University of Edinburgh

T H E S I S

on

The Effect of Air Inlet Conditions on the performance of a slow speed compression ignition oil engine.

submitted by

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for the degree of Ph.D.

UNIV. EDINBURGEN EX BIBL. *

# INTRODUCTION

In the past, a large number of experiments have been carried out by various experimenters investigating the combustion of liquid fuel in air at different temperature and pressure conditions.

The great majority of these experiments have been of a physical character, carried out in specially constructed bombs. Relatively few experiments have been carried out with normal engines.

The chief purpose of the tests described in the following pages was to investigate the combustion of fuel in a slow speed compression ignition engine and to show a connection between the phenomena of combustion and the performance of the engine as a whole.

In general, the method adopted was first to determine the load characteristics of the engine under normal, and approximately constant, atmospheric conditions, from no load to full load.

Variation of the temperature of the charge was made by preheating the inlet air, the maximum inlet air temperature obtained being about 100 °C. u/

Variation of the air consumption was effected by throttling in the induction pipe of the engine. Available equipment did not permit of supercharging the engine.

Prominence/

Prominence has been given to a thermodynamic method of estimating the rate of combustion and the method has been applied to each series of tests.

When varying the induction conditions, care was taken to maintain as near as possible a constant fuel consumption in any one series of tests. Since the thermal efficiency of the engine changes with the induction conditions, this necessitated small variations in the brake load. While it was not possible to obtain absolute constancy in the fuel consumption, it is considered that such a method is preferable to carrying out tests at constant B.H.P. but variable fuel consumption.

In conclusion, the author wishes to acknowledge the many helpful suggestions received from various members of the Engineering Department.

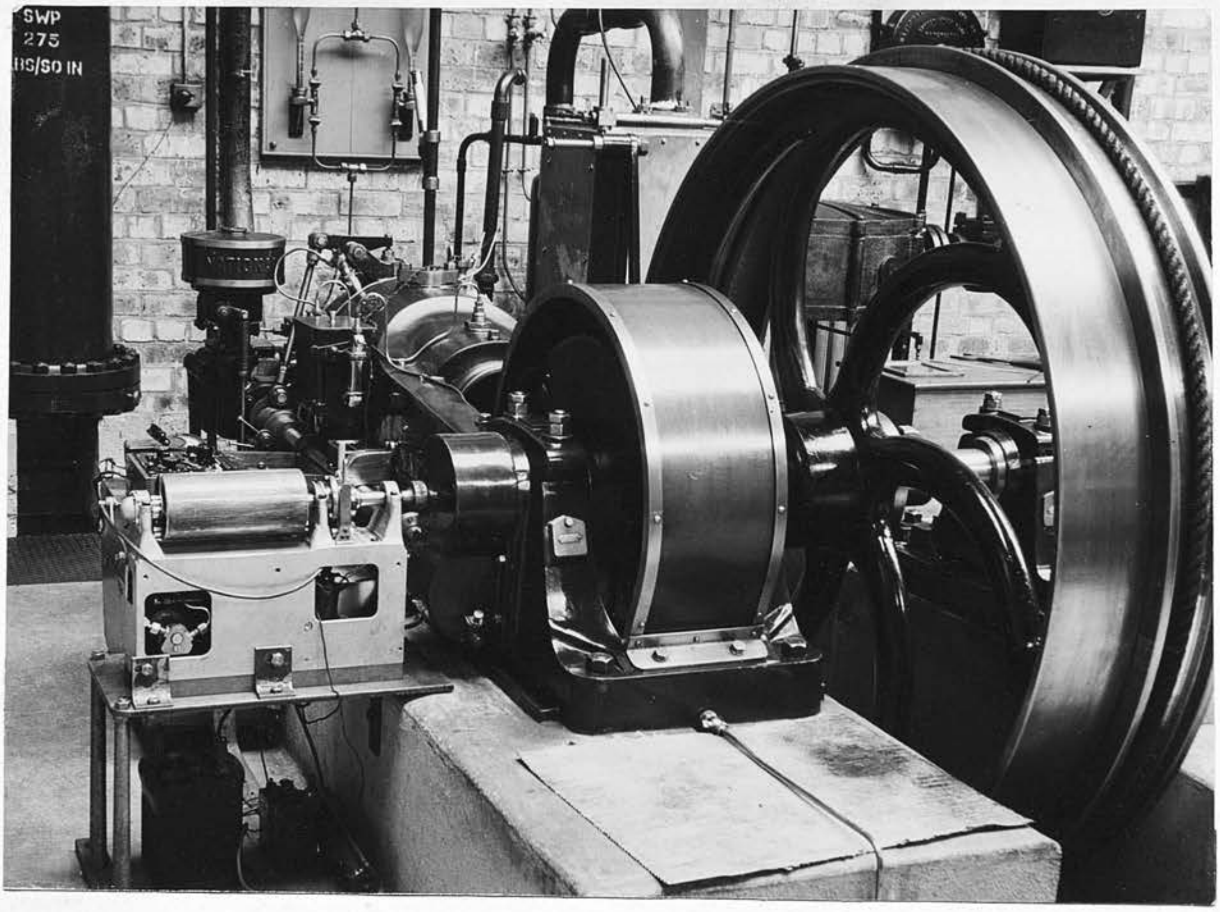


FIG. 1

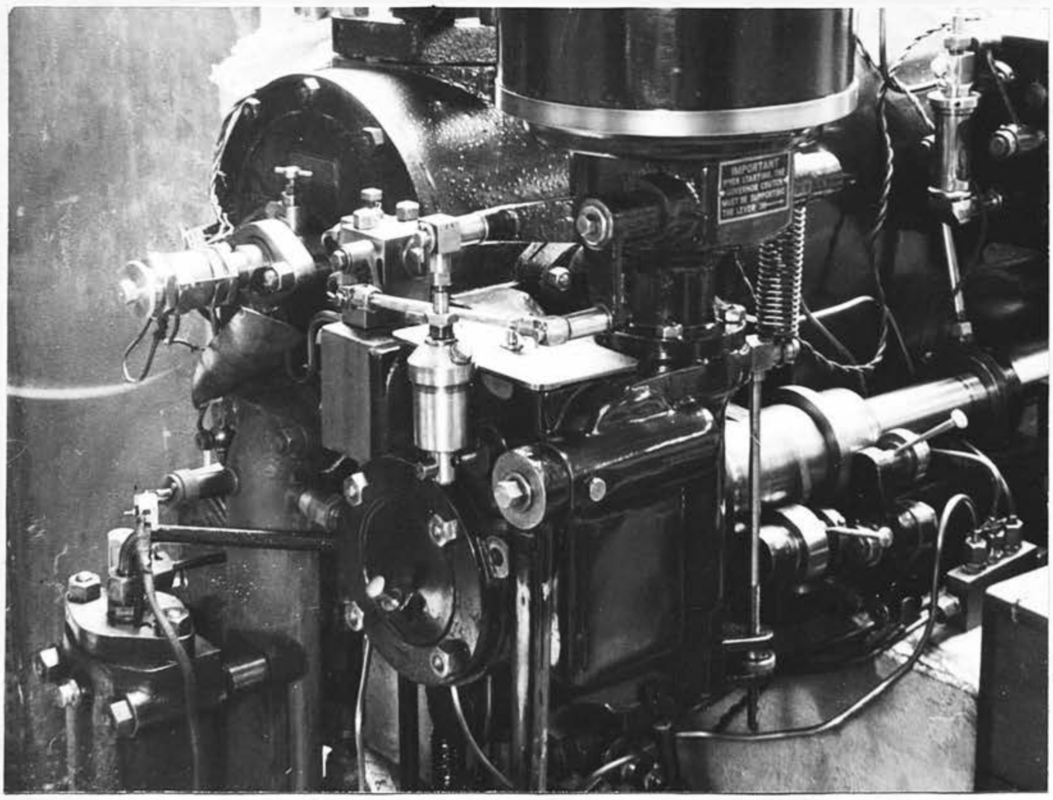

FIG. 2.

## TESTING EQUIPMENT.

ENGINE:

The Tests were carried out on a horizontal four stroke cycle, single cylinder National heavy oil engine. Cylinder Dimensions: Bore 8 inches; Stroke 16 inches. Volume Compression ratio: 13.11:1
Normal full load rating: 18 B.H.P. at 290 R.P.M.

A cam operated plunger pump delivered the fuel to the injector, the pressure of the fuel lifting the needle off its seat against the action of a spring. The fuel pump plunger has a constant stroke and the amount of oil delivered to the injector depends on the position of a bye-pass valve controlled by a governor.

The engine was fitted with a water cooled rope-brake ring on the fly-wheel for power absorption purposes and a revolution counter, driven from the camshaft, was also incorporated.

Two views of the engine and equipment are shown in Figs. 1 and 2.

FUEL SUPPLY:

For the measurement of fuel consumption, two glass flasks of double conical form, tapering to a narrow/

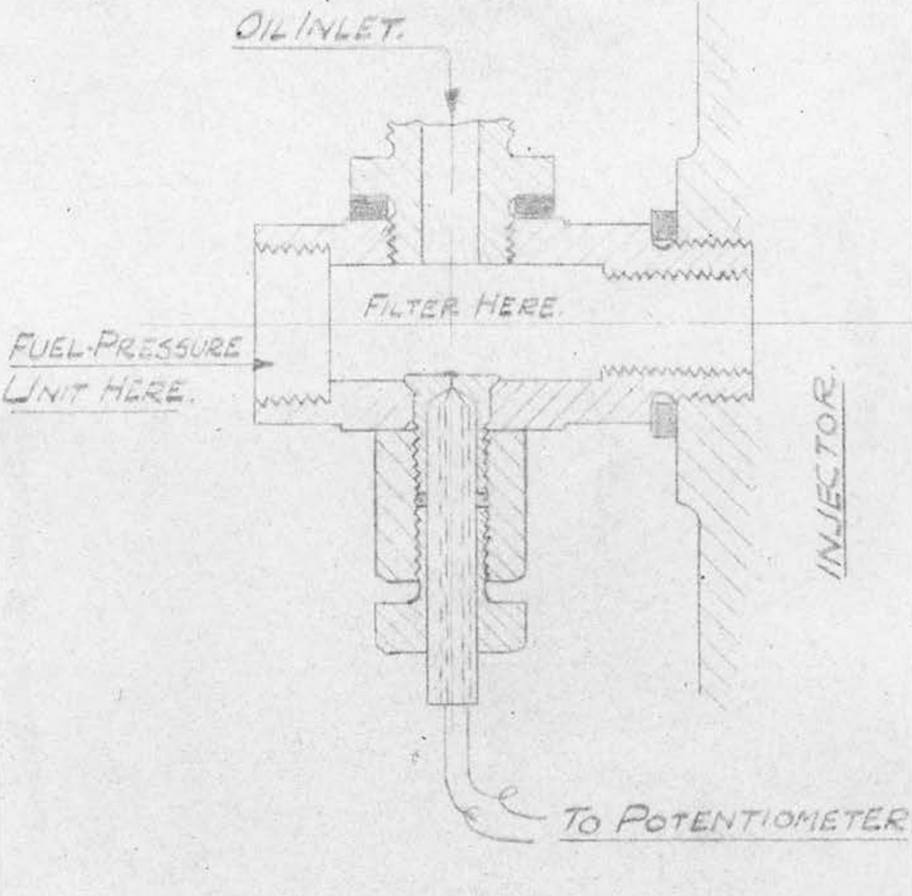


FIG 3 — THERMOCOUPLE FOR OIL TEMPERATURE

SCALE – FULL SIZE

narrow neck at each end, were employed. These had been previously calibrated and the narrow necks suitably marked. The flasks were fed from a large supply tank about 8 feet above the level of the engine fuel pump. The procedure during any test was to switch over from one flask to another, checking the fuel consumption rates for consistency, by means of a stop-watch.

Arrangement was also made for measuring the fuel temperature at the injector. A copper cadmium thermo-couple was built into the injector as close to the nozzle as practicable. The arrangement is shown in Fig. 3.

The couple was connected to a potentiometer through suitable compensating loads. Calibration was carried out by passing heated oil through the dismembered part of the injector and checking against a reliable direct reading mercury thermometer.

Records of injection pressures and timing were obtained on a Dobbie McInnes "Farnboro" Indicator and will be discussed later.

AIR SUPPLY:

Care was taken to measure accurately the quantity, temperature and pressure of the supply.

A/

e/

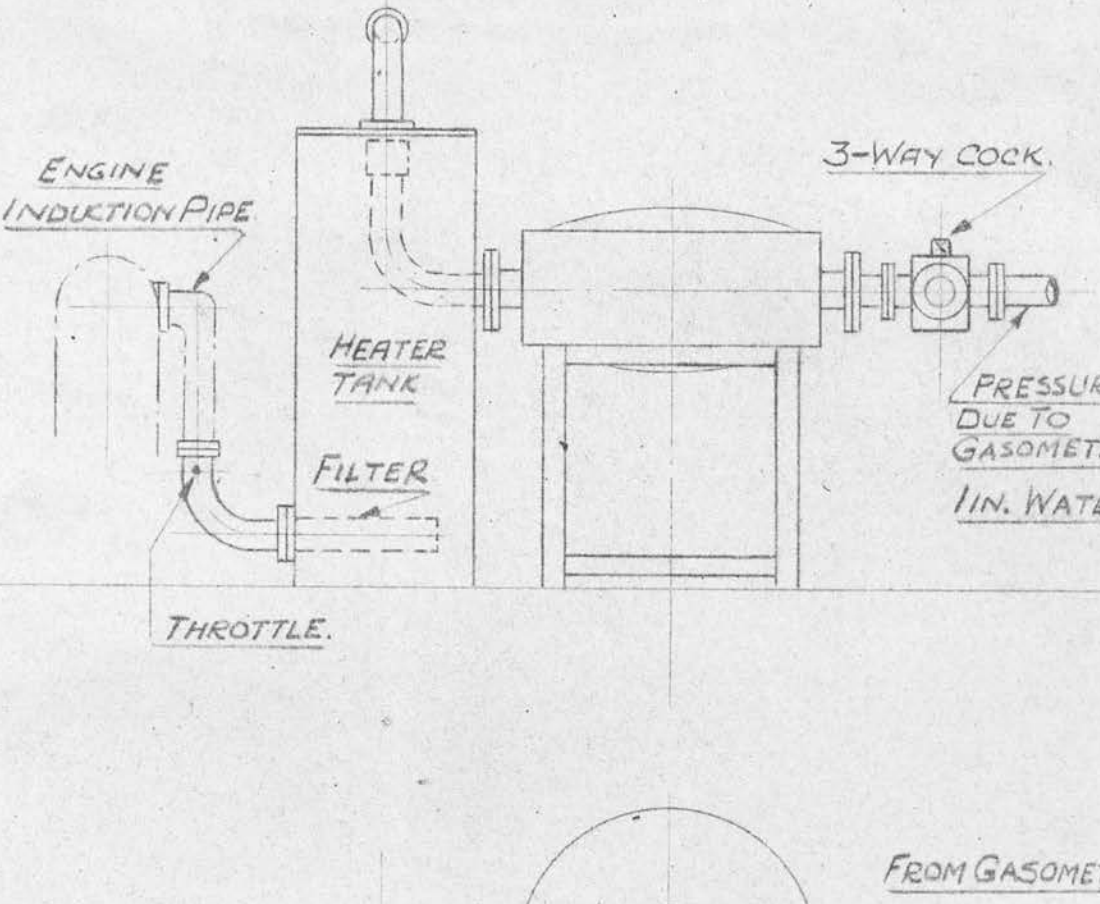


FIG 4. – AIR INDUCTION SYSTEM.

SCALE ½" = 1 FOOT

A diagrammatic arrangement of the air induction system is shown in Fig. 4.

Heater

Heating was carried out electrically in a steel leak-proof tank, air entering at the top and passing out at the bottom to the engine.

Four coils built from 20 S.W.G. Iron Wire were suspended on carriers inside the tank so as to afford as large a heating surface as possible.

Each coil absorbed about 1 Kilowatt and was connected through an insulated and leak-proof terminal to a switch on the control panel.

One of the coils was connected to a variable resistance which was arranged to vary the power carried by the coil in steps of $\frac{1}{12}$ Kilowatt.

By this means, the temperature could be regulated from minimum to maximum in steps of less than 2° C., since any number of coils could be brought into use.

Quantity Control

This was effected by a throttle plate interposed between the heater tank and the engine. Precautions were taken to avoid leakage at the spindle, one of the bearings being made "blind" while the other was fitted with a gland. The arrangement is shown in Fig. 5.

Measurement/

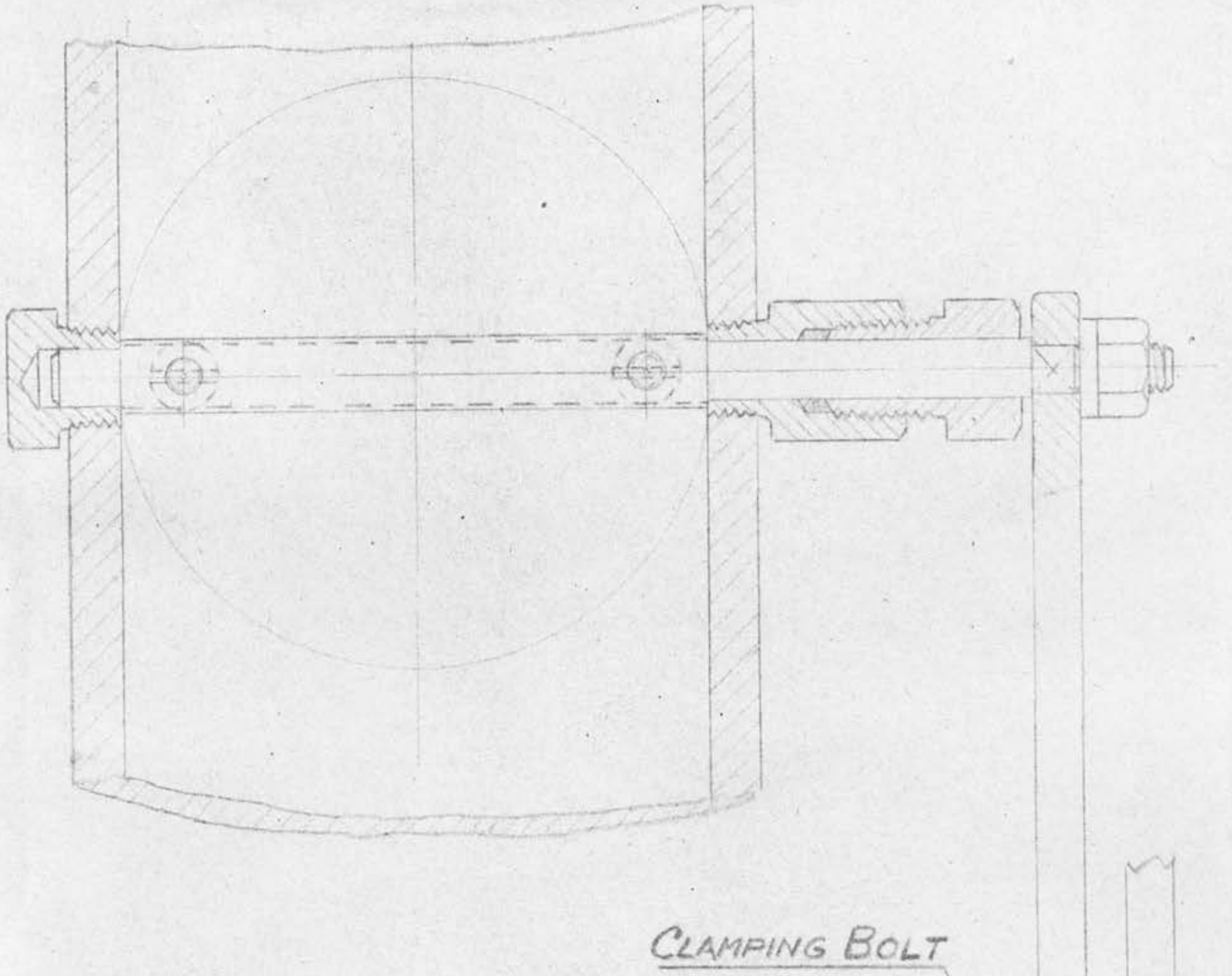


FIG 5 - THROTTLE.

SCALE - FULL SIZE.

Measurement of Air consumption was made by a calibrated gasometer, consisting of water sealed bell and container, and having a capacity of 200 cubic feet of free air.

Under normal induction conditions, this enabled air consumption to be measured over $2\frac{1}{2}$ minutes.

A description of the Air measuring equipment appeared in "Engineering" on the 16th April, 1937.

Induction Temperature

Arrangement was made to measure the temperature as close to the engine as possible - about 6 inches from the inlet valve (see Figs. 1 and 2).

Preliminary tests showed that, under certain running conditions, blow back from the cylinder gave readings which were obviously not the average induction temperature. A second arrangement was, therefore, incorporated on the other side of the throttle and about 24 inches from the inlet valve, and readings taken here were used for normal induction conditions.

INDICATOR GEAR:

An existing Dobbie McInnes "Farnboro" Electric Indicator was used to obtain records of the pressures in the engine cylinder, the induction pipe and the fuel injector.

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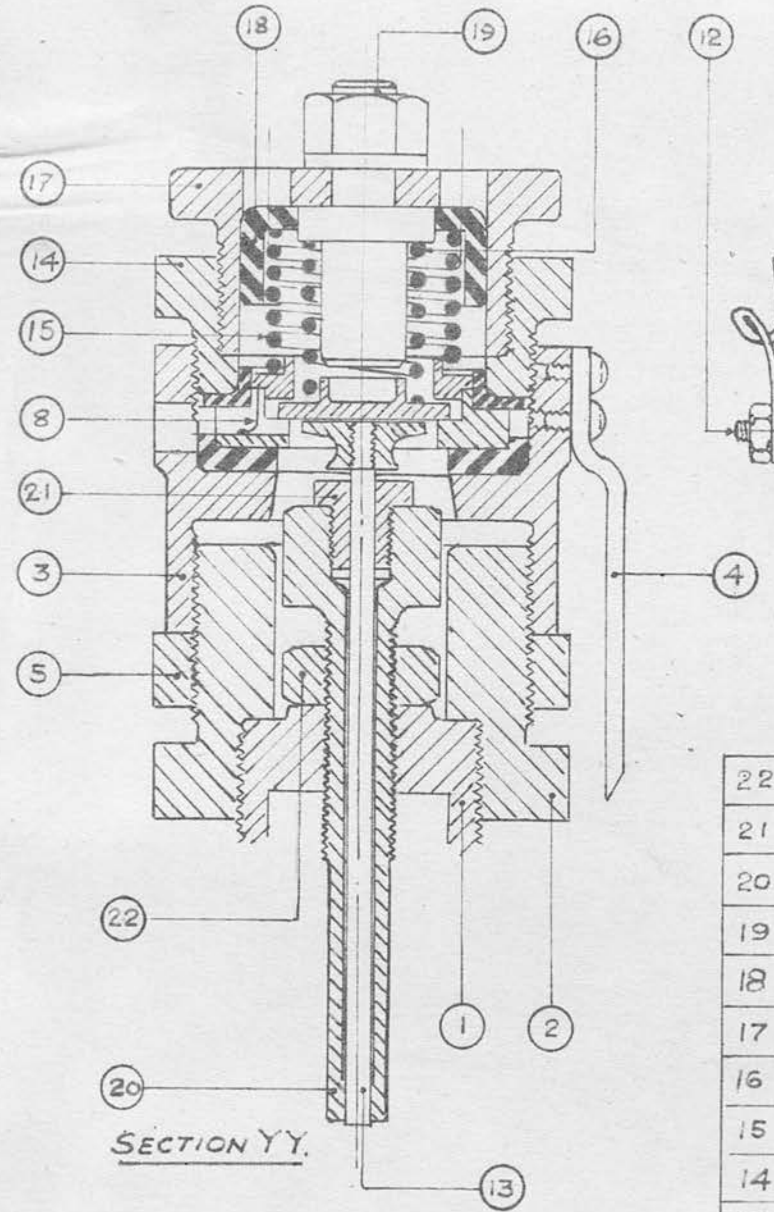


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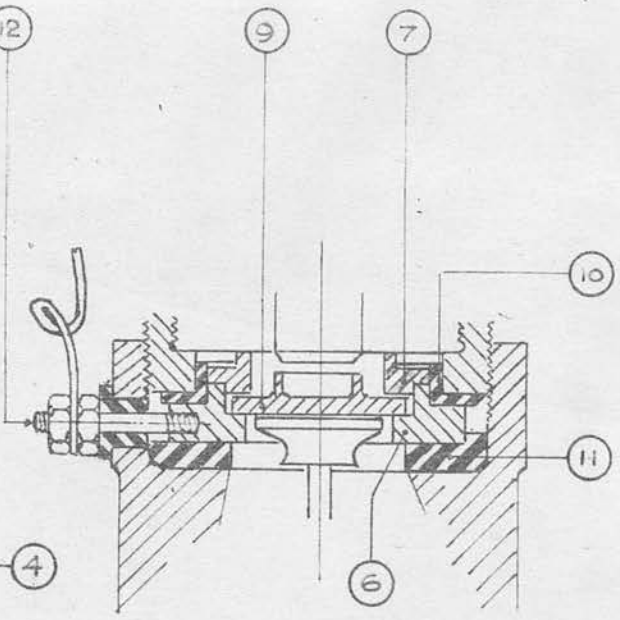


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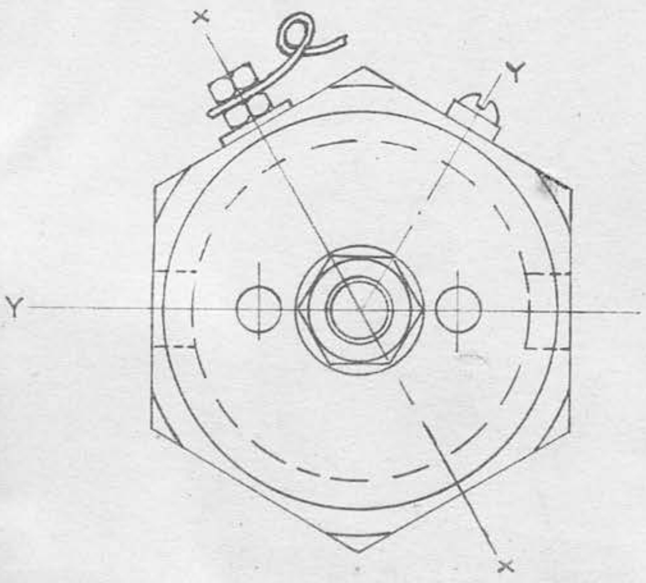


FIG. 6 – INJECTION TIMING UNIT.

SCALE – FULL SIZE.

| | |
|---|---|
| 22 | LOCK NUT FOR INJECTOR NEEDLE STOP. |
| 21 | GUIDE BUSH FOR PLUNGER. |
| 20 | STOP FOR INJECTOR NEEDLE. |
| 19 | GUIDE FOR DISC RETURN SPRING. |
| 18 | INSULATION FOR RETURN SPRING. |
| 17 | ADJUSTING NUT FOR SPRING TENSION. |
| 16 | RETURN SPRING FOR "M & B." DISC. |
| 15 | RETURN SPRING FOR UPPER SEAT |
| 14 | LOCKNUT FOR LOWER SEAT. |
| 13 | ACTUATING PLUNGER. |
| 12 | TERMINAL FOR LOWER SEAT. |
| 11 | INSULATION FOR LOWER SEAT |
| 10 | DITTO |
| 9 | "MAKE & BREAK" DISC |
| 8 | LIGHT CONTACT SPRING BETWEEN SEATS. |
| 7 | UPPER DISC VALVE SEAT. |
| 6 | LOWER DISC VALVE SEAT |
| 5 | LOCKNUT FOR ADJUSTABLE SLEEVE. |
| 4 | POINTER FOR SETTING SLEEVE. |
| 3 | ADJUSTABLE SLEEVE |
| 2 | ADAPTER. |
| 1 | INJECTOR BODY |

PARTS LIST.

To the standard equipment used for obtaining these records was added an auxiliary electric circuit which was capable of sparking out, simultaneously with any other record, a diagram of injection timing.

The diagrams obtained by the "Farnboro" Indicator are, of course, to a base of crankangle, the indicator drum in this case being directly coupled to, and in line with, the crankshaft. They represent the mean of about 50 cycles. This is a great advantage, practically a necessity, where the speed of the engine and its fuel consumption are controlled by a centrifugal governor. As will be seen later, the quantity of fuel injected during the cycle varies considerably at any one load.

Correct phasing was checked periodically by a "No-Fuel" compression and re-expansion curve.

For obtaining the pressure records, Dobbie McInnes standard units were employed - Disc Valve units for the engine cylinder and induction pipe pressures, a Differential Plunge Valve unit for the fuel pressures.

For measuring the injection timing, the author designed a unit which was constructed in the University Workshop. Fig. 6 shows a full size drawing of this unit. It consists of a make and break disc operated through a plunger by the injector needle. When connected/

connected up to the auxiliary electrical circuit in the usual manner, two sparks per cycle are obtained - one at the lifting of the injector needle and the other at the closing of the needle. No attempt was made to obtain a complete valve lift record but the "break" could be adjusted to occur at any desired position of the injector needle. This was achieved by means of the adjustable sleeve (3). Trial showed that valve lift was completed in less than 1 degree of crankangle, the complete lift being about .016 inch.

Records were made of the time at which the needle was .002 inch open on the "lift" and .007 inch open on the closing. The difference of .005 inch between these positions is accounted for by the clearance between the disc and its seat.

For converting the records from a crankangle base to a stroke volume base, a standard transparent scale was used, suitably calibrated in degrees. PV charts were prepared (see Fig. 7 <u>et seq</u>.), the base being divided by vertical ordinates into lengths equivalent to every 10 degree crankangle.

Due allowance was made for the obliquity of the connecting rod when preparing these charts.

CYLINDER/

CYLINDER JACKET WATER:

For measurement of rate of flow, two calibrated tanks were fitted on the outlet side, the procedure being to switch over from one tank to another while one was being emptied.

The supply was made up of hot and cold water obtained from the general supply to the building and mixed in a small cast iron cylinder before entering the engine.

Mercury thermometers close to the engine inlet and outlet were used to measure the temperature.

EXHAUST CONDITIONS:

The average exhaust pipe temperature was measured in all tests by means of a platinum-platinum rhodium thermo-couple. This was inserted as close to the engine as possible - about 5 inches from the exhaust valve - and connected to a direct reading indicator. At the end of each test, a check was made against the potentiometer, and proved satisfactory.

Apart from a visual examination of the exhaust - during each test - a chemical analysis was carried out when the visual examination and low air/fuel ratios indicated the possibility of incomplete combustion.

Standard/

Standard "Orsat" equipment was used with absorbents for $CO_2$, CO and $O_2$, the sample being removed at a point in the exhaust pipe about 10 feet from the engine.

## PRELIMINARY CONSIDERATIONS AND TRIALS.

### FACTORS AFFECTING HEAT LOSSES AND COMBUSTION:

The main object of these tests is to investigate the affect on the combustion of fuel in the engine cylinder of changes in

(a) the temperature of the charge; at induction?

(b) the weight of air used per cycle.

There are, of course, a number of factors which may affect the process of combustion and, as it is desirable to have a single variable in any one series of tests, care has been taken to control these other factors suitably.

One important factor is the quantity of oil injected per cycle. Consequently, it was decided that, when examining the affect of different induction conditions at one approximate load condition, the quantity of oil injected per average cycle should be kept sensibly constant. This necessitated small alterations in the brake load to suit varying Brake Thermal/ e/ Not clear

Thermal Efficiency. Quick determinations of fuel consumption (for setting purposes) were obtained by measuring the time taken to consume 0.1 lb. of oil. No alteration to load was made once the test had started.

The quantity of fuel injected is controlled by a governor and there is, therefore, a cyclical variation in the period of injection and, consequently, in the quantity of oil injected.

The use of heavy gear oil in the dash pot attached to the governor damped down the variations, though it was apt to require frequent regulation. The injection timing unit was found to be of great assistance in these adjustments.

?

The temperature of the fuel oil will, if varied, considerably affect the rate of combustion. Arrangement was, therefore, made to check the variation of temperature at the injector. The highest fuel temperature recorded in any test was 64° C.

H.H. Wolfer has stated that the combustion is not affected by fuel temperature below 100° C. and this view is supported by other authorities[1,2]. The effect of the fuel temperature has, therefore, been neglected.

That/

1. Gerrish & Ayer - N.A.C.A. Technical Note No. 565 1936
2. H.H. Wolfer - (V.D.I. - Forschungschaft No. 392) 1938

That the jacket conditions have a considerable effect on the engine performance is well known though the exact effect on combustion is not so apparent.

There appear to be two factors:-

(a) the jacket temperature, i.e. the average of inlet temperature and outlet temperature;

(b) the rate of flow of water through the jacket.

The former is of importance since it affects the conduction of heat from the cylinder walls, while the latter affects the scouring action, i.e. the removal of heat by convection. Experiments carried out in the Heat Engines Laboratory at the University of Edinburgh in 1937 showed that, at full load, combustion became very irregular with cold jacket*, and the heat given to the jacket water was greatly increased as the mean jacket temperature was lowered. Unfortunately, no attempt was made in these tests to keep the rate of flow through the jacket constant.

It was thought that the engine would be more sensitive to small changes of flow rate when the rate of flow was low. To confirm this view, a few preliminary tests were carried out at about 9 B.H.P. and a jacket temperature averaging 45° C. Table I shows the chief results.

Table/

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* M. Davidson - "Effect of Jacket Temperature
PhD. Thesis University of Edinburgh 1937.

TABLE I.

| | | | | |
|---|---|---|---|---|
| Quantity of Jacket Water per min. lb... | 19.63 | 11.53 | 6.18 | 5.53 |
| Inlet Temp. °C. .. | 37.7 | 33.4 | 24.9 | 21.8 |
| Outlet Temp. °C. .. | 51.5 | 56.5 | 65.9 | 67.4 |
| Mean Temp. °C. .. | 44.6 | 45.0 | 45.4 | 44.6 |
| Heat to Jacket water per min. C.H.U. .. | 271 | 266 | 254 | 252 |
| B.H.P. .. .. .. | 8.96 | 9.01 | 8.92 | 9.03 |

It will be seen that the variation of heat to jacket is not great for flows above 10 lbs. per minute, whereas, below 10 lbs. per minute, the heat taken up by the jacket water is considerably reduced.

The most noticeable feature observed was the difficulty in controlling the temperature at the lowest flow rates. Hence it was decided to maintain the flow during any series of tests between 15 lbs. per minute and 10 lbs. per minute except at the very highest loads, the mean jacket temperature being fixed at 40°C.

It is appreciated that the results of the above tests do not necessarily indicate any differences in the mode of combustion.

Barometric Pressure cannot be said to affect the combustion process except when air/fuel ratios are at

a/

a critical stage. In order to obtain comparative values of exhaust and suction pressures (absolute), the variation of atmospheric pressure should not be excessive. In the tests carried out, this ranged from 14.23 lbs. per sq.in. to 14.73 lbs. per sq.in.

## TEST ARRANGEMENTS.

CLASSIFICATION OF TESTS:

(1) Tests at normal induction conditions.

6 tests were carried out chiefly to determine the power characteristics of the engine.

Classification: P 1. - P 6.

(2) Tests with varying charge temperature but approximately constant weight of air.

3 tests at light load - Classification: TL 1, TL 2, TL 3.

3 tests at half load - Classification: TH 1, TH 2, TH 3.

(3) Tests at normal induction temperature varying throttle opening.

6 tests at light load - Classification: WL1 - WL6.

FUEL OIL:

The fuel oil used throughout these tests was Pool Diesel Oil.

A/

A sample was analysed by a competent authority yielding the following results:-

| | | |
|---|---|---|
| Gross Calorific Value | .. | 10,830 C.H.U. per lb. |
| Nett Calorific Value | .. | 10,100 C.H.U. per lb. |
| Specific Gravity | .. .. | .868 |

Chemical Analysis:

C 85.5%     $H_2$ 13.0%     S 1.25%

Nitrogen, Oxygen, Ash and Errors 0.25%

Theoretical Air required for complete combustion     14.4 lb. per lb.

The Nett Calorific Value has been used throughout for the determination of thermal efficiencies etc.

STEADY CONDITIONS:

In order to obtain consistent results, care was taken to ensure steady running conditions.

Not less than $1\frac{1}{2}$ hours was allowed to elapse after starting from cold before commencing a test and not less than 1 hour between tests. Temperature readings were taken every ten minutes - this being the minimum regular interval consistent with a regular measurement of air consumption and taking of pressure diagrams. Not less than two air consumption measurements were made during any one test. Two sets of indicator/

indicator diagrams were taken - one set early in the test and the other set towards the end of the test. Usually both sets proved consistent. As would be expected, some tests proved unsatisfactory either on account of jacket temperature or inlet conditions. These were rejected and subsequently repeated.

Co-ordination of Tests.

The three groups of tests mentioned above have been dealt with separately in so far as their general characteristics are concerned. Finally, information obtained from each of the three groups has been analysed and co-ordinated.

## ENGINE CHARACTERISTICS UNDER NORMAL INDUCTION CONDITIONS.

LOAD PERFORMANCE TESTS:

Tests were carried out at six different loads to determine the characteristics of the engine over its whole power range i.e. from No load to Full load.

Table II shows the more general characteristics and is followed by 6 pressure volume indicator diagrams referring to these tests (Figs. 7 - 12).

Table/

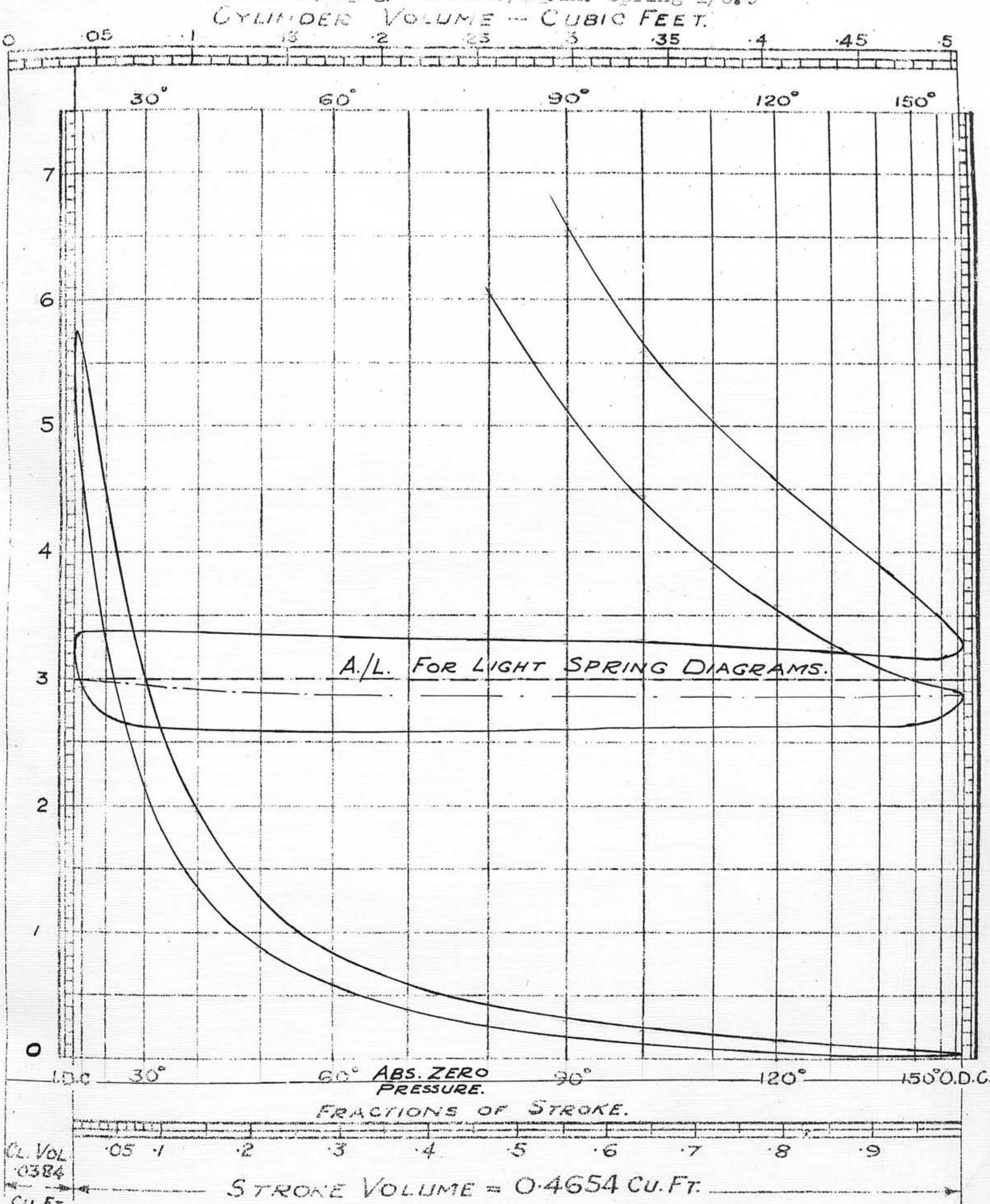
TABLE II

| Test No. | P 1 | P 2 | P 3 | P 4 | P 5 | P 6 |
|---|---|---|---|---|---|---|
| Atmospheric press. lb/sq.in.abs. | 14.23 | 14.52 | 14.23 | 14.44 | 14.45 | 14.41 |
| R.P.M. | 288.8 | 288.6 | 288.7 | 288.3 | 288.1 | 288.1 |
| B.H.P. | - | 4.98 | 8.73 | 12.44 | 15.57 | 17.81 |
| I.M.E.P. (gross) lb./sq.in. | 21.9 | 38.7 | 48.95 | 62.2 | 72.5 | 78.1 |
| I.M.E.P. (pumping) lb./sq.in. | 4.53 | 4.22 | 3.52 | 3.70 | 3.43 | 3.13 |
| I.M.E.P. (nett) lb./sq.in. | 17.4 | 34.5 | 45.4 | 58.5 | 69.1 | 75.0 |
| I.H.P. (gross) | 6.44 | 11.36 | 14.37 | 18.22 | 20.93 | 22.88 |
| I.H.P. (pumping) | 1.33 | 1.23 | 1.03 | 1.08 | 1.01 | .92 |
| I.H.P. (nett) | 5.11 | 10.13 | 13.34 | 17.14 | 19.92 | 21.96 |
| Friction H.P. | 5.11 | 5.15 | 4.61 | 4.60 | 4.35 | 4.15 |
| Mechanical Efficiency % | - | 49.3 | 65.4 | 72.6 | 78.2 | 81.0 |
| Fuel used per hour - lb. | 2.068 | 3.68 | 4.91 | 6.65 | 8.49 | 11.10 |
| Fuel/B.H.P./hour - lb. | - | .738 | .563 | .535 | .545 | .624 |
| Fuel/Gross I.H.P./hour - lb. | .321 | .324 | .342 | .365 | .407 | .485 |
| Fuel/Nett I.H.P./hour - lb. | .405 | .363 | .368 | .388 | .427 | .506 |
| B.T.E. % | - | 19.0 | 24.9 | 26.2 | 25.7 | 22.5 |
| I.T.E. on Gross I.H.P. % | 43.7 | 43.3 | 40.9 | 38.4 | 34.4 | 28.9 |
| I.T.E. on Nett I.H.P. % | 34.6 | 38.6 | 38.1 | 36.1 | 32.8 | 27.7 |
| Air used per minute - lb. | 4.3 | 4.37 | 4.25 | 4.24 | 4.22 | 4.09 |
| Volumetric Efficiency % | 87.7 | 87.9 | 86.3 | 86.2 | 85.3 | 83.2 |
| Induction Temperature °C. | 18.5 | 17.1 | 18.2 | 18.2 | 18.3 | 20.6 |
| Exhaust Temperature °C. | 101 | 165 | 217 | 265 | 381 | 515 |
| Jacket Water inlet temp. °C. | 34.0 | 32.7 | 32.7 | 28.7 | 26.5 | 22.2 |
| Jacket Water outlet temp. °C. | 46.9 | 49.7 | 50.7 | 52.9 | 53.4 | 58.9 |
| Jacket Water flow/min. - lb. | 12.21 | 11.63 | 13.86 | 13.94 | 17.07 | 16.78 |
| Heat to Jacket Water /min. lb. | 157 | 198 | 249 | 337 | 458 | 615 |
| Temp. of Fuel at injector °C. | 43 | 39.5 | 46.5 | 47 | 54 | 64 |

# "NATIONAL" HEAVY OIL ENGINE.

FIG. 7.

TEST P. 1. I.M.E.P. (gross) 21.9 lb./sq.in. Spring 1/80
I.M.E.P. (Pumping) 4.53lb./sq.in. Spring 1/6.9

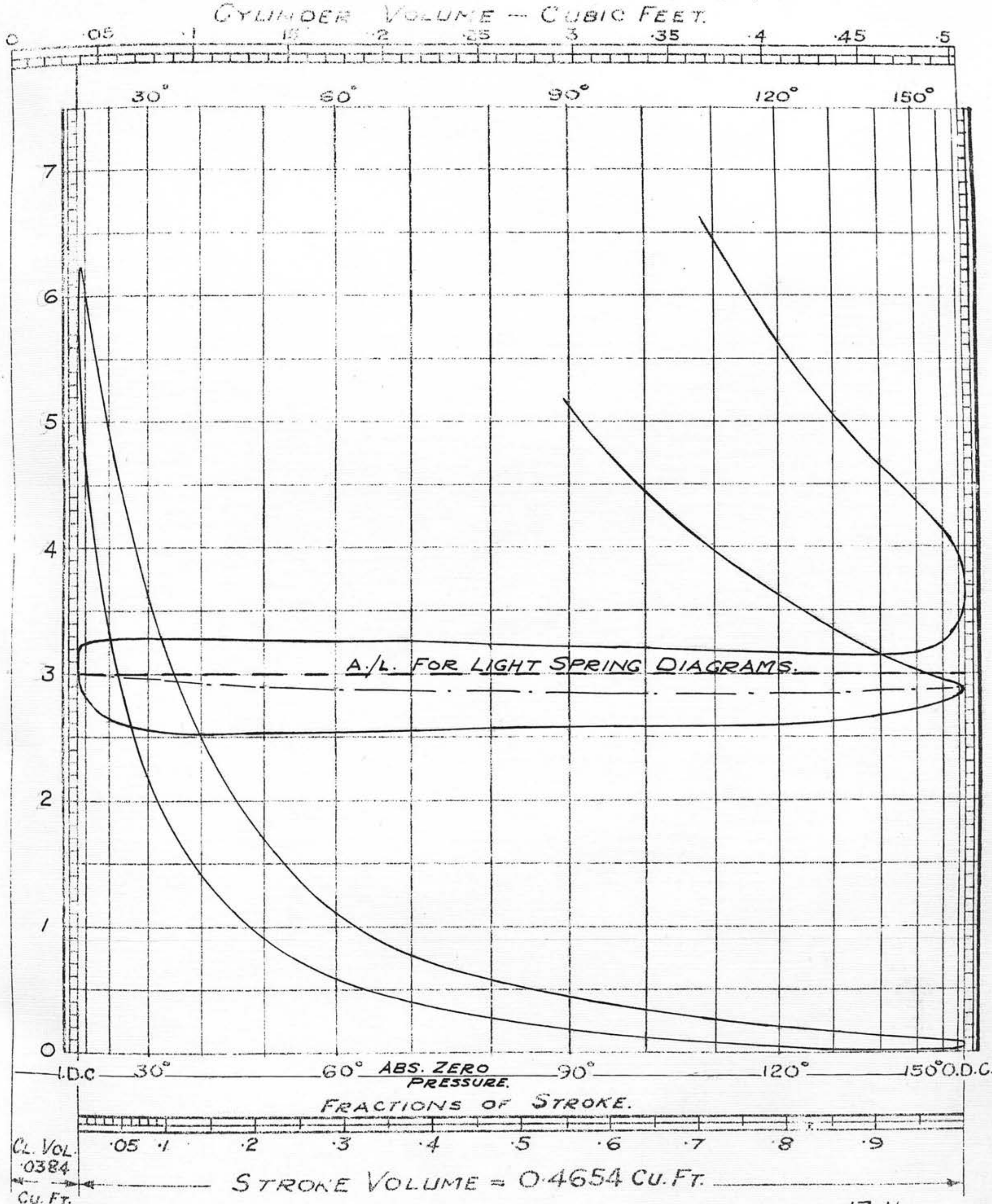



BORE 8" DIA STROKE 16" COMP$^{N}$ RATIO $\frac{13 \cdot 11}{1 \cdot 0}$

# "NATIONAL" HEAVY OIL ENGINE.

FIG. 8.

TEST P. 2. I.M.E.P. (gross) 58.7 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 4.22 lb./sq.in. Spring 1/6.9

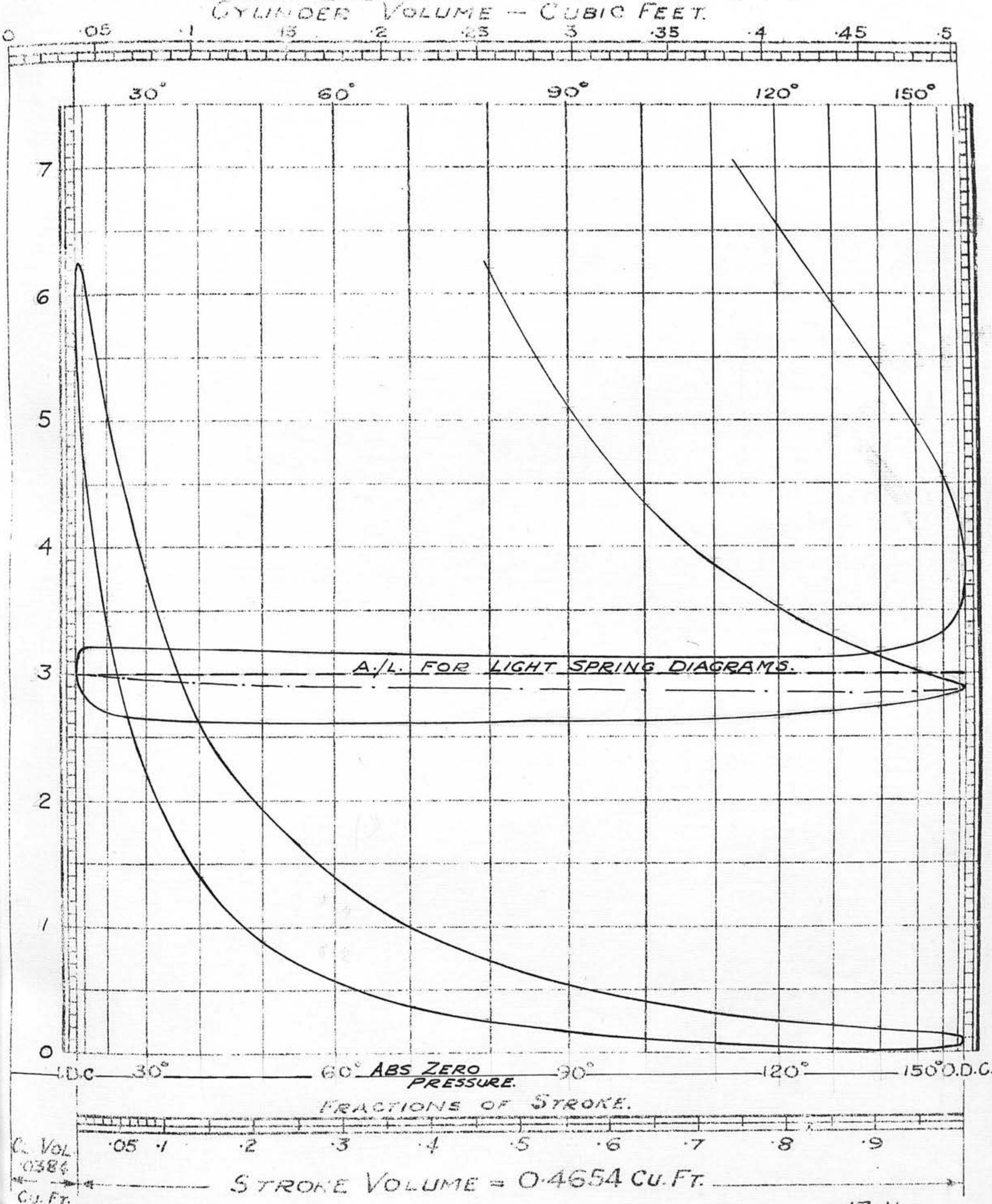



BORE 8" DIA STROKE 16" COMPⁿ RATIO $\frac{13 \cdot 11}{1 \cdot 0}$

# "NATIONAL" HEAVY OIL ENGINE.

FIG. 9.

TEST P. 3. I.M.E.P. (gross) 48.95 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 3.52 lb./sq.in. Spring 1/6.9

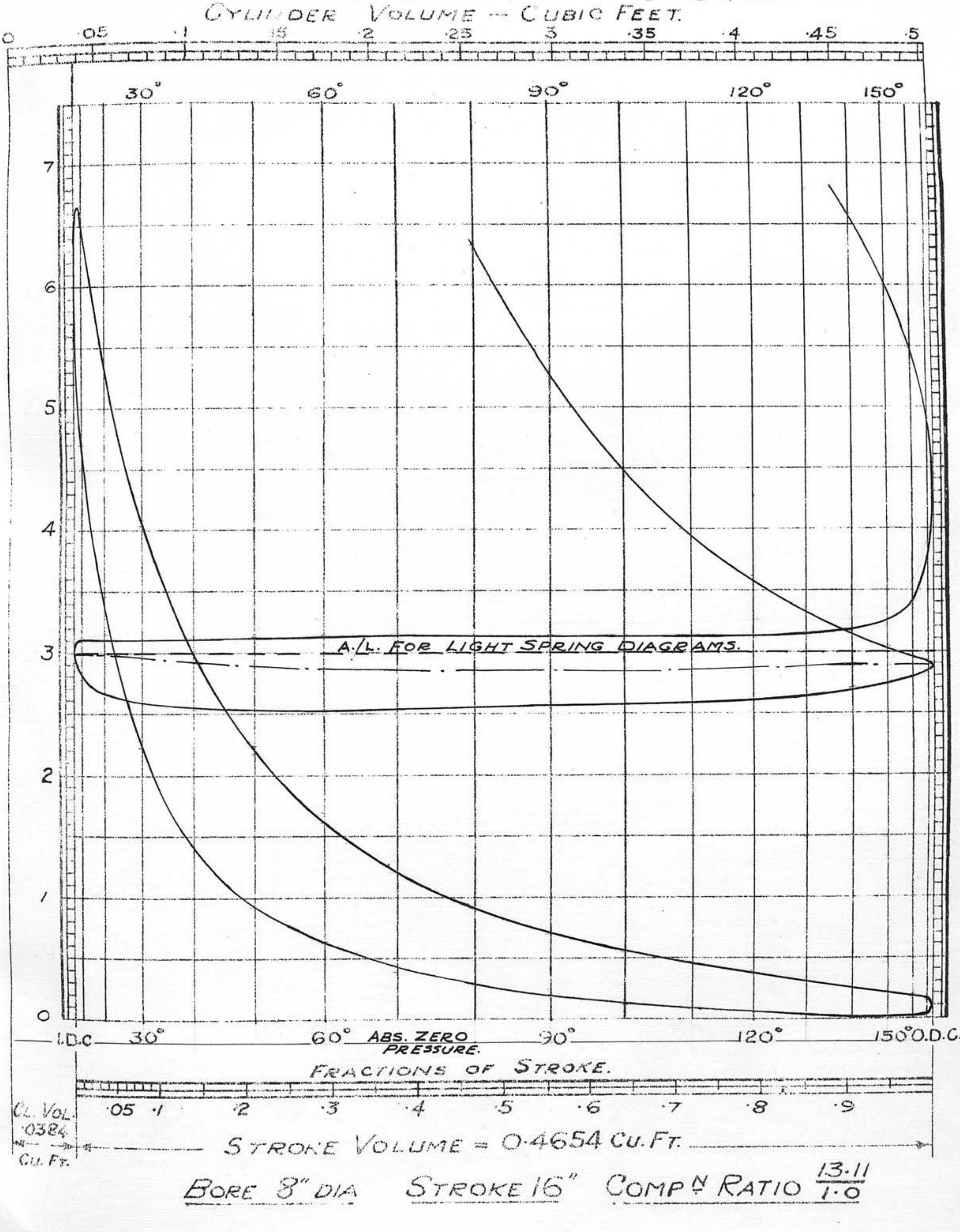



BORE 8" DIA STROKE 16" COMP^N RATIO $\frac{13 \cdot 11}{1 \cdot 0}$

# "NATIONAL" HEAVY OIL ENGINE.

FIG. 10.

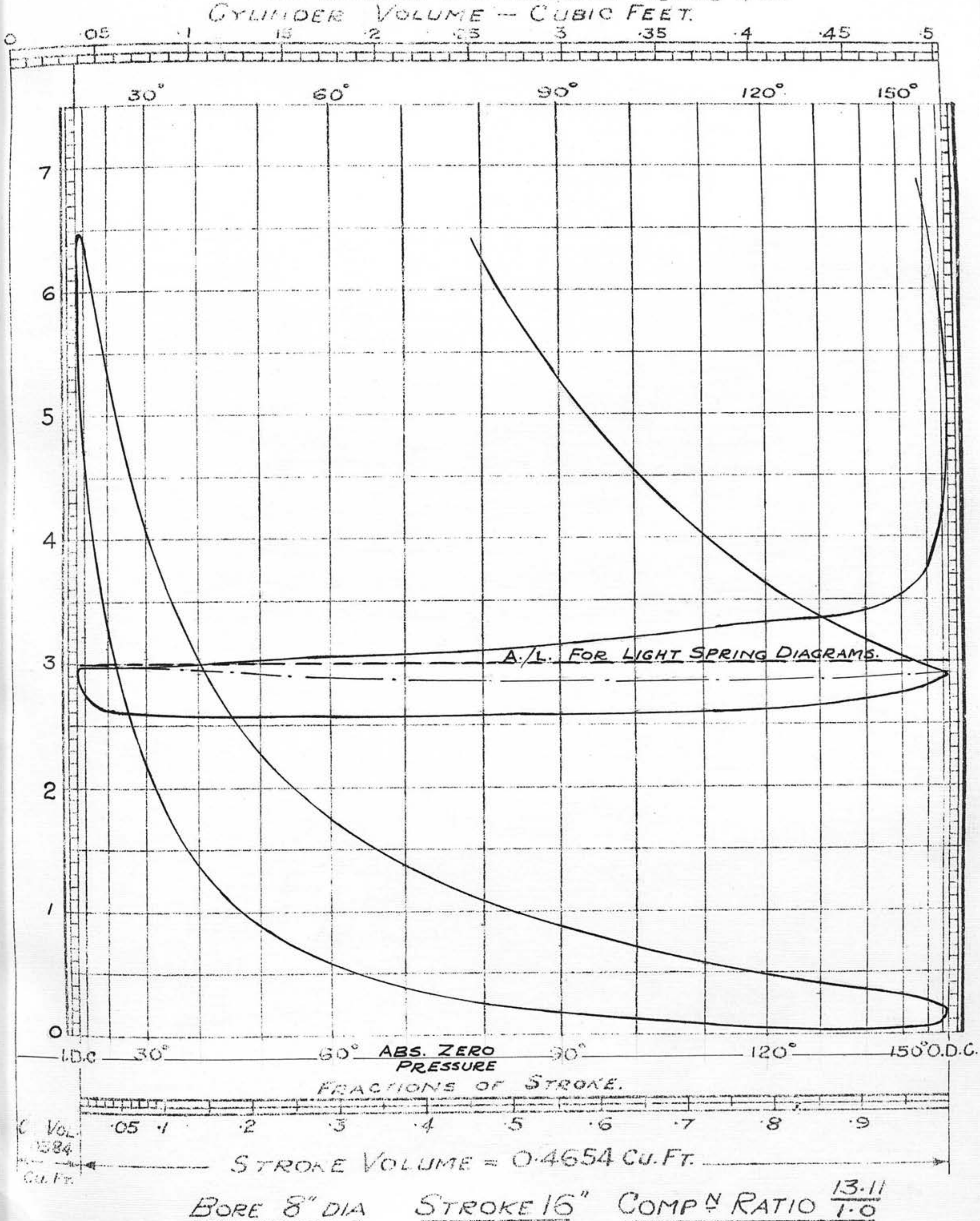
TEST P. 4. I.M.E.P. (gross) 62.2 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 3.7 lb./sq.in. Spring 1/6.9




BORE 8" DIA  STROKE 16"  COMPⁿ RATIO $\frac{13{\cdot}11}{1{\cdot}0}$

"NATIONAL" HEAVY OIL ENGINE. FIG. 11.

TEST P. 5. I.M.E.P. (gross) 72.5 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 3.45 lb./sq.in. Spring 1/6.9

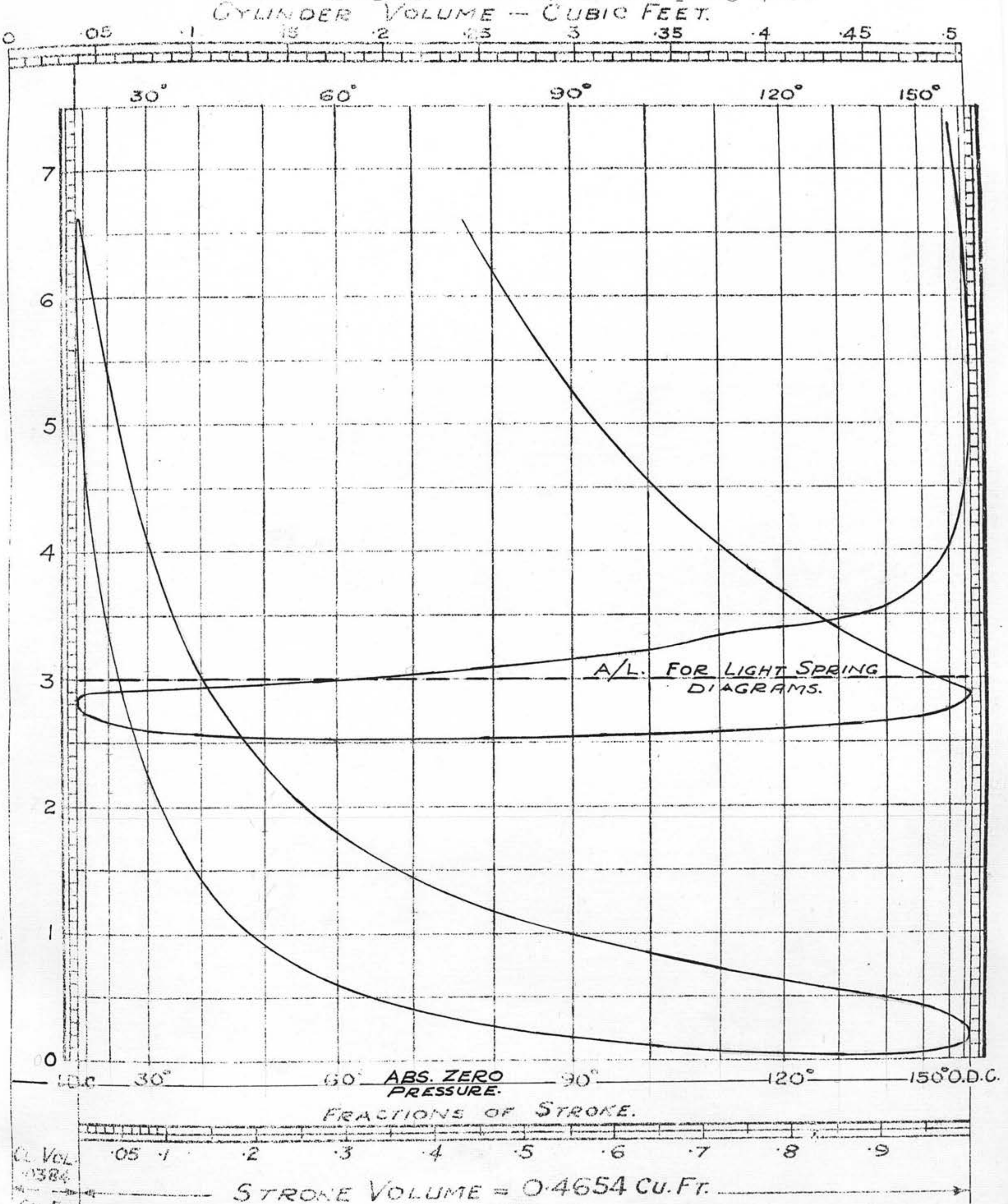



BORE 8" DIA STROKE 16" COMPN RATIO $\frac{13.11}{1.0}$

# "NATIONAL" HEAVY OIL ENGINE.

FIG. 12.

TEST p. 6. I.M.E.P. (gross) 78.1 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 3.13 lb./sq.in. Spring 1/6.9




BORE 8" DIA STROKE 16" COMPN RATIO $\frac{13.11}{1.0}$

Notes on the Nett I.H.P. and Friction H.P.

The I.H.P. (nett) has been calculated by deducting the pumping horse-power (as given by the light spring pv diagram) from the I.H.P. (gross). The horse-power expended in friction has been estimated as the difference between the B.H.P. and the nett I.H.P. It, therefore, includes piston friction, bearing friction, windage and the power required to operate the fuel pump, lubricating oil pump, inlet air valve and exhaust valve. No attempt has been made to separate these items.

Similarly, the Mechanical Efficiency has been taken as the ratio $\frac{\text{B.H.P.}}{\text{I.H.P.(nett)}}$

COMMENTS ON THE RESULTS:

Fig. 13a shows graphically the mechanical characteristics with reference to B.H.P.

The friction horse-power, calculated as the difference between two observed values not greatly different, is necessarily somewhat erratic. This would not have been so noticeable had the Friction H.P. been drawn to the same scale as the B.H.P.

It appears, however, that there is definitely a reduction in friction with increase of load and this/

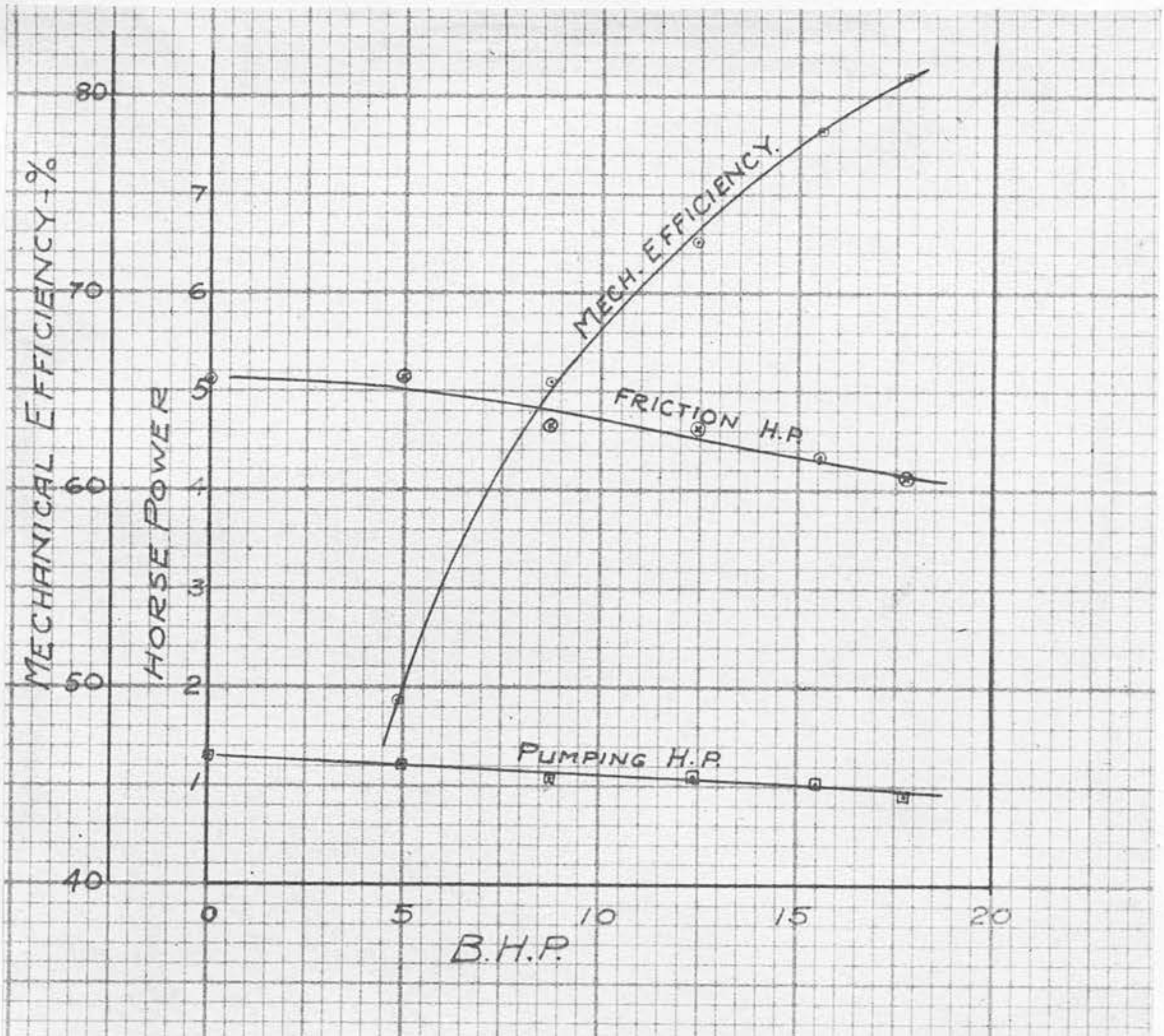



FIG 13a - MECHANICAL CHARACTERISTICS.

this may be accounted for chiefly by the reduction in piston friction, in turn due to a lower viscosity brought about by the higher temperature of the gases in the cylinder. Overloading, resulting in a more or less partial breakdown of the lubricating oil films, would no doubt raise the friction horse-power again.

The pumping power also shows a decrease with load and is due chiefly to the reduced back pressure at the higher loads. The large drop in pressure which takes place at release, must give rise to a high velocity of discharge and the inertia of gases no doubt helps in continuing the reduction in back pressure throughout the exhaust stroke, as observed at full load.

The mass of gas exhausted is sensibly constant for all loads whereas its volume (in the exhaust pipe), and consequently its Momentum, will be roughly proportional to the exhaust temperature, which increases with load. The deceleration of the piston towards the end of the stroke reduces the momentum and this change of momentum aids in continuing the reduction in back pressure right up to the end of the exhaust stroke. (See pv diagram, Fig. 12.) ?

The fuel characteristics with reference to B.H.P. are illustrated in Fig. 13b. They appear to be/

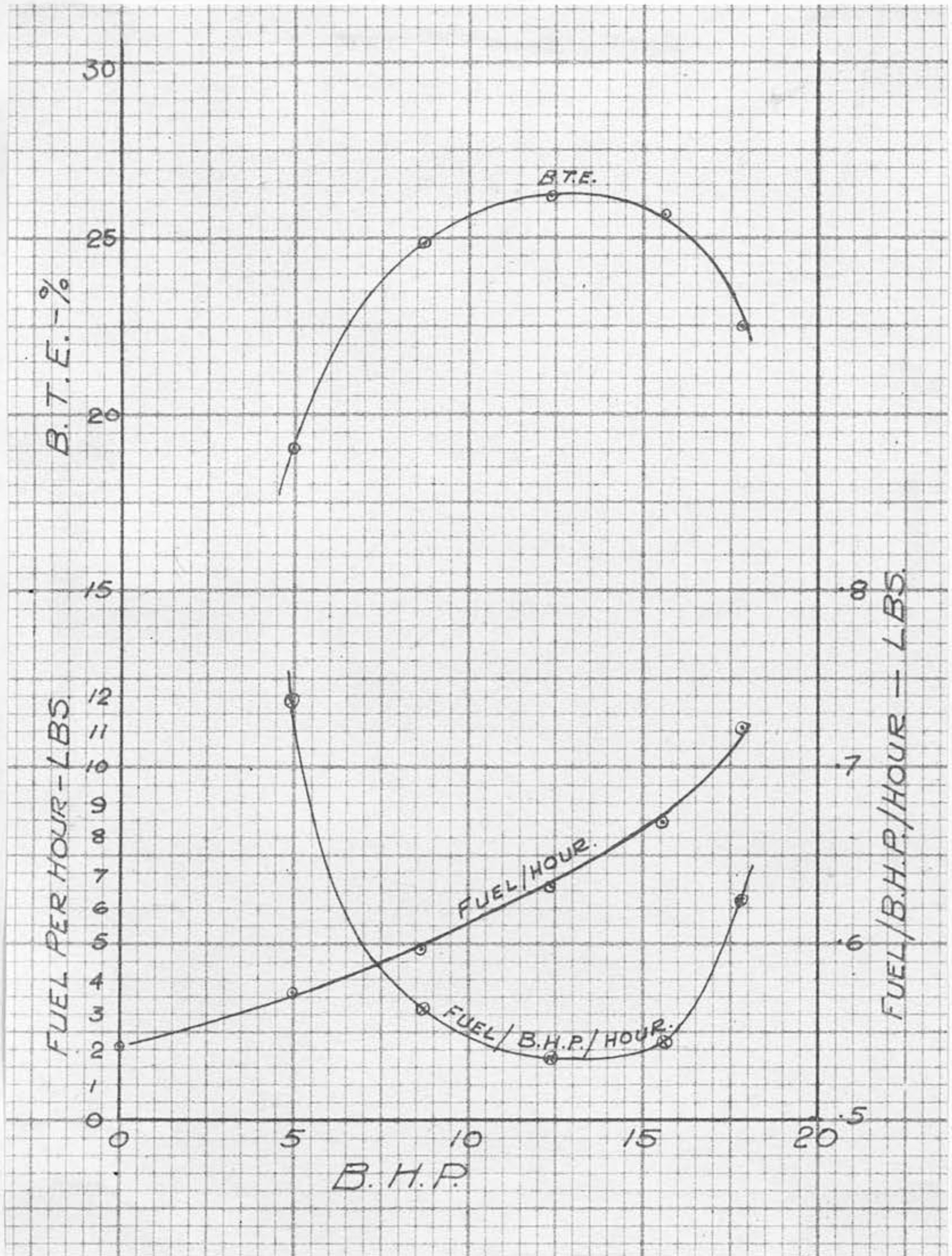



FIG. 13b - FUEL CHARACTERISTICS & B.H.P.

be typical of slow speed compression ignition engines though the rather sharp rise in fuel per hour at high loads, and the consequent fall in B.T.E. is more pronounced than usual. This feature is given more consideration later on.

Fig. 14 shows the fuel characteristics referred to I.H.P. (nett). It will be seen that the I.T.E.(on nett basis) reaches a maximum at nearly half load and that the efficiency falls off at about the same rate on either side of this load. The cause of the lower efficiency at light loads would appear to be due to the increased percentage which the pumping H.P. bears to the I.H.P. at light load. This is indicated by Fig. 15 where the I.T.E. is shown to a basis of gross I.H.P.

While it is perhaps more usual to give the I.H.P. and I.T.E. on the "nett" basis, from a thermodynamic point of view the pumping loss is really a loss external to the process of converting the fuel heat into work. Consequently, it would appear that a determination of thermal efficiency from the positive loop only of the indicator diagram is the correct method. This was adopted by Hopkinson* in 1907 and its validity very well explained by Captain Sankey+.

There/

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* Hopkinson - "Indicated Power and Mechanical Efficiency of the Gas Engine" (Proc.Inst.Mech.Eng. Vol. 4 p. 873. (1907)

\+ Sankey Discussion ibid p. 916

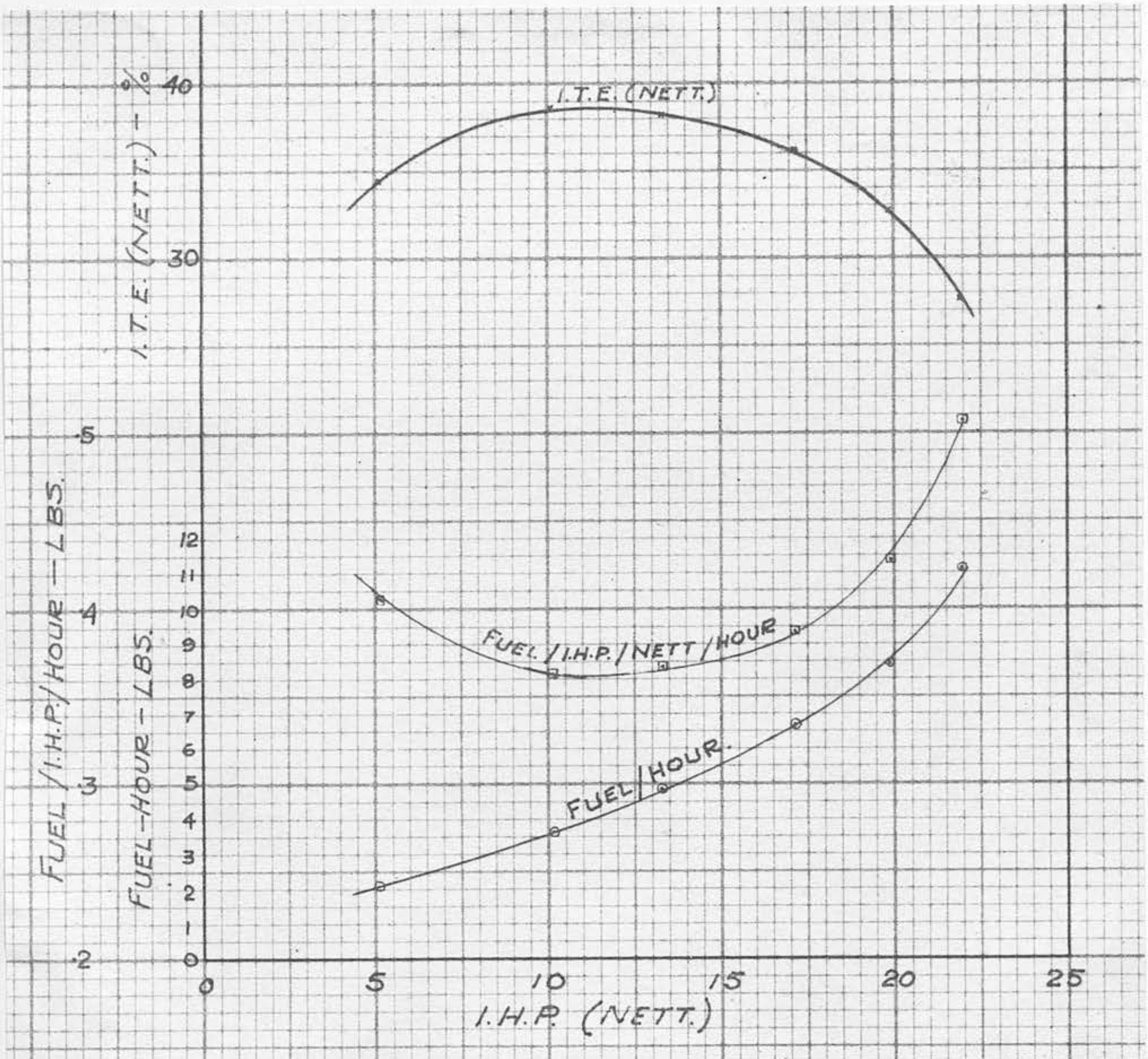



FIG. 14. — FUEL CHARACTERISTICS & I.H.P. (NETT.)

There would appear to be three main factors affecting the I.T.E.:-

(1) the specific heat. This increases:-

(a) with temperature;

(b) as the air/fuel ratio decreases.

Effect (b) is usually accompanied by higher temperatures so that increase in Power output is accompanied by a cumulative increase in Specific Heat and a lower efficiency of conversion of heat to work.

(2) the heat losses during combustion and expansion.

(3) the period of combustion.

In order to investigate the heat losses, certain additional data have been obtained, chiefly from the indicator diagram and these have been tabulated (Table III).

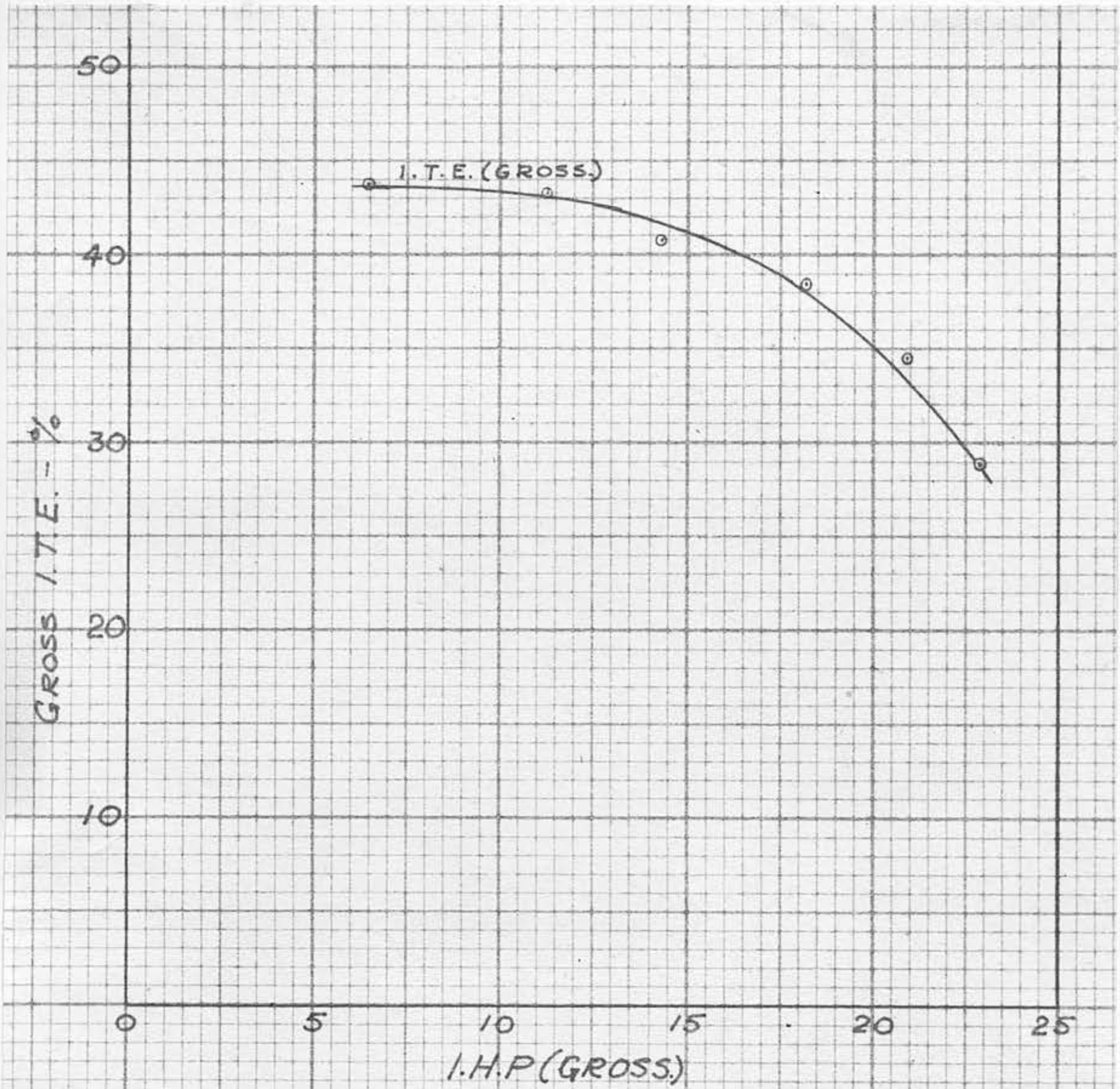



FIG. 15 — I.T.E. ON BASIS OF I.H.P. (GROSS.)

TABLE III.

| Test No. | P 1 | P 2 | P 3 | P 4 | P 5 | P 6 |
|---|---|---|---|---|---|---|
| I.H.P. (gross) | 6.44 | 11.36 | 14.37 | 18.22 | 20.93 | 22.88 |
| I.H.P. (nett) | 5.11 | 10.13 | 13.34 | 17.14 | 19.92 | 21.96 |
| Air/Fuel Weight Ratio | 124.8 | 72.3 | 52.0 | 38.5 | 29.7 | 22.1 |
| Excess Air % | 767 | 402 | 261 | 157 | 106 | 53.5 |
| Air used per cycle - lb. | .0298 | .0304 | .0296 | .0296 | .0293 | .0284 |
| Residuals in Clearance Volume - lb. | .0026 | .0021 | .00175 | .00155 | .0013 | .0010 |
| Weight of charge during Compression | .0324 | .0325 | .03135 | .03115 | .0306 | .0294 |
| Suction Temperature °C. | 31 | 47 | 49 | 56 | 67 | 75 |
| Fuel per 1000 cycles lb. | .24 | .418 | .57 | .769 | .98 | 1.287 |
| Weight of charge during Expansion - lb. | .03254 | .03292 | .03190 | .03192 | .03158 | .03069 |
| Injection begins - degrees before I.D.C. | 6.0 | 8.2 | 10.0 | 11.0 | 11.0 | 11.5 |
| Injection ends - degrees after I.D.C. | 12.0 | 21.5 | 29.5 | 40.5 | 55.5 | 72.5 |
| Period of injection degs. | 18.0 | 29.7 | 39.5 | 51.5 | 66.5 | 84.0 |
| Combustion begins approx. degrees before I.D.C. | 1.0 | 4.0 | 5.0 | 6.2 | 6.6 | 7.5 |
| Delay Period (approx.) degrees | 5.0 | 4.2 | 5.0 | 4.8 | 4.4 | 4.0 |

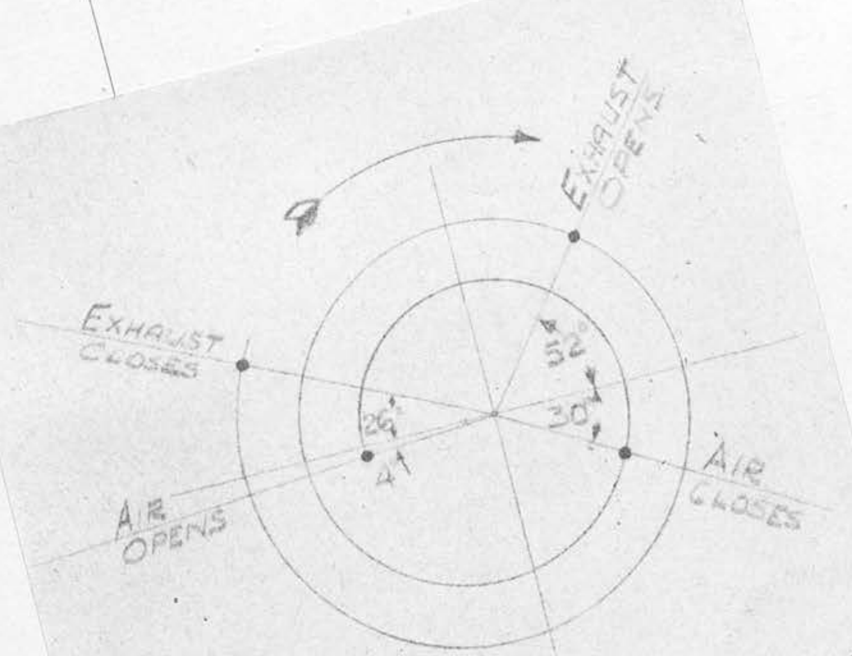



FIG 16- VALVE TIMING DIAGRAM.

NOTES ON TABLE III:

Weight of Charge.

(1) During Compression.

This is made up of:-

(a) The air consumed per cycle.

(b) The weight of residuals i.e. exhaust gases not completely evacuated from the cylinder.

The weight of air passing through the engine per cycle can be measured accurately by means of a gasometer.

The weight of residuals will be mainly dependent on the weight of exhaust gases in the clearance space at the end of the exhaust stroke. Unfortunately, there is approximately thirty degrees overlap between the opening of the inlet air valve and the closing of the exhaust valve (see Fig. 16), so that there might be expected, at full power, where the back pressure is low, a further evacuation of the exhaust during the early part of suction.

Neglecting this effect in the first instance, the weight of residuals can be estimated from the pressure, volume and temperature in the clearance space at the end of exhaust. The pressure and volume can be found from the indicator diagram. Direct measurement of the temperature of the residuals at instant of I.D.C. is not/

not practicable so the temperature recorded in the exhaust pipe about 6 inches from the exhaust valve has been used. Taking into account the various heat losses during exhaust and the fact that the thermocouples will give the mean temperature reached at that point between the cycles, it is considered that this will give a fair indication of the temperature existing in the clearance space at the beginning of suction.

Then Weight of Residuals = $W_r = \frac{p_r V_r}{R\ T_r}$ .........(1)

where R = Gas Constant for the residuals.

The value of R is dependent on the molecular constitution of the exhaust and is directly affected by the air/fuel ratio.

$$R = \frac{\text{Universal Gas Constant}}{\text{Molecular Weight of Exhaust Gas mixture}} = \frac{\bar{R}}{M_m}$$

Values of $M_m$ and R for the extreme range of air/fuel ratio employed in these tests have been calculated for the particular fuel used. They are as follows:-

| Air/Fuel Ratio | $M_m$ | R ft.lbs. |
|---|---|---|
| 124.8 | 28.86 | 96.3 |
| 22.1 | 28.89 | 96.25 |

It/

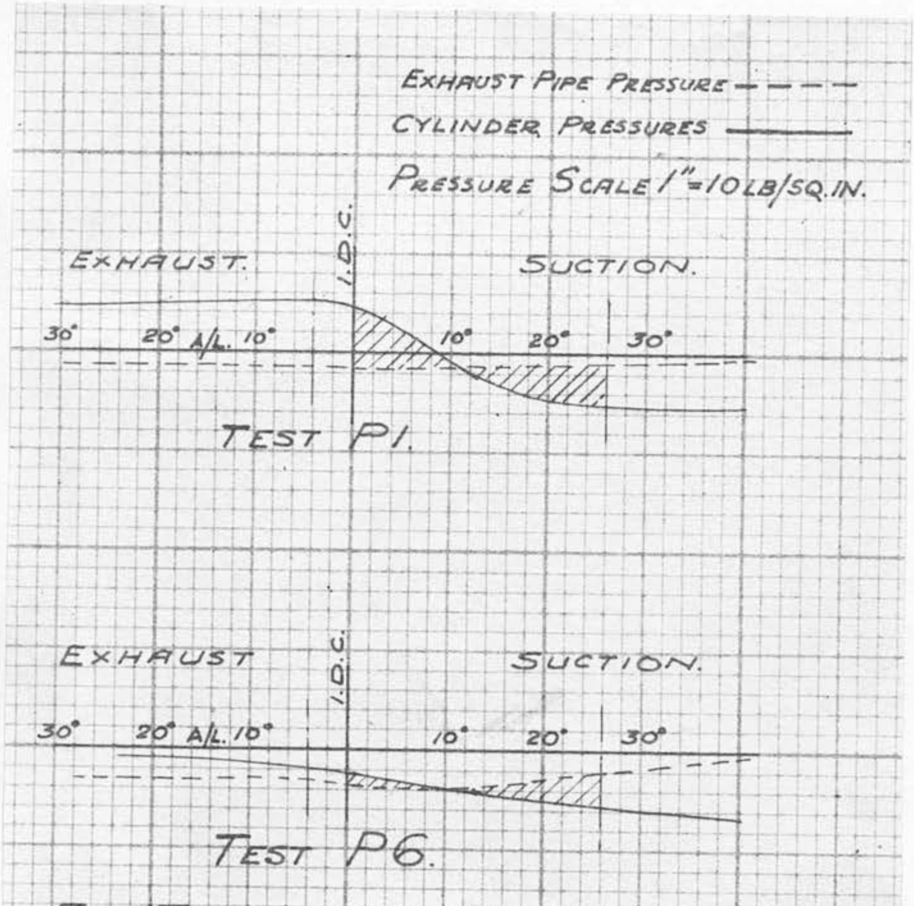



FIG 17. – EXHAUST PIPE PRESSURES.

It is considered that this difference is negligible for the purpose of calculating temperatures etc., and throughout these tests the value of R for the exhaust products and for the charge prior to combustion has been taken as 96.3 in the foot lb. Centigrade system.

To check up on the possible loss or gain of products through the exhaust valve early in the suction stroke, it was decided to fit a Dobbie McInnes Pressure Disc valve unit in the exhaust pipe and measure the pressure variations. Records were taken for conditions similar to tests P 1 and P 6 and are shown in Fig. 17 superimposed on records of cylinder pressures taken previously.

It will be seen that, in both cases, the depression during the beginning of suction is small and that, for the greater part of the first twenty-six degrees of suction stroke, the cylinder pressure was less than that of the exhaust. On the other hand, while the exhaust pressure was less than the cylinder pressure, the valve opening was greater, so that the nett effect seems to be neither gain nor loss through the exhaust valve.

Hence the weight of residuals mixed with the air at the end of suction has been taken as that obtained from/

from equation (1) on page 22.

Then Weight of Charge during compression

$$W_1 = W_a + W_r \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

where $W_a$ = Weight of Air per cycle.

(2) Weight of Charge after Combustion.

$$W_2 = W_1 + W_o \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

where $W_o$ = Weight of Fuel injected per cycle.

(3) Weight of Charge during Combustion.

$$W_2{}^1 = W_1 + xW_o \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

where $x$ = fraction of oil burned at any particular instant.

Temperature of Charge.

(1) During Compression.

Absolute Temperature $T = \dfrac{pv}{W_1R_1} = \dfrac{pv}{96.3W_1}$ deg.C.abs. .. (5)

where p and v are obtained from indicator diagram.

$W_1$ is obtained from equation (2) above.

(2) Temperature of Charge during expansion after Combustion.

$$T = \frac{pv}{96.3W_2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

where $W_2 = W_1 + W_o$ as in equation (3) above.

(3) Temperature of Charge during Combustion.

$$T = \frac{pv}{96.3W_2{}^1}$$

where $W_2{}^1 = W_1 + xW_o$ as in equation (4) above.

Figs./

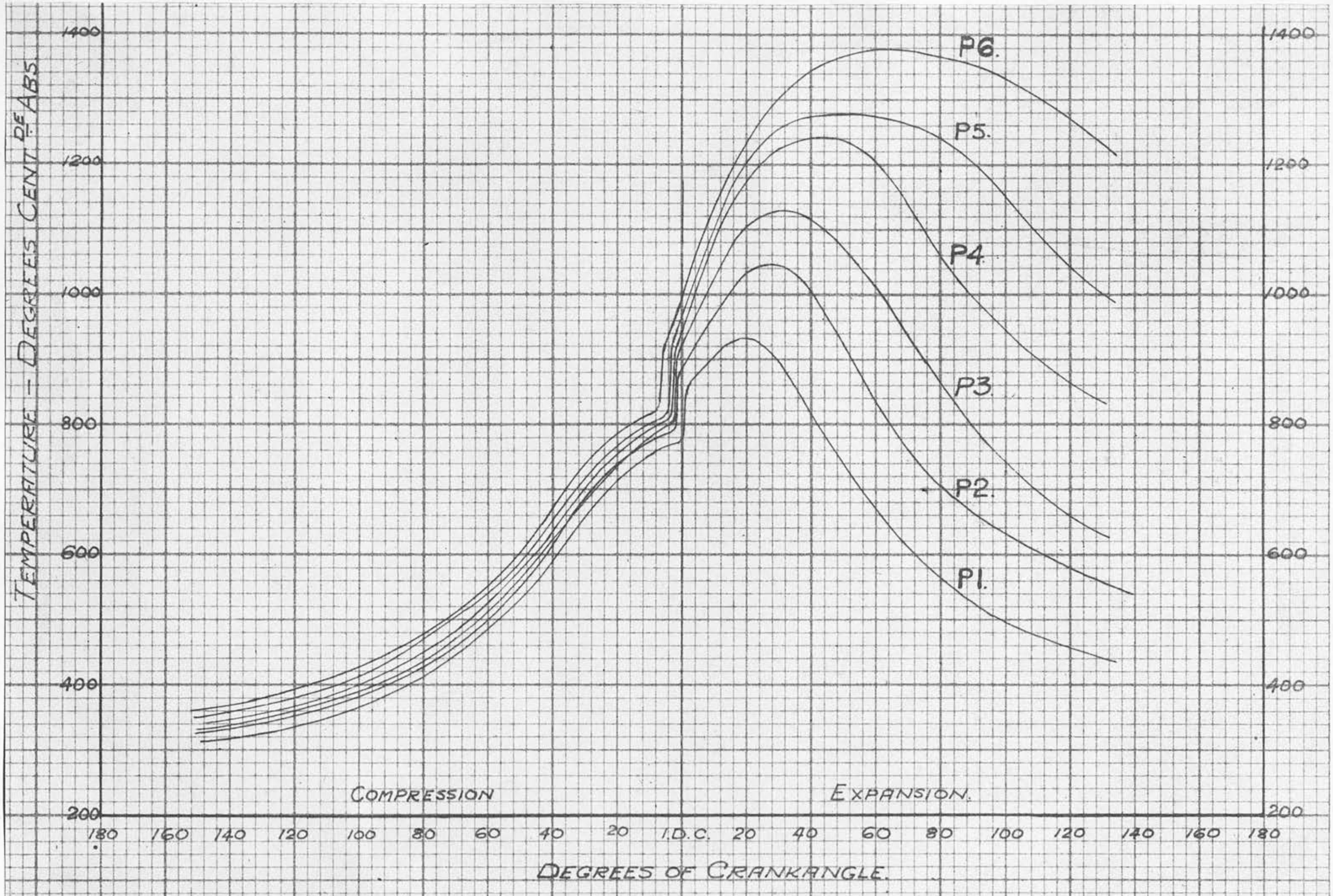


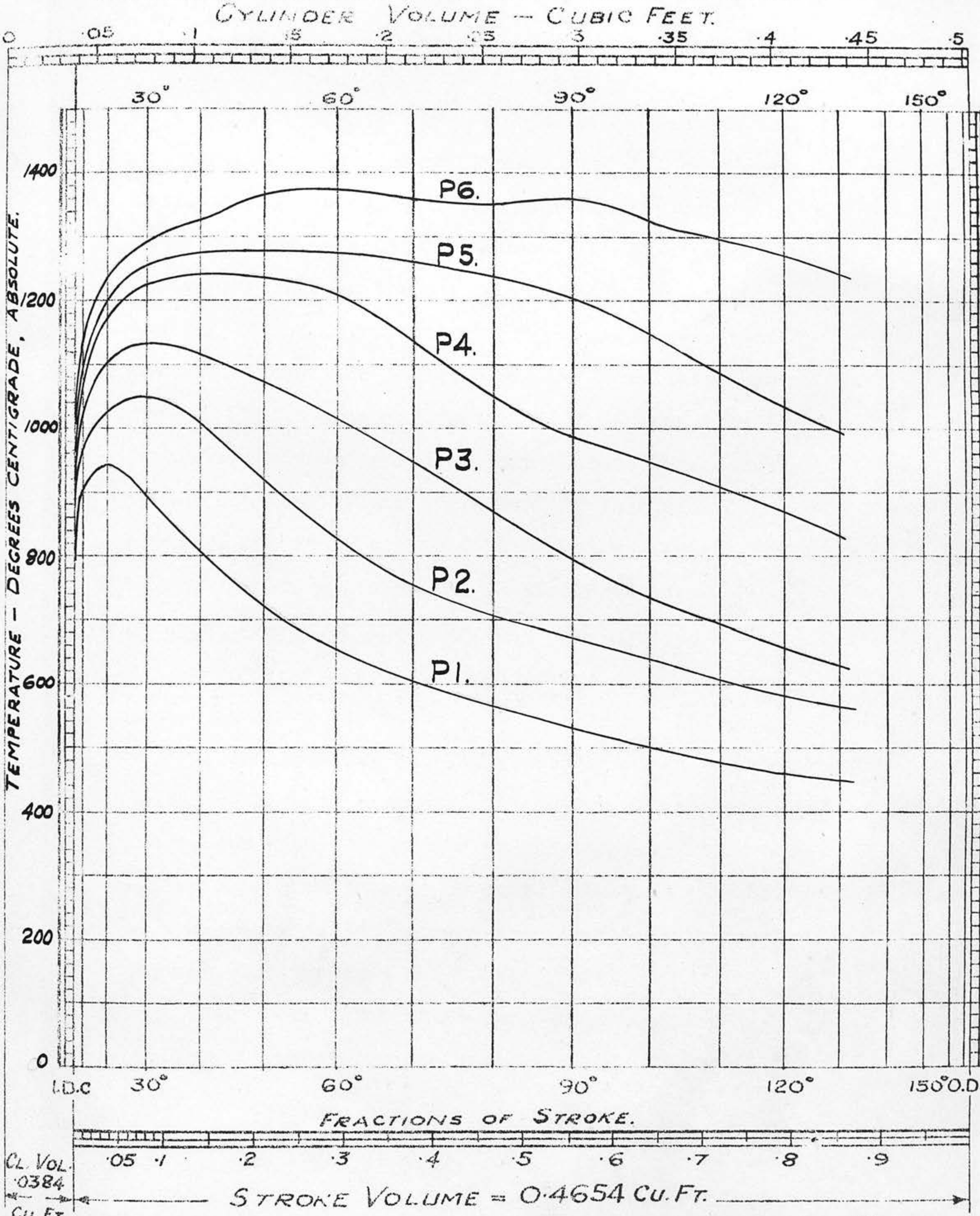

Fig. 18 — Cyclical Variation of Temperature (Tests P1–P6.)

# "NATIONAL" HEAVY OIL ENGINE.

FIG. 19.

TESTS P.1. - P.6.

VARIATION OF CYLINDER GAS TEMPERATURES DURING EXPANSION.




BORE 8" DIA STROKE 16" COMPN RATIO $\frac{13 \cdot 11}{1 \cdot 0}$

Figs. 18 and 19 show the cyclical variation of temperature inside the cylinder. It will be seen that, at high loads, the temperature during expansion is maintained at a fairly constant level for about fifty per cent. of the stroke, and tends to support the suggestion put forward by F.W. Lanchester* that the solid injection engine cycle is a mutilated Carnot cycle. While this may be true at full load, the light load diagram would certainly approximate more closely to the constant volume cycle had the injection begun earlier, and it is at light load that the engine would be expected to approach the ideal condition.

Internal Energy of the Charge.

For any change of temperature:-

Change of Internal Energy = $\int_{T_1}^{T_2} C_V dT$ for one lb. of gas.

$C_V$ is dependent both on pressure and on temperature. While the effect of temperature is fairly accurately known, our knowledge of the effect of pressure is not reliable. Such data as do exist seem to show that, at the pressures and temperatures operating in oil engines, the pressure has little effect.

Curves of Internal Energy, plotted against absolute temperature have been obtained in accordance with/

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* F.W. Lanchester - "Energy Balance Sheet of the I.C. Engine" (Proc. of Inst.Mech.E. 1939 Vol. 141 No. 4 p. 315 )

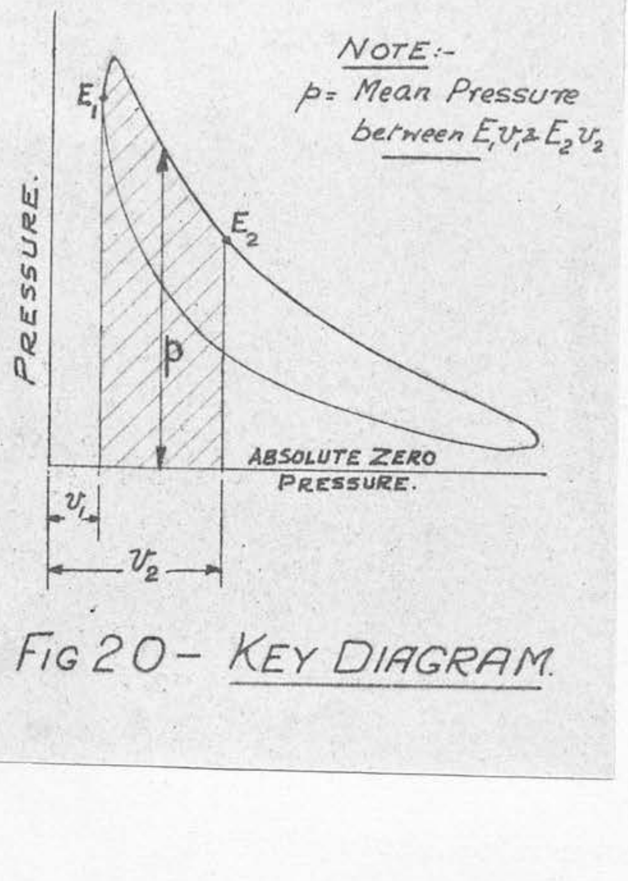



FIG 20 - KEY DIAGRAM.

with Partington & Shilling's values for the specific heat of $CO_2$, $H_2O$, etc.* No allowance has been made for the latent Heat of Steam when assessing the proportion of Internal Energy provided by the $H_2O$ content. This would appear to be correct in view of the fact that the L.C.V. of the fuel has been used in all calculations. The curves cover the range of air/fuel ratio employed and have been drawn from a zero energy at 273° C. abs. (See Chart in rear cover pocket.)

Heat Reception Curves (Figs. 21 and 22.)

These curves are based on the thermodynamic relation affecting the change of state of a mass of gas. During any such change, the nett heat received by the gas is equal to the work done by the gas together with the increase of Internal Energy.

Applying this equation to the key diagram shown in Fig. 20:-

Heat supplied $= Q = \frac{p(v_2 - v_1)}{J} + W(E_2 - E_1)$ heat units.

when $E_2 - E_1$ = Charge of Internal Energy per lb. of gas.

$W$ = Weight of gas - lbs.

i.e./

n ?

* Partington & Shilling - "The Specific Heat of Gases" pp. 208, 209 (1934 Edition)

TABLE IV.

Compression.

| Crank Angle | Cylr. Vol. cu.ft. | Pressure lb./sq.ft. | W | T | E C.H.U./lb. | W×E | Work from I.D.C. Area sq.in. | Work from I.D.C. C.H.U. | Work + W×E |
|---|---|---|---|---|---|---|---|---|---|
| I.D.C. | .0384 | 72,100 | .0326 | 882 | 107.5 | 3.50 | – | – | 3.50 |
| 10 | .0427 | 56,900 | .0325 | 777 | 88.0 | 2.86 | .38 | .205 | 3.065 |
| 20 | .0554 | 41,400 | .0325 | 733 | 80.2 | 2.61 | 1.19 | .643 | 3.253 |
| 30 | .0759 | 27,400 | .0325 | 697 | 74.0 | 2.407 | 2.13 | 1.152 | 3.559 |
| 40 | .1033 | 18,430 | .0325 | 608 | 58.5 | 1.90 | 2.97 | 1.606 | 3.506 |
| 50 | .1362 | 12,670 | .0325 | 551 | 48.5 | 1.576 | 3.62 | 1.957 | 3.533 |
| 60 | .1737 | 8,930 | .0325 | 495 | 38.5 | 1.25 | 4.16 | 2.248 | 3.498 |
| 70 | .2136 | 6,680 | .0325 | 455 | 31.5 | 1.025 | 4.58 | 2.476 | 3.501 |
| 80 | .2551 | 5,180 | .0325 | 421 | 25.3 | .823 | 4.92 | 2.66 | 3.483 |
| 90 | .2964 | 4,230 | .0325 | 401 | 21.6 | .703 | 5.15 | 2.785 | 3.488 |
| 110 | .3730 | 3,065 | .0325 | 365 | 15.5 | .503 | 5.515 | 2.98 | 3.483 |
| 130 | .4354 | 2,450 | .0325 | 340 | 11.3 | .367 | 5.73 | 3.10 | 3.467 |
| 150 | .4789 | 2,130 | .0325 | 326 | 9.2 | .299 | 5.86 | 3.166 | 3.465 |

Expansion.

| Crank Angle | Cylr. Vol. cu.ft. | Pressure lb./sq.ft. | W | T | E C.H.U./lb. | W×E | Work from I.D.C. Area sq.in. | Work from I.D.C. C.H.U. | Work + W×E |
|---|---|---|---|---|---|---|---|---|---|
| I.D.C. | .0384 | 72,100 | .0326 | 882 | 107.5 | 3.50 | - | - | 3.50 |
| 10 | .0427 | 71,400 | .0327 | 969 | 123.5 | 4.03 | .45 | .243 | 4.273 |
| 20 | .0554 | 59,000 | .0328 | 1036 | 137.0 | 4.49 | 1.55 | .838 | 5.328 |
| 30 | .0759 | 43,750 | .0329 | 1048 | 140.0 | 4.61 | 2.95 | 1.595 | 6.205 |
| 40 | .1033 | 30,800 | .03292 | 1003 | 132.5 | 4.36 | 4.32 | 2.335 | 6.695 |
| 50 | .1362 | 21,600 | .03292 | 927 | 118.5 | 3.9 | 5.445 | 2.946 | 6.846 |
| 60 | .1737 | 15,200 | .03292 | 832 | 100.7 | 3.31 | 6.36 | 3.436 | 6.746 |
| 70 | .2136 | 11,060 | .03292 | 744 | 84.5 | 2.78 | 7.02 | 3.795 | 6.575 |
| 80 | .2551 | 8,850 | .03292 | 711 | 78.5 | 2.584 | 7.58 | 4.1 | 6.68 |
| 90 | .2964 | 7,250 | .03292 | 676 | 71.5 | 2.354 | 7.95 | 4.33 | 6.68 |
| 110 | .3730 | 5,180 | .03292 | 608 | 59.6 | 1.96 | 8.6 | 4.65 | 6.61 |
| 130 | .4354 | 4,030 | .03292 | 553 | 49.9 | 1.64 | 9.06 | 4.89 | 6.53 |

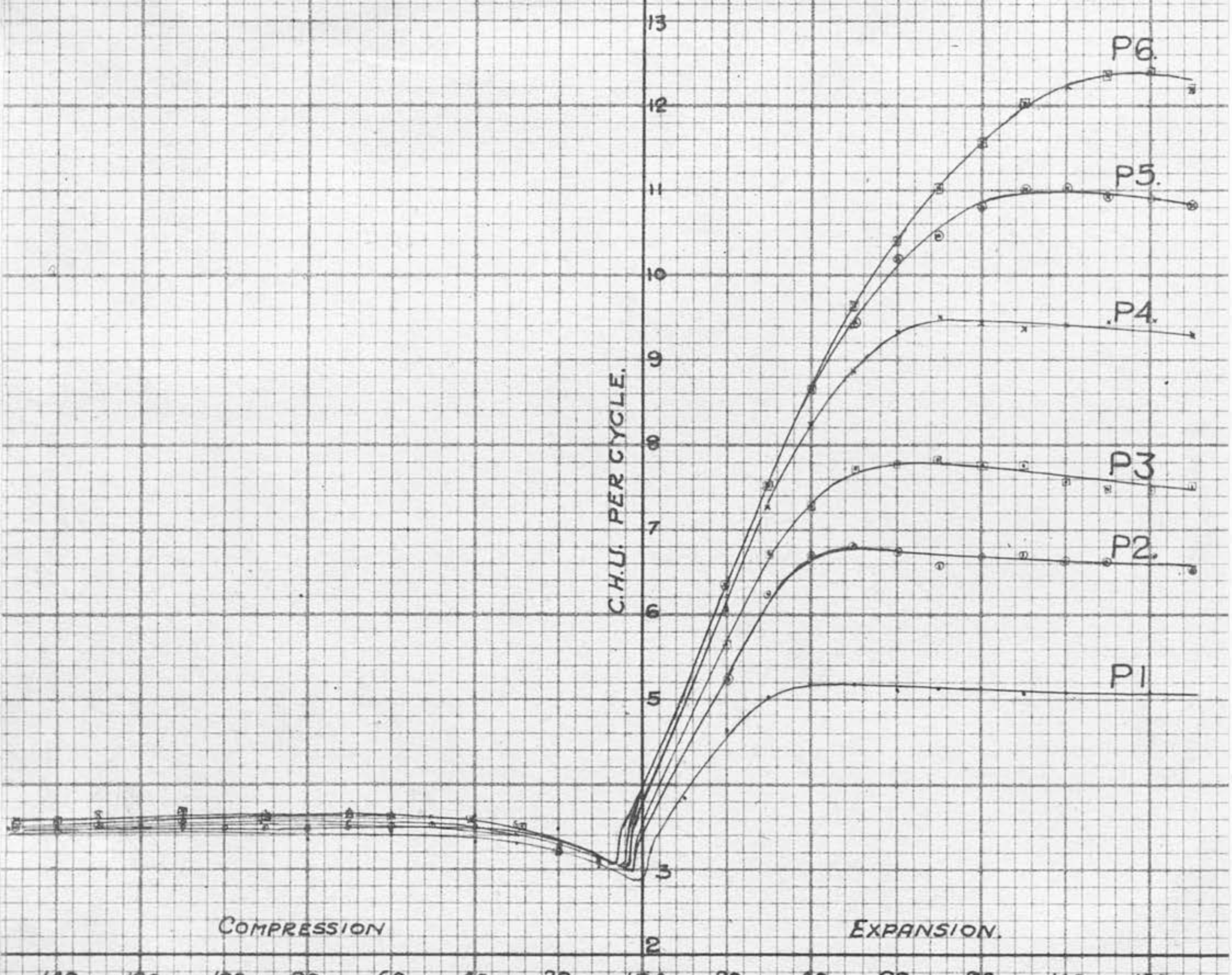



FIG. 21. TESTS P1-P6. BASIC CURVES.

The values of $\left\{\frac{p(v_2 - v_1)}{J} + WE_2\right\}$ as given by the data in the "Work + W.E." column, have been plotted together with similar values for the remaining five tests of this series, and are shown in Fig. 21.

The value of $WE_1$ - the internal energy of the gases at I.D.C. - when subtracted from the value of $\left\{\frac{p(v_2 - v_1)}{J} + WE_2\right\}$ will give the heat supplied between I.D.C. and any particular crank angle position. Hence, to determine the heat supplied between two crank angle positions, all that is required is to measure from the curve the change in the value of $\left\{\frac{p(v_2 - v_1)}{J} + WE_2\right\}$

To determine the heat supplied from the commencement of combustion, it will, of course, be necessary to determine first the crank angle positions at which combustion begins. It is customary to assume that combustion begins at that point where the cylinder pressure rises above the normal compression pressure but reference to the pressure crank angle diagram for Test P. 2. (enclosed in rear cover.pocket) shows that this point is not at all well defined. This is due partly to the slow speed of the engine but mainly to the fact that the quantity of fuel injected is never the same for two consecutive cycles. A method suggested/

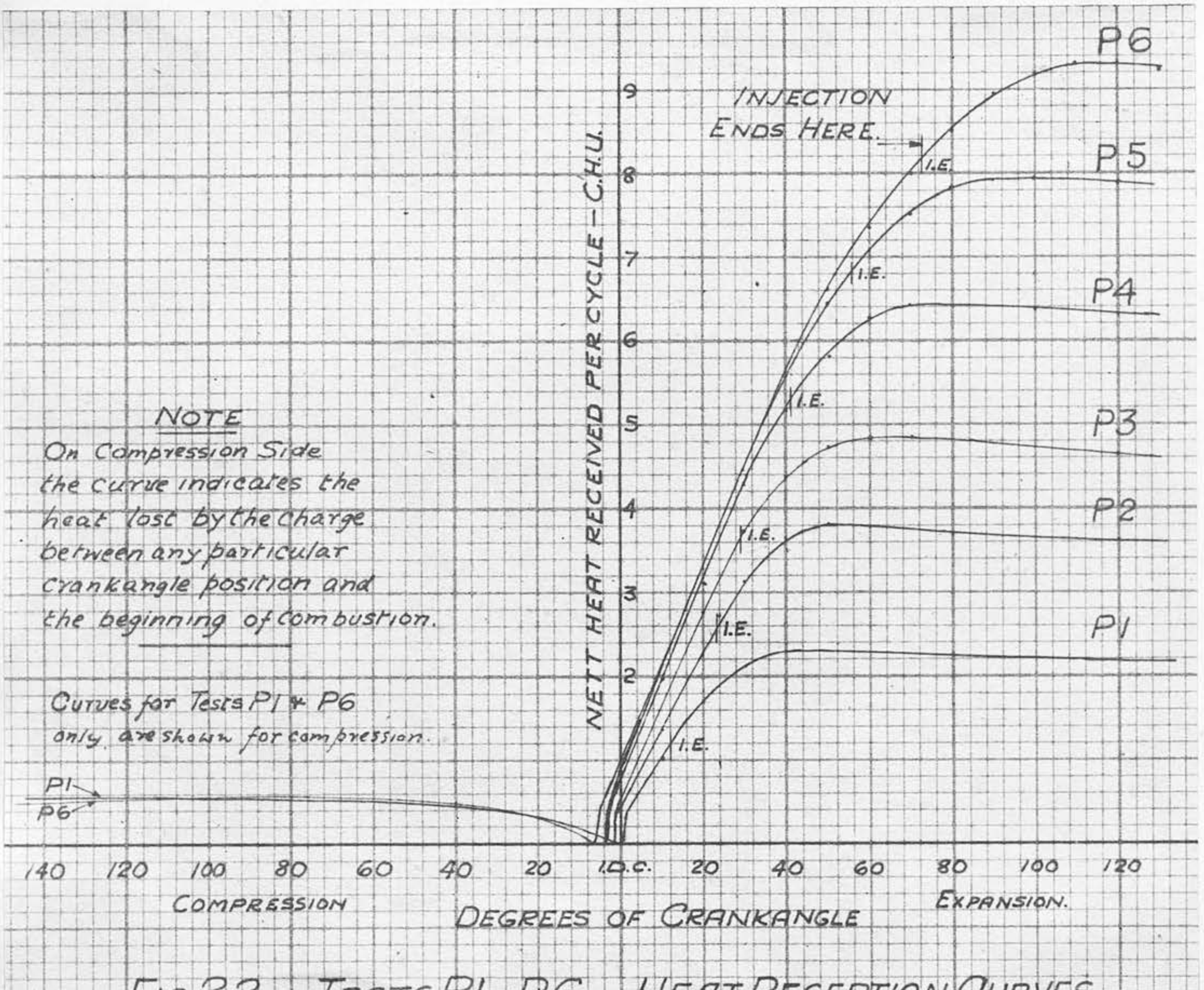



FIG. 22. TESTS P1-P6. HEAT RECEPTION CURVES.

suggested by Dr. J.Riffkin * is to use the point of inflexion of the pressure crank angle curve. While this is, perhaps, better defined, it cannot be said to represent the point at which heat from the fuel is released although up to the point at which rapid combustion takes place (as indicated by the steep pressure rise of the indicator diagram) the amount released would be quite small. In drawing up the final heat reception curves, a combination of these two methods was adopted. The first method was used to find the point at which the cylinder gases began to receive heat and the second method to find the point at which rapid evolution of heat began. No attempt has been made to follow accurately the heat reception curve during this period of rapid combustion.

Referring to Fig. 22, the most noticeable features appear to be:-

1. Combustion begins late at light load and early at full load.

This results from the fuel injection characteristics and reference to Table III (page 20) shows that/

* Dr. J. Riffkin - "Ignition Quality of Diesel Fuels" "Engineering", 6th January 1939

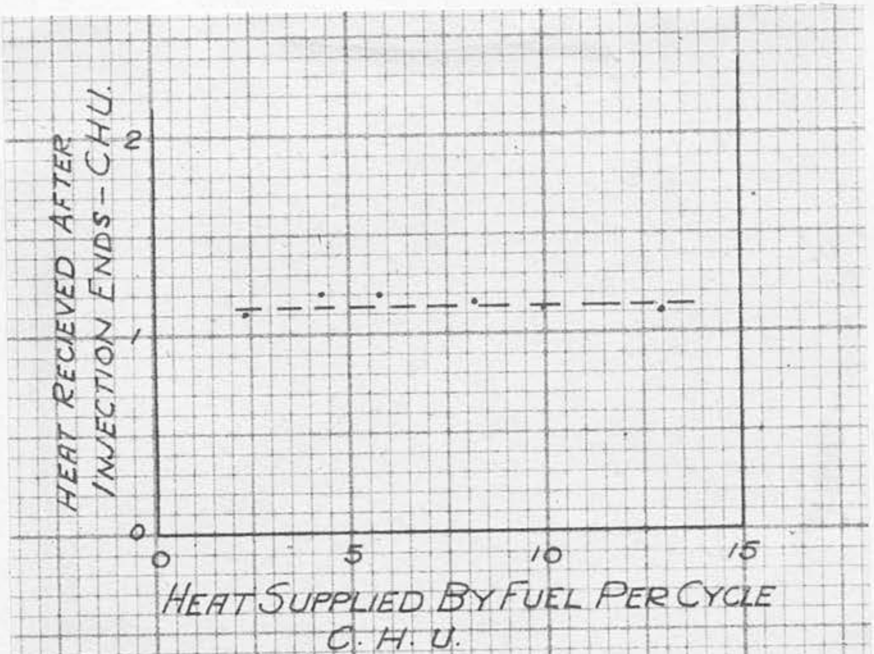



FIG. 23 - HEAT RECEIVED AFTER INJECTION ENDS.

that there is a difference of 5.5 degrees in the start of injection over the whole load range. This corresponds roughly with the heat reception curves.

2. The rate of combustion during the first forty degrees increases with the load on the engine.

The curves represent the nett heat received by the gases and, since the radiation loss etc. will increase with the load, the difference between the rates of combustion will be even more marked than that indicated by the heat reception curves.

3. The heat evolved during the period of rapid combustion immediately following the delay is small and sensibly constant for all loads - about .39 C.H.U. at no load and .49 C.H.U. at full load, i.e. 16% and 3.7% of the respective heat received during the cycle. This difference is not sufficient to have any material effect on the I.T.E.

4. The period of combustion increases with the load.

This will be directly affected by the period of fuel injection and reference to Table III (page 20) has shown that the period of injection does increase with the engine load. The crank angle position at which injection ends has been marked on the heat reception curve and it will be seen that the heat received/

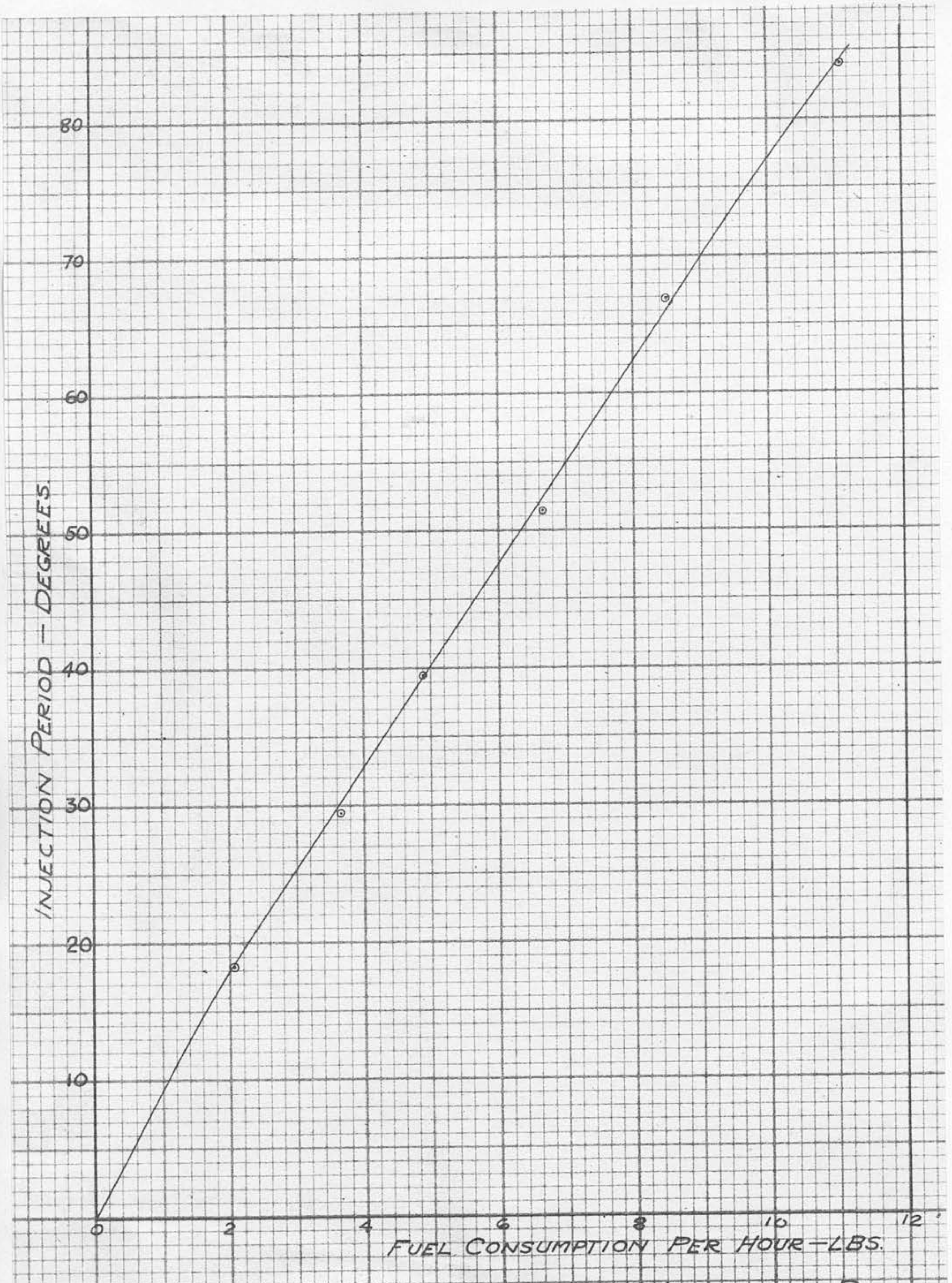



FIG. 24. FUEL CONSUMPTION & INJECTION PERIOD.

received by the gases after injection has ceased is not greatly different over the whole load range (see also Fig. 23.)

This would appear to indicate that, under all loads except the very lightest, the fuel is burned at a rate dependent on its rate of injection into the cylinder.

Fuel Injection Characteristics of the Engine.

In Fig. 24, the period of injection has been plotted against the fuel consumption in pounds per hour. This shows that the injection period is almost proportional to the fuel consumption per hour. If this is so, then it follows that the mean velocity of discharge during injection must be sensibly constant for all loads. To shed some light on this result, fuel injection pressure diagrams taken during the six tests have been plotted to a crank angle base in Fig. 25a. It will be seen that the maximum pressure during injection increases very considerably with the engine load - 1650 lbs. per sq.in. at no load, 3,300 lbs. per sq.in. at full load.

This rise of pressure with load is due to the arrangements made for governing the engine. A sensitive/

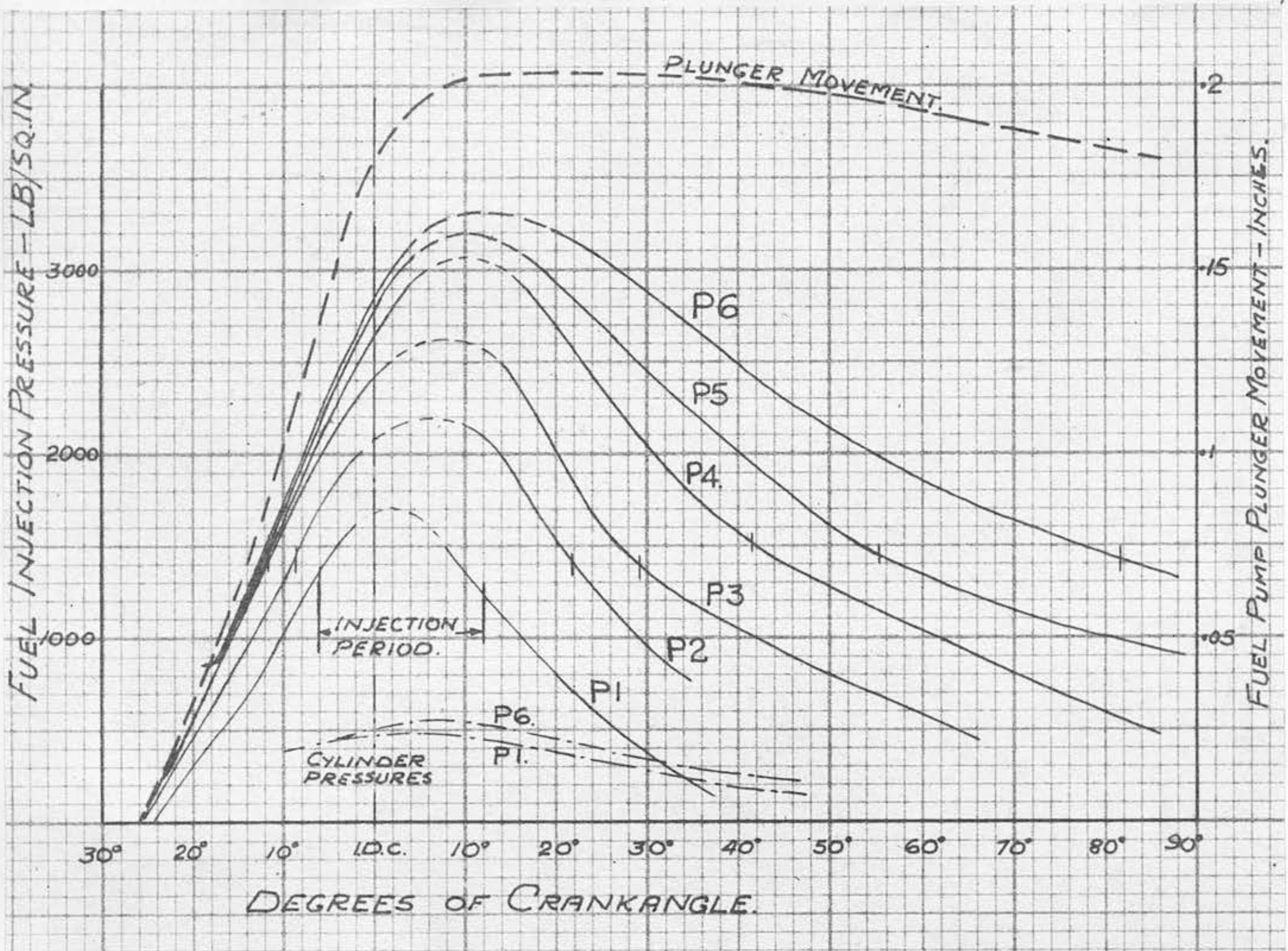
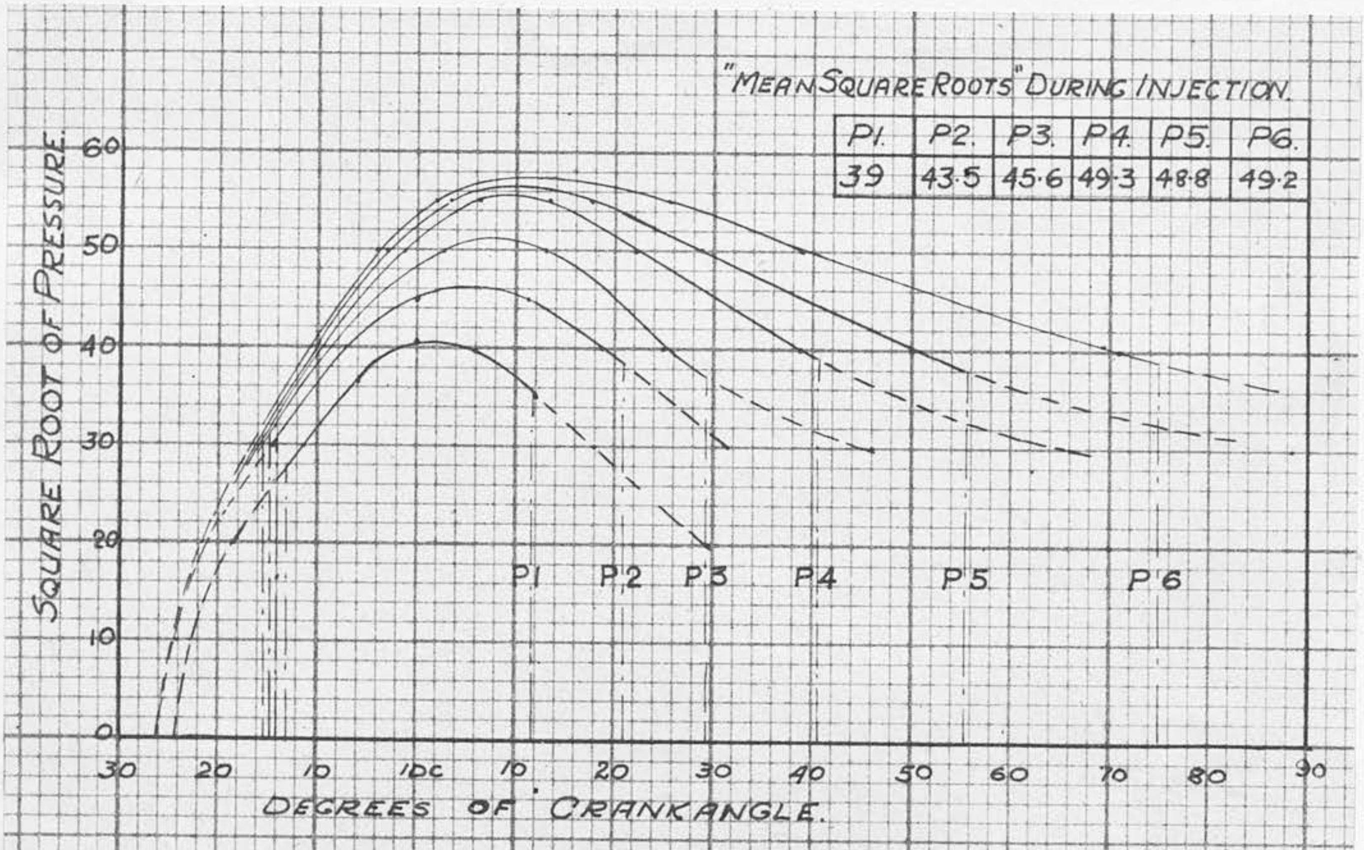



FIG. 25(a). FUEL INJECTION PRESSURES




"MEAN SQUARE ROOTS" DURING INJECTION.

| P1. | P2. | P3. | P4. | P5. | P6. |
|---|---|---|---|---|---|
| 39 | 43·5 | 45·6 | 49·3 | 48·8 | 49·2 |

FIG 25(b) SQUARE ROOT OF FUEL INJECTION PRESSURES sensitive bye-pass valve operated by the governor is kept open throughout the injection - the size of the bye-pass opening being dependent on the fuel requirements of the engine. At light load, this opening will be large and, consequently, the pressure takes longer to build up. Measurements of the fuel pump plunger movement at various crank angle positions (shown super-imposed in Fig. 25a) confirms this, the maximum pressure during injection being reached at approximately the same time for all loads. Similarly, this accounts for the lateness of injection at light load.

It is well known that the discharge from a nozzle is proportional to the square root of the pressure difference acting over the nozzle. Fig. 25b shows the square root of the fuel pressure plotted to a crank angle base and the mean square root of the pressure during the injection period has been tabulated on the graph. It will be seen that, although the pressure curves, Fig. 25a, continue to rise with the engine load, the mean square root of the pressure during injection is sensibly constant except in the case of Tests P.1 and P.2. This would seem to be in reasonable agreement with the fuel consumption curve shown in Fig. 24.

areas co[…] have been integrated

Heat/

HEAT BALANCE:

A heat balance is essentially a tabulated form of the general energy equation:-

Heat supplied = External Work done + increase of Internal Energy.

The particular form which the balance takes should depend on the purpose for which it is intended, and it can be said at the outset that the perfect form of heat balance has not yet been found and probably never will be found.

The chief difficulty in forming an exact balance lies in the fact that the engine does not work on a closed cycle. From a thermodynamic point of view and when studying the conversion of fuel heat into work done on the piston of the engine, it would appear best to strike a balance between the closing of the air inlet valve during compression and the opening of the exhaust valve at release.

Thus:-

| Heat supplied by Fuel per Cycle | C.H.U. | % |
|---|---|---|
| Indicated Work per cycle .. | | |
| *Jacket loss in compression up to beginning of combustion .. | | |
| *Jacket loss during combustion and expansion up to release .. | | |
| Exhaust Heat (above Suction Heat) | | |

In/

* The term jacket loss is intended to cover any heat loss which may be radiated from the piston to the outer atmosphere.

In such a balance, the Indicated Work would be measured from the positive loop of the diagram i.e. the gross indicated work per cycle. The jacket losses would be due to conduction and radiation from the cylinder gases and would include no part of the heat generated by piston friction, which is assumed to pass wholly into the jacket.

A similar heat balance was employed by B. Hopkinson[*] in 1908 and has been recently rejuvenated by F.W. Lancaster[×]. Both these experimenters determine Lanchester
the exhaust heat by extrapolating the compression and expansion curves up to the outer dead centre cylinder volume. The temperatures calculated from the pressures attained at this volume gave the exhaust and suction heats, the difference being classified as exhaust heat.

Both Hopkinson and Lancaster then obtained the the jacket loss by difference.

The validity of this method is questionable since it does not take into account the rounding of the toe of the diagram since release is not instantaneous and the jacket is therefore credited with the difference ?
between/

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* B. Hopkinson - "Thermal Efficiency of Gas Engines" Proc.Inst. of Mech. Engineers 1908 (2)

× F.W. Lancaster - "Energy balance sheet for the I.C. Engine" Proc.Inst. of Mech. Engineers 1939 Volume 141 (4) p. 315.

between the ideal "toe" and the actual "toe" of the diagram. This difference should be credited to exhaust. The amount concerned is not normally large though it will increase with the load on the engine.

The heat reception curves plotted in Fig. 22 afford a method of obtaining the jacket loss in compression and expansion, the exhaust heat being obtained as a difference.

Show how.

Table V gives a heat balance for tests P. 1 to P. 6 while the items are shown graphically in Fig. 26.

TABLE V

| TEST NO. | P. 1 | | P. 2 | | P. 3 • | |
|---|---|---|---|---|---|---|
| | C.H.U. per cycle | % | C.H.U. per cycle | % | C.H.U. per cycle | % |
| Fuel | 2.41 | 100 | 4.29 | 100 | 5.74 | 100 |
| Indicated Work (gross) | 1.05 | 43.7 | 1.85 | 43.3 | 2.35 | 40.9 |
| * Jacket Loss (compression) | .58 | 24.0 | .50 | 11.7 | .45 | 7.9 |
| * Jacket Loss (combustion & expansion) | .30 | 12.4 | .66 | 15.4 | 1.10 | 19.2 |
| Exhaust Loss (measured above suction temp. by difference | .48 | 19.9 | 1.28 | 29.6 | 1.84 | 32.0 |

* This term jacket loss is intended to cover any loss of heat which may be radiated from the piston direct to the outer atmosphere.

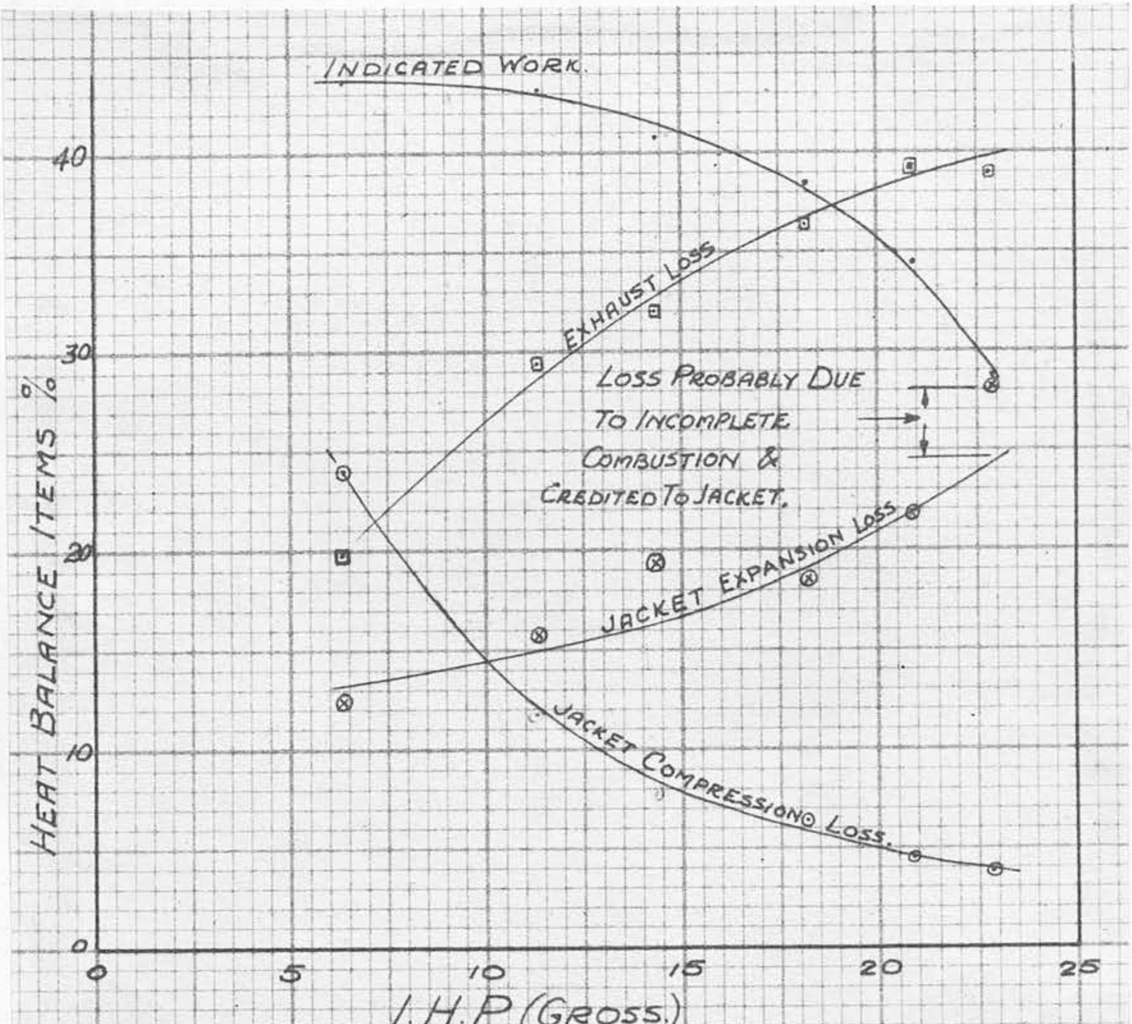



FIG. 26. TESTS P1-P6. HEAT BALANCE.

| TEST NO. | P. 4 | | P. 5 | | P. 6 | |
|---|---|---|---|---|---|---|
| | C.H.U. per cycle | % | C.H.U. per cycle | % | C.H.U. per cycle | % |
| Fuel | 7.75 | 100 | 9.93 | 100 | 12.98 | 100 |
| Indicated Work (gross) | 2.98 | 38.4 | 3.42 | 34.4 | 3.75 | 28.9 |
| Jacket Loss (compression) | .51 | 6.5 | .45 | 4.5 | .50 | 3.9 |
| Jacket Loss (combustion & expansion) | 1.44 | 18.6 | 2.17 | 21.9 | 3.67 | 28.2 |
| Exhaust Loss (measured above suction temp. by difference | 2.82 | 36.5 | 3.89 | 39.2 | 5.06 | 39.0 |

Comments on the Heat Balance.

Jacket Loss.

During compression, the loss decreases slightly with engine load, and, as would be expected, since the cylinder wall temperature will be higher.

The most noticeable feature about the compression loss is that the greater part of it takes place during the last thirty or forty degrees of the stroke. This is the period when the charge is hottest during compression but this fact in itself would seem hardly sufficient to account for the very considerable loss. It seems more likely to be due to the increased turbulence/

turbulence of the air consequent on its compression into the narrow combustion chamber.

The percentage compression loss decreases with load ~~and~~ yet it is noticeable that at no-load the compression loss represents the greatest of the three heat losses.

The loss during combustion and expansion increases more and more rapidly with the engine load and comparison with the heat reception curves, Fig. 22, shows that by far the greater proportion of heat is lost by radiation etc. during actual combustion.

Effect of Incomplete Combustion.

Although the exhaust heat loss is given in the table as obtained by difference, it is actually the jacket loss during expansion that has been obtained as the difference between Heat supplied by fuel per cycle and the sum of the Work done during expansion to release + the Increase of internal energy over the same period.  Consequently, any heat lost by incomplete combustion will be credited to the jacket loss.

This would appear to account for the fall off in exhaust heat noticeable in test P. 6.  An exhaust gas analysis was carried out during this test and, of the two samples taken, only one showed a trace of

CO/

CO - less than 0.5%. It is quite probable, of course, that combustion proceeds in the exhaust pipe and, since the sample was removed about eight feet from the exhaust valve, combustion by then may have been practically complete.

Since there is a continual loss of heat throughout combustion and expansion, it seems likely that combustion does not cease until some ten degrees or more after the maximum registered by the heat reception curves. This effect will particularly apply at full load when that portion of fuel last injected will require to "seek" its oxygen supply i.e. the last portions will burn very slowly.

Reference to the heat reception curves, Fig. 22, shows that the maximum value of heat received by the charge is not arrived at until ten degrees before release. Under these circumstances, incomplete combustion seems to be indicated.

TESTS UNDER VARIABLE INDUCTION CONDITIONS:

These tests, carried out under variable induction conditions, have been grouped together.

Tables VI, VII and VIII give results obtained when operating at different air inlet temperatures, the weight of charge being kept approximately constant. These tables are followed by six pv diagrams (Figs. 27 - 32).

Tables IX, X and XI give results obtained when operating with different air consumptions, the temperature of the inlet air being maintained approximately constant. The tables are followed by six pv diagrams (Figs. 33 - 38).

EFFECT OF AIR INLET TEMPERATURE:

In these tests, the air consumption was kept constant while the temperature was varied between 16.3° C. and 99.7° C. The main purpose of the tests was to examine the combustion process under two different air to fuel ratios, namely approximately 53:1 and approximately 28.5:1, the engine performance being examined with three different inlet air temperatures in each case.

While/ (page 48)

TABLE VI

| Test No. | LIGHT LOAD | | | HALF LOAD | | |
|---|---|---|---|---|---|---|
| | TL1 | TL2 | TL3 | TH1 | TH2 | TH3 |
| Atmospheric Temp. °C. | 17.3 | 17.8 | 18.0 | 16.8 | 18.8 | 16.3 |
| Atmospheric Pressure lb/sq.in. | 14.59 | 14.64 | 14.61 | 14.73 | 14.64 | 14.73 |
| Induction Temperature °C. | 16.3 | 62.8 | 99.7 | 15.0 | 64.2 | 100.2 |
| Air used per min. lb. | 3.21 | 3.13 | 3.095 | 3.11 | 3.08 | 3.15 |
| R.P.M. | 288.0 | 288.0 | 287.0 | 288.2 | 288.0 | 288.1 |
| B.H.P. | 3.81 | 3.78 | 3.63 | 10.8 | 10.37 | 10.18 |
| I.M.E.P.(gross) lb./sq.in. | 35.2 | 36.23 | 37.1 | 58.8 | 57.4 | 57.1 |
| I.M.E.P.(pumping) lb./sq.in. | 6.23 | 5.32 | 4.98 | 5.43 | 5.47 | 5.23 |
| I.M.E.P.(nett) lb./sq.in. | 28.97 | 30.91 | 32.1 | 53.37 | 51.93 | 51.87 |
| I.H.P.(gross) | 10.3 | 10.6 | 10.8 | 17.2 | 16.8 | 16.7 |
| I.H.P.(pumping) | 1.82 | 1.56 | 1.45 | 1.59 | 1.60 | 1.53 |
| I.H.P.(nett) | 8.48 | 9.04 | 9.35 | 15.6 | 15.2 | 15.17 |
| Friction H.P. | 4.67 | 5.26 | 5.72 | 4.8 | 4.8 | 5.0 |
| Mechanical Efficiency % | 44.9 | 41.8 | 38.9 | 69.5 | 68.3 | 67.0 |
| Fuel per hour lb. | 3.565 | 3.499 | 3.562 | 6.415 | 6.548 | 6.516 |
| Fuel/B.H.P./hour lb. | .936 | .926 | .981 | .593 | .631 | .64 |
| Fuel/I.H.P.(gross)/hr. lb. | .346 | .33 | .33 | .373 | .39 | .39 |
| Fuel/I.H.P.(nett)/hr. lb. | .42 | .387 | .381 | .411 | .43 | .429 |
| B.T.E. | 15.0 | 14.8 | 14.27 | 23.6 | 22.2 | 21.9 |
| I.T.E. on gross I.H.P. | 40.4 | 42.4 | 42.4 | 37.6 | 35.9 | 35.9 |
| I.T.E. on nett I.H.P. | 33.3 | 36.2 | 36.8 | 34.1 | 32.6 | 32.7 |
| Exhaust Temperature °C. | 196 | 208 | 221 | 344 | 352 | 356 |
| Jacket Water inlet temp. °C. | 29.4 | 27.2 | 30.2 | 29.7 | 26.1 | 22.7 |
| Jacket Water outlet temp. °C. | 50.9 | 47.3 | 52.2 | 54.7 | 56.4 | 53.2 |
| Jacket Water quantity/min. lb. | 11.52 | 12.91 | 12.3 | 14.05 | 13.2 | 14.9 |
| Heat to Jacket/min. C.H.U. | 248 | 260.8 | 270.8 | 350.5 | 400 | 454 |
| Fuel Temp. at injector °C. | 45 | 44 | 45 | 46 | 52 | 54 |

TABLE VII.

| Test No. | LIGHT LOAD | | | HALF LOAD | | |
|---|---|---|---|---|---|---|
| | TL1 | TL2 | TL3 | TH1 | TH2 | TH3 |
| I.H.P. (gross) | 10.3 | 10.6 | 10.8 | 17.2 | 16.8 | 16.7 |
| I.H.P. (nett) | 8.48 | 9.04 | 9.35 | 15.6 | 15.2 | 15.17 |
| Air/Fuel Weight Ratio | 54 | 53.7 | 52.2 | 29.1 | 28.2 | 29.0 |
| Excess Air % | 275 | 273 | 263 | 102 | 96 | 101 |
| Air used per cycle lb. | .0223 | .0216 | .02156 | .02158 | .02137 | .0219 |
| Residuals lb. | .0020 | .00196 | .00181 | .00153 | .00147 | .00142 |
| Weight of Charge during Compression - lb. | .0243 | .02356 | .02337 | .02311 | .02284 | .02332 |
| Fuel used per 1000 cycles lb. | .413 | .404 | .413 | .742 | .758 | .753 |
| Weight of Charge during Expansion - lb. | .02471 | .02396 | .02378 | .02385 | .02360 | .02407 |
| Injection begins - degrees before I.D.C. | 8.3 | 9.3 | 7.8 | 11.0 | 11.6 | 10.8 |
| Injection ends - degrees after I.D.C. | 19.0 | 19.6 | 21.0 | 40.0 | 40.2 | 40.3 |
| Period of Injection - degrees | 27.3 | 28.9 | 28.8 | 51.0 | 51.8 | 51.1 |
| Combustion begins (approx.) degrees before I.D.C. | 4.2 | 4.5 | 4.8 | 5.0 | 6.5 | 7.6 |
| Delay Period (approx.) degs. | 4.1 | 4.8 | 3.0 | 6.0 | 5.1 | 3.2 |

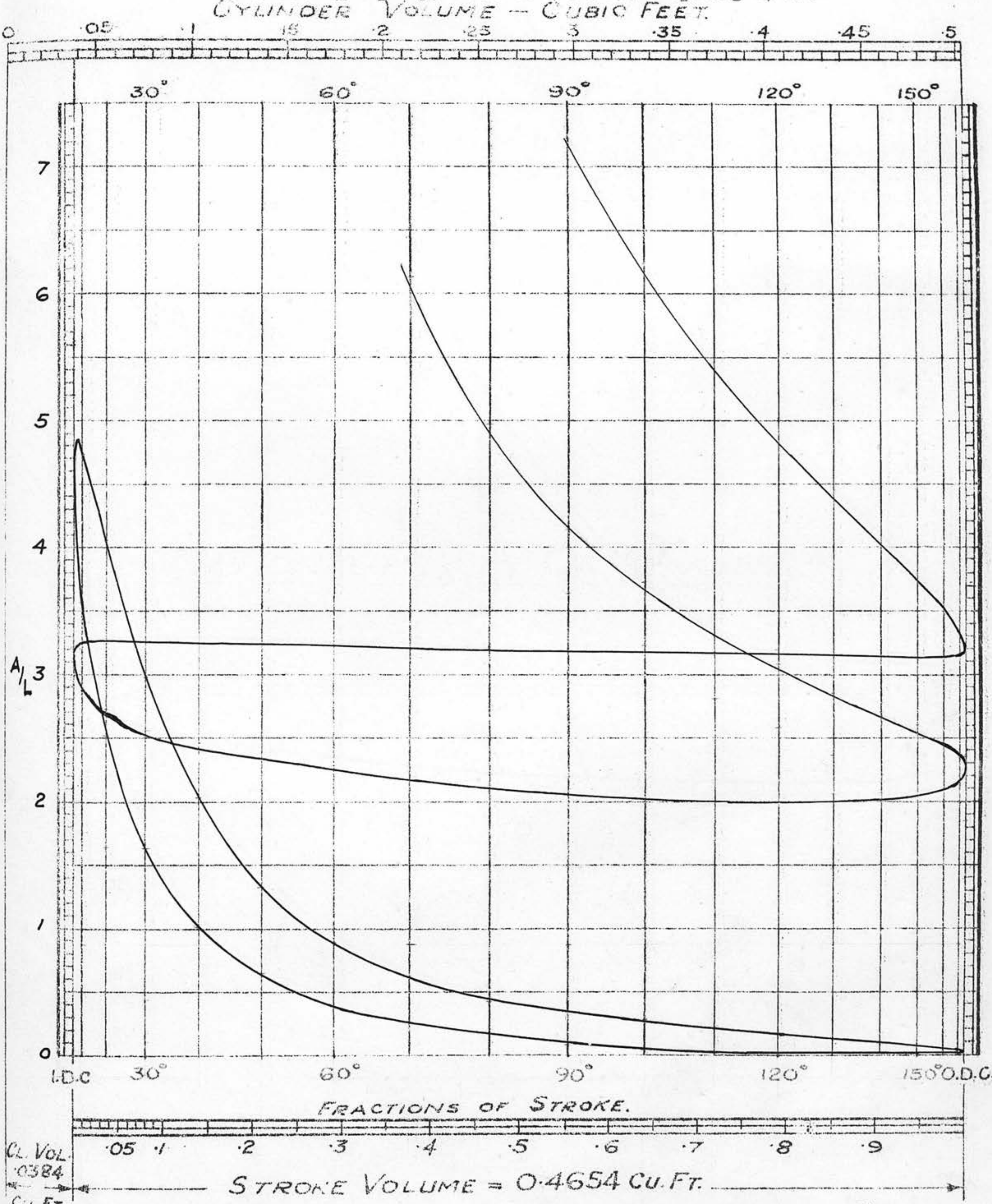
TABLE VIII

| TEST NO. | LIGHT LOAD | | | | | |
|---|---|---|---|---|---|---|
| | TL. 1 | | TL. 2 | | TL. 3 | |
| | C.H.U. per cycle | % | C.H.U. per cycle | % | C.H.U. per cycle | % |
| Fuel | 4.18 | 100 | 4.08 | 100 | 4.18 | 100 |
| Indicated Work (gross) | 1.69 | 40.4 | 1.73 | 42.4 | 1.77 | 42.4 |
| Jacket Loss (compression) | .47 | 11.2 | .54 | 13.3 | .58 | 13.9 |
| Jacket Loss (combustion and expansion) | .65 | 15.6 | .58 | 14.2 | .50 | 12.0 |
| Exhaust Loss measured above suction temp. (by difference) | 1.37 | 32.8 | 1.23 | 30.1 | 1.33 | 31.7 |

| TEST NO. | HALF LOAD | | | | | |
|---|---|---|---|---|---|---|
| | TH. 1 | | TH. 2 | | TH. 3 | |
| | C.H.U. per cycle | % | C.H.U. per cycle | % | C.H.U. per cycle | % |
| Fuel | 7.49 | 100 | 7.66 | 100 | 7.6 | 100 |
| Indicated Work (gross) | 2.82 | 37.6 | 2.75 | 35.9 | 2.73 | 35.9 |
| Jacket Loss (compression) | .41 | 5.5 | .51 | 6.7 | .67 | 8.8 |
| Jacket Loss (combustion and expansion) | 1.71 | 22.8 | 1.87 | 24.0 | 1.83 | 24.1 |
| Exhaust Loss measured above suction temp. (by difference) | 2.55 | 34.0 | 2.53 | 33.0 | 2.36 | 31.1 |

# "NATIONAL" HEAVY OIL ENGINE. FIG. 27.

TEST TL. 1. I.M.E.P. (gross) 35.2 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 6.23 lb./sq.in. Spring 1/6.9



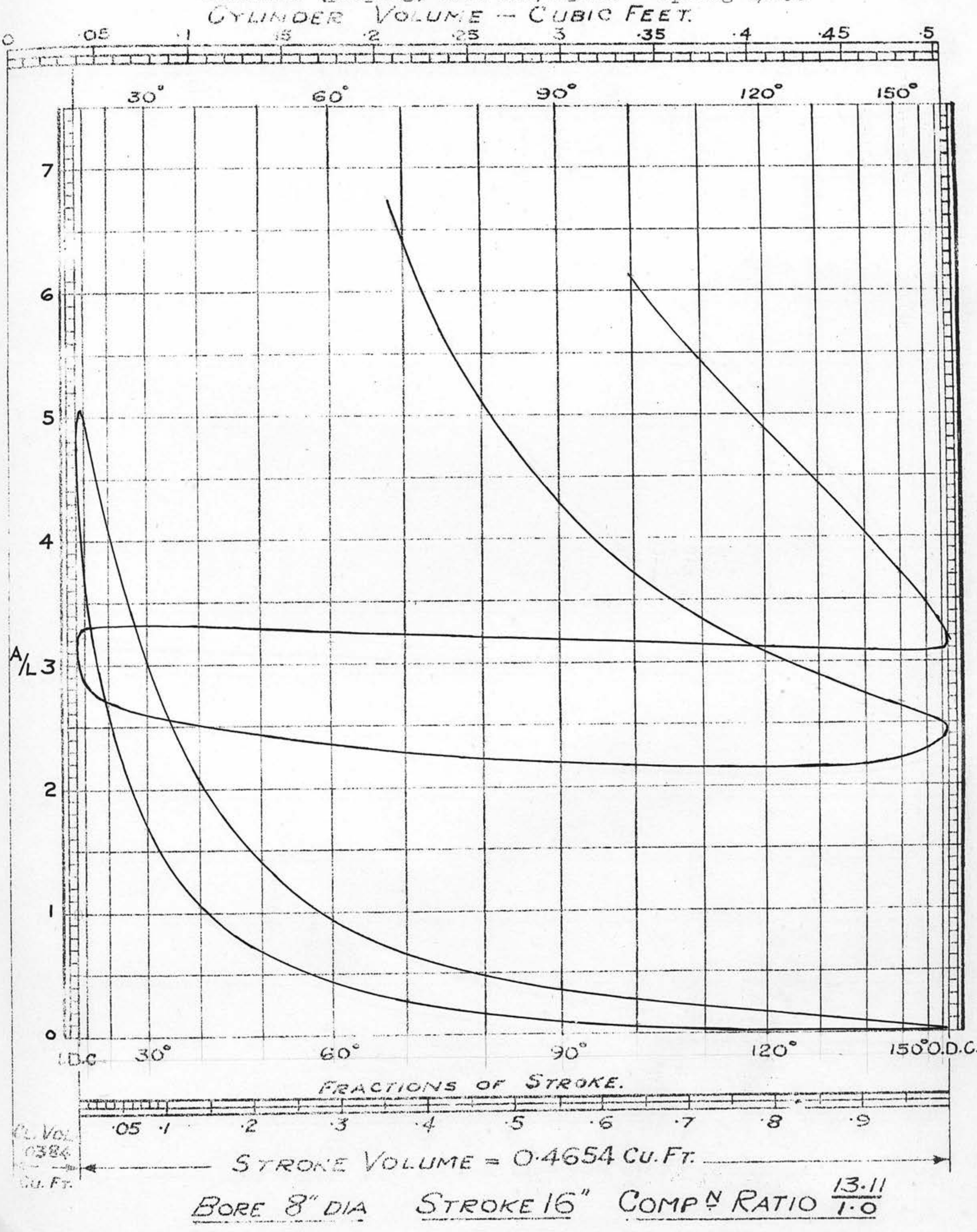

BORE 8" DIA STROKE 16" COMPN RATIO $\frac{13{\cdot}11}{1{\cdot}0}$

7.08

# "NATIONAL" HEAVY OIL ENGINE.

FIG. 28

TEST TL. 2. I.M.E.P. (gross) 36.23 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 5.32 lb./sq.in. Spring 1/6.9

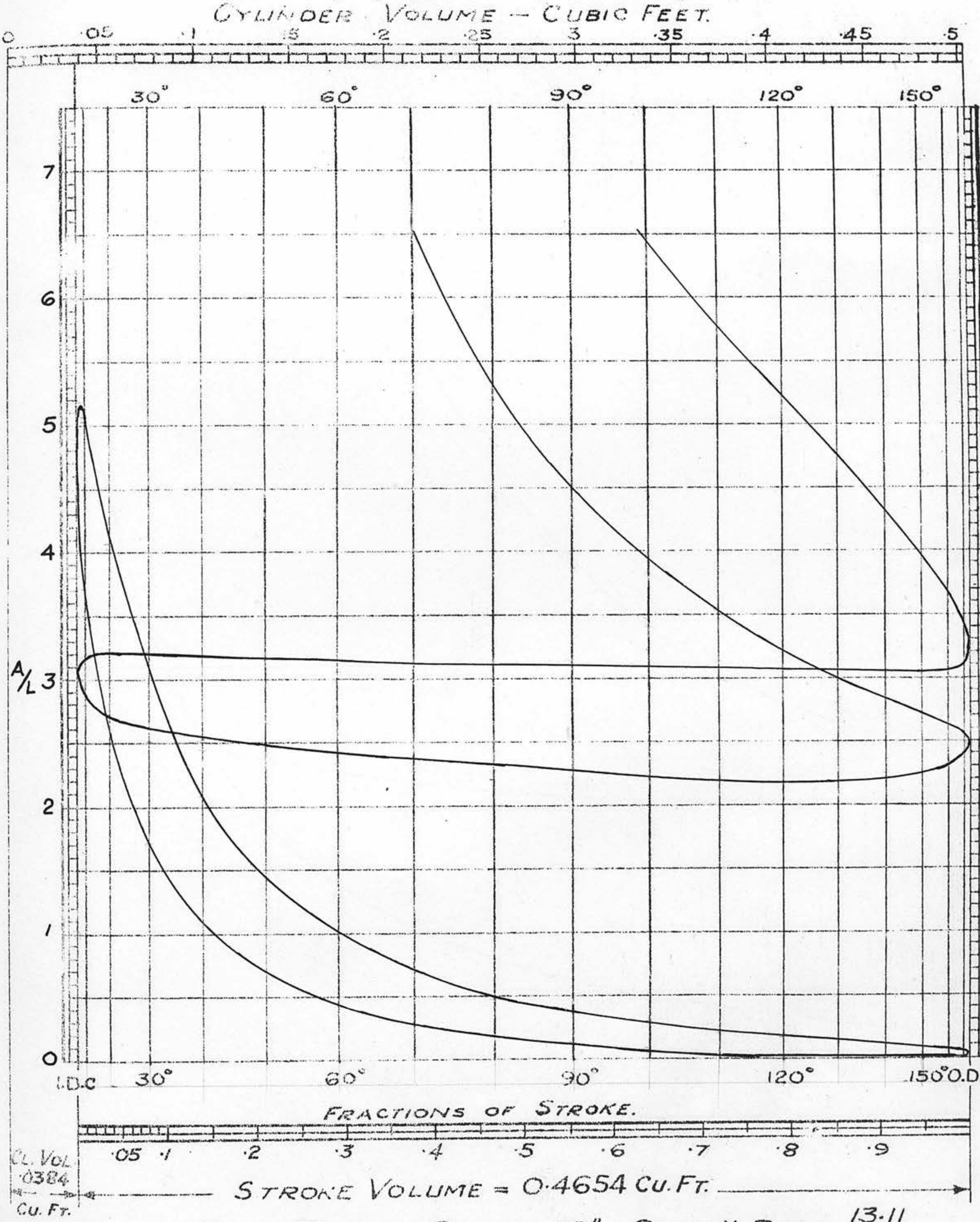



BORE 8" DIA STROKE 16" COMP^N RATIO $\frac{13{\cdot}11}{1{\cdot}0}$

# "NATIONAL" HEAVY OIL ENGINE.

FIG. 29.

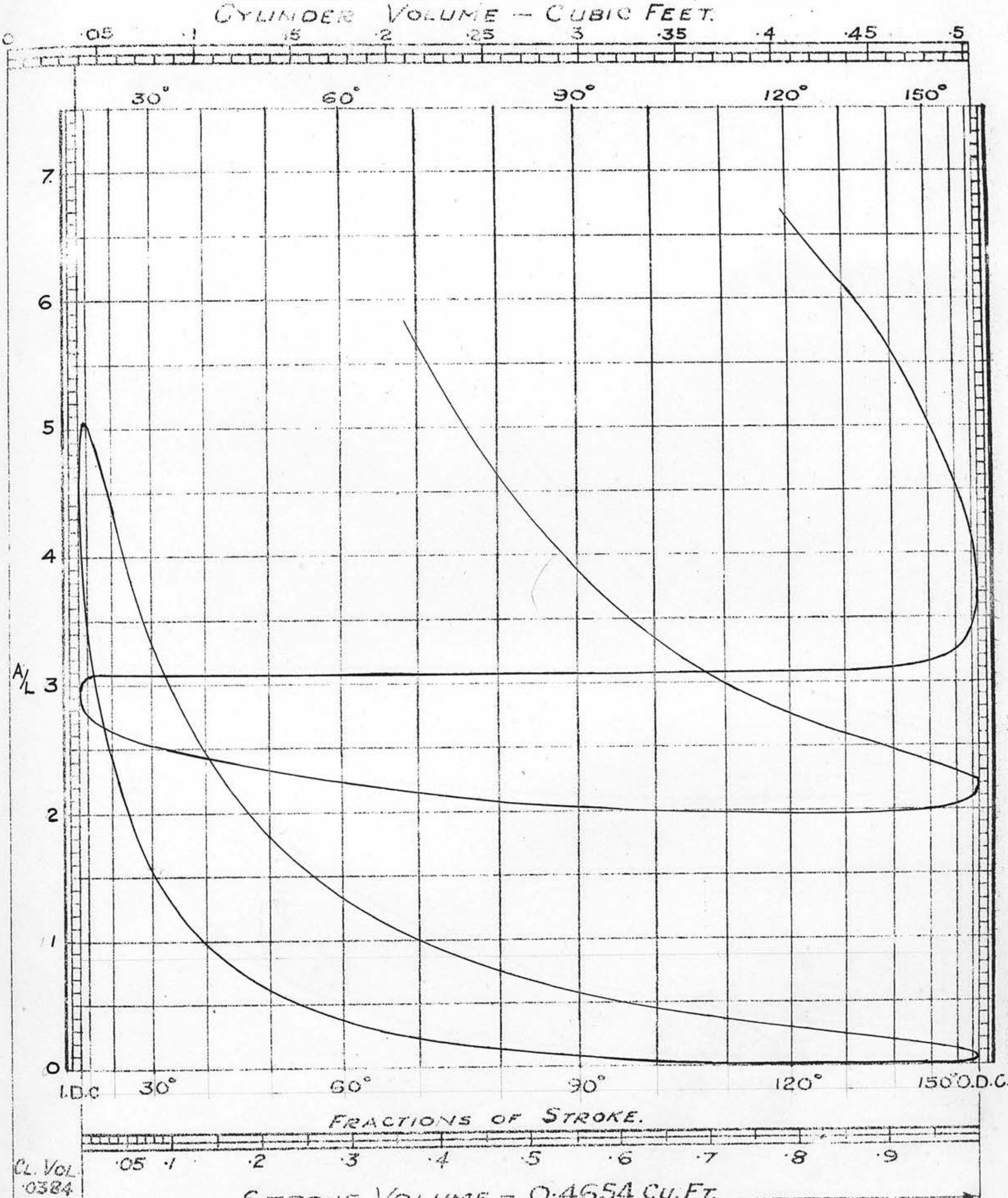
TEST TL.3. I.M.E.P. (gross) 37.1 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 4.98 lb./sq.in. Spring 1/6.9




BORE 8" DIA STROKE 16" COMP^N RATIO $\frac{13.11}{1.0}$

"NATIONAL" HEAVY OIL ENGINE. FIG. 30.

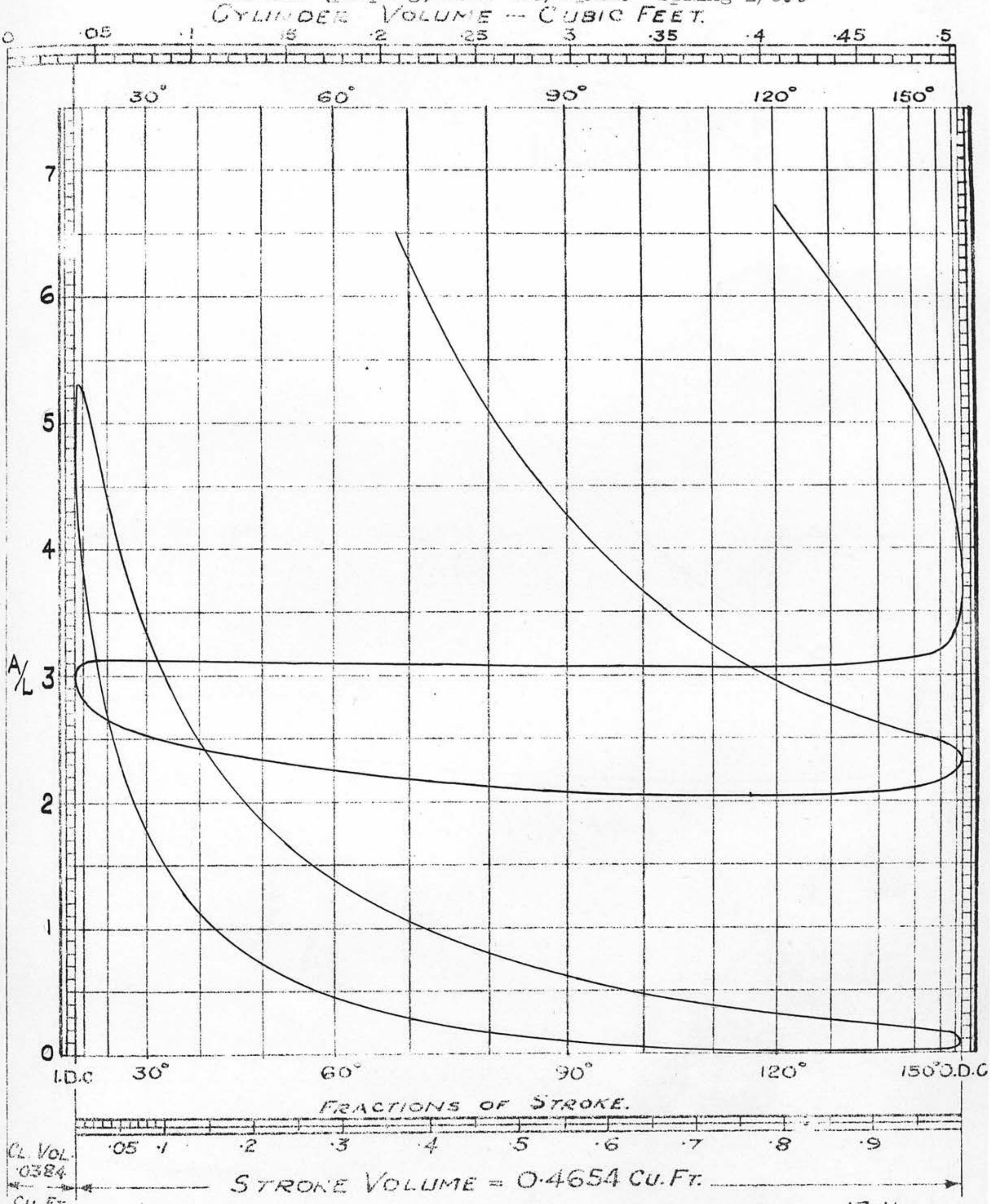
TEST TH.1. I.M.E.P. (gross) 58.8. lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 5.43 lb./sq.in. Spring 1/6.9




BORE 8" DIA STROKE 16" COMPN RATIO $\frac{13.11}{1.0}$

"NATIONAL" HEAVY OIL ENGINE. FIG. 31.

TEST TH. 2. I.M.E.P. (gross) 57.4 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 5.47 lb./sq.in. Spring 1/6.9

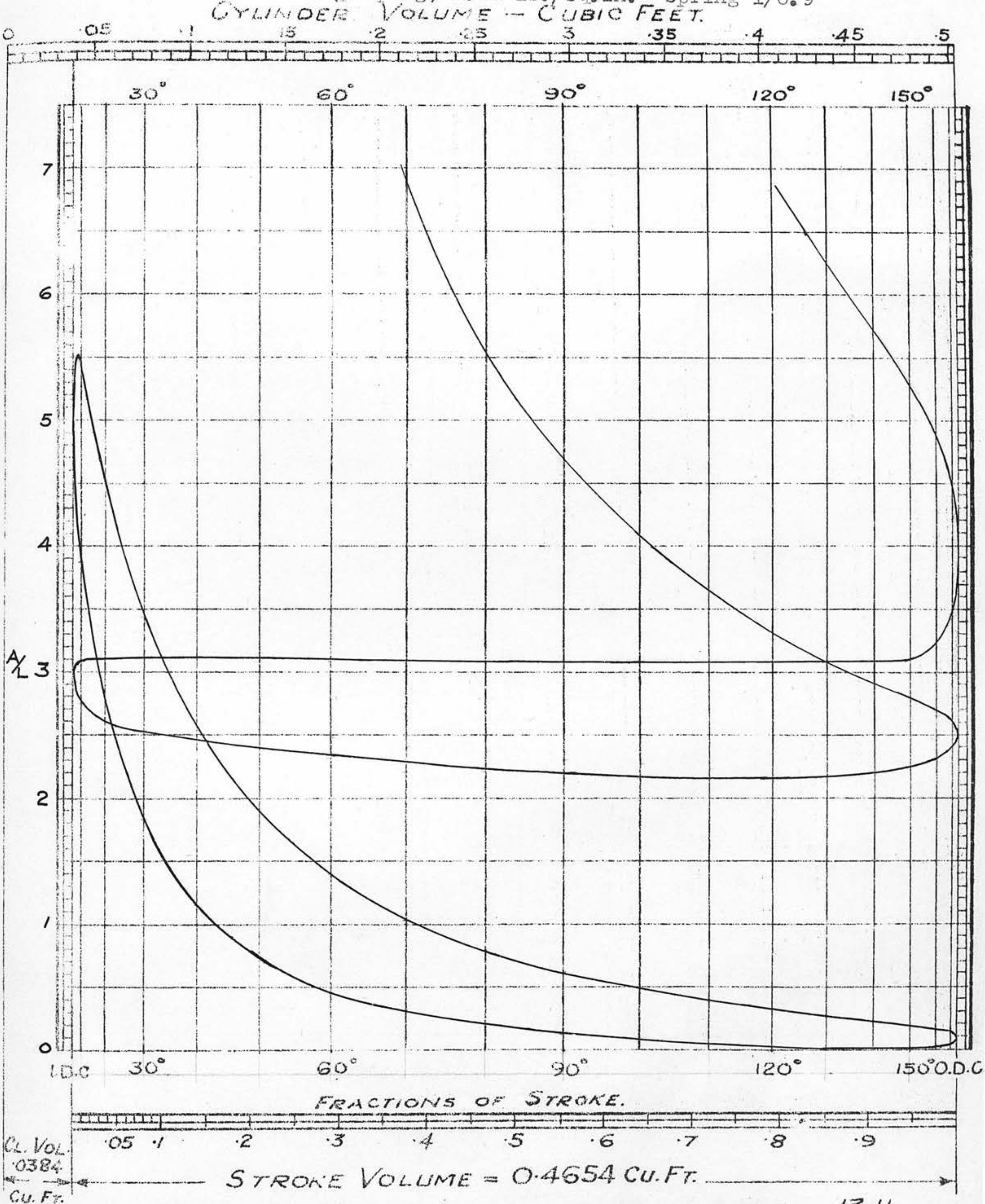



BORE 8" DIA STROKE 16" COMPN RATIO $\frac{13.11}{1.0}$

# "NATIONAL" HEAVY OIL ENGINE.

FIG. 32.

TEST TH. 3. I.M.E.P. (gross) 57.1 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 5.23 lb./sq.in. Spring 1/6.9




BORE 8" DIA STROKE 16" COMP$^{\text{N}}$ RATIO $\frac{13{\cdot}11}{1{\cdot}0}$

TABLE IX

| Test No. | WL 1 | WL 2 | WL 3 | WL 4 | WL 5 | WL 6 |
|---|---|---|---|---|---|---|
| Atmospheric Temp. $^{o}$ C. | 17.1 | 17.0 | 17.9 | 17.3 | 19.0 | 22.0 |
| Atmospheric Press. lb/sq.in. | 14.52 | 14.59 | 14.57 | 14.59 | 14.51 | 14.59 |
| Induction Temp. $^{o}$ C. | 17.1 | 18.2 | 18.0 | 16.3 | 17.9 | 21.6 |
| Air used per min. lb. | 4.37 | 3.96 | 3.425 | 3.21 | 2.552 | 21.6 |
| Volumetric Efficiency % | 87.9 | 78.8 | 68.2 | 63.6 | 51.2 | 43.3 |
| R.P.M. | 288.6 | 287.5 | 287.6 | 288 | 288 | 287.3 |
| B.H.P. | 4.98 | 4.63 | 4.405 | 3.81 | 3.823 | 3.26 |
| I.M.E.P.(gross) lb./sq.in. | 38.7 | 37.66 | 36.9 | 35.2 | 34.76 | 34.46 |
| I.M.E.P.(pumping) lb./sq.in. | 4.22 | 5.86 | 6.2 | 6.23 | 6.33 | 6.91 |
| I.M.E.P.(nett) lb./sq.in. | 34.5 | 31.8 | 32.7 | 28.97 | 28.43 | 27.55 |
| I.H.P.(gross) | 11.36 | 11.00 | 10.78 | 10.3 | 10.18 | 10.06 |
| I.H.P.(pumping) | 1.23 | 1.71 | 1.81 | 1.82 | 1.85 | 2.02 |
| I.H.P.(nett) | 10.13 | 9.29 | 8.97 | 8.48 | 8.32 | 8.03 |
| Friction H.P. | 5.15 | 4.66 | 4.565 | 4.67 | 4.5 | 4.77 |
| Mechanical Efficiency % | 49.3 | 49.8 | 49.2 | 44.9 | 46.0 | 40.5 |
| Fuel/hour lb. | 3.68 | 3.58 | 3.57 | 3.565 | 3.54 | 3.72 |
| Fuel/B.H.P./hr. lb. | .738 | .773 | .798 | .936 | .927 | .1142 |
| Fuel/I.H.P.(gross)/hr. lb. | .324 | .325 | .326 | .346 | .348 | .37 |
| Fuel/I.H.P.(nett)/hr. lb. | .364 | .385 | .393 | .42 | .427 | .463 |
| B.T.E. | 19.0 | 18.1 | 17.55 | 15.0 | 15.1 | 12.3 |
| I.T.E. on gross I.H.P. | 43.3 | 43.2 | 43.0 | 40.4 | 40.2 | 37.9 |
| I.T.E. on nett I.H.P. | 38.6 | 36.4 | 35.7 | 33.3 | 32.8 | 30.2 |
| Exhaust Temperature $^{o}$ C. | 165 | 177 | 194 | 196 | 231 | 287 |
| Jacket inlet Temp. $^{o}$ C. | 32.7 | 31.1 | 30.8 | 29.4 | 31.1 | 30.0 |
| Jacket outlet Temp. $^{o}$ C. | 49.7 | 50.5 | 51.6 | 50.9 | 51.3 | 54.1 |
| Jacket quantity/min. lb. | 11.62 | 10.79 | 10.77 | 11.52 | 12.6 | 12.27 |
| Heat to Jacket/min. C.H.U. | 198 | 209 | 224 | 248 | 255 | 295 |
| Fuel Temp. at injector $^{o}$ C. | 39.5 | 43.0 | 42.0 | 45.0 | 46.0 | 51.0 |

TABLE X

| TEST NO. | WL 1 | WL 2 | WL 3 | WL 4 | WL 5 | WL 6 |
|---|---|---|---|---|---|---|
| I.H.P. (gross) | 11.36 | 11.00 | 10.78 | 10.3 | 10.18 | 10.06 |
| I.H.P. (nett) | 10.13 | 9.29 | 8.97 | 8.48 | 8.32 | 8.03 |
| Air/Fuel Weight Ratio | 72.3 | 66.4 | 58.3 | 54.0 | 43.1 | 34.8 |
| Fuel/Air Weight Ratio | .0138 | .0150 | .01715 | .0185 | .0232 | .0288 |
| Excess Air % | 402 | 361 | 305 | 275 | 200 | 141 |
| Air used per cycle lb. | .0304 | .02756 | .02382 | .0223 | .0176 | .01503 |
| Weight of Residuals lb. | .0325 | .0297 | .02585 | .0243 | .01949 | .01675 |
| Fuel used per 1000 cycles lb. | .418 | .414 | .408 | .413 | .408 | .430 |
| Weight of Charge during expansion lb. | .03292 | .03011 | .02625 | .02471 | .0199 | .01718 |
| Injection begins - degrees before I.D.C. | 8.2 | 9.0 | 8.5 | 8.3 | 9.5 | 8.5 |
| Injection ends - degrees after I.D.C. | 21.5 | 21.5 | 22.0 | 19.0 | 20.0 | 23.0 |
| Period of injection - degrees | 29.7 | 30.5 | 30.5 | 27.3 | 29.5 | 31.5 |
| Combustion begins (approx.) degrees before I.D.C. | 4.0 | 4.0 | 3.0 | 3.0 | 2.0 | 0.5 |
| Delay Period (aaprox.) degs. | 5.2 | 5.0 | 5.5 | 5.3 | 7.5 | 8.0 |

TABLE XI

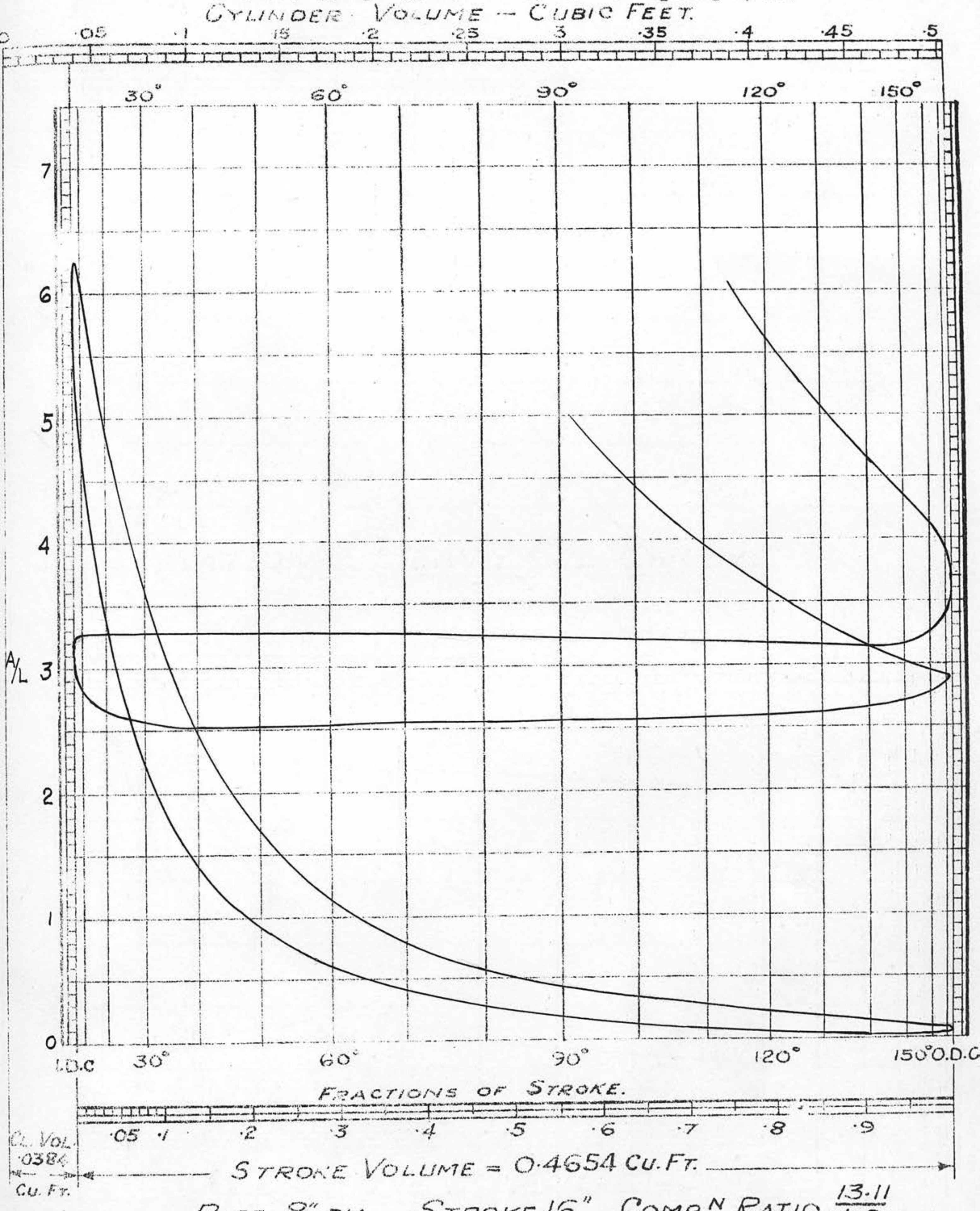
| TEST NO. | WL. 1 | | WL. 2 | | WL. 3 | |
|---|---|---|---|---|---|---|
| | C.H.U. per cycle | % | C.H.U. per cycle | % | C.H.U. per cycle | % |
| Fuel | 4.29 | 100 | 4.18 | 100 | 4.12 | 100 |
| Indicated Work (gross) | 1.85 | 43.3 | 1.80 | 43.2 | 1.77 | 43.0 |
| Jacket Loss (compression) | .5 | 11.7 | .44 | 10.5 | .46 | 11.2 |
| Jacket Loss (combustion and expansion) | .66 | 15.4 | .58 | 13.9 | .57 | 13.8 |
| Exhaust Loss measured above suction temp. (by difference) | 1.28 | 29.6 | 1.36 | 32.4 | 1.32 | 32.0 |

| TEST NO. | WL. 4 | | WL. 5 | | WL. 6 | |
|---|---|---|---|---|---|---|
| | C.H.U. per cycle | % | C.H.U. per cycle | % | C.H.U. per cycle | % |
| Fuel | 4.18 | 100 | 4.12 | 100 | 4.34 | 100 |
| Indicated Work (gross) | 1.69 | 40.4 | 1.66 | 40.2 | 1.65 | 37.9 |
| Jacket Loss (compression) | .47 | 11.2 | .43 | 10.4 | .51 | 11.7 |
| Jacket Loss (combustion and expansion) | .65 | 15.6 | .64 | 15.5 | .73 | 16.8 |
| Exhaust Loss measured above suction temp. (by difference) | 1.37 | 32.8 | 1.39 | 33.8 | 1.45 | 33.4 |

"NATIONAL" HEAVY OIL ENGINE. FIG. 33.

TEST WL.1. I.M.E.P. (gross) 38.7 lb./sq.in. Spring 1/80

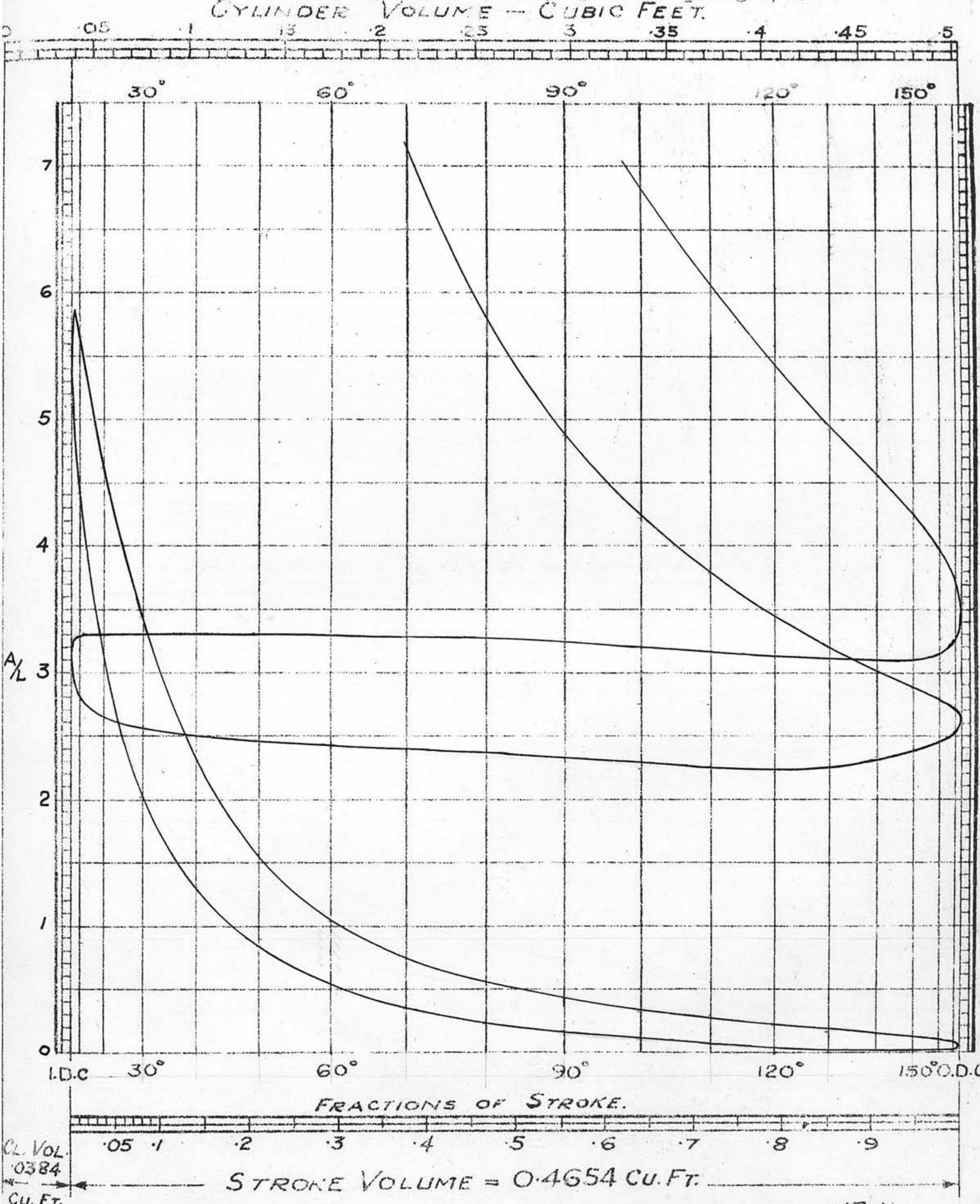
I.M.E.P. (pumping) 4.32 lb./sq.in. Spring 1/6.9




BORE 8" DIA STROKE 16" COMPN RATIO $\frac{13 \cdot 11}{1 \cdot 0}$

# "NATIONAL" HEAVY OIL ENGINE. FIG. 34.

TEST WL. 2. I.M.E.P. (gross) 37.66 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 5.86 lb./sq.in. Spring 1/6.9

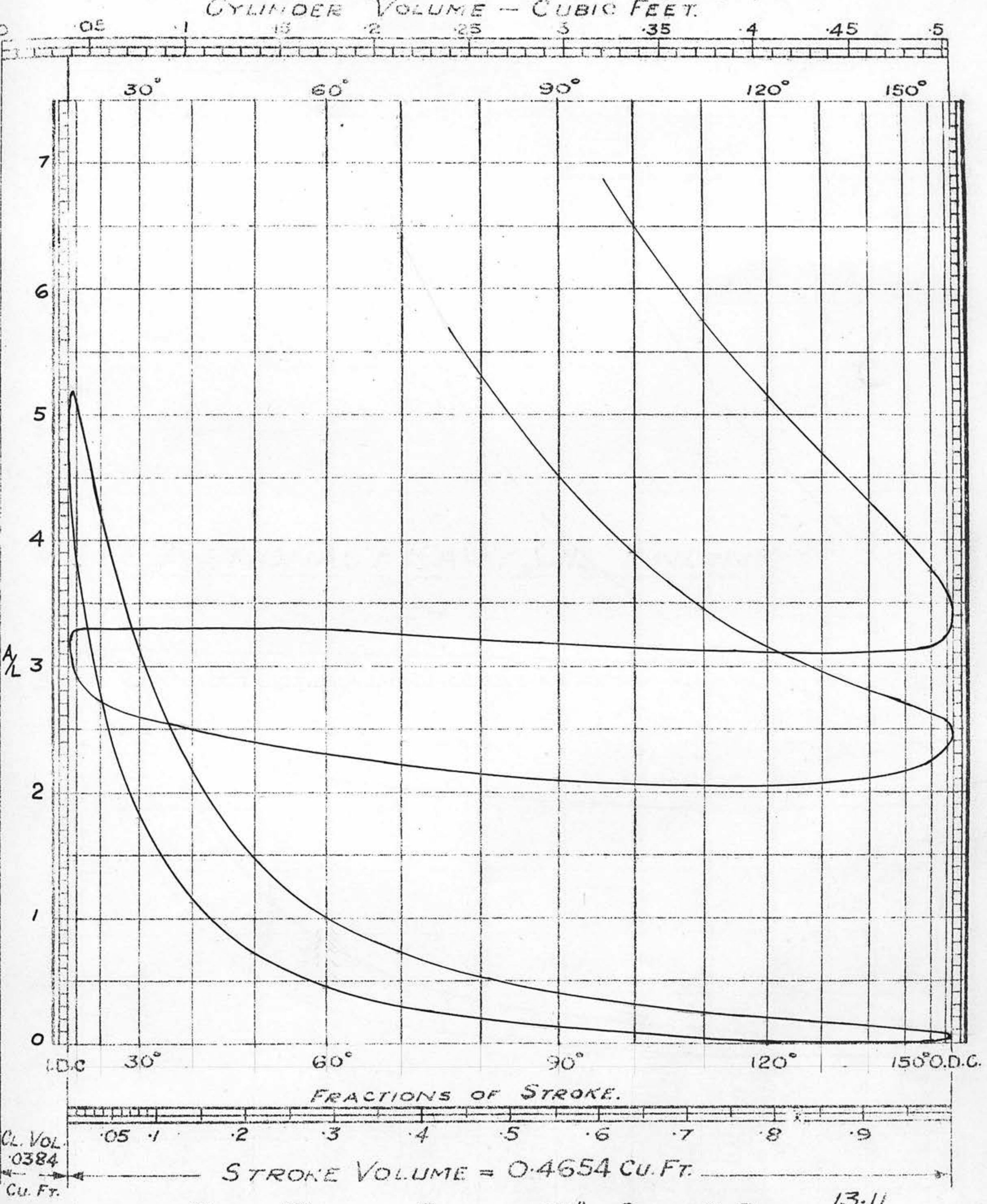



BORE 8" DIA STROKE 16" COMPN RATIO $\frac{13.11}{1.0}$

# NATIONAL HEAVY OIL ENGINE.

FIG. 35.

TEST WL.3. I.M.E.P. (gross) 36.9 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 6.2 lb./sq.in. Spring 1/6.9

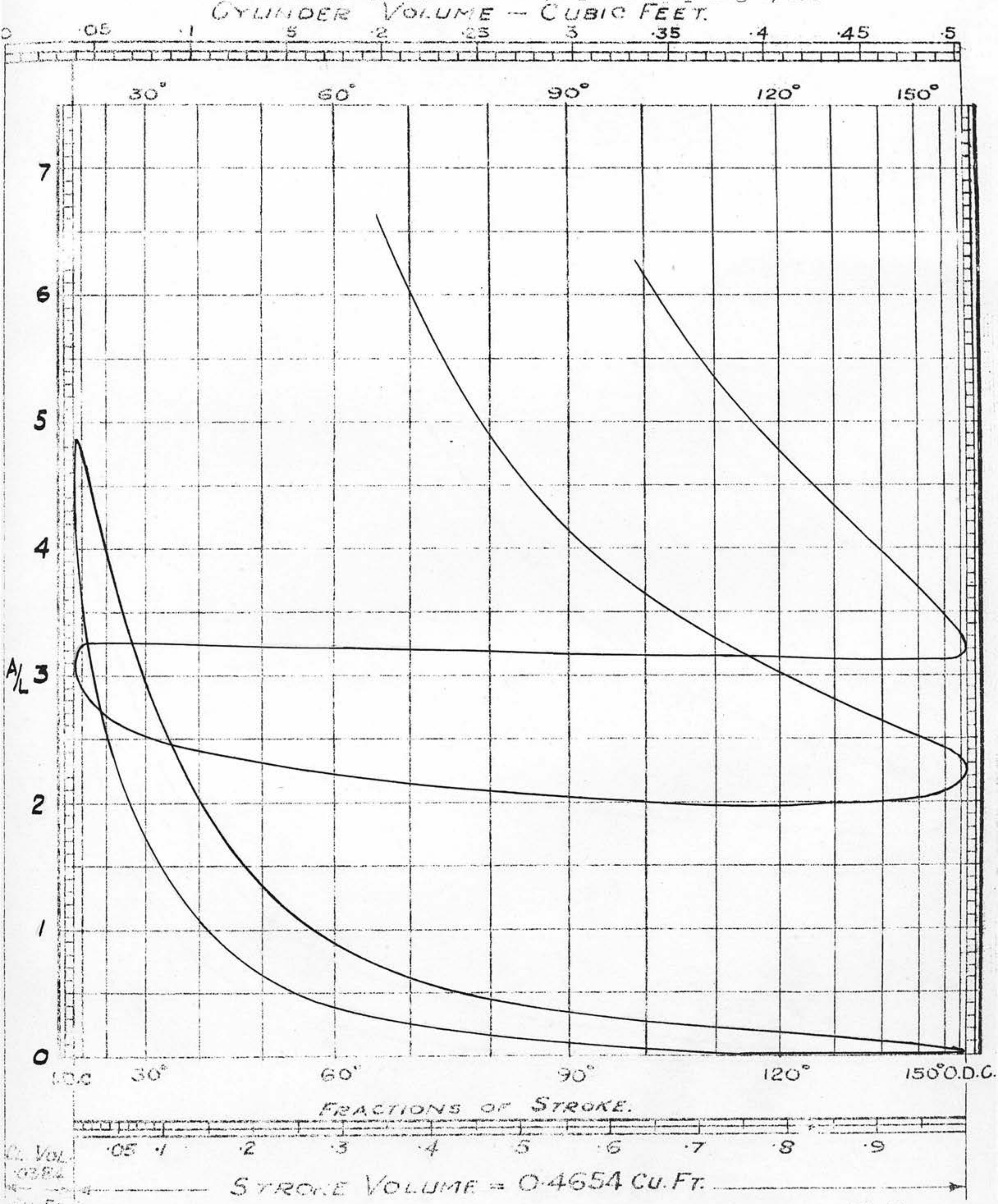



BORE 8" DIA STROKE 16" COMP^N RATIO $\frac{13{\cdot}11}{1{\cdot}0}$

# "NATIONAL" HEAVY OIL ENGINE.

FIG. 36.

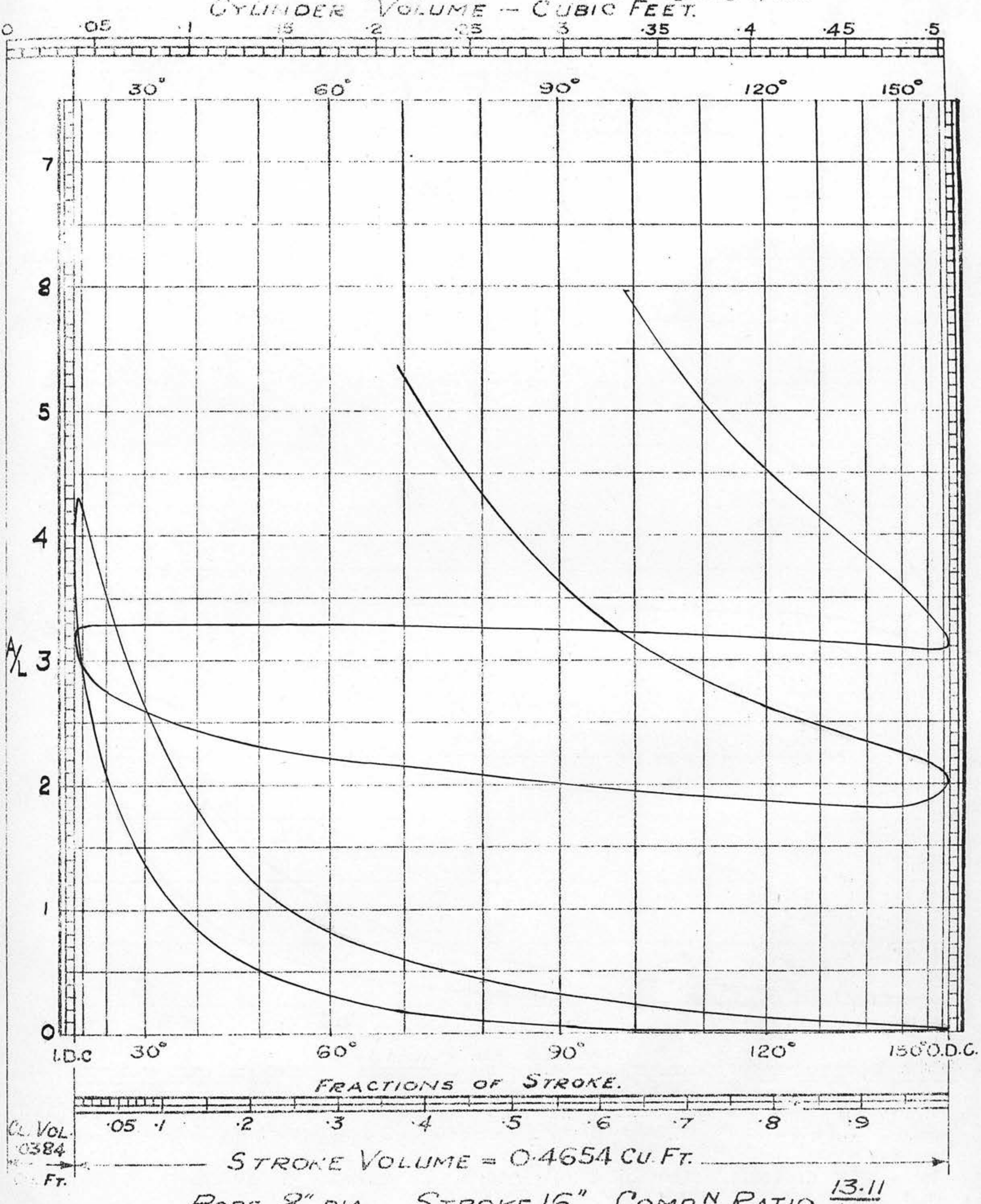
TEST WL.4. I.M.E.P. (gross) 35.2 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 6.23 lb./sq.in. Spring 1/6.9




BORE 8" DIA  STROKE 16"  COMPⁿ RATIO $\frac{13{\cdot}11}{1{\cdot}0}$

# "NATIONAL" HEAVY OIL ENGINE. FIG. 37.

TEST WL. 5. I.M.E.P. (gross) 34.76 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 6.33 lb./sq.in. Spring 1/6.9

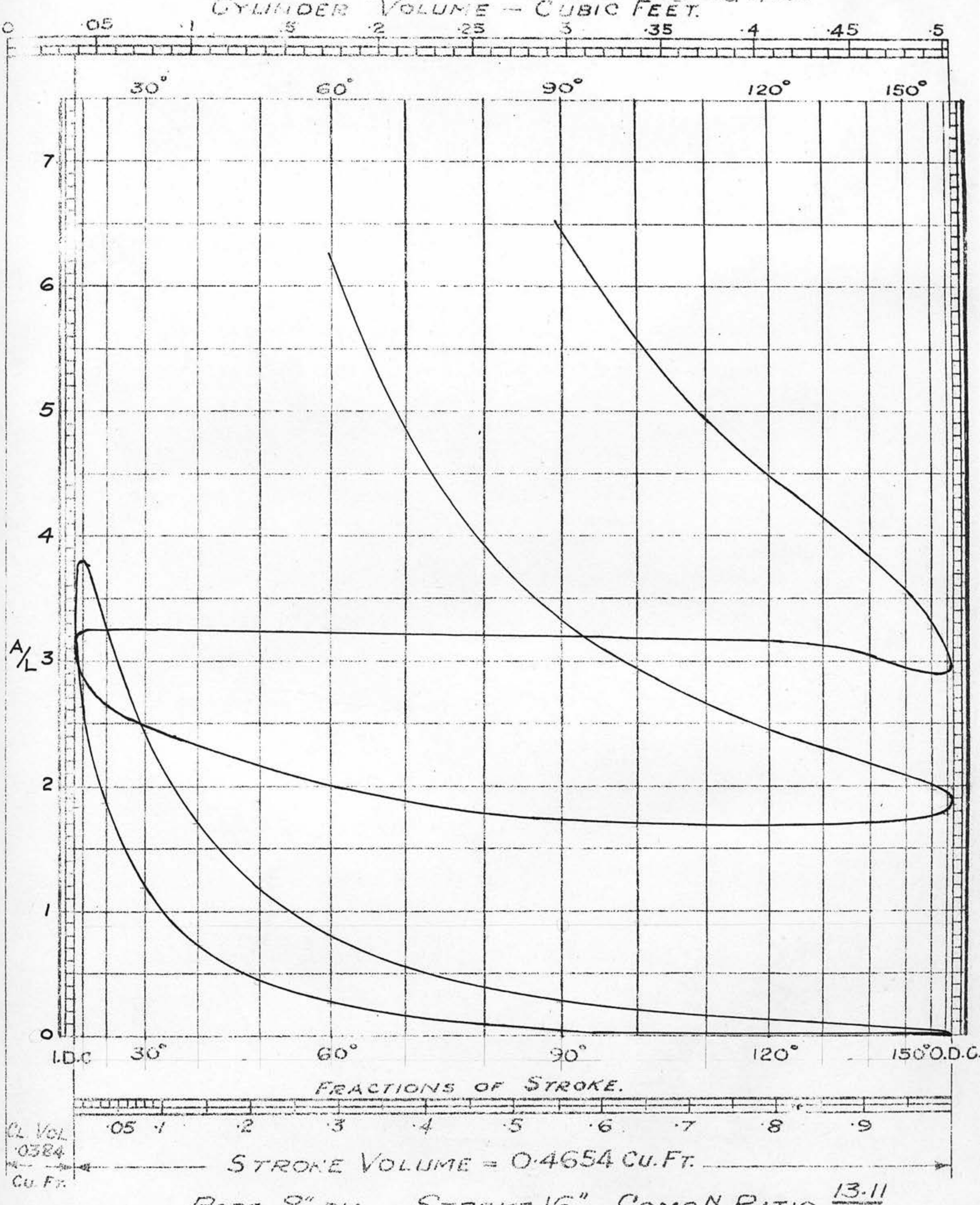



BORE 8" DIA STROKE 16" COMPN RATIO $\frac{13.11}{1.0}$

# NATIONAL HEAVY OIL ENGINE.

FIG. 38.

TEST WL.6. I.M.E.P. (gross) 34.46 lb./sq.in. Spring 1/80
I.M.E.P. (pumping) 6.91 lb./sq.in. Spring 1/6.9




BORE 8" DIA STROKE 16" COMPN RATIO $\frac{13 \cdot 11}{1 \cdot 0}$

While it must be admitted that insufficient tests were carried out to give an accurate picture of the effect of temperature variations, certain trends can be seen from the tables.

Under light load conditions, (Tests TL1 - TL3), a considerable increase in the friction horse-power was noticed as the inlet air temperature was increased. This friction horse-power includes bearing friction, piston friction, and the power to operate the valves and pumps. Of these, the piston friction is the only one likely to be affected by induction temperature. It is generally assumed that any increase in cylinder temperature results in a <u>reduction</u> of piston friction. It is doubtful whether film lubrication is maintained and "boundary" friction is more likely. Under these conditions, it may be that the substitution of a hot dry atmosphere during suction in place of the cool atmosphere obtained under normal induction conditions has resulted in less efficient lubrication. The increase in air temperature was accompanied by less throttling (in order to keep the air consumption constant). The increase in cylinder pressure brought about by this may also have been a contributory cause for the increased friction. The fact that the/

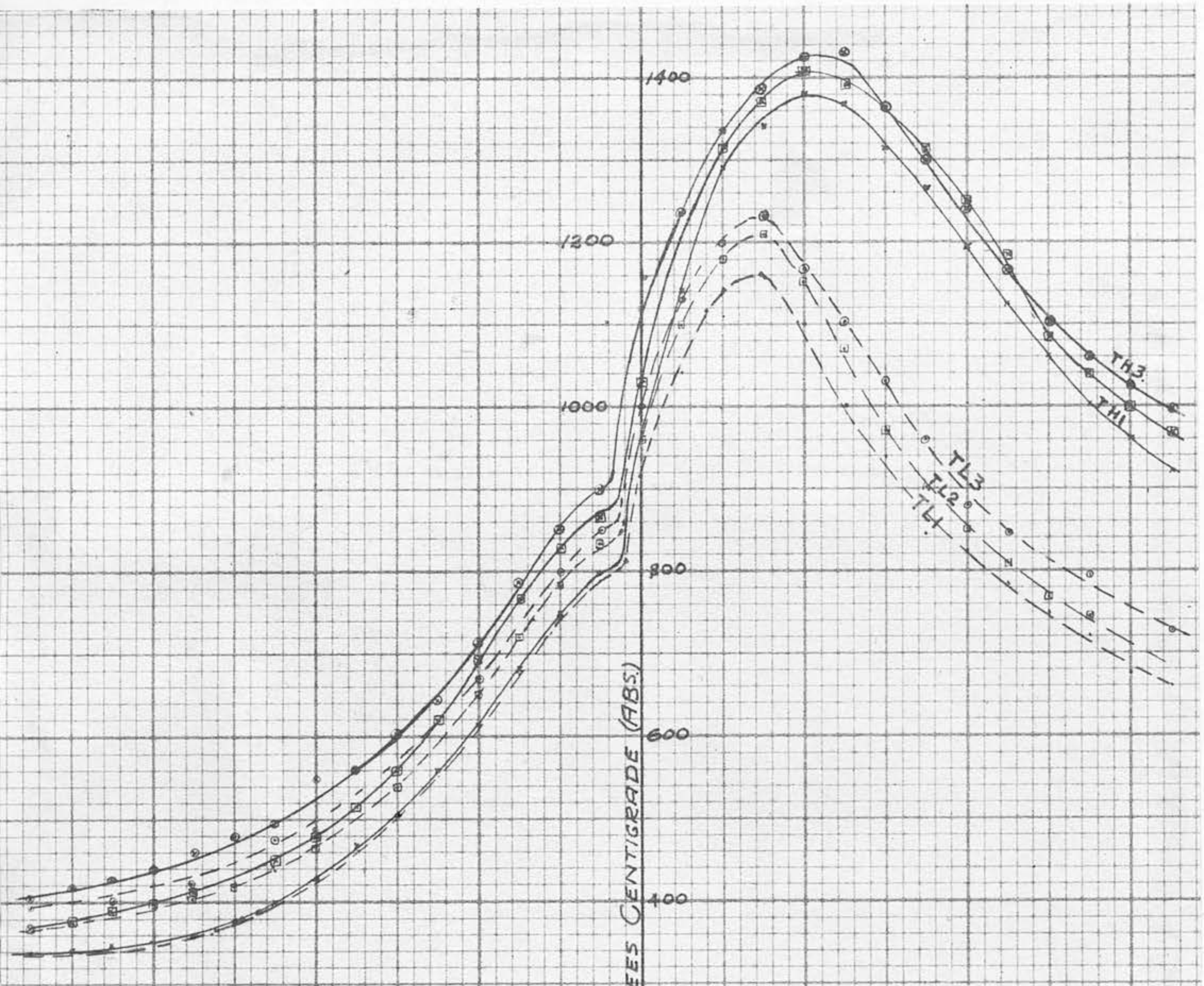



FIG 39. TESTS TL & TH. CYCLICAL CHANGE OF TEMPERATURE.

the increase in friction is maintained as the temperature increases seems to rule out any possibility of serious error in the observations and, in any case, the diagrams and other data taken appeared consistent. It is noticeable, however, that the increase obtained in the half load tests (TH1 - TH3) is much smaller.

The thermal efficiency does not appear to be greatly affected. Under light load, increase of Indicated Thermal Efficiency appears to have resulted from the temperature increase, at least for moderate increase, and this may be due to more efficient combustion due to higher temperatures existing during the combustion process. At half load, this will be offset by the greater heat loss to the jacket (see Heat Balance, Table VIII, page 44).

Fig. 39 shows the temperature changes during compression and expansion. The increase of air temperature above 64 °C. (tests TL3 and TH3) appears to have little affect on the maximum temperatures reached during expansion and this would be due to increased jacket loss during compression.

Fig. 40 shows the heat reception curves for these tests. Reference to this and to Table VI, page 42, indicate/

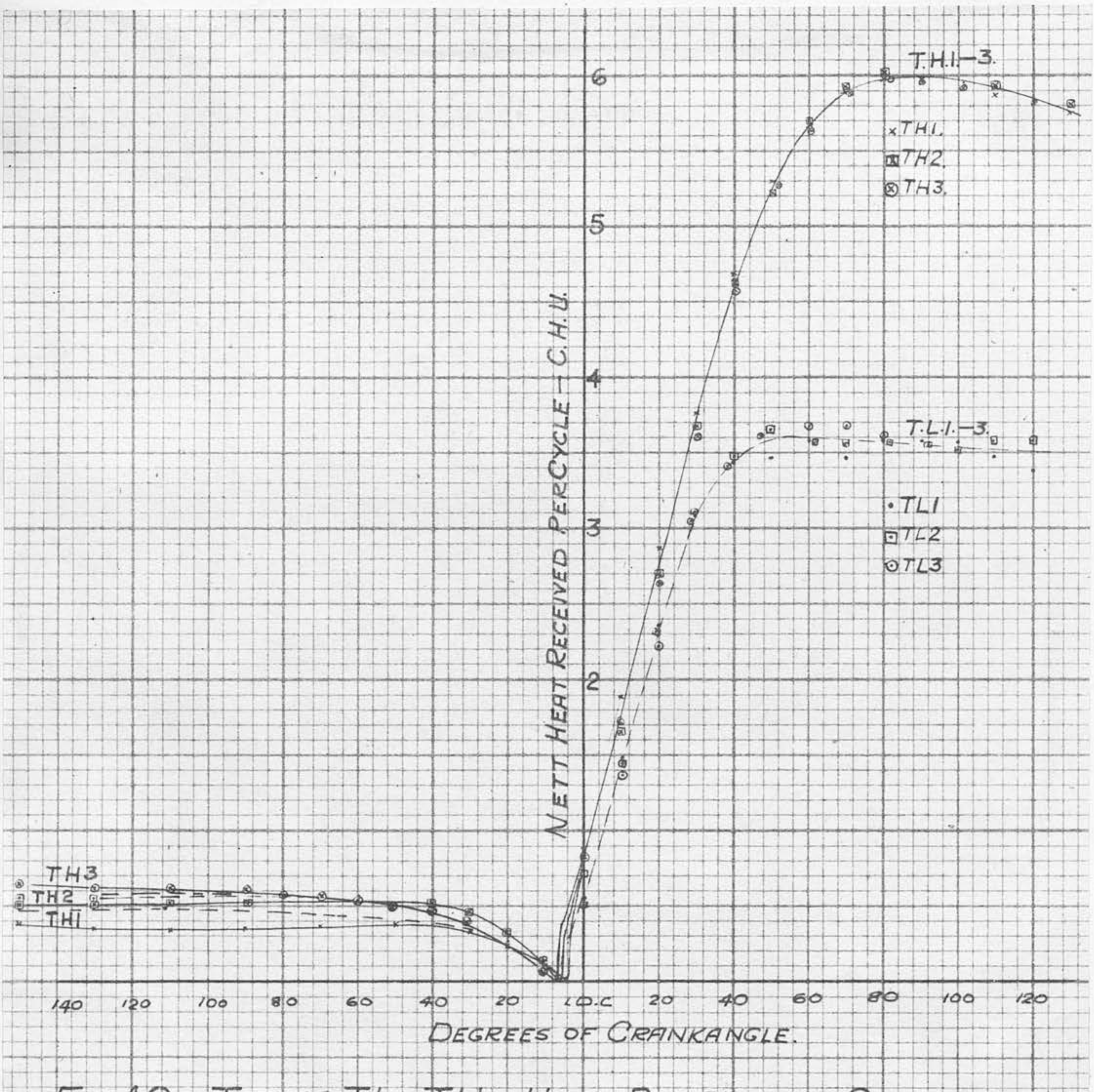



FIG 40. TESTS TL & TH. HEAT RECEPTION CURVES.

 with what?

indicate a small reduction in the delay period which is in agreement with bomb experiments by R.F. Selden* and others. ?

It would appear, however, that, for changes of air temperature up to 100 °C., there is no appreciable affect on the combustion process, the rate of heat reception being practically the same for all inlet temperatures and the rate of combustion would similarly be unaffected. The fact that the maximum temperature is reached at the same crank angle position (see Fig. 39) also confirms this view.

¶ Any ~~change~~ increase in inlet air temperature while the induction pressure is kept constant will result in a reduction of volumetric efficiency. Any change in combustion efficiency in such a case would be due essentially to the change in the quantity of air available for combustion and not to the temperatures existing inside the cylinder.

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* R.F. Selden - "Auto ignition and combustion of Diesel Fuel in a Constant Volume Bomb". N.A.C.A. report No. 617 (1938)

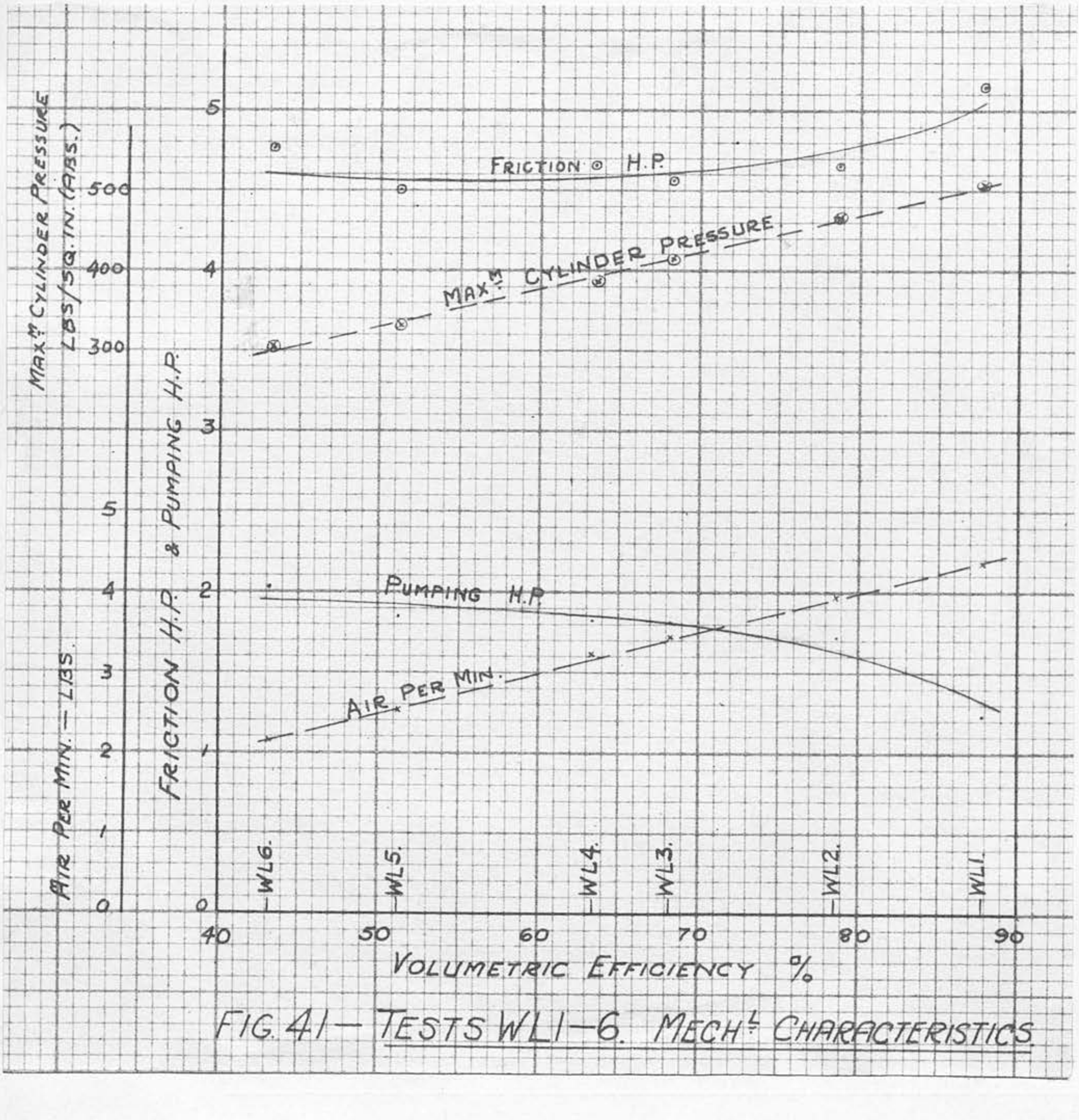


FIG. 41 – TESTS WL1–6. MECHᴸ CHARACTERISTICS

Effect of Air Density.

Note on the Volumetric Efficiency.

This has been calculated on the basis of the existing atmospheric conditions i.e.

Volumetric efficiency = $\frac{W_a RT}{144\ pv}$

where $W_a$ = weight of air consumed per cycle

$V$ = stroke volume of engine - cu.ft.

$p$ = atmospheric pressure - lb. per sq.in.abs.

$T$ = atmospheric temperature - °C. abs.

$R$ = Gas Constant for Air = 96.3

Fig. 41 shows the mechanical characteristics plotted against volumetric efficiency.

The friction horse-power does not appear to be greatly affected, though a reduction is noticeable and, allowing for small observational errors, the reduction appears to be maintained as the volumetric efficiency is reduced.

Increase in the pumping horse-power is more pronounced where the volumetric efficiency has been reduced only slightly below normal. No considerable increase in pumping power is noticeable for volumetric efficiencies below 70% and this is accounted for by the/

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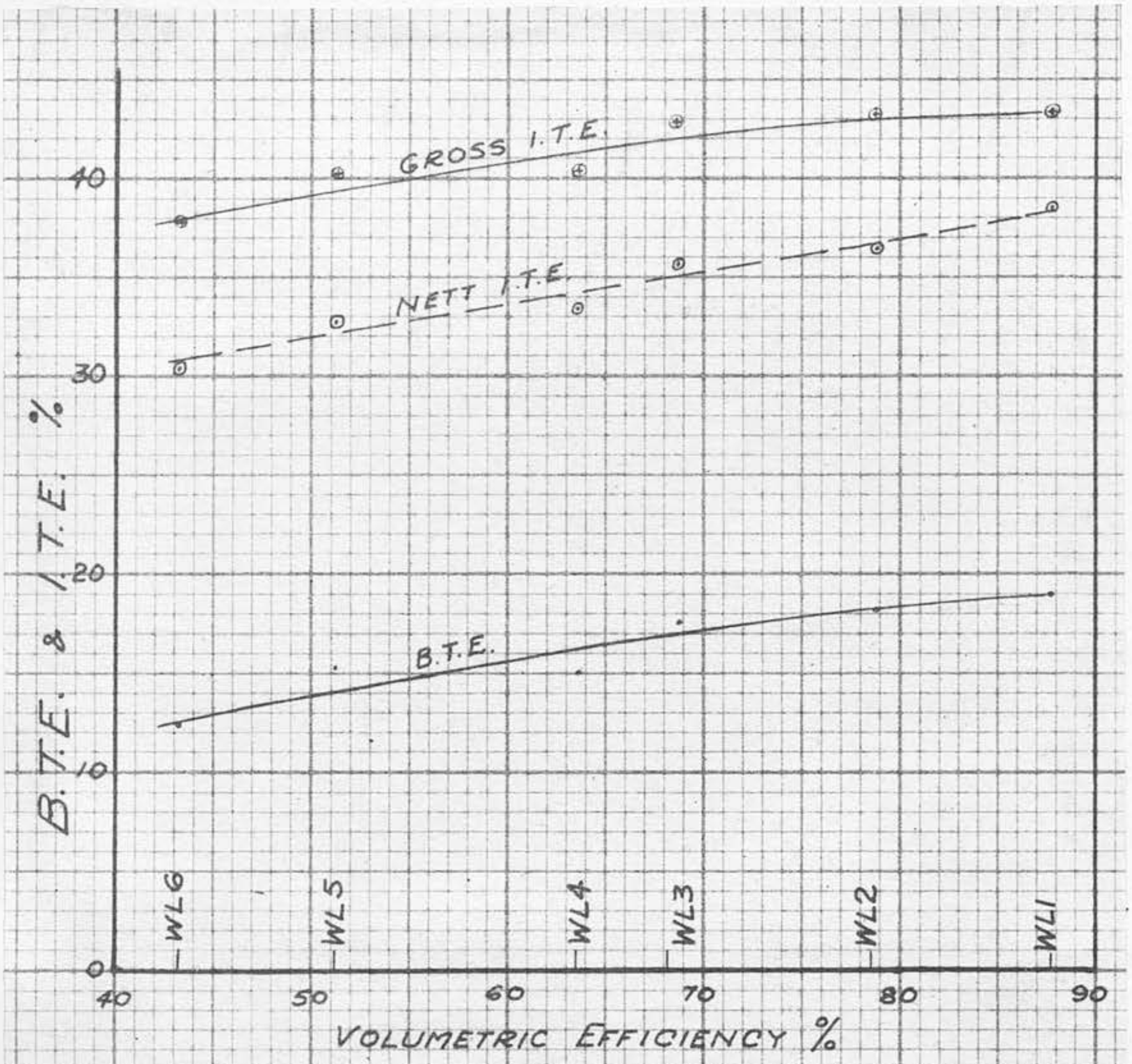


FIG 42. TESTS WL1-6. THERMAL EFFICIENCIES.

the reduction in compression pressures which offsets the reduced suction pressure (see light spring diagrams Figs. 33 - 38). The air consumption in pounds per minute is also shown. As would be expected, this is directly proportional to the volumetric efficiency. Care was taken when carrying out the tests to ensure them being under approximately the same atmospheric conditions. The speed of the engine will remain practically constant and, where necessary, adjustments were made to the governor control.

The maximum cylinder pressures have also been plotted in Fig. 41, showing a reduction in pressure almost directly proportional to the reduction in volumetric efficiency for the range covered by the experiments. This would seem to indicate that throttling the air supply might be advantageous at light load in the case of the lighter weight automobile engine.

The B.T.E. and gross I.T.E. are plotted in Fig. 42. Both show a reduction though, for volumetric efficiencies down to 65%, there appears little reduction in the gross I.T.E., the reduction in the nett/

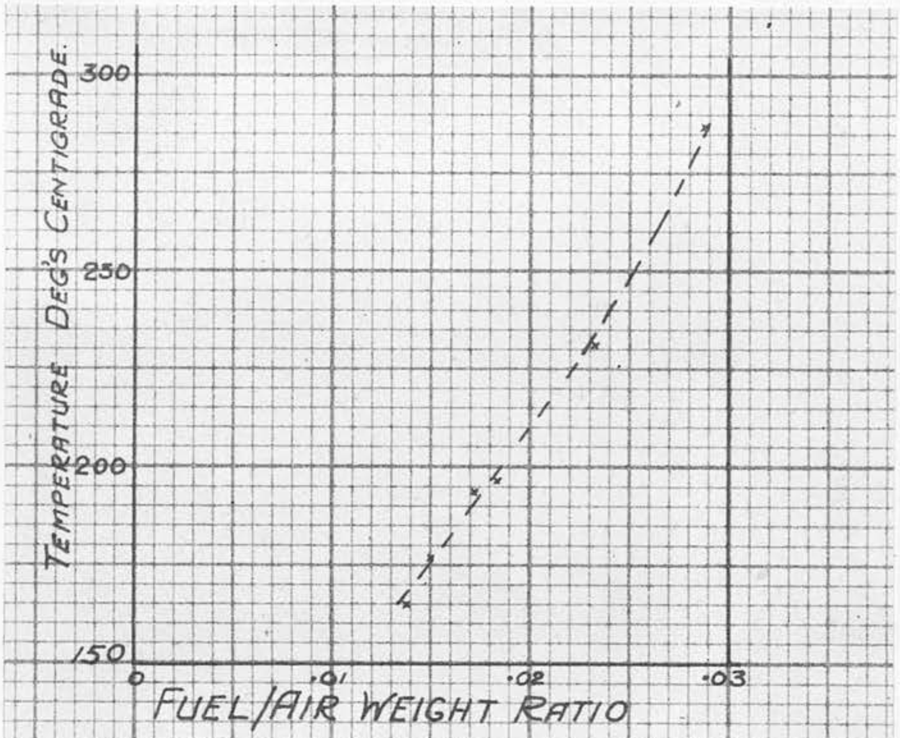


FIG 43. TESTS WL1-6. EXHAUST TEMPERATURE.

nett I.T.E. and the B.T.E. being due chiefly to the increased pumping loss - the reduction in friction helping to maintain the B.T.E. for volumetric efficiencies between 80% and 90%.

Fig. 43 shows the Exhaust pipe temperature plotted against the Fuel/Air Weight Ratio. As would be expected, the temperature varies almost directly with the ratio until the latter is increased beyond .025, though, even here, the variation of the curve from a straight line is not pronounced. This may be an indication that the period of combustion of the fuel has been unaffected by reduction in the volumetric efficiency, this being the only factor likely to affect the proportionality of exhaust temperature to Fuel/Air Weight Ratio.

Cyclical Changes of Temperature.

The cylinder gas temperatures have been calculated in accordance with the principles outlined on pages 21 and 24. The increased depression during suction may be expected to draw back a proportion of the exhaust from the pipe during the first thirty degrees of the suction stroke. To examine this feature, a recording of the exhaust pipe pressure was made for tests WL 1 and WL 6. No appreciable difference at the end of the exhaust/

Drawing of exhaust pipe ?
ch

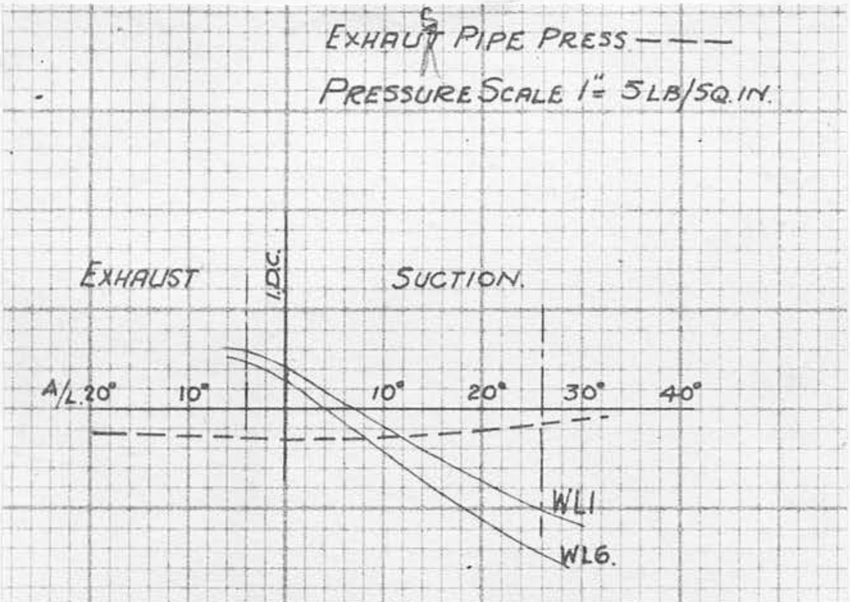


FIG 44. TESTS WL1-6.
EXHAUST PIPE & CYL^R PRESSURES.

exhaust stroke was noticeable so that the single "exhaust pipe pressure" curve, shown in Fig. 44, may be considered representative of all six tests. Superimposed are the cylinder gas pressures recorded during the same period. No very considerable difference is noticeable though, in general, the cylinder pressure during the first part of suction is reduced with the volumetric efficiency. (The pv diagrams, Figs. 33 to 38, indicate that the reduction in suction pressure is more pronounced as the stroke proceeds.) It appears, however, from Fig. 44, that, as the volumetric efficiency is reduced, more and more exhaust gases are drawn back from the pipe into the cylinder. An exact determination is practically impossible and, in any case, the amounts affected are not great, but an approximation may be arrived at by comparison with the induction and cylinder pressures obtaining in the load performance Tests P 1 - P 6.

For small pressure differences acting over a valve, the weight of gas passing per second is given by:-*

$$W = C_d A \sqrt{2g(p_1 - p_2)\rho}$$

where $C_d$ = coefficient of Discharge,

$A$ = area of Valve opening

$p_1 - p_2$ = pressure difference over valve

$\rho = \frac{\rho_1 + \rho_2}{2}$ $\rho_1$ and $\rho_2$ being the densities before and after passing the valve.

In/

* Schule - "Technical Thermodynamics" 1933 Edition p. 257.

In the case of the inlet air valve, the value of W was determined by gasometer, A was calculated from the valve lift and valve area, $(p_1 - p_2)$ obtained from the indicator diagrams, while $\ell$ was determined from the gas temperatures and pressures.

The value of $C_d$ obtained (= .16)* was substituted, in the case of the exhaust valve, for the mean conditions existing during the first thirty degrees of suction stroke in the case of test WL 6 and this gave a value of W = .01 pounds per second from which it was estimated that the flow into the cylinder from the exhaust pipe = .00018 pounds.

As this cannot be regarded as more than an approximation, the amount allowed for in the other five tests was taken in proportion to the reduction in volumetric efficiency below that of test WL 1, the amount allowed for in the case of WL 1 being zero.

It will be seen that the flow back into the cylinder is of negligible importance even in the worst case but the above method of calculation would appear to be the only method of indicating the amount of flow.

Consequently, the amount of residuals given in Table VII page 43 and Table X page 46 include a proportion allowed/

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* This low value is no doubt due to the annular shape of the valve opening and the turbulent flow.

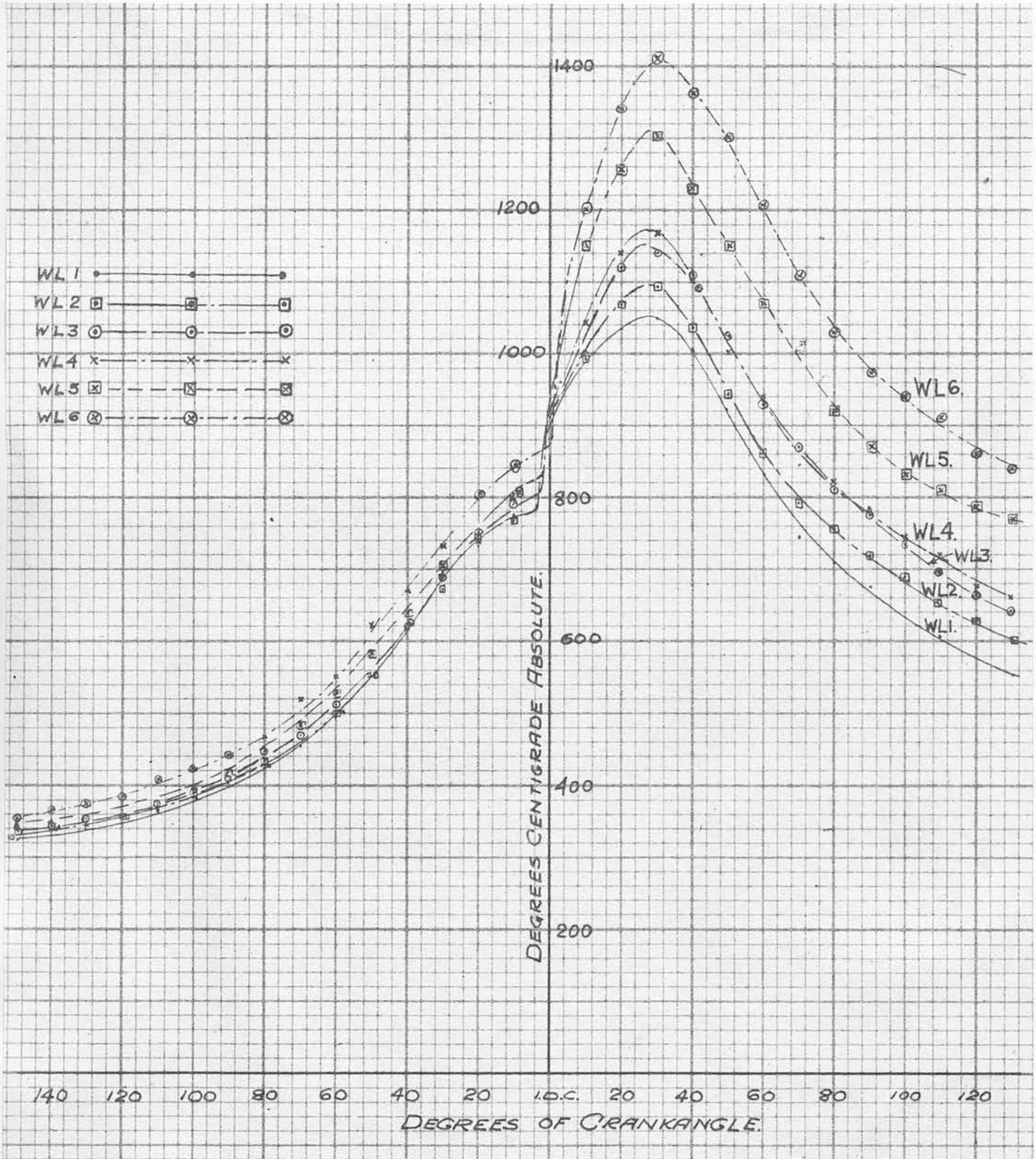


FIG 45. TESTS WL1-6. CYCLICAL CHANGE OF TEMPERATURE.

allowed for exhaust flow back into the cylinder.

A possible source of error in the charge weight is due to the exhaust valve not following the cam during the closing period. The exhaust valve spring was normally 40 pounds but an additional tension of 10 pounds increased this to 50 pounds. With a valve seat area of about $3\frac{1}{2}$ square inches, the latter force was equivalent to a pressure of 14.3 pounds per square inch. The maximum depression in the cylinder was 9.0 pounds per square inch (WL 6) though, during the valve closing period, it was not more then 3.4 pounds per square inch. Under these circumstances, there should be no leakage past the exhaust valve.

The cyclical change of temperature is shown to a crank angle base in Fig. 45. The compression temperatures are not greatly affected and this would seem to be due to the small volume of residuals associated with a high compression engine. It is noticeable, however, that decrease in the volumetric efficiency has greatest affect in raising the compression temperature when the volumetric efficiency is low.

As would be expected, the maximum expansion temperatures rise in step with the fuel/air ratio.

Heat/

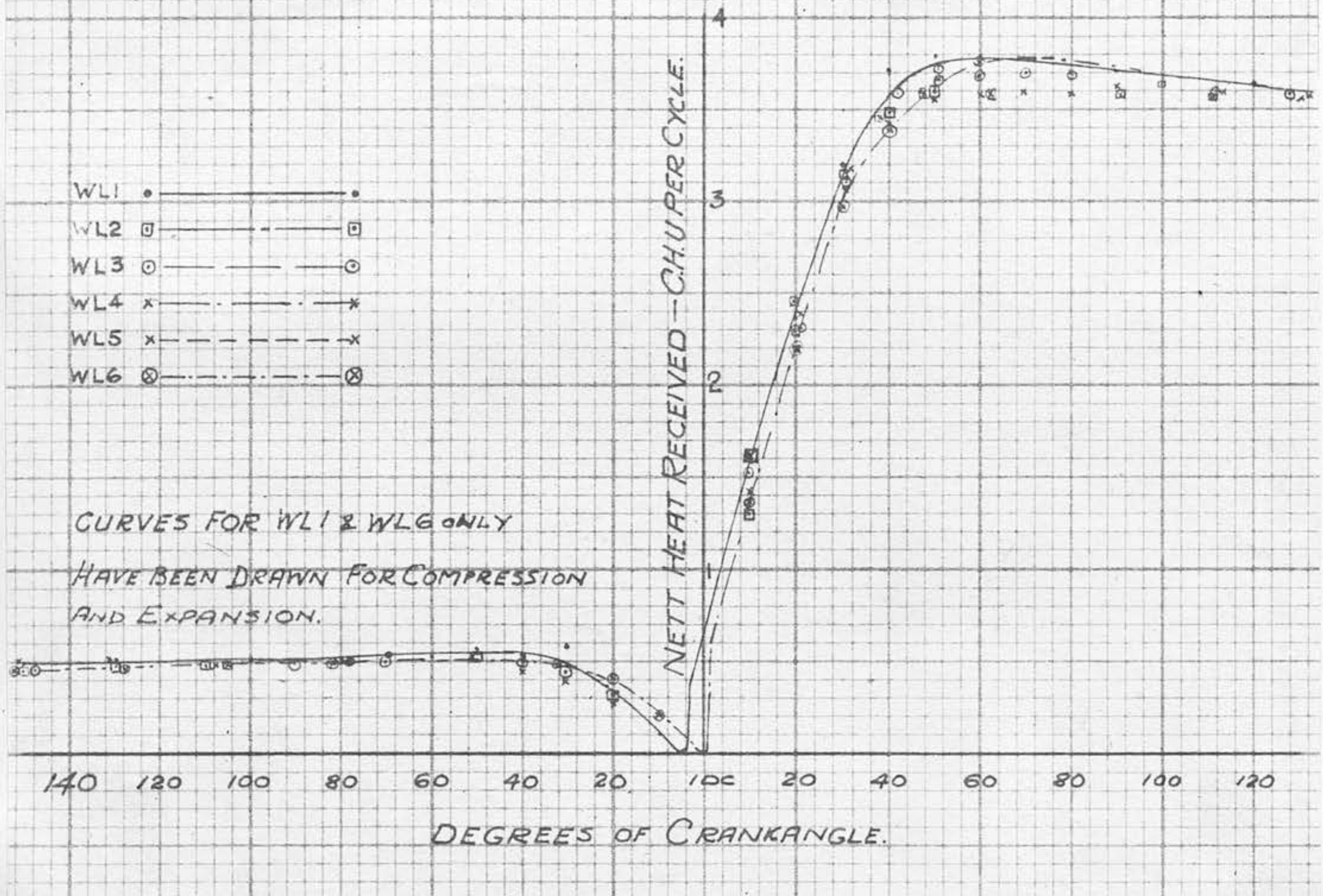


FIG. 46. TESTS WL1-WL6. HEAT RECEPTION CURVES.

HEAT RECEPTION AND RATE OF COMBUSTION:

Fig. 46 shows the heat reception curves. The most noticeable features are:-

(1) Delay in combustion increases with throttling. For the range covered, the delay period has been nearly doubled.

(2) The heat evolved during the rapid combustion period following delay is nearly constant, increasing somewhat with the degree of throttling.

(3) The rate of heat reception during the first twenty degrees of expansion is practically constant. Thereafter, combustion proceeds rather more slowly as the throttling is increased. In general, it would appear that, provided a minimum air/fuel ratio is exceeded, the rate of combustion will be dependent only on the rate of fuel injection into the cylinder (see also page 32.)

HEAT BALANCE:

The data given in Table XI (page 47) have been plotted in Fig. 47 on a percentage basis. The quantity of fuel injected per cycle is sensibly constant with the exception of tests WL 1 and WL 6 (representing an excess of 3% and 5% respectively above the mean of the other/

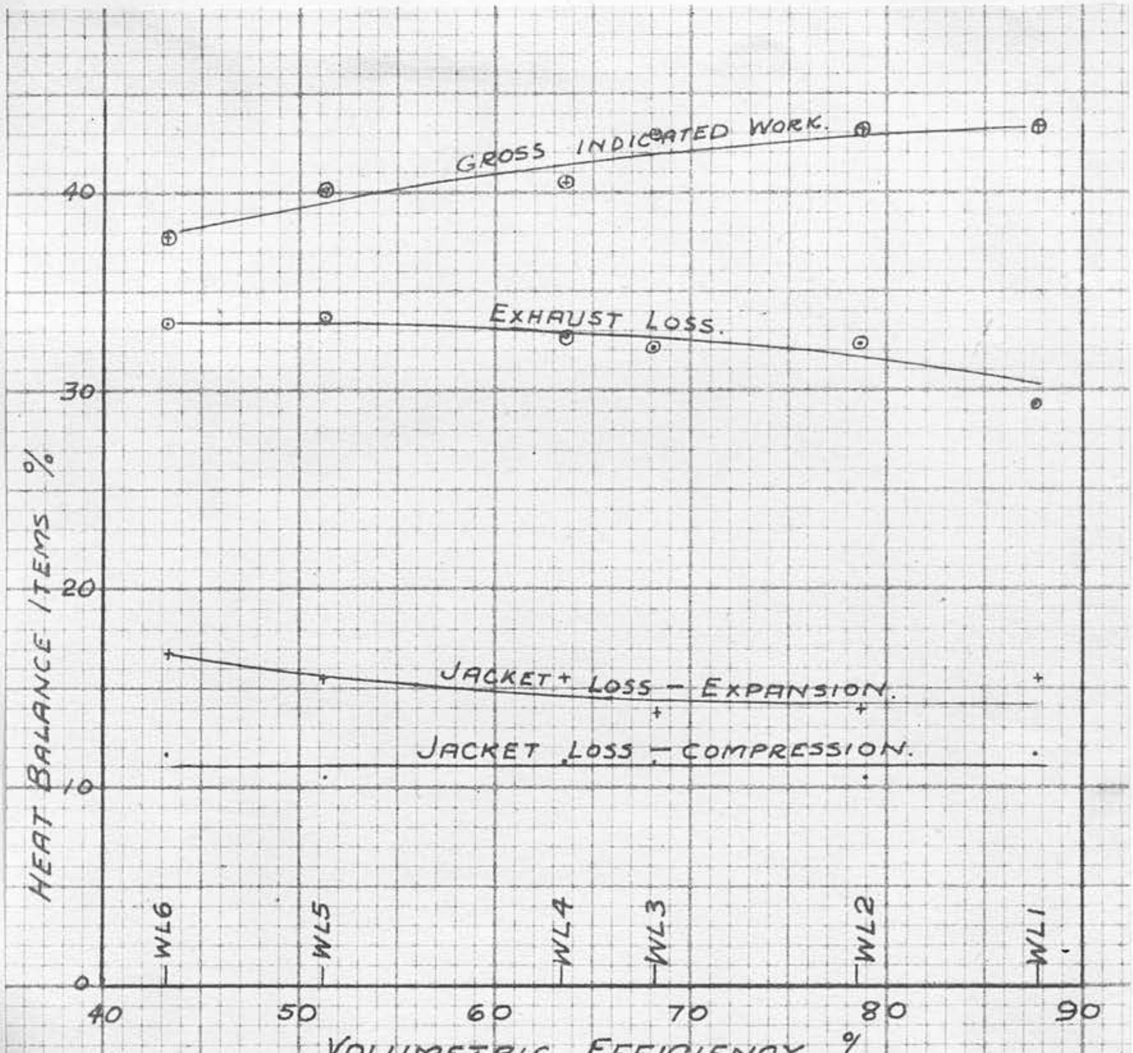


FIG 47. TEST WL1-6. HEAT BALANCE.

other four tests). Hence there will be little difference between the actual heat values and the percentage values.

The compression heat loss is practically unaffected by the change in the charge density brought about by throttling. It must be remembered, however, that, owing to the increased delay, the compression period prior to combustion has been increased at a time when the rate of heat loss is highest. Had the compression loss been taken in each test up to say six degrees before I.D.C., the loss would be seen to decrease with the charge density.

The heat loss during combustion and expansion is not greatly different in either of the four tests WL 2 to WL 5. The increased heat loss recorded in tests WL 1 and WL 6 appear then to be primarily due to the larger quantity of fuel injected.

The increase in the exhaust heat is not very marked, since the greatly increased temperature of the exhaust is counterbalanced by the reduced weight of charge. Any increase there may be in the exhaust heat is mainly accounted for by the increased specific heat at the higher temperatures induced by throttling.

It/

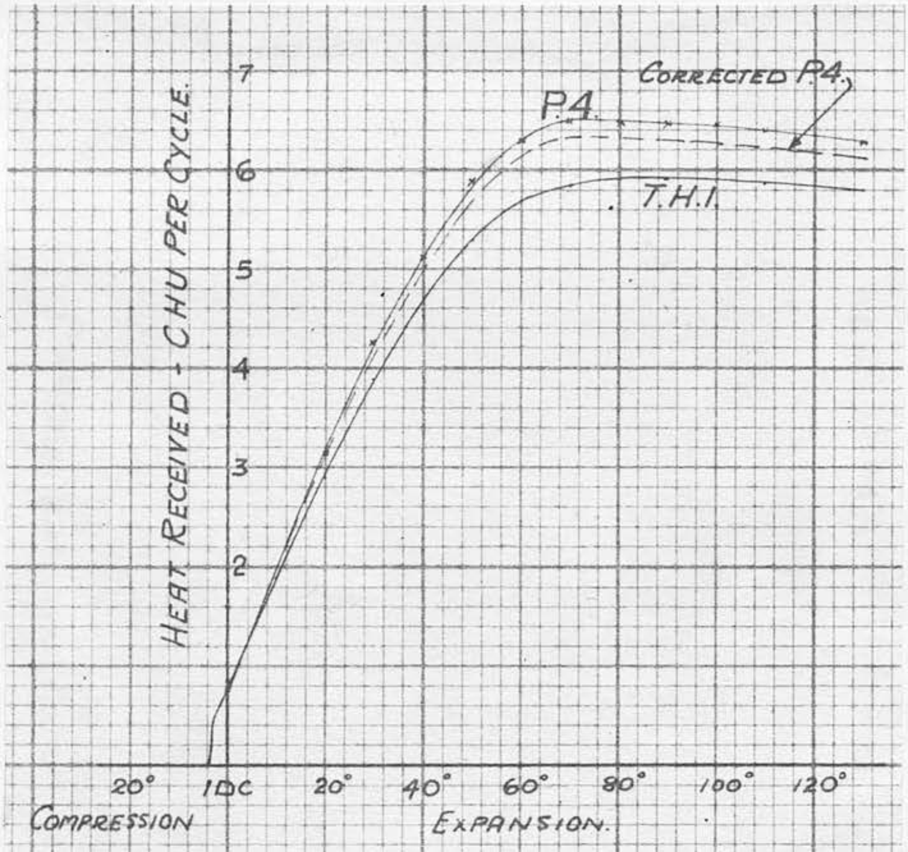


FIG 48. TESTS P4 & THI. HEAT RECEPTION.

It would seem then that the reduction in the thermal efficiency as throttling is increased is occasioned partly by the increased jacket loss and partly by the increase in the specific heat, both brought about by the higher gas temperatures.

COMPARISON OF OTHER TESTS:

It may be objected that in the above tests WL 1 to WL 6, the air used in all cases was too much in excess of that theoretically required.

It had been intended to carry out a test series at about half load as was done in the "air temperature" tests but time did not permit of this. A comparison may, however, be made between the results of test P 4 (Table III page 20) and test TH 1 (Table VII page 43) both of which employed approximately the same fuel consumption.

Fig. 48 shows the heat reception curves for these tests. The nett heat received by the charge is greater for P 4 but this may be accounted for by the discrepancy in the fuel consumptions. To correct this, resource was made to interpolating between the curves P 4 and P 3 in Fig. 22, and the resulting curve is also shown in Fig. 48.

The/

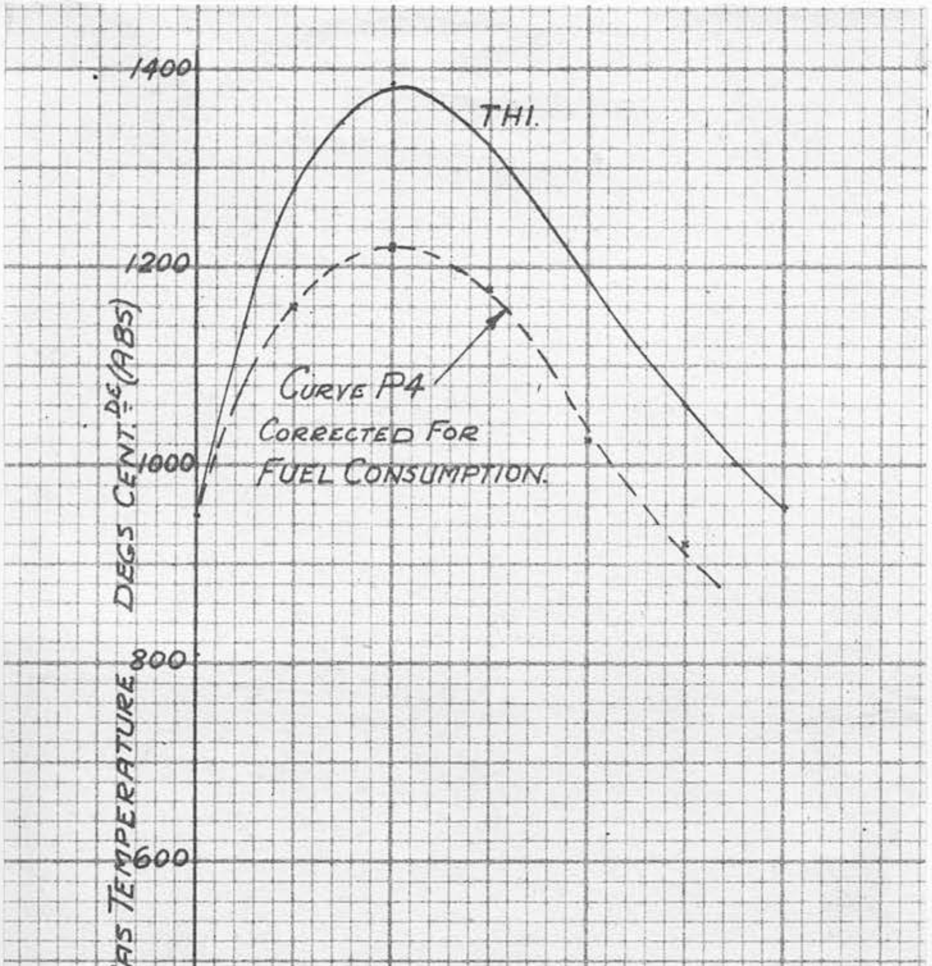


FIG 49. TESTS P4 & THI. TEMPERATURES.

The nett heat received as shown by the maximum value of the corrected curve P.4 amounts to 6.3 C.H.U. per cycle at 70° after I.D.C., whereas, for test TH 1, the corresponding values are 5.93 C.H.U. at 82° C. The heat losses during combustion and expansion up to release are:-

$7.49 - 5.82 = 1.67$ C.H.U.'s for test TH 1.

$7.49 - 6.12 = 1.37$ C.H.U.'s for test P.4 (corrected)

The combustion and expansion temperatures are shown in Fig. 49 from which it will be seen that the maximum temperature reached for test TH 1 (1380°C.abs.) is 11.3% greater than for the corrected P.4 (1222°C.abs.) The temperatures during the remainder of expansion follow in approximately the same order. Consequently, the heat loss to the jacket after combustion has ceased, should be greater in test WH 1 than in Test P. 4 (corrected). The average rate of heat loss shown by the curves in Fig. 48 for the last fifty degrees before release show, however, rather smaller loss for Test TH 1 than for Test P. 4. It would seem then that combustion continues further beyond its maximum heat reception point in Test TH 1 than it does in Test P. 4. Generally combustion will not be complete at the point of/

of maximum heat reception since these curves represent the nett heat received by the charge. If combustion is assumed to be complete at the following points:-

Test P. 4 $70^\circ + 10^\circ = 80^\circ$

Test TH 1 $82^\circ + 20^\circ = 102^\circ$

the effect of the reduction in air density will have been to increase the combustion period by:-

$$102^\circ - 80^\circ = 22^\circ.$$

Heat Loss during Combustion

The heat reception curves in Fig. 48 appear to indicate that, during the first forty degrees of the expansion stroke, the rate of combustion is approximately the same for test TH 1 as for test P. 4 (inclination of TH 1 curve is less than that of the corrected P. 4 curve, but the heat loss will be greater in the case of test TH 1, consequent on the higher gas temperatures).

The data available referring to radiation and conduction losses during combustion in a compression ignition oil engine are very scanty but it does appear that this loss far exceeds any loss during expansion after combustion

Experiments on the combustion of a fuel jet were carried out by L.R. Underwood in 1931* using a hot bulb/

* L.R. Underwood - "Combustion of an Oil Jet in an Engine Cylinder
Proceedings of Inst. of Mech. Eng. 1931 Vol. 121 p. 379.

bulb type compression ignition oil engine with bore and stroke 7.5 inches and 15 inches respectively. The rate of injection was relatively high - period of injection 15 degrees for load conditions similar to those obtaining in test P. 4 and the combustion period lasting about 50 degrees. Visual observation of the flame showed that it rapidly attained a bright white colour and remained thus until shortly after injection ceased. Thereafter, the flame gradually became irregular and duller until, at 50 degrees after I.D.C. combustion had apparently ceased. It would seem then that, if the rate of injection is slower, the rate of combustion will practically follow in step with the injection of the fuel into the cylinder and that the luminosity will not vary greatly during the injection period. (This too is in keeping with the mean charge temperatures recorded in Fig. 18.) The radiation and conduction losses should then be proportional to the period of injection.

Tests P. 1 to P. 6 have been used to plot a graph showing the variation of heat lost during the combustion period with the period of injection. The heat lost during combustion has been taken as the difference between the maximum "heat received" (as determined/

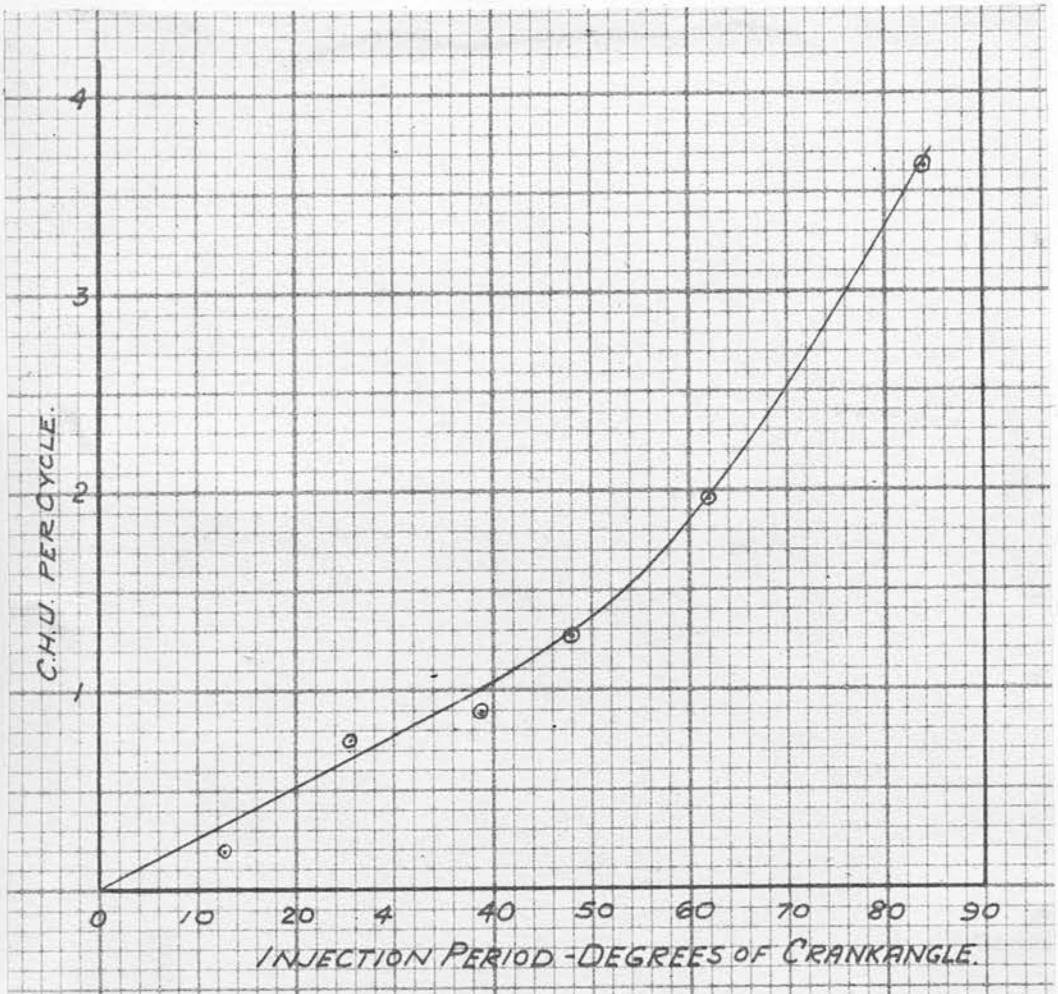


FIG 50. HEAT LOST DURING COMBUSTION

determined from Fig. 22) and the heat supply per cycle by the fuel.

Thus: Test P. 4 Maximum Heat received = 6.47
Heat supplied by fuel per cycle .. .. = 7.75
∴ Heat loss in combustion = 1.28 C.H.U.

This graph is shown in Fig. 50. Up to about 18 I.H.P. (gross), the loss is roughly proportional to the injection period but, with further load, the "loss" increased much more rapidly.

The mean temperature of the charge is higher at heavy load and this will in itself increase the rate of loss of heat to the jacket. It seems, however, that the chief factor would be that the combustion is by no means complete at the point of maximum heat reception, in the case of the full load tests.

## CONCLUSION

While it is realised that the heat reception curves that have been plotted in the foregoing pages do not give an exact picture of the combustion process, they do give a comparison during the early stages of combustion. During the last stages of injection at the heavier loads, they indicate, by their slow change of direction, that combustion is proceeding more and more slowly.

The effects of induction conditions on combustion have, perhaps, not shown up very well but it does seem to be indicated that, in a slow speed engine of this type, the process is not greatly affected provided a minimum safe air/fuel ratio is exceeded.

The observations recorded on the delay period, while somewhat erratic, this being expected in view of the slow speed of the engine and the governor fuel control, do bear out the results of other experiment~~o~~rs. Explain

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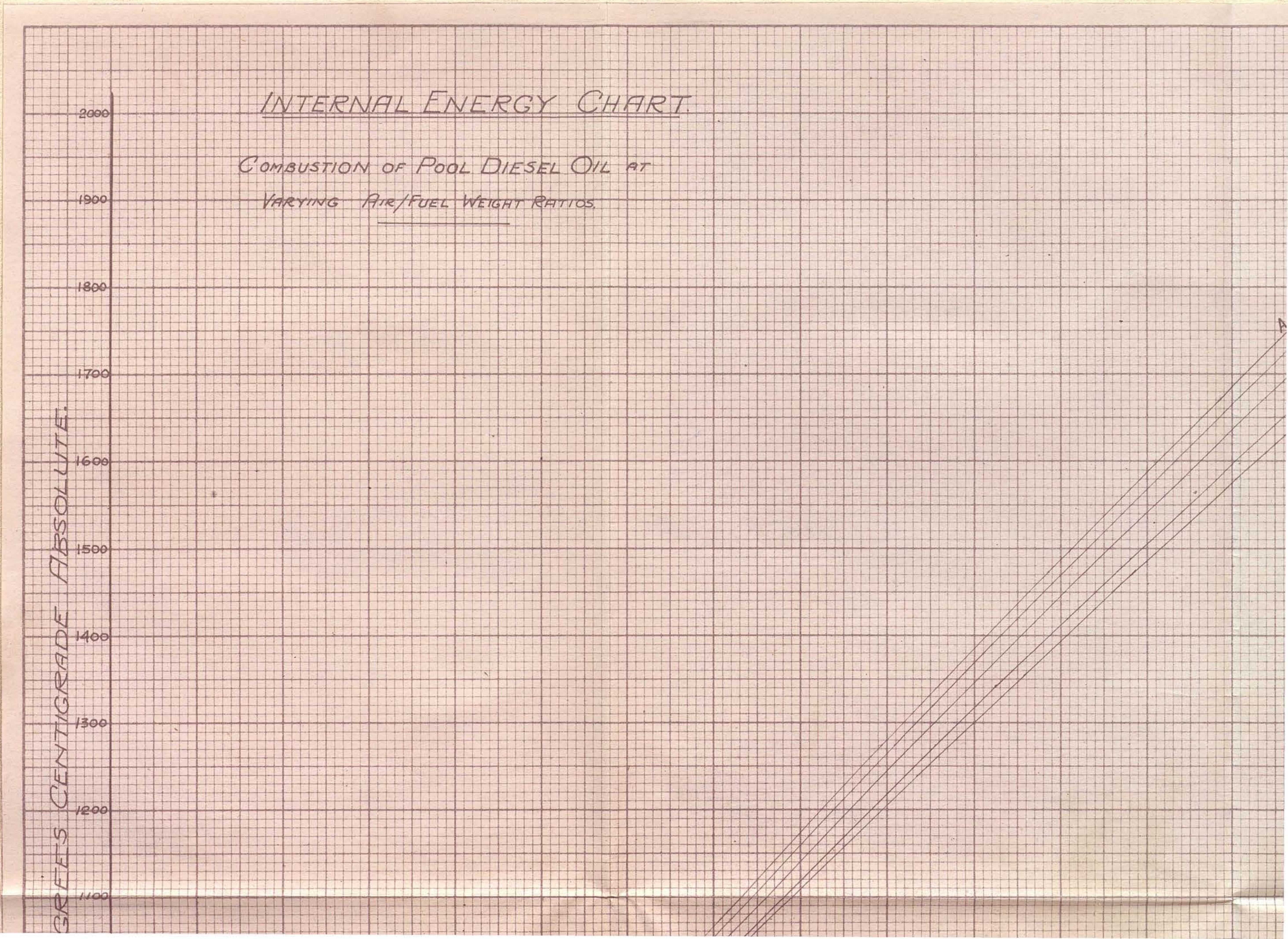
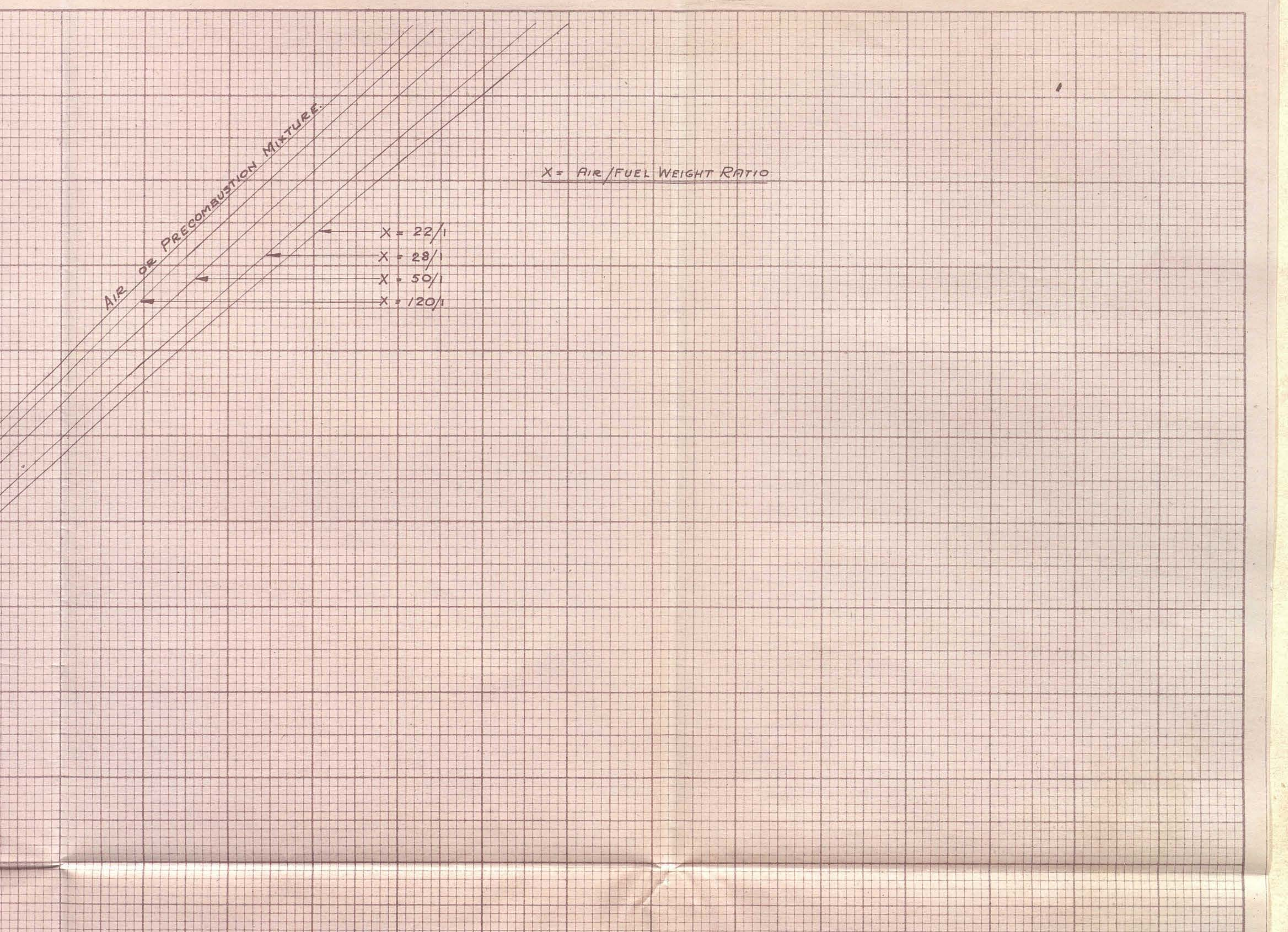
Contents of Cover Pocket.

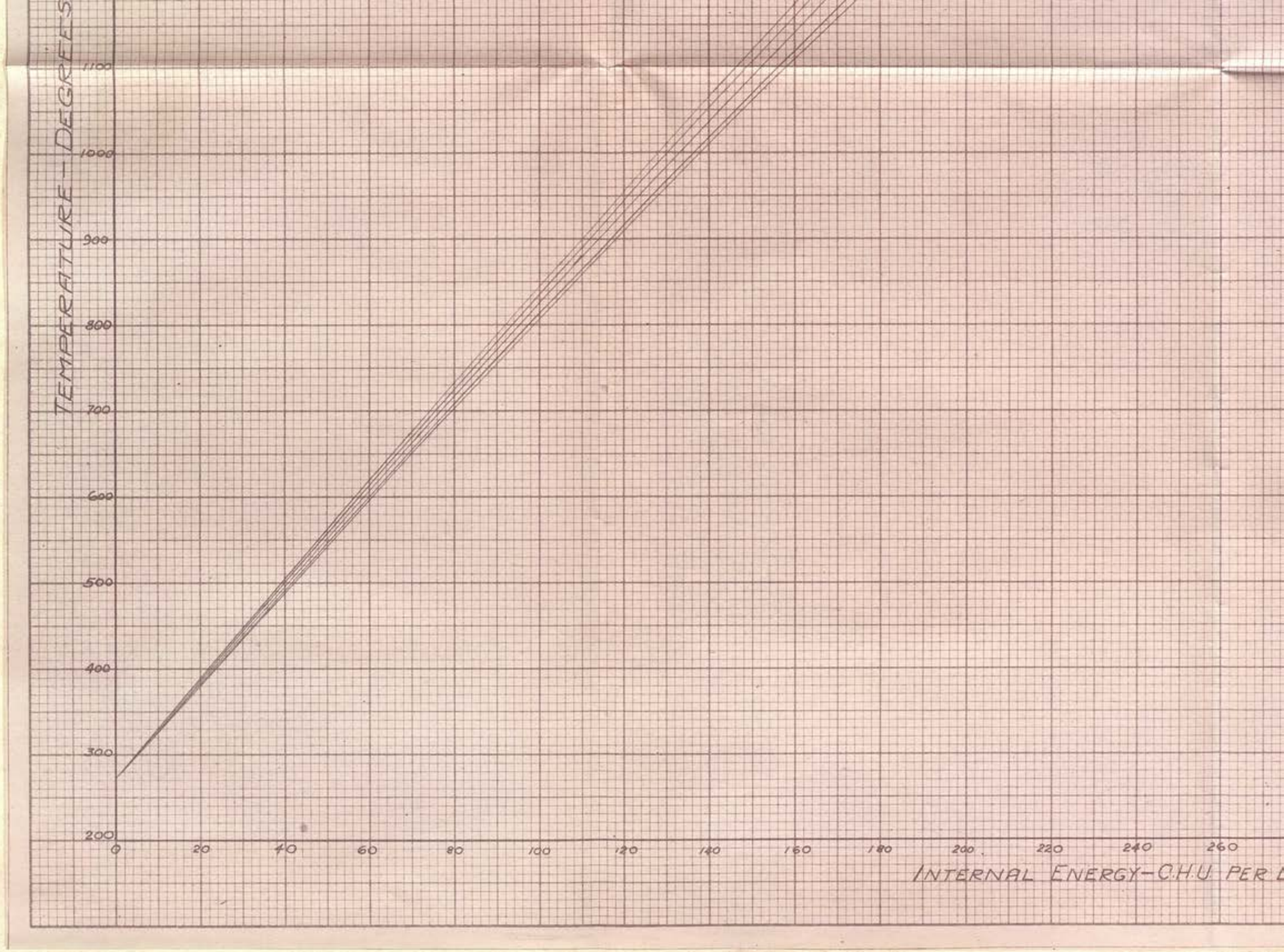
1. Energy Chart.

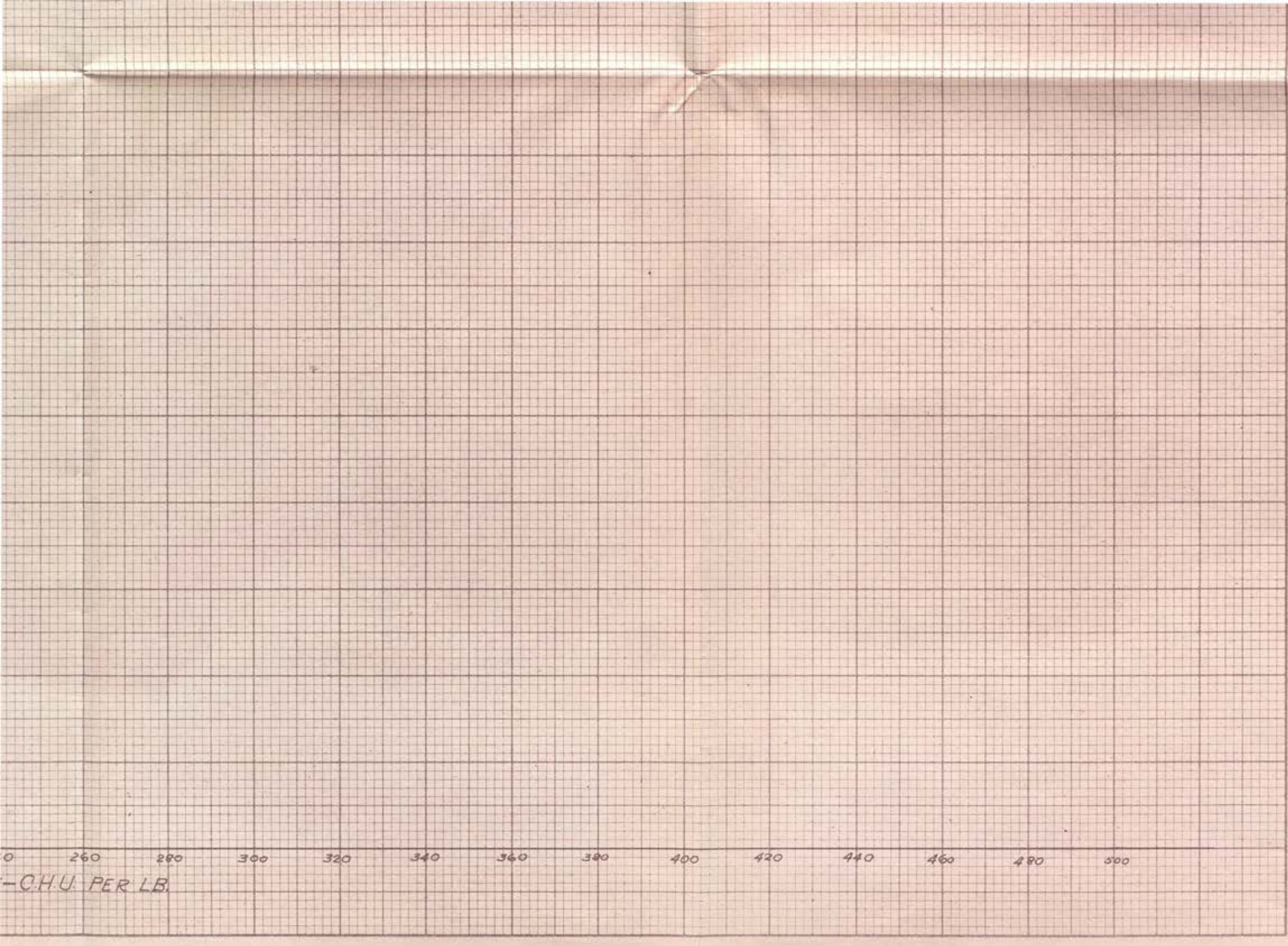
2. Crank angle Indicator Diagrams.

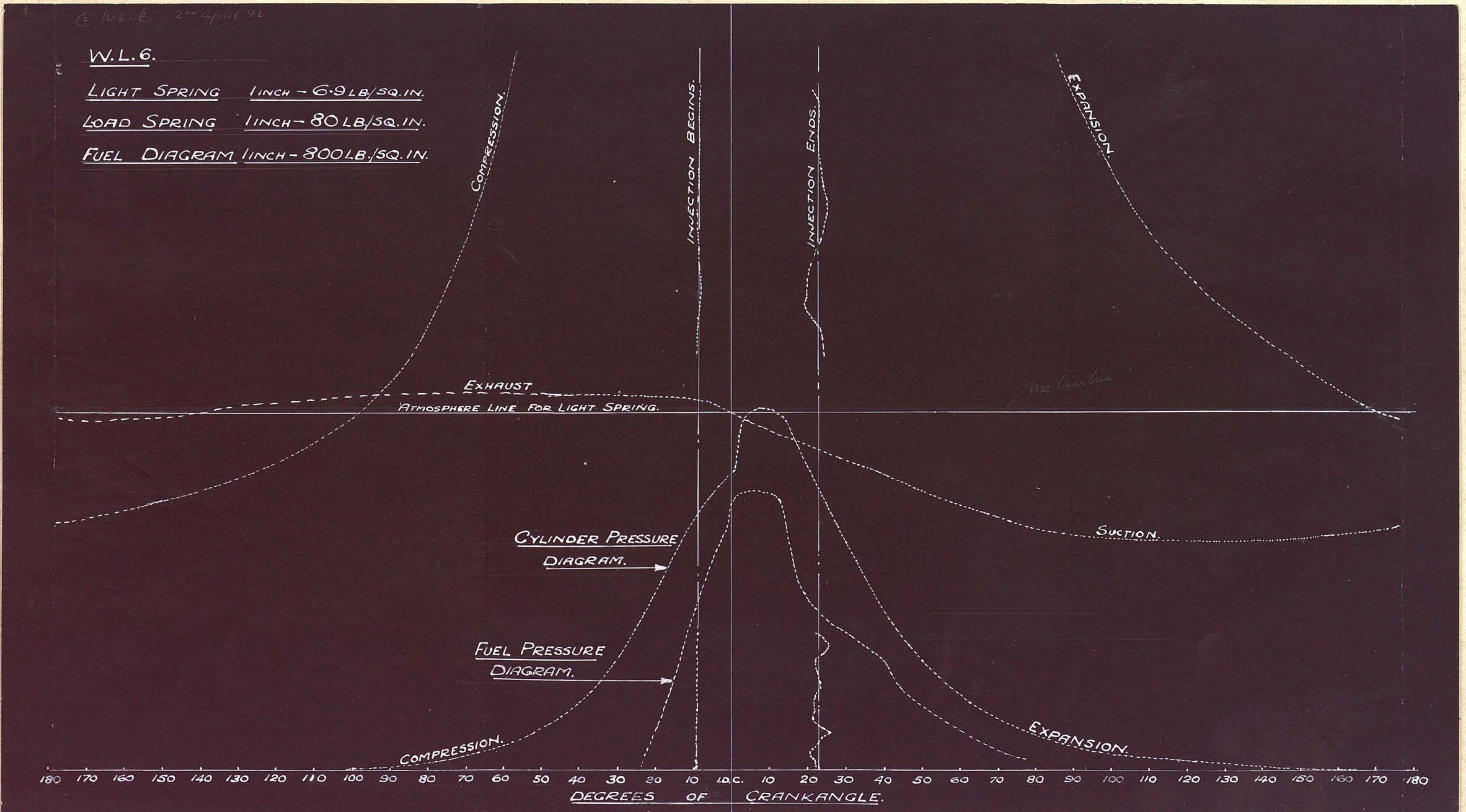

INTERNAL ENERGY CHART.
COMBUSTION OF POOL DIESEL OIL AT
VARYING AIR/FUEL WEIGHT RATIOS.
2000
1900
1800
1700
1600
1500
1400
1300
1200
1100
GREES CENTIGRADE ABSOLUTE.

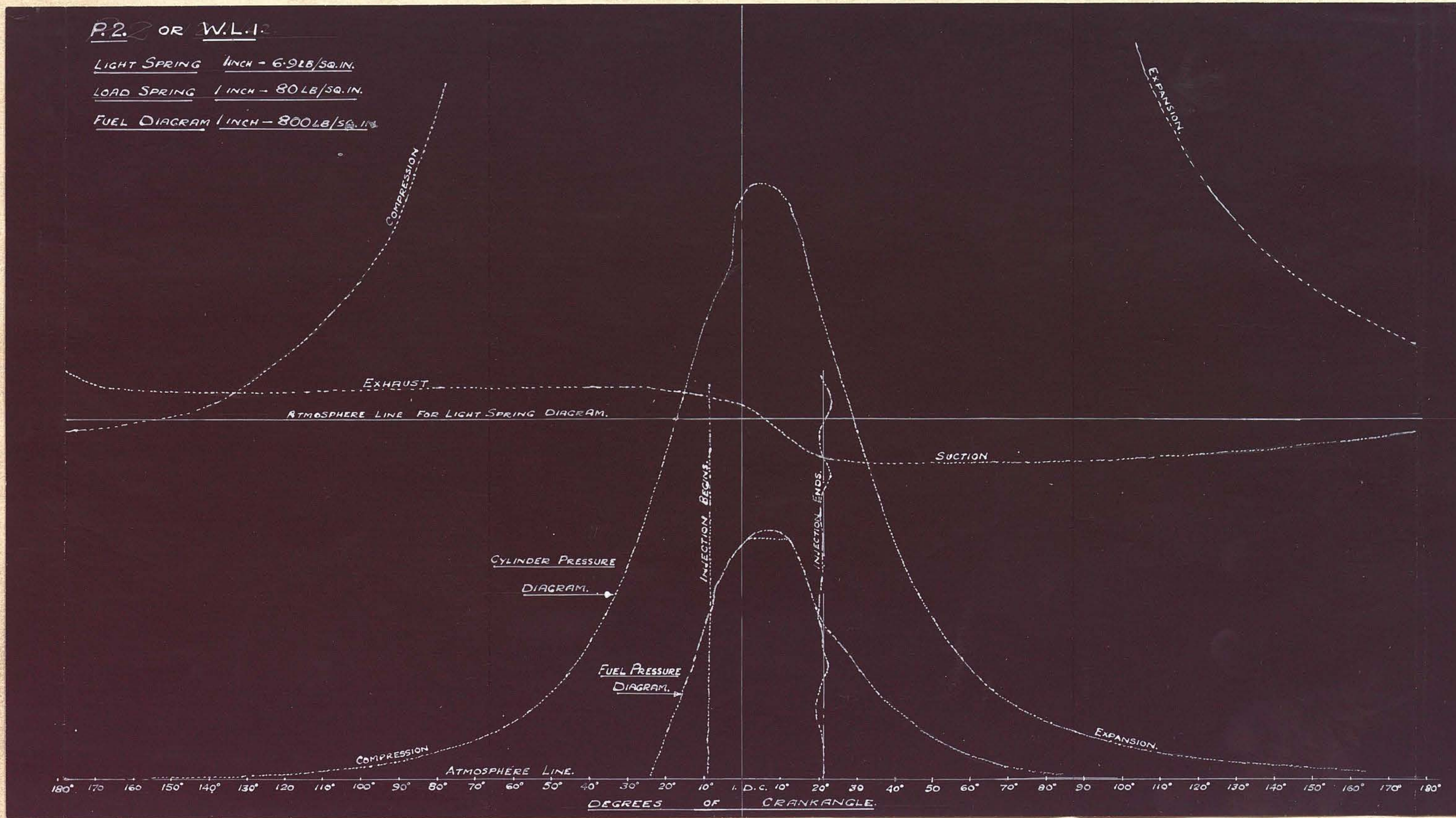


AIR OR PRECOMBUSTION MIXTURE.
X = AIR/FUEL WEIGHT RATIO
X = 22/1
X = 28/1
X = 50/1
X = 120/1


TEMPERATURE - DEGREES
1100
1000
900
800
700
600
500
400
300
200
0
20
40
60
80
100
120
140
160
180
200
220
240
260
INTERNAL ENERGY-C.H.U. PER


260 280 300 320 340 360 380 400 420 440 460 480 500
–C.H.U. PER LB.


W.L.6.
LIGHT SPRING 1 INCH - 6.9 LB/SQ.IN.
LOAD SPRING 1 INCH - 80 LB/SQ.IN.
FUEL DIAGRAM 1 INCH - 800 LB./SQ.IN.
COMPRESSION.
INJECTION BEGINS.
INJECTION ENDS.
EXPANSION.
EXHAUST
ATMOSPHERE LINE FOR LIGHT SPRING.
SUCTION.
CYLINDER PRESSURE DIAGRAM.
FUEL PRESSURE DIAGRAM.
COMPRESSION.
EXPANSION.
180 170 160 150 140 130 120 110 100 90 80 70 60 50 40 30 20 10 T.D.C. 10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180
DEGREES OF CRANKANGLE.



P.2. OR W.L.1.
LIGHT SPRING 1INCH - 6.9LB/SQ.IN.
LOAD SPRING 1 INCH - 80 LB/SQ.IN.
FUEL DIAGRAM 1 INCH - 800LB/SQ.IN.
COMPRESSION
EXPANSION.
EXHAUST.
ATMOSPHERE LINE FOR LIGHT SPRING DIAGRAM.
SUCTION
INJECTION BEGINS.
INJECTION ENDS.
CYLINDER PRESSURE
DIAGRAM.
FUEL PRESSURE
DIAGRAM.
COMPRESSION
EXPANSION.
ATMOSPHERE LINE.
180° 170 160 150° 140° 130° 120 110° 100° 90° 80° 70° 60° 50° 40° 30° 20° 10° I. D.C. 10° 20° 30 40° 50 60° 70° 80° 90 100° 110° 120° 130° 140° 150° 160° 170° 180°
DEGREES OF CRANKANGLE.