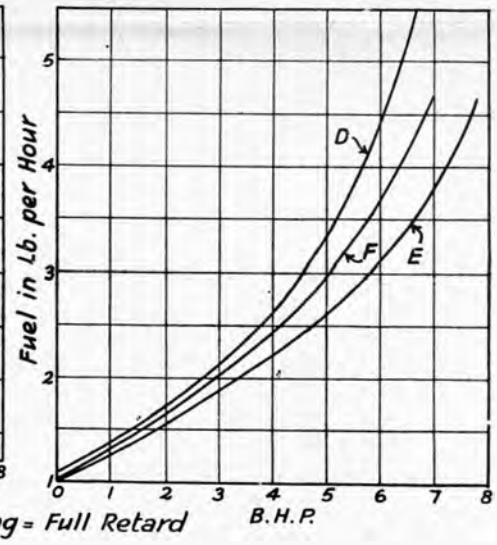


Fig. 11

Experiments on
Combustion in a High-speed Airless-injection Oil Engine

These researches were carried out at Birmingham University on a Tangye open combustion chamber engine and have particular reference to air swirl and the influence of valve shrouding upon combustion efficiency

(See pp. 89-92 of the April issue for the first part of this article)

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Question of volumetric efficiency

TO compare the test results for a number of valves with different openings, the volumetric efficiency should preferably be kept constant. A test was carried out to study the effect of variation in volumetric efficiency by throttling down on the suction side of the engine. Valve No. 1 was used in position "A." Fig. 12 shows the relation between fuel consumption and volume of air in cub. ft. per minute, at about 5·2 B.H.P., while the engine was running at 1,200 r.p.m. approximately. The air inlet temperature was about 20·5° C.

As can be seen from the curves in the above mentioned figure, the fuel per B.H.P. hour varies between 0·472 lb. to 0·544 lb. when the air drawn into the cylinder varied from 30 cub. ft. to 15 cub. ft. per minute respectively, corresponding to volumetric efficiencies of 1·0 to 0·5 on a basis of the prevailing atmospheric conditions. The exhaust temperature was noted with a platinum thermometer showing the effect of variation in volumetric efficiency on the exhaust temperature of constant load. The temperatures corresponding to the above mentioned figures were 292° C. and 507° C. respectively.

It was thought, therefore, that some means of fixing the volumetric efficiency must be devised. A blower was installed to charge the engine when the shrouded valves were tested. This method, however, gave considerable difficulty owing to the variation in the charging pressure. Due to the heating effect of the air in passing through the blower, and the effect of compression to a maximum of 7·5 lb. per sq. in., an air temperature of 56° C. was recorded just before entry to the inlet valve when valve No. IV was tested, the atmospheric temperature being about 18°C.

To make sure whether this rise of temperature would affect the results in any way, an electric heater was fitted to heat up the air before entering the cylinder. Fig. 13 gave the test results with valve No. IV in position A. A curve showing the relation between the inlet temperature of the air and the fuel consumption per B.H.P. is given. The load was kept constant, and so was the volumetric efficiency. The exhaust temperature was also recorded and plotted in the above-mentioned figure. As can be seen, the consumption at 40° C. is 0·73 lb. per B.H.P. per hour compared with 0·83 lb. per B.H.P. per hour at 70° C. A slight increase in the exhaust temperature was also observed. It was anticipated that as the inlet temperature increased, fuel consumption would decrease, as the fuel could burn more readily and combustion would be more complete. The reason suggested for the higher fuel consumption with valve No. IV (60° opening) is that the velocity of the air is much too high, even without heating. With heating, owing to the increased volume of air taken in, a still higher velocity is attained. It was therefore decided to throttle down on the induction side, when valves with big openings (small shrouding) were tested, so that the volumetric efficiency might be kept constant. In this method the authors have eliminated the effect of variation in the air inlet temperature, which in turn varies the velocity of the air through the inlet valve. Although the volumetric efficiency was low, it was constant throughout the experiments, and it is probable that the method is fairly accurate for

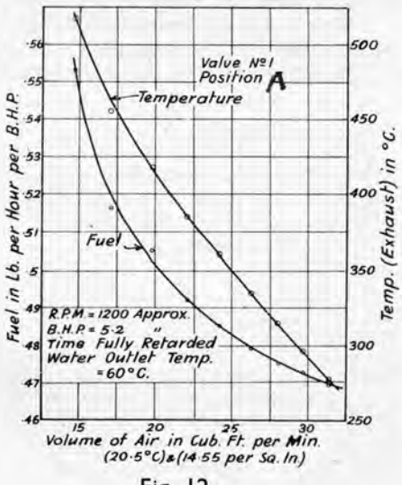


Fig. 12

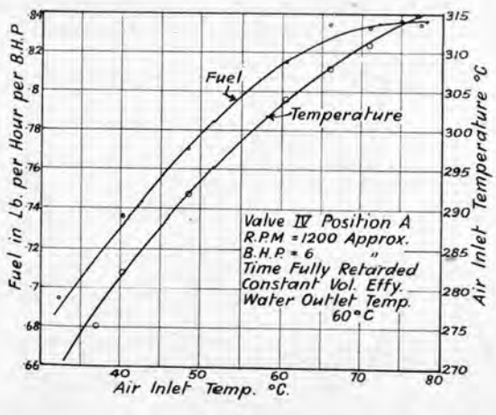


Fig. 13

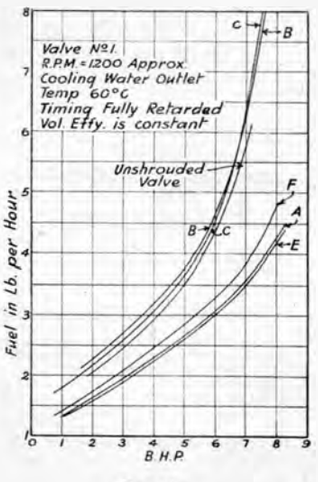


Fig. 14

studying the effect of turbulence on fuel consumption, when using different shrouded valves. The smoothing plug was used throughout the tests to be described.

The authors suggest that this method may be of some interest for aircraft compression-ignition engines with a high compression ratio. Throttling down while at low altitudes may be used. At high altitudes, normal run, full throttle may be made use of. It is realised, of course, that in an aircraft engine the exhaust pressure varies at different altitudes.

Test results

Valve No. I (120° of shroud, i.e. 240° of opening. Fig. 14 shows the results obtained for different positions. At 7 B.H.P., position E gave a fuel consumption of 3·5 lb. per hour, while for positions B and C, 6·2 lb. per hour of fuel consumption was recorded for the same B.H.P. The consumption for position F was 3·85 lb. per hour at 7 B.H.P It is interesting to note that the unshrouded valve gave a fuel consumption of the order of 5·85 lb. per hour at 7 B.H.P. These results show very clearly that for the same valve the fuel consumption can be almost doubled by varying the direction in which the air enters the cylinder, with the volumetric efficiency, inlet temperature of the air, and r.p.m. kept constant.

Valve No II (240° of shroud and 120° opening). The curves showing the relation between fuel consumption and B.H.P. are shown in Fig. 15. At 7 B.H.P. the fuel consumption for the different positions is as follows :—

F 3·53 lb. per hour.
E 3·82 ,,
A 4·03 lb. per hour.
B 6·1 ,, ,,
C 7·2 ,, ,,

These results still indicate the great variation in fuel consumption when the air on entering the cylinder is directed at different angles. Figs. 16 and 17 are results obtained from valves Nos. III (280° of shrouding) and IV (300° of shrouding), which also support the above statements.

Air direction and combustion

In Fig. 3, as was mentioned before, the inlet valve could be placed in eight different directions. Owing to the presence of symmetry, these could only be treated as five different positions, namely, A, B, E, F and C. Thus, by placing the inlet valve in any of these positions the air could be given five different directions.

Fig. 14 has been replotted on a different

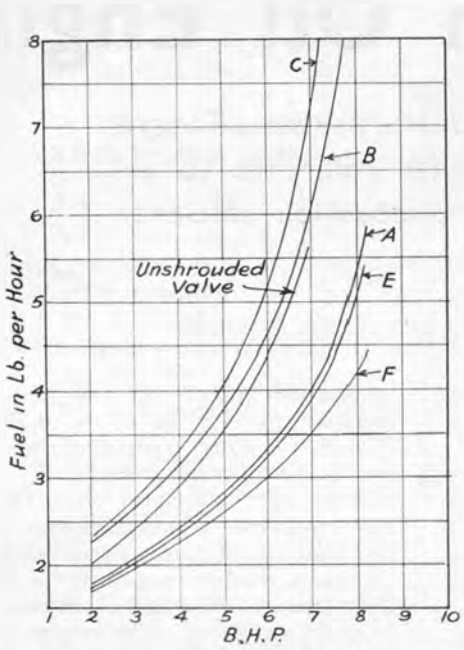


Fig. 15

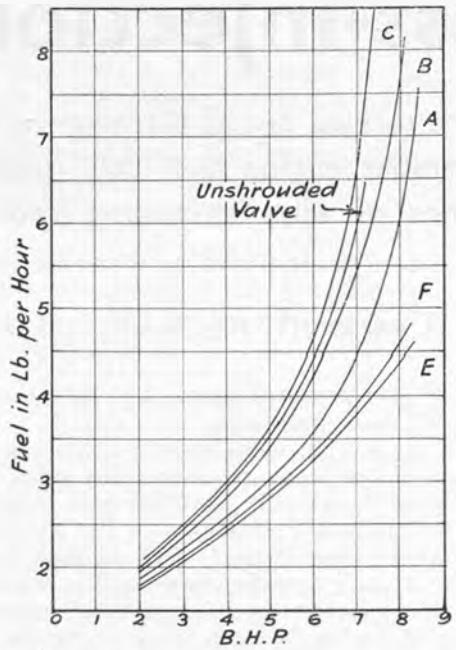


Fig. 16

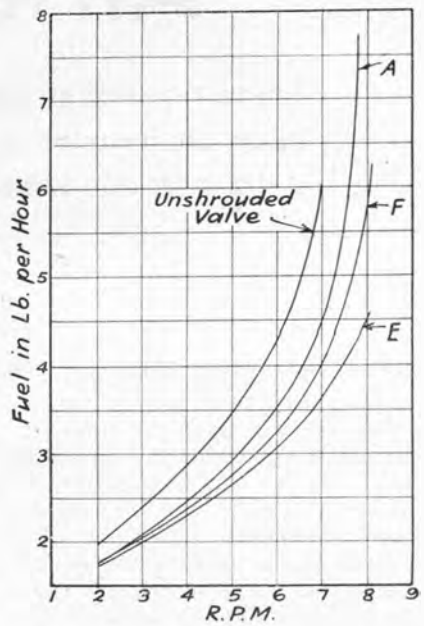


Fig. 17

R.P.M. = 1200 Approx. Cooling Water Outlet Temp. = 60°C. Timing Fully Retarded Constant Vol. Effy.

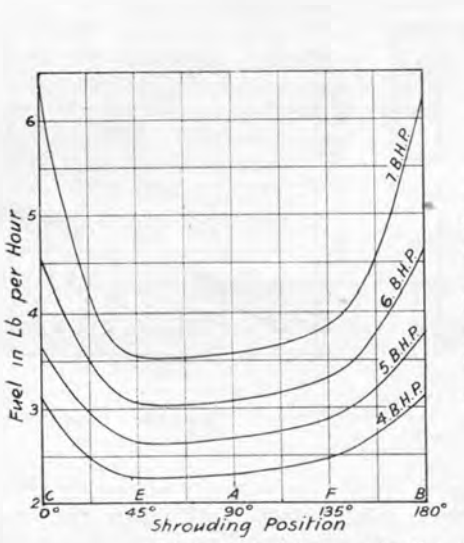


Fig. 18

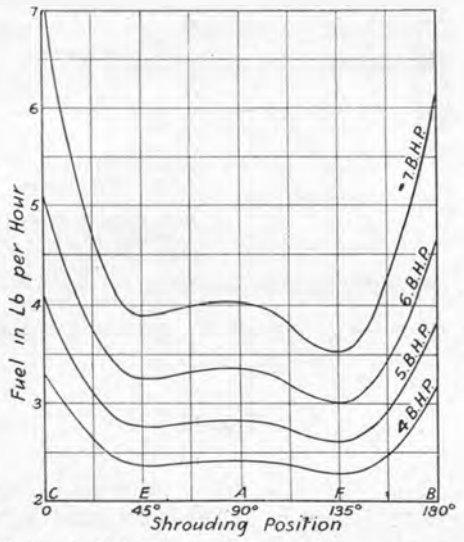


Fig. 19

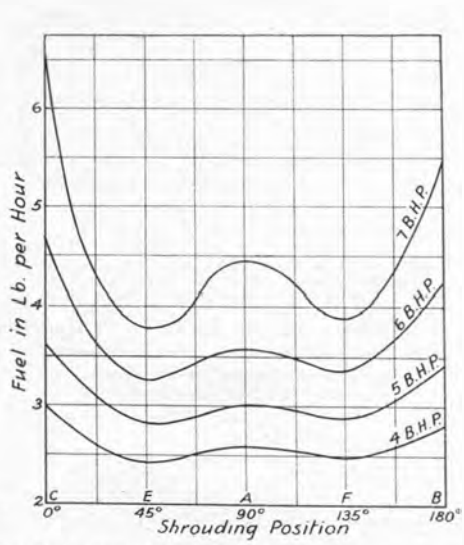
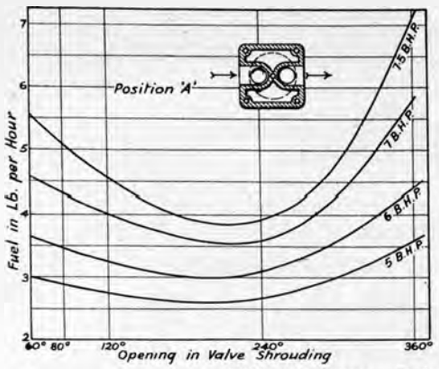
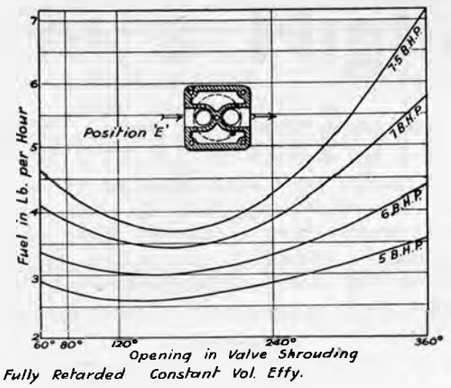


Fig. 20

R.P.M. = 1200 Approx Cooling Water Outlet Temp = 60°C. Timing Fully Retarded Vol Effy Constant

basis to show the effect of placing the inlet valve in different positions, and in this way Fig. 18 shows curves for 7, 6, 5 and 4





R.P.M. = 1200 Approx. Timing Fully Retarded Constant Vol. Effy.

Fig. 21 Fig. 22

B.H.P. plotted on a shrouding position basis. It may be interesting to note that in some positions the fuel consumption is nearly doubled for the same load. In Fig. 18, positions A, E and F gave almost the same results for the fuel in terms of lb. per hour.

Figs. 15 and 16 were replotted, as can be seen from Figs. 19 and 20, which show the effect of inlet air direction when valves II and III were used respectively. In Fig. 19, position A, slightly worse results were obtained than in positions E and F. Also Fig. 20 shows a marked variation in comparing position A with positions E and F.

The diagrams clearly indicate that when valves with big shrouding, i.e., small openings, were used, or in other words when a high velocity of air was used, position A was not the most suitable position for a satisfactory combustion. It is clear from Fig. 3, that position A is the one which should give the inlet air the proper circulatory motion.

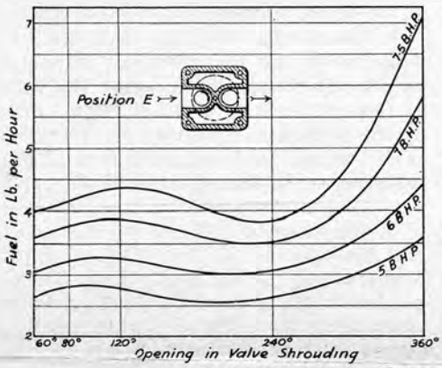
Optimum swirl

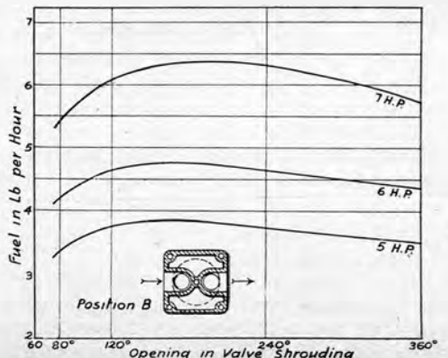
Under this heading the authors wish to discuss the importance of turbulence and how it affects the process of combustion. It has been suggested, as mentioned in the early part of this paper, that there is a definite velocity of swirl that gives minimum fuel consumption. It has been pointed out before that in the second and third phase of the process of combustion in a compression-ignition engine turbulence is very essential. The reason advanced by the authors is one which extends the views given by Mr. Ricardo.

As soon as the fuel rushes into the cylinder in a pulverised condition, the minute droplets find themselves in an atmosphere rich in oxygen. When the delay period is over, and a number of these droplets have already been ignited, the flame starts spreading. At the same time the pressure commences to rise, which marks the beginning of the second phase. The rate of pressure rise depends upon the quantity of fuel present inside the combustion chamber and the way in which this pulverised fuel comes into contact with the highly compressed air, which is at high temperature. To start with, the droplets of fuel find plenty of oxygen surrounding them, and will have no difficulty in burning. As soon as a minute fraction of a second passes, these droplets, together with the fresh pulverised fuel entering the cylinder, will be surrounded by zones where oxygen is rare; and unless the air keeps moving about inside the combustion chamber to supply the fuel with the required quantity of oxygen, the combustion process will be too slow to be completed in the short time available. Theoretically this suggests, of course, that the higher the velocity of approach between the air and the fuel the quicker the combustion takes place, and therefore the lower the fuel consumption.

The above statement, however, is not in practice correct. Recent research on the subject of combustion in a compression-ignition engine has proved that optimum swirl does exist in this type of engine. It has already been mentioned that tests carried out by Ricardo and Alcock on a sleeve-valve engine with directed swirl gave a value for optimum swirl of ten times the crank revolutions per minute.

Also, the tests carried out by the authors, as will be shown later, indicate the existence of optimum inlet velocity and, it might be argued from this, with certain valve positions, optimum swirl. The results obtained not only indicate the existence of optimum air inlet velocity alone, but show that the value of this is affected by the shrouding position. In other words, there is a definite valve opening for every shrouding position which gives the minimum fuel consumption for that position.





R.P.M. = 1200 Approx Timing Fully Retarded Vol Effy Constant

Fig. 23 Fig. 24

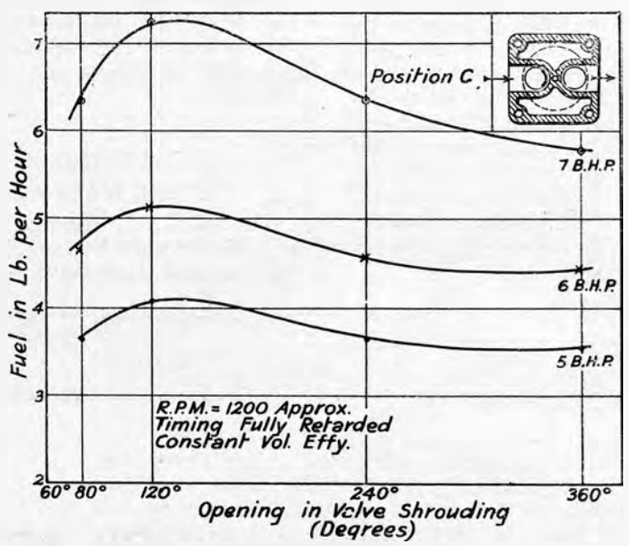


Fig. 25

Discussion of results

The results obtained and as shown in Figs. 14, 15, 16 and 17, were plotted again for each position, and the curves obtained show the relation between fuel consumption and opening in valve shrouding for 5, 6, 7 and 7·5 B.H.P.

Position A.—Fig. 21 shows very clearly that in this position the best fuel consumptions, for this type of combustion chamber, and the same volumetric efficiency and temperature, for the following B.H.P.s are :—

B.H.P.	*Fuel Consumption*
5	2·6 lb. per hour approx.
6	3 ,, ,, ,, ,,
7	3·55 ,, ,, ,, ,,
7·5	3·85 ,, ,, ,, ,,

Also, the optimum shrouding is of the order 150° (i.e., 210° opening).

At 7·5 B.H.P. the fuel varies from 3·85 lb. per hour, up to 7·1 lb. per hour.

Position F.—In Fig. 22 the best fuel consumption on this particular combustion, and for the same conditions of running, is shown as follows :—

B.H.P.	*Fuel Consumption*
5	2·6 lb. per hour approx.
6	3·0 ,, ,, ,, ,,
7	3·42 ,, ,, ,, ,,
7·5	3·7 ,, ,, ,, ,,

The optimum shrouding for this position appears to be 200° (160° opening).

Position E.—This position gave curves of peculiar shapes which may be due to the path the air takes. The curves, Fig. 23, slope down to the best position at 140° of shroud (220° opening), then up again, and finally curve down to a certain extent. The best opening appears to be 220°, where the fuel consumption is as follows :—

B.H.P.	*Fuel Consumption*
5	2·56 lb. per hour approx.
6	3· ,, ,, ,, ,,
7	3·5 ,, ,, ,, ,,
7·5	3·74 ,, ,, ,, ,,

Position B.—It is interesting to note that in this position the best fuel consumption is with the smallest valve opening and the next best is given by the unshrouded valve. The curves also show a maximum fuel consumption, instead of minimum. Fig 24 shows the test results. The maximum fuel consumption obtained in this position is with an opening of 180° (180° of shroud), and is shown below :—

B.H.P.	*Fuel Consumption*
5	3·86 lb. per hour approx.
6	4·75 ,, ,, ,, ,,
7	6·35 ,, ,, ,, ,,

Position C.—This position gave the worst possible fuel consumption ; the curves in Fig. 25 are almost the same shape as in position B, but the fuel consumptions are higher. At 140° opening (220° of shroud) the consumption is given as follows :—

B.H.P.	*Fuel Consumption*
5	4·1 lb. per hour approx.
6	5·13 ,, ,, ,, ,,
7	7·4 ,, ,, ,, ,,

Although the results tabulated in the first part of this paper, under the discussion of symmetry, were carried out while the volumetric efficiency was ignored, they support the idea of an optimum swirl to a great extent.

Conclusion

A general survey of the experiments carried out on the Tangye high-speed oil engine, supported by the experiments of Messrs. Ricardo and Alcock on a sleeve-valve engine, may give some idea of the factors affecting the combustion in a high-speed compression-ignition oil engine so far as the fuel and air are concerned.

It seems that the direction and the velocity of the air plays an important part in the fuel consumption. The width of the air jet may also have some effect on the process of combustion. In certain cases where indiscriminate turbulence exists, or when the air has not been given a rotational movement, a large quantity of fuel has been wasted, or in other words, has not been burnt completely. This is probably due to the fact that it has not met the right quantity of oxygen in the time available. On the other hand an excess of air swirl increased the fuel consumption, too. The authors have the following suggestions to make in this connection.

With excessive air swirl, the air may perhaps blow out the flame, or delay its movement.

A high swirl movement given to the air may perhaps cause the latter to drag the fuel with it.

Another effect of excessive air swirl may be to force the fuel into contact with the surface of the combustion chamber, resulting in slow and unsatisfactory burning.

APPENDIX I

Fuel used and its specification

The fuel used during the tests was supplied by Shell-Mex and B.P. Ltd., and was the same throughout the experiments.

Approximate Analysis.

Carbon	86 per cent.
Hydrogen	13 ,,
Sulphur	0·7 ,,
Undetermined (oxygen, nitrogen, etc.)	0·3 ,,
Specific gravity at 60° F	= 0·84
Flash point (closed)	= 200° F.
Higher Calorific Value. B.Th.u.s per lb.	19,600
Carbon residue (Conradson)	= 0·01 per cent.

Ash contents, negligible.
Water and sediments, negligible.

Distillation.

Initial boiling point	= 220° C.
Distilling to 300° C.	= 70 per cent. over.
Final Boiling point	= 360° C.

APPENDIX II

Calculation of quantity of air used by the engine

To measure the quantity of air, an orifice $1\frac{3}{16}$ in. in diameter was employed. The following method was used to calculate the quantity of air taken in by the engine :

Let v = velocity of air through the orifice.

h = pressure drop across the orifice.

$$L = \frac{h_1 - h_2}{\text{density}} = \frac{(h_1 - h_2)\,RT}{P}$$

where $h_1 - h_2 = H$ difference in pressure between two sides of the orifice.

Neglecting compressibility :—

$$V_2 = 2gL.$$

$$V = \sqrt{2gL} = \sqrt{\frac{2gL\,(h_1 - h_2)\,RT}{P}}$$

One inch of pressure drop = 5·2 lb. per sq. ft.

$$\therefore \quad V = \sqrt{2g \times 5{\cdot}2H\frac{RT}{P}}$$

Let Q = quantity of air used in cu. ft. per minute.

$Q = 60$ K.A.V. where—

A = Area of orifice in sq. ft. = 0·00769 sq. ft.

K = Coefficient of discharge through the orifice = 0·604.

$$\therefore \quad Q = KA\sqrt{2g \times 5{\cdot}2H\,\frac{RT}{P}}$$

$$= 4{\cdot}49\sqrt{\frac{RT}{P}}$$