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It is recognized that motor oil, in addition to lubricating, functions also as a coolant. The proportion of the total engine cooling carried by the oil varies from 2% or 3% in small automotive engines to nearly 20% in some large, stationary engines.

The purpose of this work is to determine the effect of operating variables on the amount of heat rejected to the lubricating oil, The variables studied were oil temperature, speed, spark advance, air-fuel ratio, compression ratio, and manifold pressure. The scheme developed for making the tests was to circulate oil through the crankcase of an engine, measuring its rate of flow and the temperature of the oil both as it entered and as it left the crankcase. From these measurements it was possible to calculate the amount of heat absorbed by the oil. The operating variables mentioned above were closely controlled during the runs.

The engine used was a small, one cylinder Delco, of the type used in farm lighting installations. It was fitted with controls and instruments for maintaining and measuring the variables under investigation.

The results of the tests can be summed up as follows:
The heat transfer to the lubricating oil decreases rapidly with increasing oil temperature in a straight line relationship, increases with speed in a straight line relationship, and decreases as spark is advanced, rapidly at first but leveling off with greatly advanced spark. The heat transfer rises to a maximum value as air-fuel ratio is increased to 12.6, falling off rapidly beyond that point. It decreases gradually with increasing compression ratio, and shows no variation with manifold pressure within the limits of sensitivity of the apparatus used in this investigation.

HEAT TRANSFER TO LUBRICATING OIL OF AN INTERNAL COMBUSTION ENGINE WITH VARYING OPERATING CONDITIONS

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HEAT TRANSFER TO LUBRICATING OIL OF AN INTERNAL COMBUSTION ENGINE WITH VARYING OPERATING CONDITIONS

I. INTRODUCTION

It is the tendency of most people to regard motor oil only as a lubricant. While lubrication is unquestionably the primary function of an oil, its secondary and very important use as a coolant must not be overlooked.

In any internal combustion engine, the energy inherent in the fuel appears in two forms; part of it is manifested as useful work, and the remainder as heat. The heat results partly from the burning of the fuel and partly from the work lost in friction. A typical heat balance, as measured in the laboratory and based on heat input in the fuel as 100% (1), may be tabulated as follows:

To work		24.0%
To exhaust		27.5%
To cooling		
To oil	3.5%	
To water, radiation and unaccounted for	48.5%	
Total to cooling		48.5%
Total		100.0%

It is with the 3.5% absorbed by the oil that this paper is concerned.

That oil has an appreciable effect as a coolant is agreed by many authorities (2,3,7,9,10). In a bearing

lubricated by a thick film of oil, the work of shearing the oil and rubbing the various layers upon one another is transformed into heat. Obviously this heat must be dissipated in some manner or the bearing will become infinitely hot (4). If the bearing is open to the air, circulating air currents may be sufficient to carry away the heat. However, enclosed bearings, such as those in an automobile engine, must rely on the circulating oil to keep the temperature at a safe level. In practice, the oil circulates through the bearing, rises in temperature, and runs down into the crankcase, through the bottom and sides of which the heat is dissipated (7). The temperature of the oil varies considerably with the operating conditions.

In some of the larger engines, for instance, diesel engines with pistons 12 inches and more in diameter, it is necessary to apply some sort of cooling to the pistons as well as to the cylinder walls. It is possible to circulate water through the pistons, but this necessitates objectionable packing glands to prevent contamination of the oil with water. Therefore it is the usual practice to cool the pistons by means of oil circulated through them (2).

The amount of heat rejected to the oil varies widely with the speed and design of the engine (10). A

good average value would be about 2 Btu per minute per horsepower. For the average automobile engine it would be about 1.5 Btu per minute per horsepower, w hile for the higher horsepower diesel engines the rate at which heat is absorbed by the oil runs about 2.5 Btu per minute per horsepower. On the larger engines such as described in the preceding paragraph, where the piston is oil cooled, the figure may be as high as 6 Btu per minute per horsepower. The following table showing the variation of heat to oil with increasing speed (9) was probably taken from work done on that type of engine.

Speed, RPM	Heat to oil Btu/min/bhp	Total heat to oil and water, Btu/min/bhp
800	2.9	42.5
1000	4.07	40.0
1200	5.22	36.0
1400	5.17	34.5
1600	4.92	34.0
1800	6.03	35.0
2000	6.18	36.5

This shows that at 2000 rpm the heat rejected to the oil equals 17% of that rejected to the oil and water together, quite an appreciable item.

As engine sizes become larger, the tendency for the oil to handle a larger share of the cooling becomes pronounced. In some large air cooled engines, the lubricating oil handles as much as 19% of the total engine cooling (10), while in some automotive engines the figure

is as low as 2.3%. The amount of heat absorbed by the oil in the ordinary automotive engine is not so great that it cannot be satisfactorily dissipated through the walls of the crankcase, except under extraordinary conditions. The temperature of the oil should not be allowed to exceed 200 degrees F. for any length of time. While temperatures of 250-300 degrees have been recorded under very difficult driving conditions, they are the exception rather than the rule. Oil coolers have, therefore, not been deemed necessary in the automotive field (1,10), but for the larger installations where the oil carries a considerable part of the cooling load some form of external cooler for the oil must be provided.

It is not the purpose of this research, however, to determine the merits of oil coolers, but rather to investigate the effects of operating conditions upon the rate at which heat is given up to the lubricating oil. Perhaps the information will not be particularly useful in regard to the automotive engine, where the amount of heat to the oil is not a very large percentage of the total heat loss, but as applied to larger engines it may be of some value. The author believes that very little work has been previously done on this particular subject. Some work has been done on the effect of operating variables on heat transfer to the cooling water in liquid

cooled engines (8), and several curves are to be found in Taylor and Taylor, "The Internal-Combustion Engine", presenting that data. It was, in fact, a desire to determine whether or not the lubricating oil would exhibit similar behavior that prompted the present investigation.

The method devised for making the tests was to circulate the oil through the crankcase of an engine and measure the temperature of the oil as it entered and again as it left the crankcase. It was then possible to calculate the amount of heat given up to the oil as the product of weight of oil flowing, temperature rise, and specific heat. The variables studied were air-fuel ratio, spark advance, speed, oil temperature, manifold pressure and compression ratio. These variables lent themselves quite readily to control with the equipment available. The apparatus used and the proceedures followed in carrying out this scheme will be dealt with in the following sections.

II. APPARATUS

The engine used for this investigation was a small, one-cylinder Delco, originally intended for use as a farm lighting installation. It was originally air cooled, the cylinder being provided with fins to facilitate radiation. This system, however, would not have provided a sufficiently stable operating temperature for this investigation, so the fins were turned off the cylinder, and a water jacket fitted into place. A reflux condenser was then installed to maintain a constant jacket temperature of 212 degrees F.

The unit includes a D.C. generator, the armature of which is mounted on the extended crankshaft of the engine. The generator was designed to deliver 32 volts at normal operating speeds, but in the present series of runs, at some of the higher speeds, its output was as high as 50 volts. The amperage varied from 10 to 20, and the power output ranged from 150 to 500 watts. The power was absorbed in a water resistor containing a solution of sal soda. One side of the generator circuit connected with the steel container of the solution, the other side with a moveable electrode that could be raised or lowered to any desired depth in the solution. This simple unit provided

a flexible means of adjusting the load on the engine.

The instrument used to measure the power output of the generator was an "Acroset", manufactured by the Electro Products Corp. The "Acroset" is an instrument of very wide application, but was used here only for its very convenient voltmeter and ammeter arrangement. The voltmeter has a range from 0 to 50 volts while the ammeter covers a scale from 0 to 1200 amperes. Only the sensitive 0 to 30 ampere scale was employed, however.

Transformation and measurement of the energy of the engine being thus taken care of, the next step was to provide for the circulation of the oil through the crankcase and the measurement of its flow and temperature. The oil was circulated by means of a very small gear pump, driven through a V-belt from a 1/5 horsepower electric motor. The pump was provided with a by-pass for close control of its discharge pressure. To measure the inlet and outlet temperatures of the oil, two mercury-in-glass thermometers, range 0 to 220 degrees F, were installed in the oil line as close as possible to the crankcase. The hot oil from the crankcase then passed through a small heat exchanger, the oil flowing through a coil of copper tubing, encased in a section of two-inch pipe through which cold water circulated. A small gasoline meter, of the type used for metering gasoline directly into the

carburetor of an automobile in road tests, was used to determine the quantity of oil flowing. This device consists essentially of a small horizontal cylinder surmounted by a pair of solenoids with a valve mechanism between them, and the necessary electrical contacts for connection with a 6-volt storage battery. In operation, oil is fed into one end of the cylinder, forcing a piston toward the other end where it closes an electrical contact. The resulting current energizes one of the solenoids, which actuates the valve mechanism and diverts the oil flow to the other end of the cylinder, thus forcing the piston back in the opposite direction. The piston travel is reversed in this manner at each end of the stroke, resulting in a smooth, continuous operation of the meter. A contact is provided at one end of the stroke to actuate an electromagnetic counter. The volume swept by one stroke forth and back of the piston is 1/100 gallon. The sensitivity of the electrical contacts was such that the meter had to be operated between pressure limits of 3 and 8 pounds per square inch. The optimum pressure was found to be about 5 pounds per square inch.

The lubricating oil used was Macmillan Ring-free motor oil, S.A.E. 30. The specific gravity of this oil, as measured with the Westphal balance, was 0.9275 at 78.5 degrees F. From tables in the Tagliabue Manuel (11)

specific gravities at other temperatures were found and plotted against temperature (see page 32). In order to calculate the heat absorbed by the oil from the rate of flow and the temperature rise, it was necessary to know the specific heat of the oil as well as its specific gravity. The specific heat can be calculated from the U.O.P. characterization factor, temperature, and specific gravity according to an equation (6) that provides a straight line relationship between specific heat and temperature. The U.O.P. characterization factor is calculated from the following equation:

$$K = \frac{\sqrt[3]{T_b}}{s} \qquad \text{where}$$

K = U.O.P. characterization factor
T_b = average boiling point, degrees Rankine
s = specific gravity at 60 degrees F.

The average boiling point for a narrow cut such as that represented by a given weight lubricating oil is represented closely by the 50% point of the Engler distillation. The distillation data for the oil used is as follows:

% Distilled	Temperature, degrees F.
10	693
20	708
30	717
40	725
50	734
60	742
70	750
80	756
90	758
95	760
E.P.	760

Substituting the 50% point, 1194 degrees Rankine, in the above equation gives a U.O.P. factor of 11.3. The specific heat vs. temperature curve for an oil of that characterization factor, as shown in Hougen and Watson (6), is reproduced on page 32.

The investigation of each of the variables studied -- air-fuel ratio, spark advance, speed, oil temperature, manifold pressure, and compression ratio -- involved the use of additional pieces of apparatus. For measuring the air-fuel ratio the Elliott "carbumeter" was employed. This instrument is sensitive to changes in the thermal conductivity of the exhaust gases. These variations are brought about mainly by the change in the hydrogen content of the gases, which in turn is caused by variations of the air-fuel ratio. The conductivity of the gases is translated directly into air-fuel ratio on the dial of the instrument, the scale covering the range from 10 to 16. The needle valve in the carburetor was found to have too blunt a point to give an accurate and stable adjustment of air-fuel ratio. It was necessary to remove the needle and grind it to a long, sharp point to provide the needed control.

No special equipment was necessary in the investigation of spark advance and speed. Provision was made on the engine for varying the spark by means of set

screws on the breaker point assembly, and speed was controlled by means of the load.

To supply heat to the oil in the investigation of the effect of temperature, a small electric hot plate of the type used for laboratory distillations was placed under the heat exchanger in the oil line. This equipment was found to supply quite sufficient heat for the purpose.

Manifold pressure control was obtained by installing a valve in the air intake. A section of heavy rubber hose of $1\frac{1}{4}$ inch inside diameter was clamped over the air intake of the carburetor and a 3/4 inch globe valve fitted into the free end of the hose. By throttling down the valve it was possible to obtain and steadily maintain any desired manifold vacum. The intake manifold was tapped for an 1/8 inch pipe nipple, which was connected by means of rubber tubing to an open end, mercury manometer.

No provision was made in the design of the engine for changing compression ratio, so a method had to be improvised for making this change. The spark plug originally used with the engine was a 12 mm. one, but at some time in the past a hole had been drilled and tapped in the head for a 7/8 inch plug. This hole was utilized for changing the compression ratio. A steel plug was cut and threaded so that it could be screwed into the hole to a depth of $1\frac{1}{4}$ inches, as deep as it could be placed

without contacting the piston at top center. The bore and stroke of the engine was 2.5 x 3.0 inches, a displacement of 14.7 cubic inches. The space above the piston at top center (including the volume of the 7/8 inch hole), measured by pouring in alcohol, was 2.75 cubic inches. From these measurements it can be calculated that by screwing the plug in to different depths, the compression ratio can be varied over a range of about 6.3 to 8.3.

Photographs of the equipment used are shown on page 15. Figure 1 presents a quartering view of the engine and shows quite clearly the reflux condenser and its connection to the water jacket. Just at the right of the flywheel may be seen the inlet thermometer in the oil line. The inlet oil line is the flexible, black tubing leading from the thermometer coupling over to the oil meter, which is just out of the range of the picture to the right. The outlet oil line emerges from the crankcase at the back of the engine and is seen as the upper of the two small aluminum-painted pipes emerging from behind the machine. The lower of these two pipes carries cooling water to the heat exchanger, the small, aluminum-painted, horizontal cylinder in the extreme lower right of Figure 1. Figure 2 shows the heat exchanger to better advantage. The small object just above and behind the exchanger is the gear pump, backed by an aluminum driving

pulley larger than the pump itself. Two small valves are seen just above and to the left of the pump. The upper valve is in the by-pass line, while the lower controls the line leading to the heat exchanger.

Figure 2 also shows the control board with all its instruments in place. The black-faced dial at the top is the tachometer, graduated in increments of 20 rpm. The rectangular, black instrument with the two dials is the "Acroset", the dials being the voltmeter and ammeter with which the power output of the generator was measured. Just to its right is seen the dial of the pyrometer that indicated the exhaust temperature. Below this instrument is the handwheel that controls the water resistor, the movable electrode of which dangles below the wheel on a white cord. The upright cylinder on the floor beneath it is the other part of the resistor, the container for the solution. On the floor, beside the resistor, sits the electric hot plate used for heating the oil in the oil temperature investigation. The rectangular, black instrument in the extreme lower left of the picture is the air-fuel ratio meter. (This meter is also seen in the lower right of Figure 1). The long column extending up the right hand side of the control panel is the manometer used for measuring the manifold pressure. The two heavy black cables hanging in a loop across the lower part of

the picture are the conductors carrying the current from the generator to the control board.

Just above the water resistor is a horizontal board, carrying at its left end the oil meter, and at its right the electromagnetic counter connected to the meter. It is possible, on the two photographs, to trace the path of the oil. The oil leaves the pump (Figure 2), passes through the control valve to the heat exchanger, and thence upward through the aluminum-painted pipe to the meter. It emerges from the meter and travels through the flexible black tubing to the engine (Figure 1), passing through the thermometer coupling as it enters the crankcase. At the rear of the engine, the oil passes through another themometer fitting and through the aliminum-painted pipe back to the pump.

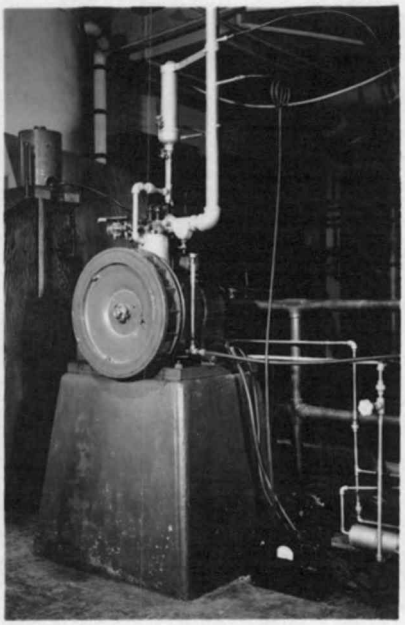

Figure 1

The Test Engine

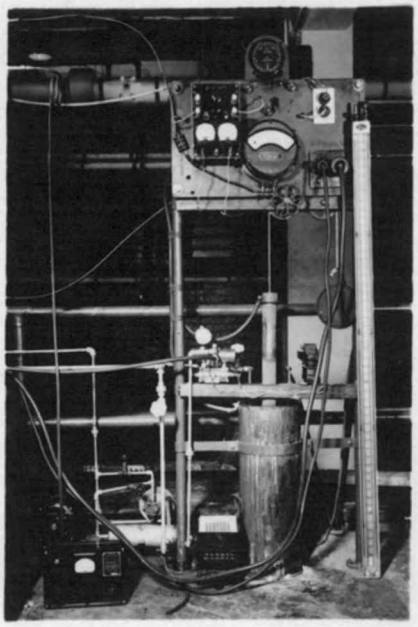

Figure 2

The Control Panel

III. PROCEDURE

The preliminary to the taking of any data in any of the runs was a warming up period of the engine. During this period all the variables were brought under control and stabilized at a value more or less arbitrarily determined beforehand. This phase of the operation usually occupied from one to two hours, depending on the length of time required for the temperatures of the inlet and outlet oil to come to equilibrium. The stability of the temperature of the room was a major factor in the length of this warming-up period. It was early discovered that the work could not be carried on while other activity was going on in the laboratory. The engine was located rather near the furnace that supplies heat and steam for the building, and it was futile to try to establish equilibrium conditions while the boilers were in operation. Also, the activity of student laboratory sections in running the various steam and gas installations, opening and closing windows, and so forth, precluded any accurate temperature measurements. It was therefore found necessary to make the runs at night or over week-ends when the operator found fewer or no disturbing elements in the laboratory,

At ten-minute intervals readings were taken of

counter number (the counter connected to the oil meter), volts, amperes, exhaust temperature, and any of the other variables pertinent to the run in question. An examination of any of the data sheets in a later section of this paper will illustrate the manner in which the data were recorded. The warming-up period was considered to be at an end when the temperature difference between inlet and outlet showed no variation (an allowance of 0.5 degree was made) for three consecutive readings. The average conditions obtaining for these three readings then established the first point on the curve, and the particular variable under consideration was then changed to a new value. Again the engine was allowed to run until the temperature difference was constant for three consecutive readings, thus establishing a second point on the curve.

By way of further explaining the material set forth in the preceeding paragraph, let us take, for example, the data recorded on page 43. This page sets forth the results of the air-fuel ratio investigation. At the end of 70 minutes equilibrium conditions were verified, the temperature difference for the readings at 50, 60, and 70 minutes varying by not more than 0.5 degree. The average air-fuel ratio for the three readings was 14.3. The rate of flow of the oil was determined by noting the increase in the counter reading over the 60 minutes

preceeding the establishment of equilibrium, in this case 66 - 10 = 56. Since the counter registers hundredths of a gallon, this means that oil was flowing at the rate of 0.56 gallons per hour. Looking up the specific gravity of the oil at the inlet temperature (see page 32), which is very nearly the temperature at the meter, it is seen that 0.56 gallons per hour is equivalent to 4.33 pounds per hour. From the specific heat curve (see page 32), the specific heat of the oil is found to be 0.435 at the average crankcase temperature. Taking the product of pounds per hour (4.33), temperature rise (32.2), and specific heat (0.435), the heat absorbed by the oil is calculated as 60.7 Btu per hour (see calculated data, page 46). This value plotted against the value of 14.3 for the air-fuel ratio gives the fifth point on the upper curve of page 36. The lower curve on that page expresses the same value as a percentage of the power output of the engine. The average reading of the wattmeter, multiplied by 3.4 to change it to Btu, was used as the divisor in calculating this percentage.

One point having thus been established, the air-fuel ratio was changed to another value, 12.0, all other variables being held constant. Conditions were maintained in this state until equilibrium was once more attained, establishing another point. The points on the

curve were not determined in order as may be seen by an inspection of the data sheet, but rather were determined in a random fashion. This was done to convince the writer that the changes in the readings were actually due to changes in the air-fuel ratio and not to "creep". In runs where this procedure was not followed, the initial point was checked at the end of the run to determine whether the equilibrium point would return to its first value.

It was thought best to determine each curve in a single run, since, if the engine were once shut down and allowed to cool off, it was feared that it might not be possible to again establish conditions so that the remaining data would fall on the original curve. This was, in fact, found to be the case in one preliminary run, the data of which is not presented since it failed to follow any kind of curve. The presence of only a few points on each of the curves is explained by the very nature of the tests. An examination of the data sheets will show that each point determined represents from 30 to 90 minutes of running of the engine. It was impossible to space the points as closely as might be desired to establish a definite curve, because it would have consumed a prohibitive amount of time. The run made in the determination of the air-fuel relationship, for example, required nearly twelve hours of steady running. Starting that run early

in the day was the reason for its taking so long to complete; with the temperature in the laboratory fluctuating, equilibrium was difficult to establish and maintain. In the later tests, the work was done after the laboratory temperature had stabilized, but, even so, considerable time was consumed in establishing each point on the curve.

The reliance on only three points to establish the curve for Btu vs. temperature (see page 33) is explained on a different basis. It was found to be impracticable to operate the engine at other than these three temperatures. The highest temperature was attained by placing the electric hot plate under the heat exchanger in the oil line and allowing the engine to run until equilibrium was reached. For the medium temperature, the hot plate was removed and no water was circulated through the heat exchanger. The low point was established by circulating water rapidly through the shell of the heat exchanger. Theoretically, it should have been possible to maintain any intermediate temperatures by proper adjustment of water and heater, but in practice it did not work out. The temperatures thus attained were far from stable, and equilibrium conditions could not be established. The compression ratio curve (see page 37) was also based on only three points. Because of the construction of the plug and the threaded hole, the plug could be

locked securely on only three positions.

In the investigation of manifold pressure, the first thought was to supercharge the engine by means of compressed air. This was attempted, but it was found that the engine was not designed for it. Gasoline dripped in a stream from the carburetor, and operating conditions were absolutely uncontrollable. The air hose from the compressor was disconnected from the air intake, the air supply throttled down, and the engine run on the vacuum side rather than under increased pressure. This arrangement was much more satisfactory, conditions being easily stabilized.

In this work it was decided to try to make an absolute separation of the variables and study them one at a time. This proved to be rather difficult, since some of the variables are so closely associated that it was nearly impossible to separate them. It was possible to keep the six variables studied under control, but adjustments of other parts of the system had to be made. For example, as the spark setting was changed it was found necessary to change the load to keep the speed at a constant value. In changing the speed, the operator had to adjust the needle valve on the carburetor to keep the air-fuel ratio constant, an adjustment that was also necessary as the manifold pressure was changed. Compression ratio

and spark setting are two items that are very closely related. In this investigation the compression ratio was varied without changing the spark in order to find the effect of compression ratio alone. Perhaps a more logical proceedure would have been to set the spark to maximum power at each compression ratio, but that would have left some doubt as to which variable was contributing most to the heat transfer.

IV. RESULTS AND CONCLUSIONS

The results of the tests have been tabulated on pages 39 to 47, and presented in graphical form on pages 33 to 38. The results are expressed as Btu per hour absorbed by the lubricating oil, and also as percent of brake horsepower.

There are three factors that can affect the amount of heat going into the oil in an engine of this type. They are mean temperature difference between combustion chamber and crankcase, coefficient of heat transfer of the piston and cylinder walls, and the rate at which the oil is sheared. The first of these is practically proportional to cylinder temperature, since the temperature of the oil in the crankcase varies very little throughout the run (except, of course, for that run wherein the oil temperature was the variable under investigation). The coefficient of heat transfer is dependent primarily on the film of gases adhering to the inner walls of the cylinder. The rate of shear of the oil is affected only by the speed of the engine.

From the form and slope of the curve relating heat transfer to oil temperature (see page 33), it is rather obvious that the temperature difference between

combustion chamber and crankcase is the governing factor. As the temperature of the oil is increased, thus decreasing the temperature differential, the amount of heat absorbed by the oil shows a sharp decline. A straight line relationship is to be expected since temperature difference is the only factor affecting heat transfer that is involved. It will be noted that at about 115 degrees the heat transfer is zero. This means that at that temperature the oil is losing heat through the walls of the crankcase as rapidly as it is receiving heat from the cylinder; therefore the oil is emerging from the crankcase at no higher temperature than that at which it entered. At temperatures above 115 degrees the flow of heat appears to be in the opposite direction, the oil giving up heat more rapidly than it receives it. This does not mean, however, that the oil is actually returning heat to the cylinder -- the cylinder temperature, as roughly indicated by the exhaust temperature (450 degrees), disputes that supposition. If the crankcase were insulated so that no heat could be lost to the atmosphere, the curve might be expected to approach zero as a limit as the oil temperature approached that of the cylinder. Since there was nothing in the change of oil temperature that could vary the power output, power was constant for the run, so the curve for percent of brake

horsepower exhibits the same form as the Btu curve.

Increasing speed has an effect on all three heat transfer factors. First, increased speed increases the cylinder temperature as indicated by the exhaust temperature (see page 40). Increased speed also increases the velocity of the gases over the cylinder walls, reducing the film thickness and thus increasing the heat transfer coefficient. Finally, the increase in speed increases the rate of shear of the oil in the bearings and the churning effect of the splash lubricating system. All three effects are additive, resulting in a sharp increase in the amount of heat to the oil as the speed rises (see page 34). The points establish a straight line relationship over the range of speeds investigated. The power output increases as the speed builds up, rather rapidly at first and then at a slower rate. This causes the percent curve to bend upward sharply in the lower range, and to approach a straight line in the upper range.

In varying the spark advance, two opposing tendencies are brought into play with regard to the average cylinder temperature. A greatly advanced spark allows more time for the cylinder walls to be exposed to the burned gases, which tends to increase the average temperature. On the other hand, a later spark produces a more intense flame and higher exhaust temperature (see page 41)

which also has the effect of raising the cylinder temperature. In the work done on heat transfer to cooling water the curves for which appear in Taylor & Taylor (8), the first tendency seems to be the governing one, the heat transfer rising gradually as the spark is advanced. In the present investigation, the second tendency seems to govern. The late spark resulted in the highest heat transfer to the oil, with the curve falling off quite sharply as the spark was advanced. With the more advanced spark settings, however, the first tendency takes effect with the result that the curve begins to level off, (see page 35). The power output falls off greatly with spark advance; hence, the percent curve rises very sharply so that at 50 degrees advance the oil is absorbing over 12% of the brake horsepower.

As air-fuel ratio is increased the amount of heat absorbed by the oil rises gradually to a maximum value and then falls off rather rapidly for the higher air-fuel ratios. Here again there are two opposing tendencies at work. As the fuel is increased, reducing the air-fuel ratio, more of the uncombined oxygen is used, thus liberating more heat. At the same time, the tendency to burn to CO instead of CO_2 increases, a reaction that liberates less heat per atom of carbon. These two tendencies balance out in the neighborhood of 8% excess fuel (8),

which is an air-fuel ratio of about 13.9. An air-fuel ratio of 15 represents the theoretical mixture. The curve for heat to oil vs. air-fuel ratio peaks at a ratio of about 12.6, a value which represents about 19% excess fuel. The peak might have been expected, from the foregoing discussion, to fall at a slightly higher ratio, but differences in performance characteristics between different engines could easily effect that much variation. It is interesting to note that the peak of the curve falls very near to the air-fuel ratio for maximum power, generally found to be about 13. The power output varied very little over the range of the experiment, so the percent curve shows the same form as the Btu curve (see page 36).

Increasing the compression ratio decreases the amount of heat absorbed by the oil as shown on page 37. There are two effects of increased compression that tend to lower the average cylinder temperature - the higher pressure shortens the period of combustion and the decreased clearance volume lessens the quantity of residual gases (5). The increased density of the charge tends to increase the combustion temperature, but the two previously mentioned, opposite tendencies apparently outweigh the density factor. Thermal efficiency increases quite markedly with compression ratio; hence, with more heat going to work, less heat is lost to the oil and the

cooling water. The power output increased slightly with compression ratio, so the percent curve drops more rapidly than does the Btu curve. Had the spark advance been varied to take full advantage of the increased compression ratio, this effect would have been more marked.

The effect of manifold pressure was so slight that the equipment used in this investigation was insensitive to it. The curve for heat to oil vs. manifold pressure will be seen to plot a horizontal straight line (see page 38). From the behavior of the exhaust temperature, however, it would appear that there is some slight effect. As manifold pressure is increased, the exhaust temperature goes up, indicating that if any variation of heat flow exists, it probably increases with manifold pressure. Power output rises sharply with increased pressure, so the percent curve shows a definite decline.

The average rate at which heat was rejected to the lubricating oil throughout the investigation was about 65 Btu per hour. The power output averaged about 475 watts. This results in an average heat transfer of about 1.7 Btu/min/hp, which compares quite well with values determined by other investigators (see page 3).

In conclusion, the results of the investigation can be summed up as follows:

1. Heat transfer to the lubricating oil decreases rapidly with increasing oil temperature in a straight line relationship.

2. Heat transfer to lubricating oil increases with increasing speed in a straight line relationship.

3. Heat transfer to lubricating oil decreases as spark is advanced, rapidly at first but leveling off with greatly advanced spark settings.

4. Heat transfer to lubricating oil rises to a maximum value as air-fuel ratio is increased to 12.6, falling off rapidly beyond that point.

5. Heat transfer to lubricating oil decreases gradually with increasing compression ratio.

6. Heat transfer to lubricating oil shows no variation with manifold pressure within the limits of sensitivity of the apparatus used in this investigation.

V. SUGGESTIONS FOR FUTURE WORK

As the reader may have gathered from the preceding sections, the engine used in this investigation was far from being a precise laboratory instrument. It was rather old, and had been in disrepair for some time before being used in the present research. It is quite possible that welding the water jacket in place distorted the cylinder to some extent. At any rate, the engine was not as tight as might have been desired. Also, it was too small for this sort of work, the heat quantities involved being so small that any errors or discrepancies were greatly magnified. Recalling what was stated in the first section, as the size of the engine is increased, a greater proportion of the cooling is handled by the lubricating oil. The point being made is that this investigation should have been carried out on a larger engine, probably at least a four cylinder, automotive model.

The test engine should be in a room of its own, where the temperature may be closely controlled, making it possible to operate at any time that is convenient rather than having to wait until after class hours (see page 16). This would make possible longer runs and the determining of a greater number of points on each curve.

If a larger engine were used, however, this factor of room temperature might not be so important.

Further work on this subject might well include combinations of the variables studied here. For instance, as compression ratio is changed, the spark might be varied to give maximum power at each ratio. Spark should logically be changed with air-fuel ratio and speed to get the best out of an engine. The different variables are so closely interwoven that varying them independently, as was done in this research, often leads to very inefficient operation of the engine.

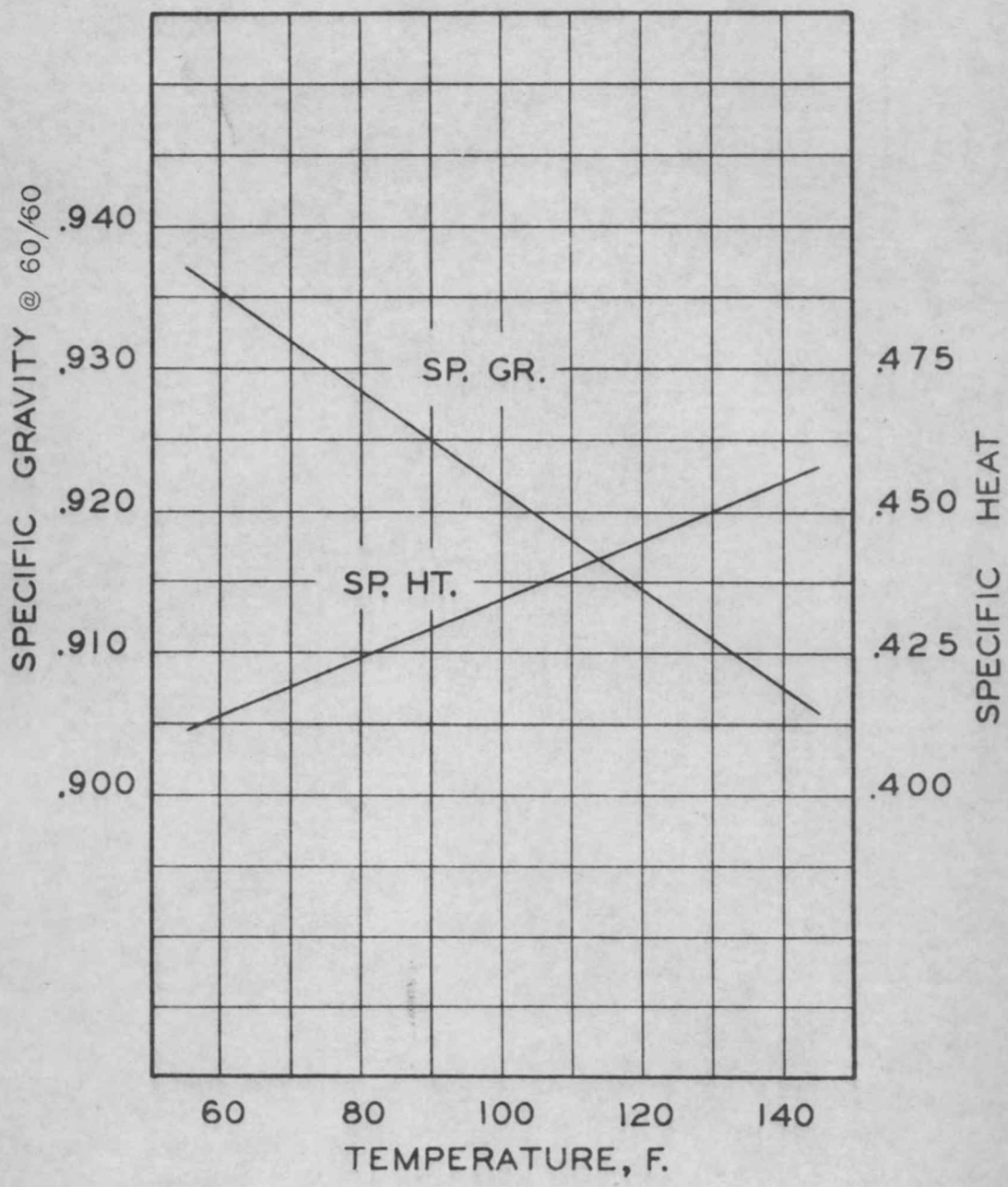

Figure 3

Specific Gravity and Specific Heat
vs. Temperature for
MacMillan Ring-free Motor Oil, S.A.E. 30

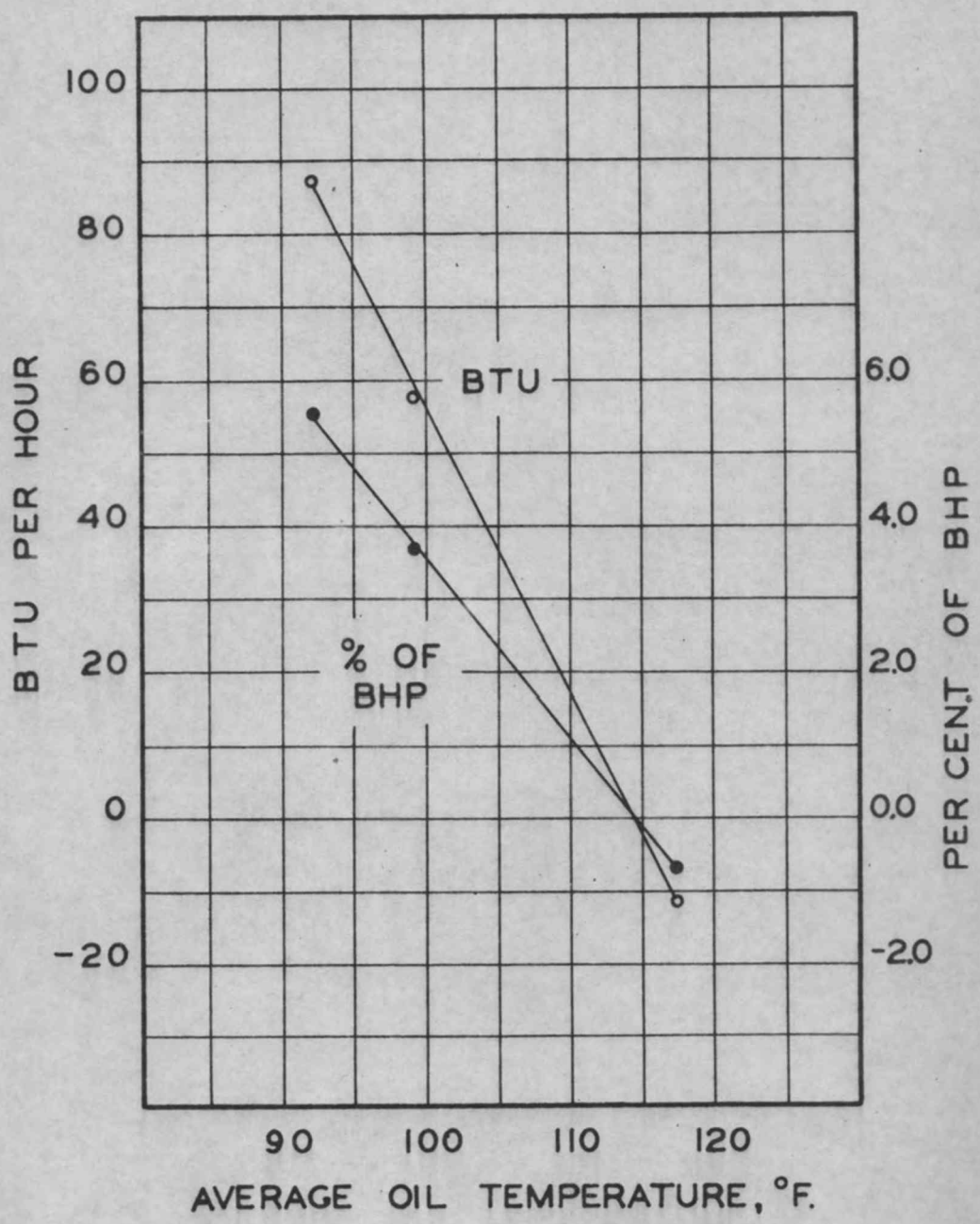

Figure 4

Heat Rejected to Lubricating Oil
vs.
Average Oil Temperature

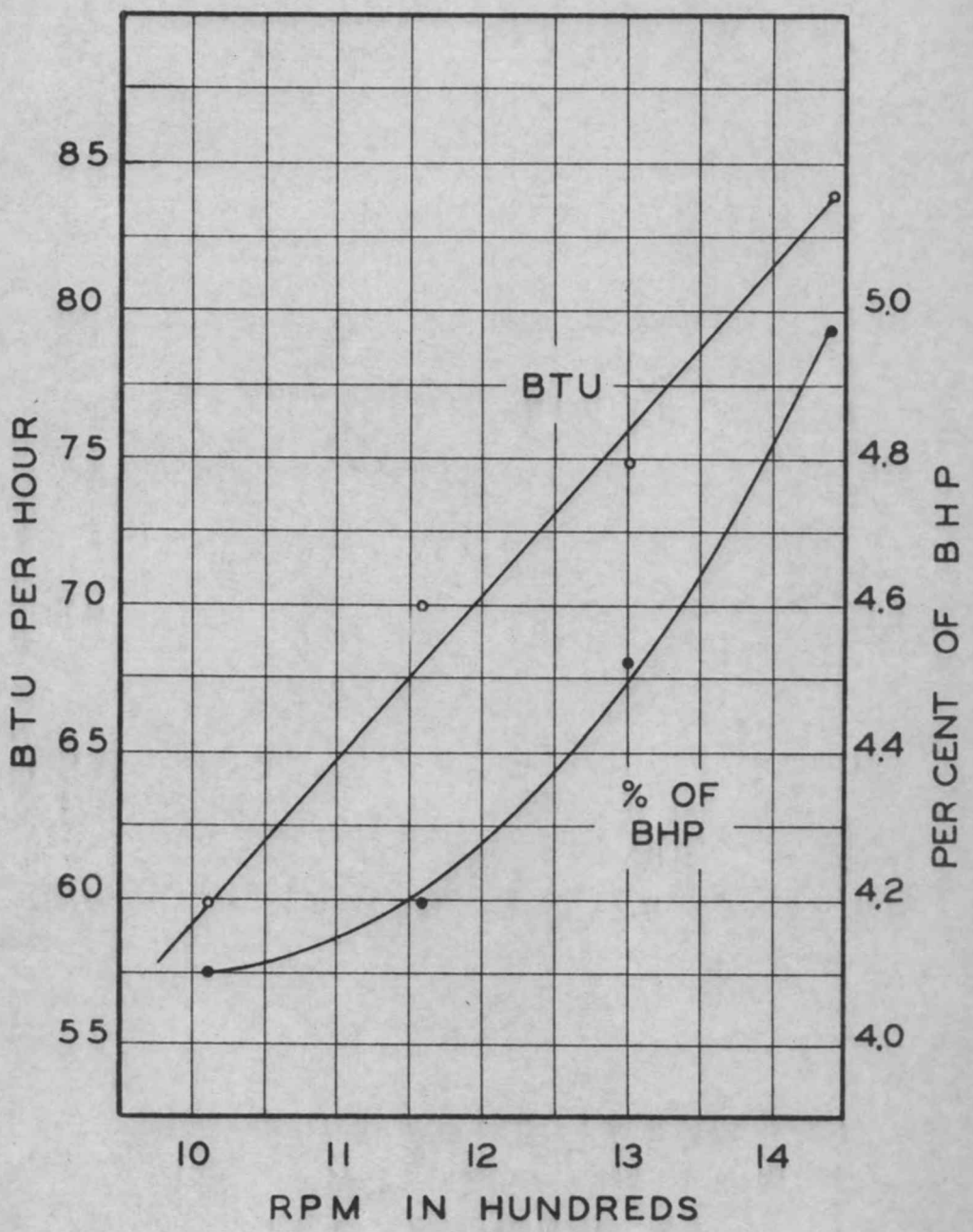

Figure 5

Heat Rejected to Lubricating Oil
vs.
Revolutions per Minute

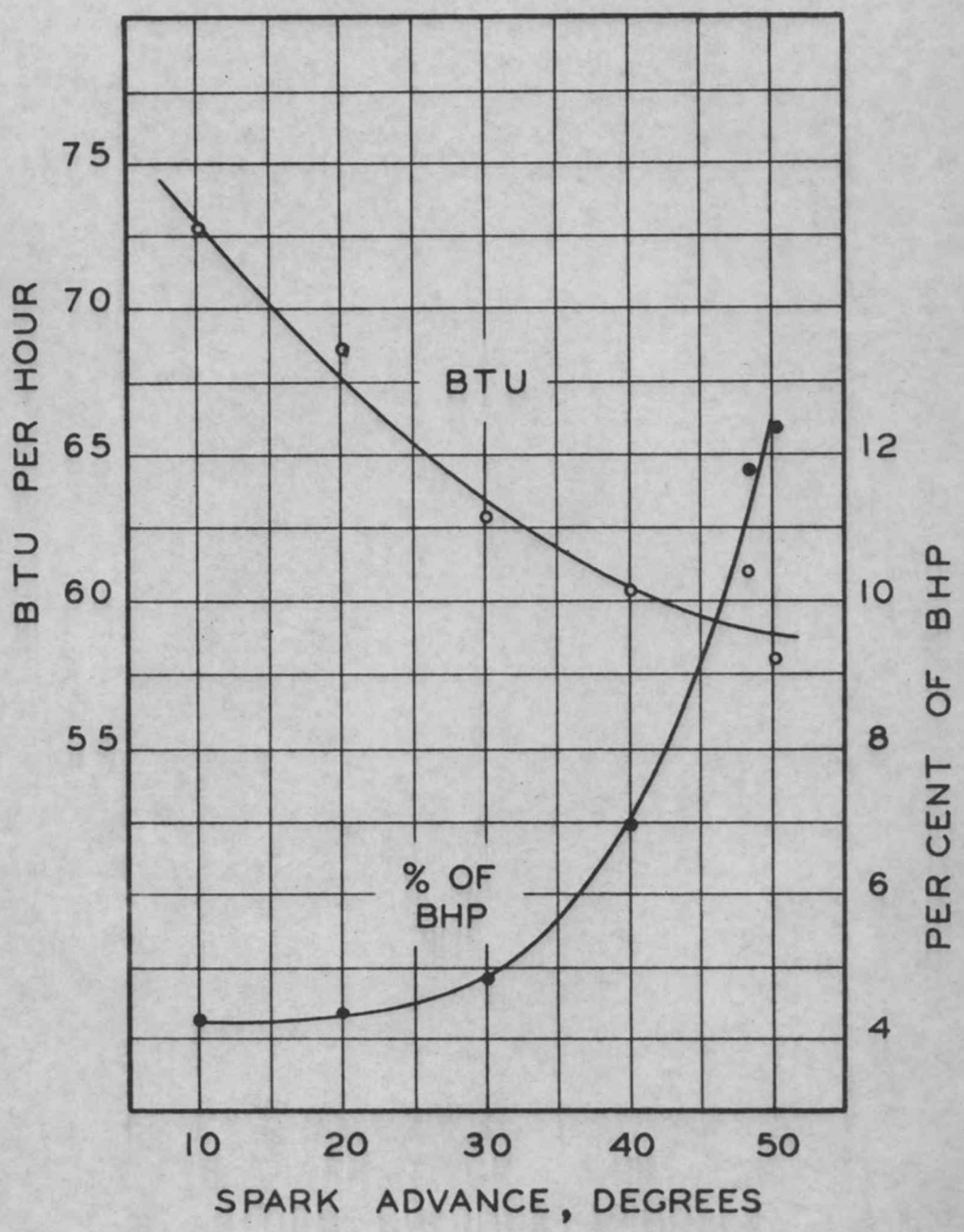

Figure 6

Heat Rejected to Lubricating Oil
vs.
Spark Advance

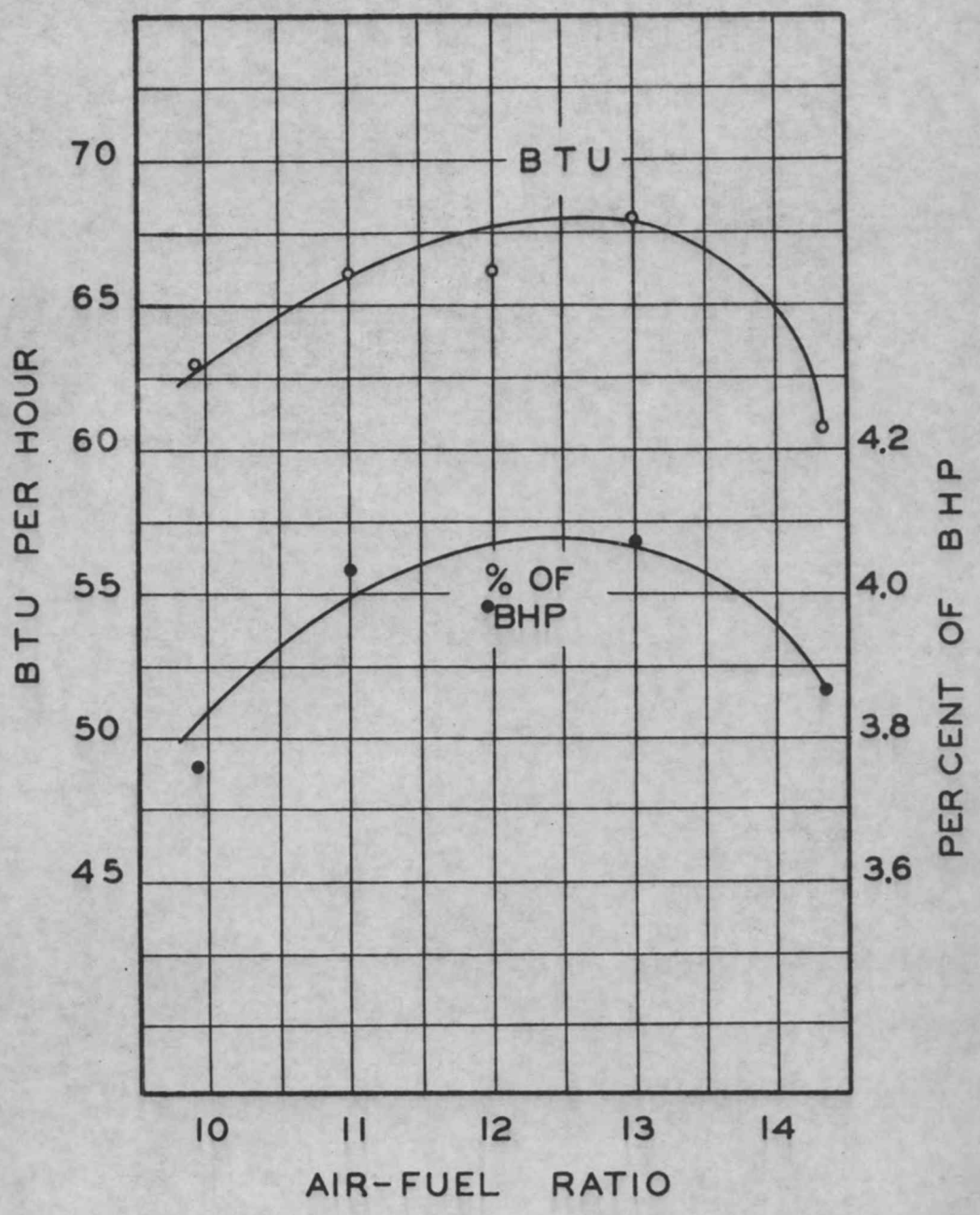

Figure 7

Heat Rejected to Lubricating Oil
vs.
Air-fuel Ratio

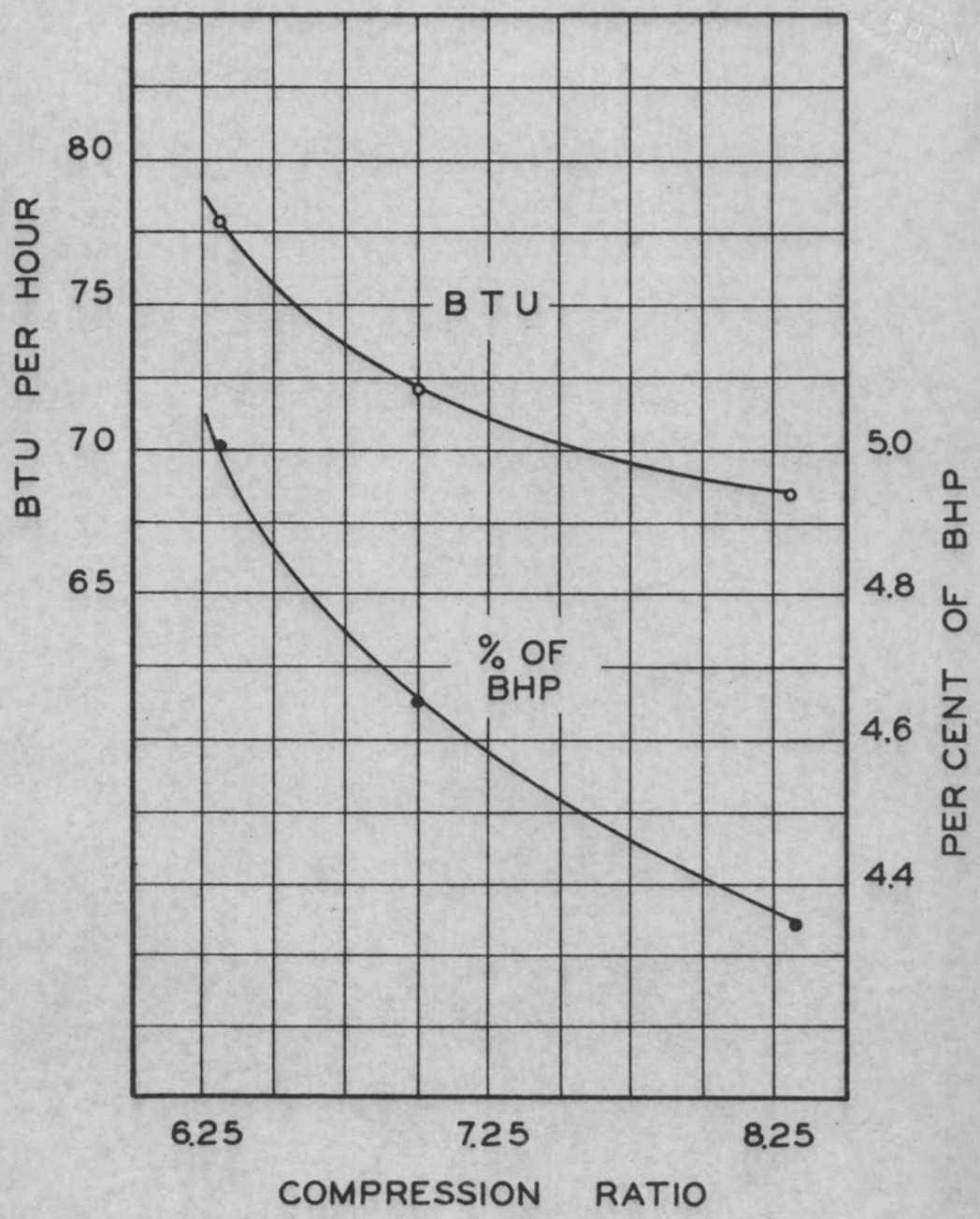

Figure 8

Heat Rejected to Lubricating Oil
vs.
Compression Ratio

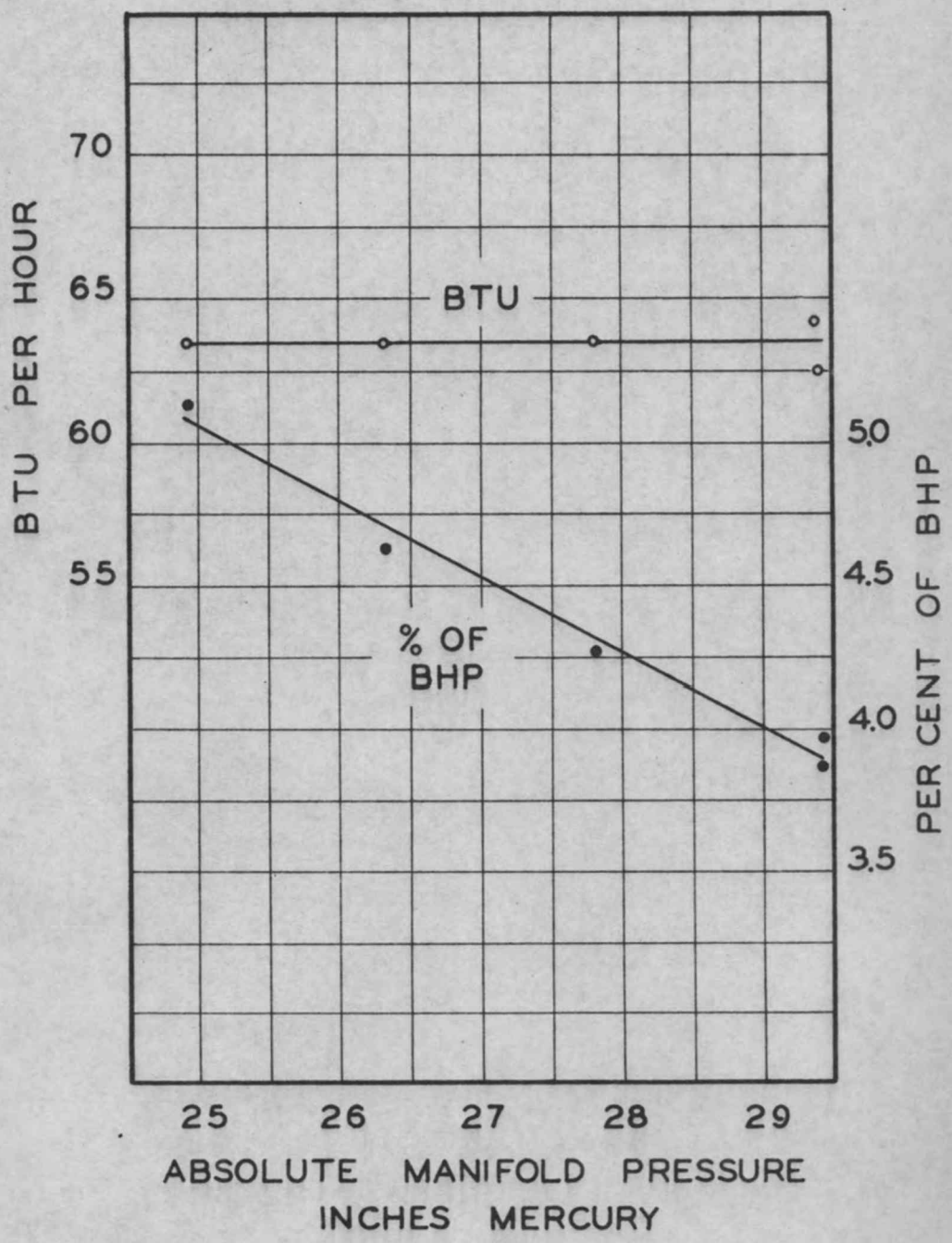

Figure 9

Heat Rejected to Lubricating Oil
vs.
Absolute Manifold Pressure

VII. TABULATED DATA

OBSERVED DATA

HEAT TO OIL vs. OIL TEMPERATURE

RPM -- 1000

Spark Advance -- 25 deg. Air-fuel Ratio -- 12.5 - 13.2

TIME min.	COUNTER	OIL TEMP. F. in	out	VOLTS	AMPS	WATTS	EXHAUST TEMP.
50	75	110.0	112.0	26.7	17.2	459	455
90	134	118.0	114.5	27.1	17.2	466	455
100	150	119.0	115.0	25.8	17.6	454	455
110	165	118.5	115.0	25.4	17.8	452	455
140	201	85.5	112.5	26.0	17.6	458	500
150	211	85.5	113.0	26.0	17.6	458	450
160	222	85.5	113.0	26.0	17.7	460	450
200	280	72.0	111.0	26.2	18.1	474	455
210	291	72.0	112.0	25.3	18.2	460	455
220	302	72.0	112.0	25.3	18.2	460	450

OBSERVED DATA

HEAT TO OIL vs. RPM

Spark Advance -- 25 deg. Air-fuel Ratio -- 12.6 - 13.3

TIME min.	COUNTER	OIL TEMP.F. in	out	VOLTS	AMPS	WATTS	EXHAUST TEMP.	RPM
140	124	79.0	114.0	33.8	14.8	501	480	1140
180	160	78.5	116.5	34.6	14.2	492	473	1150
190	169	78.5	116.5	34.5	14.2	490	473	1150
200	179	78.5	116.0	33.3	14.7	490	470	1160
240	216	79.0	119.0	40.7	12.0	489	483	1290
250	225	79.0	120.0	43.0	11.2	482	488	1310
260	234	79.0	119.5	41.8	11.5	482	490	1300
280	253	79.0	123.5	46.8	10.5	49 2	500	1420
290	262	79.0	123.5	48.0	10.2	490	500	1450
300	272	79.0	123.5	47.4	10.6	502	500	1450
320	290	79.0	112.0	26.2	16.4	430	428	1000
330	298	79.0	112.0	27.2	15.7	427	428	1020
340	307	79.0	112.0	27.1	15.8	428	428	1010

OBSERVED DATA

HEAT TO OIL vs. SPARK ADVANCE

Air-fuel Ratio -- 12.4 - 13.3 RPM -- 1000

TIME min.	COUNTER	OIL TEMP. F. in	out	VOLTS	AMPS	WATTS	EXHAUST TEMP.
Spark advance -- 50 deg.							
30	34	80.0	105.5	28.0	5.3	149	400
70	79	80.5	105.5	29.0	5.0	145	405
80	91	80.5	105.5	28.7	4.6	132	403
90	103	80.5	105.5	30.0	4.6	138	410
Spark advance -- 40 deg.							
120	138	80.0	106.0	29.0	9.0	261	410
130	149	80.0	106.0	28.5	8.7	248	411
140	160	80.0	106.0	29.0	8.7	252	415
Spark advance -- 30 deg.							
180	206	80.0	107.5	28.4	13.5	384	430
190	217	80.0	107.5	28.0	13.5	378	430
200	228	80.0	107.5	27.5	14.0	386	432
Spark advance -- 20 deg.							
220	251	80.0	109.0	27.9	16.4	457	455
260	295	80.0	111.0	25.0	18.2	456	460
270	306	80.0	111.0	26.9	17.7	476	460
280	317	80.0	111.0	27.0	17.7	478	460

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TIME min.	COUNTER	OIL TEMP. F. in	out	VOLTS	AMPS	WATTS	EXHAUST TEMP.
Spark advance -- 10 deg.							
310	349	80.0	113.5	25.8	19.6	507	480
320	360	80.0	113.5	26.0	19.4	504	480
330	371	80.0	113 .0	24.8	20.1	498	480
Spark a dvance -- 48 deg.							
370	413	80.0	109.0	28.9	5.8	168	400
380	423	79.0	108.0	30.1	4.6	138	403
390	434	79.0	108.0	30.8	5.0	154	403

OBSERVED DATA

HEAT TO OIL vs. AIR-FUEL RATIO

Spark Advance -- 25 deg. RPM -- 1000

TIME min.	COUNTER	OIL TEMP.F. in	out	VOLTS	AMPS	WATTS	EXHAUST TEMP.	A/F
10	10	78.5	107.5	27.0	15.6	421	483	14.1
50	45	77.5	110.0	25.7	17.7	455	478	14.2
60	55	78.0	110.0	26.0	17.7	460	478	14.3
70	66	78.0	110.0	25.8	18.0	464	480	14.3
260	272	84.0	115.5	24.0	19.4	466	450	12.0
300	310	85.0	120.0	26.3	18.8	495	455	12.0
310	319	84.5	119.0	25.3	19.3	488	450	12.0
320	328	84.5	119.0	26.0	19.0	490	452	12.0
520	519	81.5	115.5	26.2	18.4	483	448	12.0
560	558	81.0	113.5	26.2	18.8	493	438	10.2
570	567	81.0	113.0	25.7	19.0	489	435	9.8
580	577	81.0	113.0	25.5	19.1	488	433	9.8
620	616	80.5	114.0	27.0	18.1	490	452	12.9
630	626	80.0	115.0	26.9	18.2	490	452	13.0
640	636	80.0	114.0	26.9	18.3	492	452	13.0
670	665	80.0	113.5	25.4	18.9	481	438	11.0
680	676	80.0	113.0	25.2	19.0	481	436	11.0
690	685	80.0	113.0	24.9	19.0	480	436	11.0

OBSERVED DATA

HEAT TO OIL vs. COMPRESSION RATIO

RPM -- 1000

Spark Advance -- 25 deg. Air-fuel Ratio -- 12.8 - 13.5

TIME min.	COUNTER	OIL TEMP. F. in	out	VOLTS	AMPS	WATTS	EXHAUST TEMP.
Compression ratio -- 6.3							
30	33	78.0	110.0	26.8	17.2	462	472
70	77	78.0	113.0	26.9	17.0	458	470
80	88	78.0	113.0	25.7	17.6	454	470
90	99	78.0	113.0	25.9	17.5	454	470
Compression ratio -- 7.0							
120	131	79.0	113.0	25.3	17.9	454	455
130	141	78.5	112.5	26.2	17.5	459	458
140	151	78.5	113.0	25.5	17.7	452	455
Compression ratio -- 8.3							
0	346	77.0	108.0	26.8	17.8	477	450
40	386	77.0	111.0	26.0	17.7	460	450
50	396	77.0	111.0	26.5	17.6	467	440
60	406	77.0	111.0	26.3	17.7	467	445

OBSERVED DATA

HEAT TO OIL vs. MANIFOLD PRESSURE

RPM -- 1000

Spark Advance -- 25 deg. Air-fuel Ratio -- 12.2 - 13.1

TIME min.	COUNTER	OIL TEMP. F. in	out	VOLTS	AMPS	WATTS	EXHAUST TEMP.
Manifold vacuum -- 0.4 in. Hg							
0	229	79.0	106.0	25.9	18.3	474	450
40	269	79.0	110.0	25.8	18.5	478	450
50	279	79.0	110.5	25.6	18.6	476	450
60	288	79.0	110.5	25.4	18.7	475	450
Manifold vacuum -- 2.0 in. Hg							
80	308	79.0	111.0	26.2	16.7	438	412
90	318	79.0	111.0	26.2	16.7	438	410
100	328	79.0	111.0	26.2	16.6	435	410
Manifold vacuum -- 3.5 in. Hg							
130	358	79.0	110.0	26.9	14.7	396	375
140	368	79.0	110.0	26.7	14.9	398	375
150	379	79.0	110.0	26.7	14.9	398	380
Manifold vacuum -- 5.0 in. Hg							
170	399	78.0	109.0	27.6	13.4	370	365
180	409	78.0	109.0	27.4	13.0	356	365
190	420	78.0	109.0	27.8	12.9	359	365
Manifold vacuum -- 0.4 in. Hg							
200	430	78.0	109.0	26.0	18.7	487	445
210	440	78.0	109.5	25.4	18.8	478	450
220	450	78.0	109.5	25.1	19.0	477	450

CALCULATED DATA

VARIABLE	TEMP. RISE deg. F.	OIL FLOW lb/hr	HEAT TO OIL Btu/hr	HEAT TO OIL % of Bhp
Variable -- Air-fuel ratio				
9.9	32.2	4.49	63.0	3.77
11.0	33.2	4.57	66.1	4.03
12.0	35.1	4.33	66.2	3.96
13.0	34.2	4.57	68.0	4.07
14.3	32.2	4.33	60.7	3.87
Variable -- Spark advance				
50	25.0	5.33	58.0	12.32
40	26.0	5.33	60.3	6.96
30	27.5	5.25	62.8	4.81
20	31.0	5.10	68.7	4.29
10	33.3	5.02	72.8	4.25
48	29.0	4.87	61.4	11.78
Variable -- Average oil temperature				
117	- 3.7	6.86	- 11.3	- 0.73
99	27.3	4.83	57.7	3.70
92	39.7	5.08	87.5	5.52

CALCULATED DATA

VARIABLE	TEMP. RISE deg. F.	OIL FLOW lb/hr	HEAT TO OIL Btu/hr	HEAT TO OIL % of Bhp
Variable -- RPM				
1010	33.0	4.17	59.9	4.10
1156	37.9	4.25	70.0	4.19
1300	40.5	4.25	74.8	4.52
1440	44.5	4.33	83.8	4.97
Variable -- Manifold vacuum				
0.4	31.5	4.56	62.5	3.84
2.0	32.0	4.56	63.5	4.26
3.5	31.0	4.72	63.5	4.68
5.0	31.0	4.72	63.5	5.14
0.4	31.4	4.72	64.2	3.91
Variable -- Compression ratio				
6.3	35.0	5.11	77.9	5.02
7.0	34.1	4.89	72.2	4.65
8.3	34.0	4.65	68.5	4.34

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