A THERMODYNAMIC STUDY OF ENERGY ABSORBED BY A MOTORED FOUR CYCLE ENGINE WITH POSITIVE EXHAUST PRESSURE AS APPLIED TO COMMERCIAL EXHAUST BRAKES

by

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I. INTRODUCTION

This investigation involves the operation of a four cycle internal combustion engine. Therefore, it seems fitting at this point to review some of the terminology used and theory involved concerning four cycle engines.

Engine Design Considerations

Internal combustion engines use air for a working substance. Air is a fluid which has appreciable mass. Fluids which are in motion have inertia and this partially determines the design criteria for the valve action of an engine.

During the intake stroke of an engine, it is necessary to open the intake valve before the piston begins its downward movement in order to have a sufficient area between the valve and its seat to let airflow take place, and also to allow time for the air to be accelerated. Also at the bottom of the stroke, the valve must still be open slightly so that the inertia of the incoming air may be used to more completely fill the cylinder volume. Therefore, the intake valve must close after bottom dead center. In practice the total opening usually is in the neighborhood of 290° of crankshaft degrees.

The exhaust valve must be opened before the piston reaches the bottom of the power stroke for the same reason as the intake valve must be opened sooner. This is to allow a sufficient area between the seat and the valve for the gas to flow through and time to be accelerated. On the exhaust stroke the burned gases are pushed out by the piston and the exhaust valve closes after top dead center to take advantage of the inertia of air and to improve scavenging when supercharging is used. Consequently, the exhaust valve opens before bottom dead center and closes after top dead center. The total opening usually involves about 290° of crankshaft degrees.

Note that the end of the exhaust stroke is the beginning of the intake stroke. Therefore, at this point, both the intake and exhaust valves are open. This is commonly known as the period of valve overlap. Usually this period will be from 80 to 140 crankshaft degrees.

The valve timing of an engine depends on many factors. Among these are cylinder head design including valve sizes and port shape, speed of operation, bore and stroke, and the air flow through the engine at partial and full throttle usage.

The power delivered by an engine is dependent on two factors. The first is the torque or twisting effort delivered by the crankshaft and the second is the speed at

which this torque is produced. To compute horsepower, the following equation is commonly used:

H.P. = (Torque) (RPM) / 5250 where the torque is measured in lb-ft.

When an engine is being turned over by an external source, this is usually termed "motoring." The horsepower required to "motor" an engine under normal conditions, zero gauge intake and exhaust pressures, is given the title, "the friction horsepower of the engine."

Exhaust Brake Definition

Motor vehicles during the past fifty years have undergone considerable change. They have been made heavier and their average highway speeds have increased. Commercial highway vehicles are now able to transport goods in favorable competition with other methods of transportation, such as railroads and ships. They have achieved this by hauling heavier loads and decreasing the time to the destination. Engines now available for commercial use are relatively free from large maintenance costs and operate on a relatively inexpensive fuel - diesel.

Most of the commercial vehicles on the highway are capable of exceeding maximum legal speeds with a full capacity load on a level road or even a small upgrade. Only on steeper hills is it necessary for the vehicle to

use maximum power at reduced speed. However, in order to maintain a safe speed while descending the grade, friction type drum brakes are applied quite severely. Brake drum temperatures may reach 1600°F (13, p. 15).[1] Many trucks employ cooling water on their brakes to absorb part of the energy given up by descending the grade.

Many types of retarders (energy absorbing devices) besides friction brakes have been devised to solve the problem of absorbing the energy. Most of the retarders use a fluid which absorbs the energy and then the energy is given up by the use of a radiating surface. Often the water jacket radiator of the engine is used. These devices are quite expensive to install and carry with them a high weight penalty. Fluid retarders also require periodic maintenance and replacement of parts.

Recently a new retarding device has been introduced in this country to aid the commercial vehicle. This device has been given the title of the "exhaust brake." The Swiss have used these devices for over forty years and in Europe they are very popular (3, p. 71). The Oetiker exhaust brake, of Swiss origin, is manufactured under license in a number of countries. In France, for example, the Fowa-Oetiker exhaust brake is produced and in Britain

[1] Numbers in parentheses refer to bibliography.

Clayton Dewandre Co. Ltd. manufacture another version (1, p. 283).

An exhaust brake is a very simple mechanical device. It consists of a butterfly valve which is installed in the exhaust line of the vehicle. The operator of the vehicle may open or close the valve remotely from the cab. When the valve is closed and the engine is being motored, the engine becomes analogous to an air compressor. Therefore, it is possible to transmit energy from the vehicle through the engine into the exhaust gas and produce additional retarding. The valve should be installed as close to the engine as possible to minimize the time it takes the exhaust gases to pressurize the volume between the engine and butterfly valve.

Several advantages of exhaust brakes were summarized in the Automobile Engineer.

> Advantages of the exhaust brakes in general are as follows. When the brake is applied to reduce the speed of the vehicle down a long gradient, engine temperature does not fall greatly, because the engine is used as a device to convert mechanical energy into heat. The incorporation of an exhaust brake also reduces the danger of the engine's overspeeding. Moreover, the loading on the connecting rod bearings is reduced as long as the brake is in operation, because there is always a gas pressure acting on the piston crown to tend to offset inertia loading. Since the use of an exhaust brake eliminates the low pressures that occur when a vehicle is coasting downhill with the throttle closed, oil control problems are

> less severe. Better fuel consumption can be obtained if a fuel cut-off device is incorporated and connected to the exhaust brake control. As the maximum exhaust manifold pressures is only built up after several revolutions of the engine, the brake control can be applied as quickly as may be desirable, and there is no danger either of the development of high stresses in the transmission or of inducing a skid. (1, p. 284)

However, exhaust brakes do have some disadvantages. First, they have been found to be unsuitable for use on two cycle engines (10, p. 22). Second, on engines equipped with oil bath air cleaners, sometimes oil may be blown out of the air cleaner. This is caused by a pulsation of air in the inlet manifold due to valve overlap when the brake is applied. This condition is more serious with a small number of cylinders but it can be reduced by providing a sufficient volume of air between the engine and air filter (10, p. 22).

Other forms of exhaust brakes have been devised. One of these opens the exhaust valve at the top of the compression stroke, thus releasing the energy of the compressed gas to the atmosphere via the exhaust line. The system for doing this uses engine oil pressure coupled with a valving arrangement and is quick in action, requiring only three or four revolutions of the engine to become operative (5, p. 79). Another type, changes the camshaft timing so that the engine operates exactly as an air compressor. This type is suitable for two cycle engines (1, p. 284).

Various claims for the effectiveness of exhaust brakes have been stated. In Europe, the manufacturers claim their units will give a retarding effort equal to 80 or 90 per cent of the rated horsepower of the engine (2, p. 252). One American manufacturer claims his unit will give a braking effect at the driving wheels of the vehicle at least equal to the horsepower rating of the engine (12, p. 2).

II. OBJECT AND SCOPE

The purpose of this investigation was to determine in what manner high positive exhaust pressure absorbed the energy resulting from motoring an engine. In addition to this, a study was made of the processes that were actually being carried out in the cylinder of the engine during exhaust braking. By analyses of these processes, it was determined how the retarding effect was produced and where the energy was dissipated. Throughout the investigation it was tried to relate the results of the test to practical applications with commercial exhaust brakes.

III. THEORY AND OPERATION

The theory of an exhaust brake is as follows. By partially closing the exhaust line during the exhaust stroke, the piston of the engine builds up a pressure in the exhaust line. After a few engine revolutions, this pressure will reach a semi-equilibrium condition where the pressure is varying about a mean value. The four cycles of the engine are now as follows.

When the intake valve begins to open, there is more pressure in the cylinder than in the intake manifold because of the exhaust backpressures in the cylinder and exhaust manifold. The air rushes into the intake manifold from the cylinder and exhaust line. The amount of valve overlap determines the length of time that this transfer of air will take place. Once the exhaust valve closes, the downward motion of the piston will draw in a fresh charge of air.

The compression stroke is carried out in the normal manner, and the air expands on the power stroke until the exhaust valve opens. The compression and expansion strokes are very similar in form, and the energy absorbed during these phases is relatively small as most of the energy of compression is regained during expansion. The exhaust valve then opens and the cylinder is pressurized to the

level of the exhaust backpressure. An approximate constant pressure process is carried out during the exhaust stroke and ends with the opening of the intake valve (10, p. 21). The extra energy required to produce the approximate constant pressure process during the exhaust stroke develops the additional braking effect on the engine. This effect therefore is dependent upon the amount of exhaust backpressure used and the valve overlap present in the engine (8, p. 75).

Exhaust brakes being manufactured in the United States at present are being produced by the Power Brake Equipment Company in Portland, Oregon. Their units may be adapted to any four cycle gasoline or diesel engine. Many trucks now are equipped at the factory with these units. The price is moderate, about $275 for a unit installation (13, p. 20). These units are designed for normally aspirated or supercharged engines. The weight of the complete unit is approximately 25 pounds.

Automobile Engineer describes the operation of an exhaust brake in the following manner.

> This unit incorporates a butterfly valve actuated by a compressed air or vacuum cylinder. The valve is eccentrically pivoted in such a way that the exhaust manifold pressure produces a torque reaction on its spindle. This reaction is balanced by the actuating cylinder, so that it is possible to regulate the amount of retardation by controlling the pressure in the pneumatic cylinder.

> The valve stop is adjusted in a manner such that, when the retarding valve is fully applied, the engine can still idle. This avoids any hesitation in the transition from the braked to the driving conditions; also, it enables upward gear shifts to be made while the brake is applied. There is a second valve in the pneumatic control system; this valve is actuated by the accelerator linkage and allows the retarding valve to open when the accelerator pedal is depressed. It helps to ensure that the engine response to the throttle control is instantaneous, so that downward gear changes can be made without difficulty. The arrangement of the linkage is also such that when the accelerator is released, the fuel-injected to the engine is positively restricted to that required for idling.
>
> By virtue of these features, the brake control can be left in the maximum retardation position at all times and the brake is immediately available not only on down gradients but also for slowing down under normal driving conditions. Also, faster upward shifts can be made, because the engine decelerates more rapidly when the throttle is released and the clutch is depressed. (1, p. 284)

All these features reportedly make the exhaust brake a practical, trouble-free piece of equipment to aid the commercial vehicle.

IV. APPARATUS

The apparatus used for testing was constructed for two purposes; first, to give measurements necessary for data collection in the tests that were conducted, simulating exhaust braking, and secondly to provide a flexible piece of equipment for further study by other students.

To study exhaust braking, it was necessary to measure various parameters in essentially equilibrium conditions. In addition, these parameters needed to be measured with the best accuracy possible using the measuring devices that were in the possession of the Mechanical Engineering Department or could be constructed inexpensively.

Engine and Accessories

The Mechanical Engineering Department was fortunate to have a Waukesha single cylinder engine, including a large supply of parts, donated by the California Research Corporation, Richmond, California. The engine was an overhead valve, manifold injection unit of 16.3 cubic inch displacement. Complete information regarding this engine is available in the Appendix. The engine will operate continuously at 2400 rpm with one atmosphere supercharge pressure.

The engine was installed on a suitable base and was

coupled directly to a 15 hp. direct current electric motor. Figure 1 shows a side view of the engine. The motor was cradled in bearings of large diameter and small ball size to minimize the friction on the trunions. As a direct current motor also acts as a generator by reversing the current flow, this provided a means of measuring power output while generating and power input when motoring. To start the engine and also to motor it during the exhaust braking tests, the direct current power was supplied from a portable ac to dc converter, motor-generator set of 15 hp. capacity.

The torque of the engine, both delivered and absorbed, was measured directly on a dial spring scale. Figure 2 shows the control panel; scale is on the extreme left. This type of torque measuring device will give readings with less than one per cent error. A strobotach was used to determine the speed of the engine. Since the line voltage reverses itself 3600 times per minute, any fraction of this frequency can be determined with very little error (less than one per cent). The speed of the engine could be increased or decreased by changing the field voltage on the electric motor.

To damp out the pressure pulses that are created by a single cylinder engine in the intake and exhaust passages, tanks with a volume greater than fifty times the piston

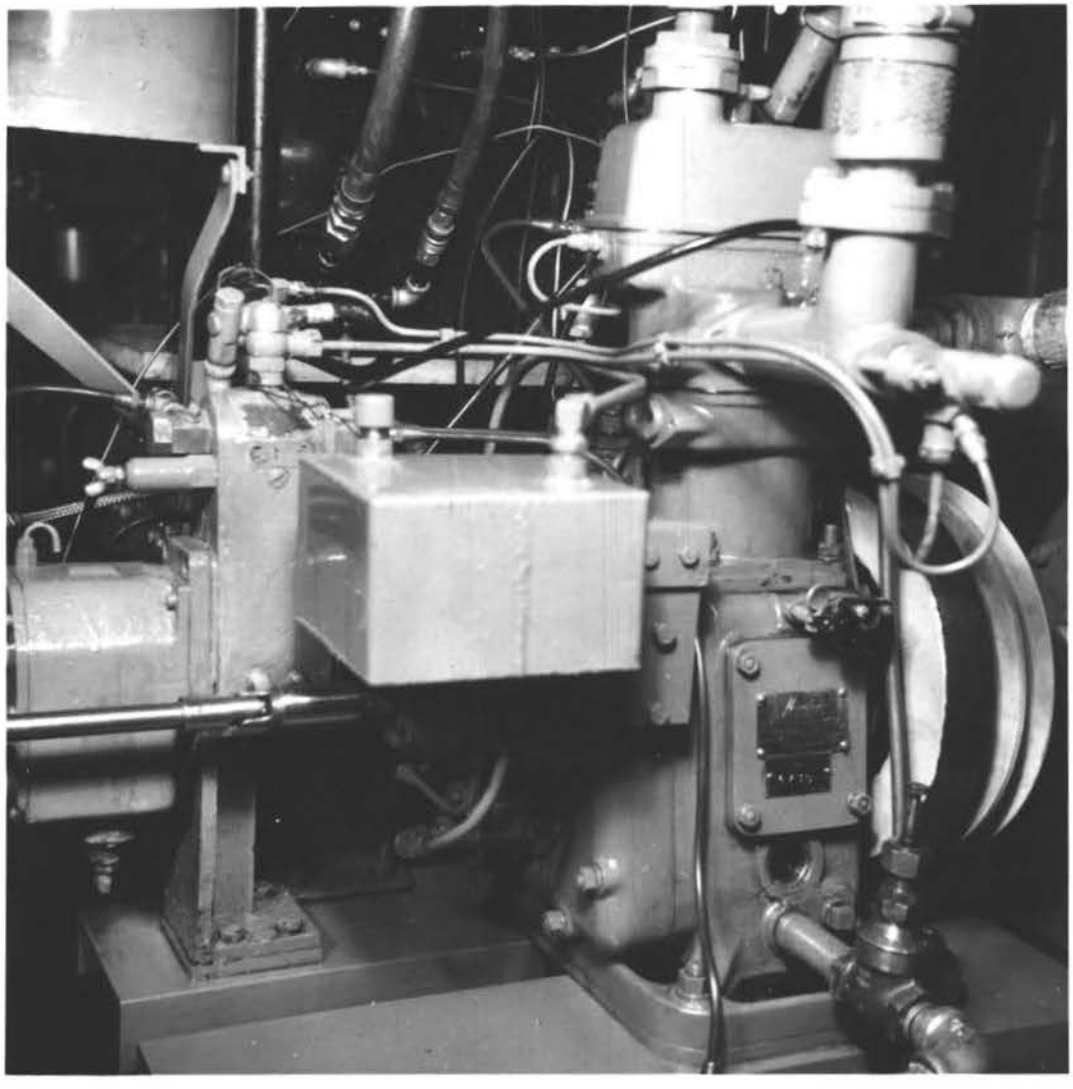

Figure 1. Side view of test engine.

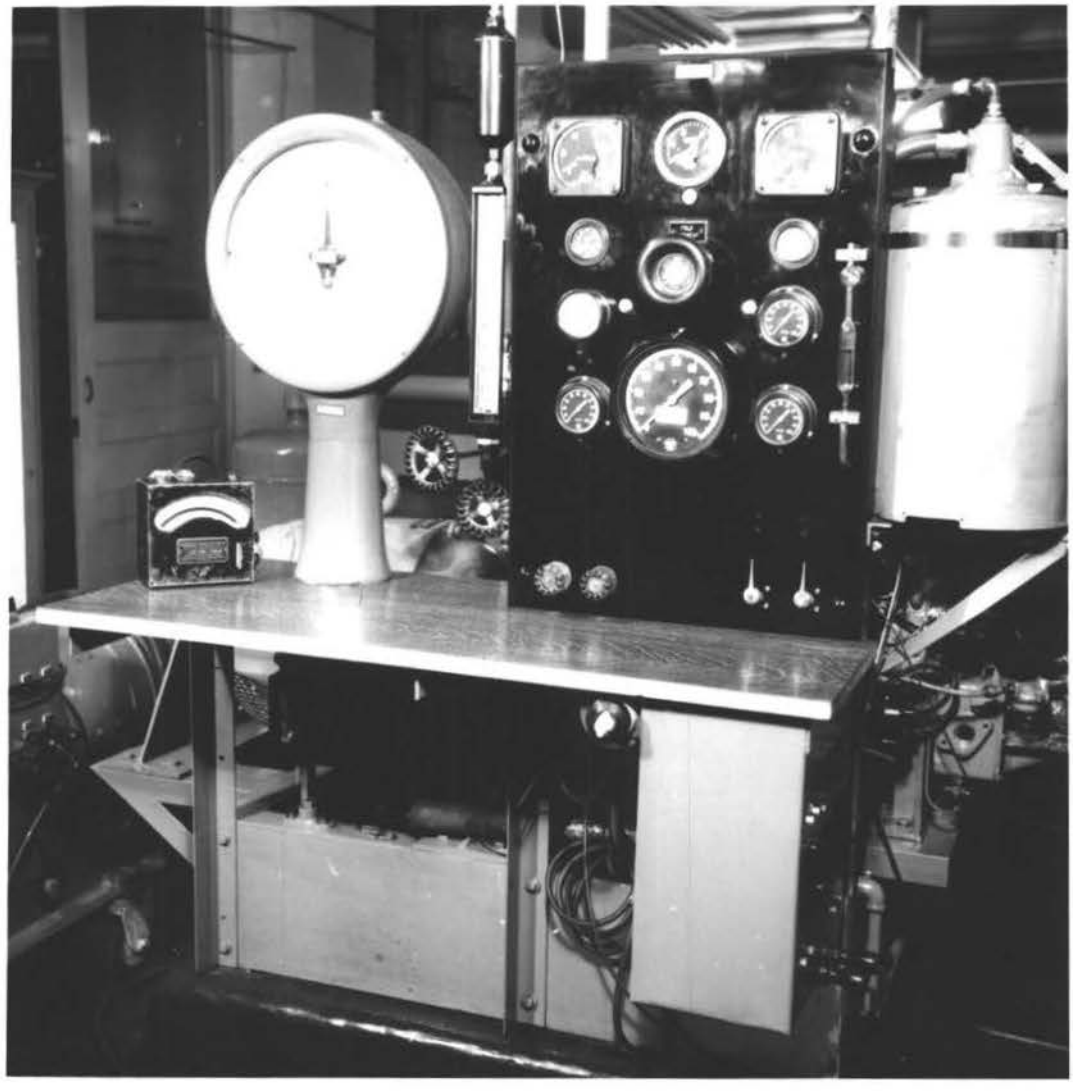

Figure 2. Control panel and dial scale.

displacement were installed on both exhaust and inlet lines (15, p. 9). In order to measure the air flow, a stainless steel orifice of 0.500 inch diameter was installed in the inlet air line. The pressure drop across the orifice was measured by a ten inch water manometer with 1/10 inch divisions. Air flow weights were computed for different upstream pressures and pressure drops across the orifice. Further information regarding these calculations may be found in the Appendix.

During the exhaust braking tests, the exhaust backpressure was manually controlled by regulating a gate valve in the exhaust line. The exhaust pressure was measured by a bourdon-tube gauge and the exhaust temperature by a chromel-alumel thermocouple located near the exhaust port.

An array of other instruments were used to measure the conditions under which the engine was operating. Among these were; oil and water temperature, oil pressure, inlet air temperature and pressure, and engine speed. All of these instruments were calibrated before their installation to insure accurate readings.

Pressure-Time Indicator

To determine the pressure in the cylinder of the engine during the exhaust brake tests, an instrument giving a pressure time diagram was needed. A custom built

balanced pressure indicator was available for this purpose. Figure 3 shows an overall view of this instrument in the foreground. The instrument uses a pressure pick-up mounted in the spark-plug hole of the engine. As the engine used had dual spark-plug holes, one was used for ignition, the other for the pressure pick-up.

The working part of the pressure pick-up consists of a thin diaphragm which is free to move up and down. Motion of the diaphragm opens and closes an electrical circuit. On one side of the diaphragm is the dynamic pressure in the cylinder of the engine; on the other, a controlled but variable nitrogen pressure. Making or breaking of the electrical circuit is accomplished when the cylinder pressure exceeds or drops below a set nitrogen pressure.

The main part of the instrument consists of a drum which rotates at crankshaft speed and an electrical circuit capable of producing a high voltage spark whenever the pressure pick-up opens or closes the circuit. The nitrogen pressure in the pick-up is also transmitted to a piston and cylinder arrangement. The force produced on the piston under the action of the nitrogen pressure is balanced by a spring which in turn is connected by a mechanical linkage to a pointer which is located immediately above the drum. The pointer is attached to the high voltage source.

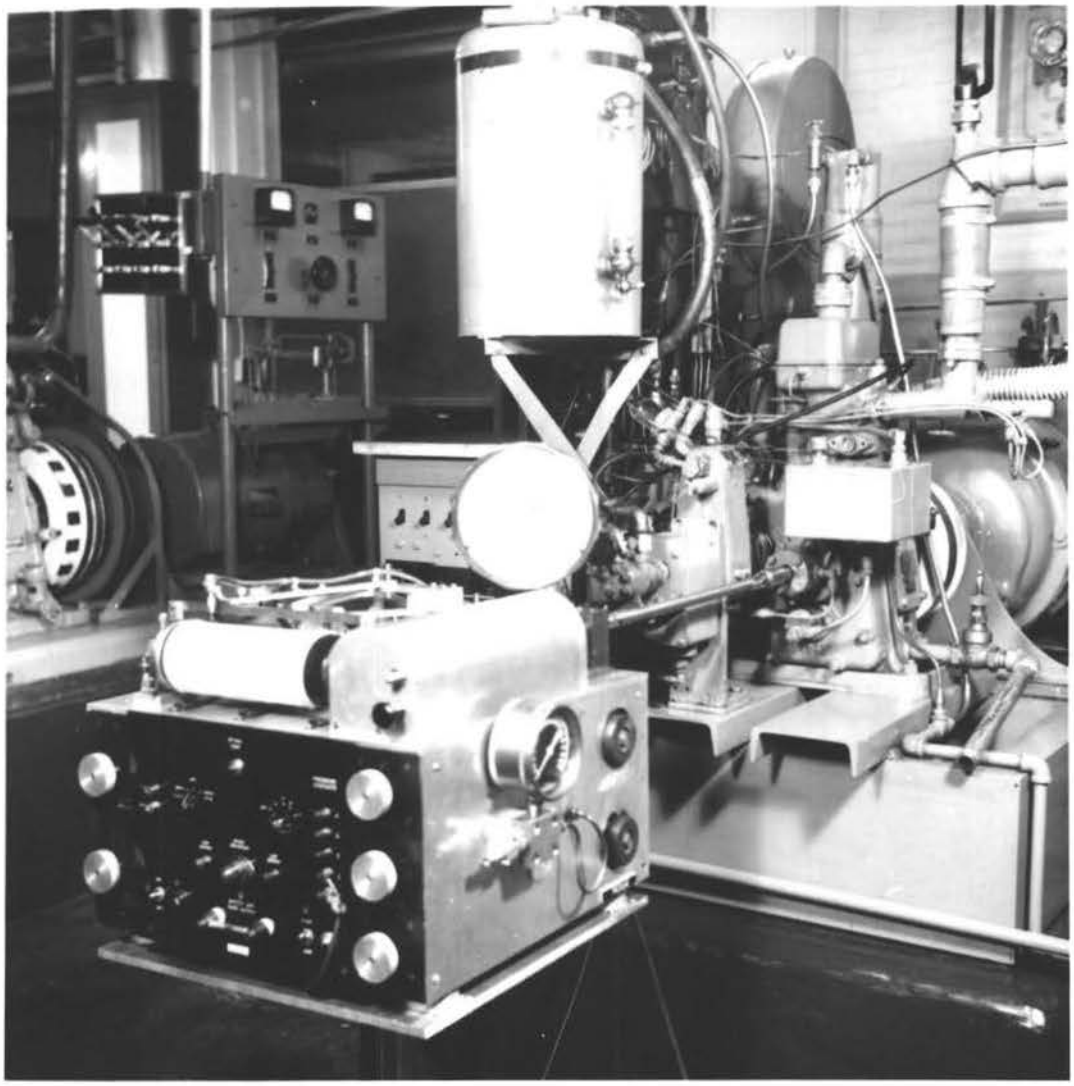

Figure 3. Balanced pressure indicator in foreground, engine and dynamometer in background, right hand side.

Consequently, whenever the pressures are equal in the pick-up, a spark is produced at a certain point on the drum. It is therefore possible to make pressure-time diagrams by using a range of nitrogen pressures.

The pressure pick-up is sensitive to 0.07 psi difference between the sides of the diaphragm and manufacturers of these instruments claim that the accuracy is within one per cent.

V. TESTING PROCEDURE

Measuring the Braking Effect

As the object of this investigation was to determine how exhaust pressure affected motoring horsepower, the exhaust pressure was the main variable in all of the tests made. It was decided to use a range of pressures from zero to 50 psig. These pressures are representative of those found in exhaust manifolds when commercial exhaust brakes are used. Tests were conducted at ten psi increments.

To determine how the braking effect would change with speed, three speeds were chosen; 1200, 1800, and 2400 rpm. These speeds were chosen for two reasons. First, they are representative of the range of speeds for the modern diesel or gasoline engine, and secondly, these speeds may be easily compared to the frequency of a strobotac (3600 cpm).

Because many diesel engines are supercharged to 5 to 7 psig pressure, additional tests were conducted at five psig intake manifold pressure to determine this effect on motoring horsepower.

Finally, in order to insure reliability, at least two points for each combination of speeds and pressures were taken. Therefore, it was required to take a minimum of 72 separate data points.

Tests were conducted in the following manner. First, the engine was started and allowed to reach operating temperature, 212°F. Second, the oil temperature was brought up to 150°F by the use of an oil heater. The fuel to the injector was then shut off and the engine motored. The exhaust pressure, intake pressure, and speed were regulated until they were the desired test conditions. It took a minimum of fifteen minutes to reach equilibrium conditions. The exhaust temperature was the best indicator to tell when equilibrium conditions were established. When equilibrium conditions were established, the following data were taken: engine rpm, exhaust pressure, exhaust temperature, scale load, intake temperature and pressure, pressure drop across orifice, orifice upstream pressure, oil and water temperature, and oil pressure.

Throughout the tests, the oil temperature was held as constant as possible (plus or minus 5 degrees) in order to minimize the variation of oil drag on the moving parts of the engine. Water jacket temperature would not hold constant as it depended on the exhaust pressure that was being used (this will be discussed in greater detail later). Periodically it was required to restart the engine to eliminate the possibility of any oil build up in the combustion chamber. Before the conclusion of the exhaust braking tests, the full load horsepower of the

engine was determined in order to compare braking to output horsepower. During the output horsepower tests, the air-fuel ratio and spark advance were varied to give maximum power for the speeds used.

Measuring Dynamic Cylinder Pressure

It was decided to obtain pressure-time diagrams at three points. These points were at 1800 rpm with zero, 20 and 40 psig exhaust pressure. Care was taken to change the nitrogen pressure at the same rate in all three cases. The pressure was increased to 220 psig and then bled off very slowly to obtain as many points as possible on the pressure-time diagrams. Again equilibrium conditions were established before any diagrams were made.

VI. RESULTS

The main variables of all the experimental points which were taken are listed in the last section of the Appendix. All of the data that were taken were not listed, only the variables which were important in the analysis of exhaust braking. The horsepower output or absorption of the engine was computed using the equation, H.P. = (Scale reading) (RPM) / 51,600; 51,600 being the dynamometer constant for the installation. Air flow values were determined by using Figure 18 which is located in the front section of the Appendix.

The results of the tests conducted are best represented by a series of 13 curves. These curves give plots of the parameters that the author felt were important to represent the variables.

Horsepower Absorption

Figure 4 shows the increase of motoring horsepower that was required for the range of exhaust pressures used. Note that the curve for 1800 rpm is almost a straight line and the ones for 1200 and 2400 rpm bend downward and upwards respectively with increasing exhaust pressure. Also note that the 5 psig manifold pressure curves are below the zero manifold pressure curves. This is slightly

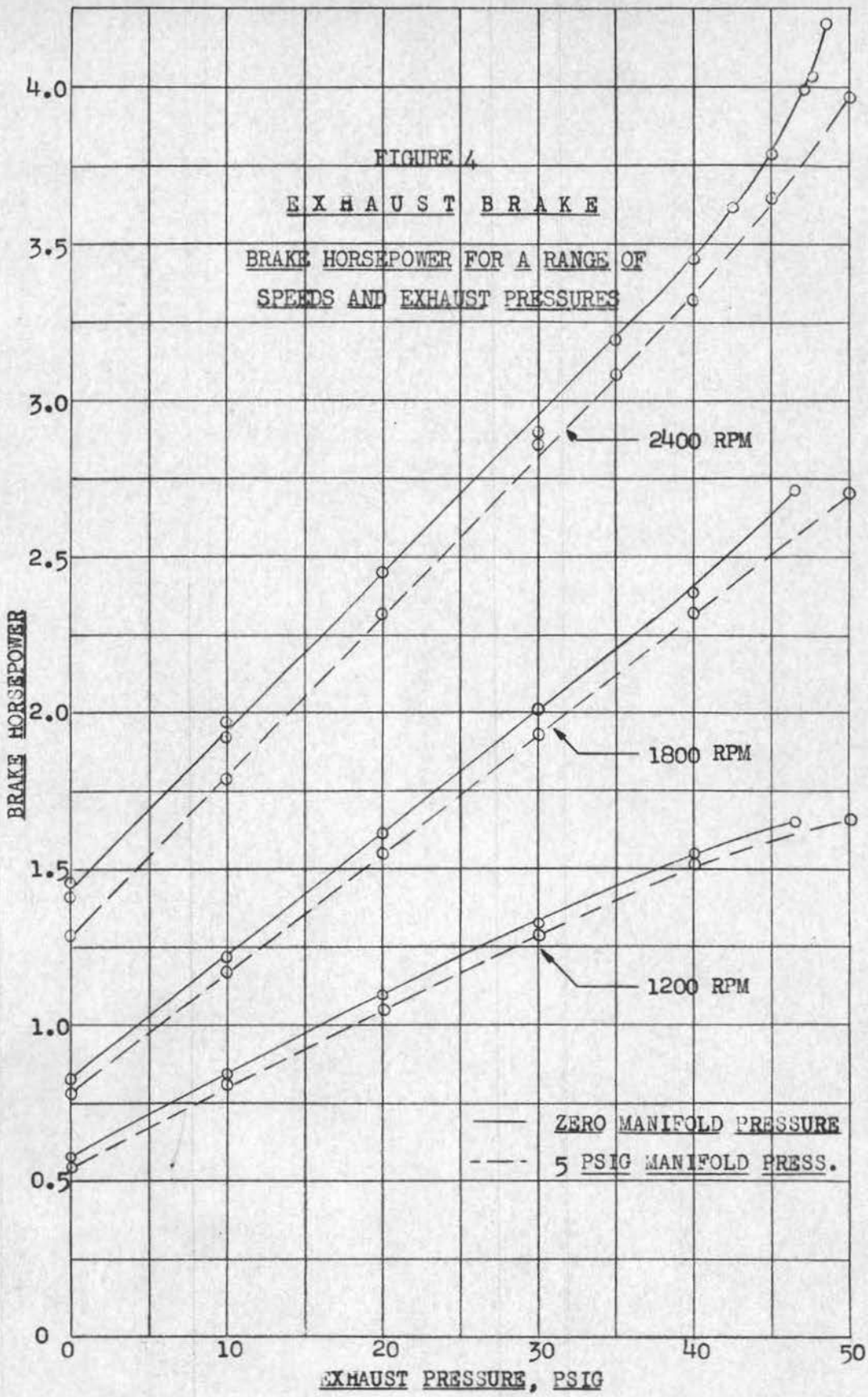
FIGURE 4
EXHAUST BRAKE
BRAKE HORSEPOWER FOR A RANGE OF
SPEEDS AND EXHAUST PRESSURES
2400 RPM
1800 RPM
1200 RPM
ZERO MANIFOLD PRESSURE
5 PSIG MANIFOLD PRESS.
BRAKE HORSEPOWER
EXHAUST PRESSURE, PSIG
0
0.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0
0
10
20
30
40
50

misleading as the horsepower values shown do not include the energy necessary to compress the inlet air to 5 psig. These values of additional energy were computed. A computation procedure for this additional energy was derived and was included in the Appendix of this presentation. Figure 5 indicates the sum of the motoring and computed horsepower required to supercharge the intake air. Summing the two, produces curves which are more representative of the actual condition. Therefore, the supercharged braking horsepower is greater than the normally aspirated condition.

Figure 6 indicates the increase in the effectiveness of the exhaust brake, calculated as the per cent of output horsepower at 2400 rpm, as exhaust pressures and engine speeds increase. The bottom curve, for zero exhaust pressure, is the normal friction horsepower of the engine.

Figure 7 gives the per cent of absorbed horsepower from exhaust braking compared to that which the engine will produce at corresponding rpm. The percentage was computed using the absorbed and output horsepower for each of the two speeds indicated. Note that at 45 psig exhaust pressure, the retardation is approximately 75 per cent of the output horsepower of the engine, either developed at 1800 or 2400 rpm.

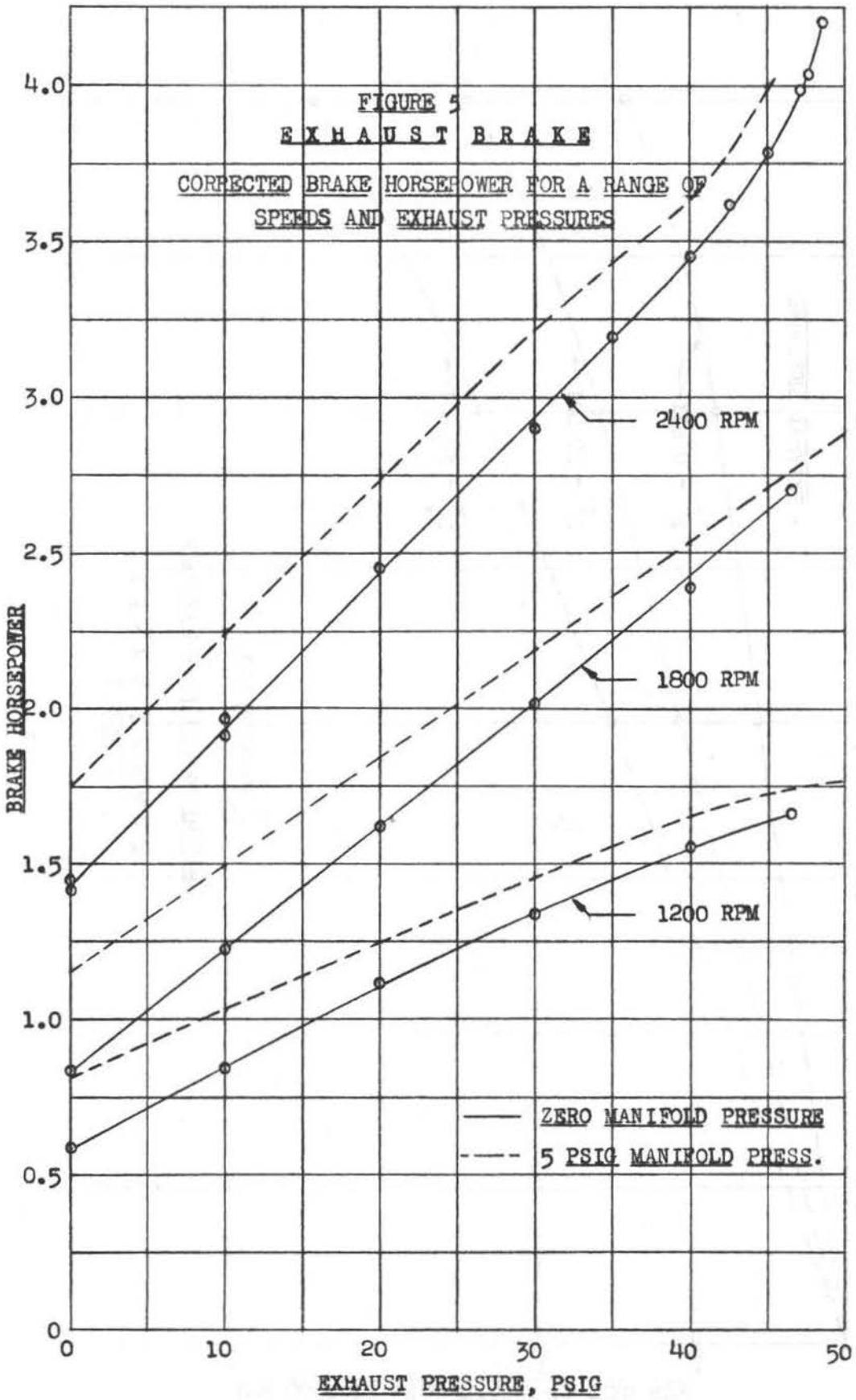
FIGURE 5
EXHAUST BRAKE
CORRECTED BRAKE HORSEPOWER FOR A RANGE OF SPEEDS AND EXHAUST PRESSURES
2400 RPM
1800 RPM
1200 RPM
ZERO MANIFOLD PRESSURE
5 PSIG MANIFOLD PRESS.
BRAKE HORSEPOWER
EXHAUST PRESSURE, PSIG
0
0.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0
0
10
20
30
40
50

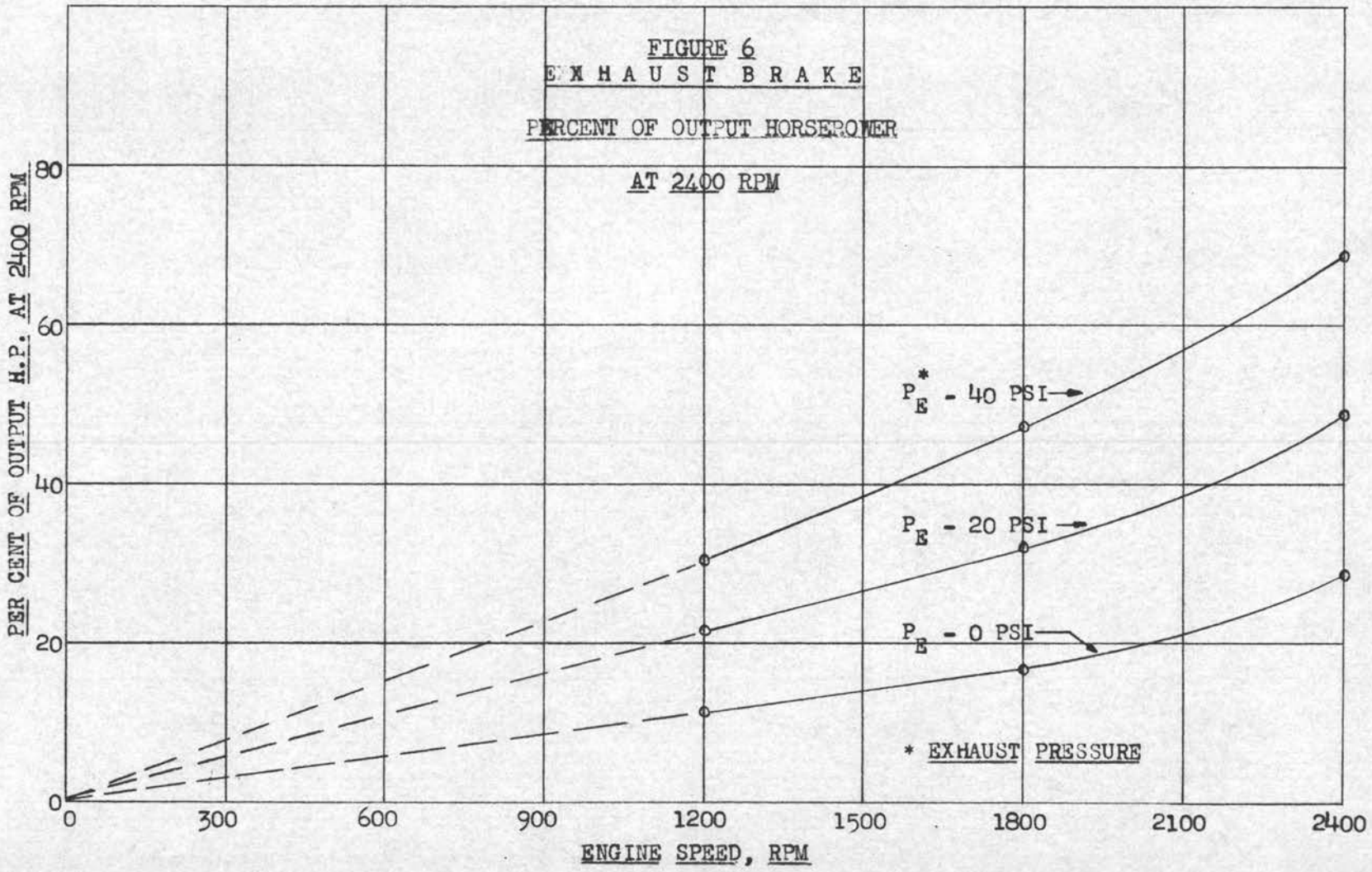
FIGURE 6
EXHAUST BRAKE
PERCENT OF OUTPUT HORSEPOWER
AT 2400 RPM
PER CENT OF OUTPUT H.P. AT 2400 RPM
0
20
40
60
80
0
300
600
900
1200
1500
1800
2100
2400
ENGINE SPEED, RPM
P_E^* - 40 PSI
P_E - 20 PSI
P_E - 0 PSI
* EXHAUST PRESSURE

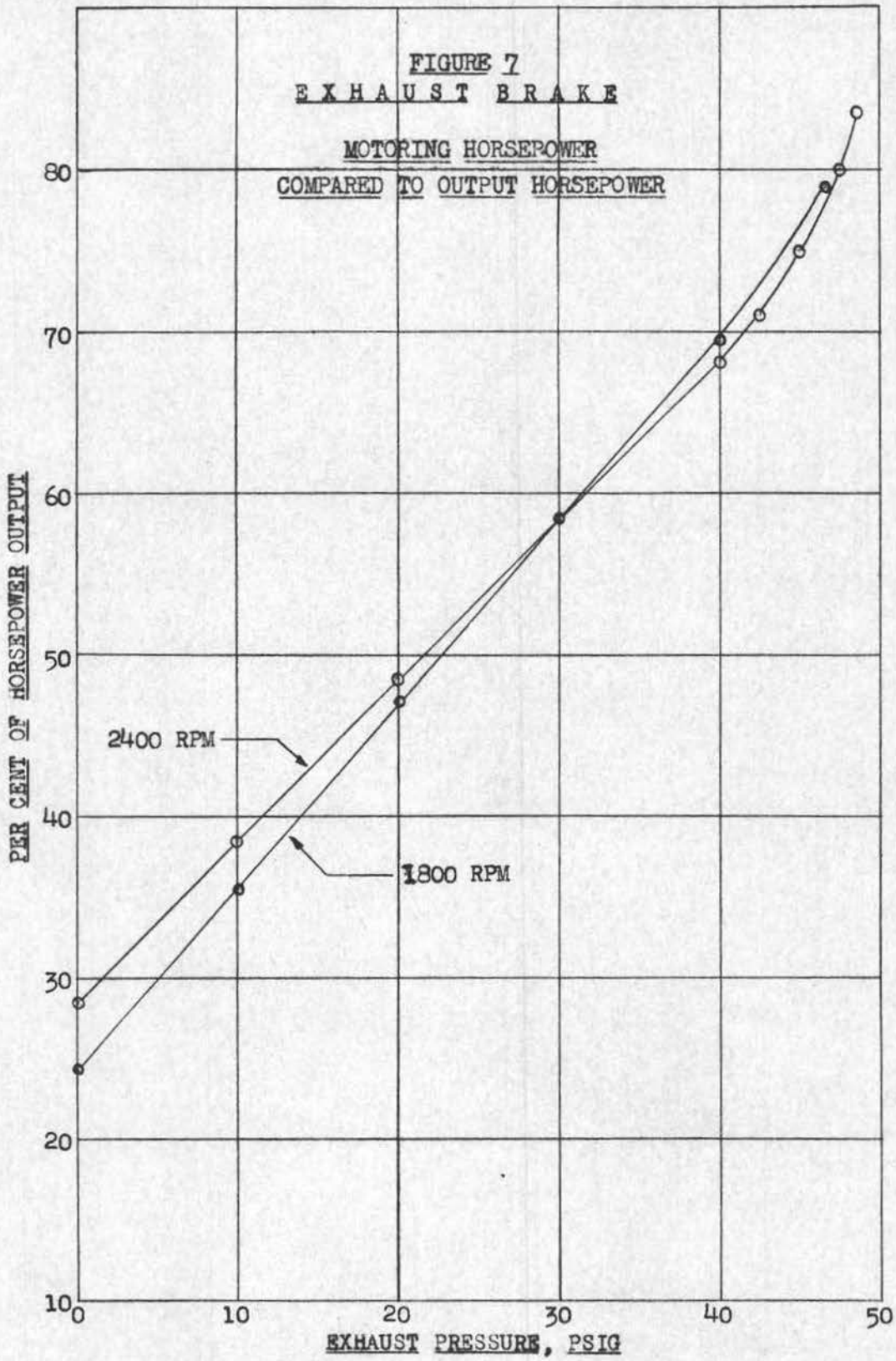
FIGURE 7
EXHAUST BRAKE
MOTORING HORSEPOWER
COMPARED TO OUTPUT HORSEPOWER
PER CENT OF HORSEPOWER OUTPUT
80
70
60
50
40
30
20
10
0
10
20
30
40
50
EXHAUST PRESSURE, PSIG
2400 RPM
1800 RPM

This is an important point since 45 psig is the exhaust pressure that is normally used in conjunction with exhaust brakes on unsupercharged engines. The points at zero backpressure give the friction horsepower of the engine, representing about 25 per cent of the output horsepower. Therefore, the extra retardation of the exhaust brake is nearly 50 per cent of the output horsepower of the engine.

Operating Variables

Figure 8 gives an indication of the exhaust temperatures for different exhaust pressures. Note that at the supercharged condition, the exhaust temperatures are considerably lower, especially at high exhaust pressures.

Figure 9 represents the decrease in air flow through the engine when exhaust pressure is increased. Note that above 45 psig, the air flow decreases sharply. Figure 10 represents the supercharged case with 5 psig intake manifold pressure where the air flow does not decrease so markedly at the lower speeds, at least up to 50 psig, the highest exhaust pressure used. Also note that the slope of the air flow lines is the same for three different speeds used.

Figure 11 gives the equilibrium water jacket temperature of the engine for a range of exhaust temperatures. Figure 12 indicates the variation of water jacket

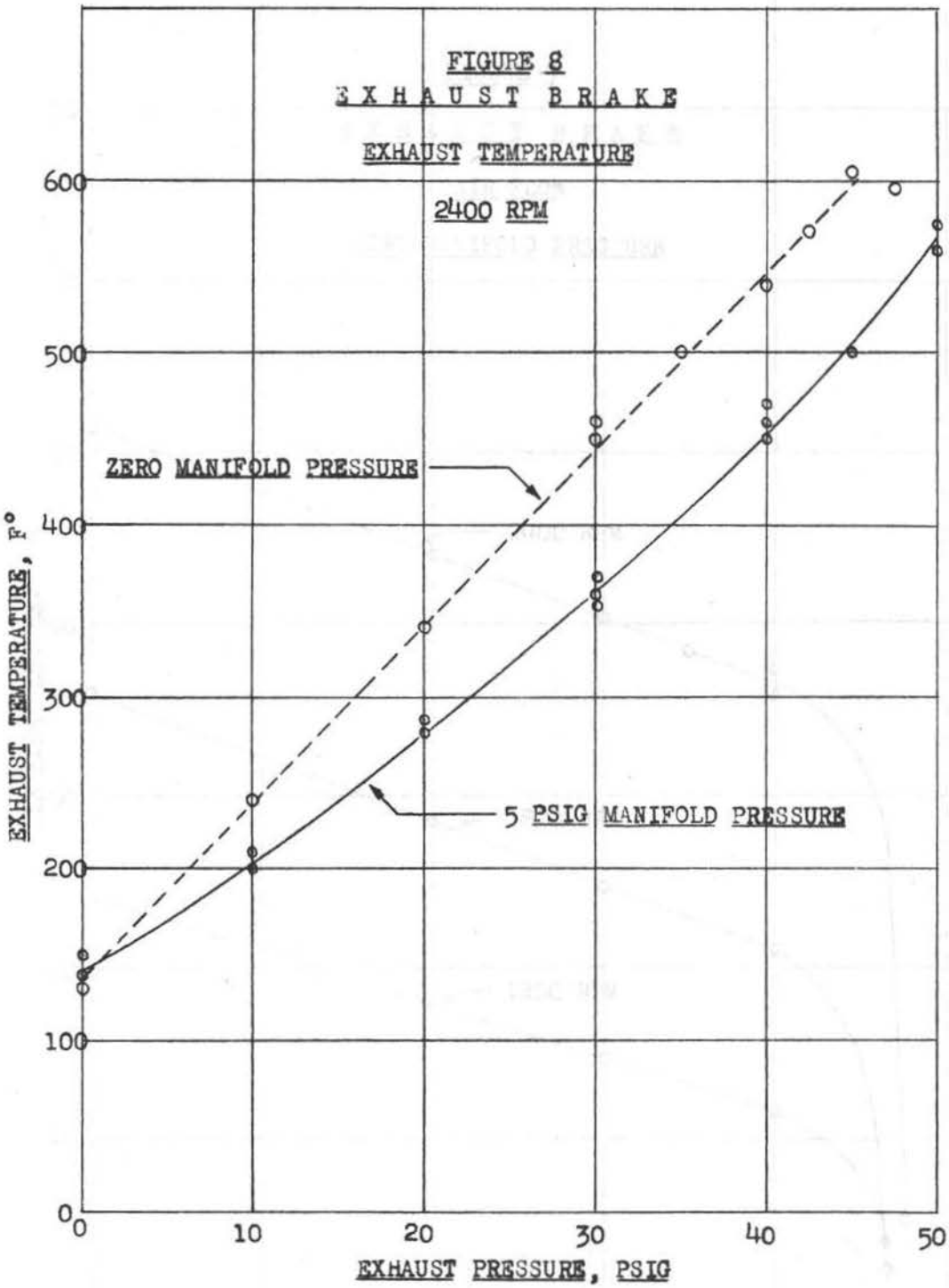
FIGURE 8
EXHAUST BRAKE
EXHAUST TEMPERATURE
2400 RPM
ZERO MANIFOLD PRESSURE
5 PSIG MANIFOLD PRESSURE
EXHAUST TEMPERATURE, F°
EXHAUST PRESSURE, PSIG
0
100
200
300
400
500
600
0
10
20
30
40
50

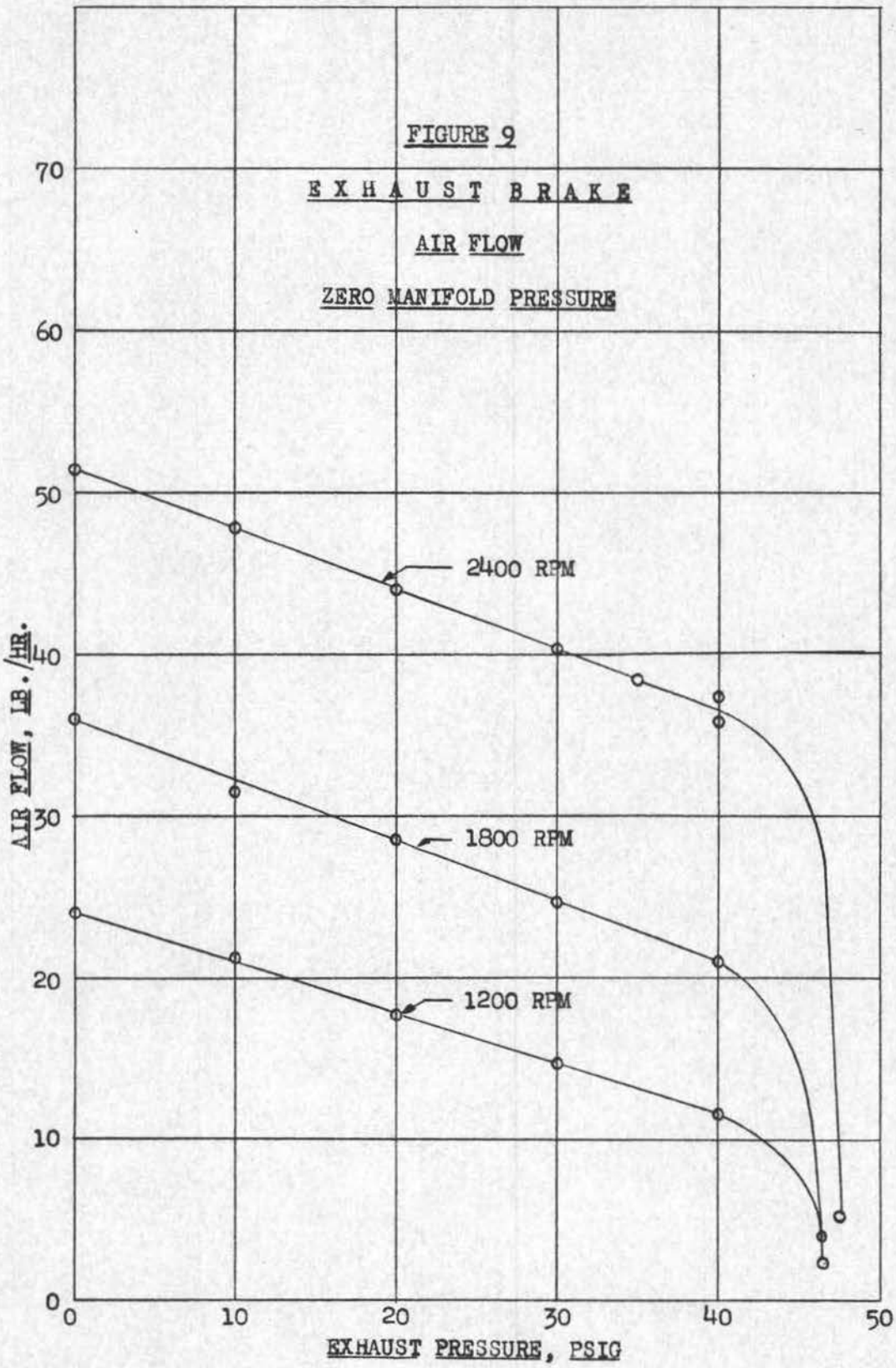
FIGURE 9
EXHAUST BRAKE
AIR FLOW
ZERO MANIFOLD PRESSURE
2400 RPM
1800 RPM
1200 RPM
AIR FLOW, LB./HR.
EXHAUST PRESSURE, PSIG
0
10
20
30
40
50
60
70

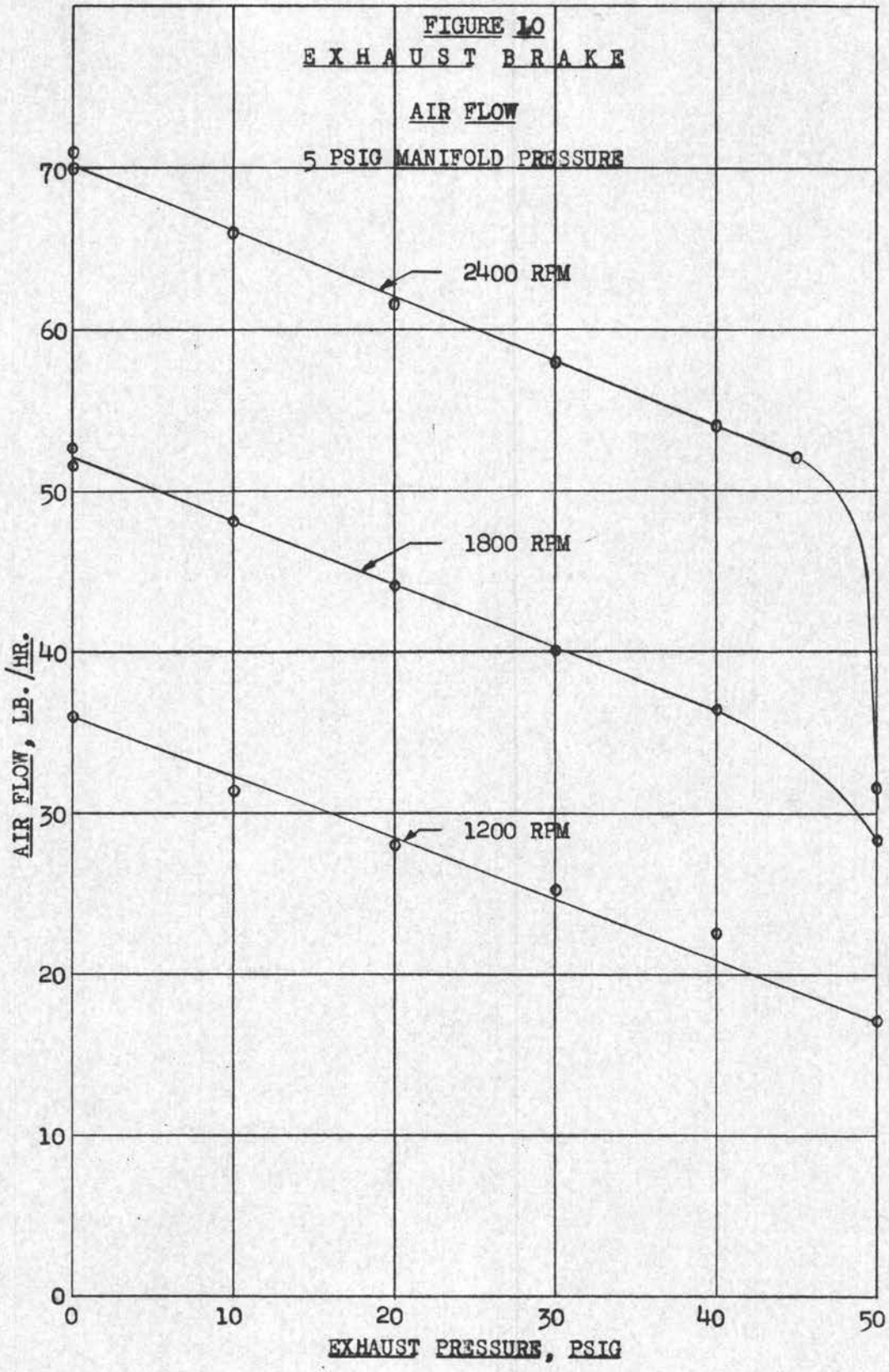
FIGURE 10
EXHAUST BRAKE
AIR FLOW
5 PSIG MANIFOLD PRESSURE
2400 RPM
1800 RPM
1200 RPM
AIR FLOW, LB./HR.
EXHAUST PRESSURE, PSIG
0
10
20
30
40
50
60
70

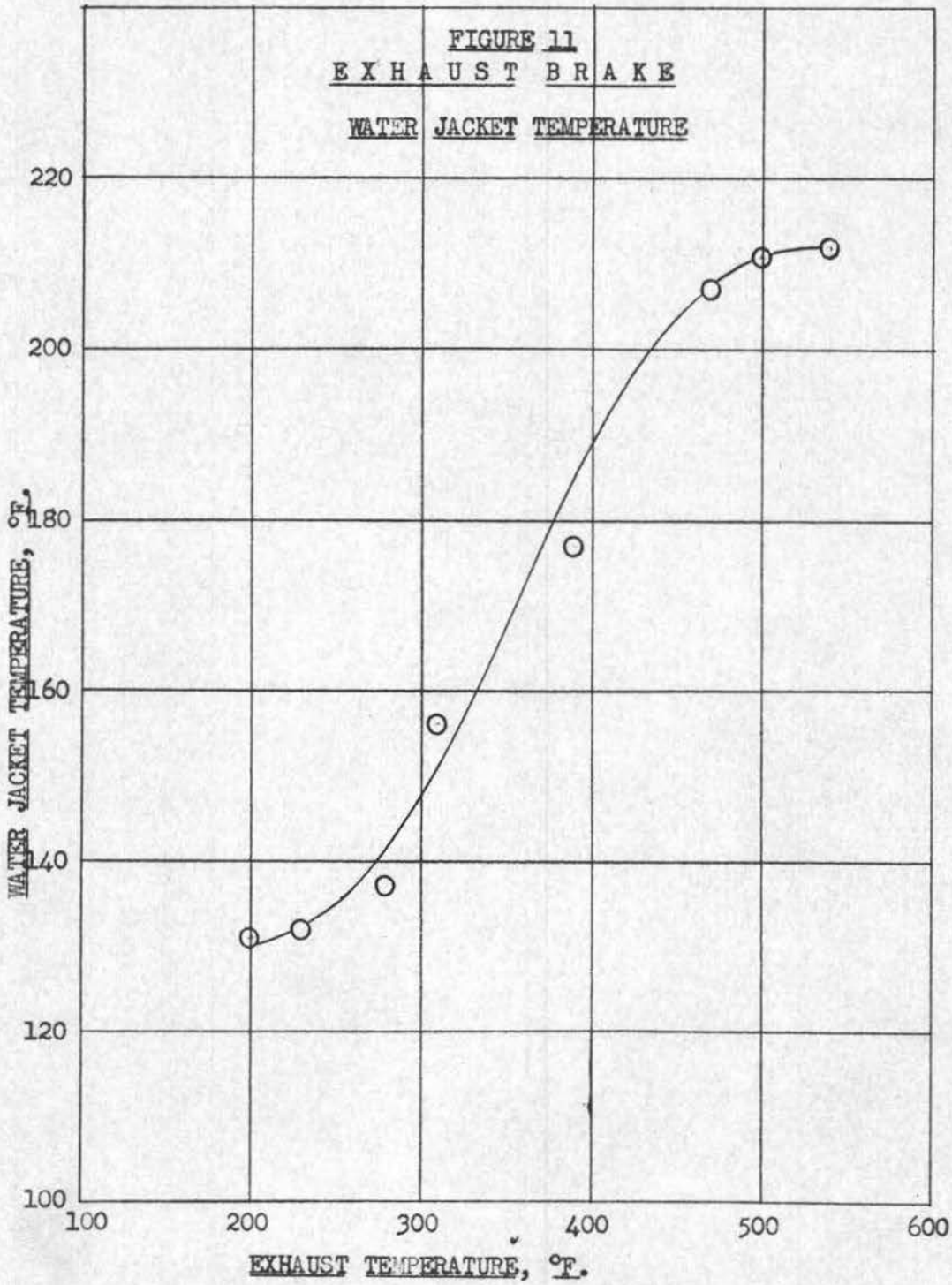
FIGURE 11
EXHAUST BRAKE
WATER JACKET TEMPERATURE
220
200
180
160
140
120
100
WATER JACKET TEMPERATURE, °F.
100
200
300
400
500
600
EXHAUST TEMPERATURE, °F.

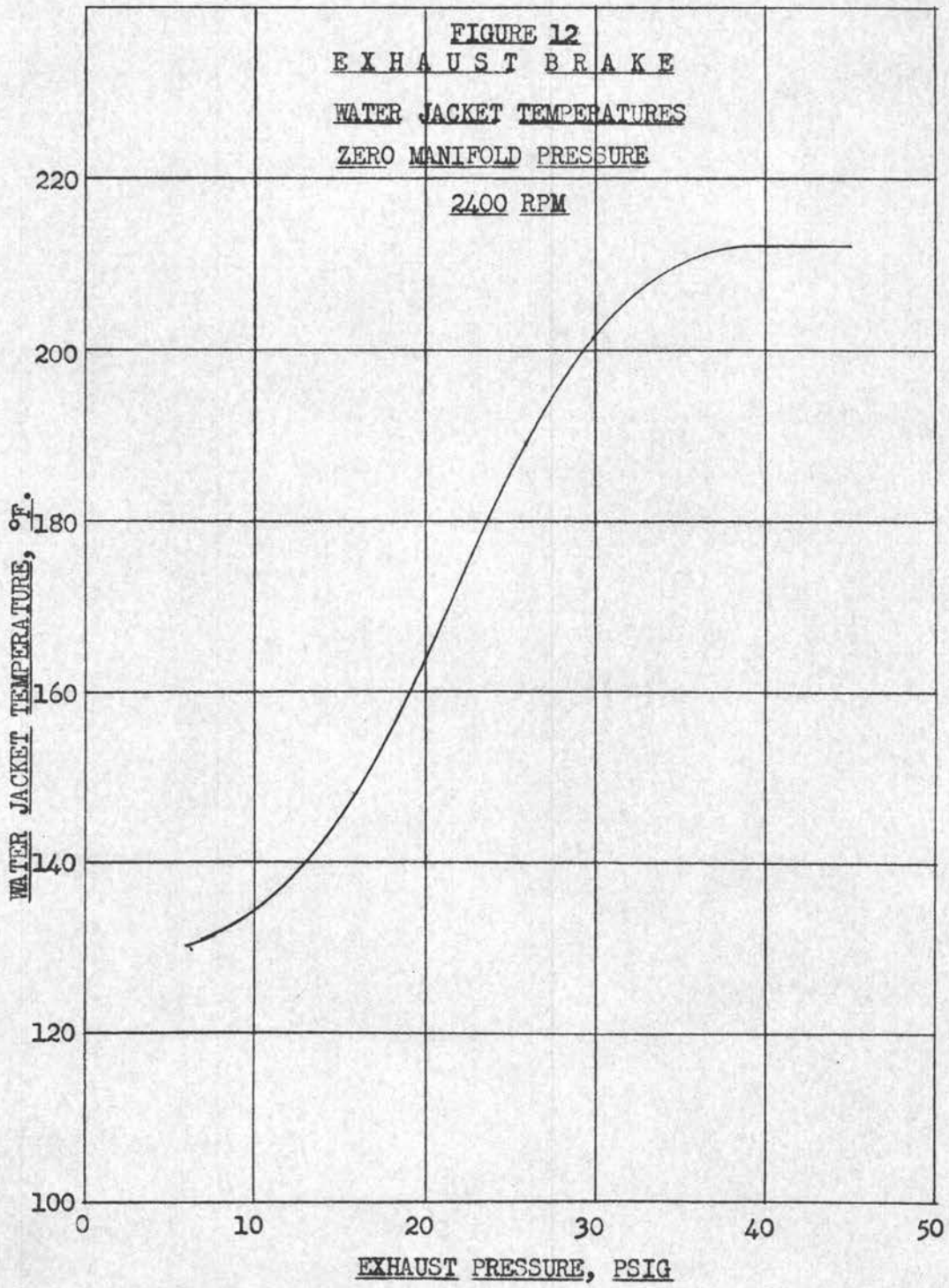
FIGURE 12
EXHAUST BRAKE
WATER JACKET TEMPERATURES
ZERO MANIFOLD PRESSURE
2400 RPM
220
200
180
160
140
120
100
WATER JACKET TEMPERATURE, °F.
0
10
20
30
40
50
EXHAUST PRESSURE, PSIG

temperature for increased exhaust pressures. This curve was generated by cross-plotting Figures 8 and 11. Note the values of Figure 12 are for 2400 rpm and zero manifold pressure. The results of Figures 11 and 12 only apply to the engine tested.

Dynamic Cylinder Pressures

The pressure-time diagram for dynamic pressures in the cylinder of the test engine without exhaust pressure is presented by Figure 13. The expansion and compression strokes are very similar in form. In fact, they are almost identical. It was not possible to record pressures below atmospheric with the balanced-pressure indicator. Therefore, the intake stroke is not shown. During the exhaust stroke, the exhaust pressure was nearly at atmospheric pressure and no distinction between the two could be made.

The effect of 20 psig exhaust pressure is shown clearly in Figure 14. Note that during the exhaust stroke, the pressure is not a constant value, but varies. The dynamic pressure increases to above 20 psig at the beginning of the stroke, then drops in the center, and finally increases at the conclusion of the stroke where the piston is reaching the limit of its upward travel.

Figure 15 indicates the effect on the pressure-time diagram when 40 psig exhaust pressure is used. The shape

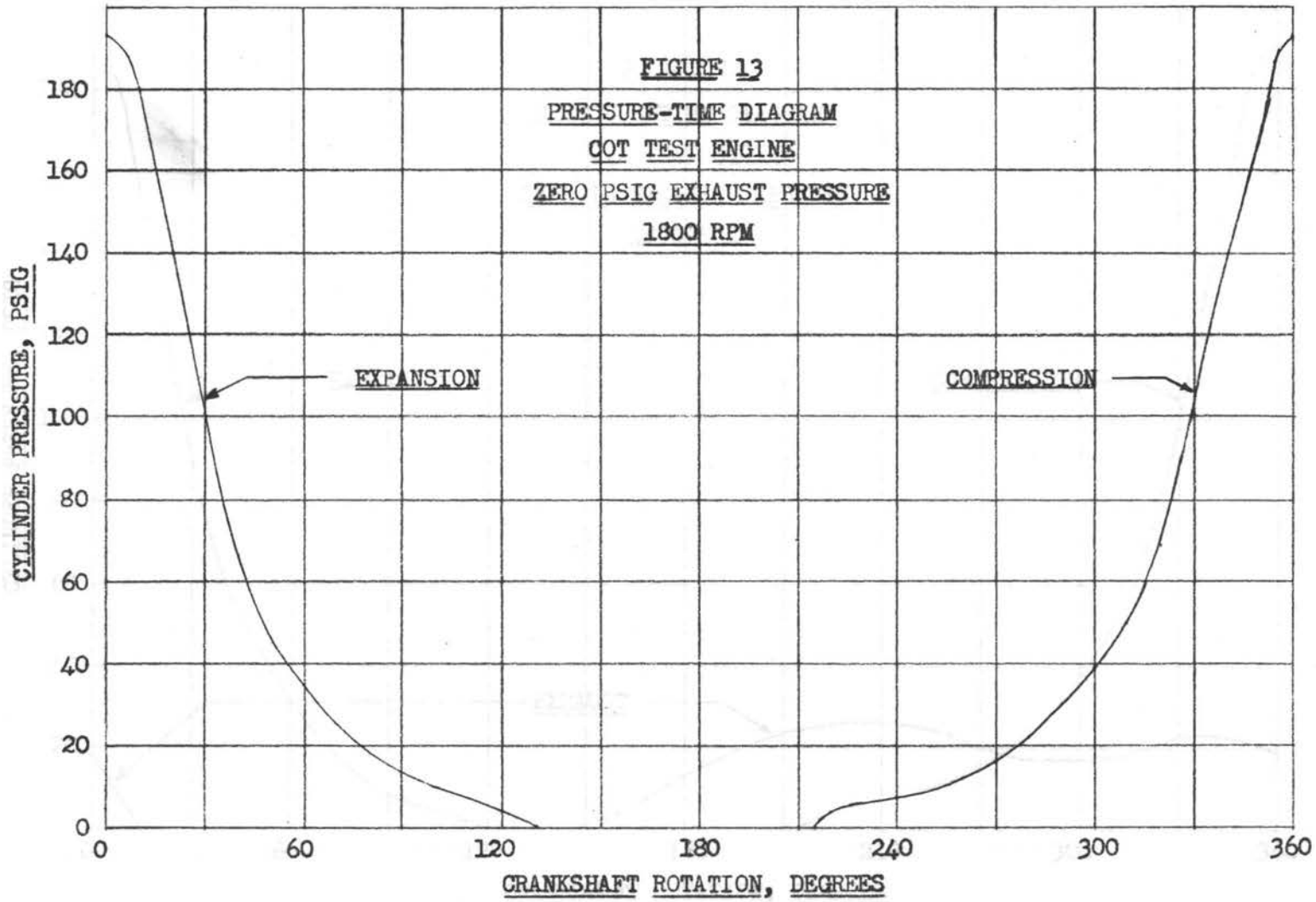
FIGURE 13
PRESSURE-TIME DIAGRAM
COT TEST ENGINE
ZERO PSIG EXHAUST PRESSURE
1800 RPM
EXPANSION
COMPRESSION
CYLINDER PRESSURE, PSIG
0
20
40
60
80
100
120
140
160
180
0
60
120
180
240
300
360
CRANKSHAFT ROTATION, DEGREES

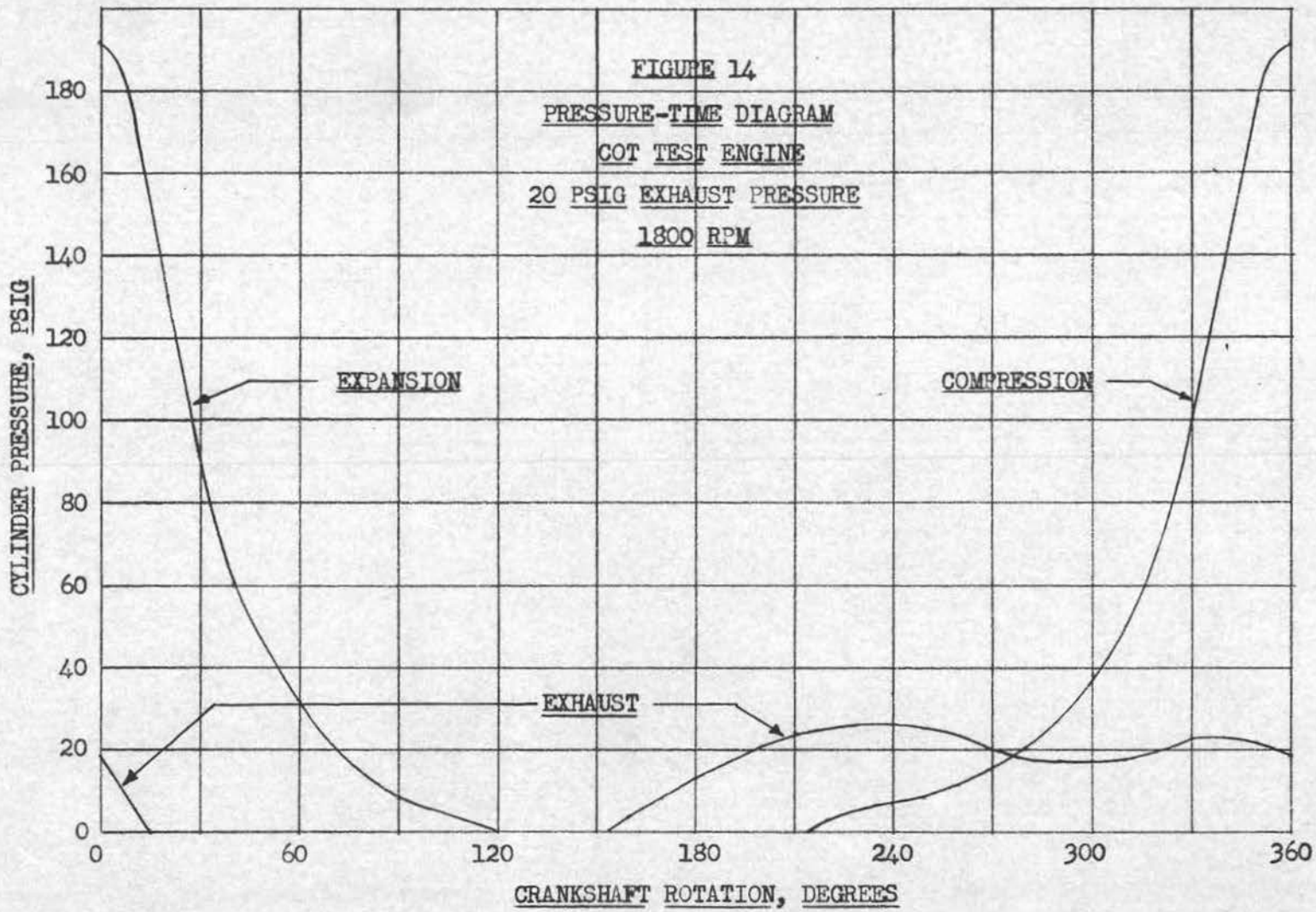
FIGURE 14
PRESSURE-TIME DIAGRAM
COT TEST ENGINE
20 PSIG EXHAUST PRESSURE
1800 RPM
EXPANSION
COMPRESSION
EXHAUST
CYLINDER PRESSURE, PSIG
0
20
40
60
80
100
120
140
160
180
0
60
120
180
240
300
360
CRANKSHAFT ROTATION, DEGREES

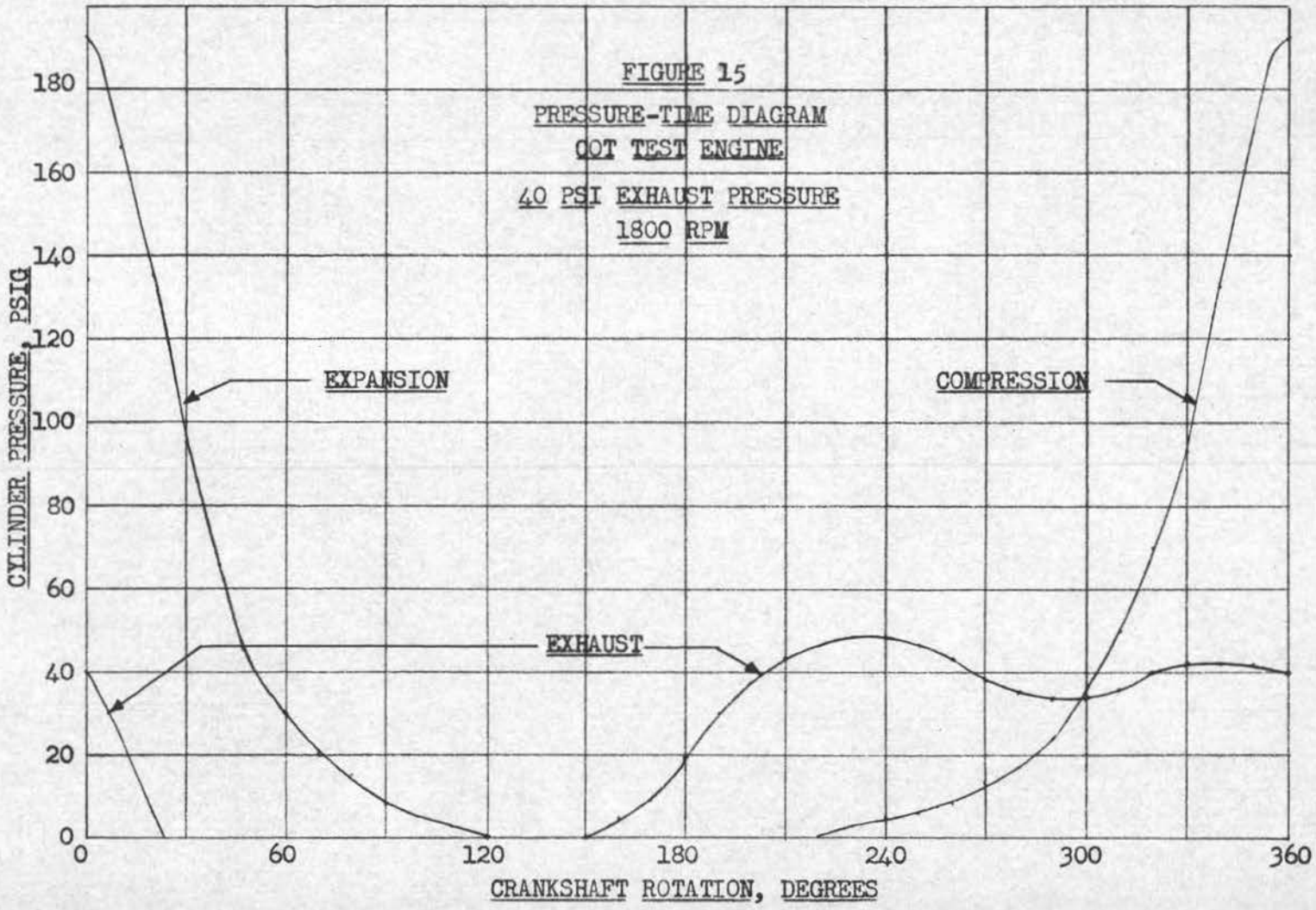
FIGURE 15
PRESSURE-TIME DIAGRAM
COT TEST ENGINE
40 PSI EXHAUST PRESSURE
1800 RPM
EXPANSION
COMPRESSION
EXHAUST
CYLINDER PRESSURE, PSIG
0
20
40
60
80
100
120
140
160
180
0
60
120
180
240
300
360
CRANKSHAFT ROTATION, DEGREES

of the curve for the exhaust stroke is very similar to Figure 14. Comparing Figures 13, 14, and 15, the expansion and compression strokes have the same form and all reach the same maximum pressure, approximately 192 psig. Also the pressure rise, due to the exhaust stroke, begins at nearly the same point in each diagram.

Figure 16 gives a comparison of 20 and 40 psig exhaust pressures, plotted on pressure-volume coordinates. Note that the dynamic pressure for 20 and 40 psig vary about a mean, approximately equal to the exhaust pressures for which the diagrams were taken.

Transferring from pressure-time to pressure-volume coordinates changes the exhaust pressure curves significantly. Points on the pressure-time curves near the top and bottom of the stroke, where the cylinder volume is changing very slowly for each degree of crank angle, are very close together on pressure-volume coordinates. Therefore, the exhaust pressure decreases to zero after opening the intake valve without an appreciable amount of piston motion even though 15 to 20° of crankshaft rotation have occurred.

Notice the area inside the pressure-volume diagram. This area represents the work that the piston of the engine does on the exhaust gases during each exhaust stroke.

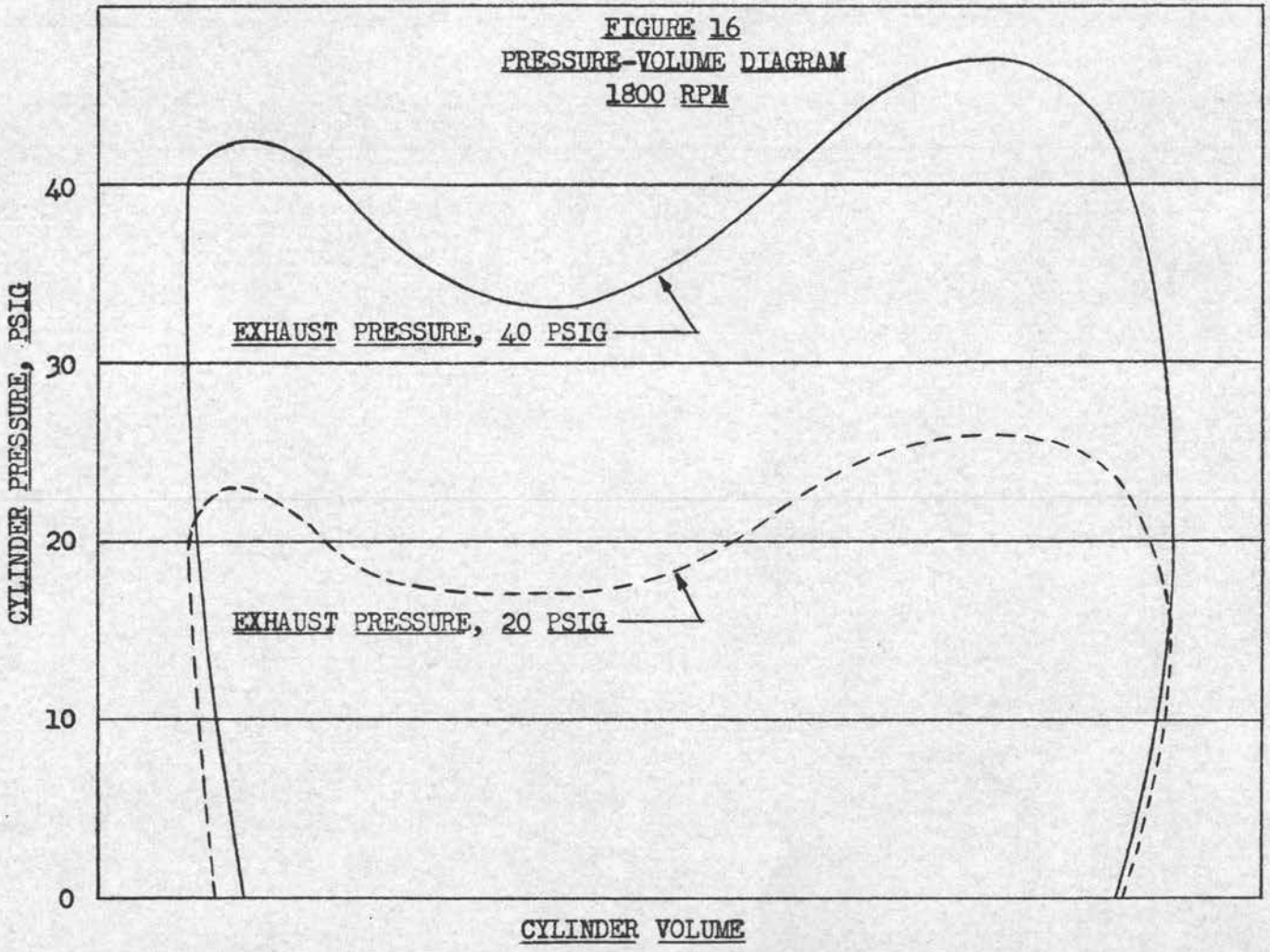
FIGURE 16
PRESSURE-VOLUME DIAGRAM
1800 RPM
CYLINDER PRESSURE, PSIG
40
30
20
10
0
EXHAUST PRESSURE, 40 PSIG
EXHAUST PRESSURE, 20 PSIG
CYLINDER VOLUME

VII. DISCUSSION

General Analysis of Exhaust Braking

The data obtained pertain to a spark-ignition engine of 16.3 cubic inch displacement with a compression ratio of 8.2 to 1. However, in this discussion these data were generalized to include diesel engines and engines in general of larger displacements.

The power producing and braking ability of an engine is approximately in direct proportion to its piston displacement, other factors being constant. Therefore, a 100 cubic inch engine has ten times the retarding ability and also ten times the horsepower output as a 10 cubic inch engine, if such items as cam timing, engine speed, head design are similar in both engines. Consequently, the results of the test may be applied to any size spark-ignition engine directly by using the ratio--braking to output horsepower.

To extend this idea to include diesel engines, an additional factor must be included. Theoretically, diesel engines operate on a constant pressure cycle whereas spark-ignition engines operate on a constant volume cycle. Therefore, the horsepower output of the two types of engines for equivalent cylinder volumes will be different. The ratio of diesel to spark-ignition horsepower output for

similar displacement engines was found, according to several investigators, to range from 0.65 to 0.95. In this discussion a mean value of 0.85 will be used.

As stated previously, a spark-ignition engine with an exhaust pressure of 45 psig, a retarding ability of 75 per cent of the rated horsepower may be realized. But with a diesel engine, a retarding effect of 75/0.85 or approximately 88 per cent is possible. This value is obtained assuming the braking horsepowers are the same for a spark-ignition and diesel engine of similar displacements using an exhaust brake. Recent research has shown this to be true.

An article in the Automobile Engineer discusses the effect compression ratio has on exhaust braking and their findings are as follows:

> It has been found that the compression ratio of an engine does not seriously affect its capacity to absorb power when used in conjunction with an exhaust brake, since the greater proportion of power is absorbed on the exhaust stroke and not on the compression and expansion strokes of the engine. (6, p. 139)

To an operator of a commercial vehicle using an exhaust brake, not only is the extra braking beneficial, but the maintaining of water jacket temperature while descending a long grade is also important.

It may be noted that even at 30 psig exhaust pressure with the test engine, a water temperature of 200°F. was

maintained. An increase in the exhaust pressure increases the water temperature. Since the test engine was cooled by a condenser which required the water in the jacket to boil to obtain extra cooling above that of natural convection and radiation of the engine proper, any exhaust pressure above 37 psig made the water boil. Thus, energy going into the water jacket when using a commercial engine with an exhaust brake is sufficient to keep the engine warm and up to operating temperature. In fact, users of the exhaust brake claim that even at 5°F. outside temperature, their engines do not drop below the thermostat temperature on a long downgrade (12, p. 8) The energy being absorbed by the water jacket during exhaust braking comes mainly from the heat transfer due to the rise in temperature of the exhaust gas resulting from the work of compression.

The air flow through the engine decreases with increased exhaust pressure. This is because of the valve overlap. As the exhaust pressure is increased, more air is transferred from the exhaust line to the intake manifold during the period of time when both valves are open. In fact, this transfer of air creates a pulsation in the intake manifold, as stated in the introduction, which will tend to blow oil out of some oil-bath air cleaners. If the pulsation is damped by sufficient air volume

between the air cleaner and intake valves, this does not occur.

The weight of air flow for 5 psig supercharge pressure is more than the flow for the normally aspirated case because of the increased air density. But, the air flow for both cases decreases uniformly until about 46 or 47 psig exhaust pressure for the unsupercharged case. At this point the exhaust pressure is holding the exhaust valve open during part of the intake stroke, allowing for more air transfer. The pressure at which this will occur depends on the valve area and force of the valve spring when the valve is on its seat.

With 5 psig manifold pressure, this condition would not occur until about 51 or 52 psig exhaust pressure. The higher exhaust pressure would be possible when the engine is supercharged because the additional 5 psig positive pressure tends to hold the valve closed on its seat.

The tests indicated that with 5 psig intake supercharge pressure, more total energy was absorbed by the engine. This is as would be expected as the energy needed to compress the air is added to the motoring energy except for that which is regained by the intake air pushing down on the piston during the intake stroke. On commercial vehicles, pop-off valves are provided between the supercharger and engine to allow for any excess pressure built

up to be eliminated to avoid damage to the induction system seals. With direct driven, constant displacement superchargers (Roots type), the manifold pressure rises quickly when the exhaust pressure restricts air flow.

As stated previously, approximately 88 per cent of the output horsepower of a normally aspirated diesel engine may be realized in exhaust braking using 45 psig exhaust pressure. However, with supercharged commercial diesel engines, the retarding effect from an exhaust brake is actually reduced for two reasons. First, usually more valve overlap is used in supercharged diesels so that the incoming pressurized air may more completely scavenge the cylinder. Consequently, the maximum exhaust pressure is reduced as airflow between exhaust and intake manifolds will take place for a longer time. Maximum exhaust pressures when using a Roots type supercharger and an exhaust brake are limited to 30 to 35 psig. Second, with supercharging, the output horsepower of the engine is increased from 30 to 40 per cent. Consequently, the ratio of retarding to output horsepower is decreased.

For example, consider an unsupercharged diesel engine whose horsepower output is 100 hp. Supercharging this engine would increase its output to approximately 135 hp. But with an exhaust pressure limited to 30 psig, the effectiveness of the exhaust brake, including the energy

required to drive the supercharger is 76 per cent based on the unsupercharged output of a diesel engine. Based on the supercharged rating of the engine, the effectiveness is reduced to 76/1.35 or 56 per cent.

Even at this lower level, the exhaust brake is still advantageous because of decreased engine wear due to higher water jacket temperatures and also less brake wear.

Direct Analysis of Exhaust Braking

The energy that is absorbed during exhaust braking may be divided into two parts. The first part is the energy necessary to motor the engine without any exhaust pressure (friction horsepower). The second part is the energy which is absorbed during the cycle where the piston is doing work on the exhaust gases by expelling them from the cylinder volume at the pressure in the exhaust line.

The friction horsepower for similar displacement engines depends on many factors. Among these are: bore to stroke ratio, compression ratio, piston speed, oil drag on cylinder walls and bearings, and valve train design. A reasonable estimate for engine friction is between 20 and 30 per cent of the rated horsepower of the engine (15, p. 332). The engine tested exhibited friction horsepower equal to 24 and 28 per cent of the horsepower rating

at 1800 and 2400 rpm respectively. This indicates the test engine was typical of engines having larger displacements.

The remaining braking, amounting to approximately 50 per cent of the rated horsepower, may be explained by the use of pressure-time data. This extra 50 per cent is derived during the exhaust stroke of the engine. The braking energy was computed by taking the pressure-time diagram of the exhaust cycle and converting it to pressure-volume. The area under the curve indicates the energy the piston transmits to the gas during the exhaust stroke.

Following is the method of calculating the exhaust braking energy. Referring to Figure 16, the piston displacement of the engine (16.334 in^3) is represented by a distance of 5.5 inches, also each inch on the vertical scale represents 10 psi.

Therefore:

Each square inch of the p-v diagram represents,

$$\frac{10\ (16.334)}{5.5} = 29.7 \text{ lb-in}$$

At 1800 rpm there are 15 exhaust cycles per second.

Also, 1 hp = 6600 lb-in/sec.

$$\text{H.P.} = \frac{29.7\ (15)}{6600} \text{ (Area of diagram)}$$

$$= 0.0675 \text{ (Area of diagram)}$$

The area of the pressure-volume diagram was determined by using a planimeter. Table 1, shown below, is a summary of the information determined.

Table 1. Data based on measured area under P-V diagram.

	For 20 psig exhaust press.	For 40 psig exhaust press.
Area of diagram	11.27 in^2	21.3 in^2
Computed p-v hp.	0.761	1.44
Measured total hp.	1.622	2.390
Subtract friction hp.	0.837	0.837
Exhaust braking horsepower	0.785	1.553
Correlation between computed and measured	97%	93%

A correlation of over 90 per cent of the horsepower computed from the pressure-time diagram compared to the measured horsepower was extremely interesting and gratifying. This high correlation indicates the additional braking effect produced by an exhaust brake is derived during the exhaust stroke. In fact, a correlation higher than indicated was actually produced, but it could not be measured.

The friction horsepower which was subtracted from the measured input horsepower was determined using zero exhaust pressure. A large percentage of friction horsepower is

derived from the friction of the piston rings on the cylinder wall. When there is a gas pressure above the piston, the pressure will also act behind the compression ring, forcing it against the cylinder wall. An increase in this force increases the drag of the ring on the cylinder wall. This is one reason why increasing compression ratio increases friction horsepower, but as stated before, this difference is very slight, especially when using an exhaust brake.

Since the friction horsepower was measured only when there was a positive gas pressure in the cylinder during two of the four strokes of the engine (compression and expansion), then the value found was slightly less than when there was a positive pressure in the cylinder during most of three out of four cycles of the engine (compression, expansion, and exhaust) as found during exhaust braking. Therefore the friction horsepower values used were slightly low. Increasing these values makes the correlation even higher than indicated, approaching 100 per cent.

VIII. CONCLUSIONS

The conclusions derived from the exhaust braking studies conducted by the author and information gathered from other sources are best represented by the following statements.

1. A spark-ignition engine with 45 psig exhaust pressure can realize approximately 75 per cent of its rated horsepower in braking effort.

2. An unsupercharged diesel engine with 45 psig exhaust pressure can absorb approximately 88 per cent of its rated horsepower in exhaust braking.

3. With supercharged diesel engines, using 30 psig exhaust pressure, the effectiveness of the exhaust brake is reduced to approximately 56 per cent.

4. The exhaust braking is derived during the exhaust stroke of the engine. Therefore, the magnitude of the exhaust pressure determines the amount of retardation above that of the normal friction horsepower of the engine.

5. Varying the compression ratio of an engine has very little effect on the exhaust braking ability of the engine.

6. An exhaust brake is beneficial in addition to the braking effect as it will maintain a commercial engine at operating temperature down a long grade, therefore,

reducing the engine wear.

IX. RECOMMENDATIONS FOR FURTHER STUDY

The size of volume between the exhaust braking valve and engine should have an effect on the horsepower that is absorbed by an engine during exhaust braking. One source indicates that there should be a large manifold volume to obtain more uniform exhaust pressure (6, p. 139).

The location of the exhaust valve was not studied in the investigation presented. Exhaust brake valves are at the present being installed at the outlet flange of the exhaust manifold on diesel engines. The volume for exhaust gas storage is therefore only the exhaust manifold of the engine. The question arises whether this is the optimum location of the exhaust brake valve. Studies of the location of this valve would give practical results which could be of importance to exhaust brake manufacturers.

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XI. APPENDIX

DESCRIPTION OF ENGINE

Manufacturer	Waukesha Motor Company, Waukesha, Wis.
Model	C O T 29
Engine Number	626285
Bore and Stroke	2 3/4 X 2 3/4
Displacement	16.334 cu. in.
Compression Ratio	8.2
Piston Material	Aluminum Alloy
Connecting Rod Length	5.5 in.
Piston Rings	4 - 2 compression; 2 oil all cast iron
Cylinder Liner Material	SAE 4140
Bearings - main	Ball bearing
Bearing - rod	Lead-flashed silver
Ignition	Magneto
Sparkplug	Champion J-6
Injection timing	50 $\pm$ 5 degrees ATC on inlet stroke
Intake duration	291°
Exhaust duration	295°
Valve overlap	80°
Valve lift	0.317 in. (see valve timing diagram, next page)
Valve lash	0.000 (Hydraulic lifters)

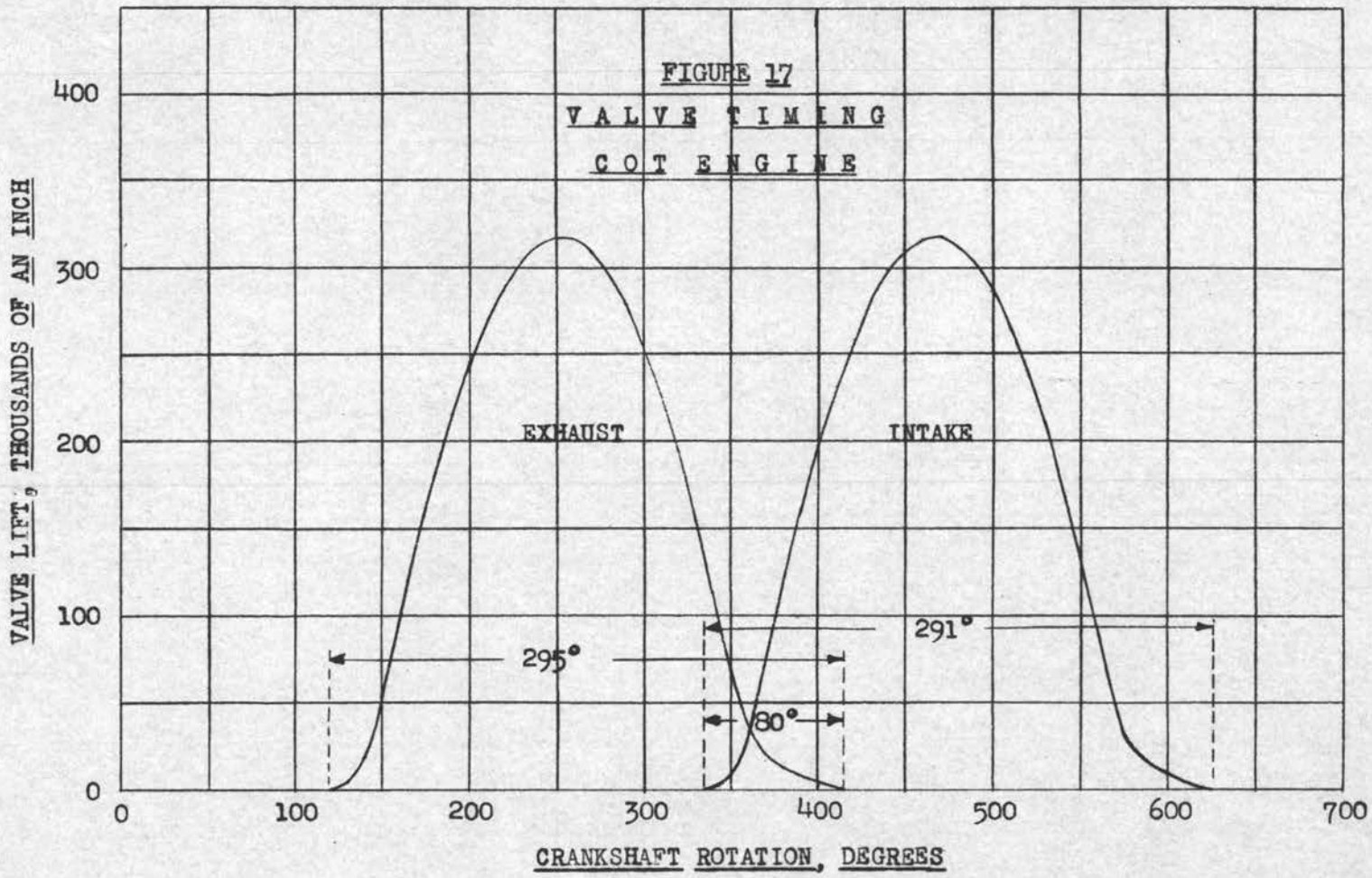
FIGURE 17
VALVE TIMING
COT ENGINE
VALVE LIFT, THOUSANDS OF AN INCH
400
300
200
100
0
EXHAUST
INTAKE
295°
291°
80°
0
100
200
300
400
500
600
700
CRANKSHAFT ROTATION, DEGREES

AIR FLOW CALCULATIONS

The following calculations are based on information from pages 66 and 67 of reference 9.

$M = A_2 KEY \sqrt{2 g_c \, \Delta p \, \rho_1}$

where

M = actual mass flow rate, lb per second

A_2 = cross sectional area of orifice

$K = MC$

M = velocity approach factor $\sqrt{\frac{1}{1-\beta^4}}$

β = orifice diameter/pipe diameter

C = orifice discharge coefficient

$Y = 1 - (0.41 + 0.35\beta^4) \, \Delta p / p_1^k$

$k = 1.4$ for air

$E = 1.000$ for $80^\circ F$

From pg 72 top curve in (6) $C = 0.605$

For a 0.500 in. orifice in a nominal 2 in. pipe (I.D. 2.067)

$\beta = 0.500/2.067 = 0.249$

$\beta^4 = 0.003844$

$$M = \frac{1}{0.996156} = 1.000^+$$

$A_2 = 0.1963 \text{ in}^2$

$g_c = 32.174 \text{ ft/sec}^2$

therefore:

$M_a = 286.2 \, Y \sqrt{\rho_1 \, \Delta p}$ for M_a in lb/hr

For the case upstream pressure = 30 psig, Δp = 10 in H_2O.

$$Y = 1 - (0.41^{+})\left(\frac{0.361}{(1.4)(44.7)}\right)$$

$$= 0.99764$$

$$\rho^1 = \frac{44.7(144)}{53.3(540)} = 0.2237$$

$$M_a = 286.2\ (0.99764)\ \sqrt{0.2237(0.361)}$$

$$= 81.25 \text{ lb/hr}$$

SUMMARY OF AIR FLOW CALCULATIONS

Upstream pressure	Δp=10"	Δp=5"	Δp=1"
90	123.8	87.9	39.35
50	97.7	69.3	31.0
30	81.25	57.5	25.75
20	71.55	50.6	22.7

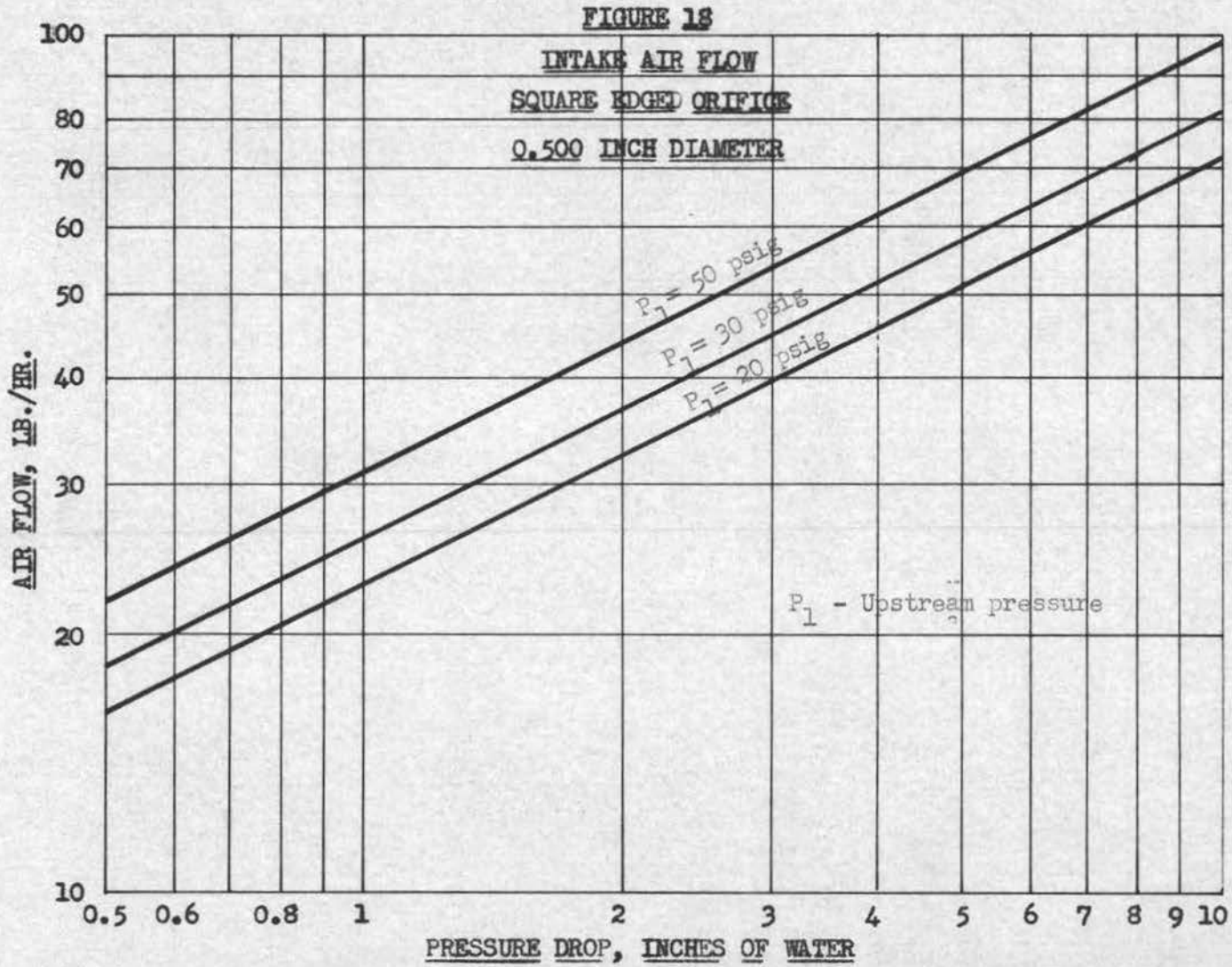
FIGURE 18
INTAKE AIR FLOW
SQUARE EDGED ORIFICE
0.500 INCH DIAMETER
AIR FLOW, LB./HR.
100
80
70
60
50
40
30
20
10
0.5
0.6
0.8
1
2
3
4
5
6
7
8
9
10
PRESSURE DROP, INCHES OF WATER
P_1 = 50 psig
P_1 = 30 psig
P_1 = 20 psig
P_1 - Upstream pressure

SUPERCHARGE HORSEPOWER CALCULATIONS

Assume air entering at 60°F, the change of enthalpy may be calculated as follows:

From air tables, (11, p. 512)

$$p_{r1} = 1.215 \qquad h_1 = 124.3 \text{ Btu/lb.}$$

Using a boost of 5 psig,

$$p_{r2} = p_{r1} \left(\frac{5 + 14.7}{14.7}\right)$$
$$= 1.630$$

Interpolating from air tables

$$h_2 = 135.7$$

$$\text{Change in enthalpy} = h_2 - h_1$$
$$= 135.7 - 124.3$$
$$= 11.4 \text{ Btu/lb.}$$

The adiabatic efficiency of a Roots type supercharger is approximately 66 per cent.

See curve - (7, p. 432)

$$\Delta h = \frac{11.4}{0.66} = 17.3 \text{ Btu/lb.}$$

Horsepower required per lb of air per hour

$$\text{H.P.} = \frac{17.3}{2545} = 0.0068 \left(\text{Air Flow, } \frac{\text{lb}}{\text{hr}}\right)$$

EXHAUST BRAKE TEST DATA

Engine Speed - 1200 rpm Zero Manifold Pressure

Run #	Ex. Press. Psig	Ex. Temp °F.	Scale Load	Upst. Press Psig	Δp in H_2O	H.P.	Air lb/hr
1	0	120	25.0	21	1.10	0.581	24.0
2	10	200	36.5	21.0	0.80	0.848	21.2
3	20	290	47.5	21.0	0.60	1.104	17.7
4	30	360	57.5	23.0	0.40	1.336	14.7
5	40	450	67.0	22.0	0.25	1.556	11.5
6	46.5	500	71.5	23.0	0.01	1.661	2.4
7	20	300+	47.5	21.0	0.50	1.104	16.2

Engine Speed - 1200 rpm Five psig manifold pressure

Run #	Ex. Press. psig	Ex. Temp °F.	Scale Load	Upst. Press psig	Δp in. H_2O	H.P.	Air lb/hr	Total H.P.
1	0	100	24.0	19.5	2.50	0.558	36.0	0.803
2	10	155	35.0	20.0	1.88	0.813	31.3	1.026
3	20	230	45.5	20.0	1.50	1.057	28.0	1.249
4	30	310	55.5	21.0	1.20	1.291	25.2	1.464
5	40	390	65.5	21.5	0.95	1.522	22.5	1.675
6	50	470	71.5	22.0	0.55	1.662	17.1	1.778

EXHAUST BRAKE TEST DATA

Engine Speed - 1800 rpm. Zero Manifold Pressure

Run #	Ex. Press psig	Ex. Temp °F.	Scale Load	Upst. Press psig	Δp	H.P.	Air lb/hr
1	0	260	24.0	19.5	2.5	0.837	36.0
2	10	285	35.0	19.0	1.9	1.222	31.4
3	20	335	46.5	20.0	1.55	1.622	28.5
4	30	435	57.5	20.0	1.15	2.01	24.6
5	40	500	67.5	20.0	0.85	2.32	21.0
1	0	230	24.0	19.5	2.5	0.837	36.0
2	10	280	35.5	19.5	1.9	1.237	31.5
3	20	350	46.5	19.5	1.6	1.622	29.0
4	30	430	57.5	19.5	1.15	2.01	24.6
5	40	500	68.5	21.0	0.82	2.39	21.0
6	46.5	540	78.0	23.0	0.03	2.72	4.0

EXHAUST BRAKE TEST DATA

Engine Speed - 1800 rpm Five psig manifold pressure

Run #	Ex. Press psig	Ex. Temp °F	Scale Load	Upst. Press psig	Δp	H.P.	Air lb/hr	Total H.P.
1	0	165	23.0	19.0	5.3	0.803	51.5	1.153
2	10	170	33.5	19.0	4.6	1.170	48.0	1.496
3	20	265	44.5	19.0	3.8	1.554	44.0	1.854
4	30	350	55.5	19.0	3.15	1.937	40.0	2.209
5	40	405	66.5	20.0	2.6	2.322	36.5	2.570
1	0	160	22.5	19.0	5.5	0.786	52.7	1.144
2	10	205	33.5	19.0	4.6	1.170	48.0	1.496
3	20	270	44.5	19.0	3.9	1.554	44.5	1.856
4	30	340	56.5	19.0	3.15	1.972	40.0	2.244
5	40	420	67.5	19.0	2.6	2.357	36.4	2.604
6	50	520	77.5	21.0	1.50	2.702	28.2	2.894

EXHAUST BRAKE TEST DATA

Engine Speed - 2400 rpm Zero Manifold Pressure

Run #	Ex. Press psig	Ex. Temp °F.	Scale Load	Upst. Press psig	Δp	H.P.	Air lb/hr
1	0	175+	29.5	19.0	5.2	1.37	51.0
2	10	255+	41.5	19.0	4.6	1.93	48.0
3	20	330+	54.5	19.0	3.95	2.53	44.8
4	30	430+	65.0	21.0	3.35	3.02	41.5
5	40	525+	74.5	20.5	2.5	3.46	36.0
6	48.5	610+	90.5	23.0	0.03	4.21	4.0
1	0	200+	30.5	19.0	5.25	1.42	51.5
2	10	270+	41.5	20.0	4.45	1.93	47.7
3	20	360+	54.5	19.5	3.6	2.53	43.0
4	30	450	66.0	20.0	3.15	3.07	40.2
5	40	520+	74.5	19.5	2.75	3.46	37.5
6	45	610	81.5	22.0	1.05	3.79	23.5
7	47.5	620	87.0	23.0	0.05	4.04	5.2
1	0	130	31.5	19.5	5.4	1.465	52.0
2	10	240	42.5	20.0	4.6	1.975	48.5
3	20	340	53.0	20.5	3.75	2.46	44.0
4	30	460	62.5	21.0	3.1	2.90	40.4
5	42.5	570	78.0	22.0	2.2	3.62	34.4
6	45	605	81.5	22.0	0.5	3.79	16.4
7	47	605+	86.0	23.0	0.05	4.00	5.2
1	30	460+	63.5	19.5	3.3	2.95	41.0
2	35	500	69.0	20.0	2.9	3.21	38.5
3	40	540	72.5	20.0	2.75	3.37	37.5

EXHAUST BRAKE TEST DATA

Engine Speed - 2400 rpm Five psig manifold pressure

Run #	Ex. Press psig	Ex. Temp °F.	Scale Load	Upst. Press psig	Δp	H.P.	Air lb/hr	Total H.P.
1	0	140	25.5	30.0	7.7	1.40	71.5	1.89
2	10	200	37.5	30.0	6.6	1.74	66.0	2.19
3	20	280	50.0	31.0	5.7	2.32	61.5	2.74
4	30	355	62.0	31.0	5.25	2.88	59.2	3.28
5	40	460	71.5	29.0	4.9	3.32	56.0	3.70
6	45	500	78.5	21.0	5.15	3.65	52.0	4.00
7	50	560	85.5	21.0	1.9	3.97	31.5	4.18
1	0	150	26.5	29.0	7.45	1.23	70.0	1.70
2	10	210	38.0	29.0	6.55	1.77	65.3	2.21
3	20	285	50.5	30.0	5.7	2.35	61.3	2.77
4	30	360	62.5	29.0	5.15	2.90	58.0	3.29
5	40	430	71.5	29.0	4.9	3.32	56.0	3.66
1	0	80	27.5	31.0	7.5	1.28	71.0	1.76
2	10	160	38.5	31.0	6.7	1.79	67.0	2.24
3	40	470	74.0	21.0	5.6	3.44	54.0	3.81
4	35	440	66.5	18.5	6.6	3.09	56.5	3.47
5	30	370	61.0	20.5	6.4	2.83	58.0	3.22
6	40	450	70.5	21.0	6.1	3.28	56.5	3.66
7	50	575	85.5	20.0	1.8	3.98	30.5	4.19

HORSEPOWER OUTPUT

Exhaust pressure - Zero psig

Inlet air pressure - Zero psig

RPM	Ex Temp °F.	Scale Load	Upst. Press psig	Δp	H.P.	Air lb/hr	BMEP lb/in^2
1800	1100	98.5	21.0	1.9	3.44	32.0	92.7
1800	1130	98.5	19.0	1.91	3.44	31.5	92.7
2400	1160	108.5	19.0	4.2	5.05	46.0	102.0
2400	1200	108.5	19.0	4.1	5.05	45.6	102.0

It was impossible to obtain horsepower readings with five psig manifold pressure as detonation was too severe.

WATER JACKET TEMPERATURES

A combination of various equilibrium conditions.

Exhaust Temp. °F.	Water Temp. °F.
540	212*
500	211
470	207
390	177
310	156
280	137
230	132
200	131

*Maximum temperature possible.