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EXPERIMENTAL DETERMINATION OF THE
AIR FLOW THROUGH THE INDIVIDUAL CYLINDERS OF
AN INTERNAL COMBUSTION ENGINE

by

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A THESIS
SUBMITTED TO THE FACULTY
IN PARTIAL FULFILLMENT OF THE
REQUIREMENTS FOR THE DEGREE OF
MASTER OF SCIENCE
IN
MECHANICAL ENGINEERING

Houston, Texas
May, 1955
ACKNOWLEDGMENT

The author wishes to acknowledge the assistance and encouragement of Dr. Alan J. Chapman, under whom the work was performed. Credit is extended to Mr. Wallace B. Diboll, Jr., for advancing the problem. Appreciation is expressed to The Rice Institute and the Shell Companies Foundation, Incorporated, for providing the equipment and financial assistance which made possible this investigation.
ABSTRACT

The overheating of certain cylinders of multicylinder engines is of vital import to those concerned with large power installations. Such overheating results in a loss of power from the engine. Therefore, engines must be underrated by manufacturers or else customers must install power plants which are somewhat oversized.

Overheating of certain cylinders of an engine has been attributed to a nonuniform distribution of the pulsating fluid (air or fuel-air mixture) flowing through the individual cylinders of the engine. For this reason, manifolds (intake or exhaust) have been adjudged as the probable cause for overheating. To date, the majority of analytical work applicable to manifolds has been limited to intake manifolds or to steady flow conditions. The experimental work related to nonsteady flow has been limited to the total overall flow of a particular engine.

The present work is concerned with the experimental procedure for measuring the pulsating flow through the individual cylinders of an internal combustion engine. Experimentation has been carried out by motoring a three cylinder, two stroke cycle, General Motors Diesel Engine. By motoring the engine with a dynamometer, complications arising from high temperatures and pressures were circumvented. The fluid flow through the individual cylinders was measured with a "Proportional Flow Meter".
<table>
<thead>
<tr>
<th>TABLE OF CONTENTS</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACKNOWLEDGEMENT</td>
</tr>
<tr>
<td>ABSTRACT</td>
</tr>
<tr>
<td>INTRODUCTION</td>
</tr>
<tr>
<td>THE APPARATUS</td>
</tr>
<tr>
<td>(A) Theory of Operation</td>
</tr>
<tr>
<td>(B) Physical Description</td>
</tr>
<tr>
<td>(C) Calibration and Experimental Procedure</td>
</tr>
<tr>
<td>RESULTS AND CONCLUSIONS</td>
</tr>
<tr>
<td>RECOMMENDATIONS</td>
</tr>
<tr>
<td>APPENDIX A - Flow Calculations</td>
</tr>
<tr>
<td>(I) Time Correction to Empty Standard Volume</td>
</tr>
<tr>
<td>(II) Formula For Flow Through An Orifice</td>
</tr>
<tr>
<td>APPENDIX B - Error Of Flow Measurement Due To Pulsations</td>
</tr>
<tr>
<td>APPENDIX C - Summary Of Experimental Data</td>
</tr>
<tr>
<td>(I) Flow Tests</td>
</tr>
<tr>
<td>(II) Exhaust Gas Temperature Tests</td>
</tr>
<tr>
<td>APPENDIX D - Calibration Curves</td>
</tr>
<tr>
<td>ILLUSTRATIONS</td>
</tr>
<tr>
<td>Figure 1 - Schematic Flow Diagram</td>
</tr>
<tr>
<td>Figure 2 - Cross-Section Of Special Orifice</td>
</tr>
<tr>
<td>Figure 3 - Special Orifice Plate and Two Restriction Orifice Nipples</td>
</tr>
<tr>
<td>Figure 4 - The Apparatus</td>
</tr>
<tr>
<td>BIBLIOGRAPHY</td>
</tr>
</tbody>
</table>
INTRODUCTION

The overheating of certain cylinders of large multiple-cylinder internal combustion engines is of considerable import to manufacturers and power plant operators as such a condition means the loss of potentially available power. The cause of overheating has been attributed to a nonuniform distribution of air to the individual cylinders. Even though this problem probably existed with the first successful internal combustion engine, no serious work has been devoted to measuring the air flow to the individual cylinders of engines using manifolds which serve more than one cylinder. Instead, a great amount of time and energy have been applied to the designing of manifolds and testing the horsepower output of the engine, then redesigning and further testing. This has been a trial and error procedure until recently when attention has been turned to so called "tuned manifolds" (7)*.

The measurement of air flow through the individual cylinders of an internal combustion engine is concerned with a dynamic system in which large pressure variations occur over relatively short periods of time. The analytical treatment of a pulsating flow is considerably more difficult than the treatment of the uniform flow system (3).

The measurement of a flow by employing an orifice or a nozzle implies the measurement of the pressure differential across the device and the correlation of the pressure differential with the velocity of the fluid moving through the device.

The pressure differential across the orifice or nozzle is propor-

* The numbers in parenthesis refer to the bibliography which concludes this work.
tional to the square of the average velocity for steady state flow of an incompressible fluid. Air at low pressure and velocity behaves very nearly like an incompressible fluid. Consequently, when the proportionality constant is determined, the flow rate for various values of the pressure differential can be calculated. The application of standard orifices and nozzles to the measurement of pulsating flow is most unsatisfactory due partly to the inability, because of inertia, of the indicating and recording mechanisms to follow the rapid pressure variations (2). Of more importance is the fact that the average pressure differential, which a recorder tends to indicate (1), is not proportional to the square of the average velocity when the velocity varies rapidly. The mathematical explanation of the error incurred in assuming that the average pressure differential is proportional to the square of the velocity is presented in Appendix B. Also, the magnitude of the error for a hypothetical case is presented. Corrections to an actual calculated flow are difficult to apply as the magnitude of the error cannot be determined (4).

Professor H. P. Bailey of Rensselaer Polytechnic Institute designed a "Proportional Flow Meter" which was installed in the upstream side of a double acting reciprocating air compressor. The results of tests on the pulsating flow of the air compressor showed that the error of the proportional flow meter was less than one-sixth the error obtained with a standard orifice and manometer and for certain ranges less than one-fortieth.

Mr. P. S. Vaughan of the Diesel Engine Division of the American Locomotive Company suggested that such a "Proportional Flow Meter" might be used to determine the air flow through the individual cylinders of an
internal combustion engine. This work is concerned with the application of the meter, slightly modified, to a three cylinder, two stroke cycle, air blower equipped General Motors Diesel Engine.
(A) **Theory of Operation**

The difficulties encountered when measuring a pulsating flow with an orifice have been presented in the "Introduction" and are considered in more detail in "Appendix B". These difficulties are minimized when the pressure differential existing across the orifice is employed indirectly rather than trying to record it and calculate the flow rate directly. A schematic diagram of the proportional flow meter is shown in Figure 1.

The pressure differential, which exists across the special orifice employed for indicating the flow rate of the pulsating fluid, is applied across a very small restriction orifice. This pressure differential is isolated in such a manner that none of the pulsating air is permitted to pass through the restriction orifice. The only air flowing through the restriction orifice comes from a vessel of known volume. As the air passes from the vessel through the restriction orifice, a displacing liquid is added to the vessel at a rate such that the pressure within the vessel is maintained constant. The liquid level in the vessel is an indication of the amount of air (at a particular pressure) remaining in the vessel.

The time required to empty the air from the vessel is related to the flow rate of the pulsating air. Knowing the volume of the vessel and the size of the restriction orifice and by measuring the time required to empty the air from the vessel, it is possible to calculate the pressure differential existing across the restriction orifice. This pressure differential is equal to the average pressure differential existing across the orifice placed in the pulsating flow. Thus, it would be possible to calculate approximately the pulsating flow rate.

A more accurate method of determining the pulsating flow rate was
used in this work. Instead of calculating the flow rate according to the average pressure differential, a calibration curve was obtained relating directly the time required to empty the air from the standard vessel to the pulsating flow rate. The work was performed with a three cylinder engine. The special orifice was in fact three orifices, one for each cylinder. Hence it was necessary to obtain a calibration curve for each cylinder. Referring the time required to empty the vessel to the proper calibration curve determines the flow rate through the individual cylinder of the engine. The procedure for obtaining the calibration curves is explained under "Calibration and Experimental Procedure".
(B) **Physical Description**

The physical description of the components of the apparatus will assume more meaning if the relationships of the components are understood. These relationships will be made clear by referring again to the schematic diagram in Figure 1.

The special orifice $O_1$, made from a three-eighths inch thick brass plate, was placed between the exhaust manifold and the engine block. Figure 2 represents a cross section of this orifice which indicates the upstream and downstream pressure taps. This particular orifice, an adaptation from Reference 1, page 37, by its thinness, minimized the dimensional change in the manifold system. Though pictured as a single orifice, it is in fact three orifices (see Figure 3) made from a single plate. By incorporating the three orifices in a single plate, alignment with the three cylinders of the engine was easily accomplished. Brass was chosen for the material of the orifice plate due to its good machinability and resistance to oxidation. Short lengths of copper tubing were silver soldered to the plate and were connected to the upstream and downstream pressure taps by drilled passages. These lengths of tubing afforded a simple means for attaching rubber tubing employed for transmitting the pressure differential across the special orifice $O_1$ to the rest of the system. Rubber plugs were used for capping off those orifices not in use. The location of the pressure taps and the cone angle of the orifice were selected quite arbitrarily.

The standard volume (see Figure 1), previously referred to as vessel, was a rectangular parallelepiped with external dimensions of 2.5 x 5.5 x 10 inches. It was constructed from one-quarter inch stainless steel plate,
welded along all edges. This material was selected on the basis of its chemical stability, thereby permitting a wide choice of displacing fluids. The thickness of the material provided the required rigidity for volumetric stability. A glass tube connected to the vessel at the top and bottom served to indicate the level of the displacing fluid. The glass tube was marked at an interval corresponding to 1000 cubic centimeters of volume in the tank. One of the fundamental qualities involved in this proportional flow meter method was the measurement of the time required for displacing this 1000 cubic centimeters of air. The standard volume plus the restriction orifice, manometers, valves, and lines could be considered as the proportional flow meter though it might be well to consider this "meter" as a technique or method of measurement rather than a particular physical device.

The restriction orifice (see Figures 1 and 3) was made by soldering a piece of 0.010 inch brass shim stock to the end of a one-quarter inch brass nipple and then drilling a hole through the shim stock. Several sizes of orifices were made so that the brass nipples provided an expedient method for changing them. The final choice of the orifice size was determined experimentally and was selected such that the time required to empty the standard volume through it was at least four minutes. The size of the orifice (0.0104 inches in diameter) was obtained by grinding a number 80 drill (0.0135 inches in diameter) and drilling at a high speed.

A muffler for damping the pulsations in the air was employed for calibration. The muffler was a drum with the entrance pipe coinciding with the axis of the drum and extending into the drum a distance of 30 inches. This pipe was drilled with many holes permitting the air to expand radially from the pipe. The outflow from the drum was through a nominal two
inch pipe whose direction was perpendicular to the axis of the drum. This outflow pipe led directly to a Daniel Simplex Orifice which was employed for measuring the flow after being damped of pulses.

The transmission of pressures throughout the system was via one-quarter inch copper tubing and/or rubber hose. The air pressures and flows were controlled by standard glass stopcocks.
(C) Calibration and Experimental Procedure

The calibration of the apparatus and the experimental procedure for employing it will be considered together as they are very closely related. An understanding of the calibration procedure will in itself unfold the experimental approach. The information obtained from calibration is the flow rate of air through the individual cylinders and the associated corrected time for emptying the standard volume of air. For this work, the calibration information has been plotted forming the Calibration Curves of Appendix D.

By referring to Figure 1 for identifying the location and relation of the various components in the system, the function of the apparatus will be more easily understood. The geometry of the exhaust system was such that separate calibration of the three special orifices $O_1$ seemed justified, a decision which proved correct. For calibration of a particular cylinder it was necessary to remove the rocker mechanisms operating the exhaust valves and fuel injectors of the two remaining cylinders so that the exhaust air from only the one cylinder passed through the exhaust manifold. The air inlet to the cylinders of this engine was controlled by ports. The fuel injectors were removed from the two cylinders not being calibrated. This prevented the compression of air in the two cylinders and distributed the air from the engine blower so that all of it did not pass through the cylinder being calibrated.

Connections from the balancing manometer $H_4$ and the restriction orifice $O_3$ were made with the upstream and downstream pressure taps, respectively, of the special orifice $O_1$. The upstream and downstream pressure taps were on opposite sides of the orifice plate as illustrated in Figure
2. The engine was then brought to a particular speed by motoring with the dynamometer. The air flow through the single cylinder then passed through the manifold system producing a pulsating pressure differential across special orifice $O_1$ and a constant pressure differential across the standard calibration orifice $O_2$.

By controlling the amount of displacing fluid flowing into the standard volume, the pressure differential across the balancing manometer could be maintained at zero. This indicated that the pressure differential existing across the restriction orifice was equal to the pressure differential across the special orifice. This is true so long as the friction drop and velocity head of the flow from the standard volume are negligible. The level of the indicating fluid of the balancing manometer remained constant even though the pressure differential across the special orifice was pulsating. This was presumably due to the damping effect of the leads from the special orifice and the inertia of the indicating fluid. The time required to empty the 1000 cubic centimeters of air from the standard volume could then be associated with the flow rate through that particular cylinder. After making the proper temperature and pressure corrections to the time interval (see Appendix A, Section I), a quantity "corrected time" was obtained which, with the flow rate, determined a point on the calibration curve of that particular cylinder. The flow rate through that cylinder was obtained by calculation using the pressure differential across the standard calibration orifice.

By choosing discriminate engine speeds, the flow rate through the particular cylinder under investigation and the corrected time to empty the 1000 cubic centimeters of air from the standard volume for each speed were obtained. It was necessary to calibrate each cylinder singly as the
standard calibration orifice $O_2$ indicated a reading dependent upon the total flow rate through the manifold system. The calibration curve for each of the three cylinders was obtained by plotting the air flow rate through the cylinder versus the corrected time for emptying the standard volume.

The engine was returned to its original condition by replacing the fuel injectors and the rocker mechanisms. In this condition, the exhausting air from all the cylinders passed through the manifold system so that with the reading of the pressure differential across the standard calibration orifice $O_2$, the total flow through the engine could be calculated.

To determine the flow rate of air through the individual cylinders with the engine being motored at a particular speed, the time intervals required to empty the standard volume through each of the three special orifices $O_1$ were measured. These time intervals were corrected and then referred to the respective calibration curves. From the calibration curves could be read the flow rate through each cylinder. It required as much as forty-five minutes (at low speeds) to obtain the three "time" readings. Over that period of time the total engine speed, hence flow rate, varied a little. Comparison of the individual flow rates required correcting them with respect to the total flow rate through the engine. These corrections were based on the assumption that the proportionality of the flow rates through the individual cylinders remained constant over the small variation of the total flow rate through the engine. The original and corrected data as well as the method for making the corrections appear in Appendix C - Flow Tests. The general arrangement of the equipment is illustrated in Figure 4.
RESULTS AND CONCLUSIONS

The Summary of Experimental Data presented in Appendix C is a tabulation of the important data observed with all of the cylinders exhausting into the manifold system. The calibration data has been excluded from this work as little or no importance can be associated with such quantities as the engine cooling water temperature, barometric pressure, manometer readings, etcetera. Instead, a summary of the calibration data is presented graphically in the form of the Calibration Curves, Appendix D.

The calibration data observed at an engine speed of 440 rpm were inconsistent with the other observed data. The three calibration points at this engine speed have been plotted and are circled for identification on the Calibration Curves, Appendix D. They have been considered invalid due to the existence of a resonance phenomena at this speed. The physical system was such that the distances from the exhaust valves to the bottom of the muffler were 145 inches and 149 inches respectively for the middle and end cylinders. With a fluid temperature of 85°F., the sound propagation velocity would be approximately 1146 feet per second. The three cylinders with an engine speed of 440 rpm would give $3 \times \frac{440}{60} = 22$ pulses per second to the system. The resonant length for the first fundamental frequency equals the propagation velocity divided by four times the pulse rate $= \frac{(1146 \times 12)}{(4 \times 22)} = 156$ inches. The engine speed for resonance in the 149 inch length would equal 460 rpm. Some variation between the actual engine speed and the speed as indicated with an electric tachometer is to be expected. Also, the actual air temperature was probably a little higher than was indicated by the thermometer, as they were separated by the material of the thermometer well. The errors possible
in reading the engine speed and the temperature of the air flowing through
the engine could account for the deviation between the calculated resonant
speed and the observed engine speed. The probability of resonance at
460 rpm was deemed sufficient for disregarding these three calibration
points when drawing the final calibration curves.

The effect of resonance can not be underestimated as it was exempli-
fied again at an engine speed of 610 rpm. During a particular testing
period at an engine speed of 610 rpm, the fluid (air) temperature was
100°F. The rubber tubing leads between the special orifice on the engine
and the indicating apparatus were ten feet in length. With this parti-
cular length of leads, engine speed, and fluid temperature, the resonant
frequency of the system would correspond to an engine speed of approx-
imately 600 rpm. The result was (see Flow Tests - Appendix C) that the
sum of the flows through the individual cylinders, as indicated by the
proportional flow meter, was 9.8 per cent greater than the total flow
through the engine as indicated by the standard calibration orifice down-
stream of the muffler. This error was three times greater than the error
experienced at other speeds. The changing of the natural frequency of
the system, by shortening the rubber tubing leads, corrected this resonant
condition.

The data obtained by using this proportional flow meter indicate a
good fluid distribution through the individual cylinders at medium engine
speeds and poor distribution (30 per cent difference at 600 rpm) at low
speeds. A similar air distribution in the manifold system of an internal
combustion engine employing a conventional carburetor could be at least
partially responsible for the poor idling characteristics so common with
them. As further proof of the uneven flow distribution of this engine,
some tests on the exhaust gas temperatures (the engine driving, not being motored with the dynamometer) showed that cylinders Nos. 1 and 2 were consistently running hotter at all speeds and loads than cylinder No. 3. The experimental data indicate that cylinder No. 3 had the greatest air flow rate at all speeds while being motored. Thus, it could be concluded that the better scavenging reduced the cylinder head temperature. The temperature data were very erratic implying, perhaps, a great dependency of the indication upon the location of the thermocouples in the exhaust stream. The thermocouples were switched several times from one exhaust stream to another (see data of Exhaust Gas Temperature Tests, Appendix C) and precise control of the location of these thermocouples was very difficult.

The concern of this work is that of applying some type meter for measuring pulsating flow. Perhaps the proportional flow meter represents the solution to the pulsating flow problem through the flow tests. The temperature tests presented in this work can be considered as merely indicating trends.
Even though considerable thought and experimentation were applied toward the design of the proportional flow meter, working with it has suggested several modifications. It is felt that by making these modifications the accuracy of the measured data would be improved, thereby reducing the difference between the sum of the air flow rates through the individual cylinders measured with the meter and the total air flow rate as indicated by an orifice downstream of the muffler.

For more precise indication of the pressure difference across the balancing manometer $H_4$, a draft gage indicating one-hundredths of an inch of water pressure could be installed in parallel with the standard manometer. As the pressure difference across the balancing manometer was as large as fifty inches of water head, an isolating needle valve on the proposed draft gage would be required to prevent expulsion of the indicating fluid. By placing this isolating valve in the line containing the indicating fluid of the draft gage, the valve could be used as a restrictor for reducing pulsations in the system as well as an isolating valve. After the pressure difference across the balancing manometer $H_4$ was brought approximately to zero, the very accurate draft gage could be brought into use by opening the isolating valve.

In conjunction with the more precise pressure indicator, a quick and more accurate control over the flow of the displacing fluid into the standard volume could be obtained with a needle valve and a plug valve in parallel in the line. The plug valve would permit rapid emptying of the displacing fluid from the standard volume and also afford a rough control of the flow into the standard volume. The needle valve would be used for
the fine adjustment of the flow rate. Some difficulty was encountered with over controlling as only a plug valve was incorporated in the apparatus.

Increasing the size of the restriction orifice would reduce the time required to empty the standard volume. This would reduce the time over which the engine speed would need be kept constant. The accuracy of the work would not be reduced as the time measurements are the most accurate data. Particles of dust would have less influence on a larger restriction orifice so that increasing the size of the orifice would possibly increase the overall accuracy of the measured flow rates.

If a choice is available, the special orifice which supplies the pressure differential to the meter should be installed where the fluid stream has the pulsations of smallest magnitude. In general, this will be on the intake side of compressors and engines. By choosing the location of the special orifice where the pulsations are a minimum, the greatest accuracy of measurements will be obtained.

The shape of the standard volume could be improved by having a smaller cross-section near the extremities of the vessel where the graduations appear. In this manner an error in reading the level of the displacing fluid at the beginning or end of a run would be minimized as the volume in the "necked-down" portions would be only a very small part of the standard volume.

With these several suggested modifications, the accuracy of the proportional flow meter could be improved. As such it could become a very useful device to those concerned with engines and compressors.
APPENDIX A - Flow Calculations

(1) Time Correction To Empty Standard Volume

The velocity of the air flowing from the standard volume through the restriction orifice O₃ (see Figure 1) is very small as the orifice opening (0.0104 inches in diameter) and the pressure differential across it are small. Hence, the velocity head is small. If the velocity head is neglected and the reading of the balancing manometer H₄ is maintained at zero by regulating the flow of the displacing fluid, then the pressure applied across the restriction orifice will be equal to the pressure existing across the special orifice O₁. Therefore, the flow rate of air from the standard volume (through the restriction orifice O₃) will be proportional to the flow rate through the special orifice O₁. The standard volume is the physical size of the vessel which contains air at room temperature. This size is considered fixed as the usual room temperature fluctuations are small. The relation between the flow rates through the orifices can be expressed as

\[ q = B Q \]

where: \( q \) = the flow rate of air from the standard volume. This can be considered as a mass flow rate (lb mole/sec.) or a volume flow rate (cu. ft./sec.), the latter of which is corrected to the conditions of \( T_s \) and \( P_s \). For the following, let us consider \( q \) as a volume rate of flow.

\[ T_s = \text{the standard base temperature}^\#. \]

\[ P_s = \text{the standard base pressure}^\#. \]

^\# For this work, \( T_s = 520^\circ \text{R} \) and \( P_s = 14.7 \text{ psia} \). These particular values were chosen as they were the basis for readily available corrections for expansion factors.
Q = the flow rate of air through the special orifice \( O_1 \) (the units to be the same as those for \( q \)).

\( B \) = the proportionality constant.

If the temperature and pressure of the air in the standard volume is at standard base conditions, then the time required to empty the standard volume can be expressed in terms of the flow rate \( Q \), thusly:

\[
t_o = \frac{V}{BQ}
\]

where:

\( t_o \) = the time required to empty the standard volume.

\( V \) = the volume of air in the standard volume at the standard base temperature and pressure \( T_s \) and \( P_s \). For this investigation, the standard volume was 1000 cubic centimeters.

The actual temperature and pressure of the air in the standard volume are \( T_3 \) and \( P_3 \) respectively.

If

\[
V_\alpha = \text{the volume of the air which fills the standard volume at the conditions } P_3 \text{ and } T_3, \text{ after being corrected to the standard conditions } P_s \text{ and } T_s,
\]

and the perfect gas relations are assumed valid (the system under consideration is of low pressure), then

\[
V_\alpha = \frac{V}{T_3 P_3} \frac{T_s P_s}{T_3 P_s}
\]

The standard volume of air is emptied from the vessel by being displaced with the displacing fluid. The time required to empty the standard volume of air (at the conditions \( T_3 \) and \( P_3 \)) is given by

\[
t_1 = \frac{V_\alpha}{BQ} = \frac{V}{BQ} \frac{T_s P_3}{T_3 P_s}
\]

where:

\( t_1 \) = the time required to displace the standard volume of air (for this work, \( t_1 \) has the units of seconds).
Note that $t_1$ is a measured quantity which is determined experimentally. For comparison calculations, it is necessary to correct $t_1$ to a value corresponding to the time that would be required to empty the standard volume were the air contained therein at the standard conditions. Plotting this corrected time $t^*$ versus the actual flow $Q$ will determine a point on the calibration curve. For calibration, it is necessary that the flow from all of the cylinders except one be bypassed from the exhaust manifold. The flow from this single cylinder then passes through the exhaust manifold, the muffler, and then through the calibration orifice $O_2$ from whence it is possible to calculate the flow $Q$ through this cylinder. Making the pressure and temperature corrections to the standard base pressure and temperature conditions, we can write a relation involving the corrected time $t^*$ thusly:

$$t^* = \frac{V_a}{B Q} \cdot \frac{T_s}{P_3} = \frac{V}{B Q} \cdot \frac{T_3}{P_3} \cdot \frac{T_1}{P_3}.$$

If $k = \frac{T_3}{T_s} \cdot \frac{P_s}{P_3}$, then the measured time that is required to empty the standard volume of air during a test multiplied by "$k" will result in the corrected time.

With all of the cylinders exhausting through the common manifold system (manifold, muffler, and piping run containing the calibration orifice), the calibration orifice will be indicating the combined flow through the engine. To obtain the flow through an individual cylinder, the meter is attached to the special orifice $O_1$ of that cylinder, and the time required to empty the standard volume of air is measured. This time interval is corrected as indicated above. Referring this corrected time interval to the calibration curve of the particular cylinder will
determine a value on the curve. This value will be the flow rate corrected to the base temperature and pressure, through the individual cylinder. By attaching the meter successively to the special orifice of the several cylinders, the individual flow rates can be determined. The sum of the individual flow rates can be checked against the flow rate as indicated by the calibration orifice $O_2$. 
## Formula For Flow Through An Orifice

The downstream flow rate was measured with a Daniel Simplex Orifice (O2 - Figure 1) using the "Hydraulic" equation with correction data. The correction factors were those published by the American Gas Association in "Gas Measurement", Committee Report No. 2. The basic formula is:

\[ Q = C' \sqrt{H_1} \sqrt{H_2} \frac{T_B}{T_f} (Y) \]

where:

- \( C' \) = the orifice flow constant - a function of Reynold's number, pressure base factor, temperature base factor, specific gravity factor, supercompressibility factor, etcetera.
- \( H_1 \) = the pressure differential across the orifice plate.
- \( H_2 \) = the absolute static pressure of the fluid either upstream or downstream, the choice of which determines the method for obtaining the expansion factor \( Y \).
- \( T_B \) = the absolute base temperature.
- \( T_f \) = the absolute fluid temperature.
- \( Y \) = the expansion factor - a function of the velocity and pressure change through the orifice.
An impressive number of articles have been published concerning pulsating flow and the errors resulting when such flows are measured with inferential meters. A simplified analysis will indicate why errors exist, but to date, a satisfactory approach for determining the magnitude of the errors is unavailable (6). When using an orifice in a rapidly pulsating flow, a manometer will tend to indicate the average pressure difference existing between the upstream and downstream pressure taps.

Consider a flow situation where \( v \) is the average fluid velocity; \( v_i \), the instantaneous velocity; \( \delta \), the deviation of the instantaneous velocity from the average (either positive or negative); and \( t \), the variable time. Symbolically

\[
\Delta p = \Delta p_0 \left( 1 + \delta \right)
\]

and by definition of average

\[
\int_0^t \delta \, dt = 0.
\]

The manometer will indicate a pressure difference which is proportional to the square of the velocity which will be equal to

\[
(k/t) \int_0^t v_i^2 \, dt = (k/t) \int_0^t v^2(1 + \delta)^2 \, dt = \left[ (v^2 k/t) \int_0^t \delta \, dt \right] 0 \int_0^t \delta \, dt + (v^2 k/t) \int_0^t \delta^2 \, dt.
\]

The first term is the pressure that would be indicated by a manometer if there were no pulsations in the velocity and in fact is the pressure difference which the manometer should read for computing the average flow velocity. The second term will approach zero (by definition of average velocity) as the range of \( t \) increases to include an infinite number of pulses. The third term is the difference between what the manometer should read and the value it will actually indicate.
Suppose that the flow condition exists in which the velocity pulses vary sinusoidally with time and with a maximum deviation from the average velocity equal to \( b \). The velocity at any time is therefore given by the equation
\[
v = v_0 \left[ 1 + \left( \frac{b}{v_0} \right) \sin t \right].
\]
The pressure difference indicated by the manometer will be \( P_1 \), where
\[
P_1 = \frac{6}{2\pi} \int_0^{2\pi} v_0^2 \left[ 1 + \left( \frac{b}{v_0} \right) \sin t \right]^2 dt
\]
\[
= 6v_0^2 \left[ 1 + \left( \frac{b^2}{2v_0^2} \right) \right],
\]
"C" being a proportionality constant. The pressure difference, which should be indicated by the manometer for the same average flow velocity \( v \) but without pulsations, is \( P \), where
\[
P = (C/t) \int_0^t v_0^2 dt = Cv_0^2.
\]
For example, if the amplitude of the pulsations were 25\% of the average velocity, the manometer would indicate a value of pressure that is in excess of the pressure due to the average velocity head. The pressure ratio will serve to indicate the error. Thus
\[
\frac{P_1}{P} = \frac{Cv_0^2 \left[ 1 + \left( \frac{0.25v_0}{2v_0^2} \right)^2 \right]}{Cv_0^2} \approx 1.03.
\]
Therefore, the computed flow rate would be greater than the actual flow rate as the indicated pressure will read three per cent greater than it should. In existing instances it is not uncommon for the flow error to be twenty to thirty per cent and will range up to one-hundred per cent in severely pulsating flows. Unfortunately, the pulsation wave shape and amplitude and its variation with time can not be predicted (5). For the time of being it is impossible to make analytical corrections for pulsating flow conditions and expect very accurate results.
### APPENDIX C - Summary of Experimental Data

#### (I) Flow Tests

<table>
<thead>
<tr>
<th>Run</th>
<th>RPM</th>
<th>Cyl. No.</th>
<th>Q</th>
<th>(kt_1)</th>
<th>(q_1)</th>
<th>(Q_c)</th>
<th>Error in % based on (Q_c)</th>
</tr>
</thead>
<tbody>
<tr>
<td>29</td>
<td>800*</td>
<td>1</td>
<td>93.2</td>
<td>603</td>
<td>30.3</td>
<td>93.2</td>
<td>30.3</td>
</tr>
<tr>
<td>30</td>
<td>800*</td>
<td>2</td>
<td>93.5</td>
<td>671</td>
<td>30.9</td>
<td>93.2</td>
<td>31.0</td>
</tr>
<tr>
<td>31</td>
<td>800*</td>
<td>3</td>
<td>92.9</td>
<td>633</td>
<td>33.0</td>
<td>93.2</td>
<td>33.1 94.4</td>
</tr>
<tr>
<td>22</td>
<td>790</td>
<td>1</td>
<td>92.5</td>
<td>570</td>
<td>31.6</td>
<td>92.5</td>
<td>31.6</td>
</tr>
<tr>
<td>23</td>
<td>790</td>
<td>2</td>
<td>92.5</td>
<td>646</td>
<td>31.3</td>
<td>92.5</td>
<td>31.3</td>
</tr>
<tr>
<td>24</td>
<td>790</td>
<td>3</td>
<td>92.3</td>
<td>636</td>
<td>32.8</td>
<td>92.5</td>
<td>32.9 95.8</td>
</tr>
<tr>
<td>13</td>
<td>750</td>
<td>1</td>
<td>90.6</td>
<td>569</td>
<td>31.6</td>
<td>90.0</td>
<td>31.4</td>
</tr>
<tr>
<td>14</td>
<td>750</td>
<td>2</td>
<td>90.0</td>
<td>631</td>
<td>31.5</td>
<td>90.0</td>
<td>31.5</td>
</tr>
<tr>
<td>27</td>
<td>610</td>
<td>1</td>
<td>71.8</td>
<td>794</td>
<td>23.3</td>
<td>71.8</td>
<td>23.3</td>
</tr>
<tr>
<td>26</td>
<td>610</td>
<td>2</td>
<td>71.8</td>
<td>990</td>
<td>26.2</td>
<td>71.8</td>
<td>26.2</td>
</tr>
<tr>
<td>25</td>
<td>610</td>
<td>3</td>
<td>71.8</td>
<td>816</td>
<td>29.3</td>
<td>71.8</td>
<td>29.3 78.8</td>
</tr>
<tr>
<td>12</td>
<td>610</td>
<td>1</td>
<td>71.8</td>
<td>772</td>
<td>24.1</td>
<td>71.8</td>
<td>24.1</td>
</tr>
<tr>
<td>11</td>
<td>610</td>
<td>2</td>
<td>72.2</td>
<td>819</td>
<td>28.7</td>
<td>71.8</td>
<td>28.6</td>
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<tr>
<td>21</td>
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<td>1</td>
<td>67.1</td>
<td>946</td>
<td>17.9</td>
<td>67.8</td>
<td>18.1</td>
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<tr>
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<td>600*</td>
<td>2</td>
<td>67.8</td>
<td>1074</td>
<td>24.9</td>
<td>67.8</td>
<td>24.9</td>
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<tr>
<td>19</td>
<td>600*</td>
<td>3</td>
<td>68.0</td>
<td>878</td>
<td>27.0</td>
<td>67.8</td>
<td>26.9 69.9</td>
</tr>
</tbody>
</table>

\(Q\) is the total flow rate through the engine.

\(Q_c\) is the total flow rate through the engine used as a base for the linear correction applied to \(q_1\).

\(kt_1\) is the corrected time required to empty the standard volume.

\(q_1\) is the flow rate through the individual cylinder as indicated by the calibration curves.

\(q_{1c}\) is the flow rate through the individual cylinder corrected the same percentage as \(Q\) ( \(q_{1c} = q_1(Q_c/Q)\) ).

Error = \(100 \left( \sum q_{1c} - Q_c \right)/Q_c\).

* These runs were made after shortening the leads from the engine to the meter.
(II) Exhaust Gas Temperature Tests

<table>
<thead>
<tr>
<th>Engine Speed RPM</th>
<th>Brake H.P.</th>
<th>Exhaust Gas Temperature in degrees Fahrenheit</th>
<th>Thermocouple Order</th>
</tr>
</thead>
<tbody>
<tr>
<td>1000</td>
<td>16.3</td>
<td>385   416   364</td>
<td>1-2-3</td>
</tr>
<tr>
<td>1000</td>
<td>16.3</td>
<td>406   414   346</td>
<td>3-2-1</td>
</tr>
<tr>
<td>1000</td>
<td>13.0</td>
<td>355   382   357</td>
<td>1-2-3</td>
</tr>
<tr>
<td>1000</td>
<td>13.0</td>
<td>424   445   396</td>
<td>3-2-1</td>
</tr>
<tr>
<td>1000*</td>
<td>13.0</td>
<td>367   405   311</td>
<td>3-2-1</td>
</tr>
<tr>
<td>1000</td>
<td>1.3</td>
<td>299   275   210</td>
<td>1-2-3</td>
</tr>
<tr>
<td>1000</td>
<td>0.7</td>
<td>294   298   208</td>
<td>3-2-1</td>
</tr>
<tr>
<td>780**</td>
<td>0.78</td>
<td>261   252   173</td>
<td>3-2-1</td>
</tr>
<tr>
<td>500</td>
<td>0.3</td>
<td>241   238   185</td>
<td>3-2-1</td>
</tr>
</tbody>
</table>

* This is the average of two separate tests with the thermocouples moved off center a distance equal to approximately 25% of the manifold diameter.

** Average values of three separate tests.
APPENDIX D - CALIBRATION CURVES

LEGEND:

CYLINDER 1 - ○
DO 2 - ●
DO 3 - ●

FLOW RATE - C.F.M.

CORRECTED TIME - SECONDS

440 R.P.M.
SPECIAL ORIFICE
(CROSS SECTION)

PRESSURE TAP

FLOW

60°

PRESSURE TAP

FIGURE 2
BIBLIOGRAPHY


