Degree of Ph. D.

THESIS

on

The Effect of Jacket Temperature on Oil Engine Performance

submitted by

Maxwell Davidson, B.Sc.

INTRODUCTION.

It is generally assumed that an internal combus:tion engine operates more efficiently when the mean temperature of the jacket water is high.

Whatever gain there may be in the efficiency, this can always be expressed in terms of heat whether such a gain be due to purely thermal or to mechanical reasons. It may happen that for such an engine run at a particular load, variation in mean jacket temper:ature produces only thermal changes. On the other hand, it is possible that modifications of a mechanical nature will be introduced, and these may or may not be conducive to increased efficiency.

The value of the mechanical loss in an engine is generally taken to represent the difference between the power developed within the cylinder and that at the shaft. This loss is caused by bearing friction, plunger operation, and piston friction. What propor:tion each bears to the whole is doubtful.

The object of the experiments under consideration was to examine - in so far as the available equipment would permit - the effects produced on an oil engine performance, by variation of the jacket water temper:ature.

These /

These effects might be classified as:

a) Thermal - combustion, etc.

b) Mechanical - friction.

While the former would generally be considered the more important, in the present case greater space is given to the latter in an attempt to separate the total friction into its components:

1) bearing and valve operation friction;

2) piston friction.

The investigation was made on one of the test units in the Heat Engines Laboratory of the Engineering Department, University of Edinburgh.

Three series of trials were run, with cold, medium and hot jackets, the powers developed in each case varying from nothing to a maximum, with the intermed-:iate values approximately equal in the three series.

APPARATUS

ENGINE:

The test unit used was a horizontal four stroke cycle, single cylinder, "National", heavy oil engine of the ordinary commercial type, having a bore of 8 inches, stroke 16 inches, and a volume compression ratio /

ratio of 13.11. The normal full load of this engine is 18 B.H.P. and the speed 290 R.P.M. A cam operated plunger pump delivered the fuel to the atomiser, the pressure of the fuel lifting the needle valve off its seat against the action of a spring. A bye-pass valve controlled by the governor regulated the quantity of fuel delivered to the cylinder. The engine was fitted with a water cooled brake ring on the flywheel for power absorption purposes. R.P.M

FUEL SUPPLY:

For measurement of fuel consumption, two glass vessels of double conical form, tapering to a narrow neck at each end, were employed. These were previous:ly calibrated and the narrow necks suitably marked. The procedure during any test was to switch over from one vessel to the other, checking the fuel consumption rates for consistency, by means of a stop watch.

AIR SUPPLY:

The air supplied to the engine was measured at frequent intervals - about every 10 minutes - by a gasometer. The latter, which had a capacity suffic:ient for about 3 minutes run of the engine, was charged and discharged continuously during tests.

A /

A description of the air measuring equipment* is inserted in the cover pocket.

INDICATOR GEAR:

For indicating purposes an existing "Farnboro" indicator was used from which load, light spring and fuel diagrams were obtained. It might be considered that the employment of an indicator of this type is unwarranted for a slow running engine, but, while the ordinary type is, with intelligent operation, no doubt quite satisfactory for power estimations on slow speed engines, it leaves much to be desired when data involving pressure and volume measurements are required with some guarantee of accuracy.

The indicator drum was driven directly from the engine crankshaft by a flexible coupling of the vernier type, the latter proving very useful in procuring the correct phasing. The top dead centre or inner centre line indicated correct phasing when its position was such as to bisect the area between the compression and re-expansion curves as sparked on the drum paper to a crank angle base during the motoring of the engine. The /

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* Engineering: April 16, 1937.

The diagrams produced by this indicator represent the change of cylinder pressure to a crank angle base. For the conversion of these to pressure-volume diagrams a transparent scale, marked off in degrees and equal in length to the circumference of the drum, is used. The pressure at any crank angle is then read off and plotted on a prepared PV chart. This chart has a base line representing the stroke volume and is divided by vertical ordinates into lengths equivalent to every 10 degrees of crank angle. In the preparation of the charts as used for this engine due allowance was made for obliquity of connecting rod.

A disc valve unit recorded the cylinder pressures and a differential valve the fuel pressures.

Objection might be taken to the use of the term "top centre" for a horizontal engine, instead of "inner centre". When preparing the PV charts, how:ever, "T.C." and "B.C." were inadvertently printed, and they are retained in the text.

<u>JACKET WATER</u>:

Two calibrated tanks were employed for the measurement /

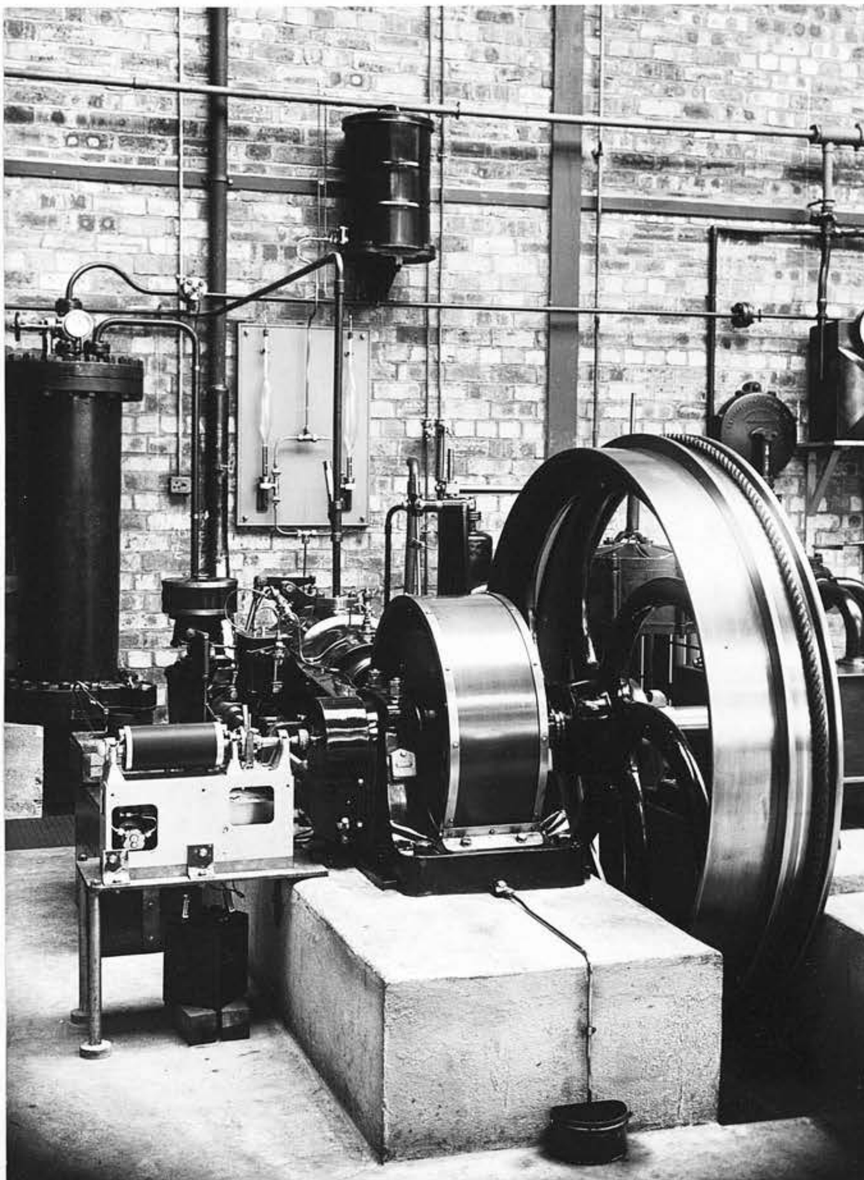


FIG. No 1.

measurement of the jacket water quantity and the rates of flow checked, the inlet and outlet jacket tempera:tures being maintained at the desired values. For the tests with the high inlet temperature a separate hot water supply was used for mixing with cold before entry to jacket.

EXHAUST THERMOCOUPLE:

The exhaust temperature was measured by a platinum-platinum rhodium thermocouple inserted in the exhaust pipe as close to the engine as possible and about four inches from the exhaust valve. Compensating leads connected the couple to a portable indicator fitted with automatic cold junction compensator.

Views of the complete test equipment are shown in Figures 1 and 2.

TEST ARRANGEMENTS AND PRECAUTIONS.

In order to obtain as high a degree of accuracy as possible great care was taken to ensure that con:ditions became steady before commencing the actual tests.

A relatively small power unit such as was used here /

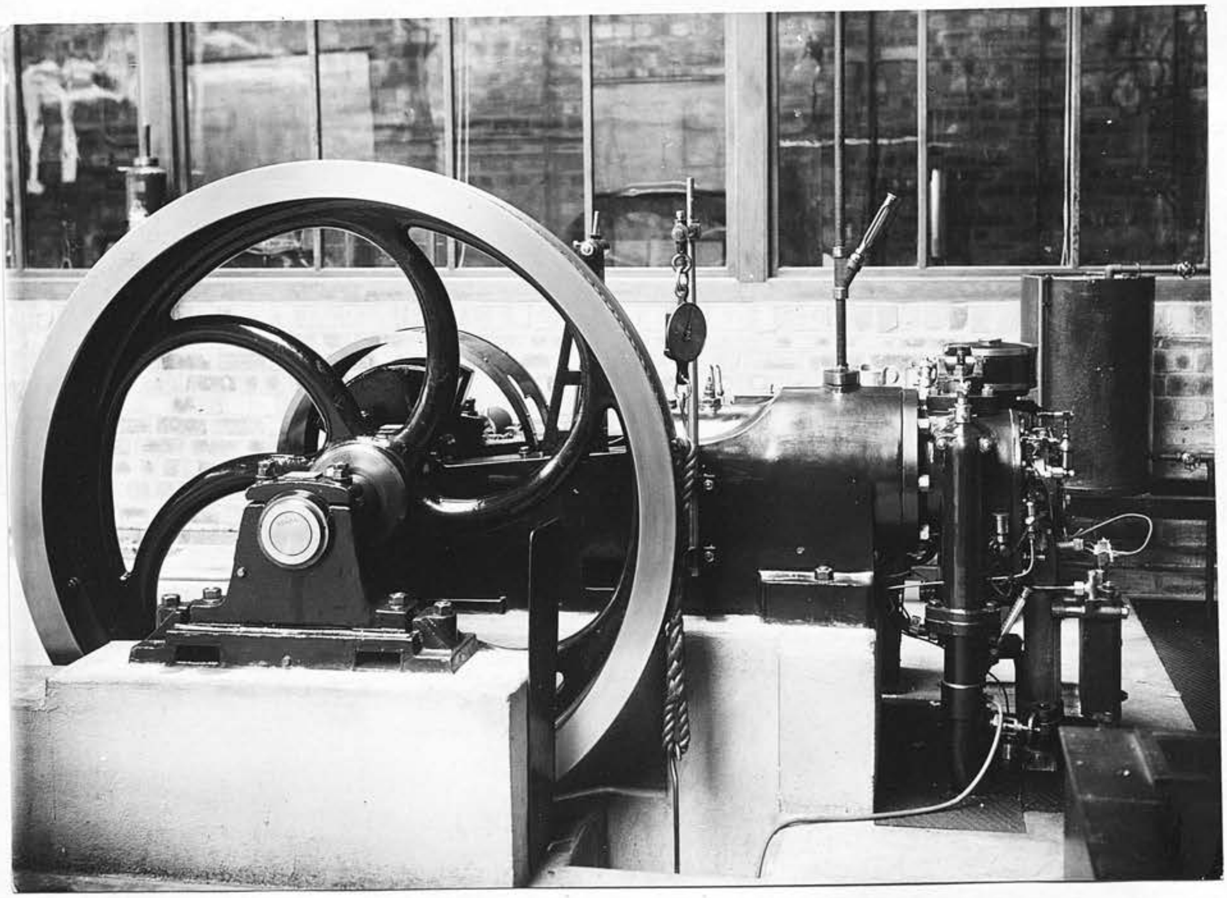


FIG. No 2.

here tends of course to reduce the time interval nec:essary for the adjustment of the single variable which in this case was either jacket temperature or load. Generally, however, the interval required for steady running conditions to prevail was as long as that of the actual test which in all cases was approx:imately one hour.

Jacket water inlet and outlet temperatures, exhaust temperature, spring balance reading, and R.P.M. were noted every 5 minutes. These were found to be consistent throughout the tests in all except 3 cases, which were repeated.

While it was desirable that the jacket inlet temperature for each series of tests should be the same, this was found impracticable due to varying seasons and severe weather conditions. During any one test however this temperature did not vary more than one and a half degrees above or below the mean, while the outlet temperature was similarly stable.

It was observed that, when developing the highest power with the cold and medium jackets, there was a certain amount of unsteady running due no doubt partly to /

to the adverse jacket conditions and to the fact that this power really represented 12 per cent. more than the normal full load. On such occasions there was a periodic fall and increase in engine speed and during the increase in speed the exhaust temperature rose about 10° above the mean.

It is, of course, appreciated that the actual temperature at the exhaust valve would be above that recorded, due to heat loss to pipe, etc., walls. The possible error caused by this is discussed later. Before and after the series of tests the indicator was checked by a standard potentiometer and found to be correct.

With regard to indicating, two sets of load, light spring and fuel diagrams were obtained during each test. They were taken in sequence, one of each type, and then the process repeated in the latter half of the test in order to procure good average diagrams. It was found, however, that even in the tests where there was the slight running irregularity, already mentioned, the respective diagrams were in very close agreement; the diagrams of course each represented a large /

large number of cycles. Having thus compared the crank angle diagrams for similarity, the required PV diagrams were later reproduced and from these the mean effective pressures and other details.

TEST OBSERVATIONS.

For classification purposes the tests have been numbered according to load and jacket conditions, e.g. 1.C.; 2.M.; 3.H.; etc,, the numeral indicating the power or load series and the letter the jacket cond-:itions, cold, medium, or hot.

The fuel oil used throughout the tests was "ESSOSTAT" Diesel Fuel Oil.

Calorific Values:-		Gross, 10,900 C.H.U. per lb.
		Nett, 10,350 C.H.U. per lb.
Weight Analysis:-		C, 85.6 to 85.8 per cent.
		H, 13.09 to 13.1 per cent.
		S, 0.45 to 0.48 per cent.
		O & N, 0 to 0.63 per cent.

Specific Gravity at 15°C., 0.845 to 0.855.

By calculation, the theoretical air required per lb. of oil for complete combustion is 14.5 lb.

During /

During most of the cold and medium jacket tests, samples of the exhaust gas were analysed by the Orsat apparatus. No trace of CO or other combustible gas could be found and these analyses were discontinued.

Table I gives the observations and general results calculated therefrom. The Brake and Indicated Thermal Efficiencies (abbreviations BTE and ITE) are based on the lower calorific value of the fuel.

Table I is followed by the pressure-volume diagrams in sequence. These are to the same pressure scale and practically the same volume scale as the actual crank angle diagrams. One set of crank angle diagrams, from which the pressure-volume diagrams were reproduced, is inserted in the cover pocket.

TABLE I /

TABLE I.

Test No.	1.C.	1.M.	1.H.	2.C.	2.M.	2.H.	3.C.
Average R.P.M.	294.2	295.8	294.0	292.5	292.5	291.0	291.0
B.H.P.	0	0	0	4.75	4.78	4.72	9.46
Jacket Water Inlet Temp. °C.	25.0	69.0	72.9	23.0	70.8	69.6	24.6
Jacket Water Outlet Temp. °C.	9.8	10.6	51.3	8.5	12.0	51.8	8.1
Jacket Water Qty./min. lb.	17.15	2.33	3.09	22.0	2.93	6.82	24.8
Heat to Jacket per min. C.H.U.	261	136	66.7	320	172	124	410
I.M.E.P. (gross) lb./sq.in.	26.5	20.9	19.25	41.8	35.5	31.7	56.5
I.M.E.P. (nett) lb./sq.in.	21.6	15.6	14.5	37.6	31.2	27.5	52.5
I.H.P. (nett)	6.44	4.69	4.32	11.2	9.26	8.13	15.55
Mech.Effy. %	-	-	-	42.5	51.7	58.0	60.9
Oil used/min. lb.	.0442	.0368	.0328	.0665	.0582	.0538	.0950
Oil/hour lb.	2.65	2.21	1.97	3.99	3.49	3.225	5.70
Oil/1000 cycles lb.	.30	.247	.2234	.454	.398	.369	.653
Oil/B.H.P./hour lb.	-	-	-	.840	.730	.684	.602
Oil/I.H.P./hour lb.	.412	.471	.456	.357	.377	.397	.367
B.T.E.	-	-	-	16.25	18.7	20.0	22.7
I.T.E.	33.15	29.0	29.9	38.2	36.2	34.45	37.3
Air used/min. lb.	4.831	4.842	4.622	4.875	4.672	4.505	4.573
Air/oil weight ratio	109	131	141	73.4	80.4	83.9	48.2
Excess Air %	653.0	807.0	872.0	406.0	454.0	478.0	232.0
Atmospheric Pressure lb./sq.in.abs.	14.5	14.8	14.25	14.6	14.53	14.07	14.01
Atmospheric Temp. °C.	14	14	15	13	13	15	14
Exhaust Temp. °C.	123	121	123	174	173	170	253

TABLE I (contd.)

Test No.	3.M.	3.H.	4.C.	4.M.	4.H.	5.C.	5.M.
Average R.P.M.	290.8	290.0	290.6	290.5	290	290.2	291.0
B.H.P.	9.26	9.27	13.65	13.75	13.45	15.7	15.75
Jacket Water Inlet Temp. °C.	70.0	71.6	24.4	68.9	71.4	25.1	69.4
Jacket Water Outlet Temp. °C.	9.0	49.5	8.0	11.6	49.8	8.0	11.3
Jacket Water Qty./min. lb.	4.1	9.15	30.0	6.45	13.10	30.0	7.04
Heat to Jacket per min. C.H.U.	250	202	492	370	283	513	409
I.M.E.P.(Gross) lb./sq.in.	48.4	47.8	67.5	64.1	62.5	72.8	70.3
I.M.E.P.(Nett) lb./sq.in.	44.7	43.3	63.9	60.1	58.6	69.1	66.3
I.H.P.(Nett)	13.2	12.75	18.85	17.7	17.25	20.35	19.6
Mech.Effy. %	70.1	72.6	72.4	77.7	77.9	77.2	80.4
Oil used/min. lb.	.0830	.0804	.1226	.1140	.1102	.1384	.1303
Oil/hour lb.	4.98	4.82	7.35	6.84	6.615	8.30	7.83
Oil/1000 cycles lb.	.570	.5535	.844	.785	.76	.954	.897
Oil/B.H.P./hour lb.	.538	.518	.540	.497	.492	.530	.497
Oil/I.H.P./hour lb.	.377	.378	.390	.387	.384	.407	.400
B.T.E.	25.4	26.2	25.3	27.5	27.8	25.75	27.5
I.T.E.	36.2	36.2	35.0	35.3	35.7	33.4	34.1
Air used/min. lb.	4.407	4.477	4.566	4.512	4.435	4.692	4.588
Air/oil weight ratio	53.2	56.8	37.3	39.6	40.3	33.9	35.2
Excess Air %	267.0	305.0	157.0	173.0	178.0	134.0	143.0
Atmospheric Pressure lb./sq.in.abs.	14.0	14.28	14.10	14.43	14.36	14.63	14.68
Atmospheric Temp. °C.	13	18	12	15	18	12	15
Exhaust Temp. °C.	245	233	322	318	310	357	352

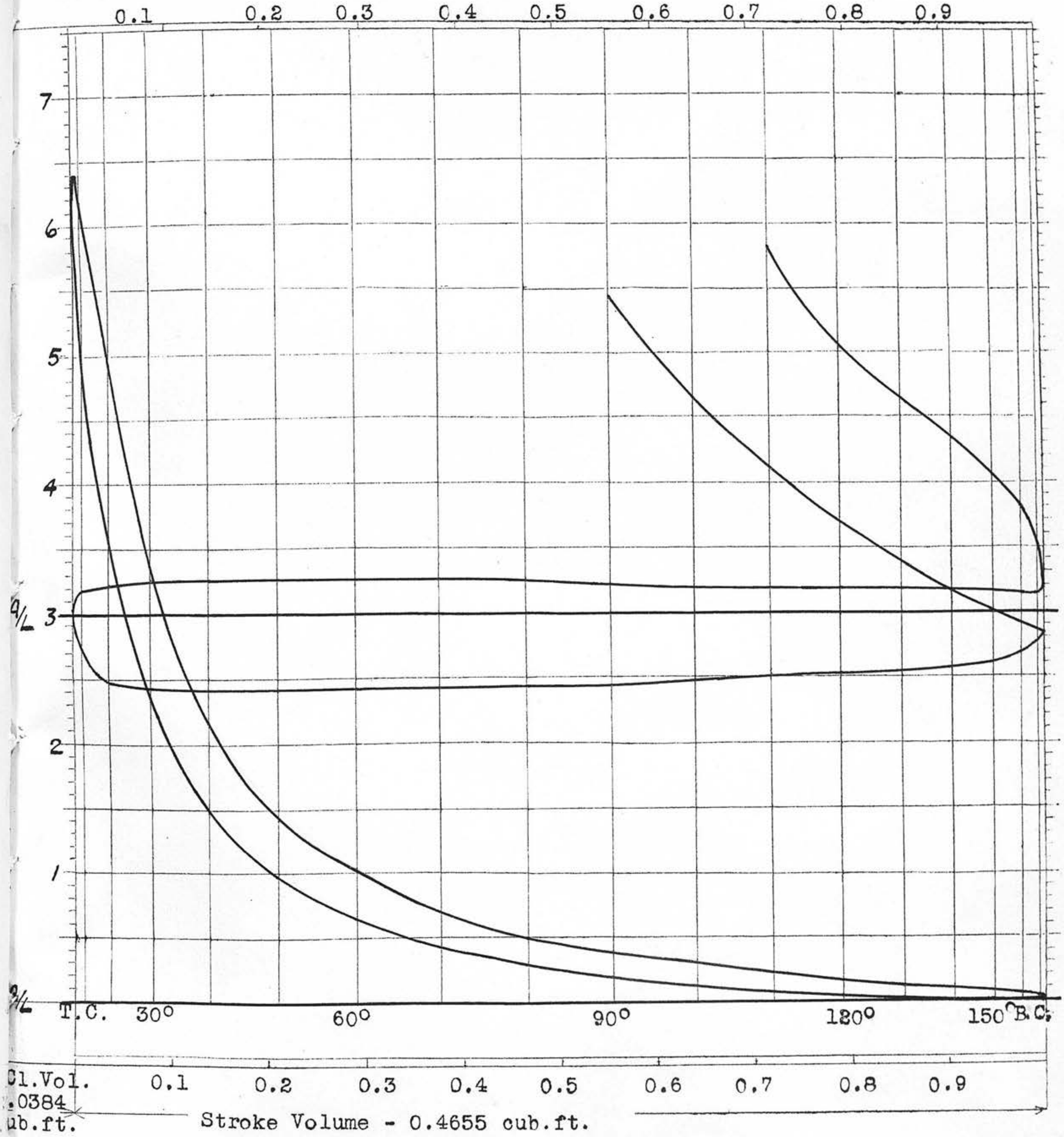
TABLE I (contd.)

Test No.	5.H.	6.C.	6.M.	6.H.	7.C.	7.M.	7.H.
Average R.P.M.	290.0	291.7	291.2	290.0	292.0	291.0	290.0
B.H.P.	15.4	18.4	18.6	17.8	19.9	19.7	19.2
Jacket Water Inlet Temp. ^{o}C.	71.8	23.4	69.5	70.2	23.6	68.6	70.6
Jacket Water Outlet Temp. ^{o}C.	49.1	8.0	10.6	50.8	8.0	10.7	50.0
Jacket Water Qty./min. lb.	14.81	38.1	9.16	21.63	42.0	9.95	22.48
Heat to Jacket per min. C.H.U.	337	587	540	420	655	576	464
I.M.E.P.(gross) lb./sq.in.	70.0	81.7	81.1	79.5	87.5	85.1	83.4
I.M.E.P.(nett) lb./sq.in.	65.8	77.8	76.6	75.6	83.9	81.6	79.9
I.H.P.(nett)	19.4	23.1	22.7	22.25	24.9	24.1	23.55
Mech.Effy. %	79.5	79.8	81.8	80.6	79.7	81.8	81.6
Oil used/min. lb.	.1265	.1700	.1650	.1531	.1895	.1797	.1693
Oil/hour lb.	7.6	10.20	9.90	9.2	11.47	10.78	10.16
Oil/1000 cycles lb.	.873	1.164	1.132	1.055	1.297	1.235	1.168
Oil/B.H.P./hour lb.	.493	.554	.533	.5165	.572	.546	.5315
Oil/I.H.P./hour lb.	.392	.442	.436	.414	.461	.447	.432
B.T.E.	27.7	24.6	25.6	26.5	23.85	25.0	25.7
I.T.E.	34.9	30.9	31.3	33.05	29.9	30.5	31.6
Air used/min. lb.	4.483	4.516	4.488	4.415	4.670	4.499	4.360
Air/oil weight ratio	35.5	26.5	27.2	28.8	24.6	25.0	25.75
Excess Air %	145.0	82.7	84.8	98.6	69.6	72.5	77.6
Atmospheric Pressure lb./sq.in.abs.	14.58	14.47	14.8	14.44	14.86	14.83	14.43
Atmospheric Temp. ^{o}C.	18	15	14	18	11	13	18
Exhaust Temp. ^{o}C.	346	447	440	425	478	473	460

FIGURE NO.3.

OIL ENGINE. Bore 8". Stroke 16". Compr.Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



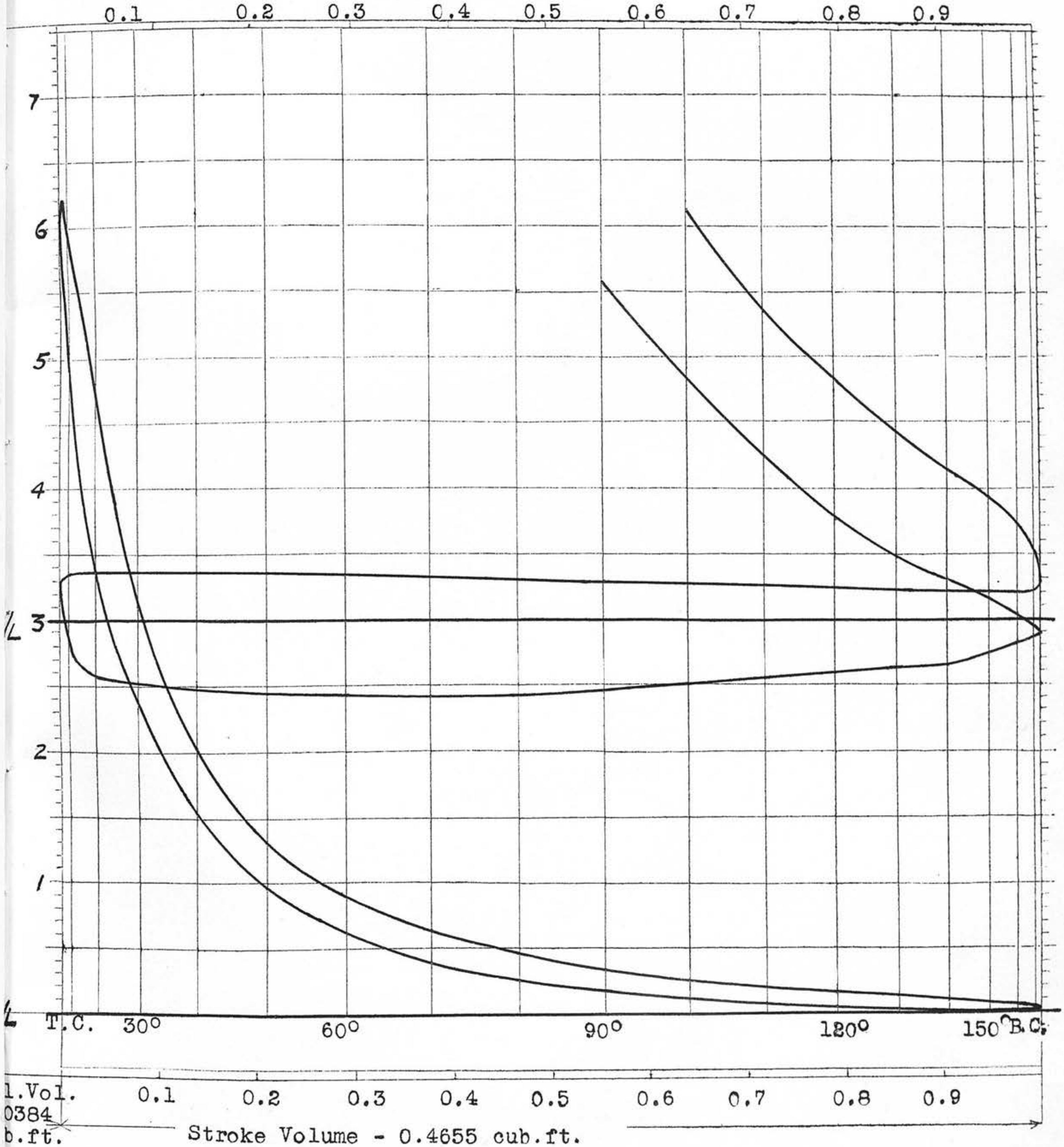
TEST NO. 1.C. p_m (gross) 26.5 lb./sq.in.

p_m (nett) 21.6 lb./sq.in.

FIGURE NO.4.

OIL ENGINE. Bore 8". Stroke 16". Compr.Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



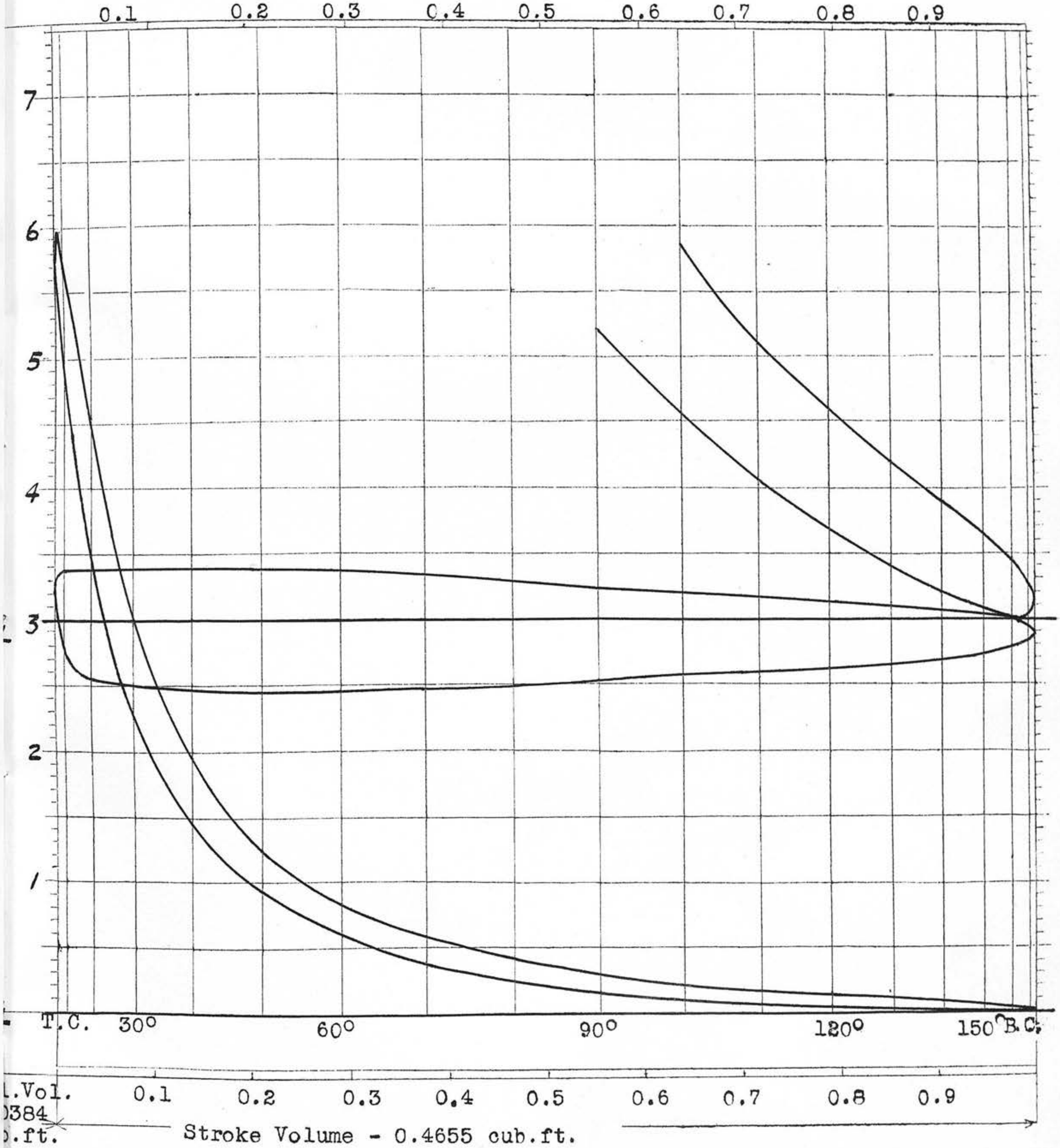
TEST NO. 1.M. p_m (gross) 20.9 lb./sq.in.

p_m (nett) 15.6 lb./sq.in.

FIGURE NO. 5.

OIL ENGINE. Bore 8". Stroke 16". Compr. Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



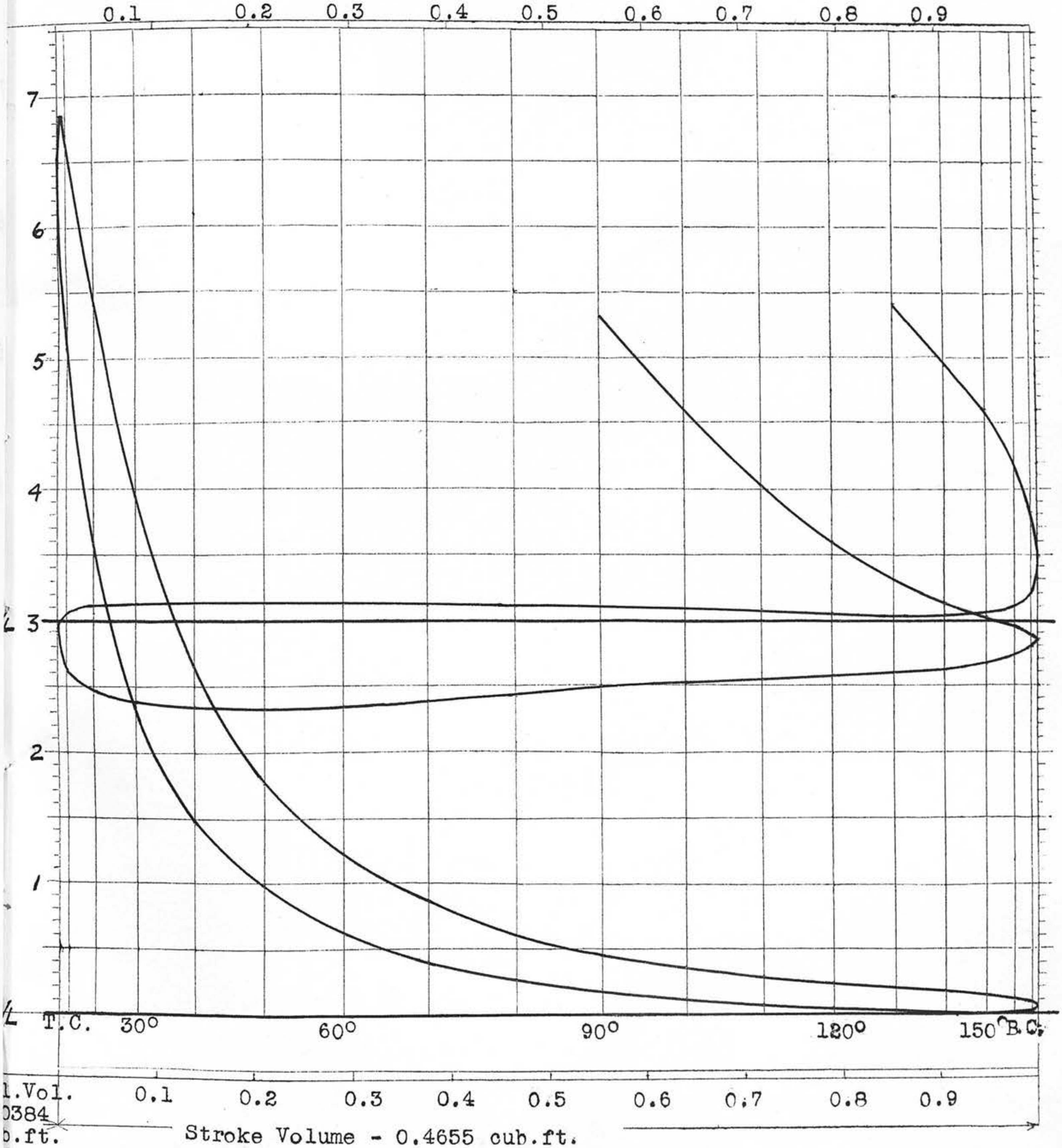
TEST NO. 1.H. p_m (gross) 19.25 lb./sq.in.

p_m (nett) 14.5 lb./sq.in.

FIGURE NO.6.

OIL ENGINE. Bore 8". Stroke 16". Compr. Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



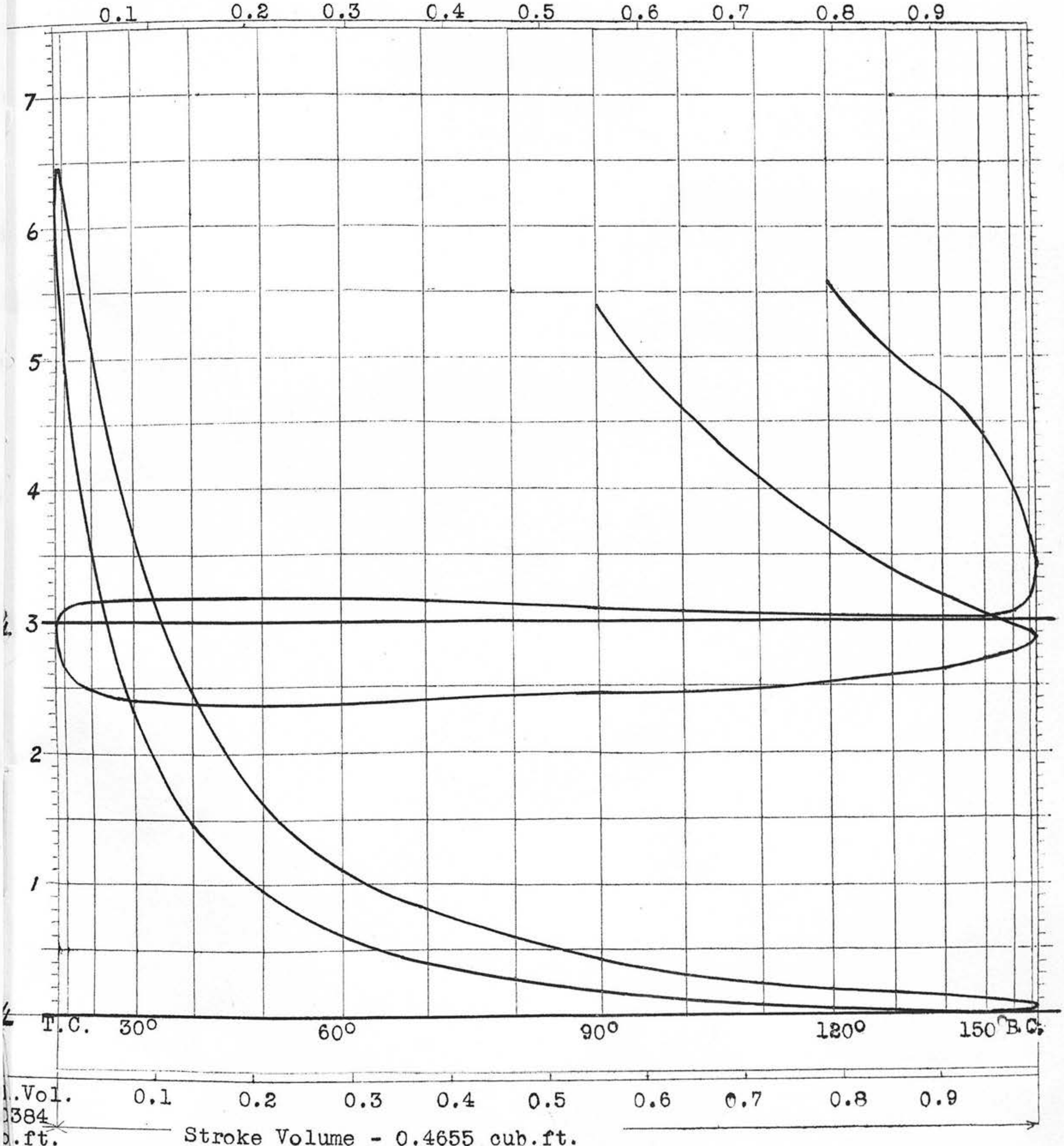
TEST NO. 2.C. p_m (gross) 41.8 lb./sq.in.

p_m (nett) 37.6 lb./sq.in.

FIGURE NO.7.

OIL ENGINE. Bore 8". Stroke 16". Compr. Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



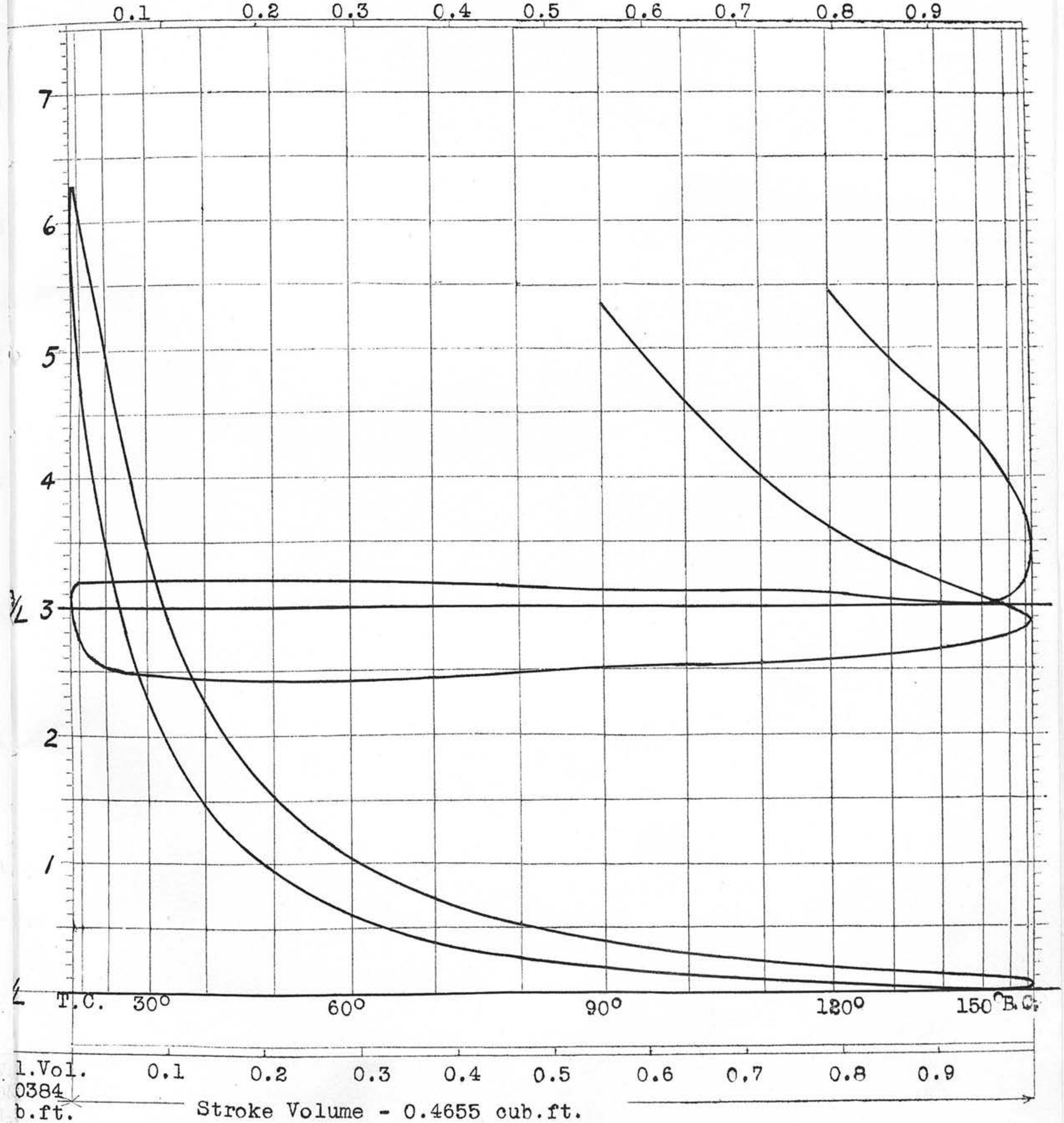
TEST NO. 2.M. p_m (gross) 35.5 lb./sq.in.

p_m (nett) 31.2 lb./sq.in.

FIGURE NO.8.

OIL ENGINE. Bore 8". Stroke 16". Compr.Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



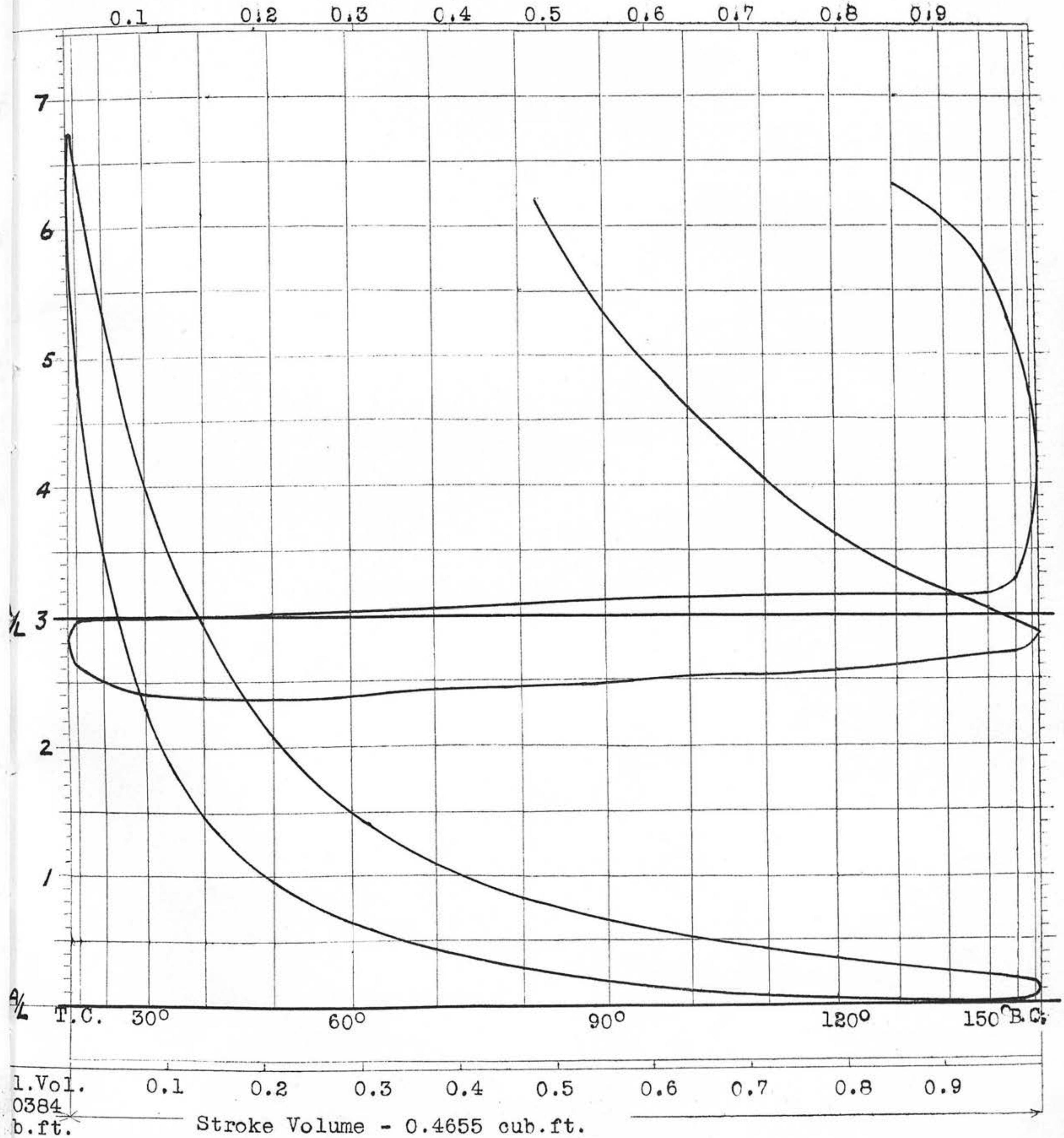
TEST NO. 2.H. p_m (gross) 31.7 lb./sq.in.

p_m (nett) 27.5 lb./sq.in.

FIGURE NO.9.

OIL ENGINE. Bore 8". Stroke 16". Compr.Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



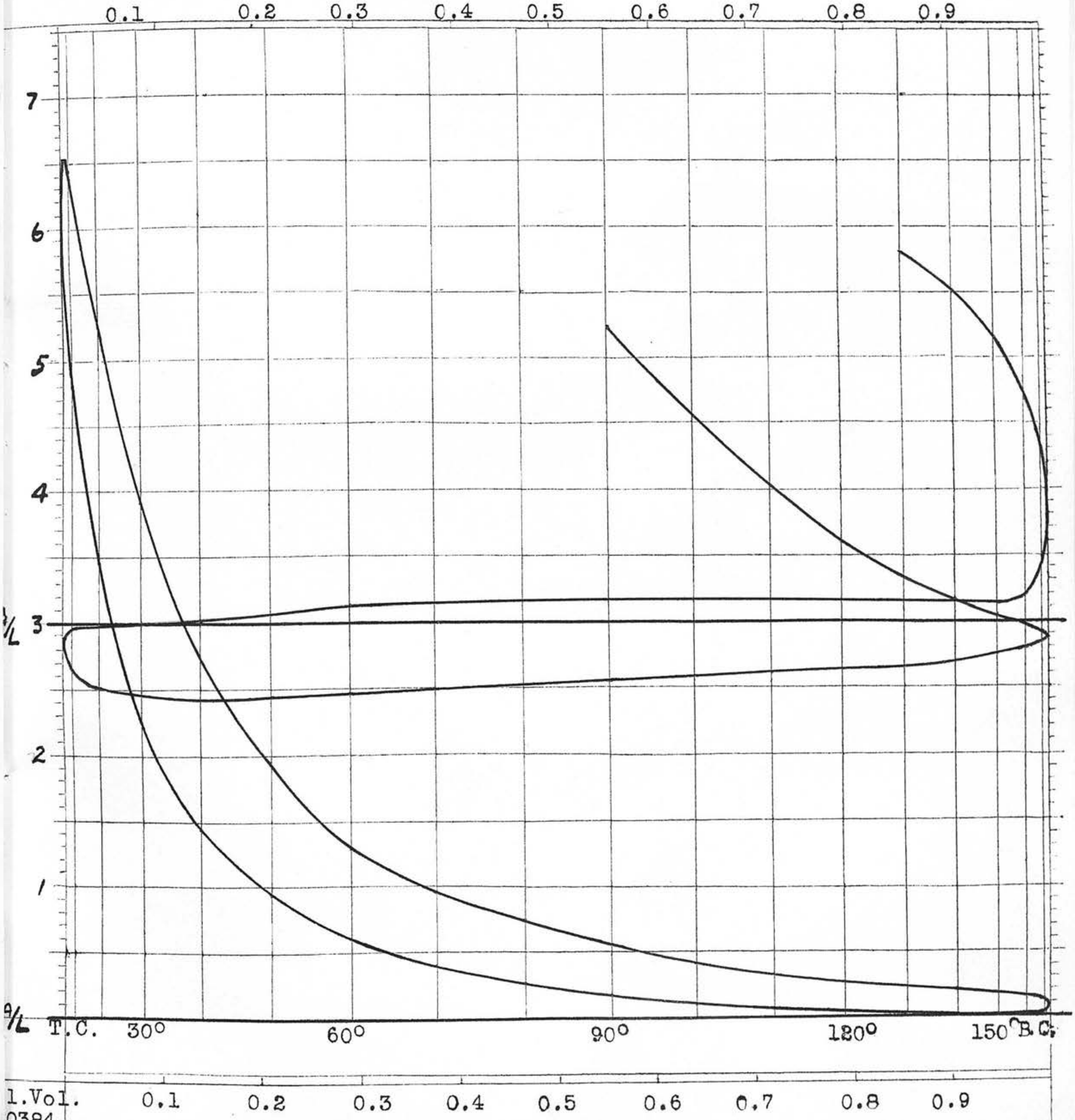
TEST NO. 3.C. p_m (gross) 56.5 lb./sq.in.

p_m (nett) 52.5 lb./sq.in.

FIGURE NO. 10.

OIL ENGINE. Bore 8". Stroke 16". Compr. Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



Cl. Vol. 0.0384 cub.ft.

Stroke Volume - 0.4655 cub.ft.

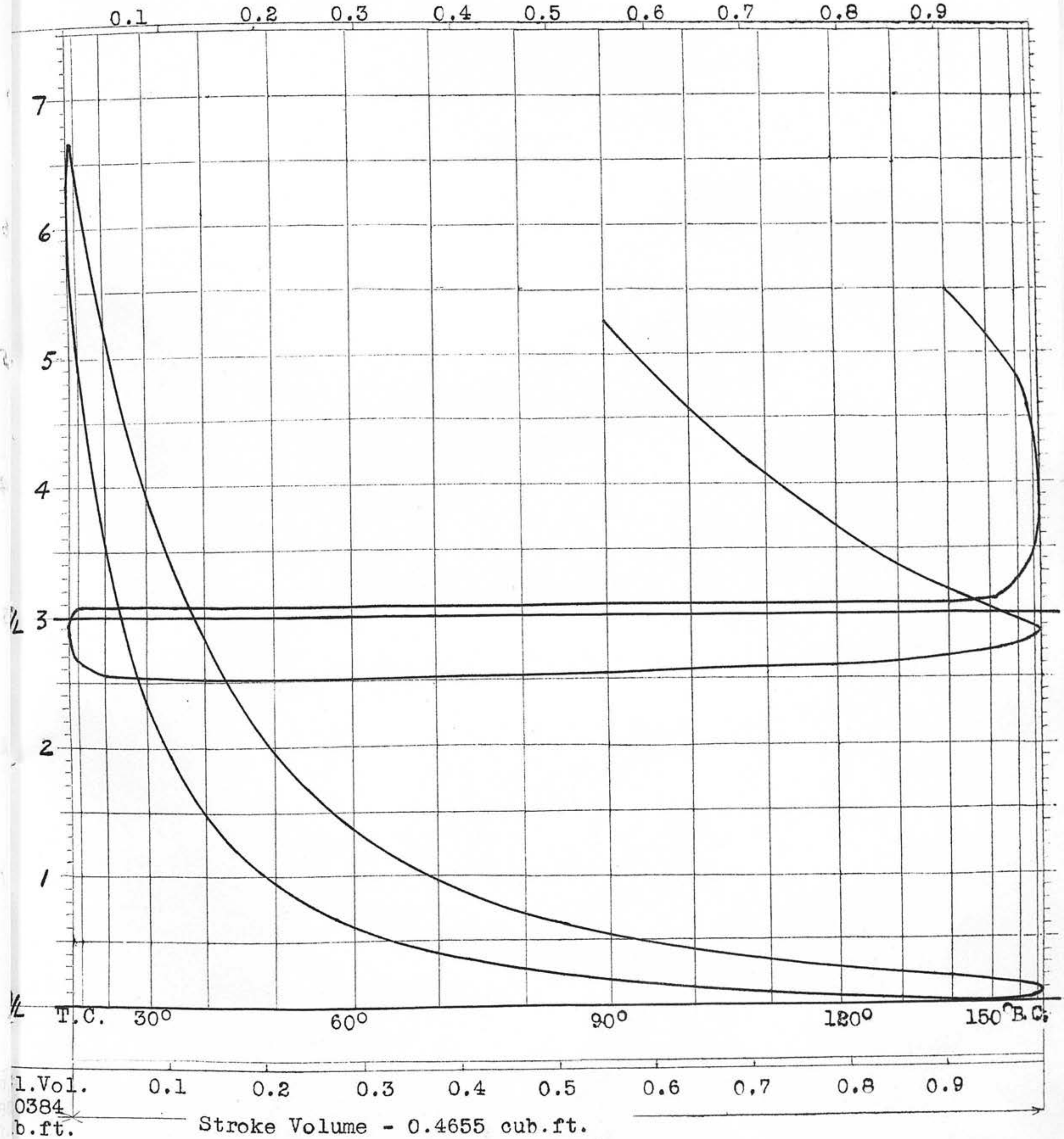
TEST NO. 3.M. p_m (gross) 48.4 lb./sq.in.

p_m (nett 44.7 lb./sq.in.

FIGURE NO.11.

OIL ENGINE. Bore 8". Stroke 16". Compr.Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



Stroke Volume - 0.4655 cub.ft.

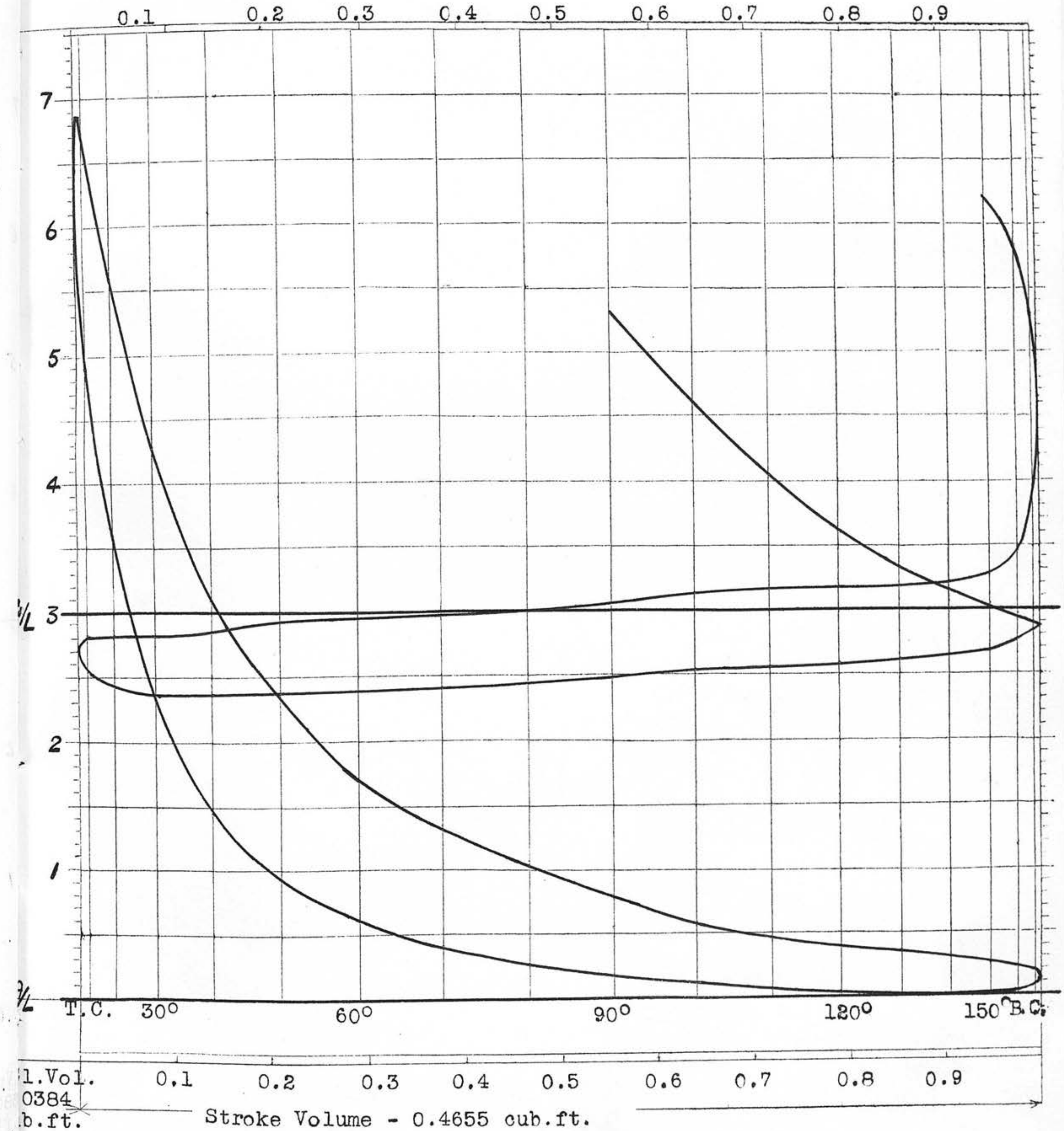
TEST NO. 3.H. p_m (gross) 47.8 lb./sq.in.

p_m (nett) 43.3 lb./sq.in.

FIGURE NO.12.

OIL ENGINE. Bore 8". Stroke 16". Compr.Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



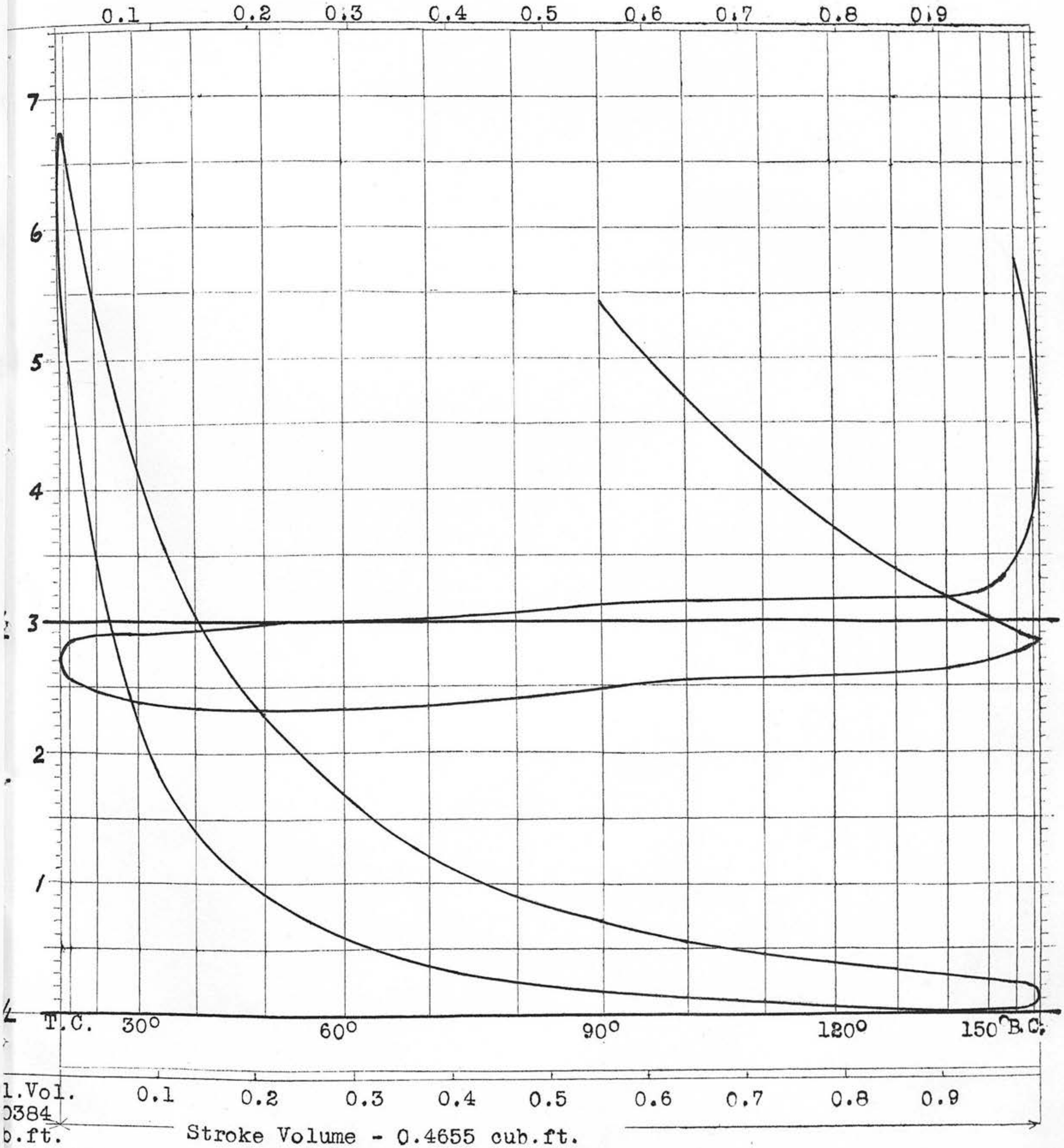
TEST NO. 4.C. p_m (gross) 67.5 lb./sq.in.

p_m (nett) 63.9 lb./sq.in.

FIGURE NO.13.

OIL ENGINE. Bore 8". Stroke 16". Compr.Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



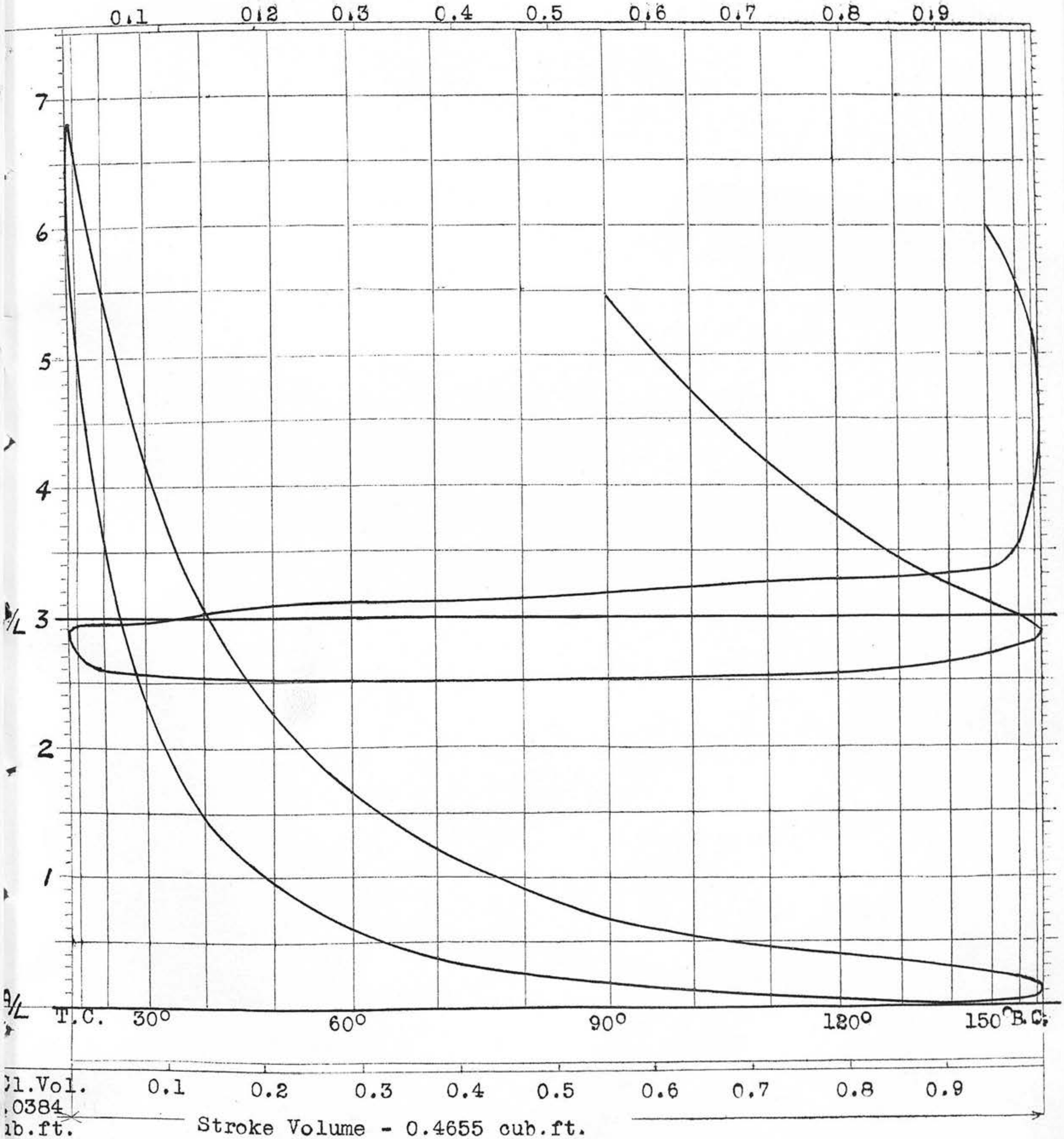
TEST NO. 4.M. p_m (gross) 64.1 lb./sq.in.

p_m (nett) 60.1 lb./sq.in.

FIGURE NO.14.

OIL ENGINE. Bore 8". Stroke 16". Compr.Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



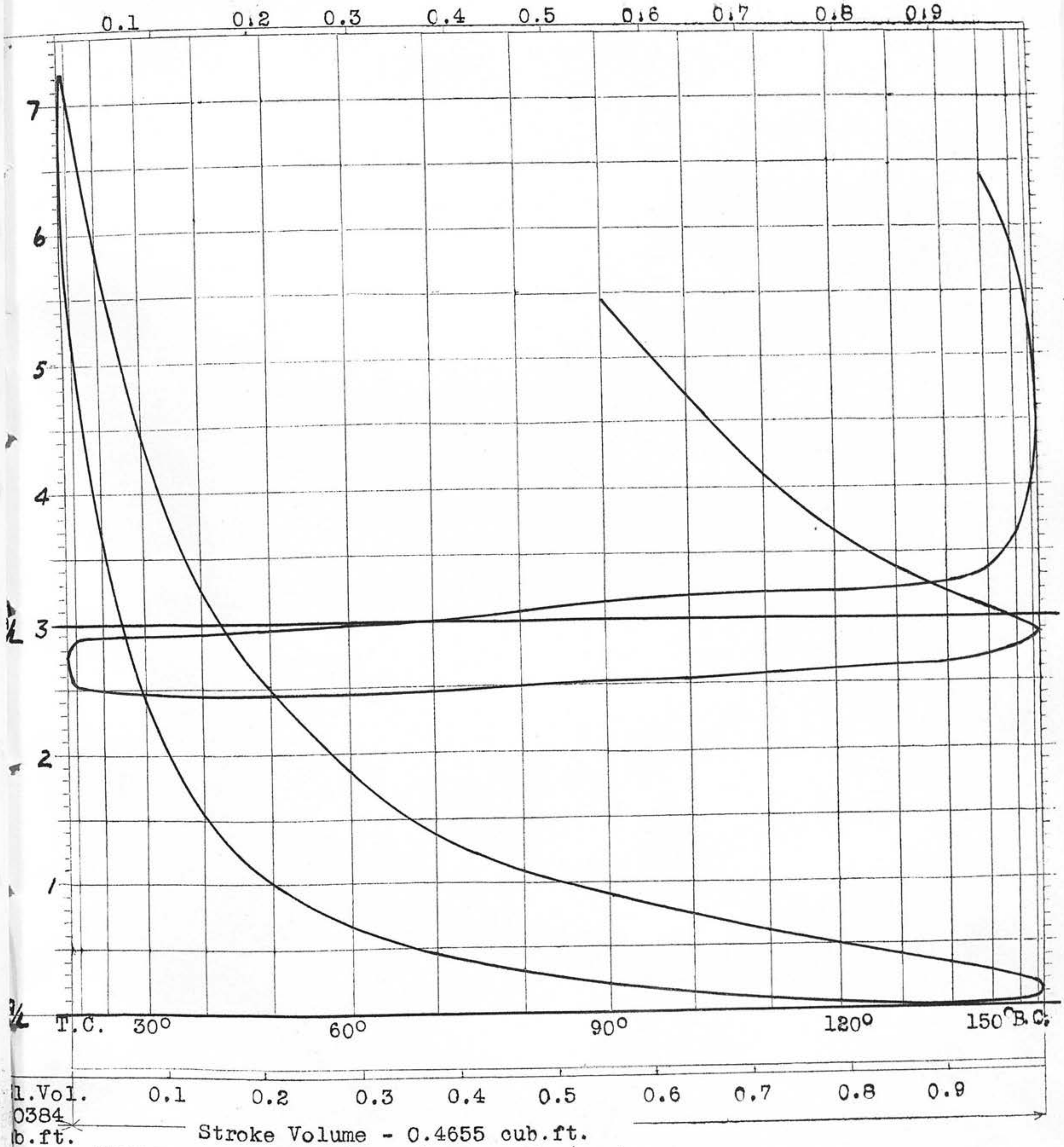
TEST NO. 4.H. p_m (gross) 62.5 lb./sq.in.

p_m (nett) 58.6 lb./sq.in.

FIGURE NO.15.

OIL ENGINE. Bore 8". Stroke 16". Compr. Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



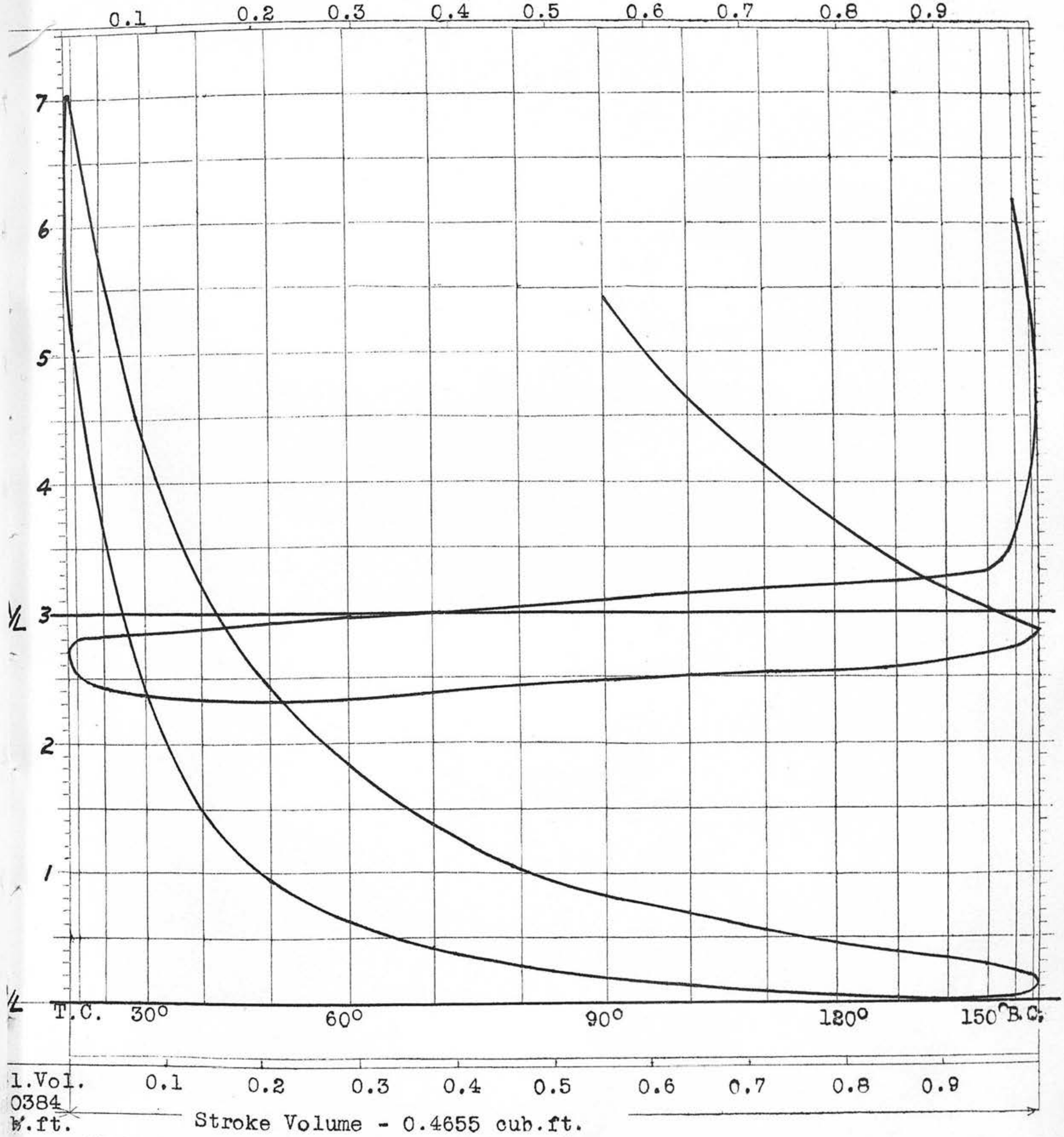
TEST NO. 5.C. p_m (gross) 72.8 lb./sq.in.

p_m (nett) 69.1 lb./sq.in.

FIGURE NO.16.

OIL ENGINE. Bore 8". Stroke 16". Compr.Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



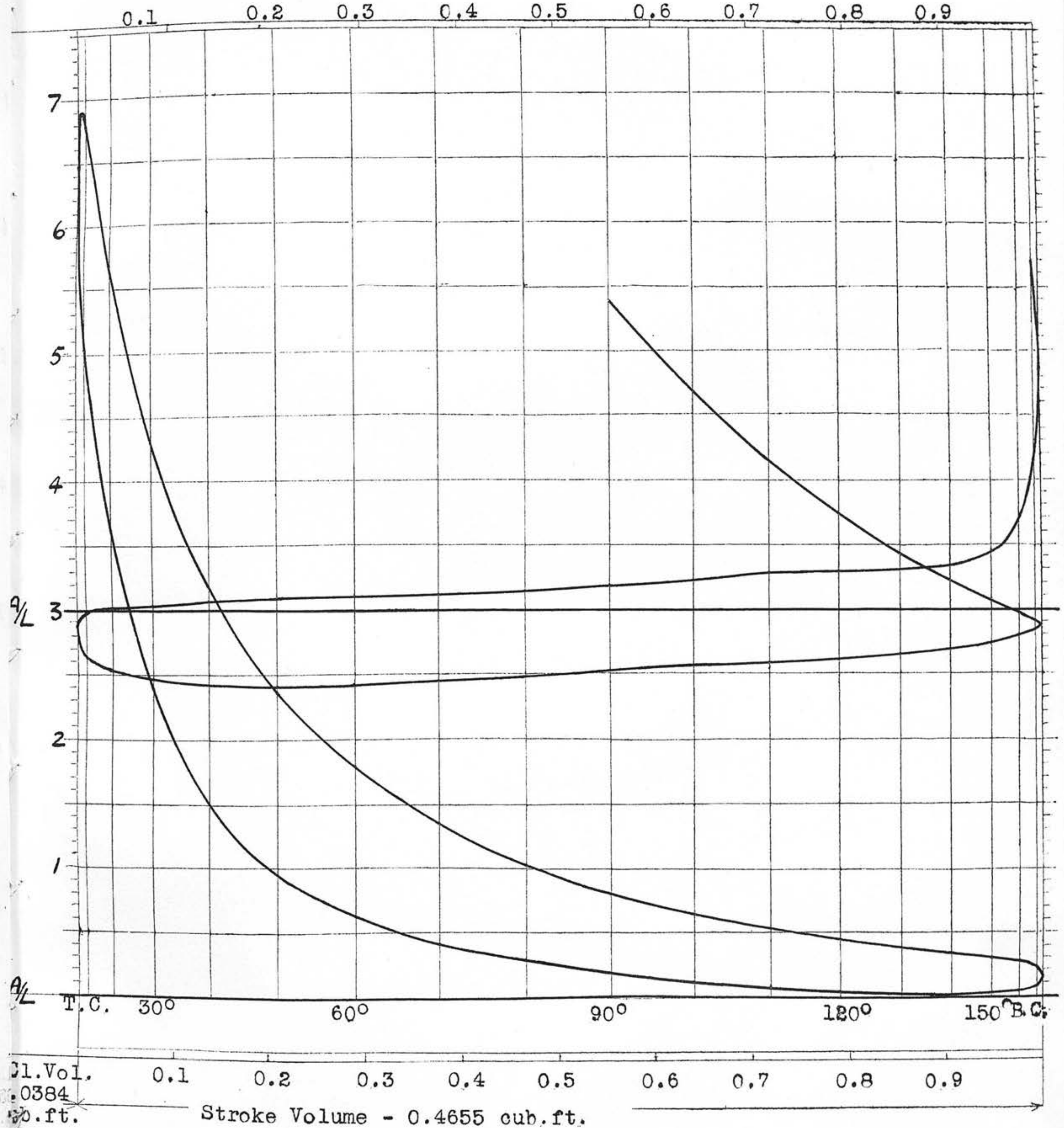
TEST NO. 5.M. p_m (gross) 70.3 lb./sq.in.

p_m (nett) 66.3 lb./sq.in.

FIGURE NO.17.

OIL ENGINE. Bore 8". Stroke 16". Compr. Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



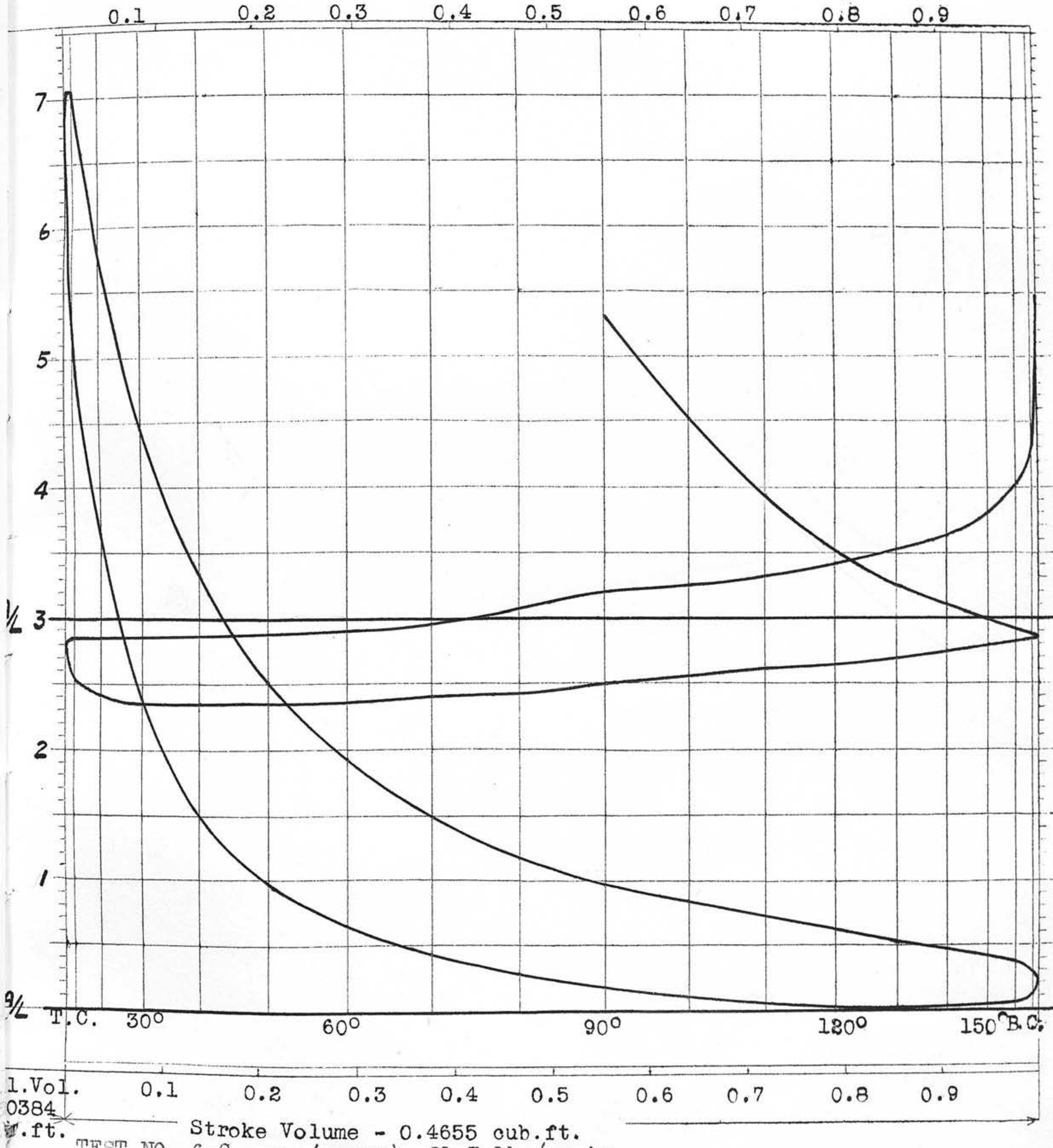
TEST NO.5.H. p_m (gross) 70.0 lb./sq.in.

p_m (nett) 65.8 lb./sq.in.

FIGURE NO.18.

OIL ENGINE. Bore 8". Stroke 16". Compr. Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



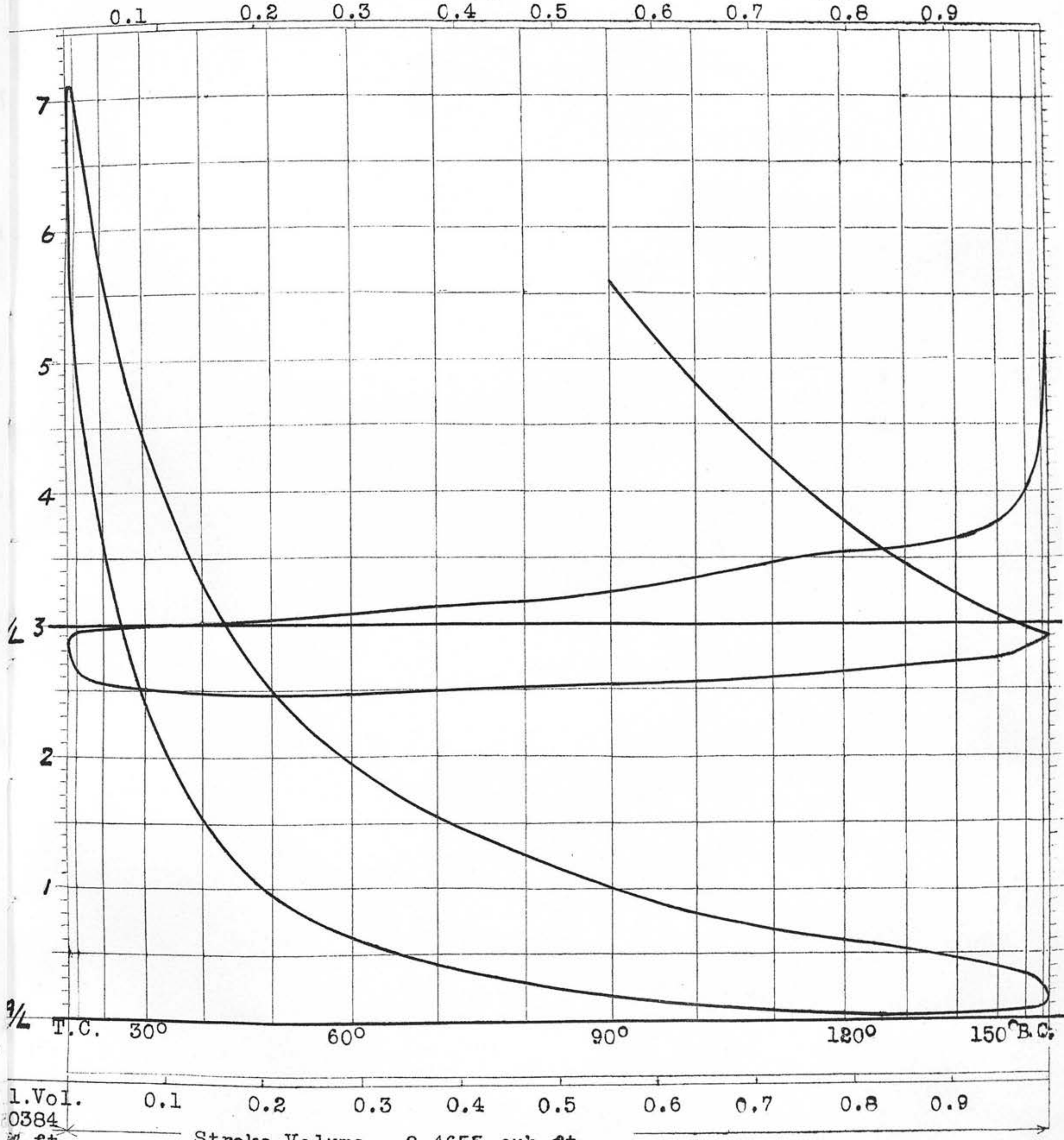
TEST NO. 6.C. p_m (gross) 81.7 lb./sq.in.

p_m (nett) 77.8 lb./sq.in.

FIGURE NO.19.

OIL ENGINE. Bore 8". Stroke 16". Compr.Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



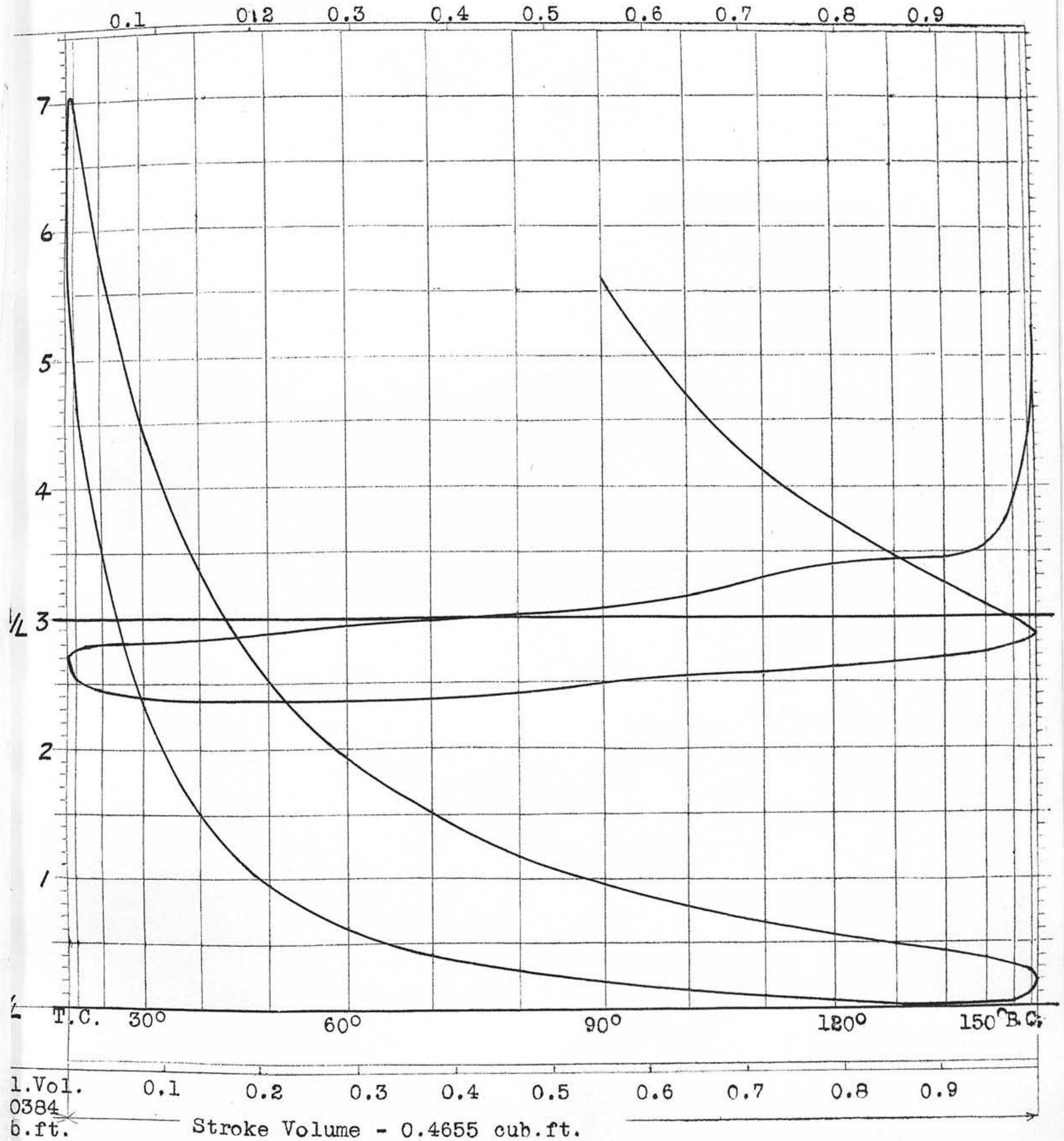
Test No. 6.M. p_m (gross) 81.0 lb./sq.in.

p_m (nett) 76.6 lb./sq.in.

FIGURE NO.20.

OIL ENGINE. Bore 8". Stroke 16". Compr. Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



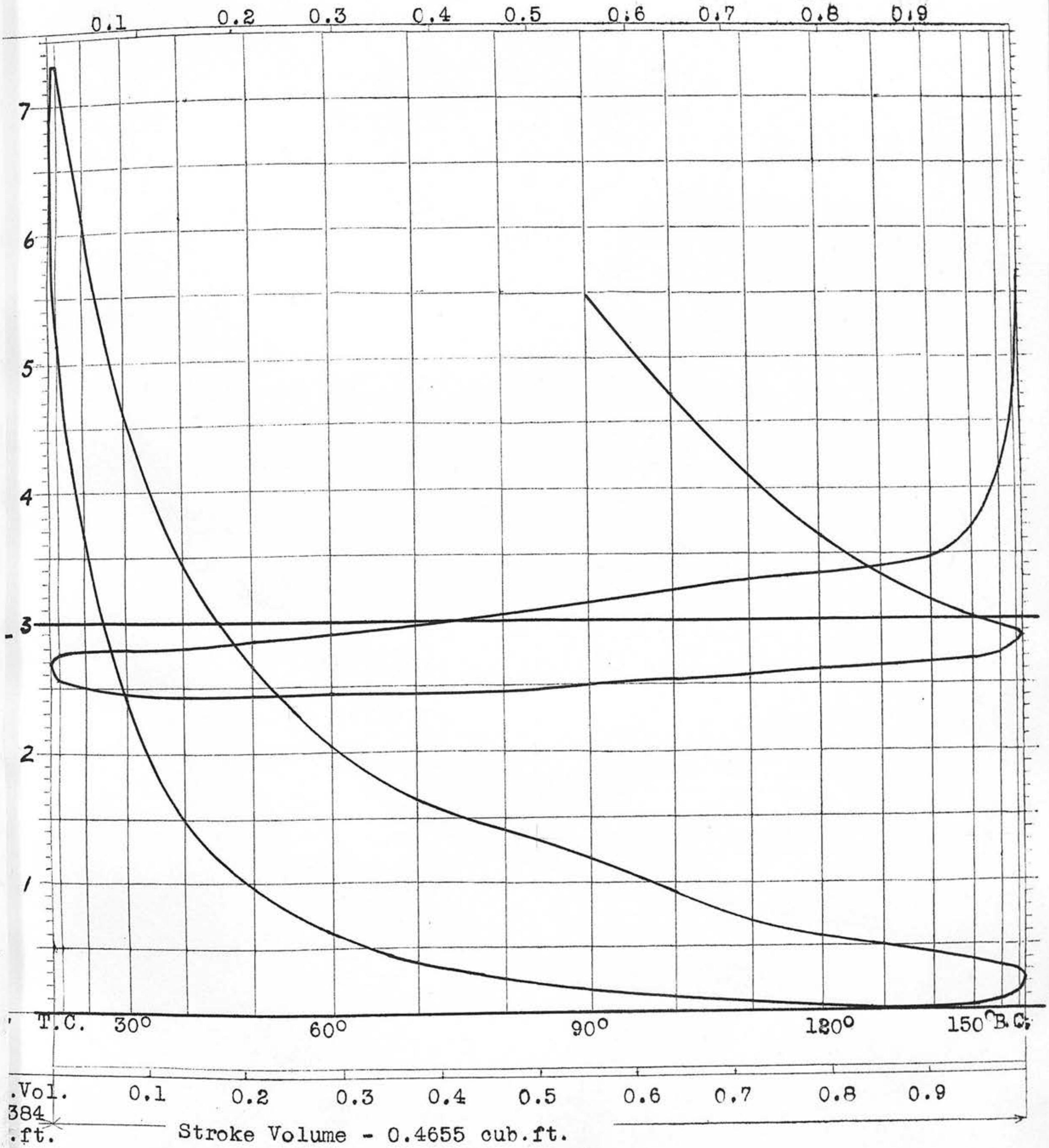
TEST NO. 6.H. p_m (gross) 79.5 lb./sq.in.

p_m (nett) 75.6 lb./sq.in.

FIGURE NO.21.

OIL ENGINE. Bore 8". Stroke 16". Compr.Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



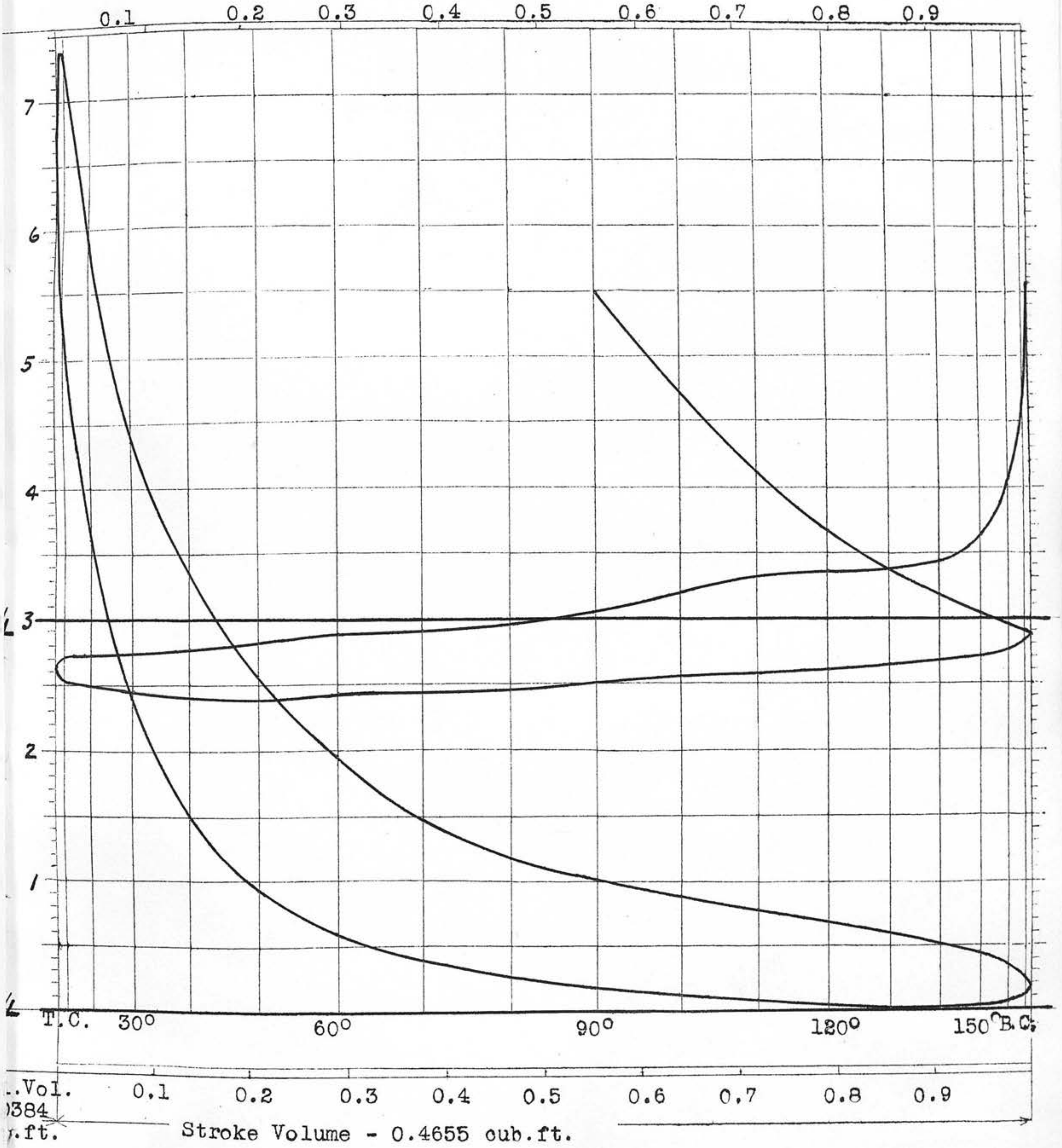
TEST NO. 7.C. p_m (gross) 87.5 lb./sq.in.

p_m (nett) 83.9 lb./sq.in.

FIGURE NO.22.

OIL ENGINE. Bore 8". Stroke 16". Compr. Ratio $\frac{13.11}{1.0}$.

SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



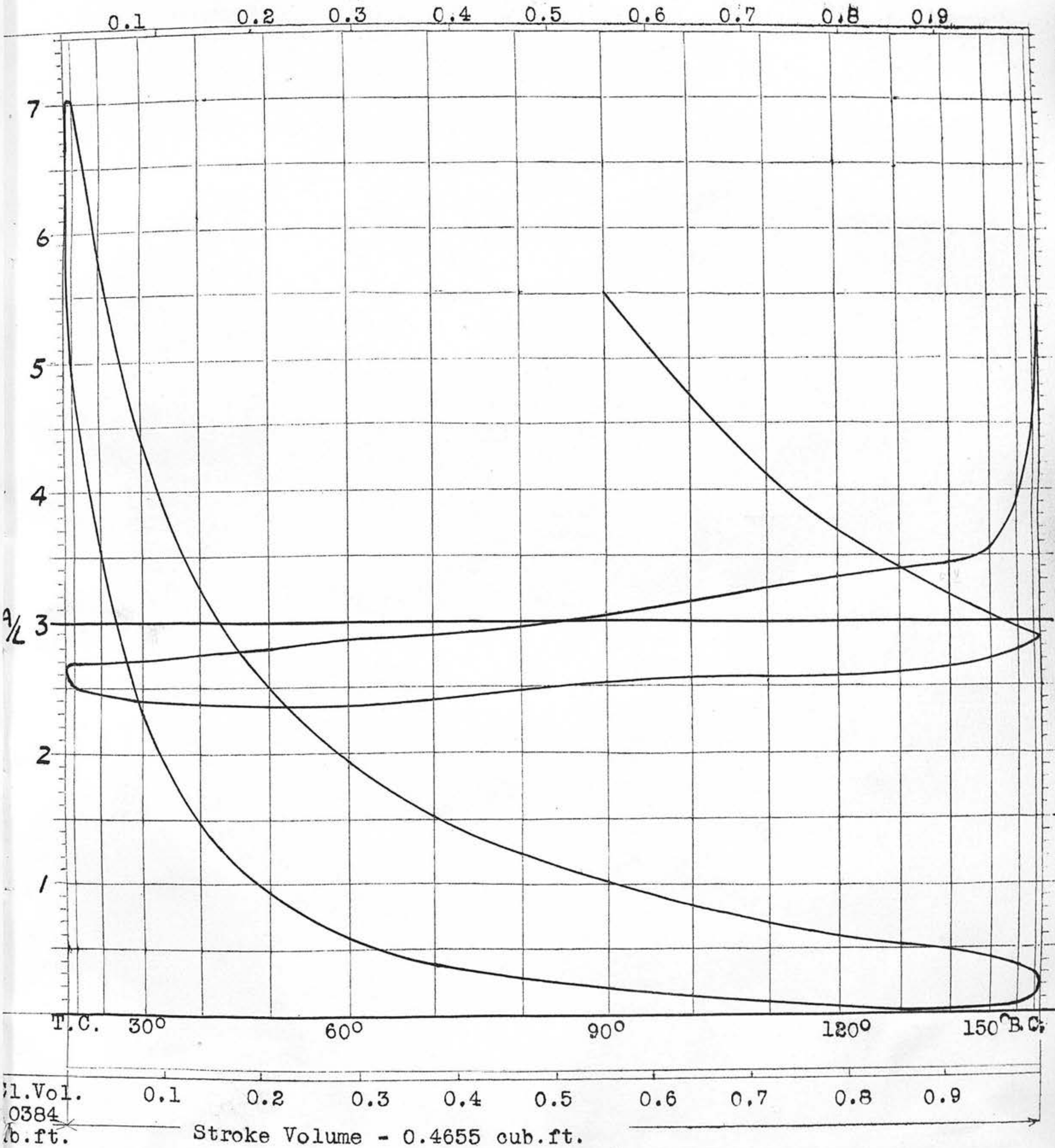
TEST NO. 7.M. p_m (gross) 85.1 lb./sq.in.

p_m (nett) 81.6 lb./sq.in.

FIGURE NO. 23.

OIL ENGINE. Bore 8". Stroke 16". Compr. Ratio $\frac{13.11}{1.0}$.

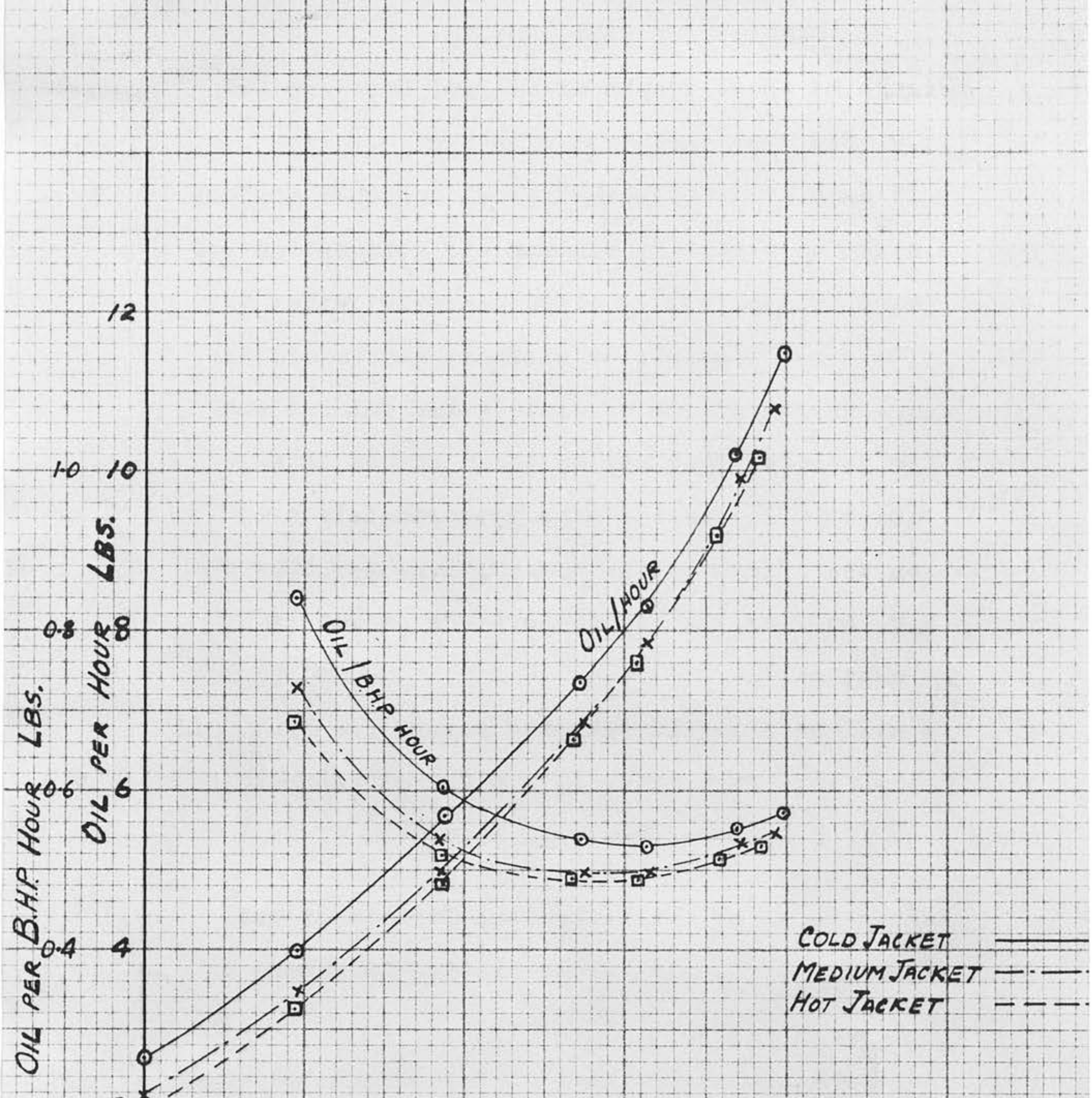
SPRINGS:- LOAD DIAGRAM $\frac{1}{80}$: LIGHT SPRING DIAGRAM $\frac{1}{6.9}$



TEST NO. 7.H. p_m (gross) 83.4 lb./sq.in.

p_m (nett) 79.9 lb./sq.in.

FIG. Nº 24.



OIL PER B.H.P. HOUR LBS.
OIL PER HOUR LBS.
12
1·0 10
0·8 8
0·6 6
0·4 4
0·2 2
0 0
5
10
15
20
B.H.P.
OIL / HOUR
OIL / B.H.P. HOUR
COLD JACKET
MEDIUM JACKET
HOT JACKET

COMMENTS ON TEST RESULTS.

Figure 24 shows graphs of Oil per hour and Oil per B.H.P. per hour to a base of B.H.P.

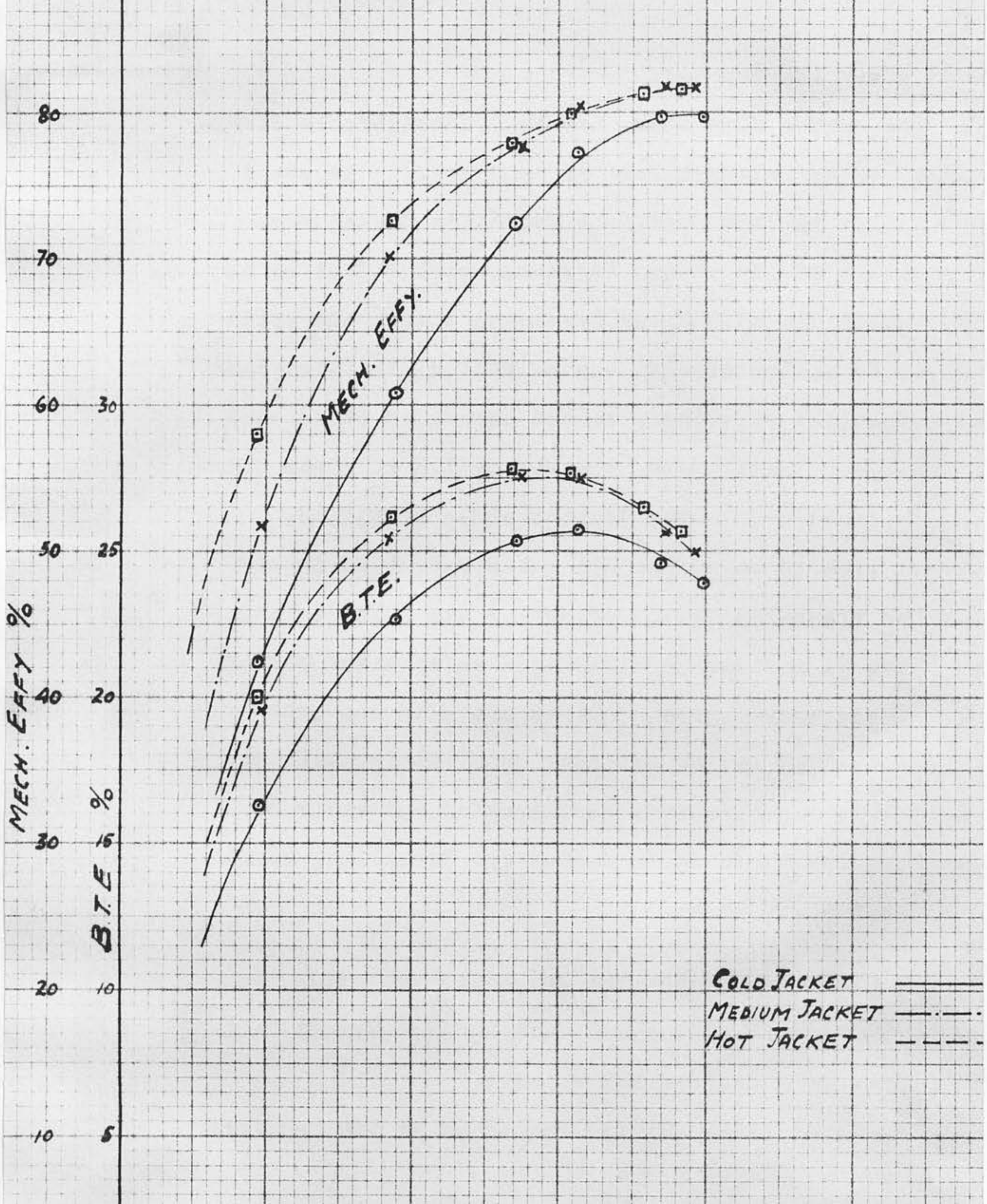
Both sets of graphs are typical for such engines and as might be expected, minimum oil consumption per B.H.P. hour (0.492 lb.) is given when the jacket is hot. It is not possible to estimate accurately the mean temperatures of the water jackets, but for comparative purposes they may be taken say as 16°C., 40°C., and 61°C. Despite this approximately even variation the effect on the consumptions per hour and per B.H.P. hour of the low temperature jacket is by far the most marked.

It will be observed that the powers for least oil consumption vary from about 14 B.H.P. for the hot jacket to 16 B.H.P. for the cold jacket.

Figure 25 shows the Mechanical Efficiency and Brake Thermal Efficiency values plotted to a base of B.H.P.

The Mechanical Efficiency curves follow the usual form of increasing to a maximum with a very slight drop in the case of the cold jacket, when the normal rated /

FIG. Nº 25.


MECH. EFFY.
B.T.E.
MECH. EFFY %
B.T.E %
B.H.P.
0
5
10
15
20
10
20
30
40
50
60
70
80
5
10
15
20
25
30
COLD JACKET
MEDIUM JACKET
HOT JACKET

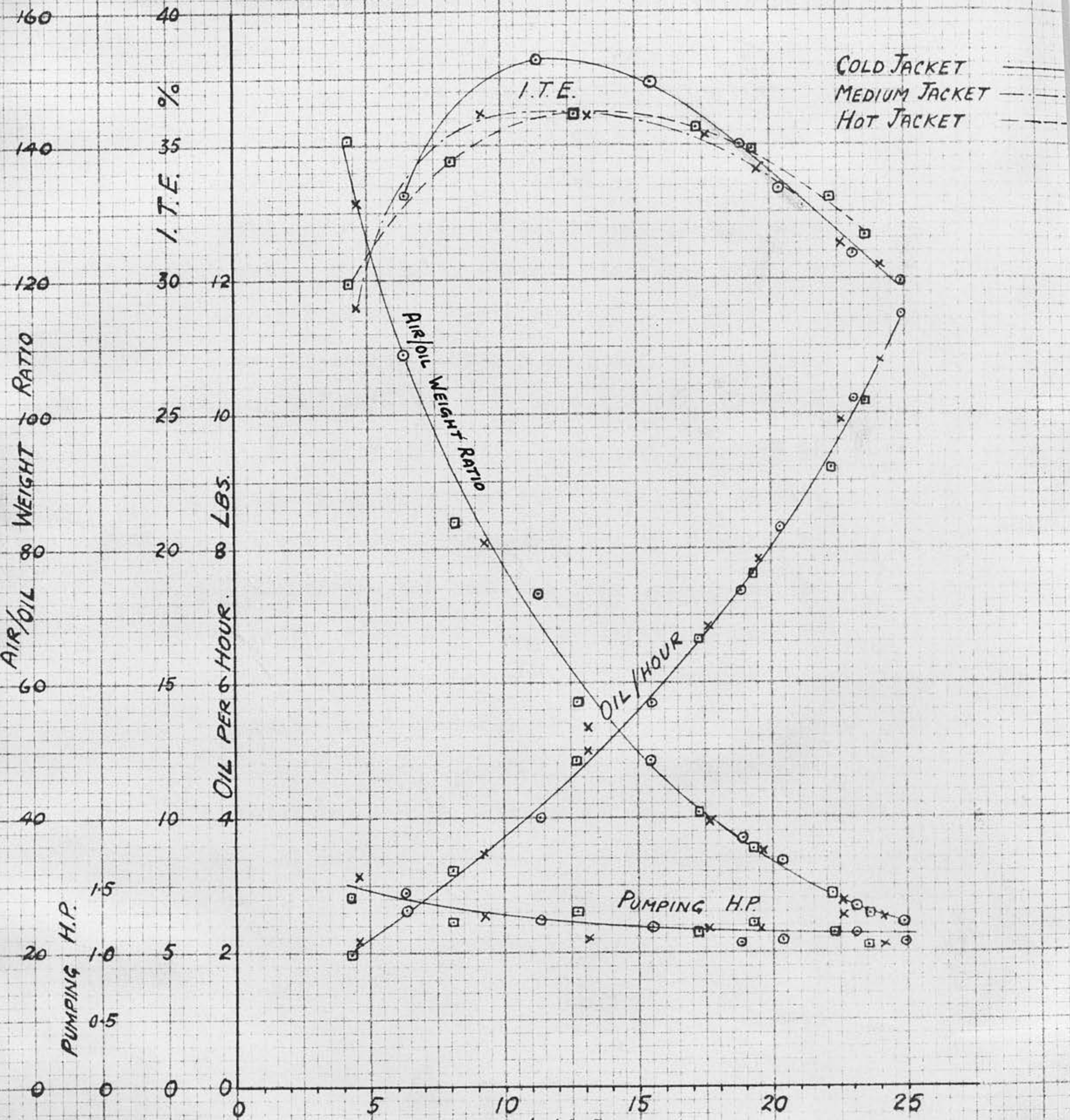
rated power is exceeded.

The curves also show, however, considerable diver-:gence in the neighbourhood of half load. It would appear that in this region, a reduction in the mean temperature of the jacket is accompanied either by an increase in the nett power developed within the cylin-:der or by an increase in the mechanical friction loss. The Mechanical Efficiency values have been calculated on the I.H.P. nett basis, i.e. gross I.H.P. minus pump-:ing or fluid H.P. The Brake Thermal Efficiency curves, like those of the Oil per B.H.P. hour in the previous figure, show the effect of the cold jacket, but there is not such a divergence as with the Mechan-:ical Efficiency curves. An increase in the nett power developed in the cylinder when running with the cold jacket may therefore be suspected as partly res-:ponsible for the drop in Mechanical Efficiency round about half load.

Figure 26 shows the Oil per hour, Indicated Ther-:mal Efficiency, Pumping H.P. and the $\frac{\text{Air}}{\text{Oil}}$ Weight Ratio curves all to a base of I.H.P. nett.

Comparing these curves with those in the previous figure,/

FIG. Nº 26.


COLD JACKET
MEDIUM JACKET
HOT JACKET
I.T.E.
AIR/OIL WEIGHT RATIO
OIL/HOUR
PUMPING H.P.
AIR/OIL WEIGHT RATIO
I.T.E. %
OIL PER HOUR. LBS.
PUMPING H.P.
160
140
120
100
80
60
40
20
0
40
35
30
25
20
15
10
5
0
12
10
8
6
4
2
0
1·5
1·0
0·5
0
0
5
10
15
20
25
I.H.P. (NETT)

figure, the jacket temperature does not seem to have a very marked effect on the oil consumption as far as the nett power developed within the cylinder is con:cerned. The pumping H.P. curve, which is slightly concave upwards, indicates a variation of approximately 0.5 H.P. throughout the tests. It will be seen that there is no appreciable variation due to jacket temp:erature. The Indicated Thermal Efficiency curves, however, do show to a certain extent that between the I.H.P.s of 6 and 18 there is a higher efficiency with the cold jacket, while beyond 18 I.H.P. the hot jacket gives the greater efficiency. The $\frac{\text{Air}}{\text{Oil}}$ Weight Ratio values vary as the I.H.P., and here also no great change is produced by the jacket temperature variation.

It is, of course, realised that the variations in efficiency are not great, and had the scale, which is similar to that to which the Brake Thermal Efficiency was drawn, been reduced, this apparent discrepancy might not have merited much attention.

Two possible reasons for the reduced indicated thermal efficiency and for the hot and medium jackets within the range of powers stated may be, weak air-fuel mixtures, and low volumetric efficiency.

Figure /

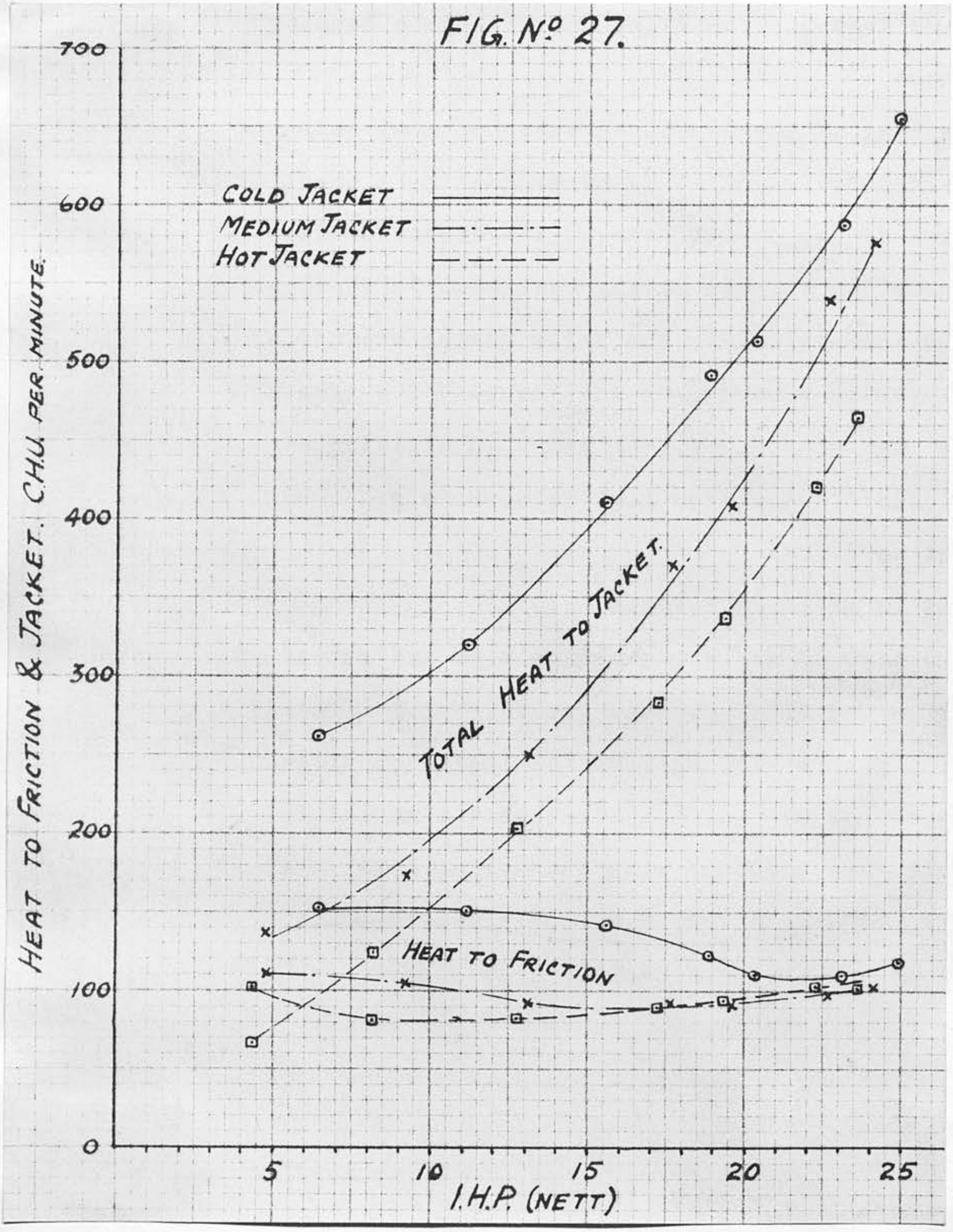


FIG. Nº 27.
COLD JACKET
MEDIUM JACKET
HOT JACKET
HEAT TO FRICTION & JACKET. C.H.U. PER MINUTE.
TOTAL HEAT TO JACKET.
HEAT TO FRICTION
I.H.P. (NETT)
700
600
500
400
300
200
100
0
5
10
15
20
25

Figure 27 shows the Heat per minute lost to the Jacket and that lost to Friction, both to a base of I.H.P. nett.

how does h
get Frict

The heat to the jacket is taken as that obtained by the simple calculation temperature rise × quantity. This value of the jacket loss is therefore for the whole cycle, no attempt having been made to distinguish between the heat lost to the walls during expansion and that during exhaust, as was the method suggested by Clerk[+] and adopted by Hopkinson[*] in arriving at what he considered a true heat balance, e.g.

Heat to I.H.P.

Heat in gases at release

Heat lost to walls during ignition and expansion.

(Note the above I.H.P. is the gross I.H.P., derived from the positive loop only of the indicator diagram.)

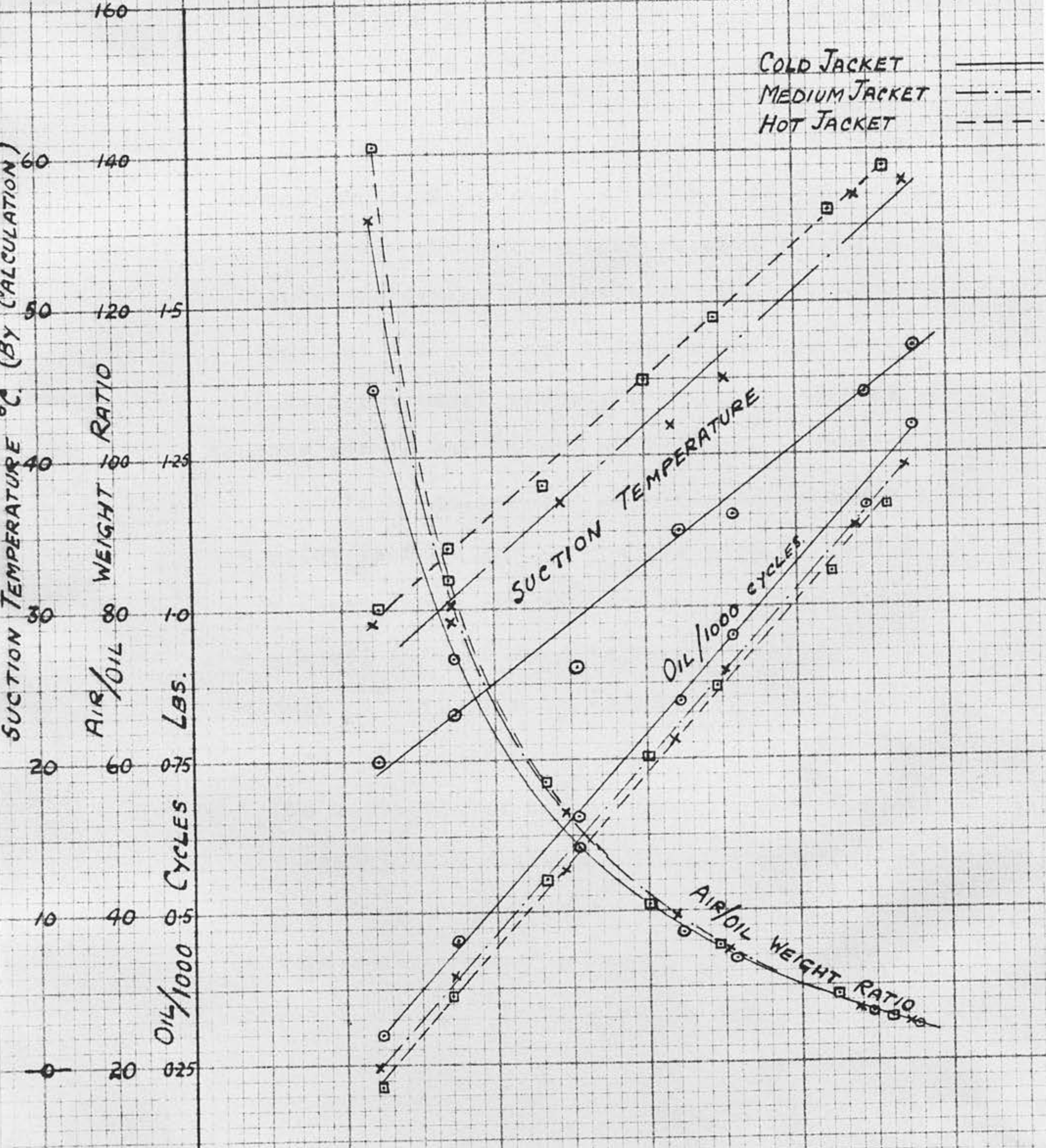
The curves of heat to jacket show a marked in-:crease in this quantity with increasing load. At low loads there is a very considerable difference between the hot and cold jacket quantities.

The frictional loss curves are interesting; drawn as they are to the same scale, for comparison purposes /

+ Proc. Inst.C.E., vol.169, 1907.

* Proc. I.Mech.E., 1908: vol.II, page 427.

FIG. № 28.


COLD JACKET
MEDIUM JACKET
HOT JACKET
SUCTION TEMPERATURE °C. (BY CALCULATION)
AIR/OIL WEIGHT RATIO
OIL/1000 CYCLES LBS.
SUCTION TEMPERATURE
OIL/1000 CYCLES
AIR/OIL WEIGHT RATIO
EXHAUST TEMPERATURE °C.
0
10
20
30
40
50
60
0
20
40
60
80
100
120
140
160
0·25
0·5
0·75
1·0
1·25
1·5
0
100
200
300
400
500

purposes, as the jacket water loss. They are repro:duced again on Figure 38 to a base of B.H.P., which perhaps is more appropriate.

Figure 28 shows the Oil Consumption per 1000 cycles, the $\frac{\text{Air}}{\text{Oil}}$ Weight Ratio, and the Suction Temper:ature, all to a base of Exhaust Temperature.

No provision was made for the recording of gas temperatures at the exhaust valve, and the values plotted as exhaust temperature are those measured by the thermocouple situated in the centre of the exhaust pipe as close as possible to the engine. It is realised that in passing through the exhaust valve and before reaching the thermocouple the spent gases will have lost a certain amount of heat causing thereby a temperature drop. The value of this drop in temper:ature would be more or less proportional to that actually recorded for each of the three series of tests, so that as far as the general arrangement of the curves in Figure 28 is concerned, no serious error is involved, and what is read as Exhaust Temperature should strictly speaking be Exhaust Pipe Temperature.

F.R.B.Watson /

F.R.B.Watson in his experiments on Cylinder Temp-:eratures* carried out on a 25 H.P. hot-bulb engine found that the temperature variation between the exhaust valve and a point in the centre of the exhaust pipe close to the engine amounted to from 40°C. to 100°C. over the full load range, the larger value being more or less constant between half and full loads, at which loads the exhaust pipe thermocouple was record-:ing 283 to 568°C.

Considering this range of temperature drop and also the temperature range recorded in the exhaust pipe, during Watson's experiments, namely 139 to 568°C., it is not unreasonable to assume that in the present case, where the recorded temperature range was 121 to 478°C., the temperature drop would vary from about 30 to 90°C. according to the load.

The $\frac{\text{Air}}{\text{Oil}}$ Weight Ratio curves are consistent throughout the range of exhaust temperature.

It may be mentioned that no account was taken of the residuals in the clearance volume when calculating the values of the $\frac{\text{Air}}{\text{Oil}}$ weight ratio.

The /

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* Proc. I.Mech.E., 1928: vol.IV, p.935.

The oil consumption was determined per cycle in order to cancel the effect of the small speed variation and 1000 cycles were adopted as the unit for simplicity. That the oil consumption per 1000 cycles should vary more or less as a straight line function of the exhaust temperature, as shown in Figure 28, would tend to show that the jacket temperature did not seriously affect the process of combustion.

It may be inferred therefore that inefficient combustion was not the reason for the indicated ther:mal efficiencies between the I.H.P.s 6 and 18 with the hot and medium jackets being lower than with the cold jacket.

The curve of estimated suction temperature, also shown in Figure 28, is discussed in the next section.

Further data, most of which is derived directly or indirectly from the load, light spring and fuel diagrams, is given in Table II.

TABLE II /

TABLE II.

Test No.		1.C.	1.M.	1.H.	2.C.	2.M.	2.H.	3.C.
Pumping H.P. (Fluid Loss)		1.465	1.59	1.42	1.25	1.28	1.24	1.18
E.V. begins to open	Above or below Atmos:phere	11.6	9.8	8.3	16.7	14.0	13.2	23.0
At end of expansion stroke		1.3	1.9	0.8	3.5	3.0	2.8	7.5
At end of exhaust stroke		0.2	2.1	1.7	-.4	-.3	0.6	-.9
At end of suction stroke		-.9	-.7	-.7	-.8	-1.0	-.7	-1.0
Press.at end of Exhaust (lb./sq.in.abs.)		14.7	16.9	15.95	14.2	14.23	14.67	13.11
Press.at end of Suction (lb./sq.in.abs.)		13.6	14.1	13.55	13.8	13.53	13.37	13.01
Wt. of air used/cycle (lb.)		.03280	.03270	.03142	.03330	.03192	.03097	.03142
Wt. of residuals in clearance volume (lb.)		.00213	.00246	.00231	.00182	.00183	.00190	.00143
Total weight at end of Suction (lb.)		.03493	.03516	.03373	.03512	.03375	.03287	.03285
Suction Temp. ^{o}C. (by calculation)		20	29	30	23	29	34	26
Volumetric Efficiency (%)		93.2	91.2	91.1	93.7	90.2	91.1	92.2
Max. Fuel Injection Pressure (lb./sq.in.)		1560	1450	1420	1950	1840	1750	2310
Max. Gas Pressure (lb./sq.in.)		512	496	476	548	516	502	536

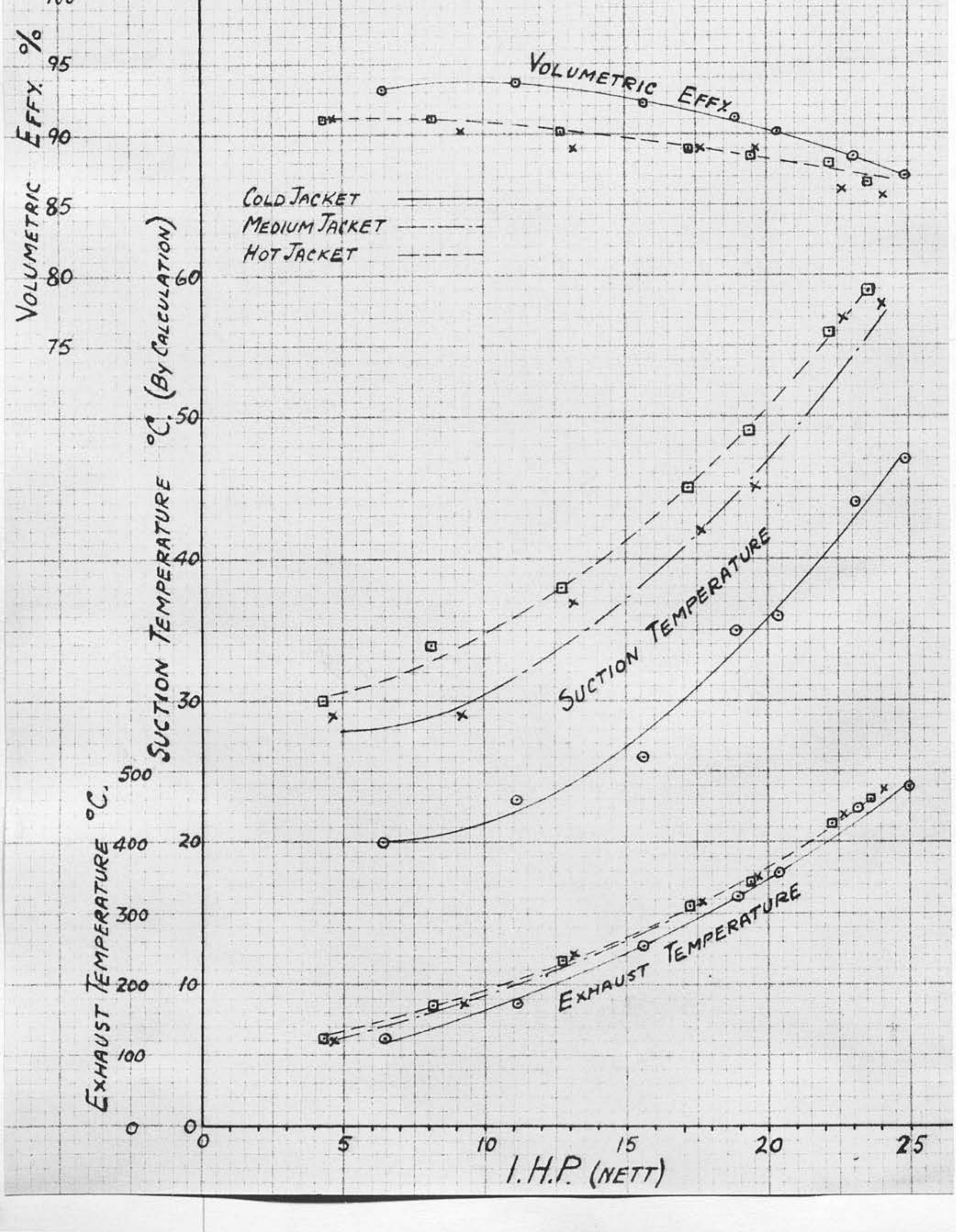
TABLE II (contd)

	Test No.	3.M.	3.H.	4.C.	4.M.	4.H.	5.C.	5.M.
	Pumping H.P. (Fluid Loss)	1.09	1.325	1.065	1.18	1.15	1.09	1.18
Above or below Atmosphere	E.V. begins to open	19.3	18.4	27.2	26.6	28.0	32.8	31.5
	At end of expansion stroke	5.6	6.0	11.3	11.5	11.8	10.9	12.3
	At end of exhaust stroke	-1.0	-.3	-1.9	-1.8	-.7	-1.6	-2.0
	At end of suction stroke	-.9	-.9	-.8	-1.0	-.9	-.9	-.9
	Press.at end of Exhaust (lb./sq.in.abs.)	13.0	13.98	12.2	12.63	13.66	13.03	12.68
	Press.at end of Suction (lb./sq.in.abs.)	13.1	13.38	13.3	13.43	13.46	13.73	13.78
	Wt. of air used/cycle (lb.)	.03030	.03085	.03141	.03100	.03055	.03230	.03155
	Wt. of residuals in clearance volume (lb.)	.00144	.00158	.00118	.00122	.00134	.00119	.00116
	Total Weight at end of Suction (lb.)	.03174	.03243	.03259	.03222	.03189	.03349	.03271
	Suction Temp. °C. (by calculation)	37	38	35	42	45	36	45
	Volumetric Efficiency (%)	89	90.2	91.2	89	89	90.2	89
	Max.Fuel Injection Press. (lb./sq.in.)	2170	2140	2600	2480	2410	2780	2640
	Max.Gas Pressure (lb./sq.in.)	522	530	549	536	544	576	562

TABLE II (contd.)

	Test No.	5.H.	6.C.	6.M.	6.H.	7.C.	7.M.	7.H.
	Pumping H.P. (Fluid Loss)	1.24	1.155	1.30	1.15	1.07	1.03	1.03
bove or elow tmos-here	E.V. begins to open	31.2	40.0	42.0	39.2	43.0	43.0	46.4
	At end of expansion stroke	12.8	17.1	15.2	15.3	18.4	17.8	16.7
	At end of exhaust stroke	-1.0	-1.2	-1.0	-2.0	-2.0	-2.3	-2.6
	At end of suction stroke	-.8	-1.0	-.8	-.8	-.9	-.9	-.8
	Press. at end of Exhaust (lb./sq.in.abs.)	13.58	13.27	13.8	12.44	12.86	12.53	11.83
	Press. at end of Suction (lb./sq.in.abs.)	13.78	13.47	14.0	13.64	13.96	13.93	13.63
	Wt. of air used/cycle (lb.)	.03095	.03100	.03085	.03043	.03195	.03090	.03009
	Wt. of residuals in clearance volume (lb.)	.00125	.00106	.00111	.00102	.00098	.00096	.00093
	Total Weight at end of Suction (lb.)	.03220	.03206	.03196	.03145	.03293	.03186	.03102
	Suction Temp. ^{o}C. (by calculation)	49	44	57	56	47	58	59
	Volumetric Efficiency (%)	88.6	88.6	86	88	87	85.6	86.7
	Max.Fuel Injection Pressure (lb./sq.in.)	2540	2960	2760	2660	3010	2780	2720
	Max. Gas Pressure (lb./sq.in.)	550	565	567	561	584	589	561

FIG. № 29.



Volumetric Effy. %
Suction Temperature °C. (By Calculation)
Exhaust Temperature °C.
Volumetric Effy.
Cold Jacket
Medium Jacket
Hot Jacket
Suction Temperature
Exhaust Temperature
I.H.P. (Nett)
100
95
90
85
80
75
60
50
40
30
20
10
0
500
400
300
200
100
0
5
10
15
20
25

Curves of Exhaust Temperature or Exhaust Pipe Temperature, Suction Temperature and Volumetric Effic- :iency have been plotted to a base of I.H.P. nett, and are shown in Figure 29. The Exhaust Temperatures are represented by a smooth curve.

The temperatures at the end of suction have been calculated from:

a) the weight of residuals in the clearance volume at the end of the exhaust stroke, taking the pressure as that indicated on the light spring diagrams and assum- :ing the temperature to be that recorded in the exhaust pipe;

b) the weight of air taken in per cycle as calculated from that used per minute;

c) the pressure at the end of suction, as indicated on the light spring diagrams.

Objection might be taken in these Suction Temper- :ature calculations to the using of the temperatures recorded from the exhaust pipe as applying to the residuals. In this connection it is interesting to note that by assuming that the correct exhaust temper- :atures vary from 30°C. to 90°C. above those as read, the /

the calculated suction temperatures for the first and last tests are found to be 23°C. and 59°C., that is, the lowest is increased 3° and the highest is unaltered.

A suction temperature frequently assumed for four stroke cycle gas engines is 100°C.

Callendar and Dalby* measured the temperature of the gases in a gas engine during the suction stroke. They found the suction temperature at full load varied from 95°C. to 125°C. according to the mixture strength. For that engine the clearance volume was $\frac{1}{3.68}$ of the stroke volume. The suction temperature is, however, governed chiefly by the proportion of exhaust products left in clearance volume, which mix with and heat the incoming charge, and also by the heat given up to the entering gases by the hot walls of the cylinder and piston.

It is to be expected therefore that with a rela-:tively high compression engine such as the one under discussion, where the exhaust products left in the cylinder represent only a small fraction of the total weight of the charge, their effect can not be apprec-:iable. Even with a high exhaust temperature, conditions /

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* Proc. Royal Society A. vol.80, 1907, p.57.

conditions are not materially altered since, assuming similar pressures, the weight of the exhaust products is correspondingly reduced. The exhaust temperature has, however, an indirect bearing on the suction temperature in so far as its magnitude represents the minimum temperature to which the cylinder walls have previously been exposed. why

In Figure 28 the calculated suction temperature is plotted to a base of exhaust temperature and it will be seen that for any exhaust temperature throughout the range of recorded values, there is a variation in suction temperature of about 10°C. for the hot and cold jackets. The suction temperatures are more or less proportional to the exhaust temperature and for any one type of jacket the increase is about 30°C. between no load and full load. The volumetric efficiency values as plotted on Figure 29 have been taken as the ratio of the volume of air drawn in per cycle at atmospheric pressure and temperature to the stroke volume of the engine. The highest value of this efficiency is 93.7% obtained with the cold jacket when developing about 5 B.H.P. or 11 I.H.P.

All /

All the curves show a reduction in value at the higher loads.

The quantity of fresh charge drawn into the cylinder is quite independent of the heat given to it by the products in the clearance volume provided the specific heats are equal, and any increase of volume of the fresh charge due to heat given up by the pro:ducts is compensated by a corresponding decrease in the volume of the products of combustion. In actual practice, of course, the specific heats are not equal, the divergence being greater where the combustible mixture is rich. The air entering the cylinder receives heat from the cylinder walls, this amount depending on the gas temperatures previously existing and also on the heat flow through the wall to the jacket. This reception of heat by the entering air reduces the specific volume and consequently the volu:metric efficiency and explains the dropping of the curves at high loads as shown in Figure 29.

why

The speed of the entering air past the inlet valve also affects the volumetric efficiency, but for any one engine run at constant speed this factor would be /

be sensibly constant over all loads.

For equal temperatures and piston speeds the mean pressure is dependent almost solely upon the volumetric efficiency and alternatively for a given mean pressure the higher the volumetric efficiency the lower the maximum temperature that can be employed.

Also for similar indicated powers, the indicated thermal efficiency is inversely proportional to the amount of oil used, and, where the $\frac{\text{Air}}{\text{Oil}}$ weight ratio is constant for these powers, then the indicated thermal efficiency is directly proportional to the weight of air used.

Assuming that in the present case the $\frac{\text{Air}}{\text{Oil}}$ weight ratio is sensibly constant for similar I.H.P.s, and this would appear to be the case from Figure 26, and neglecting the varying atmospheric conditions existing throughout the tests, also the relatively small addit:ional weight of fuel, then under these conditions the weight of air used is proportional to the weight of the charge and therefore to the volumetric efficiency, in which case the Indicated Thermal Efficiency is pro:portional to the volumetric efficiency for similar indicated /

indicated powers. While the assumed similar conditions only exist to a close approximation in the present case it would nevertheless appear that the small gain in Indicated Thermal Efficiency between the I.H.P.s 6 and 18 for the cold jacket is due mostly to the high volumetric efficiency.

The light spring diagrams show that as the I.H.P. is increased the pressure at the end of the exhaust stroke is reduced, the variation being fairly consid-:erable. These exhaust pressure variations are primarily due to the inertia of the gases and to the different temperature and pressure conditions existing when the exhaust valve opens. The exhaust pipe was connected to a muffler which had a capacity about twice that of the cylinder, and the length of this portion of the pipe was approximately 7 feet; and, owing to structural conditions, there was a further 20 feet of exhaust pipe beyond the muffler.

It may be therefore that with the high loads there was a substantial lead in the gas velocity over that of the piston due to the inertia of the gas column in the exhaust pipe. This of course would reduce the pressure at the end of exhaust, whereas the opposite effect /

effect would be created at the very light loads where there was no appreciable drop in pressure at the end of expansion, the piston in this case forcing the gases out. The variation in volumetric efficiency through-:out the range of loads is not very great, and while the values obtained for the low loads are what would be expected for a slow running engine, the values at the high loads may have been boosted by the exhaust pipe effect.

It is appreciated that overlap of the air inlet and exhaust valves (25 degrees in the engine under test) may also produce false volumetric efficiency values. These may be either above or below the correct value depending on the pressure at the end of exhaust, since under these conditions a portion of the exhaust may find its way into the air inlet pipe, or as when the pressure is less air may be induced to pass through the cylinder into the exhaust pipe before the valve has closed.

The probability of the latter taking place was foreseen and before the tests observations were taken in order to ascertain the magnitude if such a loss was in effect taking place.

Turning /

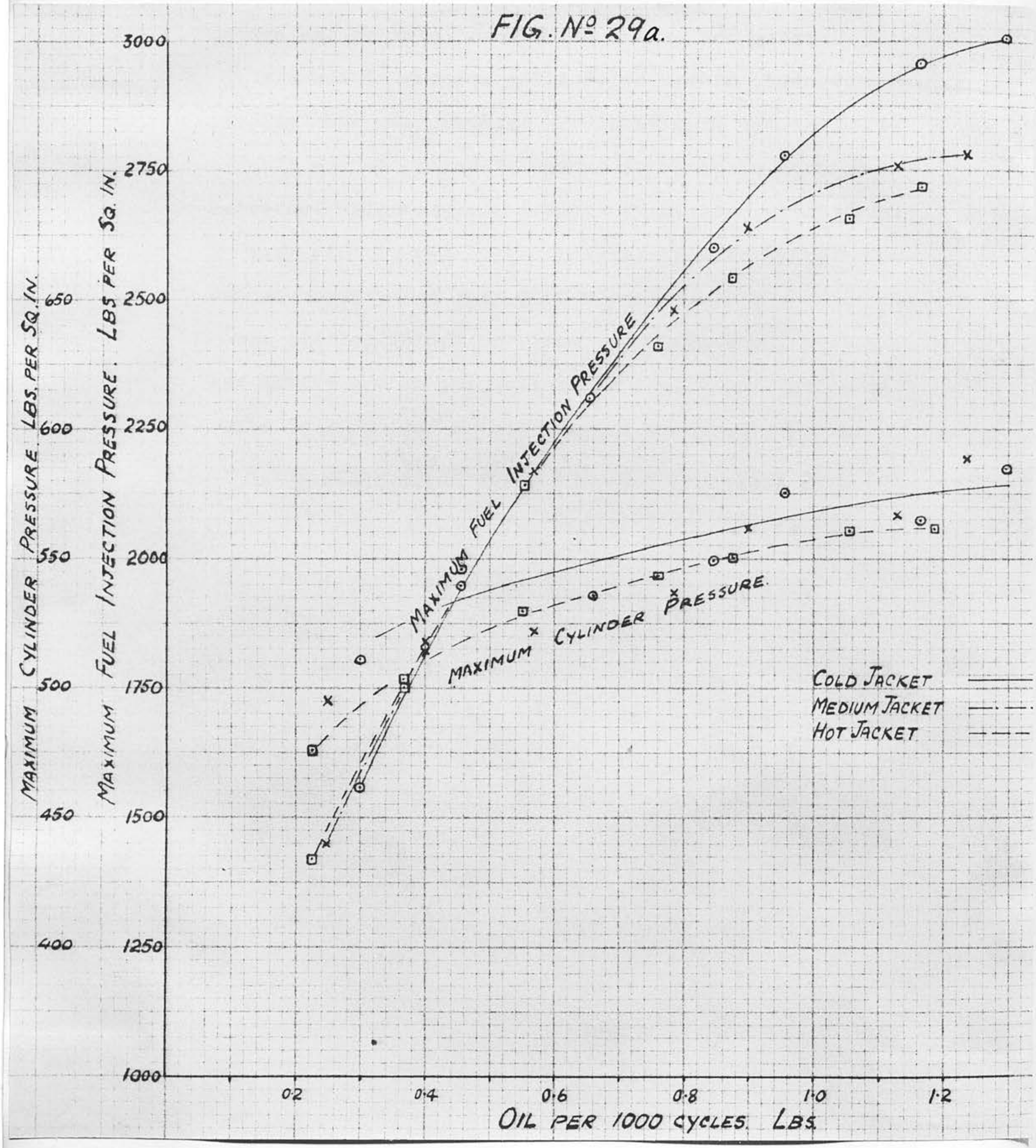


FIG. Nº 29a.
MAXIMUM CYLINDER PRESSURE LBS. PER SQ. IN.
MAXIMUM FUEL INJECTION PRESSURE. LBS PER SQ. IN.
3000
2750
2500
2250
2000
1750
1500
1250
1000
650
600
550
500
450
400
0·2
0·4
0·6
0·8
1·0
1·2
OIL PER 1000 CYCLES. LBS.
MAXIMUM FUEL INJECTION PRESSURE
MAXIMUM CYLINDER PRESSURE.
COLD JACKET
MEDIUM JACKET
HOT JACKET

Turning the engine until that point in the cycle was reached, midway between the opening of the air valve and the closing of the exhaust, air was led, under a pressure of about $2\frac{1}{2}$ inches of water, to the air inlet valve, and in 20 minutes the gasometer indicated a discharge of 2 cubic feet. In view of this very small loss produced under a $2\frac{1}{2}$ inch water pressure and representing about $\frac{1}{2}$ per cent. on the actual air con:sumption, the effect of overlapping of the valves was neglected, though it is appreciated that such an effect would be accentuated by the low exhaust pressures ob- how low :served at the higher loads.

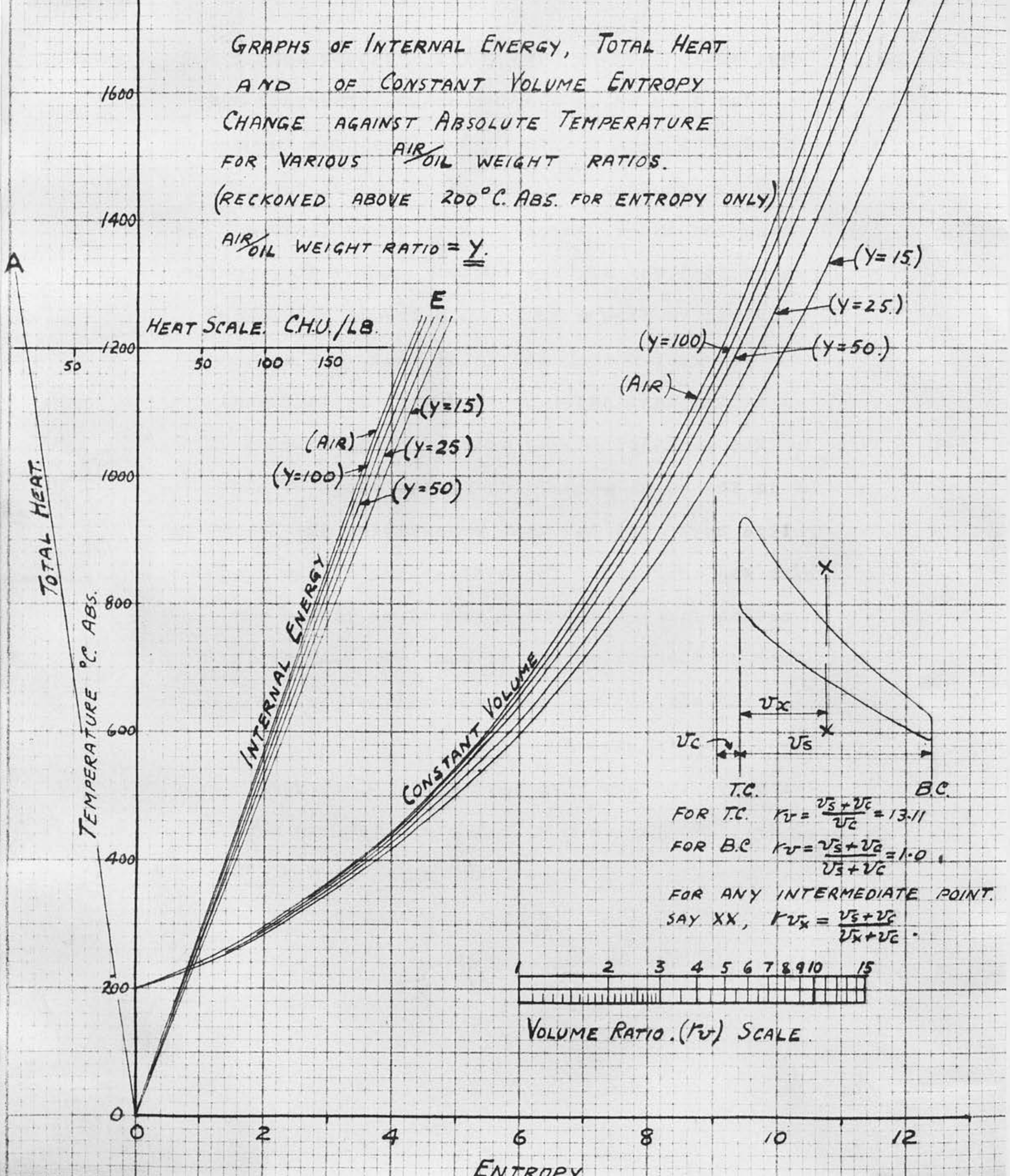
Figure 29A shows the maximum fuel injection pressures plotted to a base of oil used per 1000 cycles. The differential valve, which enabled the fuel diagrams to be taken, was fitted on the fuel pipe line as near to the injector as possible. From the diagrams, which had a pressure scale of 800 lb. per inch, it was not possible to locate with certainty the instant of valve opening and therefore no attempt has been made to measure the combustion time lag and as to how this is effected by gas temperature.

It /

It is difficult to account for the variation in pressure at the higher oil supplies. In an endeavour to throw light on this, the corresponding values of maximum cylinder pressures have been plotted; these however are not very regular and therefore no deductions relating to the oil pressure can be made from them. For similar cylinder conditions and similar injection periods the velocity past the nozzle would vary accor:ding to the oil supply. For equal oil quantities, equal injection pressures would therefore be expected under the above conditions, provided the viscosity of the oil did not vary.

For similar cylinder conditions, injection periods and oil supplies, the velocities past the nozzle should be the same. Assuming that there is no change in the viscosity of the oil, the pressures creating these velocities should also be similar. As the injector passed through the jacketed cylinder head there may have been some slight change in viscosity of the oil due to varied heat conduction from the jacket, and this may in part account for the increased pressures. On the other hand, average cylinder pressures during the period of injection, instead of maximum cylinder pressures /

FIG. Nº 30.

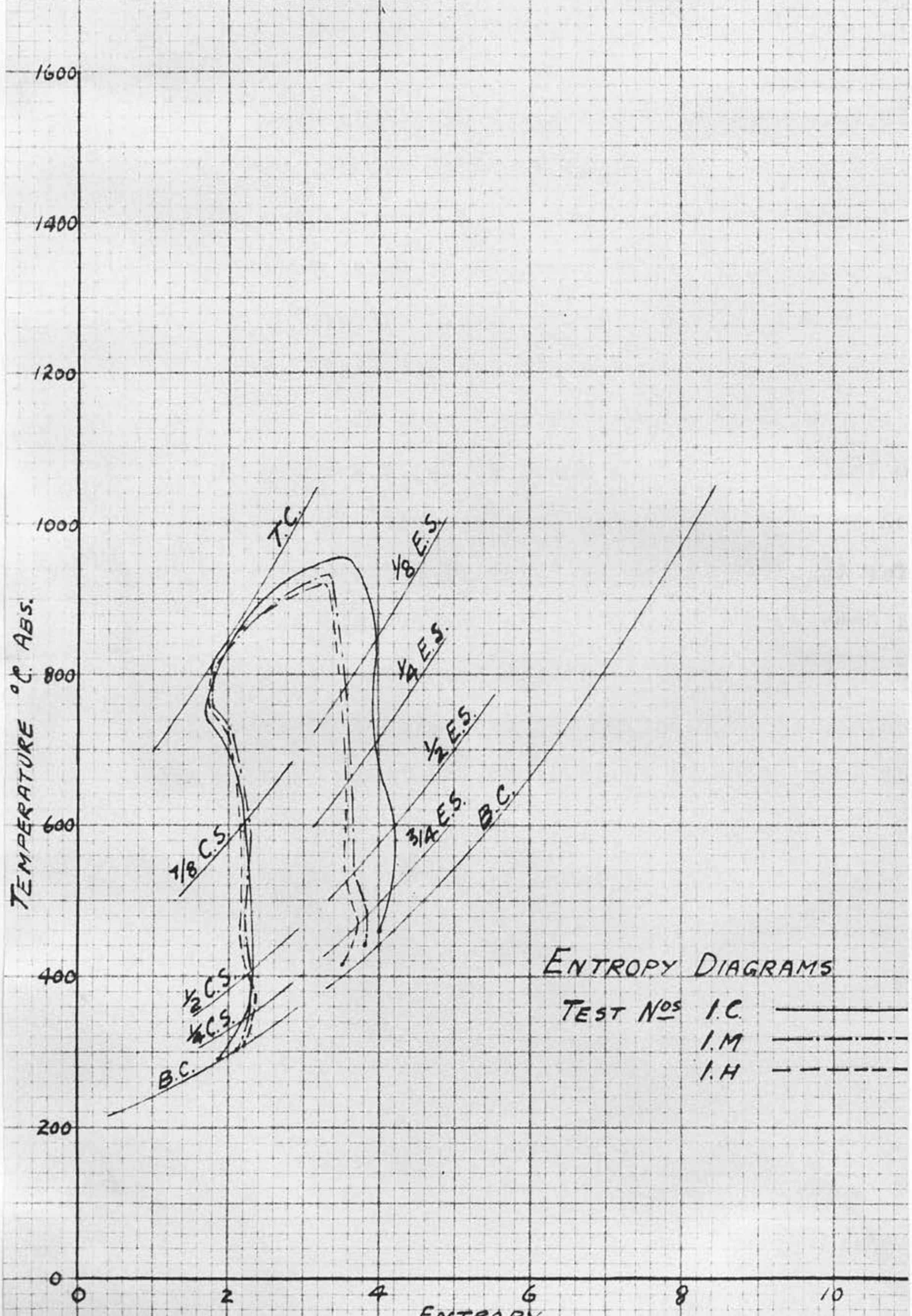

GRAPHS OF INTERNAL ENERGY, TOTAL HEAT AND OF CONSTANT VOLUME ENTROPY CHANGE AGAINST ABSOLUTE TEMPERATURE FOR VARIOUS AIR/OIL WEIGHT RATIOS.
(RECKONED ABOVE 200° C. ABS. FOR ENTROPY ONLY)
AIR/OIL WEIGHT RATIO = Y.
HEAT SCALE. C.H.U./LB.
50
50
100
150
TOTAL HEAT.
A
E
TEMPERATURE °C. ABS.
1600
1400
1200
1000
800
600
400
200
0
(AIR)
(Y=15)
(Y=25)
(Y=100)
(Y=50)
INTERNAL ENERGY
CONSTANT VOLUME
(Y= 15)
(Y=25)
(Y=100)
(Y=50)
(AIR)
X
X
v_x
v_c
v_s
T.C.
B.C.
FOR T.C. $r_v = \frac{v_s + v_c}{v_c} = 13.11$
FOR B.C. $r_v = \frac{v_s + v_c}{v_s + v_c} = 1.0$
FOR ANY INTERMEDIATE POINT. SAY XX, $r_{v_x} = \frac{v_s + v_c}{v_x + v_c}$
1
2
3
4
5
6
7
8
9
10
15
VOLUME RATIO. (r_v) SCALE.
0
2
4
6
8
10
12
ENTROPY

pressures, would perhaps have given a better comparison.

In order to examine the combustion processes and what effects, if any, the variation in jacket temper:ature had on the reception and rejection of heat by the working substance, temperature-entropy diagrams were constructed for all the tests. A key diagram for these is given in Figure 30. This figure also shows curves of Internal Energy and Total Heat, but as these are used in the next section, they will be discussed later. The constant volume entropy change curves, derived in the usual manner, cover the range of air-fuel ratios used, and in their preparation the following procedure was adopted. The composition of the products of combustion for the air-fuel ratios as marked was determined on the assumption of complete combustion. As previously stated, no imperfect combustion was detected during the tests.

The molecular specific heat values used were derived from Partington & Shilling's constants for Molecular Specific Heat equations, which gave "K_v" as a quadratic function of the absolute temperature. The entropy change per mol reckoned above 200°C. absolute for convenience, was calculated at suitable temperature intervals /

FIG. Nº 31.



1600
1400
1200
1000
800
600
400
200
0
TEMPERATURE °C ABS.
T.C.
1/8 E.S.
1/4 E.S.
1/2 E.S.
3/4 E.S.
B.C.
1/8 C.S.
1/2 C.S.
1/4 C.S.
B.C.
ENTROPY DIAGRAMS
TEST Nos 1.C.
1.M
1.H
0
2
4
6
8
10
ENTROPY.

intervals for the constituent gases CO_2, H_2O, etc. The entropy changes per mol at constant volume, for various temperature ranges, were then obtained for gas mixtures, the calculated composition of which represen:ted the products of certain $\frac{\text{Air}}{\text{Oil}}$ weight ratios. Values were also obtained for air and all these were plotted to a base of entropy as in Figure 30.

The logarithmic volume ratio scale is similar to that used by Professor Goudie in his paper* on Energy Charts for the Calculation of Standard Efficiencies of Internal Combustion Engines. The logarithmic scale was used by him in order to obtain temperatures at the end of adiabatic compression and expansion processes.

That a logarithmic scale of volume ratio is a function of the entropy scale to which the constant volume curves are drawn is shown by considering an adiabatic.

The /

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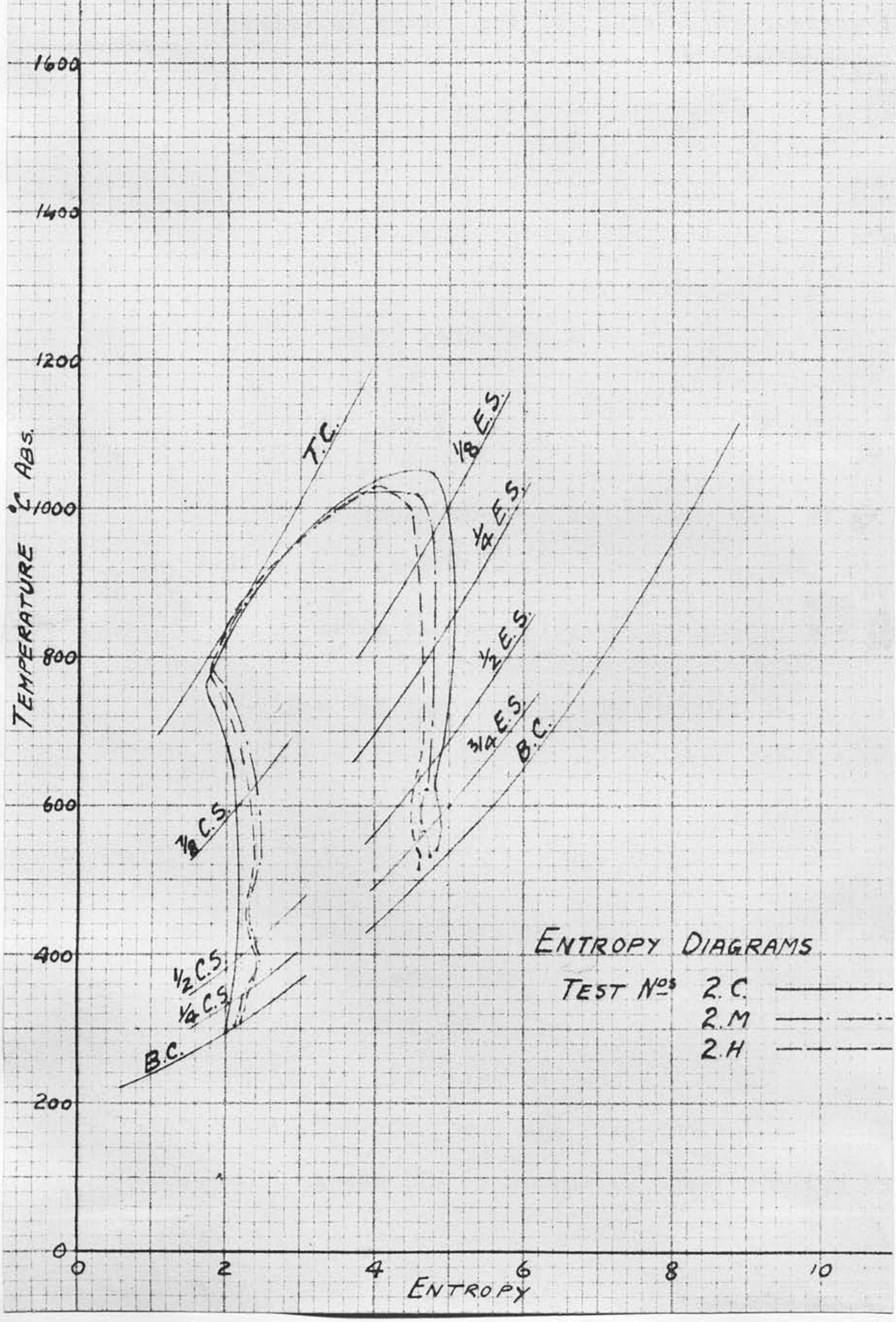
* Proc. Inst. of Engineers & Shipbuilders in Scotland, 1929.

FIG. № 32.


1600
1400
1200
1000
800
600
400
200
0
0
2
4
6
8
10
TEMPERATURE °C ABS.
ENTROPY
T.C.
1/8 E.S.
1/4 E.S.
1/2 E.S.
3/4 E.S.
B.C.
1/8 C.S.
1/2 C.S.
1/4 C.S.
B.C.
ENTROPY DIAGRAMS
TEST N^os 2.C.
2.M
2.H

The general expression for the change in entropy per mol is:

$$\phi_2 - \phi_1 = \int_{T_1}^{T_2} K_v \frac{dT}{T} + \frac{\overline{R}}{J} \int_{v_1}^{v_2} \frac{dv}{v}$$

where K_v = Molecular specific heat

$\overline{R}$ = Universal gas constant

for an adiabatic $\phi_2 - \phi_1 = 0$ and therefore

$$\int_{T_1}^{T_2} K_v \frac{dT}{T} = - \frac{\overline{R}}{J} \int_{v_1}^{v_2} \frac{dv}{v} = - \frac{\overline{R}}{J} \log_e \frac{v_2}{v_1}$$

and if v_2 is less than v_1 as for compression

$$= \frac{\overline{R}}{J} \log_e \frac{v_1}{v_2}$$

where $\frac{v_1}{v_2}$ = adiabatic compression volume ratio

i.e change in entropy at constant volume per mol

$$= 1.985 \log_e \frac{v_1}{v_2}$$

$$= 4.57 \log_{10} \frac{v}{v_2}$$

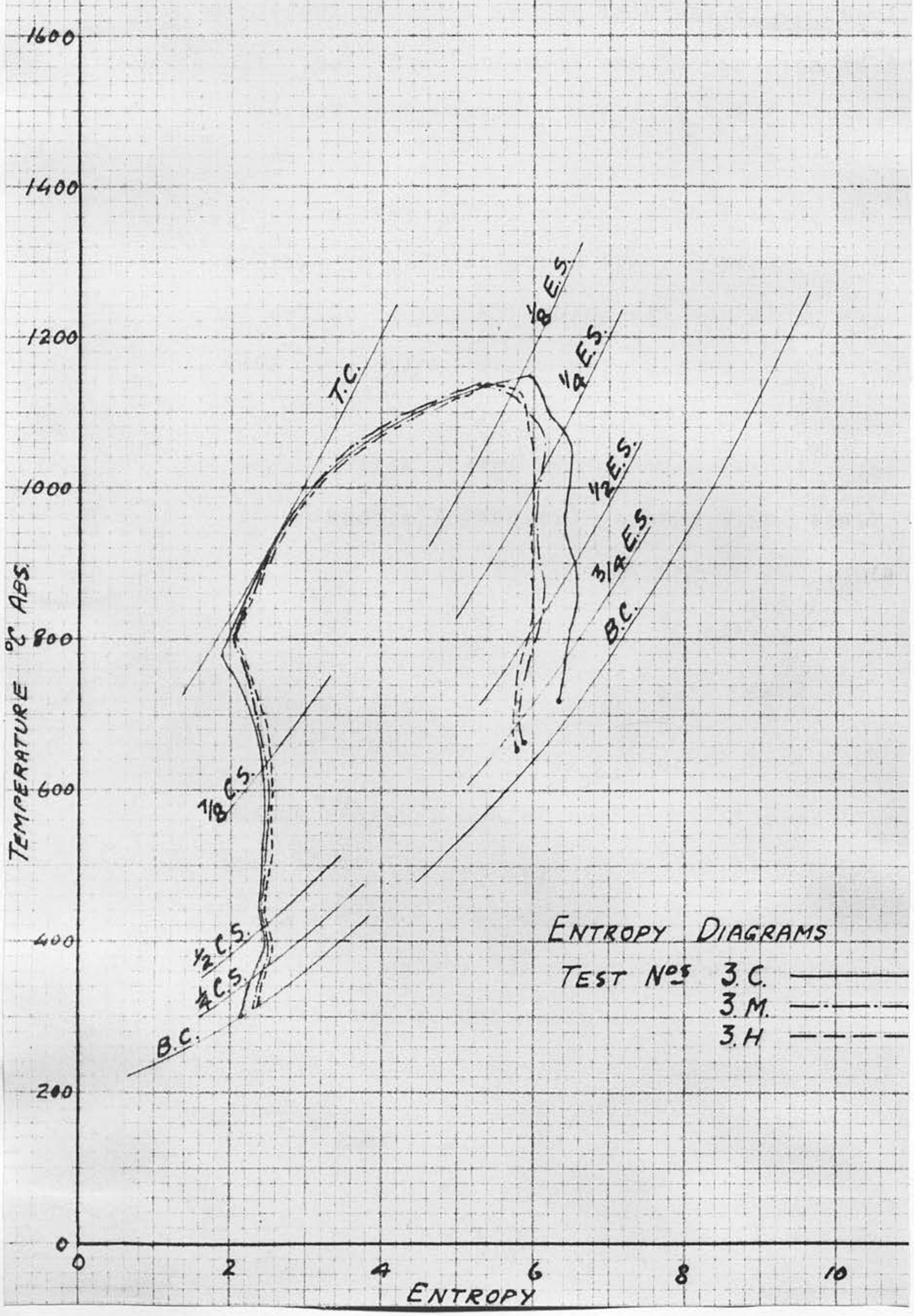
$$= 4.57 \log_{10} r_v$$

where r_v = adiabatic volume compression ratio.

We see therefore that a logarithmic scale of volume ratio is a function of the entropy scale and that this applies to any gas or mixture of gases for which the constant volume curves have been drawn since $\overline{R}$, the universal gas constant, is common to all gases.

A /

FIG. No 33.



1600
1400
1200
1000
800
600
400
200
0
TEMPERATURE °C ABS.
0
2
4
6
8
10
ENTROPY
T.C.
1/8 E.S.
1/4 E.S.
1/2 E.S.
3/4 E.S.
B.C.
1/8 C.S.
1/2 C.S.
1/4 C.S.
B.C.
ENTROPY DIAGRAMS
TEST Nos 3.C.
3.M.
3.H

A logarithmic scale of volume ratio is therefore obtained by marking off the values of 4.57 $\log_{10} r_v$ on the entropy scale with the corresponding values of r_v.

In the present case the scale is used to locate points in the entropy field corresponding to various crank angle positions for which the actual volume is known, and therefore also the volume ratio, r_v, (see PV diagram), and at which the temperature has been estimated. At any temperature level, which in the entropy field represents an isothermal expansion or compression, the change in entropy

$$= \frac{\bar{R}}{J} \log_e \frac{v_2}{v_1}$$

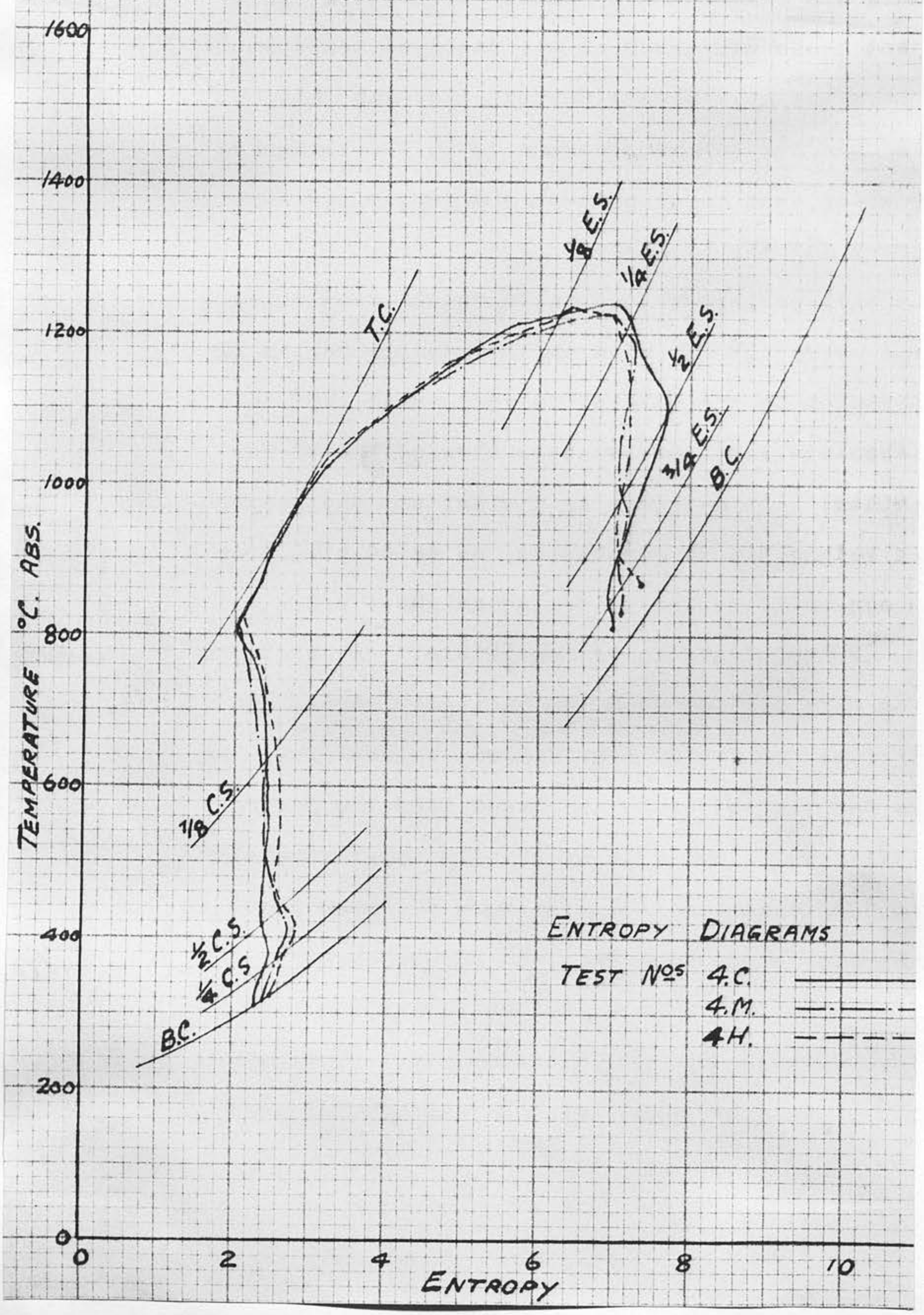
$$= 1.985 \log_e \frac{v_2}{v_1}$$

$$= 4.57 \log_{10} \frac{v_2}{v_1}$$

If v_1 represents the constant volume curve as drawn, for the point B.C. in the PV diagram, and v_2 is less than v_1, then the change in entropy = $- 4.57 \log_{10} \frac{v_1}{v_2}$ = - the particular r_v value on the scale. The negative sign indicates measurements to the left.

For any temperature and crank angle position, the point in the entropy field is located by using dividers to /

FIG. No. 34.

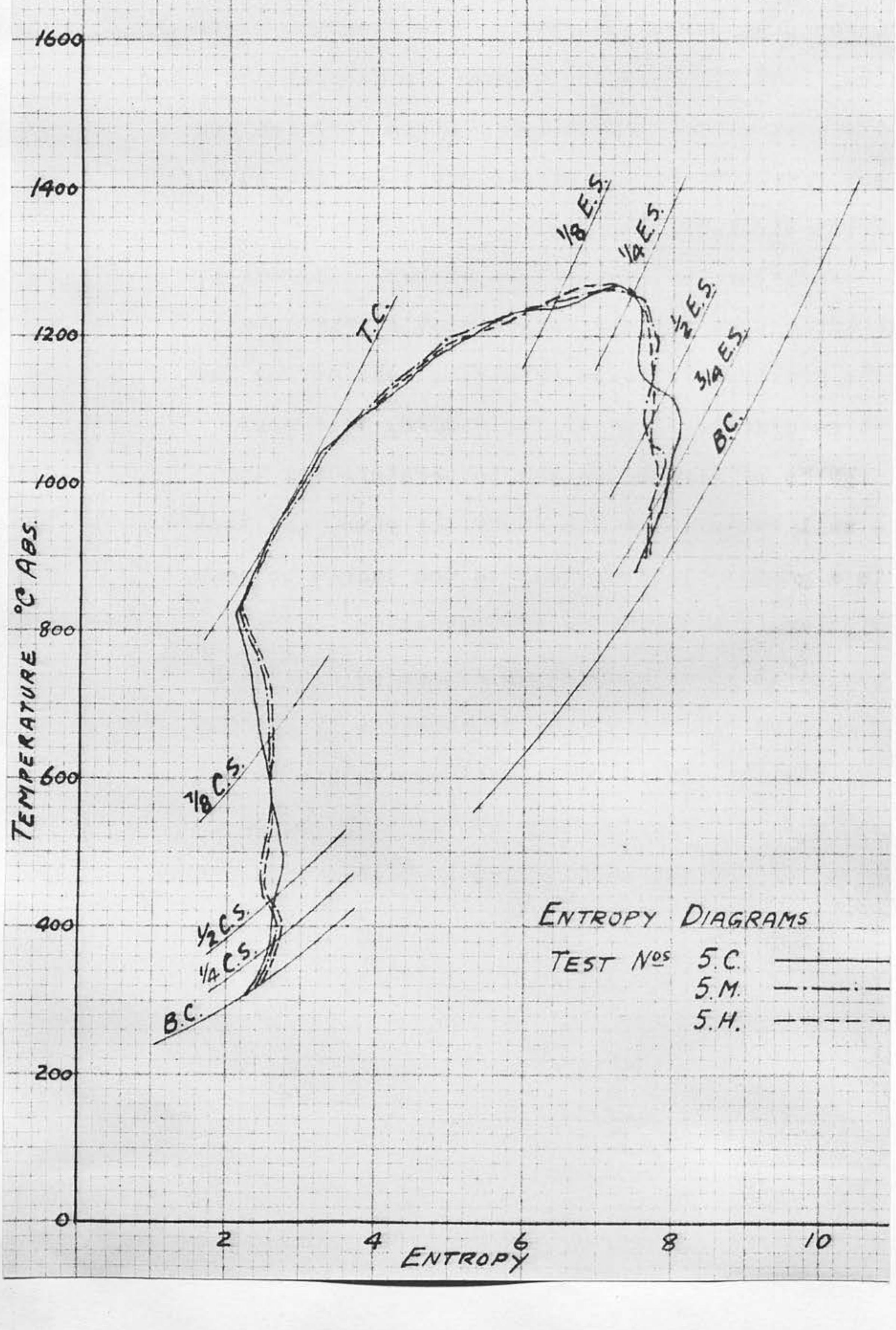


1600
1400
1200
1000
800
600
400
200
0
TEMPERATURE °C. ABS.
0
2
4
6
8
10
ENTROPY
T.C.
1/8 E.S.
1/4 E.S.
1/2 E.S.
3/4 E.S.
B.C.
1/8 C.S.
1/2 C.S.
1/4 C.S
B.C.
ENTROPY DIAGRAMS
TEST Nos 4.C.
4.M.
4H.

to measure the volume ratio values, care being taken to project these distances at the required temperature levels from the appropriate curves, i.e. the "air" curve is used for the compression up to the beginning of combustion, and on or between the particular air-fuel curves for the expansion.

For this purpose volume ratio values were calcul:ated for every 10^{o} of crank angle from the beginning of compression to the exhaust valve opening, 50^{o} before B.C. Absolute temperatures were also estimated for these points. In determining the latter, the contrac:tion in volume due to combustion, as calculated from the gas constants or the molecular weights, i.e. $\frac{R_p}{R_m} = \frac{M_m}{M_p} = \frac{V_p}{V_m}$, was found to be negligible. Absolute pressures were derived from the indicator cards, and the additional weight of the fuel was allowed for, the increase for the latter being spread over the period of injection which could be ascertained approximately from the fuel diagrams. In the same way when plotting the TØ points, the change from the pre-combustion to the post-combustion constant volume curves was made gradually in order that the diagrams might represent as true a picture as possible. No attempt has been made /

FIG. Nº 35.



1600
1400
1200
1000
800
600
400
200
0
TEMPERATURE °C ABS.
2
4
6
8
10
ENTROPY
T.C.
1/8 E.S.
1/4 E.S.
1/2 E.S.
3/4 E.S.
B.C.
1/8 C.S.
1/2 C.S.
1/4 C.S.
B.C.
ENTROPY DIAGRAMS
TEST NOS 5.C.
5.M.
5.H.

made to close the diagrams, and in fact no useful purpose would be served thereby. The diagrams there:fore only show the compression and the expansion stroke up to the exhaust valve opening. Lines representing different fractions of the stroke are added for guidance, see Figures 31 to 37.

In comparing the temperature entropy diagrams it will be noted that for the compression stroke in all cases the gas appears to be receiving heat during the first three-eighths or so of the stroke, that near seven-eighths of stroke the gas temperature and the average wall temperature are about the same, and that there is a general loss of heat to the jacket between this point and the beginning of combustion. The temperatures at which combustion commences rise with increasing load from about 750°C. absolute to 850°C. absolute, but the effect of jacket temperature is not apparent in this respect except at the light loads which show for any one set a slightly higher temperature with the hot jacket.

At light loads the difference in size of the diagrams is considerable, the cold jacket producing the /

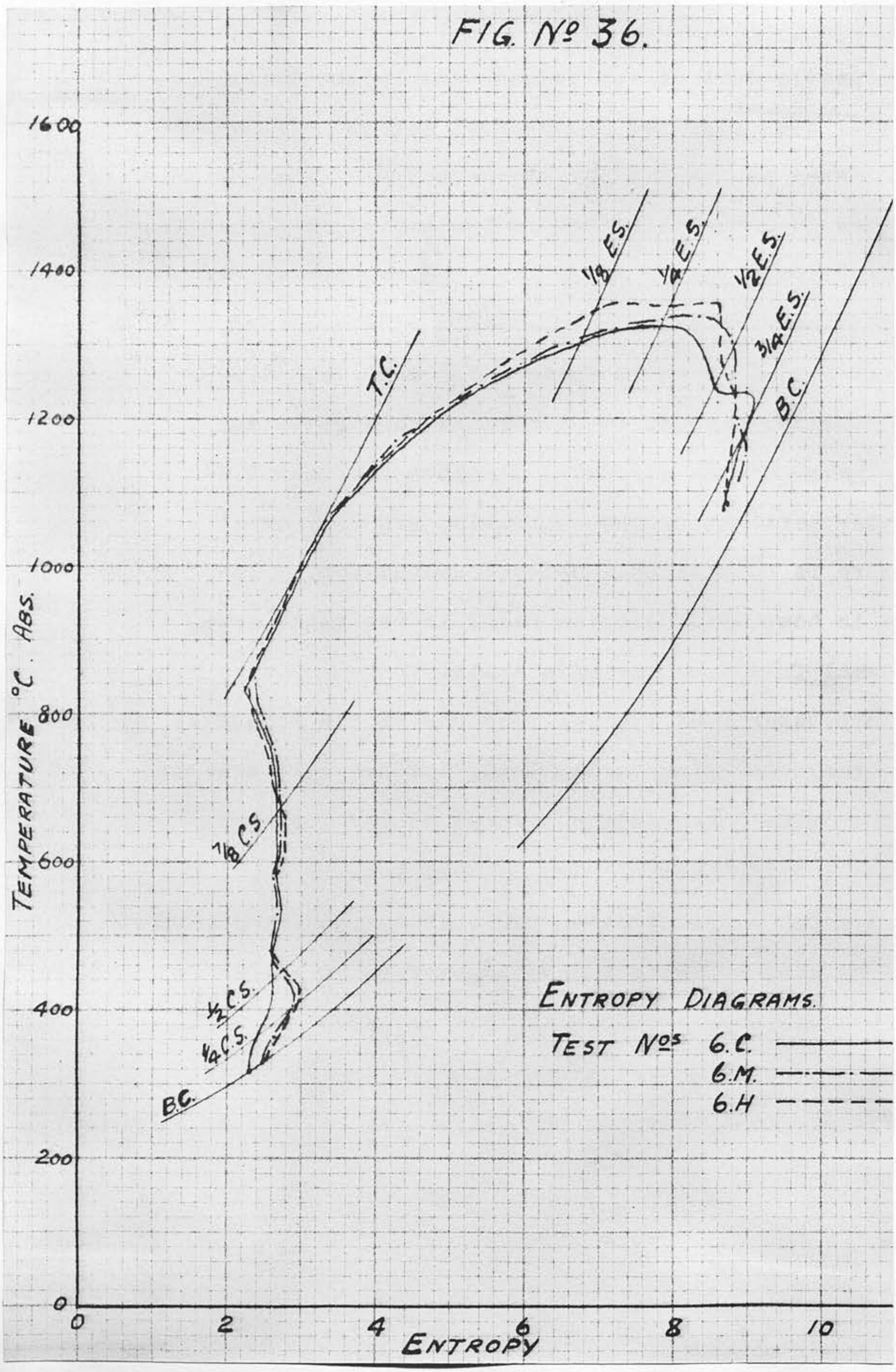


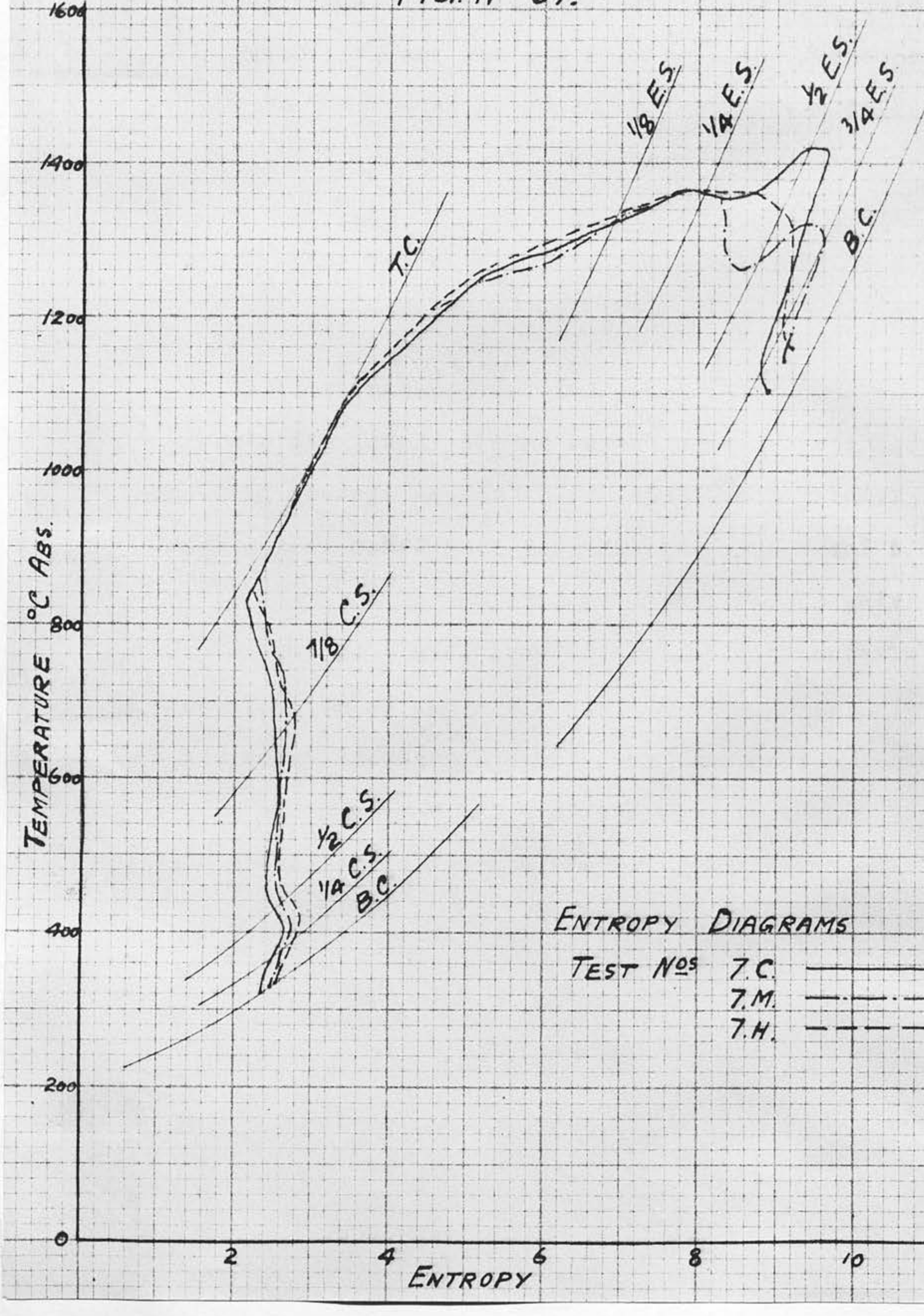
FIG. № 36.
1600
1400
1200
1000
800
600
400
200
0
TEMPERATURE °C ABS.
0
2
4
6
8
10
ENTROPY
T.C.
1/8 E.S.
1/4 E.S.
1/2 E.S.
3/4 E.S.
B.C.
1/8 C.S.
1/2 C.S.
1/4 C.S.
B.C.
ENTROPY DIAGRAMS.
TEST Nos 6.C.
6.M.
6.H

the largest; this is, of course, due to the increased amount of heat supplied in the fuel per lb. of charge, as indicated by the oil per hour - B.H.P. curves shown in Figure 24. The variation in the period of combustion is well illustrated. At no load the combustion has ceased somewhere about one-sixteenth of the stroke, while at full load as in tests 6.C., 6.M., and 6.H., the combustion extends to between three-eighths and five-eighths of the stroke. In tests 7.C. and 7.M. the later portions of the combustion are very erratic, and there is considerable delayed combustion. The engine is however developing about 12 per cent. over its normal full load in these tests.

Temperatures at similar portions of the expansion stroke vary according to the load, but not very greatly with the jacket temperature, except at the lighter loads. For instance, in the no load tests 1.C., 1.M., and 1.H., for a period extending over about five-eighths of the stroke, there is a gas temperature variation of from 50°C. to 80°C. between the hot and the cold jackets, the cold jacket giving the higher values as would be expected from the $\frac{\text{Air}}{\text{Oil}}$ weight ratios.

Taken /

FIG. Nº 37.



1600
1400
1200
1000
800
600
400
200
0
2
4
6
8
10
TEMPERATURE °C ABS.
ENTROPY
T.C.
1/8 E.S.
1/4 E.S.
1/2 E.S.
3/4 E.S.
B.C.
1/8 C.S.
1/2 C.S.
1/4 C.S.
B.C.
ENTROPY DIAGRAMS
TEST Nos 7.C.
7.M.
7.H.

Taken as a whole, the combustion process appears to be more well defined with the hot jacket, though the rate of combustion, as might be inferred from the upward slope of the expansion curve, is not appreciably increased.

The effect of the variation in jacket temperature on the frictional loss will now be considered.

This frictional loss includes bearing friction, valve and fuel plunger operation etc. loss, and also piston friction, but does not of course include pumping or fluid loss, which has already been taken into account in arriving at the I.H.P. nett value.

Unfortunately this frictional loss - in view of the exclusion of pumping loss, it might be more correct:ly termed mechanical loss - does not appear divisible with any degree of accuracy into its components:

1) bearing friction, valve and plunger operation, etc.,

and

a) piston friction.

A method adopted by certain experimenters in order to ascertain the former was to motor the engine with the piston removed, but this would obviously give too /

too low a figure for the following reasons:

a) The removal of the piston involves the elimination of the load on the bearings due to fluid pressures and inertia of reciprocating parts.

b) The pressure against which the exhaust valve operates is atmospheric.

For these reasons no attempt was made when carrying out the tests to estimate, by motoring the engine, the possible value of the individual components of the mechanical loss.

In the later branches of this work, however, another method of separating piston friction from bearing and plunger operation friction is discussed.

Professor Hopkinson, during his tests in the laboratory of Cambridge University on a Crossley gas engine* having a single cylinder $11\frac{1}{2}$" bore × 21" stroke, speed 180 R.P.M., estimated the loss due to bearing friction and valve operation as 2.7% of the I.H.P. gross.

He also found that the mechanical loss varied according to the jacket temperature. In order to investigate this point independently of either indicator or brake, he motored the engine and varied the temperature /

* Proc. I.Mech.E. 1907, vol.IV.

temperature of the water jacket from 21°C. to 82°C., and made tests with normal and excess lubrication and water injections in addition to ordinary lubrication. During these experiments the exhaust valve cover was removed so that there was no loss from air resistance or compression. He obtained the following results:-

	Power absorbed H.P.
Engine hot (82°C) normal lubrication	4.0
Engine cold (21°C) normal lubrication	6.5
Engine cold (21°C) excess lubrication	4.7
Engine cold (21°C) water injected	2.7

From this it will be seen that the purely fric-:tional losses of the engine vary considerably with increased jacket temperature, 2.5 H.P. for normal lubrication. A separate determination of frictional loss was made with the piston and connecting rod removed; this included the main bearing friction valve operation and driving belt losses.

It was found that:

Friction of main bearings, side shaft valve gear, etc. amounted to 1.4 H.P.

By subtraction, the piston and crank pin friction in these experiments varied, therefore, from 1.3 to 5.1 H.P.

The /

The normal value of the piston friction with jacket at 82°C. was 4 - 1.4 = 2.6 H.P.

At this normal working Hopkinson calculated the mechanical efficiency as 87.8% at full load, 41 I.H.P. and 180 R.P.M.

The details were as follows:-

I.H.P. - 41: B.H.P. - 36: Mechanical Efficiency 87.8%

Pumping Losses	- 1.4 H.P.	- 3.4%	of I.H.P.
Piston Friction	- 2.5 H.P.	- 6.1%	of I.H.P.
Other Friction	- 1.1 H.P.	- 2.7%	of I.H.P.
	5.0 H.P.	12.2%	of I.H.P.

Regarding these figures he stated that the apportionment of the piston and other friction appears uncertain, since there was no load on the piston when the engine was motored round, and the compression would increase the friction to some extent.

It is reasonable to anticipate in any internal combustion engine an increase in bearing etc. friction - as distinct from piston friction - at the higher powers because of the increased fluid pressure. What:ever the value of this increase may be, it is a justifiable assumption that, at similar powers, and as in the present case similar speeds, this bearing friction /

friction will be constant under any jacket temperature condition, since the latter can in no way affect the friction external to the cylinder and piston. So that, while the approximate value of the bearing and valve friction has not been determined for the tests under consideration, the ordinates between the curves in Figure 27 give a ready indication of the reduction of piston friction due to increase in jacket temperature at all loads.

From these curves it will be observed that the jacket temperature has a considerable effect on the piston friction, particularly so at light loads. It is very difficult to arrive at an indication, with any degree of accuracy, of the mean jacket temperature existing under the separate conditions, and for pur:poses of comparison the average of the inlet and outlet temperatures has been taken as the mean jacket temperature. The term jacket in these experiments is to be taken as including that portion round the cylin:der head which, from the water circulation point of view, was one with the main body.

The average mean jacket temperatures for the three /

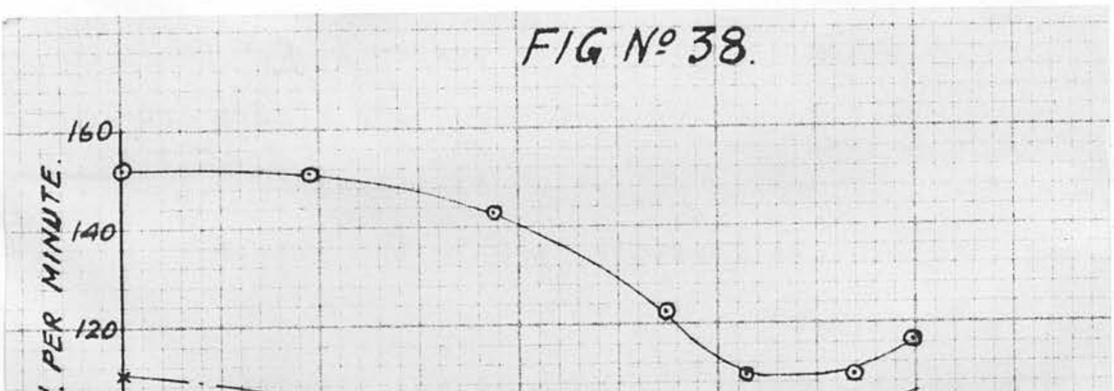

160
140
120
100
80
60
40
20
0
MECHANICAL LOSS IN C.H.U. PER MINUTE.
0
5
10
15
20
B.H.P.
COLD JACKET
MEDIUM JACKET
HOT JACKET

FIG Nº 38.

three series of tests can be taken then as 16°C., 40°C., and 61°C., though it will be appreciated that these figures, while suitable for comparison purposes, must be considered as only approximate.

Figure 38 shows the Mechanical Losses in C.H.U. per minute plotted to a base of B.H.P., and Figure 39 shows the Piston Friction reduction expressed in H.P. for jacket temperature increases within the limits obtained in these experiments.

At 5 B.H.P. by increasing the mean jacket temperature from 16 to 61°C. the energy spent in piston friction is reduced by 3 H.P.;

At 10 B.H.P. by 2.3 H.P.;

At 15 B.H.P. by 0.8 H.P.;

At 18 B.H.P. by 0.5 H.P.

By increasing from 16 to 40°C. the reduction is at 5 B.H.P., 2 H.P.; and at 10 B.H.P., 2.15 H.P., and by increasing from 40 to 60°C., the reduction is at 5 B.H.P., 1 H.P.; and at 10 B.H.P., .25 H.P.

At the higher loads the effect of increasing jacket temperature from 40 to 61°C. on piston friction is /

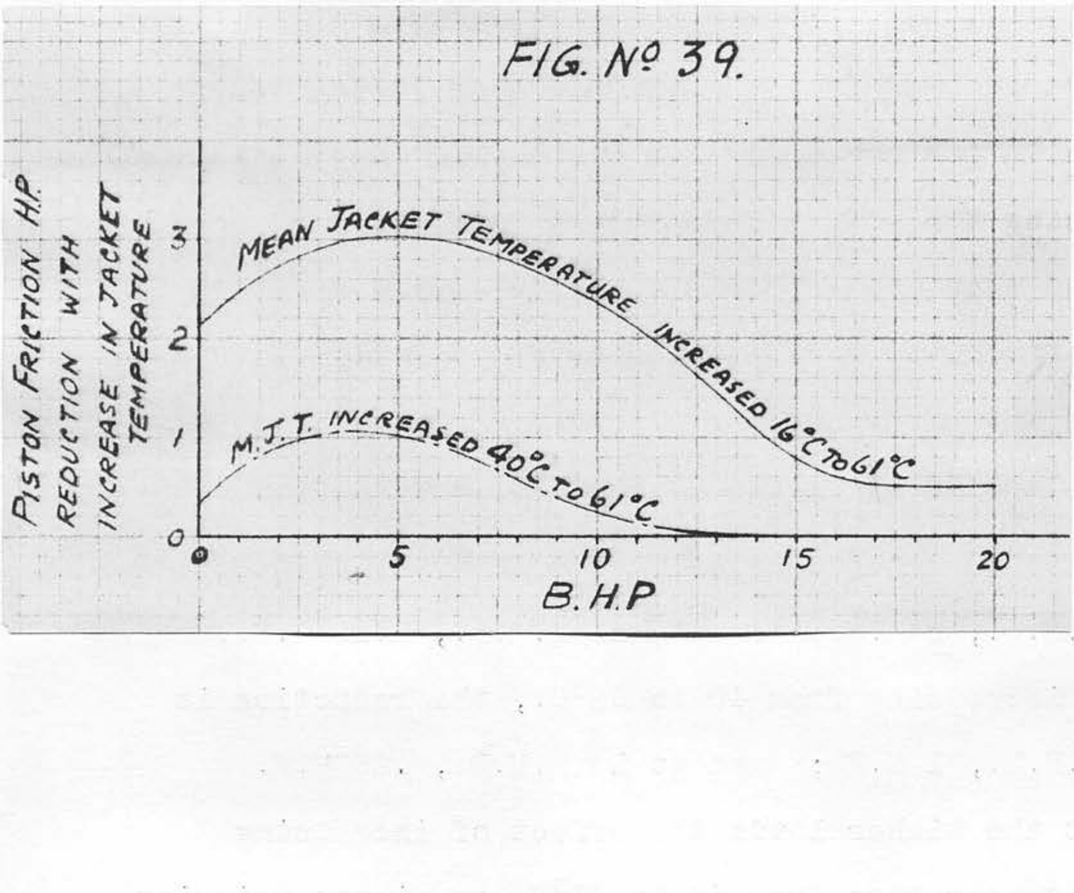


FIG. Nº 39.
PISTON FRICTION H.P. REDUCTION WITH INCREASE IN JACKET TEMPERATURE
MEAN JACKET TEMPERATURE INCREASED 16°C TO 61°C
M.J.T. INCREASED 40°C TO 61°C
0
1
2
3
0
5
10
15
20
B.H.P

is not apparent, though when the cold jacket temperature is increased to 40 or 61°C. a reduction of .5 H.P. is possible. It is of interest to compare these results with other values.

Hopkinson in his experiments already referred to found that by increasing the jacket outlet temperature from 21°C. to 81°C. and motoring the engine the piston friction was reduced by 2.5 H.P. (see page 42)

Dr. Mucklow in his experiments on Piston Temper-:atures in a Solid Injection Oil Engine* estimated that increasing the jacket temperature from 25°C. to 85°C. produced a saving of 2.8 H.P.on the energy spent in piston friction, when the engine was developing 38 H.P. i.e. at 60% of its rated full load. In Dr. Mucklow's description of the tests no mention is made of the jacket inlet temperature, but taking this temperature as 10°C., then the mean jacket temperatures in his case (assumed for comparative purposes only) would be 17.5 and 47.5°C. and the saving in piston friction represented $\frac{2.8}{38.0}$ = 7.4% of the B.H.P. developed.

In the present instance the reduction in friction H.P. for a jacket temperature rise 16 to 61°C. when developing /

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* Proc. I.Mech.E., 1932: vol.123, p.365.

developing 60% rated full load i.e. $10\frac{3}{4}$ B.H.P. is found to be 2.1 H.P. representing a saving of $\frac{2.1}{10.75}$ = 19% of the B.H.P. developed.

By increasing the mean jacket temperature from 16 to 40°C. when developing the same power, e.g. $10\frac{3}{4}$ B.H.P., the reduction of piston friction is found to be 1.9 H.P. representing a saving of $\frac{1.9}{10.75}$ = 17.7% of the B.H.P. developed.

It would however be more accurate to express the saving in piston friction as a percentage of the total mechanical loss, particularly when comparing different powered engines. Unfortunately no indicated power was measured in Dr. Mucklow's experiments and the value of piston friction reduction was obtained not as in this present case from the mechanical loss, but by estimations from the fuel saved and the brake thermal efficiencies. This method of calculating the piston friction saving derived from the Brake Thermal Effic-:iency involves considerations of combustion, improved or otherwise, as well as of friction, and for this reason might give a value above or below that directly deduced from the observed mechanical losses.

It /

It is known that the viscosity or body of a lubricating oil is a measure of its fluidity. It is also known that for any one oil the viscosity varies as the temperature, i.e., the colder the oil the more sluggish it becomes. It is to be expected therefore that a reduction of piston friction is due to a change in the viscosity of the oil consequent on the rise in temperature of the oil film.

The evidence of the temperature-entropy diagrams would suggest that any rise in temperature of the oil film is not due directly to the cylinder gas temper:atures, since there is no great variation in these, and in fact, at the light loads, the mean gas temper:atures are really higher with the cold jacket. To throw further light on this point mean gas temperatures have been calculated, on an equal time interval basis, for suction, compression, expansion and exhaust, and also for the whole cycle. There is remarkably little variation in these mean temperatures. At the low loads the cold jacket gives the highest mean for the cycle, and at the high loads the hot jacket. It would appear therefore that the mean temperature of the /

the oil film is dependent chiefly on the temperature gradient across the cylinder walls to the jacket. In the paper by Professor Dalby on Heat Transmission* it is estimated that in the case of a boiler, 98 per cent. of the total temperature head between the gas and the water is absorbed by the gas film.

In the oil engine, of course, this film is com:plicated by two other factors, namely the presence of lubricating oil and possibly carbon deposits. There is also a considerable change in temperature throughout the cycle.

Estimations based on Heat Transmission data could therefore only give values which might and might not be near the actual. It is not unreasonable to suppose however that the major portion of the temperature grad:ient would be used up in reaching the metal of the cylinder wall. In that case, for any particular gas temperature, the inside wall temperature and also the oil film temperature would be governed chiefly by that of the jacket. Sulzer, in his paper on Temperature Variation and Heat Stresses in Diesel Engines,+ measured the /

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* Proc. Inst.Mech.E., 1909.

+ Trans. Inst.N.A., 1926.

the cylinder wall temperature about 0.5 m.m. from the inner surface. With gas temperature fluctuations of 1000°C. the fluctuations at this point were 14°C. above and 8°C. below the mean of 240°C. The cylinder wall in this case was 2 inches thick and the jacket temperature 40°C., though whether this was the outlet or the mean is not stated. If the mean temperature, taken from the recorded values as 240°C., is considered as the inside wall temperature, then the temperature drop across the gas film would be about 80 per cent. of the maximum temperature range.

In estimating the reduction in piston friction with increased jacket temperature, the values of such have been obtained under similar power conditions, so that the actual piston loads which are partially res-:ponsible for these, can be considered as equal. It would seem therefore that reduction in piston friction with increased jacket temperature is due principally to the lowering of the viscosity of the lubricating oil as its temperature is raised. While this state-:ment may in some manner be an explanation of the con-:ditions at any one power, it does not fit so well when /

when the full range of powers is considered. In Figure 38 the curves of Heat to Friction are wide apart at the low powers and close together at the high powers. At the high powers the gas temperatures are very much increased, the estimated mean cyclic temper:atures exceeding those at no load by $300^{\circ}C$. The effect of this would be to increase the inner wall mean temperature by an approximately equal amount for the cold, medium and hot jackets. The proportionate increase in the case of the cold jacket would therefore be greatest, tending to equalise the viscosities. This may therefore in part account for the proximity of the curves. Another reason may be that at these high powers considerable heat is given to the oil film by direct conduction through the hot piston. If the oil film is unbroken, the piston friction is dependent only on the viscosity or resistance to flow of the oil, since in a case such as the one under discussion, the mean piston speed is constant and also the surface area.

This is the important point

It is very doubtful however whether under the conditions existing the oil film can be maintained. A high viscosity oil will, within certain limits, tend to /

to preserve the film to a greater degree than a low one, for similar loads. On the other hand the high viscosity oil will increase the resistance of the film to shear. Where the film is broken then the friction:al force will depend to a certain extent on the piston loads and the character of the surfaces. Whether it is actually possible to get the piston and rings riding on an oil film in the manner of a shaft bearing is debatable. The sharp edges of the rings would tend to scrape off the oil rather than to draw it under the rings. On the other hand there must be some oil between the rings and the cylinder, otherwise there would be no lubrication. It may be assumed that some sort of lubrication is secured whether of film support, partial or so-called "greasy". Probably partial lubrication would be the most accurate estimate.

As the engine powers increase, so also will the piston loads, the mean gas temperatures and the oil film temperatures. The viscosity of the oil therefore decreases as the piston loads increase.

Dealing now with the shape of the individual curves of Heat to Friction as shown on Figure 38; if we consider the bearing friction constant throughout the /

the whole range of powers, then it would appear that between no load and 7 B.H.P., with the hot jacket, the decreasing rate of oil viscosity has a greater effect on the piston friction than the increasing rate of the friction due directly to increasing piston loads and partial lubrication. Beyond 7 B.H.P., while the vis:cosity of the oil is still decreasing, i.e. with increasing temperature, the rate of increasing piston loads with partial lubrication now has the greater effect.

In the same way the medium jacket curve would indicate that at about 13 B.H.P. the effect of decreased viscosity is balanced by increased piston loads on the partially lubricated cylinder walls.

For the cold jacket there is a value of 17 B.H.P. at which the equalisation in the effects takes place.

These conclusions may not be fully justified, but in view of the many disturbing influences which exist during the movement of the piston, the more elaborate use of data adapted from theoretical considerations would not give results other than of a somewhat sketchy nature.

As /

As applied to an internal combustion engine the heat balance should be a statement of the way in which the total amount of heat passed into the engine has been employed.

In the experiments made by the Institution of Civil Engineers Committee* on gas engines, the full load balance sheet was given as:

Exhaust Waste ....
Jacket Water ....
Radiation etc. ....
B.H.P. ....

In the above, radiation etc. includes engine friction as well as radiation proper.

Clerk, in his paper+ read before the Institution of Civil Engineers, endeavoured to correct this measurement from several points of view. He used I.H.P. instead of B.H.P., so that the friction of the engine was no longer included under radiation, and by reasoning upon other figures found in the report, he gave the heat balance as:

Exhaust Waste ....
Jacket Water & Radiation .....
I.H.P. ....

Even this adjusted balance sheet was, however, erroneous in so far as too much heat appeared under Jacket Water and /

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* Proc. Inst.C.E., vol.163.

+ Proc. Inst.C.E., vol.69, 1907.

and Radiation and too little under Exhaust Waste. The reasons given for this were that the hot gases discharging round the exhaust valve impinge on the water jacketed space, that when the hot gases in the cylinder have fallen to atmospheric pressure, the cyl-:inder remains full of these high temperature gases which, during the exhaust stroke, lose heat to the jacket, and also that piston friction generates heat and this may flow either to the jacket or disappear in radiation and conduction.

The heat balance as it stood therefore was again modified, although this could not be done by any of the methods which had up to that time been used.

Clerk now brought into use the values found by him for the specific heats of the constituent gases. He estimated from the indicator diagram the gross I.H.P. (i.e. from positive loop), also the heat in the gases at the end of expansion, and further, the heat flow to the jacket during explosion and expansion. He then gave the balance sheet in the following terms:

Heat flow during explosion and expansion ....
Heat contained in gases at end of expansion ....
Heat to I.H.P. ....

This /

This he considered was the most accurate heat distribution balance sheet.

Frequently heat balances are given somewhat in this form:

```
Heat supplied  ............
Heat to I.H.P.(B.H.P.
              (Friction ...
Heat to Jacket  ...........
Heat to Exhaust  ..........
Heat to Radiation etc.
     (by difference) ......
```

This type of heat balance may be misleading, so far as the friction heat, jacket heat, exhaust heat, and radiation are concerned. It has, however, been adopted in the present case for reasons which will appear presently.

In arriving at the figures in Heat Balance No.1, the following methods were adopted:

The heat supplied was based on the lower calorific value of the oil.

The heat to I.H.P. was taken on the I.H.P.(nett).

The heat to friction equal to I.H.P.(nett)-B.H.P.

The heat to exhaust was obtained by subtracting the total heat of the entering air and fuel at atmos-:pheric temperature from the total heat of the exhaust gases /

gases at exhaust temperature. For this purpose curves of what might be termed Total Heat and Internal Energy at various temperatures, for air and also for the products of certain $\frac{\text{Air}}{\text{Oil}}$ weight ratios, were prepared. These curves, which are shown to a reduced scale on Figure 30, were drawn with due allowance for variation in specific heat with temperature; complete combustion was, however, assumed in all cases.

When referring to the preparation of the Constant Volume Entropy curves it was mentioned that the mole:cular specific heat values as used for the constituent gases were derived from Partington and Shilling's equations. Here again those values were used to obtain the Internal Energy per mol (reckoned above 0^{o}C. absolute) for the constituent gases at various temper:atures. The Internal Energies per lb. (reckoned above 0^{o}C. absolute) were then obtained at various temperature intervals for the products of the $\frac{\text{Air}}{\text{Oil}}$ ratios marked, and also for air.

The Total Heat of a gas or mixture per lb. at absolute temperature T is derived from the Internal Energy per lb. at the same absolute temperature by adding /

adding the quantity $\frac{R}{J}$ T (all these values being reckoned above $0^{o}C$. absolute). $\frac{R}{J}$ does not vary appreciably over a wide range of air-oil mixtures and no sensible error is caused by the use of a mean value (a mean value since R refers to a particular gas)

This term $\frac{R}{J}$ T is a straight line OA through the origin.

In Figure 30, the intercept at any temperature level between the axis OY and any of the curves OE, when measured on the Heat Scale, gives the value of the Internal Energy per lb. reckoned above $0^{o}C$. absolute.

In the same way the intercept between any of the curves OE and the line OA gives the total heat.

<u>TABLE III</u> /

TABLE III.

Heat Balance No.1 (based on one minute)

Test No.	1.C.		1.M.		1.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	458	100	382	100	340	100
Heat to I.H.P.	152	33.15	110.5	29.02	101.8	29.9
Heat to B.H.P.	-	-	-	-	-	-
Heat to Friction	152	33.15	110.5	29.02	101.8	29.9
Heat to Jacket	261	56.9	136	35.6	66.7	19.6
Heat to Exhaust	136.6	29.8	136.8	35.8	130.5	38.3
Radiation etc.	-91.6	-19.85	-1.3	-.42	41.0	12.2
Total	458	100	382	100	340	100

Heat Balance No.1 (contd.)

Test No.	2.C.		2.M.		2.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	689	100	603	100	556	100
Heat to I.H.P.	263	38.2	218.2	36.2	191.5	34.45
Heat to B.H.P.	111.8	16.25	112.8	18.7	111.0	20.0
Heat to Friction	151.2	21.95	105.4	17.5	80.5	14.45
Heat to Jacket	320	46.45	172	28.5	124.3	23.32
Heat to Exhaust	197.5	28.7	189.0	31.3	182.5	32.8
Radiation etc.	-91.5	-13.35	23.8	4.0	57.7	10.43
Total	689	100	603	100	556	100

TABLE III (CONTD.)

Heat Balance No.1 (based on one minute)

Test No.	3.C.		3.M.		3.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	985	100	860	100	832	100
Heat to I.H.P.	366	37.3	311	36.2	300.5	36.17
Heat to B.H.P.	223	22.66	218.2	25.4	218.5	26.2
Heat to Friction	143.0	14.64	92.8	10.8	82.0	9.97
Heat to Jacket	410	41.6	250	29.06	202	24.3
Heat to Exhaust	280.0	28.45	261.0	30.3	255.0	30.65
Radiation etc.	-71.0	-7.35	38.0	4.44	74.5	8.88
Total	985	100	860	100	832	100

Heat Balance No.1 (contd.)

Test No.	4.C.		4.M.		4.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	1270	100	1180	100	1142	100
Heat to I.H.P.	444	35.0	417	35.32	407	35.7
Heat to B.H.P.	321.4	25.3	324	27.5	317	27.8
Heat to Friction	122.6	9.7	93.0	7.82	90.0	7.9
Heat to Jacket	492	38.7	370	31.3	283	24.75
Heat to Exhaust	380.0	29.9	366.0	31.0	352	30.75
Radiation etc.	-46.0	-3.6	27.0	2.38	100.0	8.8
Total	1270	100	1180	100	1142	100

TABLE III (CONTD.)

Heat Balance No.1 (based on one minute)

Test No.	5.C.		5.M.		5.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	1433	100	1350	100	1310	100
Heat to I.H.P.	479	33.4	461.5	34.12	457	34.9
Heat to B.H.P.	369.2	25.75	371	27.5	363	27.7
Heat to Friction	109.8	7.65	90.5	6.62	94.0	7.2
Heat to Jacket	513	35.8	409	30.3	337	25.7
Heat to Exhaust	439	30.6	415	30.7	392	29.9
Radiation etc.	2.0	0.2	6.45	4.88	124.0	9.5
Total	1433	100	1350	100	1310	100

Heat Balance No.1 (contd.)

Test No.	6.C.		6.M.		6.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	1761	100	1710	100	1587	100
Heat to I.H.P.	544	30.9	535	31.3	524	33.05
Heat to B.H.P.	434	24.62	437.6	25.6	419.5	26.5
Heat to Friction	110.0	6.28	97.4	5.7	104.5	6.55
Heat to Jacket	587	33.3	540	31.55	420	26.5
Heat to Exhaust	544	30.9	530	31.0	498.0	31.4
Radiation etc.	86.0	4.9	105.0	6.15	145.0	9.05
Total	1761	100	1710	100	1587	100

TABLE III (CONTD.)

Heat Balance No.1 (based on one minute)

Test No.	7.C.		7.M.		7.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	1962	100	1861	100	1755	100
Heat to I.H.P.	586.5	29.9	567.5	30.52	555	31.6
Heat to B.H.P.	468.0	23.85	465	25.0	452.7	25.7
Heat to Friction	118.5	6.05	102.5	5.52	102.3	5.9
Heat to Jacket	655	33.4	576	30.9	464	26.42
Heat to Exhaust	617.5	31.45	576	30.9	539	30.7
Radiation etc.	103.0	5.25	141.5	7.68	197	11.28
Total	1962	100	1861	100	1755	100

Heat Balance No.1 shows very distinctly the mis-:leading nature of this method of assessment. For instance it will be noted that for low loads with the cold jacket as in Test Nos.1.C., 2.C., 3.C., etc., the radiation loss is - 19.75%, - 13.35%, - 7.35% and so on. This arrangement has, however, been used in the present case to obtain further knowledge of the piston friction. In the last section, various values for the piston friction reduction with increased jacket temperature were obtained. Now an attempt will be made to estimate the actual piston friction for any load under each of the three jacket conditions.

Take the case of Test No.1.C. In this test, as in all the cold jacket tests, the inlet temperature to the jacket was in the region of 8^{o}C. and the outlet 24^{o}C., giving a mean jacket temperature of say 16^{o}C. These temperatures, inlet and outlet, were in fact such as to reduce the jacket radiation to the minimum, since the atmospheric temperature was in the region of 15^{o}C. An assumption may be made, therefore, that for the cold jacket tests the jacket radiation is zero.

As /

As regards the piston radiation, this will depend on the engine load. It does not matter, for present purposes, what proportion of the piston friction heat may or may not have passed into the exhaust, since this would be accounted for in the exhaust heat, at the expense of the jacket heat. It is to be expected, however, that most, if not all, would pass to the jacket water. In this particular test, namely 1.C., the piston radiation would be very small, and it will in the first instance at least be neglected.

Under those assumed conditions (that is, where the radiation from the piston as well as from the jacket is neglected), the piston friction only can account for the 19.85% excess. Here then is an in-:dication of the value of the piston friction at no load with the cold jacket.

Objection might be taken to this roundabout method of estimating the piston friction, but, provided care is taken and reliable indicator cards are obtainable, the value as estimated should be a reasonably accurate one. It might be pointed out, of course, that while the indicated and brake values are no doubt reliable, the /

the exhaust temperature is not, since this, as already mentioned, was measured about four inches from the exhaust valve. This intervening distance was, how:ever, jacketed, so that what heat the exhaust gases lost the jacket gained. Moreover, this method should give a more reliable value for the piston friction than that obtained by motoring the engine with and without the piston in place, since these are not actual running conditions.

Such a method of deducing the piston friction heat would not, of course, be applicable with the cold jacket at the higher loads, in view of the very consid:erable increase in piston radiation which could not of course be neglected. The actual excess heat corresponding to the 19.85% is 91.6 C.H.U. per minute, which on the assumption of zero piston radiation must be piston friction. Subtracting this value from the heat to jacket, 261 C.H.U., the new value of the latter is 169.4 C.H.U. per minute, i.e. assuming all of the friction has passed to the jacket and nothing to the exhaust, though whether this assumption is justifiable or not does not affect the case. As the total frictional heat is 152 C.H.U. per minute, then the /

the value of the bearing friction is 152 - 91.6 = 60.4 C.H.U. per minute.

Proceeding on similar lines with Test 2.C., since it is only with the cold jackets that jacket radiation may be neglected, the excess there is 13.35% and the actual excess heat 91.5 C.H.U.

The new value of the jacket heat is then 320 - 91.5 = 228.5 C.H.U. per minute, and the bearing friction 151.2 - 91.5 = 59.7 C.H.U. per minute.

This follows closely the previous value, although deduced from different heat quantities.

As the loads increase the piston radiation will increase; the bearing friction may also increase, although very slightly. At the higher loads piston radiation could not be neglected and therefore this method of estimating the bearing friction would not apply. For instance, it would appear that in Test 5.C. the piston radiation has reached a value approx:imately equal to the heat generated by piston friction. This however does not enable the bearing friction to be estimated. In thus estimating the bearing friction, a mean value of which might be taken as 60 C.H.U. per minute /

why

minute, the piston radiation has been neglected.

Using now this value for the bearing friction, an approximate figure for the piston radiation heat for Test 1.C. may be obtained.

A revised Heat Balance is made out for say the first five of the cold jacket tests. This is similar to Heat Balance No.1 except for the insertion of bearing and piston friction in place of total friction, together with the consequent modifications in jacket heat and radiation. The values are arrived at in the following way:

Bearing friction is taken as 60 C.H.U. throughout.

The piston friction equals the total friction minus the bearing friction.

The jacket water heat is reduced by the piston friction heat and the radiation values are by difference.

TABLE IV /

TABLE IV.

Revised Heat Balance for Cold Jacket Tests.
(based on one minute)

Test No.	1.C.		2.C.		3.C.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	458	100	689	100	985	100
Heat to I.H.P.	152	33.15	263	38.2	366	37.3
Heat to B.H.P.	-	-	111.8	16.25	223.0	22.66
Heat to Bearing Friction	60	13.1	60	8.7	60	6.08
Heat to Piston Friction	92	20.1	91.2	13.21	83.0	8.42
Heat to Jacket	169	36.9	228.8	33.2	327.0	33.19
Heat to Exhaust	136.6	29.8	197.5	28.7	280.0	28.45
Radiation	0.4	0.1	-.3	-0.06	12.0	1.20
Total	458	100	689	100	985	100

TABLE IV (CONTD.)

Test No.	4.C.		5.C.	
	C.H.U.	%	C.H.U.	%
Heat supplied	1270	100	1433	100
Heat to I.H.P.	444	35.0	479	33.4
Heat to B.H.P.	321.4	25.3	369.2	25.75
Heat to Bearing Friction	60	4.72	60	4.18
Heat to Piston Friction	62.6	4.92	49.8	3.47
Heat to Jacket	429.4	33.84	483.2	33.78
Heat to Exhaust	380.0	29.9	439.0	30.6
Radiation	16.6	1.32	31.8	2.22
Total	1270	100	1433	100

The radiation values given in this revised Heat Balance may now be used to obtain an approximation to the piston radiation for Test 1.C. This was done by plotting the radiation heat values to a base of anti-:cipated average piston temperatures. The average piston temperatures were taken, in the absence of other data, as the mean cylinder temperatures, and a smooth curve was drawn through the points.

It is known that the radiation from a black body varies as the fourth power of the absolute temperature, and an equation of the form $R = bT^4 + c$ was assumed to fit the curve. From this equation, taking the rad-:iation at atmospheric temperature (288^o absolute) as equal to zero, the value of the piston radiation for Test 1.C. equalled approximately 3 C.H.U. It is very difficult to arrive at any satisfactory conclusion as to how this 3 C.H.U. radiation would have affected the other quantities in the Heat Balance No.1. It is to be expected that had it been possible to run the engine in such a way as to prevent piston reduction, that under this theoretical condition a portion of this heat might have been used up in reducing any cooling effect which /

which the piston had on the gases in the cylinder, but most of it would have gone into the cylinder jacket and for that reason, instead of 91.6 C.H.U. appearing as piston friction, the value would have been 94.6 C.H.U., representing an excess of 20.6% instead of 19.85 %. Under these conditions the bearing friction would amount to 152 - 94.6 = 57.4 C.H.U. per minute, and the actual heat to jacket from the cylinder 261 - 94.6 = 166.4 C.H.U. per minute. This reasoning may appear super:fluous in view of the very small quantity of heat (3 C.H.U.) under consideration. As pointed out, how:ever, when discussing Heat Balance No.1, the discrep:ancy of 19.85% in Test 1.C. could only be held to represent piston friction provided jacket and piston radiation were both zero. The assumption of zero jacket radiation still holds, but having found an approximate value for piston radiation, this is now used in order to determine the bearing friction, which for Test 1.C. is found to be 57.4 C.H.U. A mean value of 57 C.H.U. per minute for the bearing friction at all loads has therefore been adopted in the Final Heat Balance, and while this assumption of constancy may not /

not be considered as correct, any small variation such as is probable would not materially affect the other values. In this final balance no attempt has been made to produce what might be called a Thermo-dynamic Heat Balance, i.e. by distinguishing between heat lost to walls during expansion and that lost during exhaust. It is principally devoted to the variables outside the actual working substance, since the latter appears from the tests to be not so greatly affected by the change in jacket temperature.

TABLE V /

TABLE V.
Final Heat Balance (based on one minute)

Test No.	1.C.		1.M.		1.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	458	100.0	382	100.0	340	100.0
Heat to I.H.P.	152	33.1	111	29.0	102	29.9
Heat to B.H.P.	-	-	-	-	-	-
Heat to Bearing Friction	57	12.4	57	14.9	57	16.7
Heat to Piston Friction	95	20.7	54	14.0	45	13.2
Heat to Jacket (from gases)	166	36.5	82	21.6	22	6.5
Heat to Exhaust	137	29.8	137	35.8	130	38.3
Radiation (by difference)	3	0.6	52	13.7	86	25.3
Total	458	100.0	382	100.0	340	100.0

TABLE V (CONTD.)

Test No.	2.C.		2.M.		2.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	689	100.0	603	100.0	556	100.0
Heat to I.H.P.	263	38.2	218	36.2	191	34.4
Heat to B.H.P.	112	16.2	113	18.7	111	20.0
Heat to Bearing Friction	57	8.3	57	9.4	57	10.2
Heat to Piston Friction	94	13.7	48	8.0	24	4.3
Heat to Jacket (from gases)	225	32.7	124	20.6	101	18.1
Heat to Exhaust	198	28.7	189	31.3	182	32.8
Radiation (by difference)	3	0.4	72	12.0	81	14.6
Total	689	100.0	603	100.0	556	100.0

TABLE V (CONTD.)
Final Heat Balance (based on one minute)

Test No.	3.C.		3.M.		3.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	985	100.0	860	100.0	832	100.0
Heat to I.H.P.	366	37.3	311	36.2	300	36.2
Heat to B.H.P.	223	22.7	218	25.4	219	26.2
Heat to Bearing Friction	57	5.8	57	6.6	57	6.8
Heat to Piston Friction	86	8.7	36	4.2	25	3.0
Heat to Jacket (from gases)	324	32.9	214	24.9	177	21.3
Heat to Exhaust	280	28.4	261	30.3	255	30.7
Radiation (by difference)	15	1.5	74	8.6	99	12.0
Total	985	100.0	860	100.0	832	100.0

TABLE V (CONTD.)

Test No.	4.C.		4.M.		4.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	1270	100.0	1180	100.0	1142	100.0
Heat to I.H.P.	444	35.0	417	35.3	407	35.7
Heat to B.H.P.	321	25.3	324	27.5	317	27.8
Heat to Bearing Friction	57	4.5	57	4.8	57	5.0
Heat to Piston Friction	66	5.2	36	3.1	33	2.9
Heat to Jacket (from gases)	426	33.6	334	28.3	250	21.9
Heat to Exhaust	380	29.9	366	31.0	352	30.8
Radiation (by difference)	20	1.5	63	5.3	133	11.6
Total	1270	100.0	1180	100.0	1142	100.0

TABLE V (CONTD.)
Final Heat Balance (based on one minute)

Test No.	5.C.		5.M.		5.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	1433	100.0	1350	100.0	1311	100.0
Heat to I.H.P.	479	33.4	461	34.1	457	34.9
Heat to B.H.P.	369	25.8	371	27.5	363	27.7
Heat to Bearing Friction	57	4.0	57	4.2	57	4.3
Heat to Piston Friction	53	3.7	34	2.5	37	2.8
Heat to Jacket (from gases)	480	33.5	375	27.8	300	22.9
Heat to Exhaust	439	30.6	415	30.7	392	29.9
Radiation (by difference)	35	2.4	98	7.3	162	12.4
Total	1433	100.0	1350	100.0	1311	100.0

TABLE V (CONTD.)

Test No.	6.C.		6.M.		6.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	1761	100.0	1710	100.0	1587	100.0
Heat to I.H.P.	544	30.9	535	31.3	524	33.0
Heat to B.H.P.	434	24.6	438	25.6	419	26.5
Heat to Bearing Friction	57	3.2	57	3.3	57	3.6
Heat to Piston Friction	53	3.0	40	2.4	48	3.0
Heat to Jacket (from gases)	534	30.4	500	29.2	372	23.4
Heat to Exhaust	544	30.9	530	31.0	498	31.4
Radiation (by difference)	139	7.9	145	8.5	193	12.1
Total	1761	100.0	1710	100.0	1587	100.0

TABLE V (CONTD.)

Final Heat Balance (based on one minute)

Test No.	7.C.		7.M.		7.H.	
	C.H.U.	%	C.H.U.	%	C.H.U.	%
Heat supplied	1962	100.0	1861	100.0	1755	100.0
Heat to I.H.P.	587	29.9	668	30.5	555	31.6
Heat to B.H.P.	468	23.9	465	25.0	453	25.7
Heat to Bearing Friction	57	2.9	57	3.1	57	3.3
Heat to Piston Friction	62	3.1	46	2.4	45	2.6
Heat to Jacket (from gases)	593	30.3	530	28.6	419	23.9
Heat to Exhaust	617	31.4	576	30.9	539	30.7
Radiation (by difference)	165	8.4	187	10.0	242	13.8
Total	1962	100.0	1861	100.0	1755	100.0

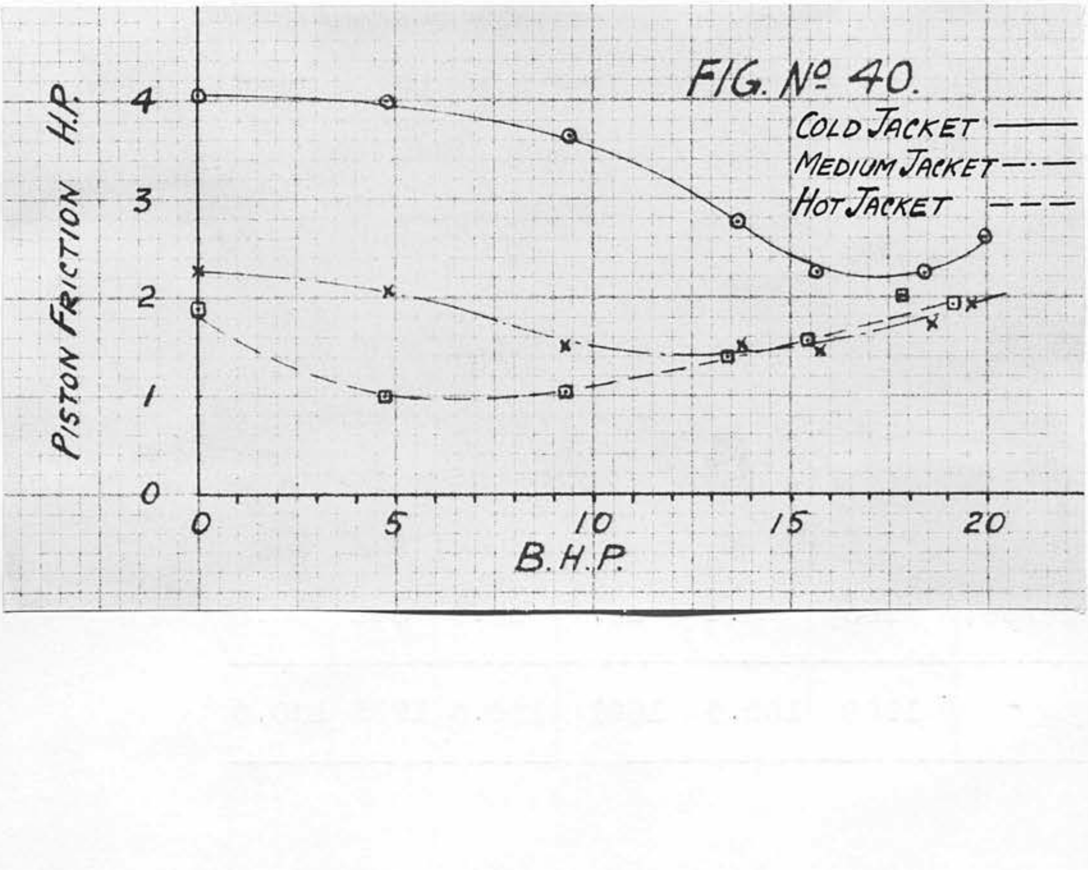
FIG. No 40.
COLD JACKET
MEDIUM JACKET
HOT JACKET
PISTON FRICTION H.P.
0
1
2
3
4
0
5
10
15
20
B.H.P.

Figure 40 shows the values of the Piston Friction H.P. plotted to a base of B.H.P. The curves, as would be expected, follow the shape of the Heat to Friction curves in Figure 38. The maximum Piston Friction H.P., 4.1, occurs with the cold jacket at no load, while the minimum, about 1 H.P., is with the hot jacket when the engine is developing 7 B.H.P. Taking the bearing friction and plunger operation as 57 C.H.U. per minute, equivalent to 2.42 H.P. at all loads say, then the range of piston friction varies from 70% above this value to about 60% below it.

With the cold jacket, the piston friction H.P. varies from 4.1 to 2.2, i.e. it is almost halved as the power developed at the brake increases from nothing to 17 B.H.P., while for the hot jacket, over the same power range, the variation is only 0.75 H.P.

Figure 41 shows the values of the ratio

$\frac{\text{Piston Friction H.P.}}{\text{I.H.P.(nett)}}$ plotted to a base of B.H.P.

This ratio, about 62% and 44%, at no load, for the cold and hot jackets respectively, falls to an approx:imately common value for all jackets of 10% at the maximum /

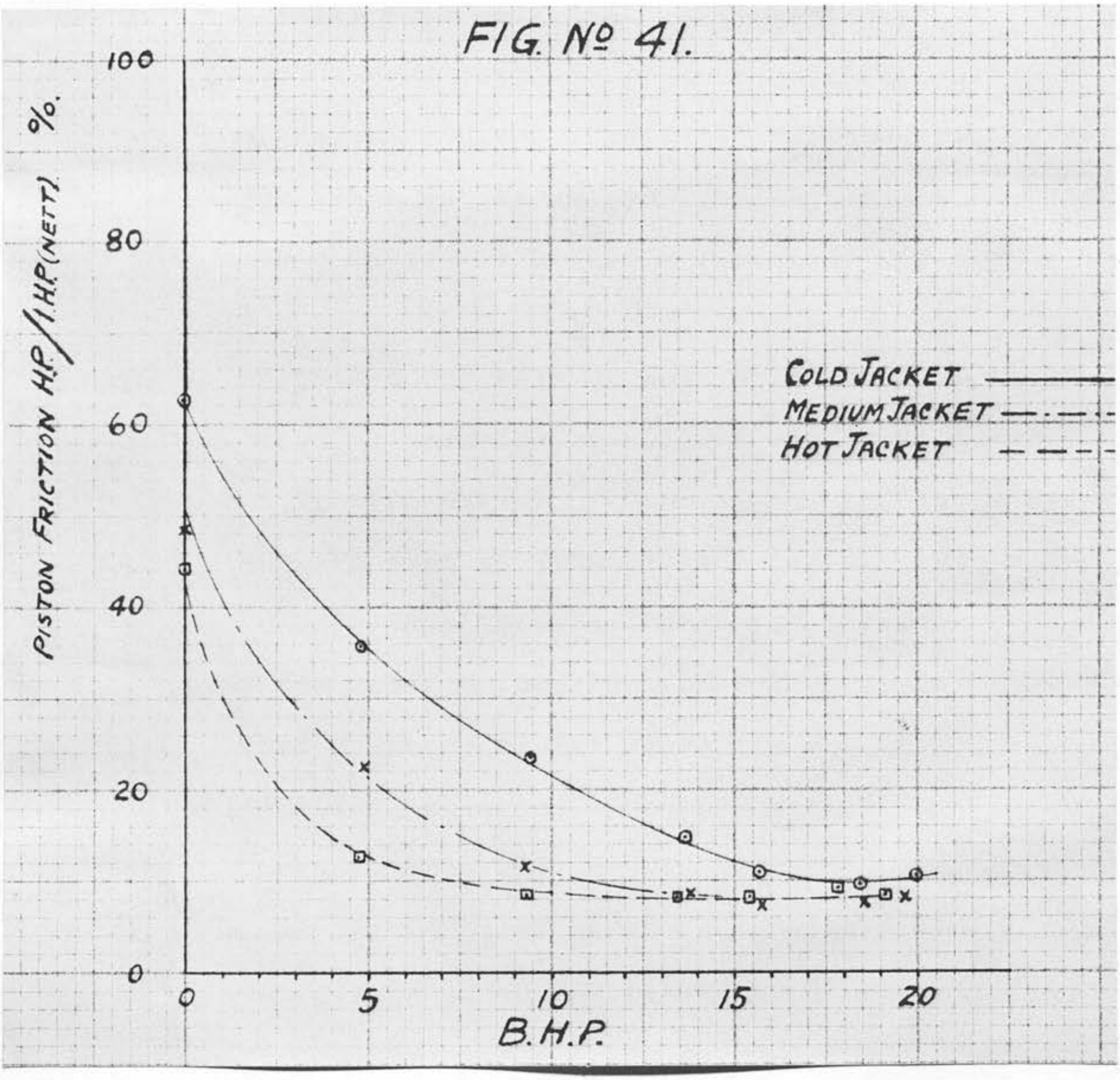

FIG. № 41.
PISTON FRICTION H.P./I.H.P.(NETT) %.
100
80
60
40
20
0
0
5
10
15
20
B.H.P.
COLD JACKET
MEDIUM JACKET
HOT JACKET

maximum loads. With the hot jacket, the fall in per:centage is particularly rapid in the early power stages, a drop from 44% to 12% occurring in the first 5 B.H.P., while beyond this power the value decreases gradually to the minimum of 8%. These curves clearly indicate the beneficial effect of the hot jacket in reducing to a minimum the fraction of the power devel:oped within the cylinder, which is wasted on piston friction.

No attempt has been made to plot the radiation values recorded in the Final Heat Balance. These increase according to the load. In the case of the hot and medium jackets the increase is more or less uniform throughout, but with the cold jacket the increase is small at the low and medium loads and then beyond 17 B.H.P. the radiation values increase very rapidly. The very considerable increase in radiation, at the high loads, in the case of the cold jacket, suggests a relatively high piston temperature, since this radiation can be taken as primarily piston rad:iation. This would reinforce the suggestion which was made, when considering the variation in viscosity of /

of the lubricating oil, that the approachment of the Heat to Friction curves at the high loads, as shown in Figure 38, would be partly due to the heat from the hot gases conducted through the piston, to the oil film.

One further point in connection with the radiation values may be mentioned. While those for the cold jacket may be taken as due primarily to piston rad-:iation, the others represent a combination of jacket and piston radiation. If it is assumed that for any particular jacket condition the jacket radiation is the same for all loads, then an approximate value for that due to piston and that to jacket can be ascertained. It is doubtful of course whether this is a justifiable assumption when the mean jacket temperature is over atmospheric.

If it is, then by considering the actual piston radiation at no load, for the medium and hot jackets, as equal to that for the cold jacket, namely 3 C.H.U. per minute, the jacket radiation for the hot and medium jackets is obtained by subtraction, namely 49 and 83 C.H.U. per minute respectively, and from the assumption of constancy the piston radiation values are found for any load.

GENERAL COMMENTS.

As the main results of the tests have already been discussed at some length, there is no intention of dealing further with these, but, before attempting to draw any conclusions, mention might be made of the methods which have been adopted in arriving at these results. It is agreed that frequent assumptions have been made, and the validity of certain of these may be questioned.

Perhaps the two deserving of greatest criticism, since so much depends on them, are:-

1) That the mean jacket temperature is the average of the inlet and outlet valves;

2) That no radiation takes place from the cylinder jacket when the mean jacket temperature (arrived at as above) is approximately equal to the atmospheric temperature.

Regarding the first of these, it is appreciated that in the case of the medium jacket where the temperature range was considerable, this method may not be satisfactory, but with the cold and hot jackets, where the temperature range was small, no great error would /

would be involved, and it is difficult to see how a closer estimate could be obtained without very com:plicated measurements.

The validity of the second assumption is depend:ent to a certain extent on that of the first. It is reasonable to expect that the top portion of the jacket would be at the higher or outlet temperature, but the effect of this would be counteracted by the cold lower portion, since the bulk of the jacket over:hung the foundation and was therefore exposed to the atmosphere all round.

The unorthodox method of estimating the piston friction from a heat balance might also be questioned, particularly in view of the many variables involved. It can only be said in extenuation of this method that, provided reasonable care is taken, the value as found is from actual running conditions.

CONCLUSIONS /

CONCLUSIONS.

The rate of combustion does not appear to be seriously influenced by the jacket temperature. There is of course a very considerable heat loss to the jacket when the latter is cold, as is shown in the Final Heat Balance, but it is probable that the major portion of this occurs during the exhaust stroke. Against this greater heat loss to the walls must, however, be credited the gain due to increased volume-:tric efficiency. As far as the power developed within the cylinder is concerned it may be said that the efficiency is not greatly affected by change in the jacket temperature. The piston friction on the other hand is very considerably influenced by the jacket temperature, especially at light and moderate loads. This would appear to be due almost entirely to the change in the nature of the lubricating oil film with change in temperature. It is known that for most, if not all lubricating oils there is considerable variation in viscosity through a moderate range of temperature, and it may be suspected that as far as the piston friction is concerned this change in viscosity is the deciding factor.

what gain by greater vol. η since all fuel is burnt & no CO

Under /

Under the severe conditions imposed by the combustion of the fuel in the cylinder other influences may be introduced which would affect the piston friction to a greater or less degree, such as dilution and de:composition. The first of these would tend to reduce the viscosity of the lubricant, and the second would certainly be detrimental to the maintenance of the oil film. There are also many other disturbing influences but it is doubtful whether during the relatively short period of the tests the existence of any of these would be effective in altering the piston friction to any extent.

In conclusion, the author desires to acknowledge the many helpful suggestions received from Sir Thomas Hudson Beare.

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