The University assumes no responsibility for the accuracy or correctness of any of the statements or opinions expressed in this thesis.
ACKNOWLEDGEMENTS

The author wishes to express his sincere appreciation to Professor Helmut W. Engelman, of the Department of Mechanical Engineering at the Ohio State University, for his valuable and patient guidance and advice throughout this study; to Mr. Dwight Moseley and the Mechanical Engineering staff for their cooperation and aid during the experimentation; to fellow graduate student Richard Springman for his interest; and finally to the author's wife, Gwen, for her interest and encouragement during this study.
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>STATEMENT OF THE PROBLEM.</td>
<td>v</td>
</tr>
<tr>
<td>CHAPTER I</td>
<td></td>
</tr>
<tr>
<td>INTRODUCTION AND HISTORY.</td>
<td>1</td>
</tr>
<tr>
<td>THEORY INVESTIGATED.</td>
<td>5</td>
</tr>
<tr>
<td>CHAPTER II</td>
<td></td>
</tr>
<tr>
<td>AIRFLOW MEASUREMENT.</td>
<td>10</td>
</tr>
<tr>
<td>ENGINE AND EQUIPMENT.</td>
<td>17</td>
</tr>
<tr>
<td>PROCEDURE.</td>
<td>19</td>
</tr>
<tr>
<td>CHAPTER III</td>
<td></td>
</tr>
<tr>
<td>TEST RESULTS.</td>
<td>20</td>
</tr>
<tr>
<td>CONCLUSIONS.</td>
<td>33</td>
</tr>
<tr>
<td>BIBLIOGRAPHY.</td>
<td>34</td>
</tr>
<tr>
<td>APPENDIX</td>
<td></td>
</tr>
<tr>
<td>Laminar Flow Meter Calibration</td>
<td>36</td>
</tr>
<tr>
<td>Spark Plug Adapters.</td>
<td>40</td>
</tr>
<tr>
<td>Frequency Plot</td>
<td>44</td>
</tr>
</tbody>
</table>

iv
STATEMENT OF THE PROBLEM

The purpose of this thesis was to construct and calibrate a laminar air flow meter and to verify the application of a mathematical model suitable for the design of multicylinder manifolds.
CHAPTER I

INTRODUCTION AND HISTORY

The possible ways of increasing the volumetric efficiency of an engine over the past few years have received much consideration. The approach studied has been to try to increase the kinetic energy of the air during the induction process thus achieving a "ram-charge" effect or supercharging effect in the cylinder. Many manifold configurations have been proposed and they vary in size and complexity.

The simplest theory uses the concept of organ pipe resonance in the intake manifold to achieve an increase in the air pressure in the cylinder at valve closure, thus increasing the mass of air inducted into the cylinder. Various authors have investigated this approach: Morse, Boden and Schecter, (2) Lichty (3) and Platner and Moore (4) are representative of this approach. The main disadvantage of this approach is that the number of organ pipe oscillations occurring in the intake manifold must have very nearly an integral relationship with engine
cycle time. This condition only occurs at certain narrow engine speed ranges thus limiting its speed range of gain. Another disadvantage is that desirable gain requires lengthy pipes thus complicating the fabrication of intake manifolds for multicylinder engines.

Another approach to intake tuning is based on Helmholtz resonance of the gases in the inlet pipe and cylinder cavity. The use of Helmholtz resonance in designing an intake manifold for a single cylinder engine has been verified by Thompson and Engelman. The basic parameters involved in the design of the manifolds were shown to be cylinder displacement, intake pipe length, and intake pipe cross-sectional area. Thompson showed that this type of resonance offers a much broader range of gain than does the organ pipe resonance, and under certain conditions both types of resonance can be detected. Usually the organ pipe resonance initiated by manifold configuration is superimposed on the Helmholtz resonance effect. This is observed as small peaks and valleys on the Helmholtz breathing curve.
Since power is proportional to volumetric efficiency, then volumetric efficiency is used as a basis of comparison for different intake manifold configurations. In order to determine volumetric efficiency effectively the volume of air inducted through the intake manifold must be measured very accurately.

Air flow has commonly been measured by using nozzles or orifices, but in numerous special cases, laminar flow meters have been applied. As an engine breathes, pressure pulses are initiated in the intake manifold as the valves open and close, and further pulsation occurs as a result of the intermittent flow to each cylinder. This pulsation is applied to the nozzle or orifice, and because non-linearity of the pressure-velocity relationship (pressure drop proportional to velocity squared) leads to considerable uncertainty in the air-flow measurement. The larger the number of cylinders, the more these pulses overlap, and the better the flow measurement becomes. However, large surge chambers are necessary to minimize flow pulsation at the nozzle or orifice.
Manifold changes may result in significant alteration of the pulsation, even with a multicylinder engine. (In one case reported in a private communication, the reversed flow through the carburetor on a six-cylinder engine threw liquid fuel vertically up out of the inlet several feet.) Consequently, there is no assurance that nozzle or orifice readings can even be compared for two different manifolds.

Therefore this thesis will attempt to verify the application of Helmholtz resonance design for intake manifolds on a multicylinder engine and also to develop a technique of measuring air flow which will not be sensitive to pulsation or which will permit linear averaging of pressure instrumentation readings if fluctuations occur.
THEORY INVESTIGATED

The Helmholtz resonance effect in the intake manifold was first developed by Professor H. W. Engelman.\(^{(7)}\) Later refinements were added to Engelman's approach by Thompson\(^{(5)}\) both of whom did their experimentation on a single cylinder engine. Thompson writes the equation for the tuning peak:

\[
fr = \frac{162}{K} c \sqrt{\frac{A}{LV}} \sqrt{\frac{R-1}{R+1}} \text{ rpm}
\]

where \(K = 2.1\) of 2.2 for most conventional engines

- \(c = \text{speed of sound, feet per second}\)
- \(V = \text{displacement of cylinder, cubic inches}\)
- \(L = \text{inlet pipe length, inches}\)
- \(A = \text{inlet pipe cross-sectional area, square inches}\)
- \(R = \text{compression ratio}\)
- \(162 = \text{constant incorporating units}\)

Later Eberhard\(^{(1)}\) continued the development of a design equation based on the Helmholtz theory for multicylinder engines. The actual equation developed was derived from an electrical analog to the acoustic model. The equation developed was a function of manifold and cylinder dimen-
sions which makes it usable for design. Eberhard writes
his equation for peak tuning as:
\[ f_1 = \frac{162}{K} c \sqrt{\frac{(ab+atl)}{2ab}} - \frac{\sqrt{(ab+atl)^2 - 4ab}}{2ab} \frac{1}{\sqrt{L_1}} \frac{1}{\sqrt{V}} \sqrt{\frac{R-1}{R+1}} \text{ rpm} \]
where  
\[ c = \text{speed of sound, feet per second} \]
\[ K = 2.1 \text{ of 2.2 for most conventional engines} \]
\[ V = \text{displacement of one cylinder, cubic inches} \]
\[ (L/A)_1 = \text{inlet pipe inductance, inches}^{-1} \]
\[ R = \text{compression ratio} \]
\[ a = \frac{(L/A)_2}{(L/A)_1} \text{, the inductance ratio of the feeder pipe to the inlet pipe} \]
\[ b = \frac{V_2}{V_1} \text{, the capacitance ratio of the volume of the idle inlet pipes to the volume of the cylinder at midstroke,} \]
\[ V_1 = \frac{V}{2} \times \frac{R+1}{R-1} \]

Eberhard used a Cummins V-6 diesel engine in his experimental verification using only three cylinders per intake manifold. His results showed that the design equation was valid and that tuning did occur over a broad speed range. However, the engine had three equally spaced induction strokes and there was negligible overlap between induction cycles among the three cylinders involved.
But in other engine configurations, such as an inline four cylinder engine, there is intake and exhaust valve overlap which results in pressure pulsations which could hinder breathing. Therefore this thesis will attempt to verify the Helmholtz resonance theory applied to a four cylinder gasoline engine.

Along with the potential problem in breathing caused by the pulsations is their effect on air flow measurement. Most velocity head measuring instruments yield information which is directly proportional to the flow velocity squared. These pulsations cause fluctuations in the flow at the meter which in turn cause flow velocity fluctuations. The meter then produces readings which have a nonlinear relationship to the true reading. The nonlinear characteristics of the meter make it impossible to determine a true average reading thus introducing possible sources of error. To eliminate this problem a laminar air flow meter was developed. The laminar or linear air flow meter yields readings, in this case pressure drop, which are directly proportional to the flow velocity to the first power. The meter was designed to have sufficient
cross-sectional area so that the flow was always laminar having a Reynolds number approximately equal to 1200. Coupled with sufficient area, the flow was directed through long cylindrical tubes. At this point a pressure drop occurs across each tube due to viscous effects between the air and surface of the tubes. Theoretically the pressure drop should be due entirely to viscous effects of the laminar flow through the tubes. Then

\[ p = C \rho \nu \]

where \( C \) is a constant incorporating compressibility and temperature effects of the air, \( \rho \) equals density and \( \nu \) equals flow velocity. The linear relationship permits an accurate average pressure drop to be determined regardless of the size of the flow fluctuations. Once the pressure drop is established then the flow velocity of the air can be determined which in turn yields calculation of the volume flow rate in the form

\[ Q = A \nu \]

where \( Q \) equals volume flow and \( A \) is the average cross-sectional area through which the flow progresses. Based
on the above discussion a flow meter was constructed and experimentally calibrated.

Therefore this thesis will explain the construction and calibration of the laminar air flow meter and then use it to determine the actual volume flow rates of a four cylinder gasoline engine. These flow rates along with cylinder pressures are then used to evaluate the applicability of Helmholtz tuning in multicylinder manifold design.
CHAPTER II

AIRFLOW MEASUREMENT AND CYLINDER PRESSURES

Airflow to the engine was determined by using a laminar airflow meter. Many configurations were considered when attempting to build the meter at low cost. It was found that corrugated cardboard “B flute” size was a very good way of creating long narrow passages for the air to flow through thus achieving laminar flow. Since the frictional properties of the cardboard were unknown a length to diameter ratio was established experimentally which would yield a static pressure drop of 1 to 2 inches of water. The cardboard was cut, painted, and then stacked into layers until a desired cross-sectional area was obtained. The paint was used to stabilize the cardboard against changing air conditions, such as humidity changes, and thus helped to maintain its calibration state. The cardboard section whose final dimensions were 6 by 4-1/8 by 17-1/2 inches nominal composing 32 layers was supported in a plywood box with appropriate connections permitting
it to be attached directly to the carburetor as shown in Figure 1 on page 12.

The flow meter was calibrated against an air flow nozzle manufactured and calibrated by the Mechanical Engineering Department at Ohio State University. The nozzle and flow meter were connected in series and an exhaust fan was used as a flow source. Theoretically a log-log plot of nozzle pressure drop versus laminar pressure drop should result in a straight line of slope 2 since nozzle pressure drop varies as the square of velocity and laminar pressure drop varies linearly with velocity. However, this did not occur and a slope of approximately 1.7 was obtained. This indicated that the pressure drop in the flow meter was not totally attributed to viscous effects but that there were kinetic effects present also. This suggested that the calibration equation be modeled as

\[ p = A \mu Q + B \mu Q^2 \]

where \( p \) = pressure drop, inches water
\( A, B \) = constants
\( \mu \) = viscosity ratio
Figure 1. Flow Meter

Figure 2. Manifolds
Q = volume flow, cubic feet per second
\( \phi = \) density of air, pounds per cubic foot

The \( A \phi Q \) is a viscous term and the \( B \phi Q^2 \) is the kinetic term. As the flow velocity increases and becomes approximately turbulent the kinetic term becomes more predominant until all viscous effects are negligible. Tests indicated that the flow meter was almost laminar up to 1.8 inches of water pressure drop which was the limit of the blower. However, this was more than satisfactory since pressure drops during the experiment never exceeded 1.2 inches of water. The temperature effects (density and viscosity) were confirmed by calibration in 85 degree room air and in 50 degree outside air.

A final calibration equation was determined and was written as

\[
P = 0.372863 \times 768 Q + 0.00165273 \frac{P}{\phi} Q^2
\]

where \( T = (1 + \phi P/460) \); \( \phi_P = \) room temperature

\( P = \) barometric pressure, in Hg

with \( P \) and \( Q \) as previously defined. The constants arise from conversions of viscosity and density to a function.
of room temperature and barometric pressure. For a more complete derivation, see Appendix page 37. This equation was believed to be accurate to within 1.0 percent. The calibration equation was then solved for volume flow as a function of pressure drop and then tabulated calibration tables were calculated by computer for different combinations of ambient temperature and barometric pressure.

Cylinder compression pressures were also recorded during the experimentation to be used for comparing the amount of supercharge gain of the three manifolds. Mercury manometers were built for this purpose as shown in Figure 3 on page 15. Sparkplug adaptors were made consisting of a 0.030 inch orifice and were installed in place of the regular spark plugs. See Figure 4 on page 15. These were connected to the wall mercury manometers.

It may be assumed that the average cylinder pressure measured through the orifices is proportional to the total pressure in the cylinder. This has been verified by Ahuja (8) and has also appeared in a publication by Thompson and Engelman (5).
Figure 3. Nanometers

Figure 4. Spark-Plug Adapters
Another parameter valuable for comparative purposes is the mean effective pressure which shows how well the engine is using its size to produce work. Mean effective pressure is defined as that theoretical constant pressure which can be imagined exerted during each power stroke of the engine to produce work equal to the actual work. Therefore, frictional mean effective pressure can be used as a parameter for comparing the pumping work for the three different manifolds.
ENGINE AND EQUIPMENT

The engine used during the experimentation was an International four cylinder gasoline engine, model 4-152E, having a compression ratio of 8.19:1. Each cylinder had a bore of 3.875 inches and a stroke of 3.2125 inches for a total displacement of 38 cubic inches per cylinder. The decision was made that the engine should be run unfired, to avoid the complication of two-phase manifold flow. Since the objective was establishment of the validity of a mathematical model, and the results without gasoline are directly applicable to diesel engines and air compressors, motoring tests appeared to be appropriate. The non-fired engine was coupled to and driven by a General Electric LC dynamometer, serial number 7218138. Engine air flow was measured by using a laminar air flow meter, as shown in Figure 1 on page 12.

The stock manifold and two prototype manifolds were used on the engine. Figure 2 on page 12 gives a general view of prototype No. 2 and the stock manifold for comparison. The valve timing is such that the intake valve
opens at 18 degrees BTC and closes at 58 degrees ABC, and
the exhaust valve opens at 58 degrees BBC and closes at
18 degrees ATC. The valve timing was assumed to be very
nearly the same as for a fired engine since the water
temperatures were kept near the normal operating range.
The engine exhaust was connected to the building exhaust
system. The engine water temperature was not controlled
by the engine thermostat but was controlled remotely. Oil
temperatures, when excessively high, were reduced by a
small blower situated under the engine.

The average engine speed was determined by using
a Standard Electric Time Company revolution counter and
timer, type SG-6. Torque was measured with the dynamo-
meter arm Fairbanks beam balance. Temperatures were re-
corded from a Brown Electronik Thermocouple Potentiometer.
PROCEDURE

Each test run was started by letting the engine reach operating temperatures at a speed of 2100 rpm. The warm-up period was usually one-half hour since room temperature for all runs averaged 71°F. However, stabilization was achieved within ten to fifteen minutes immediately following a previous run.

The time required for sub-runs was approximately five minutes. Data was monitored for a sub-run test period of nearly one minute. This procedure was very reliable in producing consistent and repeatable data.
CHAPTER III

TEST RESULTS

The test results are depicted in Figures 5 through 11. Figures 5 through 8 show a comparison of cylinder pressure (for all three manifolds) versus engine speed for each cylinder. Noticeable pressure gains delivered by prototype manifolds No. 1 and No. 2 are observed over a speed range from 1200 to 2500 rpm. These gains measure nearly 10 percent. Prototype No. 1 always gave rise to higher cylinder pressures than did the stock manifold over the speed range tested. However prototype No. 2 cylinder pressures fell below the stock manifold cylinder pressures above 2500 rpm. This is due to the fact that its cross-sectional area was not large enough to permit the engine to breathe normally at high speeds. Figure 9 which depicts volume flow versus engine speed also verifies the last statement to be true. The flow in the two prototypes became constricted thus reducing the volumetric efficiency of the engine at higher speeds. Figure 10 shows a plot of volumetric efficiency versus engine speed
comparing the three manifolds. Again it is noted that prototype number two created the greatest flow restriction. For comparison, the flow area to each cylinder of prototypes No. 1 and No. 2 and the stock manifold were 0.825, 0.772, and 1.719 square inches respectively.

The peaks and valleys observed on the curves for cylinder pressures versus engine speed are organ pipe resonance superimposed on the Helmholtz resonant curves.(6) If one pictures a smoothed Helmholtz curve a noticeable gain in volumetric efficiency is observed for prototype number one in a speed range of from about 1700 rpm to 2500 rpm. As shown in Figures 9 and 10, reductions in volume flow rate and volumetric efficiency are observed in the range from 1200 to 1600 rpm. This is believed to be caused by an unknown resonant condition occurring in the intake manifold. An attempt was made to determine what kinds of resonance were present in the experiment in hopes of explaining this phenomenon. A Bruel and Kjaer frequency analyzer was used to obtain frequency plots for various engine speeds at the flow meter entrance. Figure 15 in the Appendix shows the results of this data.
The only conclusion which can be stated is that the fundamental of engine gulp and higher harmonics were easily identified and that there were also subharmonics present. Therefore, no statement is made at this time concerning the elimination of the resonant phenomenon. A more refined acoustical study is recommended.

The mathematical model developed by Eberhard(1) listed on page 6 was applied to all three manifolds. A tuning peak at 2260 rpm was determined for manifold number one and a tuning peak of 2167 rpm was determined for manifold number two. Analysis of Figures 5 through 8 for cylinder pressures neglecting the superimposed organ pipe resonance reveals very good agreement between the data and mathematical model. The stock manifold was also modeled but it should be noted that it is not of the same basic configuration as was Eberhard's model. The model as shown in Figures 12 and 13 can be thought of as a resonant volume connected to each cylinder by equal length runner pipes. The inlet pipes for the stock manifold vary in length between the outer and inner cylinders by 43 percent. In Eberhard's model, the stock manifold has
two "modes": cylinders 1 and 4 have "long" pipes and a relatively smaller secondary volume, while cylinders 2 and 3 have "short" pipes and a larger secondary volume. Calculated values for the two modes differed only slightly, and a peak was predicted at their average, 2240 rpm. This is in agreement with the data, again independent of the superimposed organ pipe resonance, shown in Figures 5 through 8.

The frictional mean effective pressure for the two prototype manifolds was on the average a small percentage higher than for the stock manifold as shown in Figure 11 which is a plot of frictional torque, directly proportional to the frictional mean effective pressure, versus engine speed. This indicates that the pumping work of the two prototype manifolds is a little greater than for the stock manifold but this must be weighed against the fact that the prototype manifolds yield more power. See Figure 14.
Figure 5
Cylinder No. 1  - Prototype No. 1
Cylinder Pressure - Prototype No. 2
vs
Engine Speed  - Stock
Figure 6

Cylinder No. 2 — Prototype No. 1
Cylinder Pressure — Prototype No. 2
vs
Engine Speed — Stock
Figure 7

Cylinder No. 3  □- Prototype No. 1
Cylinder Pressure ◦- Prototype No. 2
vs
Engine Speed   △- Stock

Engine Speed, rpm
Figure 8
Cylinder No. 4  Ø- Prototype No. 1
Cylinder Pressure Ø- Prototype No. 2
vs
Engine Speed △- Stock

Engine Speed, rpm
Figure 9. Volume Flow Rate Versus Engine Speed

- □ - Prototype No. 1
- ○ - Prototype No. 2
- △ - Stock
Figure 10. Volumetric Efficiency Versus Engine Speed

- □- Prototype No. 1
- ○- Prototype No. 2
- △- Stock
Figure 11. Friction Torque Versus Engine Speed

- Prototype No. 1
- Prototype No. 2
- Stock
- Figure 12. Eberhard's Model of the Intake Manifold for a Four Cylinder Engine.

- Figure 13. Electrical Analog of the Model for the Intake Manifold for a Four Cylinder Engine.
Figure 14. Power Curve Obtained from Undergraduate Laboratory Data.
CONCLUSION

Two conclusions can be identified with the experimental results. One is that a low cost and very accurate laminar flow meter can be constructed and calibrated to be used for measuring the volume flow rate of air inducted as pulsating flow by a multicylinder engine.

Also it is seen from the results of this investigation that the mathematical model proposed is a valid one. The supercharge gain results from the Helmholtz resonance and not from organ pipe resonance. The model is applicable to four-cylinder engines where valve overlap which was relatively small initiates pressure pulsations in the intake manifold which possibly could have suppressed Helmholtz resonance. Increased pumping losses in the two prototypes are offset by the increased supercharge of the cylinders resulting in increased power over the stock manifold.


APPENDIX
Calibration of Laminar Flow Meter

As stated on page 11 in Chapter II a log-log plot of the pressure drop across a nozzle versus the pressure drop across the laminar flow matrix (laminations of corrugated cardboard) should theoretically yield a straight line having slope equal to 2. However, for the data obtained the slope was approximately equal to 1.7. The deviation from the theoretical line increased with increasing pressure drop. This indicated that some of the pressure drop was caused by a kinetic energy loss term present causing the non-linearity. Therefore, the calibration equation was modeled to include a linear viscous term and a kinetic energy term proportional to the square of velocity. Flow velocities were checked to insure that the flow was laminar or having a Reynolds number approximately equal to 1200.

The basic equation was written as

\[ p = A \mu Q + B Q^2 \]  \hspace{1cm} (1)

where \( p \) = pressure drop across the laminar matrix, inches of water

\( \mu \) = viscosity ratio
\[ \rho = \text{density of air} \]
\[ Q = \text{volume flow rate directly proportional to flow velocity.} \]
\[ A, B = \text{constants} \]

This type of model was an attempt at fitting the curve obtained from the plotted data. This equation can be written as a function of temperature, barometric pressure, and volume flow rate which make its use more desirable. Adiabatic expansion and compressibility effects at the matrix entrances were determined to be much less than 1 percent therefore they were neglected. Viscosity can be written as a function of temperature in the form:

\[ \eta = \eta_0 \left( \frac{T}{T_0} \right)^{0.768} \]

with \( \eta_0 \) at \( T_0 = 0.01709 \) at the triple point of water and one atmosphere pressure. Therefore the viscosity can be written directly as

\[ \eta = A \left( T \right)^{0.768} \]

where \( T = (1 + \frac{^0F}{460}) \), \(^0F = \text{room temperature. Similarly density can be written as} \]

\[ \rho = B \frac{F}{T} \]
where \( P = \) barometric pressure, inches of mercury
\( T = \) degrees absolute
\( B = \) constant

Therefore, equation (1) can be written as
\[
p = C T^{0.768} Q + D \frac{P}{T} Q^2
\]

where \( C \) and \( D \) incorporate all previous constants. Data points were used to calculate \( C \) and \( D \). The flow rate \( Q \) was determined from the corresponding pressure drops across the nozzle. For a nozzle
\[
\Delta P = M C_D \frac{Q}{Q} v^2
\]
where \( C_D = \) nozzle discharge coefficient
\( v = \) flow velocity, feet per second
\( M = \) units conversion constant
and others are as previously defined. Then volume flow rate can be calculated as
\[
Q = \sqrt{\frac{P}{M C_D \frac{Q}{Q}}} \times \text{nozzle throat area}
\]

The final equation was determined and written as
\[
\Delta P = 0.372863 (T)^{0.768} Q + 0.00165273 \frac{P}{T} Q^2
\]
where \( T = 1 + \frac{O_P}{460} \)
\( O_P = \) room temperature in degrees Fahrenheit
\( P = \) barometric pressure in inches of Mercury
The above equation was found to yield less than 1 percent error. Then the equation was solved for volume flow rate as a function of laminar pressure drop, temperature, and barometric pressure. By using the computer, calibration tables were computed for various combinations of pressure drop, temperature, and barometric pressure. From these tables one could easily determine the engine air flow rate versus engine speed.
Spark-Plug Adapters

Spark-plug adapters were machined so that each had a nominal 0.030 inch orifice in the end. All four adapters were used in each cylinder for prototype manifold No. 1. A plot of cylinder pressure versus engine speed was made for each cylinder with all four adapters. However, due to probable inaccuracies or differences in machining in orifice size or throat length, which causes the discharge coefficients to be different, for reversing flow, no two read exactly the same. But a pattern was established. Orifices in adapters labeled 5 and 8 were fairly consistent and 6 and 7 were consistent. 5 and 8 were always greater than the average of the four readings and 6 and 7 were always less. Since the average of 5 and 6 was within a few percent of the total of all four averaged, it was decided to only use these two in cylinders 1 and 2. The same applied to orifices 7 and 8 and they were used in cylinders 3 and 4. This procedure cut experimental time in half since only two orifices were used per cylinder.
Figure 16 shows the average deviation from the average pressure versus orifice pressure reading for orifices 5 and 6. This curve was obtained by plotting the deviations for each and then a line was drawn through the average deviation and origin. Therefore, the average cylinder pressure can be obtained from one orifice reading by adding or subtracting the appropriate amount from the curve corresponding to its pressure reading. Figure 17 shows the same information for orifices 7 and 8.
Lines are gulp frequency and harmonics common to all manifolds. Points are peaks noted for individual manifolds.

Figure 15. Engine Speed Versus Frequency

- Prototype No. 1
- Prototype No. 2
- Stock