AN INVESTIGATION OF THE HEAT REJECTION TO THE ENGINE OIL IN AN INTERNAL COMBUSTION ENGINE

by

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and

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Acceptance:

Instructor in Charge of Thesis,

414 Beacon Street
Boston 15, Massachusetts
February 10, 1944

Professor G. W. Swett
Secretary of the Faculty
Massachusetts Institute of Technology

Dear Sir:

In accordance with the requirements for the degree of Bachelor of Science, we herewith respectfully submit a thesis entitled "An Investigation of the Heat Rejection to the Engine Oil in an Internal Combustion Engine".

Sincerely yours,

Lester Simon

William A. Jack

ACKNOWLEDGMENT

The authors wish to express their appreciation to Professor A. R. Rogowski for his advice and interest in conducting this investigation and also to the Sloan Laboratory personnel for their valuable assistance.

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INTRODUCTION

INTRODUCTION

The purpose of this investigation was to determine and analyze the nature of the heat rejection to the engine oil under various operating conditions. This included a verification of previous work[1*] and a separation of the components of total heat rejected. By analyzing the contributing effects of each component the general nature of the total heat rejection was determined to be a function of only two variables, the friction and the indicated horsepower.

* References are to bibliography, page 29.

PREVIOUS INVESTIGATIONS

An investigation of the literature available showed there has been little work done on the problem of heat rejection to the engine oil and the closely related question of engine friction. Kaneb and Hoey[1] found that the total heat rejection to the oil showed an increase which was linear with speed at constant IMEP. Increasing load at a constant speed showed a slight increase in heat transfer at low loads while at higher loads the transfer was much greater. Over the operating range the heat transfer increased slightly less than fifty per cent as the load was quadrupled. As these tests were run at only two speeds, 1800 and 2000 RPM, definite conclusions could not be drawn although trends were established.

The object of an investigation by McLeod[2] was to determine the degree of error in the motoring test and to establish a correlation between it and the true engine friction. This aspect will be discussed later in conjunction with the motoring tests made in this investigation (see page 20).

SUMMARY OF RESULTS

It was found that the heat rejected to the engine oil was some function of the indicated and the friction horsepower. Empirical equations were developed for the heat rejected due to friction and heat rejected from the cylinder gases. These separate relationships were combined to obtain an empirical relationship for the total heat rejected to the engine oil. All are given below: first in general form, then with the constants evaluated for the particular engine tested. All relationships for heat transfer are in Btu/minute.

General:

$$Q_f = K_8(FHP)^{n1} \quad \text{(friction heat rejected to oil)}$$

$$Q_r = K_5(IHP)^{n2} \quad \text{(heat rejected to oil from cylinder gases)}$$

$$Q_t = K_8(FHP)^{n1} + K_5(IHP)^{n2} \quad \text{(total heat rejected to oil)}$$

Plymouth Engine Tested:

$$Q_f = 1.4(FHP)^{1.5}$$

$$Q_r = 0.82(IHP)^{1.1}$$

$$Q_t = 1.4(FHP)^{1.5} + 0.82(IHP)^{1.1}$$

The observed data showed that the engine speed exerts much influence on the heat rejected to the oil.

The heat rejected to the oil increased threefold over the range of speed tested (1200 to 2400 RPM). It would, therefore, be necessary to use large oil coolers with large high-speed engines.

The change in heat rejected over the range of output tested (3 to 46 BHP) at constant speed was small compared to the variation found for the speed range, and it remained approximately the same at all speeds. At high engine speeds, load or output had comparatively little effect on the heat rejected to the oil. At low speeds, load became more influential because of the small heat transfer involved.

It was also made clear that the oil is an important factor in absorbing the heat rejected from the engine. The oil system absorbed nearly a quarter the total heat rejected from the engine -- the remainder was absorbed by the cooling water system or cooling air surfaces.

APPARATUS

The tests represented were performed on a six-cylinder 1936 Plymouth engine having the following specifications:

Bore - - - - - - - - - - - -	3-1/8 inches
Stroke - - - - - - - - - - -	4-3/8 inches
Piston Displacement - -	201.3 cubic inches
Compression Ratio - - -	6.70
Rated Output - - - - - - -	82 brake horsepower at 3600 RPM

Load for firing and power for motoring was supplied by a dynamometer having the following specifications:

Current - - - - - - - - - -	147 amperes
Voltage - - - - - - - - - -	250 volts
Output - - - - - - - - - - -	100 horsepower

The engine was of standard manufacture, but necessary alterations were made to facilitate testing. A description of the engine set-up previously used for similar tests on this engine is given below.[1]

Old Set-up

The crankcase pan was lagged with asbestos to prevent appreciable heat transfer from the oil to the atmosphere.

The engine oil pump was disconnected and replaced by an external electric motor-driven gear pump. The oil was pumped from an external reservoir to the engine through the port which originally contained the pressure relief valve in the lubricating system of the engine. The oil circulated through the engine and returned to the external reservoir by gravity. A long pipe was hinged to a stand beneath the engine. This pipe could be swung so that it bypassed the gravity oil flow from the engine and emptied it into a weighing pan placed on a platform balance. By means of this weighing system, the rate of oil flow could be determined.

Several heat exchangers using steam as the heating element were installed in the external oil feed line. Cooling water was also piped to these heat exchangers in order to effect a more sensitive regulation of the oil inlet temperature. Thermometers were placed at the inlet and the outlet oil ports. These thermometers were located as near the ports as possible in order to obtain accurate readings of the inlet and outlet temperatures.

An oil pressure gage connected to the inlet oil port was used for the purpose of determining the relative rate of oil flow through the engine. To keep the flow nearly constant, a gate valve was installed in the external oil line before the engine inlet port.

The engine exhaust line was attached to the laboratory trench pump system, and the air intake system was left unchanged from the original engine.

Cooling water was pumped to the engine jacket and to a cooler surrounding the exhaust pipe. A thermocouple was inserted in the jacket cooling system, and a gate valve was used to regulate cooling water flow and the resultant jacket temperature. A surge tank was added to this cooling system to maintain a fairly constant rate of flow and at the same time to keep the cooling water temperature more stable.

New Set-up

To carry out the test routine decided upon, it was found necessary to make several changes and additions to the original arrangement as described above. The new set-up used is shown diagrammatically in Fig. 4, page 30, and in Figs. 5 and 6, pages 31 and 32, respectively.

First, the original asbestos layer was removed from the crankcase and replaced by a fresh and slightly thicker application of asbestos. Chicken wire, served to bind the asbestos coating to the pan. These precautions were taken to prevent damage to the insulation from excessive vibration.

Next, the oil-weighing system was found to be unsatisfactory. The bypass pipe to the weighing pan was a source of some error in determining rate of oil flow. This rate of flow was measured from the end of the bypass giving not the true rate of flow through the engine, but the rate of flow through the engine as modified by the time lag produced by the friction in the long pipe. Moreover, the process of emptying the contents of the weighing pan into the reservoir after each run was an unnecessary difficulty.

These disadvantages in measuring the rate of oil flow were eliminated by rearranging the weighing system so that oil from the engine outlet port flowed by gravity directly into a five-gallon tank. From this intermediate tank the oil flowed by gravity through a short pipe to another five-gallon tank placed on a platform balance. A two-way valve was attached to the end of this short pipe so that flow to the weighing tank could be stopped quickly. The suction side of the external oil pump and its bypass line also were led from this weighing tank.

Now, the oil flow to the engine could be measured accurately by determining the time necessary for a given weight of oil to be pumped out of the weighing tank and into the engine. Once equilibrium had been established, the rate of flow measured represented the rate of flow of oil to the engine itself. The oil flowed continuously; no time had to be taken to remove the contents of one tank and replace the contents of the other.

During the trial runs, it was found that the oil pressure in the engine became very low at high engine speeds. To compensate for this drop in pressure (which also meant a corresponding drop in the rate of oil flow), a gate valve was inserted in the pump bypass. Shutting this valve raised the flow and pressure from the discharge end of the pump.

To maintain approximately constant back pressure, the exhaust was disconnected from the laboratory line. This prevented a change in exhaust conditions in case the laboratory line was being evacuated.

A steam line was connected to the water jacket cooling system so that jacket temperatures similar to those during firing could be maintained for the motoring runs.

An exhaust gas analyzer was used to determine the fuel-air ratio supplied to the engine. A strobotac was used to check the accuracy of the cablė-driven tachometer. Both of these instruments were soon abandoned in view of the slight effect produced by a change in fuel-air ratio and the difficulties encountered from use of the strobotac as compared to the accuracy possible with the other measurements in the tests.

The rest of the original set-up was left intact. It was considered adequate for the accuracy desired and proved to be so in the ensuing tests.

METHOD OF TESTING

Preliminary Tests

In this research there was a considerable number of variables to deal with. The dependent variable in all tests was some portion of the heat rejected to the oil, i.e., total heat rejected, friction heat, or that portion rejected from the hot engine parts. The independent variables were oil pressure, water jacket temperature, oil inlet and outlet temperatures, spark advance, fuel-air ratio, speed, and load. All other factors were considered to be constant. For best results only one independent variable should be varied, with all others remaining constant; but, to accomplish the purpose of this investigation in the limited time available, it was decided to determine the relative importance of each variable on the heat rejection to the oil. Runs were taken to find the allowable operating range for each variable without impairing the accuracy of the heat transfer measurements. Greater accuracy than that of the temperature readings was unwarranted. In each of the following tests, all other variables were held rigidly constant.

Over the total pressure range, 40 to 80 psig., it was found that the greatest change in heat rejected amounted to about ten per cent from the mean value, 65 psig. which had been selected for use in the testing. It was therefore decided that a variation between 60 and 70 psig. would be permissable as this allows a percentage error of not more than five per cent. (See Fig. 7)

Water jacket temperature changes have some effect on heat rejection. Fig. 8 shows the maximum change over the range tested to be 14%. With reference to this curve, it was considered adequate to operate within a temperature range of 155 degrees to 170 degrees, Fahrenheit. The maximum possible variation in total heat rejected over this range then becomes about three per cent.

The oil inlet and outlet temperature have a marked effect on the total heat transfer. Fig. 9b indicates that as much as 74% change in heat rejected occurs for the small outlet temperature range of 185 to 195 degrees Fahrenheit. Heat rejected is almost as sensitive to inlet temperature changes. Fig. 9a shows the quantitative changes in heat rejection with respect to temperature while Fig. 9b gives the percentage variation. From this preliminary test, it was decided to keep the mean temperature of the oil at a relatively constant value. These readings of temperature introduced the controlling error of all the results. With the range selected the maximum error introduced was under fifteen per cent.

The change in the heat rejection with change in spark advance was never more than three per cent over a wide range. (See data sheet Number 1, test Number 5.)

For the first tests, a check on the fuel-air ratio indicated no appreciable change during operation. The fuel-air ratio was then assumed to be constant at 0.0785.

A strobotac was used during the first few tests but it was found that for the accuracy desired, speed could be maintained within the limiting range by merely using the RPM dial indicator.

Main Tests

With limitations necessary for sufficient accuracy in view, the actual testing was begun. A series of constant speed tests was performed with the brake horsepower varying. Allowing twenty to thirty minutes for the establishment of equilibrium for each run, measurements of brake load, jacket temperature, oil pressure, oil inlet and outlet temperature, speed, and rate of oil flow were then taken. To reduce the number of tests, constant brake horsepower settings were taken during the speed runs by calculating the correct brake load in advance. The speed range was 1200 to 2400 RPM with increments of 200 RPM. The brake horsepower range was from 3.2 to 46.

Motoring runs were obtained separately from the corresponding firing runs. Methods of maintaining similar operating characteristics will be explained below. For the motoring runs, measurement of speed, friction (brake) load, jacket temperature, oil pressure, oil inlet and outlet temperature, and rate of oil flow were taken as before.

The speed and load ranges were limited by the characteristics of the cradle dynamometer.

Measurements

Engine speed was read on a tachometer. Brake and friction loads were read from the balance of the dynamometer. Jacket temperatures were obtained from a dial gage connected to a thermocouple in the cooling system. Oil pressure readings were taken from a dial gage connected to the inlet side of the oil system at the engine inlet port. The oil inlet temperature was measured by a mercury thermometer placed in a well at the engine inlet port and the oil outlet temperature was measured by a mercury thermometer at the exit passage from the engine. A running balance was taken for the rate of oil flow. First, the weighing tank was allowed to fill up. Then the two-way valve in the line leading from the intermediate tank was closed. The usual weight of oil used during the running balance was ten pounds.

All measurements were an average of at least three runs after equilibrium had been established. Particular attention was paid to the reading of the oil temperatures as they were the major source of error.

DIRECT HEAT LOSSES

General Theory

Heat transfer within the internal combustion engine may be divided into two processes. One is the heat given off to the engine parts by the working fluid, and the other is the heat produced by friction of the moving parts.

It has been found by dimensional analysis[4] of the variables involved that the heat transfer from the working fluid may be represented by

$$Q_r = \Delta TAC_p(\rho S)^n\left(\frac{1}{\mu}\right)^{n-1} \tag{1}$$

where ΔT = mean temperature difference between the gas and the cylinder walls

A = exposed area

C_p = mean specific heat of the gas

ρ = mean gas density

S = mean piston speed

μ = mean gas viscosity

n = exponent determined by experiment

Hence, for a given engine operating with approximately the same outside cylinder temperature (ΔT is constant), the heat transfer from the working fluid is seen to be some function of piston speed (or RPM) and gas density.

$$Q_r \propto (\rho S)^n \quad \text{or} \quad Q_r \propto (\rho \times RPM)^n \qquad (2)$$

Now, the indicated mean effective pressure of an engine may be represented by

$$IMEP = J\rho (F \times E \times \eta_t) \times \frac{1}{144} \qquad (3)$$

where J = mechanical equivalent of heat (a constant)

ρ = density of gases in cylinder

F = fuel-air ratio

E = heating value of fuel

η_t = indicated thermal efficiency

The heating value of the fuel, fuel-air ratio, and the indicated thermal efficiency were essentially constant for the tests performed. Therefore IMEP depends only on the cylinder density of the working fluid.

$$IMEP \propto \rho$$

From which equation (2) becomes

$$Q_r = K_1 (IMEP \times RPM)^{n_1} \qquad (4)$$

Since indicated horsepower is directly proportional to IMEP at constant speed, equation (4) may be written

$$Q_r = K_2 (IHP)^{n_1} \qquad (5)$$

Either equation (4) or (5) is suitable for determining the heat rejection from the working fluid.

Equation (1) was developed by first obtaining an expression for the heat transferred from an air-cooled cylinder to the air and then by making suitable assumptions[4], a relationship was obtained for the flow of heat from the hot gases to the cylinder walls. As the flow of the gases in the cylinder is turbulent,[5] the exponent n_2 should be about 0.8.

Discussion of Results:

Part of the heat dissipated in an internal combustion engine goes to the cooling water, while the rest of it is removed by the oil and crankcase. Judge[3] showed that of the total heat dissipated by an engine, approximately fifty-five per cent was given up to the cooling waster and approximately forty-five per cent to the engine oil and crankcase. The results obtained in this investigation were much lower. Lubricating oil vaporizing from the underside of the piston and the cylinder walls counted for some of this transfer. Direct conduction to the crankcase from which the heat is transferred to both the oil inside and the air outside. accounted for the remainder. Since the crankcase was lagged with insulation, all the heat transferred from it was assumed to go to the lubricating oil, none to the atmosphere.

It was shown earlier that the total heat rejection is an exponential function of IHP and speed (equation 5) and that in all probability the value of the exponent should be less than unity. By reference to the data obtained during this investigation, the general verification of the theory developed above was found.

By taking the test data for all runs made and plotting the heat rejected from the hot engine parts Q_r -- determined by subtracting the motoring heat rejection Q_f from the total rejection Q_T -- as a function of indicated horsepower for the range of speed used on log-log graph paper, Fig. 10 was obtained. Except for deviations at low loads all the curves had approximately the same slope. By replotting the same points for Fig. 11 and weighing the scatter of the data, the general slope, n_2, was obtained. K_5 was then determined (see Appendix). n_2 had a value of approximately one. With the development of Eq.(5) it was stated that the usual value of n_2 would be around 0.8. The reason for the higher value that was found in this investigation may be due to the change in the direction or proportion of heat transfer in cylinders as conditions change. The oil splashed onto the cylinder walls and the change in piston ring action may be the chief influence. The piston may transfer less heat through the rings and more to the oil at higher speeds. As the speed increases, there is probably a change in the type of lubrication with a corresponding change in resistance to heat transfer in the cylinders. As a result of this the piston may ride up on the oil film and a larger percentage of heat will be rejected to the oil.

The resulting empirical formula developed for the heat rejected to the oil due to cylinder gases became for this particular engine

$$Q_r = 0.82(IHP)^{1.1} \text{ Btu/minute} \qquad (5a)$$

Sample calculations in the Appendix show that this equation is within the accuracy of the observed data.

FRICTION HEAT

General Theory

So far, only a portion of the heat rejected to the oil has been discussed. The remaining portion due to friction also can be explained in a manner similar to that of rejection from the hot gases. First, it is necessary to separate the friction into two types -- coulomb and viscous friction.

Coulomb friction is independent of the relative velocity of the sliding surfaces, but viscous friction will increase as some function of the speed. The coefficient of viscous friction is a function of RPM.

$$f = \phi\left(\frac{\mu N}{P}\right)$$

where ϕ = some function

μ = viscosity of lubricant

N = RPM

P = unit bearing pressure

Professors E. S. Taylor and C. F. Taylor[6] assume that the coefficient of friction is nearly proportional to $\frac{\mu N}{P}$.

$$\text{Friction} = fP \propto \mu N$$

and $Q_f = K_3(\text{Friction} \times N)$

or $Q_f = K_4(\mu N \times N)$

For a given lubricating oil at a given condition μ is a constant, and

$$Q_f = K_5 N^2 \text{ (For viscous friction)} \qquad (6)$$

Adding the effect of constant coulomb friction to that of the varying viscous friction should give the heat rejected to the oil due to friction. However, no method of testing has yet been devised to separate coulomb and viscous friction. In addition there exist regions of partial lubrication which do not obey either of the above laws. Consequently the results of this research contain only heat losses due to total friction. Unfortunately, the theoretical formulae developed for determining the coulomb and viscous friction separately cannot be used, but they do give an insight into the nature of the resulting combined friction. There should be some intermediate formulation for total friction heat rejected to the oil lying between those for coulomb and viscous friction, but closer to viscous which is the determining factor since it increases with the square of the speed.

This relationship for friction heat rejected should depend on speed (as is obvious from equation (6) for viscous friction). The exponent which the speed is to should be less than 2, the value for viscous friction alone.

Discussion of Results

The friction heat rejected to the oil and to the cooling water should be the friction horsepower of the engine. It is impossible to separate the amount of friction heat loss to the oil from that to the water jackets. The proportion to each varies under different conditions. Since coulomb friction (mostly piston friction) varies linearly with speed, and viscous friction (bearings) varies with the square of the speed, it follows that a greater proportion of the heat will be rejected to the oil at higher speeds. The part of the friction heat rejected to the oil may be expressed by

$$Q_f = X(FHP)^{n_1}$$

where X is some function of speed and design.

From a comparison with the heat rejected due to cylinder gases, the following equation for heat rejected due to friction will be assumed.

$$Q_f = K_8(FHP)^{n_1} \qquad \text{(for combined friction)} \qquad (8)$$

For the purpose of finding the constants in this relationship, the results of the motoring test were used. The plot of FHP vs. RPM (Fig. 12) indicates a linear proportionality between the two variables. This corroborates the analogy between equations for heat rejected due to friction and that due to the cylinder gases. This also justifies the assumption that total friction heat rejected is some function of FHP.

The plot of the friction heat rejected as a function of friction horsepower (Fig. 13) shows that the function is an exponential. This may be explained by the fact that at higher FHP's there is a greater per cent of the heat rejected to the oil than to the cooling water.

Heat rejected to the oil comes mostly from the bearings, and bearing friction power loss increases with the square of the engine speed. Heat rejected to the cooling water comes mostly from the cylinders where friction power loss increases linearly with the speed (because of partial lubrication of the cylinder walls). It follows that the curve rises slowly at low friction horsepower and more rapidly at higher values of FHP. Thus, the exponential shape of the curve is justified.

From the logarithmic plot of friction heat rejected vs. FHP (Fig. 14), the exponential, n , is found to be 1.5. By dividing values for Q_f by corresponding values for FHP raised to the exponent, 1.5, an average value for K_g was determined. (See Appendix, page 34). Thus the empirical formula developed for the heat rejected to the oil due to friction becomes for this particular engine

$$Q_f = 1.4(\mathrm{FHP})^{1.5} \tag{8a}$$

Sample calculations in the Appendix show that this equation is within reasonable accuracy when compared with the observed data.

The errors resulting from the usual method of measuring friction are explained in a paper by McLeod[2]. From his experiments it was found that there are many errors tending to make the friction measured by the motoring method deviate from the true firing friction. McLeod further stated that these errors were not all in the same direction, but tended to compensate each other making the final deviation small as is shown below.[2]

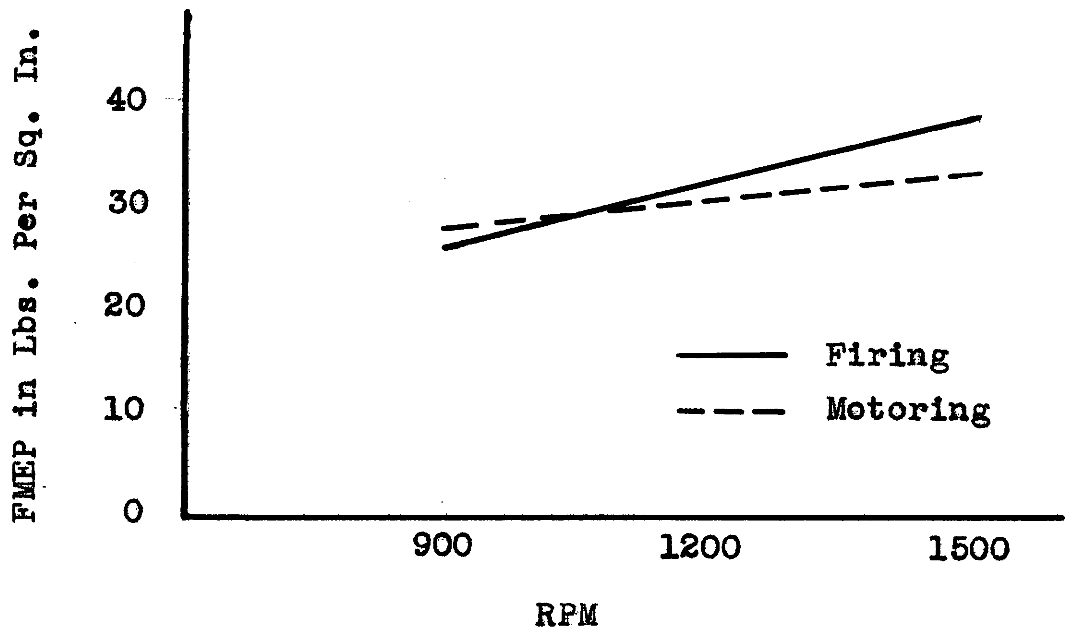

Fig. 1

Performing the motoring tests all together under controlled conditions similar to those present during the firing tests obviated the fime effect on the friction measurements. This time effect arises from the change in conditions which takes place during the delay when a motoring test is taken following each firing test without the operating conditions being controlled. It is presented by McLeod[2] and reproduced below.

The close correlation between the observed data and the calculations obtained by use of the formula derived indicates that the friction measurements were satisfactory for the accuracy desired and that the relationship

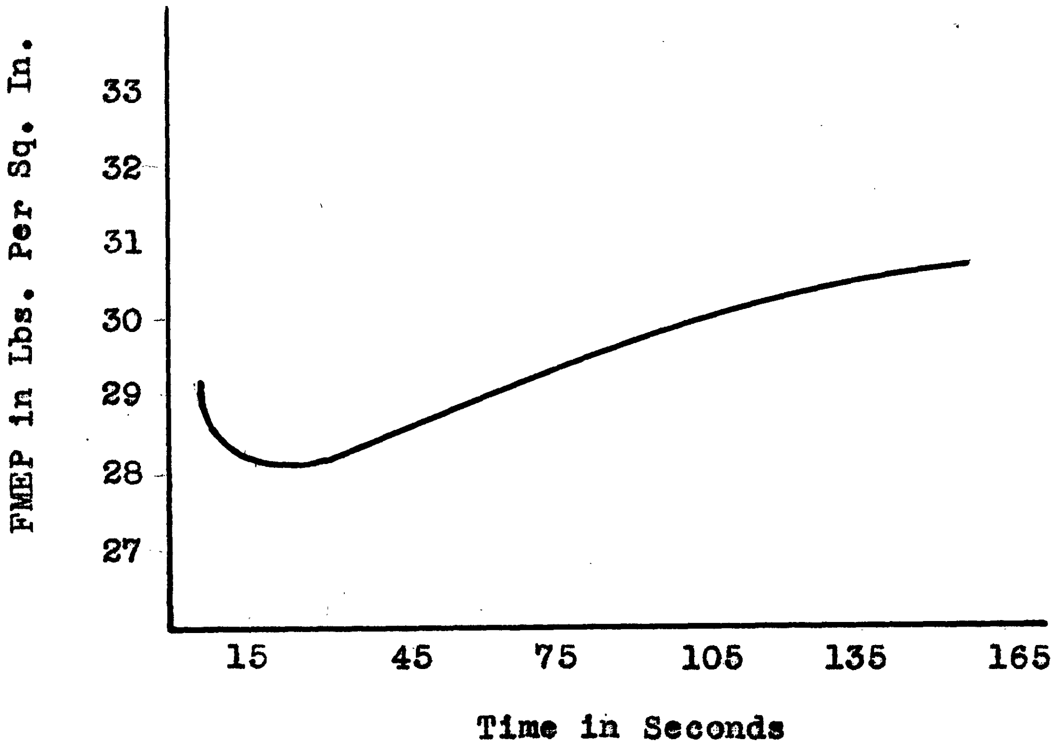

Fig. 2

between heat rejected to the oil and friction horsepower is valid.

TOTAL HEAT REJECTION

In the preceding sections empirical relationships have been developed separately for the two components of the heat rejected to the engine oil. By a combination of the two results the total heat rejection may be found for any speed and load as long as the water jacket and oil temperatures are within the prescribed limits, which cover the normal operating range.

$$Q_t = Q_f + Q_r \qquad \text{Btu/minute} \qquad (9)$$

$$Q_t = K_8(FHP)^{n_1} + K_5(IHP)^{n_2} \qquad \text{"} \quad \text{"} \qquad (10)$$

$$Q_t = 1.4(FHP)^{1.5} + 0.82(IHP)^{1.1} \qquad \text{"} \qquad (11)$$

Equation (11) checks the experimental data obtained during the tests within the limitations of the accuracy of about ten per cent.

An inspection of Figs. 15 to 20 shows the general nature of the total heat rejection over varying ranges of speed and load. The following comments on these curves are offered in explanation:

Increased IMEP, Fig. 15, at constant speeds a slight increase in total heat rejection to the oil. Except for the run at 1200 RPM all the curves have approximately the same slope. The run at 1200 RPM will not be discussed as it obviously contains errors. Previously it was stated that at constant speed, the mechanical friction would be constant but this may not be the case and in all probability the increase in heat rejection was due to the increased piston friction. Under normal operating conditions, the friction of the journal bearings is independent of the bearing pressure and therefore does not vary with varying cylinder pressures. A very large part of the mechanical losses of an engine arise from friction of the pistons and rings. This is due to the poor lubrication conditions combined with high relative velocities and large surfaces in contact. From research done by M.P.Taylor[6], the effect of cylinder pressure on mechanical friction was determined. The nature of the relationship is shown in Fig. 3 below.

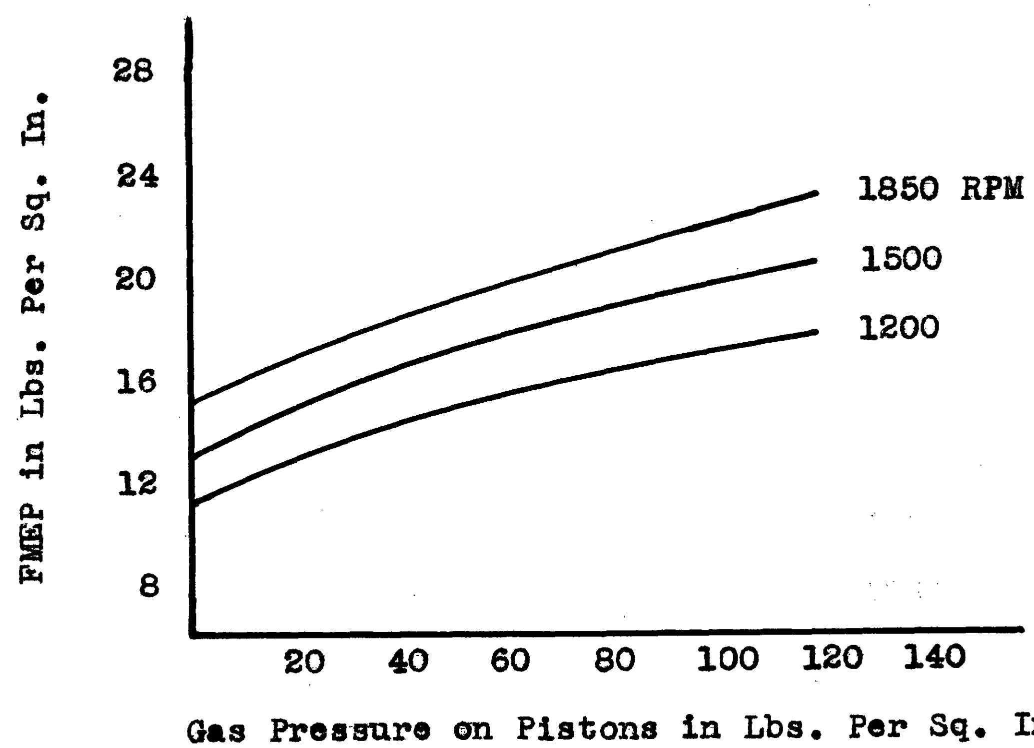

Fig. 3

One of the chief causes for the shape of the above curve is that the cylinder pressure is communicated to the spaces behind the piston rings, thereby increasing the pressure of the rings against the cylinder walls. This relationship is in all probability the main cause of the increase in total heat rejection with increasing load.

The effect of increasing engine speed at constant IMEP's on total heat rejection, Fig. 19, is practically linear. This checks the result obtained by Kaneb and Hoey[1] and has been discussed previously.

Fig. 20 is of more practical value in determining the size of the oil cooling system needed. The relationship between heat rejection and engine speed at constant BHP is in the form of an exponential which follows from the reasoning on pages 14 and 15.

When running at low loads at speeds above 1600 RPM, the total heat transfer to the oil began to decrease to a minimum and then increased as more load was applied. The explanation for this has not been ascertained.

From comparison of the plotted results it is obvious that speed exerts a major influence on heat rejected to the engine oil. Load is relatively unimportant.

FUTURE INVESTIGATIONS

If further investigation is to be done along this line, more accurate temperature control is an absolute necessity. As was shown earlier, the most serious error in all runs was in the temperature reading of the oil. A one-degree error in reading amounted to approximately a ten-per cent error in the heat rejection calculations. Also, time should be allowed for stable operation.

The limitations of load and speed were set by the cradle dynamometer, so if higher speeds are to be investigated this will necessitate a change in the set-up.

For further confirmation of the nature of the heat rejection, it would be necessary to operate the engine under different water jacket and oil temperatures.

A more complete investigation of heat rejected at low loads should be made to determine the reason for the minimum in the power curves at constant speeds.

Until more research is done upon piston and ring friction, it will be impossible to determine more exact relationships.

The empirical formulae for determining the heat transfer to the oil should be checked on other engines before being accepted. Also, these formulae should be modified so that some relationship for geometrically similar engines may be obtained. It would then be possible to predict the heat rejected to the oil for various sizes of similarly designed engines. This would be advantageous in design work.

APPENDIX

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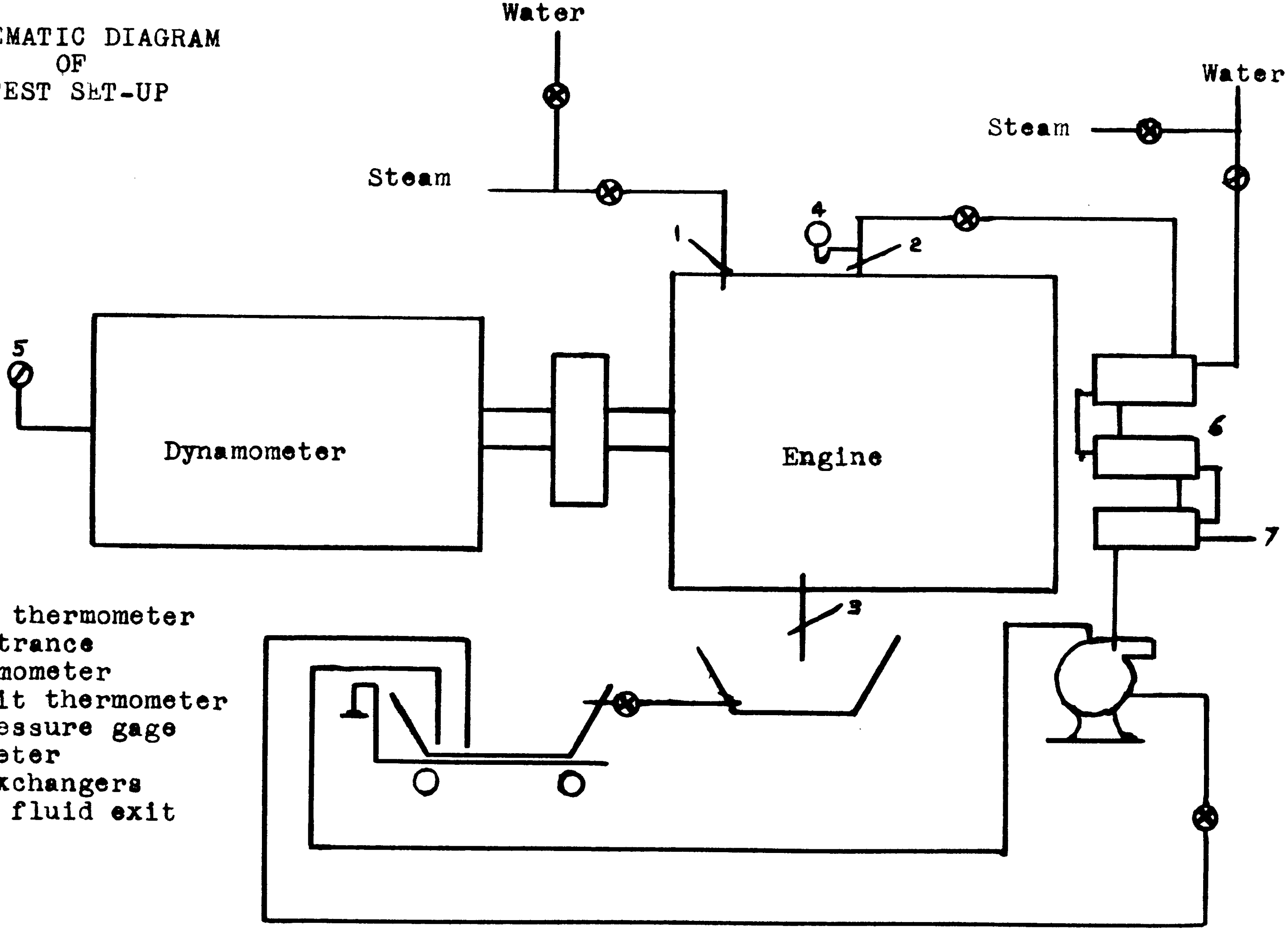
SCHEMATIC DIAGRAM
OF
TEST SET-UP



1 Jacket thermometer
2 Oil entrance thermometer
3 Oil exit thermometer
4 Oil pressure gage
5 Tachometer
6 Heat exchangers
7 Heater fluid exit

FIG. 4

SLOAN LABORATORY
M.I.T.
JANUARY 1944

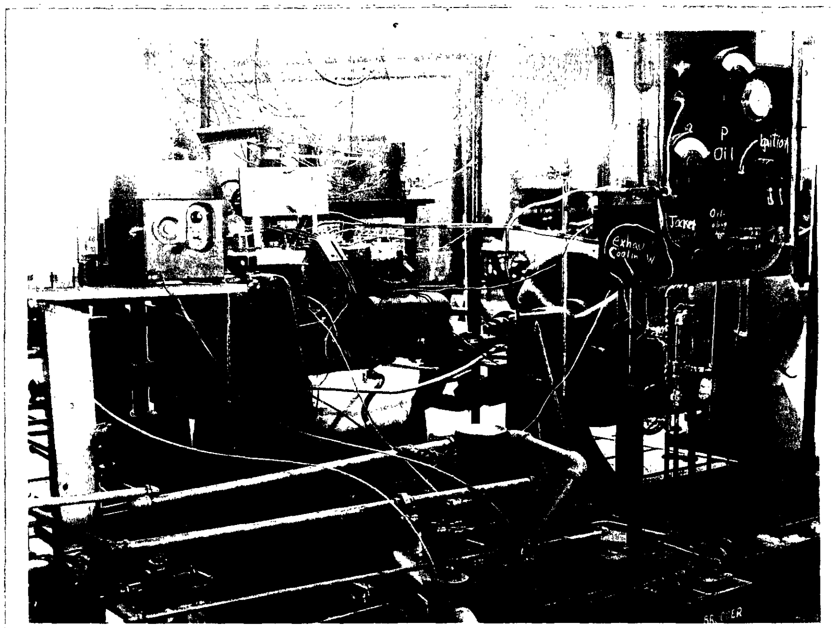


Fig. 5.

View of Engine Showing Control Panel, Asbestos Insulated Crankcase, External Oil System (Pump, Entrance Thermometer, Heaters, Pressure Valve)

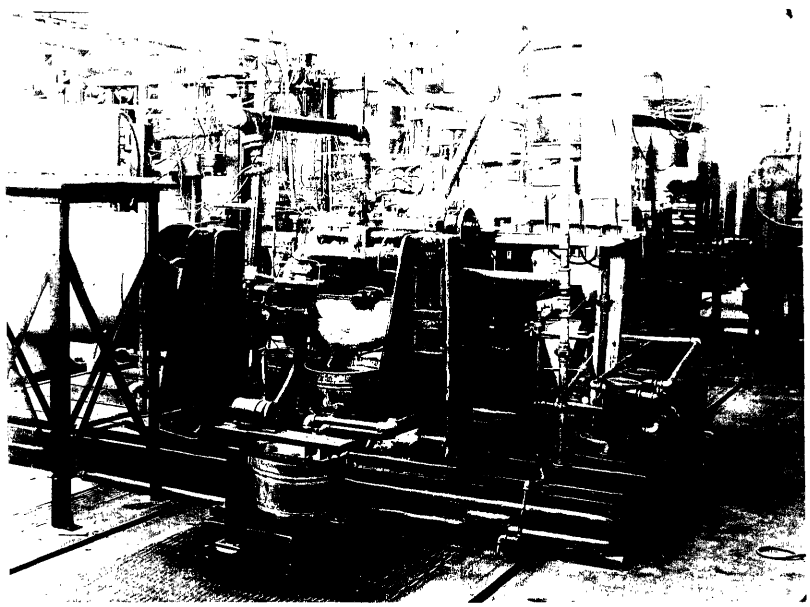


Fig. 6.

View of Engine Showing Flow Measuring System, Oil Pump Suction Line And By-Pass, Exit Thermometer

CALCULATIONS

Heat Transfer

$$Q = W\ C_p \Delta T$$
$$= (10)(0.5)(10)$$
$$= 50\ \text{BTU/min.}$$

where C_p = specific heat of oil at constant pressure
W = rate of oil flow in lbs./min.
ΔT = oil temperature difference, °F.

Brake Horsepower

$$Bhp = \frac{\text{Brake load} \times \text{RPM}}{4000}$$

where Dynamometer Constant = 4000

$$= \frac{(87.5)(1800)}{4000}$$
$$= 39.4$$

Friction Horsepower

$$Fhp = \frac{\text{Friction load} \times \text{RPM}}{4000}$$
$$= \frac{(26.6)(1800)}{4000}$$
$$= 12.0$$

Indicated Horsepower

$$Ihp = Fhp + Bhp$$
$$= 12 + 39.4$$
$$= 51.4$$

Mean Effective Pressure Constant

$$Hp = \frac{\text{Brake load} \times \text{RPM}}{4000}$$

$$Hp = \frac{MEP \times \text{Piston Displacement} \times RPM}{33000 \times 2}$$

$MEP = 0.971 \times$ Brake load

Piston Displacement $= 201 \ in^3$.

Equation (5a)

$$Q_r = K_5 (Ihp)^{n_2}$$

$n_2 = 1.1$ (from Fig. 11)

Q_r	Ihp	$(Ihp)^{1.1}$	$K_5 \ Q_r \div (Ihp)^{1.1}$
21.2	20.5	27.7	0.77
25.0	23.4	32.0	0.78
34.0	30.9	44.0	0.78
40.0	35.3	50.0	0.80
45.0	37.4	54.0	0.83
50.0	39.3	57.0	0.87
57.0	43.6	64.0	0.89

$(K_5)_{ave} = 0.82$

$$Q_r = 0.82 (Ihp)^{1.1}$$

Equation (8a)

$$Q_f = K_8 (Fhp)^{n_1}$$

$n_1 = 1.5$ (from Fig. 14)

Speed	Q_f	Fhp	$(Fhp)^{1.5}$	$K_8 \ Q_f \div (Fhp)^{1.5}$
1200	30.5	6.9	18.0	1.7
1400	35.2	8.5	25.5	1.4
1600	44.0	10.2	32.5	1.4
1800	57.5	12.0	41.6	1.4
2000	77.0	13.9	52.0	1.5
2200	91.0	16.2	65.0	1.4
2400	120	18.4	78.5	1.6

$(K_8)_{ave} = 1.4$

$$Q_f = 1.4(Fhp)^{1.5}$$

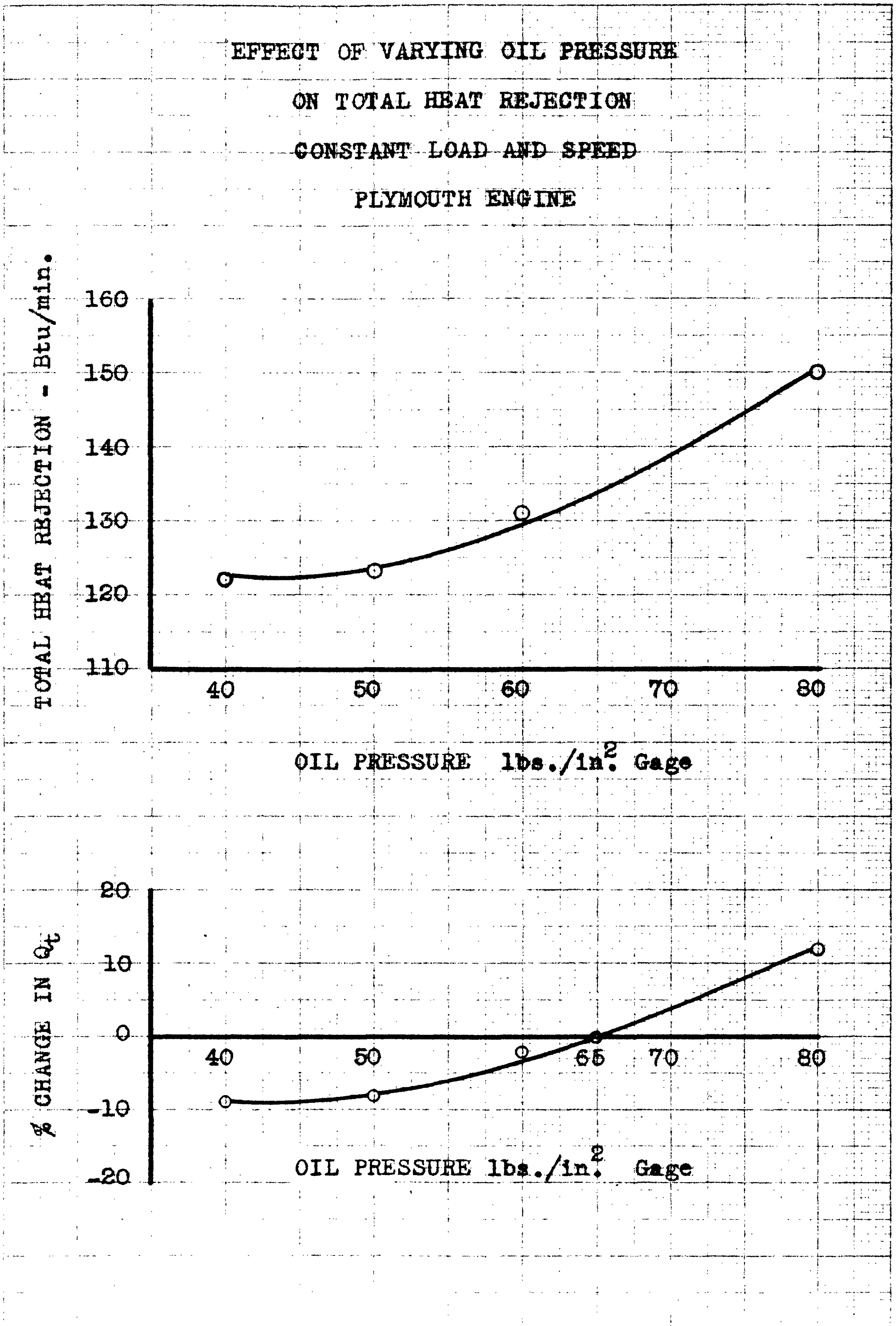
Equation (11)

$$Q_t = 1.4(Fhp)^{1.5} + 0.82(Ihp)^{1.1}$$

Tabulation of total heat rejection as calculated from equation (11) and test results with percentage error.

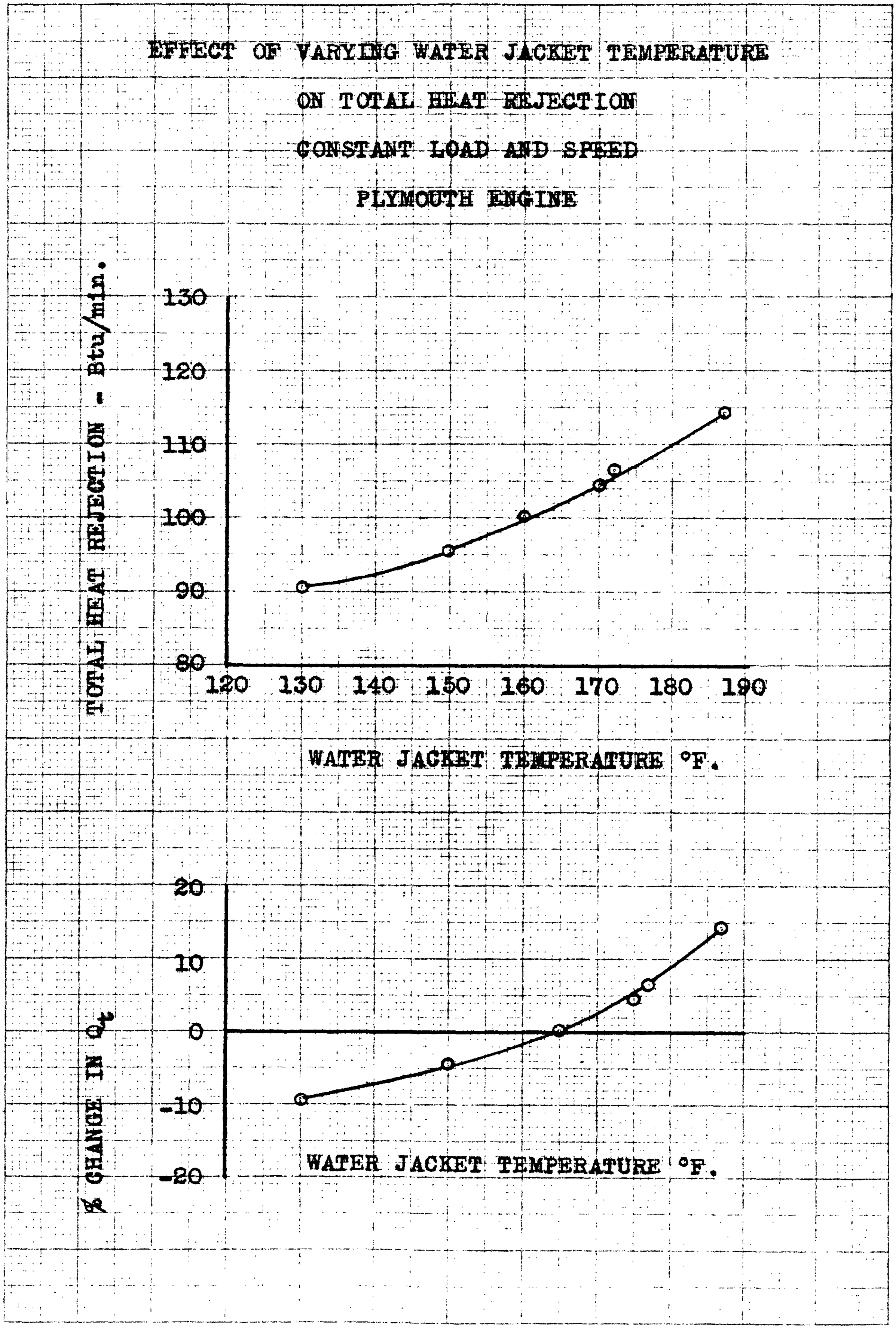
Speed	Ihp	Fhp	Formula Q_t	Test Q_t	% error
1200	23.9	6.9	52.0	51.0	+1.96
1400	34.0	8.6	75.5	84.1	-10.2
1400	21.1	8.7	59.2	57.8	+2.4
1600	31.6	10.2	82.0	85.6	-3.5
1800	21.5	12.0	82.1	81.5	+0.74
2000	17.1	13.9	91.5	103.5	-11.6
2000	35.3	13.9	114.3	117.0	-2.3
2200	41.6	16.2	148	150	-1.3
2400	64.3	18.4	191.4	191	+0.2
2400	43.8	18.4	163.5	170	-3.8

FIG. 7



SLOAN LABORATORY
M.I.T.

FIG. 8



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JANUARY 1944

FIG. 9a

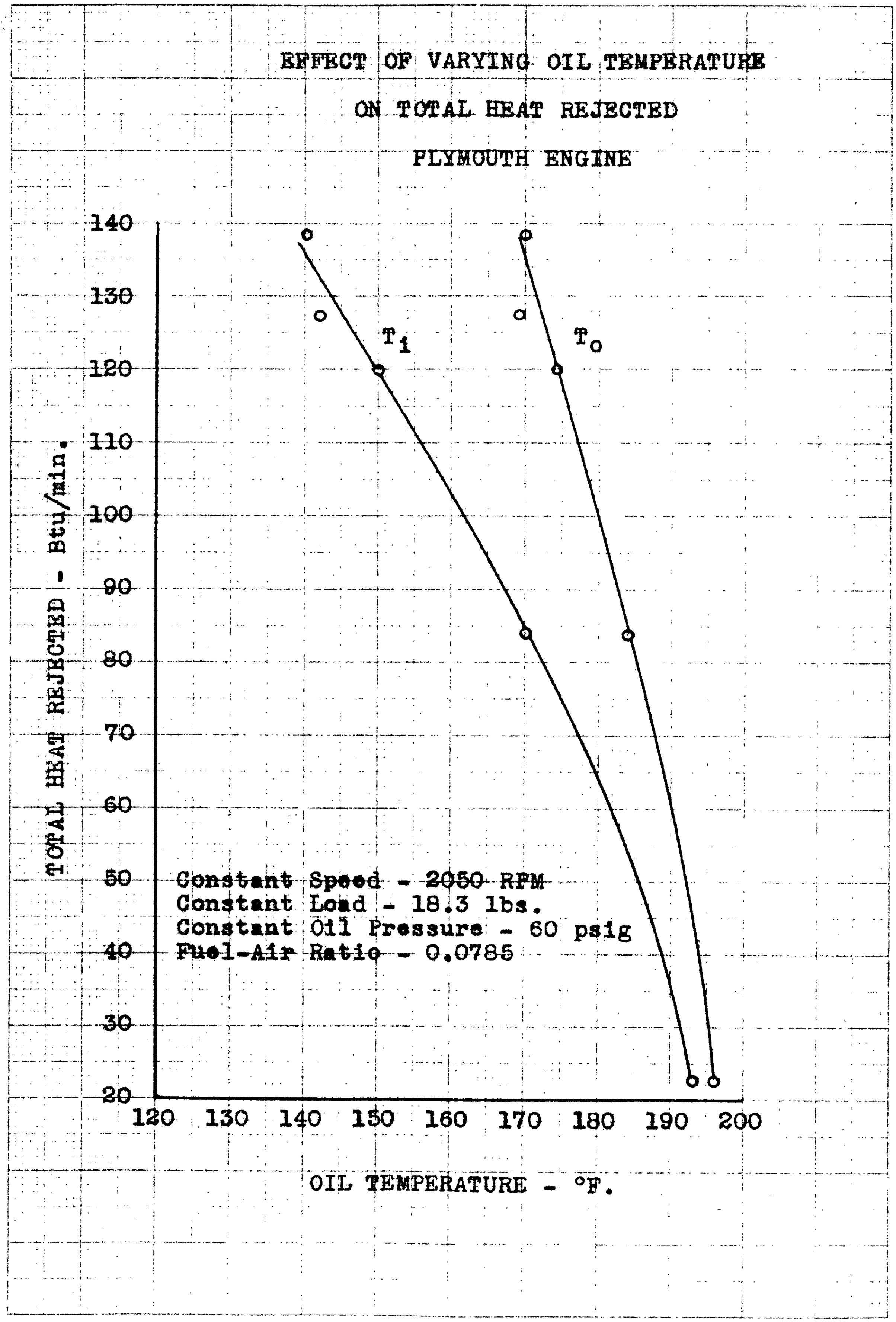

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FIG. 9b

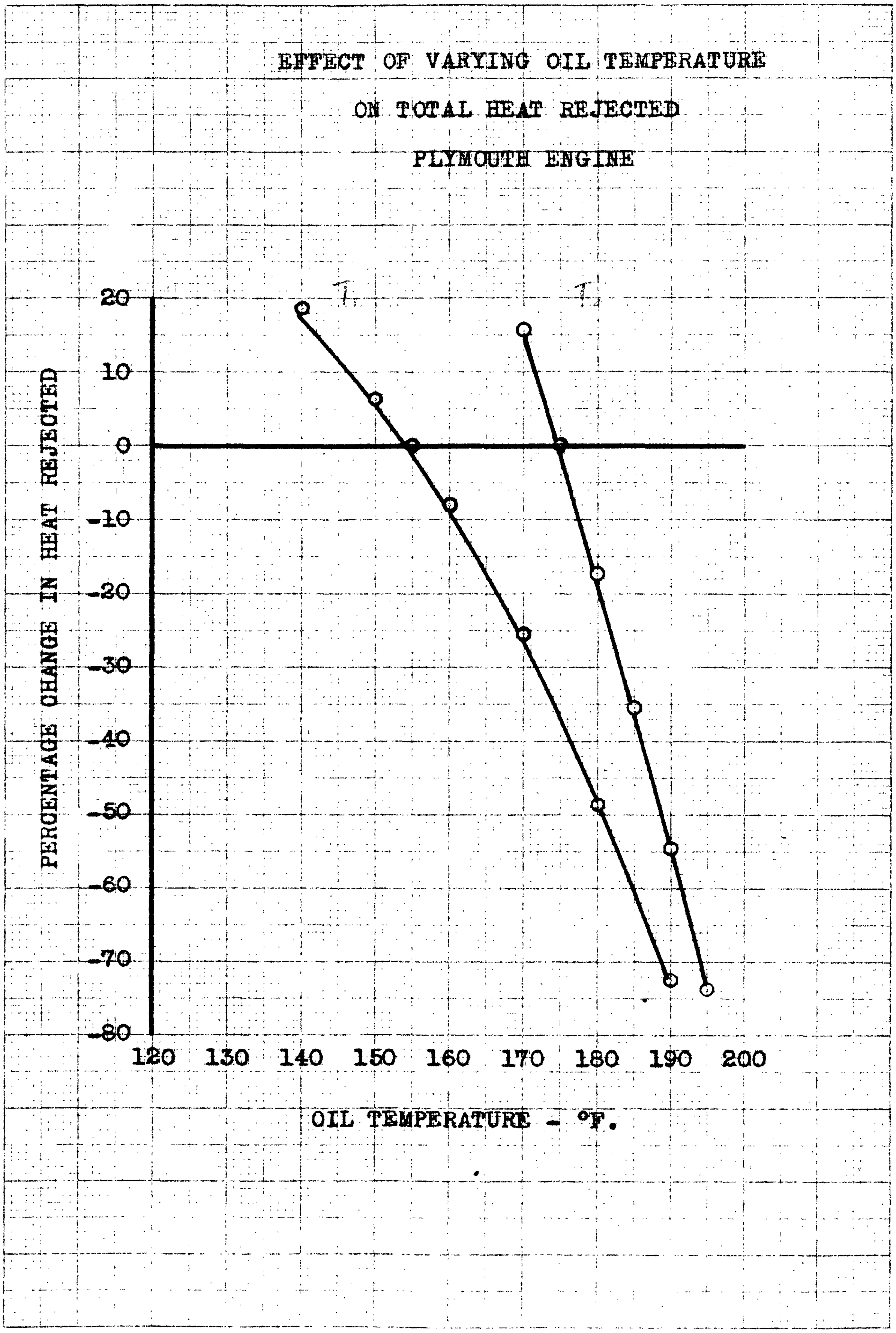

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FIG. 10

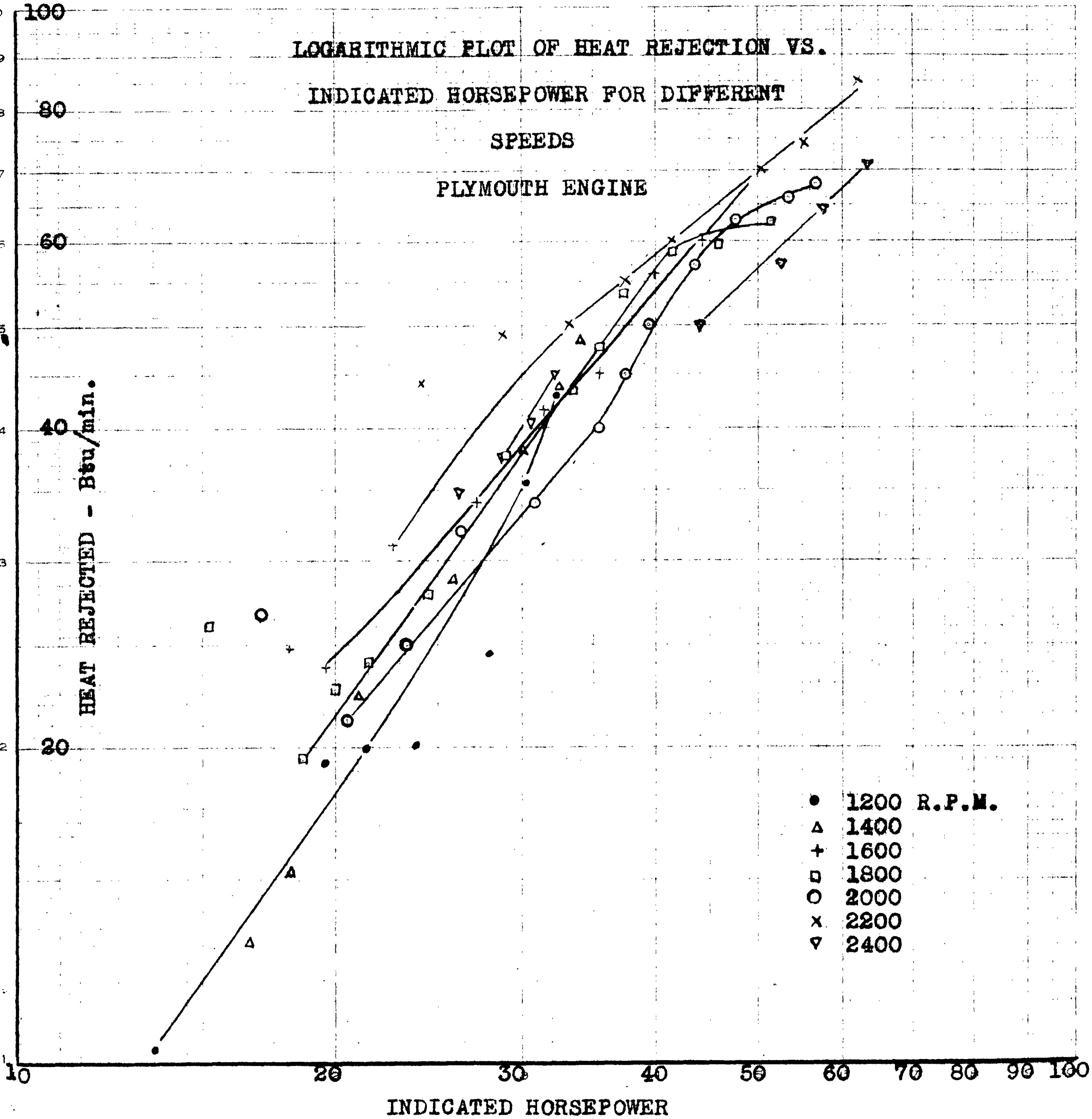

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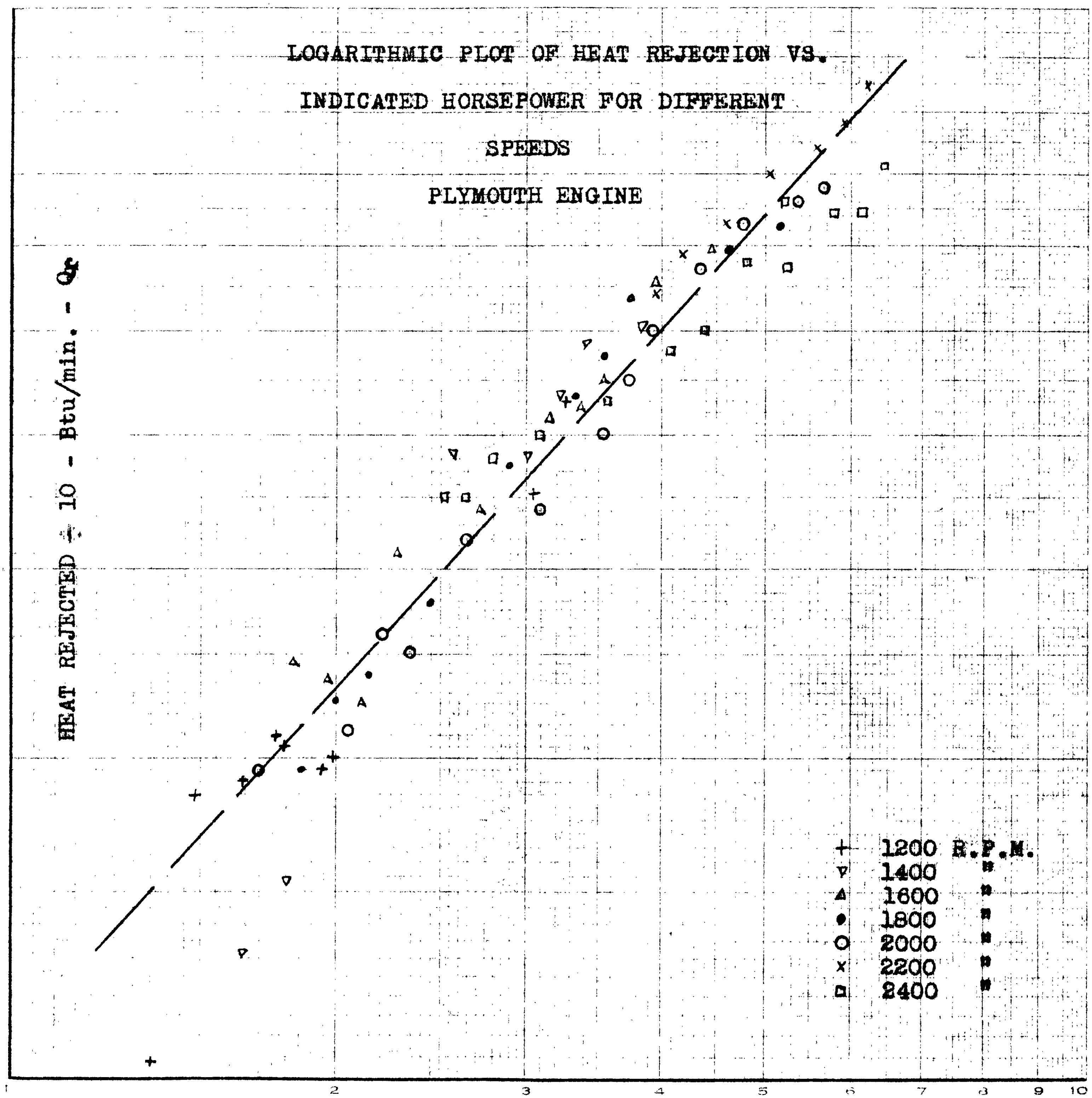
FIG. 11



INDICATED HORSEPOWER ÷ 10

SLOAN LABORATORY
M.I.T
JANUARY 1944

FIG. 12

FRICTION HORSEPOWER VS. ENGINE SPEED

PLYMOUTH ENGINE

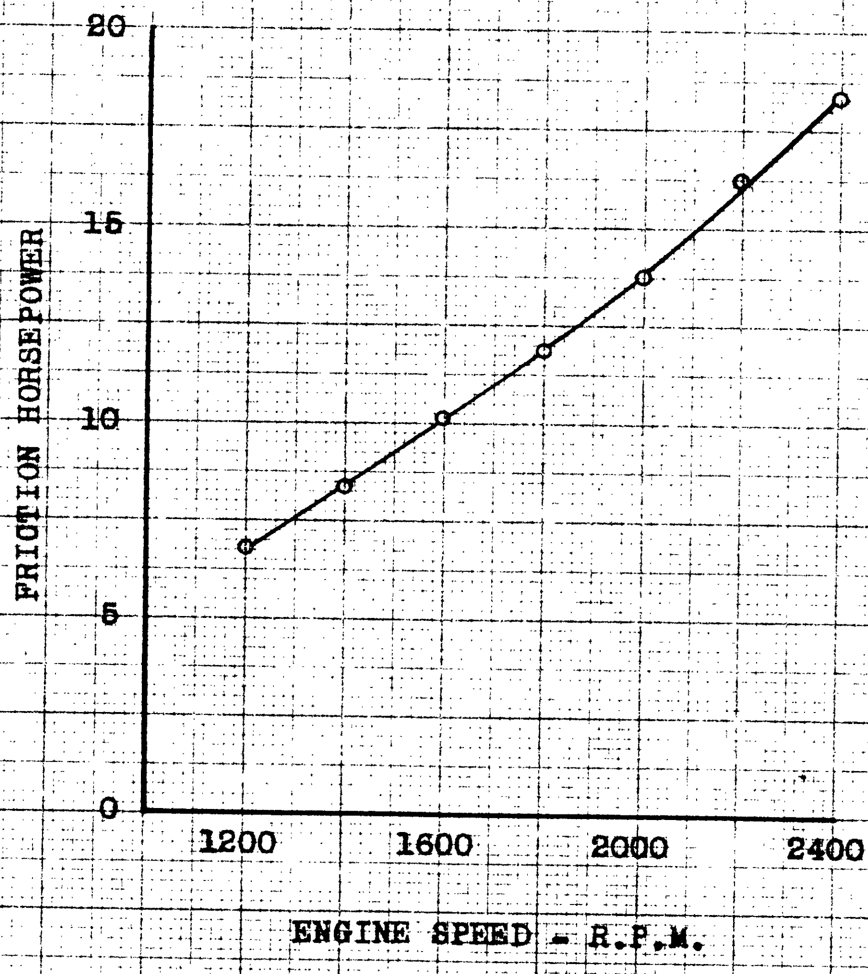


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FIG. 13

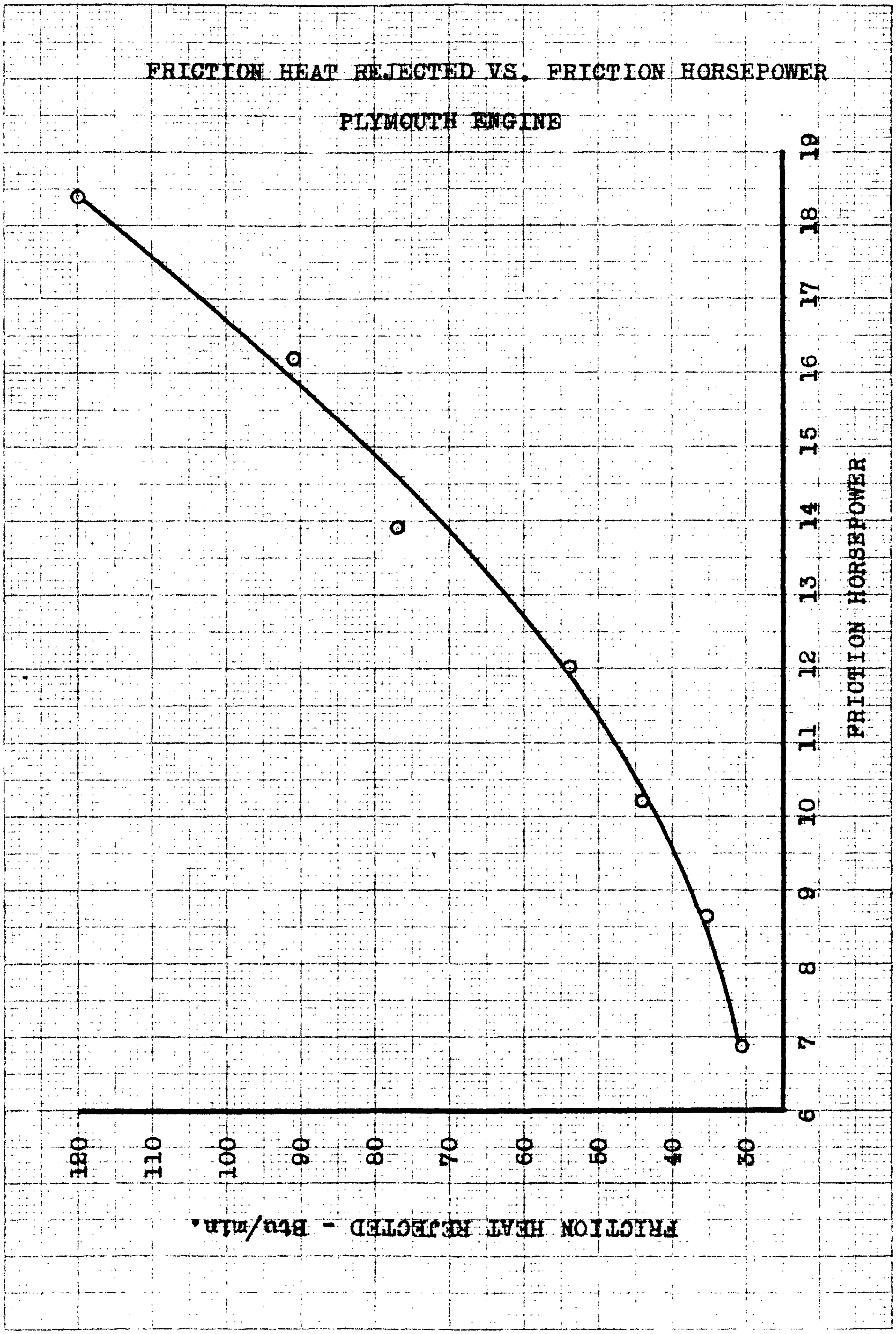

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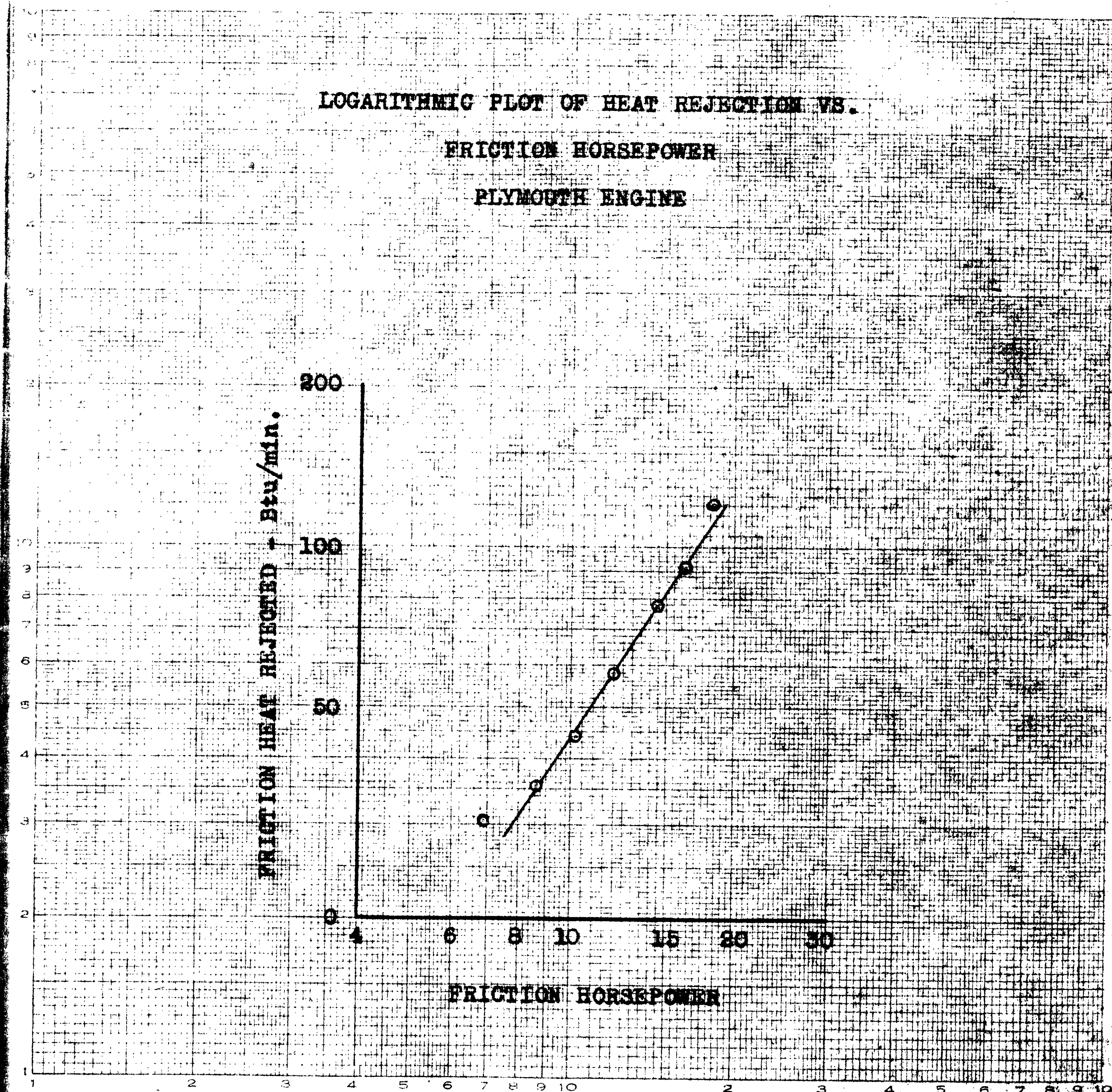
FIG. 14



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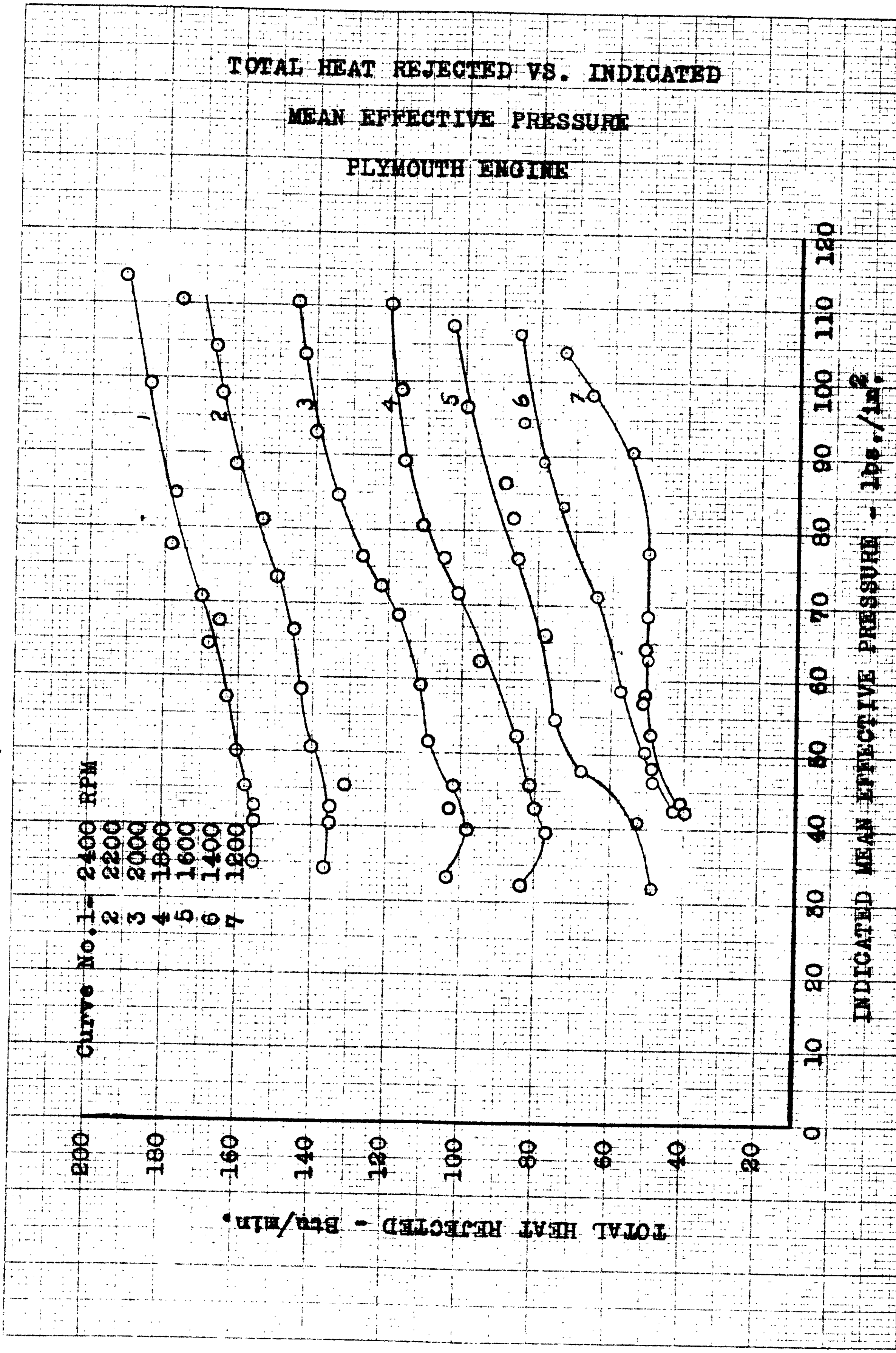
JANUARY 1944

FIG. 15



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M.I.T.
JANUARY 1944

FIG. 16

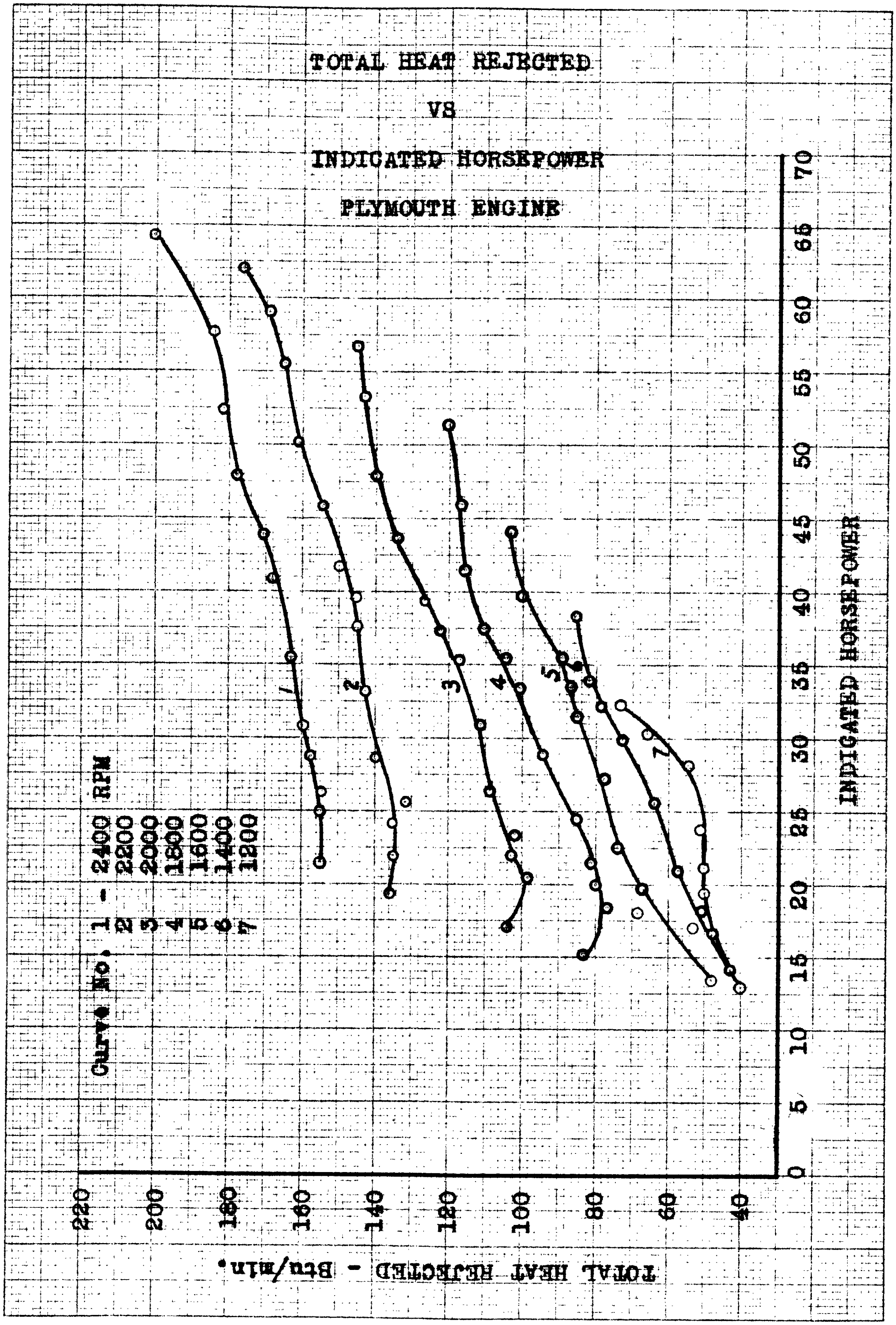

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FIG. 17

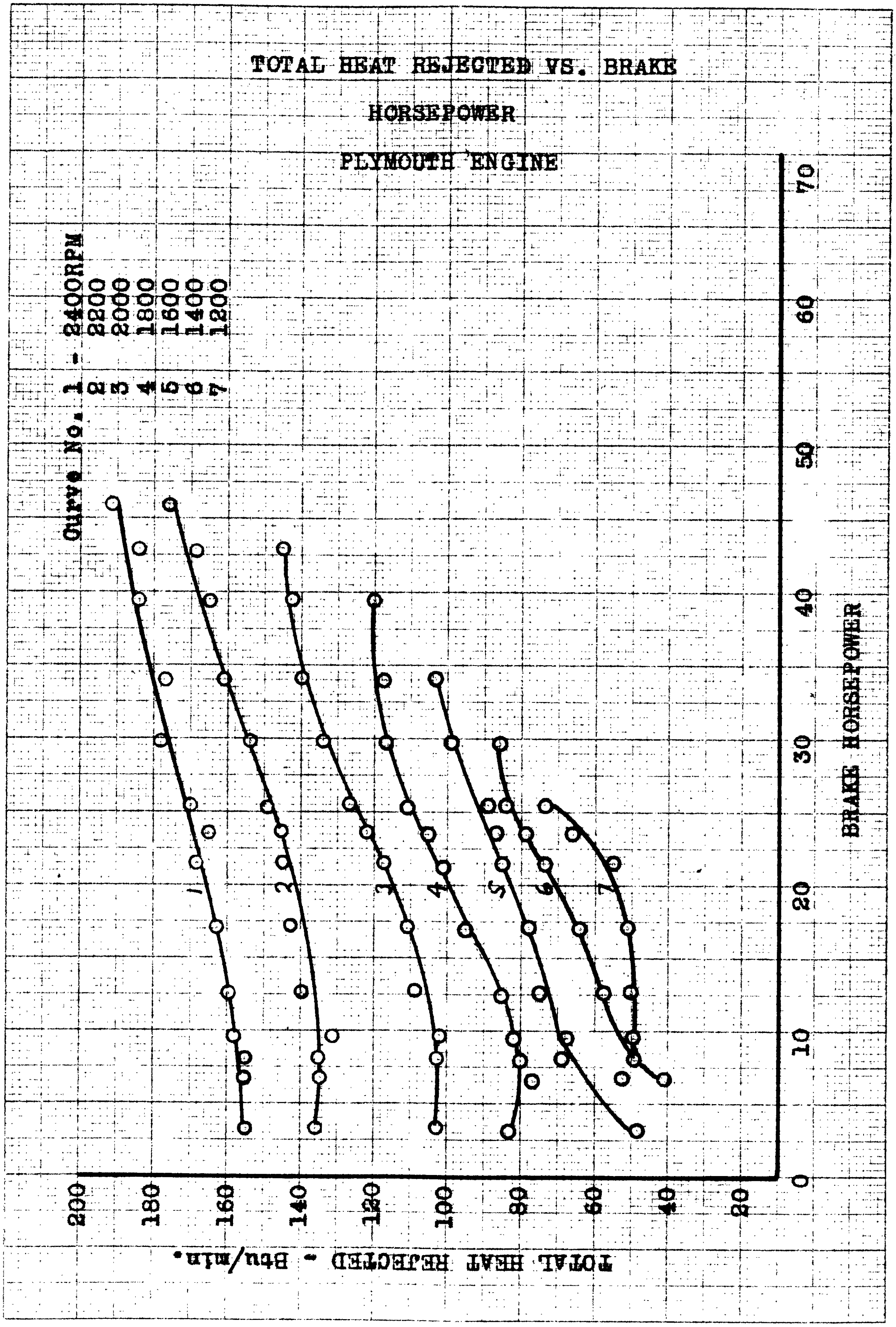

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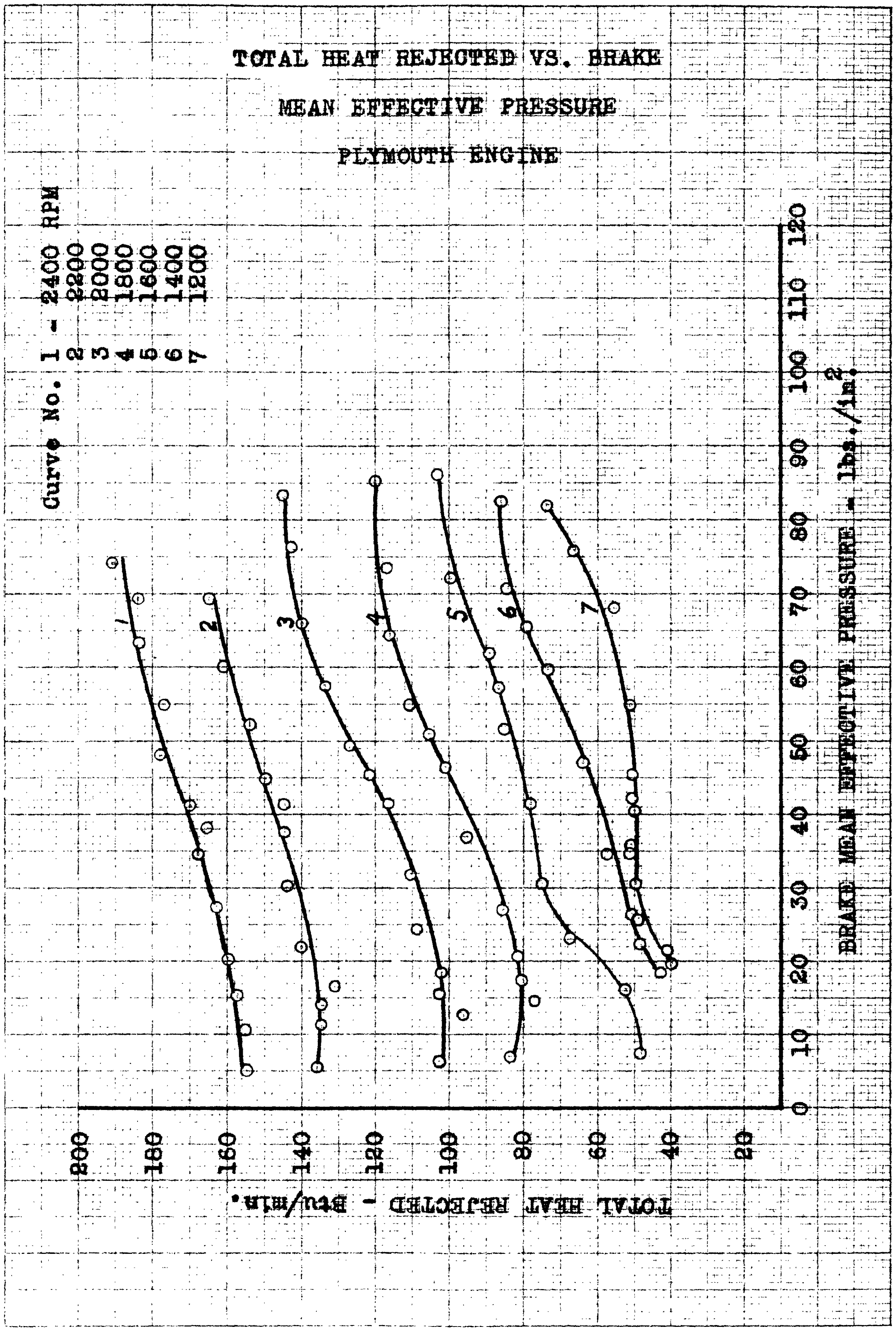
Fig. 18



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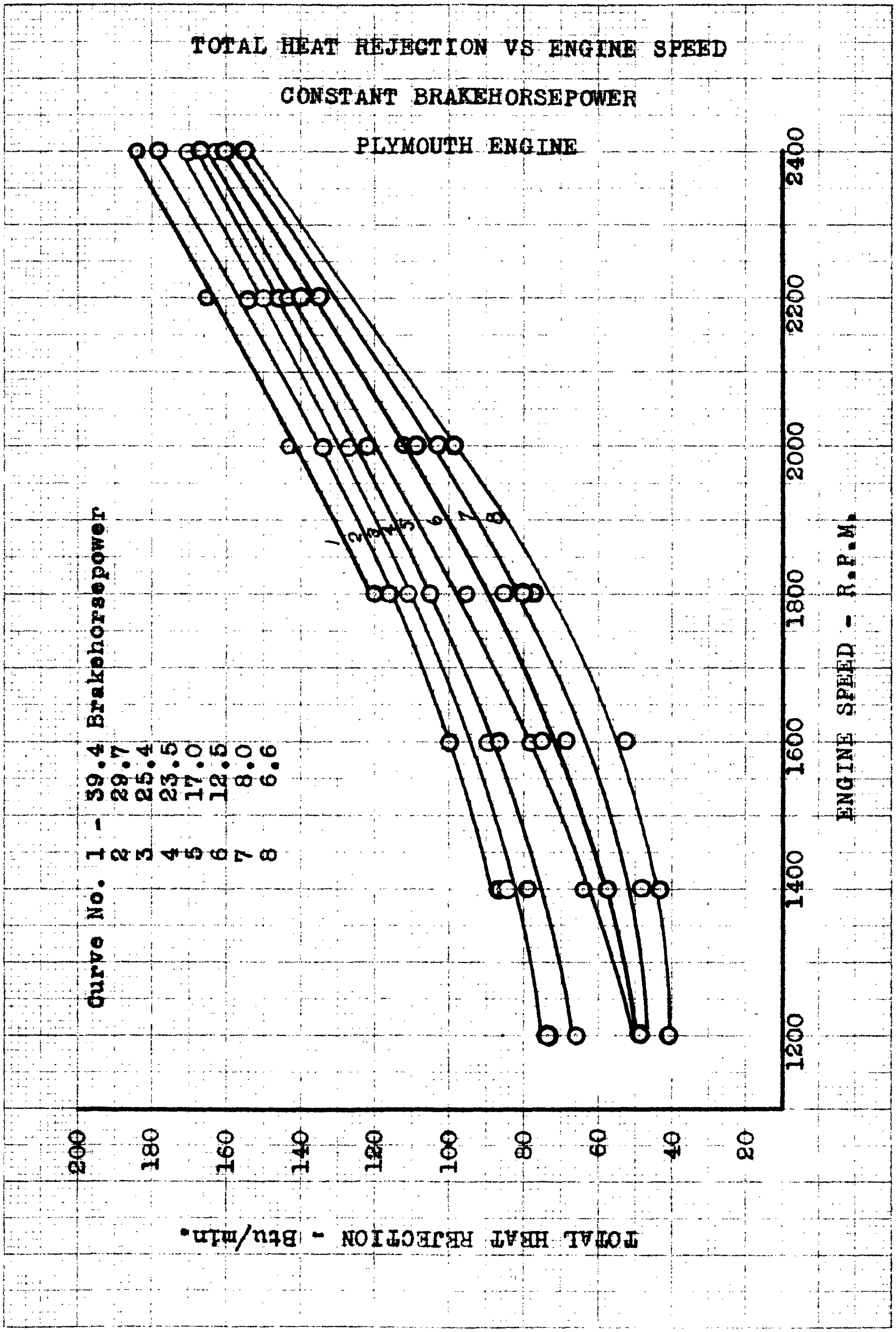
JANUARY 1944

FIG. 19



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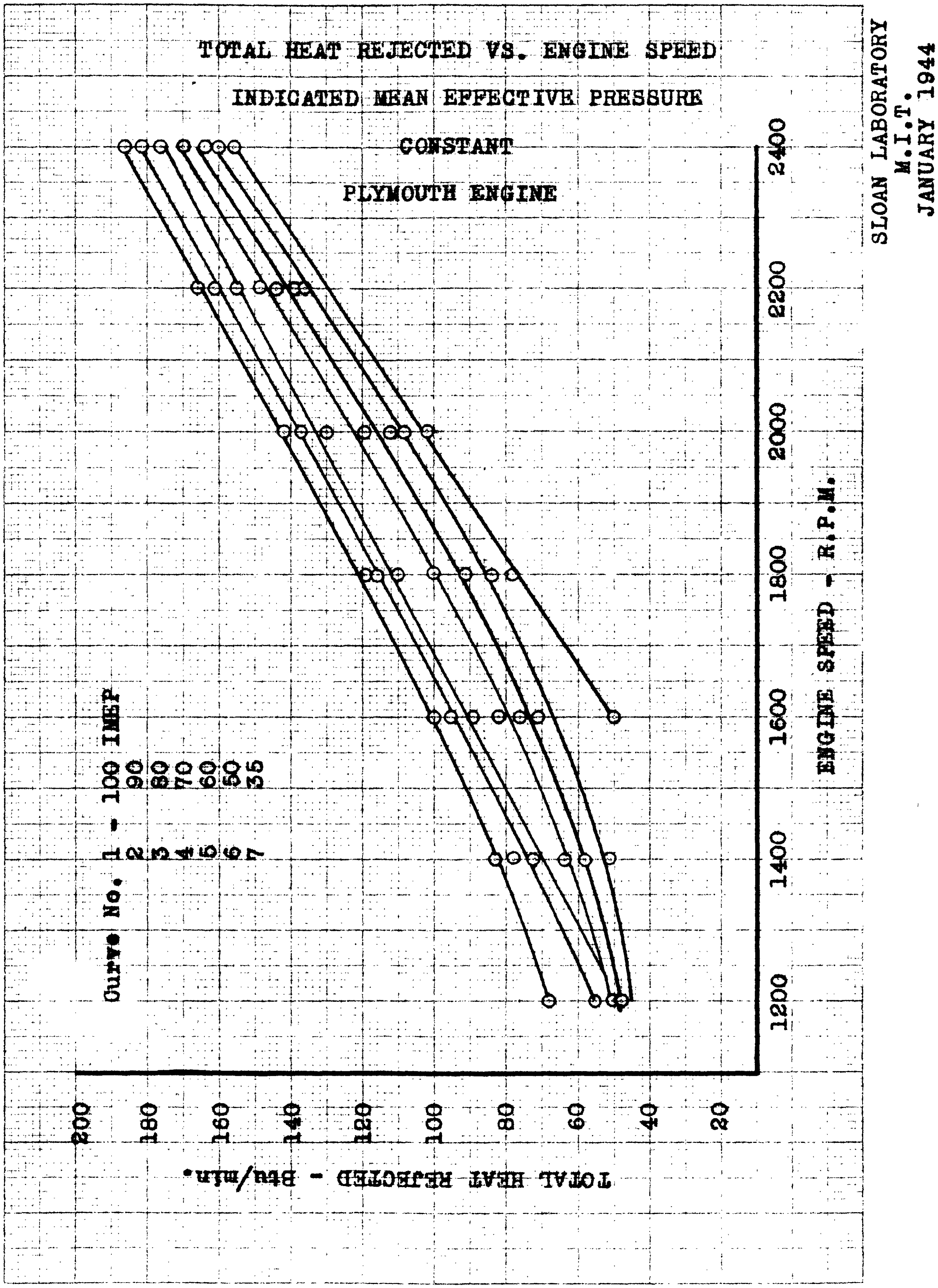
JANUARY 1944

Fig. 20


TOTAL HEAT REJECTED VS. ENGINE SPEED
INDICATED MEAN EFFECTIVE PRESSURE
CONSTANT
PLYMOUTH ENGINE
Curve No. 1 - 100 IMEP
2 90
3 80
4 70
5 60
6 50
7 35
1200 1400 1600 1800 2000 2200 2400
ENGINE SPEED - R.P.M.
200 180 160 140 120 100 80 60 40 20
TOTAL HEAT REJECTED - Btu/min.
SLOAN LABORATORY
M.I.T.
JANUARY 1944

PERIMENT NO. *A-1* TITLE *Water Jacket etc* DATE *January 1944* SLOAN LABORATORY

GINE *1936 Plymouth* FUEL *Automatic* S.G. ______ WET BULB ______ DRY BULB ______

RE *3 1/8* STROKE *4 7/8* COMPRESSION RATIO *6.70* BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS BMEP = B.L. X .971 BHP = B.L. X RPM / 4,980

REMARKS	TIME	RUN	RPM	B.L.	F.L.	TEMP. OIL	TEMP. JAC	OIL PRES.	P_1	P_E	T_1	AIR CONS.	FUEL CONS.	F/A	S.A.	Oil Temp T_i	Oil Temp T_o	ΔT	w	Time	W
	Test No. #1			#			°F									°F	°F	°F	#	sec.	#/min
Varying Water Jacket Temperature		1	1800	48.5			135	62						.0785	B.P.	153.5	170.5	17	10	56.2	10.7
		2					155	62								152	170	18	10	56.6	10.62
		3					165	61								156.5	174.5	18	10	54.0	11.1
		4					175	61								154.5	173.5	19	10	54.4	11.
		5					177	61								156.5	175.5	19	10	53.4	11.2
		6					187	60								155.5	176	20.5	10	52.6	11.2
	Test #2																				
Varying Oil Pressure		1	2050	18.3			[illegible]	40						0.0785	B.P.	142	172	30	13 3/4	79	9.1
		2					[illegible]	50									170	28	13 1/2	71.5	9.
		3					[illegible]	60									169	27	11 3/4	72	9.7
		4					[illegible]	80									[illegible]	26	6 1/2	32	8.
	Tests #3 + 4																				
Varying Oil Temp.		1	2050	18.3			165	60						.0785	B.P.	193	196	3	4.5	18	15.3
		2					168	60								170	184	14	11	55	12.5
		3					168	60								150	174	24	10	60	10.1
		4					165	60								142	169	27	11	70	9.45
		5					167	60								140	170	30	10	65	9.27
	Test #5																				
Changing Spark Advance		1	1800	48.5			155	61						.0785		157	175	18	10	54.0	11.1
		2					155	61								156	174	18	10	54.6	11
		3					155	61								156	174	18	10	54.6	11
		4					155	61								155	173	18	10	56.6	10.7

COURSE
GROUP
NAMES

W	q_T Btu/min	q_F Btu/min	q_R Btu/min	Bhp	Fhp	Ihp	Bmep psi	Imep psi
	40	30.5	9.5	6.15	6.88	13	19.8	42.1
	44.8		10.3	6.60		13.5	21.4	43.
	49		18.5	8.00		14.9	25.9	48.
	49.5		19.1	9.5		16.4	30.7	52.
	51.5		21.0	10.7		17.6	34.8	57.
	51		20.5	11		17.9	35.8	58.
	50		19.5	12.5		19.4	43.6	62.8
1	50.5		20.0	13		19.9	42.1	64.
2	50.5		20.0	14.4		21.3	46.6	68.
2	51		20.5	17		23.7	55.0	77.
0	55		24.5	21.4		28.3	68.2	92.
2	66		35.5	23.5		30.4	76.0	98.
5	73.5		43.0	25.4		32.3	82.0	104.
3	43.0	35.2	7.8	6.6	8.65	14.2	18.4	42.
5	48.3		13.1	8.00		16.6	22.2	46.
1	50.5		15.3	9.5		18.1	26.3	50.
5	57.8		22.6	12.5		21.1	34.6	58.
7	64.0		25.8	17.0		25.6	47.2	71.
8	59.4		—	21.4		30.0	59.3	83.
74	85.6		—	23.5		32.1	65.	89.
5	84.1		48.9	25.4		34.0	72.5	94.
7	85.9		50.7	29.7		38.3	8[illegible].4	10[illegible]
5	79.0		43.8	23.5		32.1	65.2	89.
5	73.5		38.3	21.4		30.0	58.3	83.

PERIMENT NO. S-1 TITLE Constant Speed DATE January 1944 SLOAN LABORATORY

NGINE 1930 Plymouth FUEL Automotive S.G. ______ WET BULB ______ DRY BULB ______

ORE 3¼ STROKE 4⅜ COMPRESSION RATIO 6.7 BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS BMEP = B.L. X 0.971 BHP = $\frac{B.L. \times RPM}{4000}$

OURSE

ROUP

AMES

REMARKS	TIME	RUN	RPM	B.L. #	F.L. #	TEMP. OIL	TEMP. JAC °F	OIL PRES. #/in²	P_I	P_E	T_I	AIR CONS.	FUEL CONS.	F/A	S.A.
	Test #B														
		1	1200	20.5	22.9		165	70						.0785	B.P.
		2		22			165	70							
		3		26.7			162	68							
		4		31.6			167	67							
		5		35.8			168	66							
		6		36.8			160	67							
		7		41.8			164	66							
		8		43.3			160	65							
		9		48			168	65							
		10		56.6			165	64							
		11		70.2			155	64							
		12		78.3			167	62							
		13		84.5			172	61							
	Test #2														
		1	1460	18.9	24.7		155	65						.0785	B.P.
		2		22.9			162	66							
		3		27.1			163	65							
		4		35.6			167	64							
		5		48.6			160	62							
See 11		6		61.0			170	62							
See 10		7		67.1			168	60							
		8		72.5			162	63							
		9		84.7			164	64							
Check 7		10		67.1			160	64							
Check 6		11		61.0			158	65							

RUN	Oil T_i °F	Temp T_o °F	ΔT °F	w #oil	Time sec	w #/min	q_T Btu/min	q_F Btu/min	q_R Btu/min	Bhp	Fhp	Ihp	Bmep psi	Imep psi
Test #B														
1	156	164	8	10	60	10	40	30.5	9.5	6.15	6.88	13	19.9	42.1
2	158	166	8	10	59	10.2	40.8		10.3	6.60		13.5	21.4	43.6
3	154	164	10	10	61.5	9.8	49		18.5	8.00		14.9	25.9	48.1
4	154	164	10	10	60.5	9.9	49.5		19.1	9.5		16.4	30.7	52.9
5	158	168	10	10	58.2	10.3	51.5		21.0	10.7		17.6	34.8	57.0
6	156	166	10	10	58.5	10.2	57		20.5	11		17.9	35.8	58.0
7	156	166	10	10	60	10	50		19.5	12.5		19.4	40.6	62.8
8	156	166	10	10	59.4	10.1	50.5		20.0	13		19.9	42.1	64.3
9	156	166	10	10	59	10.2	50.5		20.0	14.4		21.3	46.6	68.8
10	156	166	10	10	58.5	10.2	51		20.5	17		23.7	55.0	77.2
11	155	166	10	11	60	10.0	55		24.5	21.4		28.3	68.3	92.4
12	155	168	10	13	59	10.2	66		35.5	23.5		30.4	76.0	98.2
13	158	172	10	14	57	10.5	73.5		43.0	25.4		32.3	82.0	104.8
Test #2														
1	156	165	9	10	63	9.53	43.0	35.2	7.8	6.6	8.65	14.2	18.4	42.4
2	156	166	10	9	56	9.65	48.3		13.1	8.00		16.6	22.2	46.2
3	158	168	10	8	47.3	10.1	50.5		15.3	9.5		18.1	26.3	50.3
4	157	168	11	7	40	10.5	57.8		22.6	12.5		21.1	34.6	58.6
5	158	170	12	10	56.2	10.7	64.0		28.8	17.0		25.6	47.2	76.2
6	158	169	11	10	55.5	10.8	59.4		—	21.4		30.6	59.3	83.3
7	156	172	16	10	55.8	10.74	85.6		—	23.5		32.1	65.2	89.2
8	152	168	16	10	57	10.5	84.1		48.9	25.4		34.0	70.5	94.5
9	152	168	16	10	56	10.7	85.9		50.7	29.7		38.3	82.4	106.4
10	151	166	15	10	57	10.5	79.0		43.8	23.5		32.1	65.2	89.2
11	150	164	14	10	57.2	10.5	73.5		38.3	21.4		30.0	59.3	83.3

XPERIMENT NO. S-2 TITLE Constant Speed DATE January 4 1942 SLOAN LABORATORY

NGINE 1936 Plymouth FUEL Automotive S.G. ______ WET BULB ______ DRY BULB ______

ORE $3\frac{1}{8}$ STROKE $4\frac{3}{8}$ COMPRESSION RATIO 6.70 BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS BMEP = B.L. X 0.971 BHP = $\frac{\text{B.L. X RPM}}{4000}$

REMARKS	~~TIME~~	RUN	RPM	B.L.	F.L.	TEMP. ~~OIL~~	TEMP. JAC	OIL PRES.	~~P_I~~	~~P_E~~	~~T_I~~	~~AIR CONS.~~	~~FUEL CONS.~~	F/A	S.A.	Oil T_i	Temp T_o	ΔT	W	Time	w	q_T	q_F	q_R	Bhp	Fhp	Ihp	Bmep	Imep
				#	#		°F	psig								°F	°F	°F	# oil	sec.	#/min	Btu/min	Btu/min	Btu/min					
	Test #8																												
		1	1600	8.0	25.4		170	65						0.0785	B.P.	159	169	10	10	62	9.68	44.4	44.0	4.4	3.2	10.2	13.4	7.27	32.0
		2		16.5			160	67								157	168	11	10	63	9.52	52.5		8.5	6.6		16.8	16.0	47.1
		3		20.0			158	63								158	172	14	10	61	9.84	65.8		24.8	8.0		18.2	19.4	44.1
		4		23.7			160	65								156	170	14	10	62	9.68	67.8		23.8	9.5		19.7	23.0	47.7
		5		31.2			161	66								154	169	15	10	60	10.0	75.6		31	12.5		22.7	30.3	54.7
		6		42.5			163	66								154	169	15	10	58	10.35	78.0		34	17		27.2	41.3	66.0
		7		53.4			163	65								155	171	16	10	56	10.75	85.6		41.6	21.4		31.6	51.9	76.1
		8		58.7			165	63								156	172	16	10	55.4	10.83	86.5		42.6	22.5		33.7	57.0	81.7
		9		63.5			165	64								155	171	16	10	54	11.10	83.6		45	25.4		35.6	61.7	86.4
		10		74.1			155	65								157.5	174.5	17	7	35.8	11.74	99.8		55.2	29.7		37.9	72.5	92.7
		11		85.0			160	65								152	172.5	16.5	8	43	11.20	113.5		59.5	34		44.2	80.6	107.3
	Test #9																												
		1	1800	7.1	26.6		155	68						0.0785	B.P.	153.5	172	16.5	10	59.2	10.1	83.5	57.5	26	3.2	12.0	15.2	6.9	32.3
		2		14.7			150	63								157	172	15	10	58.4	10.25	77.0		19.5	6.6		16.6	14.3	37.7
COURSE		3		17.8			159	68								157	172.5	15.5	10	58.0	10.35	83.1		22.6	8.0		20	17.2	42.7
GROUP		4		21.1			164	67								157	172.5	15.5	10	57.2	10.5	81.5		24	9.5		21.5	20.5	45.9
NAMES		5		27.7			163	65								160	176	16.0	10	56.2	10.7	85.4		27.9	12.5		24.5	26.1	52.3
		6		37.8			159	65								156.5	174	17.5	10	55.0	10.9	95.2		37.7	17		29	36.8	62.2
		7		47.5			155	65								154	172.5	18.5	10	55.1	10.9	101.6		43.5	21.4		33.4	45.1	71.5
		8		52.2			163	65								153	172	19.0	10	54.2	11.1	105.2		47.7	23.5		35.5	50.7	76.1
		9		56.5			155	65								156.5	176	19.5	10	52.5	11.4	111		53.5	25.4		37.4	54.9	80.3
		10		66			168	65								155.5	175.5	20	10	51.8	11.58	116		58.5	29.7		41.7	64.1	89.5
		11		75.5			162	64								156.5	176	19.5	8	40.0	12.0	117		59.5	34		46	73.4	97.6
		12		87.5			160	64								154	174	20	8	40.0	12.0	130		62.5	39.4		51.4	85.0	110.4

XPERIMENT NO. S-3 TITLE Constant Speed DATE January 1944 SLOAN LABORATORY

NGINE 1936 Plymouth FUEL Automotive S.G. ______ WET BULB ______ DRY BULB ______

ORE 3 1/2 STROKE 4 3/8 COMPRESSION RATIO 6.7 BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS BMEP = B.L. X .971 BHP = $\frac{B.L. \times RPM}{4000}$

REMARKS	TIME	RUN	RPM	B.L.	F.L.	TEMP. OIL	TEMP. JAC	OIL PRES.	P_I	P_E	T_I	AIR CONS.	FUEL CONS.	F/A	S.A.	Oil Temp T_i	T_o	ΔT	w	Time	w	q_T	q_r	q_R	Bhp	Fhp	Ihp	BMEP	IMEP
				#	#		°F	psig								°F	°F	°F	#	sec.	#/min	Btu/min	Btu/min	Btu/min				psi	psi
	Test #10																												
		1	2000	6.4	27.9		162	63						.0785	B.P.	154	174	20	10	58	10.35	103.5	77	26.5	3.2	13.9	17.1	6.2	33.3
		2		13.2			160	63								155	174	19	10	58	10.35	98.2		21.2	6.6		20.5	12.8	39.9
		3		16.0			165	63								154.5	174	19.5	10	57	10.52	103		26.0	8.0		21.9	15.5	42.6
		4		19.0			167	63								157	176	19	10	56	10.72	102		25.0	9.5		23.4	18.5	45.6
		5		25.0			160	64								152.5	173.5	21	10	57.6	10.4	109		32.0	12.5		26.4	24.3	51.4
		6		34			160	63								152	172.5	20.5	10	55.4	10.83	111		34	17.0		30.9	32.0	59.[illegible]
		7		42.8			166	64								151	172.5	21.5	10	55.4	10.83	117		40	21.4		35.3	41.6	68.7
		8		47			160	63								151	173	22	10	54.2	11.1	122		45	23.5		37.4	45.6	72.7
		9		50.8			160	64								150.5	173.2	22.7	10	53.4	11.25	127		50	25.4		39.3	49.4	76.3
		10		59.4			160	63								152	175.5	23.5	10	52.6	11.4	134		57	29.7		43.6	57.6	84.[illegible]
		11		68			164	61								153	177	24	10	51.6	11.63	140		63	34.0		47.9	66.0	93.[illegible]
		12		78.8			165	60								153.5	177.7	24.2	10	50.8	11.8	143		66	39.4		53.3	76.5	103.[illegible]
		13		85.8			163	61								154.5	178	23.5	10	48.4	12.4	145		68	42.9		56.8	83.4	110.5
	Test #11																												
		1	2200	5.8	29.5		162	71						.0785	B.P.	153.5	177	23.5	10	52	11.5	136	91	45	3.2	16.2	19.4	5.6	34.[illegible]
OURSE		2		12			165	64								150	174.5	24.5	10	54.6	11	135		44	6.6		22.8	11.7	40.3
ROUP		3		14.5			162	64								151	175.5	24.5	10	54.6	11	135		44	8.0		24.2	14.1	42.7
AMES		4		17.3			165	64								150	174	24	10	55.2	10.9	131		40	9.5		25.7	16.8	45.[illegible]
		5		22.7			164	63								152	177	25	10	53.6	11.2	140		49	12.5		28.7	22.0	50.6
		6		30.9			165	63								150.5	176	25.5	10	53.1	11.3	143		52	17.0		33.2	30.0	58.6
		7		38.9			161	62								152.5	177	24.5	10	51	11.8	145		54	21.4		37.6	37.8	68.4
		8		42.7			162	65								153.5	178	24.5	10	51	11.8	145		54	23.5		39.7	41.5	70.[illegible]
		9		46.2			165	62								154	179	25	10	50	12	150		59	25.4		41.6	44.9	73.5
		10		54			165	62								152	178	25	10	48.6	12.4	154		63	29.7		45.9	52.5	81.1
		11		61.8			163	62								152	178	26	10	48.6	12.4	161		70	34.0		50.2	60.0	88.6
		12		71.5			165	61								152	176	26	10	47.4	12.7	165		74	39.4		55.6	69.5	98.[illegible]
		13		78			160	61								156	181.5	25.5	10	46.5	12.9	169		78	42.9		59.1	75.8	104.[illegible]
		14		83.4			155	61								156	183	27	10	46	13.05	176		85	45.9		62.1	82.0	110.[illegible]

EXPERIMENT NO. S-4 TITLE Constant Speed DATE January 1944 SLOAN LABORATORY

ENGINE 1936 Plymouth FUEL Automotive S.G. ______ WET BULB ______ DRY BULB ______

BORE $3\frac{1}{8}$ STROKE $4\frac{3}{8}$ COMPRESSION RATIO 6.70 BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS BMEP = B.L. X .971 BHP = $\frac{\text{B.L. X RPM}}{4000}$

REMARKS	~~TIME~~	RUN	RPM	B.L.	F.L.	TEMP. ~~OIL~~	TEMP. JAC	OIL PRES.	P_I	P_E	T_I	~~AIR CONS.~~	~~FUEL CONS.~~	F/A	S.A.	Oil T_i	Temp. T_o	ΔT	W	Time	w	q_T	q_F	q_R	Bhp	Fhp	Ihp	BMEP	IMEP
				#	#		°F	psig								°F	°F	°F	# oil	sec	#/min	Btu/min	Btu/min	Btu/min					
	Test	#12																											
		1	2400	5.30	39.7		165	66						0.0785	B.P.	152.5	178	25.5	10	49.4	12.15	155	120	35	3.2	18.4	21.6	5.15	35.6
		2		11.9			160	65								153.5	179	25.5	10	49.2	12.2	155.5		35.5	6.6		25.0	10.7	40.5
		3		13.3			160	65								154	180	25.5	10	49.2	12.2	155		35	8.0		25.4	12.9	42.7
		4		15.9			157	65								154	179.5	25.5	10	48.2	12.45	158		38	9.5		27.9	15.4	45.2
		5		20.8			160	65								154.5	180	25.5	10	47.8	12.5	160		40	12.5		30.9	20.2	50.0
		6		28.3			164	65								154.5	180.5	26	10	47.8	12.5	163		43	17		35.4	27.5	57.3
		7		35.7			163	63								156.5	182.7	26.2	10	46.6	12.9	168		48	21.4		40.8	34.7	64.5
		8		39.2			163	62								157.5	183	25.5	10	46.4	12.35	165		45	23.5		41.9	38.1	67.9
		9		42.4			165	62								158	184	26	9	41.2	13.1	175		50	25.4		43.8	41.1	70.9
		10		49.5			165	62								155.5	182	26.5	9	40.4	13.4	178		58	29.7		48.1	48.1	77.9
		11		56.6			165	59								151.5	178.5	27	7	32.2	13.1	177		57	34		52.4	55.0	84.8
Check on 11		12		56.6			170	61								153	181	28	10	45	13.3	186		66	34		52.4	55.0	84.8
		13		65.5			160	57								155	183	28	5	27.4	13.1	189		64	39.4		57.8	63.6	93.4
		14		71.5			160	58								156	183.5	27.5	6	27	13.35	184		64	42.9		61.3	69.4	99.3
		15	↓	76.4	↓		165	58						↓	↓	156	184	28	5	22	13.65	191	↓	71	45.9	↓	64.3	74.1	103.5

COURSE
GROUP
NAMES

EXPERIMENT NO. M-1 TITLE Motoring DATE January 1944 SLOAN LABORATORY

ENGINE 1936 Plymouth FUEL —— S.G. WET BULB DRY BULB

BORE 3 1/8 STROKE 4 3/8 COMPRESSION RATIO 6.7 BAROMETER (ACT.) (CORR.)

CONSTANTS BMEP = B.L. X .971 BHP = $\frac{\text{B.L. X RPM}}{4000}$

REMARKS	TIME	RUN	RPM	B.L.	F.L.	TEMP. OIL	TEMP. JAC	OIL PRES.	P_I	P_E	T_I	AIR CONS.	FUEL CONS.	F/A	S.A.	Oil T_i	Temp T_o	ΔT	W	Time	w	q_F	Fhp	q_F/Fhp	FMEP
					#		°F	psig								°F	°F	°F	#oil	sec	#/min	Btu/min			
	Test #13 (Motoring)																								
		1	1200		22.9		165	65								154	162	8	10	78.6	7.62	36.5	6.88	4.44	22.2
		2	1400		24.7		163	63								155.5	164	8.5	10	72.6	8.26	35.2	8.65	4.10	24.0
		3	1600		25.4		166	70								159	168.2	9.2	10	62.6	9.6	44	10.2	4.32	24.7
		4	1800		26.6		165	61								161.5	172	11.5	10	60	10	57.5	12.0	4.79	25.4
		5	2000		27.9		165	67								156	171	15	9	52.4	10.3	77	13.9	5.54	27.1
		6	2200		29.5		165	67								160	176	16	7	37	11.4	91	16.2	5.62	28.6
		7	2400		30.7		165	61								160	180	20	5	25	12	120	18.4	6.52	29.8

COURSE

GROUP

NAMES

EXPERIMENT NO. P-1 TITLE Constant Bhp DATE January 1944 SLOAN LABORATORY

ENGINE 1936 Plymouth FUEL Automotive S.G. WET BULB DRY BULB

BORE 3 1/8 STROKE 4 3/8 COMPRESSION RATIO 6.70 BAROMETER (ACT.) (CORR.)

CONSTANTS BMEP = B.L. X .971 BHP = $\frac{\text{B.L.} \times \text{RPM}}{4000}$

REMARKS	~~TIME~~	RUN	RPM	B.L.	F.L.	TEMP. ~~OIL~~	TEMP. JAC	OIL PRES.	P_I	P_E	T_I	~~AIR CONS.~~	~~FUEL CONS.~~	F/A	S.A.	Oil T_i	Temp. T_o	ΔT	W	Time	w	q_T	q_F	q_R	Bhp	Fhp	Ihp
				#	#		°F	psig								°F	°F	°F	#oil	sec.	#/min	Btu/min	Btu/min	Btu/min			
	Test	#14																									
		1	1600	8.0	25.4		170	65						0.0785	B.P.	159	169	10	10	62	9.68	48.4	44.0	4.4	3.2	10.2	13.4
		2	1800	7.1	26.6		155	68								153.5	170	16.5	10	57.2	10.1	83.5	57.5	26.0		12.0	15.2
		3	2000	6.4	27.9		162	63								154	174	20	10	58	10.35	103.5	77.0	26.5		13.9	17.1
		4	2200	5.8	29.5		162	71								153.5	177	23.5	10	52	11.52	136	91.0	45.0		16.2	19.4
		5	2400	5.3	30.7		165	66								152.5	178	25.5	10	47.4	12.15	155	120	35.0		18.4	21.6
	Test	#15																									
		1	1200	22	22.9		165	70						0.0785	B.P.	158	166	8	10	59	10.2	40.8	30.5	10.3	6.6	6.9	13.5
		2	1400	18.9	24.7		145	65								156	165	9	10	62	9.53	43	35.2	13.1		8.65	16.6
		3	1600	16.5	25.4		145	67								157	168	11	10	63	9.52	52.5	44.0	8.5		10.2	16.8
		4	1800	14.7	26.6		[illegible]	68								157	172	15	10	58.4	10.25	77	57.5	19.5		12.0	18.6
		5	2000	13.2	27.9		160	63								155	174	19	10	58	10.35	95.2	77.0	21.2		13.9	20.5
		6	2200	12	28.5		165	64								150	177.5	27.5	10	54.6	11.3	135	91.0	44.0		16.2	22.8
		7	2400	11	30.7		160	65								153.5	179	25.5	10	44.2	12.2	155.5	120	35.5		18.4	25.0
COURSE																											
GROUP																											
NAMES																											
	Test	#16																									
		1	1200	26.7	22.9		162	69						0.0785	B.P.	154	164	10	10	61.5	9.8	49	30.5	18.5	8.0	6.9	14.9
		2	1400	22.7	24.7		162	66								156	166	10	9	56	9.65	48.3	35.2	13.1		8.65	16.6
		3	1600	20	25.4		180	63								158	172	14	10	61	10.4	68.8	44.0	24.8		10.2	18.2
		4	1800	17.8	26.6		159	68								157	172.5	15.5	10	58	10.3	80.1	57.5	22.6		12.0	20.0
		5	2000	16	27.9		165	63								154.5	174	19.5	10	57	10.52	133	77.0	26.0		13.9	21.9
		6	2200	14.5	28.5		162	64								151	175.5	24.5	10	54.6	11	135	91.0	44.0		16.2	24.2
		7	2400	13.3	30.7		160	65								154.5	180	25.5	10	44.0	12.2	155	120	35		18.4	26.4

EXPERIMENT NO. P-2 TITLE Constant Bhp DATE January 1944 SLOAN LABORATORY

ENGINE 1936 Plymouth FUEL Automotive S.G. ____ WET BULB ____ DRY BULB ____

BORE 3 1/4 STROKE 4 3/8 COMPRESSION RATIO 6.70 BAROMETER (ACT.) ____ (CORR.) ____

CONSTANTS BMEP = B.L. X .971 BHP = B.L. × RPM / 4000

REMARKS	~~TIME~~	RUN	RPM	B.L.	F.L.	TEMP. ~~OIL~~	TEMP. JAC	OIL PRES.	~~P_I~~	~~P_E~~	~~T_I~~	~~AIR CONS.~~	~~FUEL CONS.~~	F/A	S.A.	Oil T_i	Temp. T_o	ΔT	W	Time	w	q_T	q_F	q_R	Bhp	Fhp	Ihp
				#	#		°F	psig								°F	°F	°F	# oil	sec	lb/min	Btu/min	Btu/min	Btu/min			
	Test #17																										
		1	1200	31.6	22.9		167	67						0.0785	B.P.	154	164	10	10	62.5	9.92	49.6	30.5	19.1	9.5	6.9	16.4
		2	1400	27.1	24.7		163	65								158	168	10	8	47.3	10.1	52.5	35.2	15.3		8.65	18.1
		3	1600	23.7	25.4		160	65								156	170	14	10	[illegible]	9.56	67.8	44.0	23.8		10.2	19.7
		4	1800	21.1	26.6		164	67								157	172.5	15.5	10	57.2	10.5	81.5	57.5	24		12.0	21.5
		5	2000	19.1	27.9		167	63								157	176	19	10	56	10.72	102	77.0	25		13.9	23.4
		6	2200	17.3	29.5		165	64								150	174	24	10	55.3	10.9	131	91.0	40		16.2	25.7
		7	2400	15.9	30.7		167	65						↓	↓	154	179.5	25.5	10	48.2	12.45	158	120	38	↓	18.4	27.9
	Test #18																										
		1	1200	41.8	22.9		164	66						0.0785	B.P.	154	166	10	10	60	10	50	30.5	19.5	12.5	6.9	19.4
		2	1400	35.6	24.7		167	64								157	168	11	7	43	13.5	57.8	35.2	22.6		8.65	21.1
		3	1600	31.2	25.4		160	66								154	159	15	10	60	10	75	44.0	31.0		10.2	22.7
		4	1800	27.7	26.6		163	65								158	176	18	10	56.2	10.7	85.4	57.5	27.9		12.0	24.5
		5	2000	25.0	27.9		160	64								155.5	173.5	21	10	57.6	10.4	109	77.0	32		13.9	26.4
		6	2200	22.7	29.5		164	63								152	177	25	10	53.6	11.2	140	91.0	49		16.2	28.7
COURSE		7	2400	20.8	30.7		160	65						↓	↓	154.5	180	25.5	10	47.8	12.5	160	120	40	↓	18.4	30.9
GROUP																											
NAMES																											
	Test #19																										
		1	1200	56.6	22.9		165	64						0.0785	B.P.	156	166	10	10	60	10.20	51	30.5	20.5	17.0	6.9	23.9
		2	1400	48.6	24.7		162	62								158	170	12	10	57.2	10.55	64	35.2	28.8		8.65	25.6
		3	1600	42.5	25.4		163	66								154	169	15	10	56	10.35	78	44.0	34.0		10.2	27.2
		4	1800	37.8	26.6		159	65								156.5	174	17.5	10	55	10.9	95.2	57.5	37.7		12.0	29.0
		5	2000	34	27.9		162	62								152	172.5	20.5	10	55.4	10.83	111	77.0	34		13.9	30.9
		6	2200	30.9	29.5		165	63								155.5	176	25.5	10	51	11.3	143	91.0	52		16.2	33.2
		7	2400	28.3	30.7		159	65						↓	↓	154.5	180	26	10	48.6	12.5	183	120	43	↓	18.4	35.4

EXPERIMENT NO. P-3 TITLE Constant Bhp DATE January 1944 SLOAN LABORATORY

ENGINE 1936 Plymouth FUEL Automotive S.G. WET BULB DRY BULB

BORE $3\frac{1}{8}$ STROKE $4\frac{3}{8}$ COMPRESSION RATIO 6.70 BAROMETER (ACT.) (CORR.)

CONSTANTS BMEP = B.L. X .971 BHP = $\frac{\text{B.L. x RPM}}{4000}$

REMARKS	~~TIME~~	RUN	RPM	B.L.	F.L.	TEMP. ~~OIL~~	TEMP. JAC	OIL PRES.	P_I	P_E	T_I	~~AIR CONS.~~	~~FUEL CONS.~~	F/A	S.A.	Oil T_i	Temp T_o	ΔT	W	Time	w	q_T	q_F	q_R	Bhp	Fhp	Ihp
				#	#		°F	psig								°F	°F	°F	# oil	Sec	#/min	Btu/min	Btu/min	Btu/min			
	Test #20																										
		1	1200	72.2	22.9		155	64						.0785	B.P.	155	166	11	10	60	10	55	30.5	24.5	21.4	6.9	28.3
		2	1400	[illegible]	24.7		153	65								150	164	14	10	57.2	10.5	73.5	35.2	38.3		8.65	30.0
		3	1600	53.4	25.4		153	65								155	171	16	10	55	10.7	85.6	44.0	41.6		10.2	31.6
		4	1800	47.5	26.6		155	65								154	172.5	18.5	10	55	10.9	101	57.5	43.5		12.0	33.4
		5	2000	42.8	27.9		155	64								151	172.5	21.5	10	55.4	10.83	117	77.0	40.0		13.9	35.3
		6	2200	38.9	29.5		161	62								152.5	177	24.5	10	51	11.8	145	91.0	54.0		16.2	37.6
		7	2400	35.7	30.7		163	63								156.5	183.7	26.2	10	46.6	12.9	168	120	48.0		18.4	40.8
	Test #21																										
		1	1200	78.3	22.9		167	62						0.0785	B.P.	155	168	13	10	57	10.5	66	30.5	35.5	23.5	6.9	30.4
		2	1400	67.1	24.7		160	64								151	165	15	10	57	10.52	79	35.2	43.8		8.65	32.1
		3	1600	58.7	25.4		165	63								156	172	16	10	55.4	10.83	86.5	44.0	42.6		10.2	33.7
		4	1800	52.2	26.6		163	65								153	172	19	10	59.2	11.1	105.2	57.5	41.7		12.0	35.5
		5	2000	47.0	27.9		160	63								151	173	22	10	54.2	11.1	122	77.0	45.0		13.9	37.4
		6	2200	42.2	29.5		162	65								153.5	178	24.5	10	51	11.9	145	91.0	54.0		16.2	39.7
COURSE GROUP NAMES		7	2400	39.2	30.7		163	62								157.5	183	25.5	10	46.4	12.95	165	120	45.0		18.4	41.9
	Test #22																										
		1	1200	84.5	22.9		172	61						0.0785	B.P.	158	172	14	10	57	10.5	73.5	30.5	43.0	25.4	6.9	32.3
		2	1400	72.5	24.7		162	63								156	172	16	10	57	10.53	84.1	35.2	48.9		8.65	34.0
		3	1600	63.5	25.4		155	64								155	171	16	10	54	11.1	89	44.0	45.0		10.2	35.6
		4	1800	56.5	26.6		155	65								156.5	176	19.5	10	52.5	11.4	111	57.5	53.5		12.0	37.4
		5	2000	50.8	27.9		158	64								153.5	176.2	22.7	10	53.4	11.25	137	77.0	50.0		13.9	39.3
		6	2200	46.2	29.5		165	62								154	179	25	10	50	12	150	91.0	59.0		16.2	41.6
		7	2400	42.4	30.7		165	62								158	184	26	9	46.2	13.1	170	120	50.0		18.4	43.8

EXPERIMENT NO. P-4 TITLE Constant Bhp. DATE January 1944 SLOAN LABORATORY

ENGINE 1936 Plymouth FUEL Automotive S.G. ______ WET BULB ______ DRY BULB ______

BORE $3\frac{1}{8}$ STROKE $4\frac{3}{8}$ COMPRESSION RATIO 6.70 BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS BMEP = B.L. X .971 BHP = $\frac{B.L. \times RPM}{4000}$

REMARKS	~~TIME~~	RUN	RPM	B.L.	F.L.	TEMP. ~~OIL~~	TEMP. JAC	OIL PRES.	P_I	P_E	T_I	~~AIR CONS.~~	~~FUEL CONS.~~	F/A	S.A.	Oil T_i	Temp T_o	ΔT	W	Time	w	Q_T	Q_F	Q_R	Bhp	Fhp	Ihp
				#	#		°F	psig								°F	°F	°F	# Oil	Sec	#/min	Btu/min	Btu/min	Btu/min			
	Test #23																										
		1	1400	84.7	24.7		164	64						.0785	B.P.	152	162	16	10	56	10.7	85.9	35.2	50.7	27.7	8.65	38.5
		2	1600	79.1	25.4		155	68								157.5	174.5	17	7	35.8	11.74	99.8	44.0	55.8		10.2	39.9
		3	1800	66	26.6		162	65								155.5	175.5	20	10	51.8	11.58	116	57.5	58.5		12.0	41.7
		4	2000	53.4	27.9		160	63								152	175.5	23.5	11	52.6	11.43	134	77.0	57.0		13.9	43.6
		5	2200	54	29.5		165	62								152	178	25	10	48.6	12.36	154	91.0	63.0		16.2	45.9
		6	2400	49.5	30.7		165	62								155.5	182	26.5	9	40.4	12.4	178	120	58.0		18.4	48.1
	Test #24																										
		1	1600	85.0	25.4		160	65						.0785	B.P.	152	170.5	18.5	8	43	11.2	103.5	44	59.5	34	10.2	44.2
		2	1800	75.5	26.6		162	64								156.5	176	19.5	8	49	12	117	57.5	59.5		12.0	46
		3	2000	68	27.9		164	61								153	177	24	10	54.6	11.63	140	77.0	63.0		13.9	47.9
		4	2200	61.8	29.5		163	62								152	178	26	10	48.6	12.36	135.5	91.0	70.0		16.2	50.2
		5	2400	56.6	30.7		165	57								154.5	172.5	27	7	32.2	13.1	177	120	57.0		18.4	52.4
COURSE																											
GROUP	Test #25																										
NAMES		1	1800	87.5	26.6		160	64						.0785	B.P.	154	174	20	8	49	12	120	57.5	62.5	39.4	12.0	51.4
		2	2000	78.8	27.9		165	60								153.5	177.7	24.2	10	50.8	11.8	143	77.0	66.0		13.9	53.3
		3	2200	71.5	29.5		165	61								152	176	24	10	49.4	12.7	165	91.0	74.0		16.2	55.6
		4	2400	65.5	30.7		160	57								155	183	28	6	27.4	13.1	184	120	64.0		18.4	57.8
	Test #26																										
		1	2000	85.8	27.9		163	61						.0785	B.P.	154.5	178	23.5	13	48.4	12.4	145	77.0	68	42.9	13.9	56.8
		2	2200	79	29.5		160	61								156	181.5	25.5	10	43.5	12.9	159	91.0	78		16.2	59.1
		3	2400	71.5	30.7		160	58								156	183.5	27.5	6	27	13.35	184	120	64		18.4	61.3