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PURDUE UNIVERSITY

ENGINEERING EXPERIMENT STATION

LAFAYETTE, INDIANA

PR

PROGRESS REPORT ON STUDY OF MULTI-CYLINDER ENGINE MANIFOLDS

By

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ARMY AIR FORCES COOPERATIVE RESEARCH PROJECT

M-125-1

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PURDUE UNIVERSITY

OFFICE OF THE DEAN OF ENGINEERING
LAFAYETTE, INDIANA

July 10, 1944

Commanding General
Army Air Forces Materiel Command
Wright Field
Dayton, Ohio

Subject: Letter of transmittal for report "Progress Report on Study of Multicylinder Engine Manifold", Contract No. W535 ac-38886, Engineering Experiment Station Project M-125-1.

Gentlemen:

This progress report completes approximately the first half of an investigation on the study of multicylinder engine manifolds as covered by Contract No. W535 ac-38886, Engineering Experiment Station Project M-125-1. This particular report has been concerned primarily with an investigation of the performance of a wide variety of designs of experimental manifolds. No attempt was made to work out a specific design for any given condition, but rather to study the various types to see the effect of changes in various parts of the manifold system on its performance.

The results of this investigation indicate that there are two types of vibration which take place in multicylinder manifolds that have an effect on the volumetric efficiency of the engine. The first is a ramming action and the second is the vibration of the air in the intake pipe against the spring effect of the manifold and cylinder volume. It was found that the individual intake pipe gave higher volumetric efficiency than the multi-pipe manifold; however, the tests indicate that it might be possible to produce equally high volumetric efficiencies in multicylinder manifolds if the two-type vibration can be tuned to give maximum effects at the same speed.

This investigation will be continued. Considerable attention will be given to the possibility of setting up electrical models of manifold system to permit the study of preliminary designs. It is thought that this method has some possibilities with regard to getting preliminary information on manifolds which could later be interpreted in terms of actual manifold design to reduce the amount of experimental work necessary in connection with their development.

Respectfully submitted,

A. A. Potter

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Dean of the Schools of Engineering
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PURDUE UNIVERSITY

Engineering Experiment Station

Progress Report on Study of Multicylinder Engine Manifold

Army Air Forces Cooperative Research Project
M-125-1
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I. SUMMARY

This report is concerned with preliminary investigations of multicylinder intake manifolds. The chief purpose of the experimental work was to obtain general ideas about the factors affecting the performance of a rather wide variety of manifolds. All of the experimental work for this report was done on a 4-cylinder, 4-stroke cycle aircooled engine with a displacement of 20 in.^{3} per cylinder. This engine was selected because it had individual intake ports which were adaptable to a wider variety of manifolds than dual ports common to many engines.

In order to determine relative volumetric efficiencies, compression pressures were measured while the engine was motored by a dynamometer. Balanced pressure gages were used for measuring compression pressures. Diagrams of pressure variations in the manifolds were obtained by photographing the screen of a cathode ray oscillograph which was used with magnetic pressure pickups.

Several manifold arrangements were used. Some data were obtained with individual pipes on the cylinders. A dual port arrangement was used on different pairs of cylinders in order to determine the effects of uniformly and non-uniformly spaced intake strokes on the performance of a dual port manifold. Four cylinder manifolds of several types were used. The first was rake type with a large diameter header and relatively large branch pipes. The second was a commercial manifold of cast aluminum. It provided well streamlined flow paths and the shortest possible distances from the inlet to the cylinders. The third type was a small diameter (3/4" pipe) rake type manifold. This manifold provided rather high resistance paths of different lengths

to the cylinders. The fourth type of manifold was made of 3/4" pipe and it had loops of pipe of considerable length between the branch pipes to the cylinders. In this manifold the differences in inertia and friction of the paths to the cylinders were greatly exaggerated. The last two manifolds were used with inlets at the end of the header and also at the center. Both of these were used with and without balance pipes connecting between the ends of the header. All manifolds were tested with a variety of pipe lengths and diameters connected to the inlets.

The tests indicated that there are two types of vibration that have important effects on the volumetric efficiencies of multicylinder manifolds. The first is a ramming action and the second is a vibration of the air in the intake pipe against tho spring of the manifold and cylinder volume. The speeds at which both of these types of vibration will produce their maximum beneficial effects can be calculated by the use of Equation 7 in this report.

It was found that individual intake pipes gave higher volumetric efficiencies than any of the other manifold tests; however, the tests indicated that it might be possible to produce equally high if not higher volumetric efficiencies with multicylinder manifolds if the two types of vibration were "tuned" to give maximum effects at the same speed. Single cylinder and dual port manifolds produced only a ramming effect except with very long pipes. These manifolds gave a volumetric efficiency curve that peaked at one speed and fell off above and below that speed. The dual port manifolds gave volumetric efficiencies about 5% lower than single cylinder manifolds. When the intake strokes of the cylinders on the dual port manifold were uniformly spaced with respect to time, the two cylinders had equal volumetric efficiencies. When the valve opening periods of the two cylinders overlapped, the cylinder with the valve closing at the time when the other valve was opening, received a reduced amount of charge.

All the 4-cylinder manifolds except the last one had similar characteristics. The cylinders were quite well balanced regardless of intake position. When no intake pipe was attached to the header, the principle vibration was a ramming action that produced volumetric efficiency curves similar to those obtained with single cylinder manifolds. When an intake pipe was attached an additional vibration of the mass in this pipe against the spring action of the manifold volume produced a peak in the volumetric efficiency curve. When the effects of this vibration were combined with the effects of ramming, flat volumetric efficiency curves frequently occurred. The addition of a balancing pipe to the third 4-cylinder manifold reduced the peak value of volumetric efficiency.

The fourth 4-cylinder manifold with its exaggerated inertia and friction effects gave extreme unbalance between cylinders. The addition of a balancing pipe to this manifold reduced but did not eliminate entirely the unbalance between cylinders. The balance pipe also improved the volumetric efficiencies obtained at high speeds because it provided two paths to each cylinder which had lower effective inertia and friction than the single path.

II. SCOPE

This report gives the results of a preliminary investigations of multi-cylinder intake manifolds. The chief purpose of the experimental work was to get a general idea of the possibilities of a wide variety of manifolds. No attempt has been made to develop a refined theories or a mathematical analyses for multi-cylinder manifolds. Most of the experimental work reported here was done on a four-cylinder engine.

III. EXPERIMENTAL APPARATUS

A. Engine

The engine used for the four-cylinder manifold studies was a four-cylinder Henderson motorcycle engine. It was selected because each intake port was separate thus making possible a wider variety of manifolds than would be possible with a dual port arrangement. The engine dimensions and an intake valve lift curve are given in Fig. 1. The small displacement of the cylinders (20 cu. in.) as compared with most aircraft engines makes it necessary to use relatively long manifold sections in order to obtain results similar to the results that would be obtained with larger engines.

For the experimental work the engine was motored with an electric dynamometer. Fig. 2 shows the engine and dynamometer. It was possible to reach a maximum speed of 3400 rpm with this arrangement.

B. Manifolds

The manifolds were built up of standard iron pipe and pipe fittings. This type of construction made it possible to readily change the dimensions of different sections of the manifolds. The pipe fittings introduced a rather large amount of friction in the system, but since the primary purpose of these experiments was to obtain general ideas of the characteristics of different systems, greater refinement did not seem necessary.

The types of multi-cylinder manifolds used are shown in Figures 3, 4, 5, 6, and 7. Some single cylinder data were obtained first for use in checking previous single cylinder data and for comparison with multi-cylinder data. The first multi-cylinder manifold used was a dual port arrangement shown in Fig. 3. This manifold was used on cylinders 1 and 2 and also on cylinders 3 and 4. The engine had a firing order of 1-3-4-2; therefore, with the dual port manifold on cylinders 1 and 2, the intake pulses were unevely spaced and there was 60° of overlap of the valve openings. With the dual port arrangement on cylinders 2 and 3, the intake processes were evenly spaced and there was no overlap of the valve openings on the two cylinders. This manifold was used with a number of different pipe lengths and diameters connected to it.

The second multi-cylinder manifold used, as shown in Fig. 4, is a rake type manifold with a header of 1-1/4" pipe and branch pipes of 1" pipe. The branch pipe length was 12-1/4" as shown in the drawing. Three different lengths of intake pipe were used. Only an end inlet was used because the test results indicated that there would be only a very small difference in results with a center inlet on this particular manifold. Center inlets and balancing pipes were used on other manifolds.

The third multi-cylinder manifold was a commercial cast aluminum manifold shown in Fig. 5. The flow paths to the cylinders were short and relatively large in diameter with well rounded corners. This manifold should have, relatively, a very low resistance.

The fourth multi-cylinder manifold used was a rake type made of 3/4" pipe throughout (Fig. 6). This manifold was used with an end inlet and also with a balancing pipe connecting the two ends of the header as shown by the dashed lines. With the balancing pipe, inlets at the end of the header and also at the center of the balancing pipe were used.

A fifth type of four-cylinder manifold (Fig. 7) was used. This manifold had long sections of pipe between the branch pipes to the cylinders. These sections were made long by using loops of pipe between cylinders. The dashed lines in Fig. 7 show the position of a balancing pipe and various inlet positions that were used.

C. Instruments

The relative volumetric efficiencies of the cylinders were determined by measuring the compression pressures with balanced pressure gages mounted in each cylinder (see Fig. 2). The electrical contacts in the balanced pressure gage were connected through an amplifier to a strobotac so that the strobotac flashed each time that the contacts in the balanced pressure gage made contact. The contacts made only when the cylinder pressure exceeded the pressure applied to the top of the balanced pressure gage. The pressure applied to the top of the gage was adjusted by needle valves in the pressure control unit shown just under the Bourdon gage in Fig. 8. The pressure applied was read from the Bourdon gage. The balancing pressure just sufficient to stop the flashing of the strobotac was equal to the peak compression pressure in the cylinder. Readings could be made on the various cylinders by turning the selector switch on the amplifier.

Diagrams of the pressure variations in the manifold were obtained with a magnetic type of pressure pick-up used in conjunction with an integrating circuit and a cathode ray oscillograph. Records of the pressure variations were made by photographing the screen of the oscillograph with a 35 mm camera. Pressure diagrams obtained in this way are shown in this report.

The pressure pick-ups were calibrated on a rotating valve device shown in Fig. 9. One of the magnetic pick-ups is shown screwed into the top of the valve. The rotating valve was driven by a motor and alternately exposed the diaphragm of the pick-up to the air pressure in the small tank, shown in the photograph, and then to atmospheric pressure. The valve action was such that a square pressure wave was applied to the pick-up. The flat top on the wave represented the tank pressure and the flat bottom of the wave represented atmospheric pressure. The pressure in the tank was measured with a manometer. In order to prevent changes in amplifier gain from affecting the readings, a known a.c. calibration voltage was substituted for the pick-up output voltage for setting the gain and checking amplification during both calibration and use of the pick-ups.

Engine speeds were determined with a strobotac.

Fig. 10 shows the entire experimental set-up with the pressure measuring devices attached. In this photograph, the two-cylinder manifold is shown with two magnetic pick-ups attached at the elbows leading into the cylinders. Most of the data on this manifold were obtain with only one pick-up attached at the center of the Tee as indicated in Fig. 3. The balanced pressure units can be seen in the tops of the cylinders. The balancing pressure tube connecting the gages with the pressure control unit is also shown.

IV. THEORY

The pressure phenomena that occur in engine manifolds may be classified into several groups. These phenomena will be defined and discussed now in order to aid in the discussion of the various manifolds that follows.

The first type of phenomena will be called ramming. A ramming action always exists in single cylinder intake pipes and also in the branch pipes leading into the individual cylinders on multi-cylinder manifolds. During the first part of the intake stroke the air in the pipe is accelerated and during the last part of the intake process when the piston slows down, stops and reverses its direction, the kinetic energy gained by the air in the pipe tends to keep it moving into the cylinder. This action will cause the air in the cylinder to be compressed, in some cases, to pressures considerably above atmospheric pressure. If the mass and spring constant for the intake system are properly proportioned the ramming action can be made to give peak volumetric efficiency at any desired speed.

The mass of air in the pipe is given by the following equation:*

$$m = \frac{\pi d^2}{4} \rho l = al\rho \text{, lbs/386, } \frac{\text{lb-sec}^2}{\text{in.}} \qquad (1)$$

where d = pipe diameter, inches

a = cross sectional area of pipe, in.^{2}

l = pipe length, inches

ρ = air density in mass units, lb/in.^{3}/386, $\frac{\text{lb sec}^2}{\text{in}^4}$

386 = the acceleration of gravity, in/sec^2

The spring constant for the volume of air involved is given by the following equation:

$$k = \frac{\pi^2 d^4 \rho c^2}{16 V} = \frac{a^2 \rho c^2}{V} \qquad (2)$$

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* Morse, Vibration and Sound, page 201

where d = pipe diameter, inches

ρ = air density in mass units, $\frac{\text{lb sec}^2}{\text{in.}^4}$

c = velocity of sound in air, in./sec = 13,900 in./sec for air at 100° F

V = total volume involved, in.^{3}

a = cross sectional area area of pipe, in.^{2}

The velocity of sound in air can be calculated from the following equation:

$$c = 12\sqrt{gkRT} \quad \text{in./sec} \tag{3}$$

where g = acceleration of gravity, ft/sec^2 = 32.2

k = ratio of specific heats = 1.4 for air

R = gas constant, ft-lb/lb-°F = 53.3

T = absolute temperature of air, °F abs.

Good results are obtained when the maximum cylinder volume plus 1/2 of the pipe volume are used in Eq. 2. This maximum cylinder volume may be calculated from the following equation:

$$V_c = D\left(\frac{r}{r-1}\right), \text{ in.}^3 \tag{4}$$

where V_c = maximum cylinder volume, in.^{3}

D = displacement volume, in.^{3}

r = compression ratio

About one-half of the pipe volume should be added to this volume for calculating sthe spring constant for the total volume. One-half of the pipe volume is given by the following equation:

$$V_p = \frac{\pi d^2 l}{8} = \frac{a\,l}{2} \tag{5}$$

The total volume V is then

$$V = D\left(\frac{r}{1-1}\right) + \frac{a\,l}{2} \tag{6}$$

The natural frequency of the system can then be calculated as follows:

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}} = \frac{1}{2\pi}\sqrt{\frac{a^2 \rho\, c^2}{V\, a\, l\, \rho}} = \frac{1}{2\pi}\sqrt{\frac{a\, c^2}{V\, l}} \text{ cycles/sec}$$

or

$$f = \frac{60}{2\pi}\sqrt{\frac{a\, c^2}{V\, l}} \text{ cycles/min} \tag{7}$$

The peak volumetric efficiency due to ramming occurs at approximately 70% of the natural frequency speed given by Eq. 7 for single cylinder systems.

In multi-cylinder manifolds the air in the interconnecting pipes between cylinders produces some ramming action along with the air in the individual branch pipes. It is difficult to calculate, by any simple means, the speed at which the peak ramming action will occur; however, an understanding of the factors affecting the ramming action as shown by Eq. 7 aids in making intelligent modifications of the manifold which will produce the desired manifold performance.

In all manifolds, single cylinder or multi-cylinder, there may be vibrations similar to the vibrations in organ pipes. The lowest natural frequency of a pipe closed at one end can be calculated by dividing the velocity of sound by four times the pipe length. Odd harmonics also exist in such a pipe, that is, natural frequencies at 3, 5, 7, etc. times the lowest natural frequency exist. A pipe with both ends open has a lowest natural frequency twice that of a pipe closed at one end and all harmonics may exist. If there is any extra volume at the closed end of a pipe the lowest natural frequency can be calculated from Eq. 7, in fact, Eq. 7 can be used for calculating the lowest natural frequency of a closed end pipe with no extra volume if 1/2 the pipe volume is used for V in the equation.

In most engine manifolds the pipes are so short that the lowest natural frequency of the pure organ pipe type of vibration is far above the frequency of the exciting forces so that vibrations of this type are not excited to any great extent and their effects are minor. Usually the vibrations that occur are the result of a mass of air in a pipe vibrating against the spring action of a volume on the end of the pipe.

In single cylinder manifolds there is a damped free vibration of the organ pipe type existing in the intake pipe after the intake valve closes. This vibration starts anew each time the valve closes and does not build up from one cycle to the next so that no true resonant effect exists. Usually the effects produced by this vibration are not of very great importance. With moderately long pipes of large diameter there may be sufficient velocity of the air existing in the pipe at the time of the valve opening for the next cycle to produce some extra humps in the volumetric efficiency curves. With very long pipes which have natural frequencies of the same order as the engine cycle frequency, there is a possibility of resonant vibrations occurring. Previous tests run on the C.F.R. engine have shown that this type of vibration will not produce volumetric efficiencies as high as those by ramming action alone.

Pure ramming action does not involve any resonant vibration effect, in other words, there is no accumulation of vibrational energy from one cycle to the next. With certain manifolds there is a type of vibration that may become resonant. It is excited by the cyclic forces produced by the repetition of intake strokes, rather than by each individual intake stroke as in the case of ramming. The results of some of the four-cylinder manifold tests indicated that the combined exciting forces of the four cylinders produced a vibration of the mass of air in the intake pipe to the manifold against the spring effect of the volume of the manifold and cylinders. This type of vibration resonates at a speed where the frequency of the exciting force is equal to the natural frequency of the system. Resonance may be produced in a system of this type with short intake pipes (small mass) because the exciting frequency is high in multi-cylinder engines and the natural frequency is usually quite low due to the large volume that acts as a spring.

The speed at which the maximum pressure amplitudes in the intake system will be obtained is the speed at which the frequency of the exciting force is equal to the natural frequency of the system. The amplitude of the pressure waves will be less at both higher and lower speeds. The natural frequency will be affected by the same factors as those shown in Equation 7; however, the exact values of volume and pipe length to be used are sometimes difficult to determine. The speed for resonance is not necessarily the speed at which vibrations of this type will produce the maximum volumetric efficiency. In order to obtain the maximum volumetric efficiency the phase relation between the engine cycle and the pressure waves must be such that a high pressure is produced at the valve at the time of valve closing. At low speeds the driving force and the displacement of the mass are in phase. At speeds near the natural frequency the displacement begins to lag behind the force and at speeds well above the natural frequency the displacement lags behind the force by nearly 180°. Except in cases where the damping is very large the phase angle changes quite rapidly from low values to nearly 180° within a narrow range of speeds near the resonant speed; therefore, it is likely that, if benefit is obtained from this type of vibration, the peak effect will occur close to the resonant speed regardless of what phase angle is necessary to produce the peak effect. If a leading phase angle is required to produce beneficial effects, then only detrimental effects can be produced by the vibration because the displacement always lags behind the force to a greater or less extent.

V. EXPERIMENTAL RESULTS

A. Single Cylinder Data

Figures 11, 12, and 13 show curves of compression pressure versus speed for a number of different pipe sizes on a single cylinder. The general shape of all of these curves is the same as for the curves previously obtained on the C.F.R. engine. These curves are given in Army Air Forces Cooperative Research reports for Engineering Experiment Station Project M-108-6, Contract No. 41-12093. They all show a tendency to peak at one particular speed with peaks of lesser importance distributed along the length of the curves. The major peak apparently is the result of a ramming action. Peak speeds calculated by multiplying the natural frequency from Eq. 7 by 0.70 have been tabulated in Table I for the single cylinder combinations using 3/4" and 1" pipe shown in Figs. 11 and 12. Actual peak speeds and pipe dimensions are also given in the table. The 1-1/4" pipe data from Fig. 13 does not show a single peak that is definite enough to check by this method.

The lengths of 1-1/4" pipe required to obtain peaks at speeds within the speed range of the engine are excessive and do not permit the assumption that ramming action is the only factor affecting volumetric efficiency.

The peak pressures for the curves for 1" and 3/4" pipes lie between 115 and 120 psia while there are rather sharp peaks for the 1-1/4" pipe that exceed 120 psia. These peaks are higher than any that were obtained with multi-cylinder combinations. The probably reasons for this fact are given later in the discussion of multi-cylinder manifolds.

B. Two Cylinder Data

A dual port manifold, shown in Fig. 3, was connected to cylinders 2 and 3 and run with two lengths of 1" inlet pipe. The firing order of the engine was 1-3-4-2 so that the intake strokes on cylinders 2 and 3 were spaced 360° of crank rotation apart. Since the manifold is symmetrical and the intake strokes are evenly spaced with respect to time, the volumetric efficiencies of the two cylinders should be nearly identical throughout the speed range.

The data obtained with this arrangement is shown in Fig. 14 and 15. The two cylinders perform nearly identically in both cases. The speed at the peak with a 1" x 18-1/2" pipe connected to the Tee is about 3000 rpm. The average peak pressure is about 112 psia. This represents about 5% less volumetric efficiency than for single cylinder operation. The speed at the peak with a 1" x 42" pipe was about 2500 rpm and the average peak pressure 114 psia.

This two cylinder system is apparently quite similar to a single cylinder system in vibration characteristics. The two cylinders act independently on the mass in the pipe and there is little interaction between cylinders. The speed for peak volumetric efficiency can again be calculated from Eq. 7. The volume to be used in the equation should be the sum of one cylinder volume, the volume of the ports and Tee, and one-half of the pipe volume. The mass to be used is the mass of air in the pipe exclusive of the mass in the Tee. The calculated peak speeds are 3100 rpm and 2580 rpm for the 18-1/2" pipe and the 42" pipe respectively.

There are probably two causes for the fact that the volumetric efficiency is lower with the two cylinder manifold than with the single cylinder manifolds; first additional friction, introduced by the Tee, tends to lower the volumetric efficiency and second, the volume of air in the Tee and ports must be compressed along with the cylinder air by the ramming action of the mass in the pipe so that some of the energy available for compressing cylinder air is wasted.

The same two cylinder manifold was connected to cylinders 1 and 2. The firing order is such that the intake valve on cylinder 1 opens 180° of crank travel after the intake on cylinder 2 opens. Since the intake valves remain open for 60 degrees after bottom center, the valve on cylinder 2 remains open for 60° of crank travel while the valve on cylinder 1 is open. This overlap of valve opening and uneven spacing of the intake strokes might be expected to cause the two cylinders to have quite different volumetric efficiencies.

Figs. 16 and 17 show the results of tests with the two cylinder manifold on cylinders 1 and 2 with the same pipe sizes that were used on cylinders 2 and 3. Fig. 17 shows the results with a 1" x 18-1/2" pipe. The two cylinders have curves that are similar in shape to the curves for cylinders 2 and 3 as shown in Fig. 15 and the peaks occur at about the same speed (2900 rpm). Cylinder 2 is about 3 psi lower than cylinder 1. The difference in pressures is about the same for all speeds. The difference is probably due to the fact that at the time when the air would ordinarily ram into cylinder two the valve on cylinder one is also open so that the pressure increase in cylinder two due to ramming is reduced. Pressure diagrams shown later for other pipe sizes on this two cylinder arrangement indicate that this is the cause for the difference in volumetric efficiency.

Fig. 16 shows the results of a test of the two cylinder manifold on cylinders 1 and 2 with a 1" x 42" inlet pipe. In this case, the curves have rather irratic shapes. Cylinder 1 has a peak volumetric efficiency at 1700 rpm and a nearly constant volumetric efficiency slightly lower than the peak at the higher speeds. The curve for cylinder 2 crosses the curve for cylinder 1 at 2100 rpm, reaches a peak at 2300 rpm and then falls below the curve for cylinder 1. The two cylinders are rather badly unbalanced (about 11 psi difference) at speeds above 2800 rpm. The reasons for the irratic shapes of the curves are the same as those given in the discussion of Figs. 20 and 2

Fig. 18 shows the results of a test of the two cylinder manifold on cylinders 1 and 2 with a 1/2" x 12" intake pipe. Fig. 21 shows pressure diagrams obtained with a magnetic pressure gage at the center of the Tee for the same manifold and pipe arrangement. The curves of Fig. 18 show a tendency for cylinder 2 to have a uniformly lower volumetric efficiency than cylinder 1. The speed at the peak for this pipe size is low (1800 rpm) as might be expected because Eq. 7 shows that the natural frequency is directly proportional to the square root of pipe area. The peak value of the pressure is low (average of the two cylinders = 99 psia) due to the high friction of the small pipe.

The pressure diagrams of Fig. 21 aid in explaining the difference in volumetric efficiency between the two cylinders and for explaining the variation with speed. It may be noted that the shape of the pressure curve during the intake stroke of cylinder 2 is almost identical to the shape during intake for cylinder 1 except near the time of valve closing. In general there is a pressure increase due to the ramming action of the air just before valve closing. Cylinder 1 gets the full benefit of this action because it is the only cylinder open to the pipe at the time that the ramming action occurs. The pressure increase due to ramming into cylinder 2 is suddenly stopped when the valve due to the ramming into cylinder 2 is suddonly stopped when the valve on cylinder 1 opens causing cylinder 2 to receive less air. The pressure at the time of valve closing for cylinder 2 is always less than the pressure at the time of valve closing for cylinder 1. This difference in pressure agrees with the tendency shown in Fig. 18. The pressure diagrams also show that the inertia of the air in the pipe is too low to give peak pressures at the time of valve closing for speeds below 1800 rpm and too high for speeds above 2400 rpm. This, also, agrees with the trends shown by the curves in Fig. 18.

The data shown in Fig. 19 were obtained with the two cylinder manifold on cylinders 1 and 2 with a 3/4" x 13-1/2" intake pipe. Pressure diagrams obtained at the pipe tee with the same manifold are shown in Fig. 22. These figures show the same tendencies as those just discussed above. The principle difference is that the peak occurs at a higher speed (2700 rpm) and the peak pressure is higher (average = 110 psia). The higher peak speed is due to a larger pipe diameter. The higher peak pressure is due to a reduction of friction as a result of using the larger pipe diameter. It is interesting to note that little is gained on the peak pressure by making the pipe 1" in diameter rather than 3/4"; compare Fig. 19 with Fig. 17. This would indicate that a pipe as large as 3/4" has a negligible amount of friction compared with other friction in the system.

Figs. 20 and 23 show data for a 3/4" x 31" pipe connected to cylinders 1 and 2. The compression pressure curves of Fig. 20 show rather irratic shapes much the same as those obtained with the 1" x 42" pipe shown in Fig. 16 so the same explanations will generally apply in both cases. The principle cause for the more irratic shape of these curves appears to be the greater spring effect of the longer pipe which permits more carry-over of velocity in the pipe from one intake stroke to the next and more carry-over of vibrations from one engine cycle to the next.

The pressure diagram for 1000 rpm shows a rather low pressure amplitude and about equal pressures at the time of valve closing for the two cylinders. This shows up on the compression pressure curves as nearly equal volumetric efficiencies for the two cylinders.

At 1400 rpm the pressure diagram indicates that considerable velocity toward the cylinder remains in the pipe from vibrations of the previous engine cycle when the valve on cylinder 2 opens. Because of this existing velocity, the pressure reduction during the first part of the intake stroke required for accelerating the air is small and a rather early pressure build-up due to ramming results. This early pressure build-up permits cylinder 2 to receive a relatively large amount of air without interference from the valve opening on cylinder 1. The pressure does not fall rapidly enough after the valve on cylinder 1 opens to cause a very great loss of charge from cylinder 2 before its valve closes. The velocity in the pipe at the time when cylinder 1 starts inducting its charge is apparently nearly zero so that a rather large pressure drop is required for accelerating the air in the pipe. This also causes the ramming action to come rather late so that cylinder 1 does not receive the full benefit of the ram before its valve is closed. The action discussed above causes cylinder 2 to have a higher than normal volumetric efficiency and a cylinder 1 to have a less than its peak volumetric efficiency so that the volumetric efficiencies of the two cylinders are not greatly different. This explains the peak in the compression pressure for cylinder 2 shown in Fig. 20.

At speeds between 1800 rpm and 2400 rpm there is apparently little carry-over of velocity between cycles or between intake strokes so that the performance is much the same as for shorter pipes.

At 2600 rpm and above there is apparently considerable carry-over of velocity from the intake stroke of cylinder 2 so that very little pressure drop and not very much time is required for acceleration of the air during the intake stroke of cylinder 1. This causes the ramming action to occur earlier than normal and tends to keep the volumetric efficiency of cylinder 1 high as shown by Fig. 20. The volumetric efficiency of cylinder 2 drops off partly because of late ramming action and partly because the opening of the valve on cylinder 1 permits some of the air to ram into cylinder 1 rather than forcing it all into cylinder 2.

The carry-over of velocities that causes the actions described above can occur only in relatively long pipes where there is enough compressibility in the air at the cylinder end of the pipe to permit an appreciable amount of air motion in the pipe when the valve is closed.

C. Large Diameter Rake Type Four Cylinder Manifold

A four cylinder manifold of the type shown in Fig. 4 was connected to the engine. The data obtained with the No. 4 cylinder end of the header plugged and the other end open with no inlet pipe attached is shown in Fig. 24. The shapes of the curves for all of the cylinders are similar to the shapes obtained with a single pipe arrangement, indicating that the ramming action of the air mass in the branch pipes is the most important factor affecting the performance of this manifold; however, the peak speed calculated from Eq. 7 for a single pipe of the same length as the branches is 4000 rpm. Actually the peak occurs at about 3000 rpm so that there must be some additional effect introduced by the header. The indication is that some of the mass in the header takes part in the ramming along with the masses in the branch pipes in determining the peak speeds. The compression pressures for the various cylinders are within 5 psi of one another and all of the cylinders peak at about 3000 rpm. It is rather surprising to find that all of the cylinders so nearly alike in their action when the lengths of pipe from the inlet to the cylinders are quite different for the various cylinders. The value of the pressure at the peak is high (115 psia) and compares favorably with the values obtained for single cylinder manifolds.

This manifold with a 14" length of 1-1/4" pipe attached to the inlet end was tested. The results of this test are shown in Fig. 25. Pressure diagrams obtained at points just outside the valve ports are shown in Fig. 26. The compression curves for the various cylinders are still closely grouped and all of the curves reach a high value at a rather low speed (1800 rpm) and remain nearly flat up to the maximum speed for the test. The average high value is about 107 psia which is considerably below the 115 psia obtained with no inlet pipe attached. The pressure diagrams show a moderately high amplitude at low speeds and a low amplitude at high speeds which is opposite from the results obtained with all previously tested manifolds. There is evidently an interference between different modes of vibration which tends to reduce the amplitudes at the high speeds. The ramming action of the individual branch pipes evidently tends to cause a peak in volumetric efficiency at a high speed while the vibration of the mass in the inlet pipe against the spring effect of the manifold and cylinder volume tends to give a boost at low speeds and tends to partially cancel the ramming action of the individual pipes at high speeds. The tendency to cancel the ramming action is the result of a phase shift. The rather high peak previously obtained with this manifold and no inlet pipe, or actually a very short effective inlet (Fig. 24), may be due to the fact that these two modes of vibration were synchonized.

The natural frequency of the vibrating system made up of the mass in the intake pipe and the spring effect of the manifold and cylinder volumes can be calculated from Eq. 7 in the following manner:

volume of manifold and ports	= 72 in.^{3}
volume of one cylinder	= 26 in.^{3}
volume of 1/2 of 14" intake pipe	= 10.5 in.^{3}
Total volume	= 108.5 in.^{3}

There is some question about the amount of cylinder volume to be added. Due to overlap of the valve openings two cylinders are connected to the manifold at the same time during parts of the cycle.

The natural frequency from Eq. 7 is

$$f = \frac{60}{2\pi}\sqrt{\frac{a\ c^2}{V\ l}} = \frac{60}{2\pi}\sqrt{\frac{1.5 \times 13{,}900^2}{108.5 \times 14}} = 4180 \text{ cycles/min}$$

Two intake strokes occur each revolution so the rpm at resonance would be 1/2 of the cycles/min for the natural frequency. The rpm at resonance would then be 2090 rpm. Fig. 25 shows that the peak effect from this vibration occurs at about that speed.

Figs. 27 and 28 show results of tests on the same manifold with two additional lengths of pipe on the inlet. These tests gave the same flat type of volumetric efficiency curve; however, the high value was reached at lower speeds. The high value was reached at 1600 rpm with a 23" length of pipe and at 1200 rpm with a 40" length of pipe. There is a slight upward trend of the flat portion of the curve for the 40" length.

D. Four-Cylinder Manifold with Low Inertia Branches

A commercial type of cast manifold was available for this engine. A photograph of the manifold is shown in Fig. 5. This manifold was tested with intake pipes of different sizes connected to the carburetor flange. Figures 29, 30 and 31 show the results of these tests. The shape of this manifold is such that the lengths of the flow paths from the inlet to the various cylinders are as short as possible which should reduce to a minimum the effective inertia of the air in the different branches.

The first test of this manifold, Fig. 29, was run with a 3" length of 1" pipe connected to the carburetor flange. This length of pipe should approximate the effect of a carburetor. The results of the test show that the compression pressures gradually increase with speed up to a maximum value of about 107 psia at 3400 rpm. Whether they would have flattened out and stayed constant for higher speeds could not be determined. The principle cause for the tendency to give a peak at a speed above 3400 rpm is evidently the vibration of the mass in the short inlet pipe against the volume of the manifold and cylinders because the individual branches are too short to cause a peak due to ramming at this speed.

Fig. 30 shows the results of a test with an 18" length of 1" pipe connected to the carburetor flange. This length of pipe causes a peak compression pressure of about 109 psia to occur at 2000 rpm. There is a slight drop in the compression pressure at the higher speeds. Again the principle cause for the peak seems to be the vibration of the mass in the inlet pipe against the volume of the manifold and cylinders. The drop at higher speeds is probably due to the natural tendency for this type of vibration to have a lower pressure amplitude at these speeds (See Section IV). There would also be a tendency at the higher speeds, for the phase angle between the displacement of the mass and the driving force to increase. This may cause part of the drop.

The results of a test run with a 17" length of 3/4" pipe on the inlet are shown in Fig. 31. The results of this test are esstentially the same as those of the previous test. The peak occurs at a slightly lower speed (1800 rpm) due to a reduction of the natural frequency of the vibrating system which results from the use of the smaller diameter pipe. The peak value (106 psia) is slightly lower because of the greater friction in the 3/4" pipe. The small change due to the friction in the pipe indicates that pipe friction, in this particular case, is not a very important factor in effecting the volumetric efficiency.

It is interesting to note that the peak compression pressures obtained with this smoothly streamlined manifold are in general very little higher and in some cases not as high as those obtained with more crudely-built manifolds made of pipe fittings.

The resonant speeds for the system made up of the mass in the intake pipe and the spring effect of the manifold and cylinder volume were calculated from Eq. 7. For details of the method refer to the discussion of Fig. 21 and 22. The resonance speeds for the three pipes used are: 3500 rpm, 1840 rpm, and 1450 rpm. The actual speeds at which the compression pressures reached high values are: over 3400 rpm, 2000 rpm, and 1600 rpm. These speeds are a little higher than the calculated resonance speeds which indicates that a rather large lagging phase angle is required to obtain maximum volumetric efficiency.

E. Rake Type Manifold Made of 3/4" Pipe

It was thought that the relatively large diameters of the manifolds used for the previous tests might tend to minimize the possible effects of inertia and friction in the individual branches; therefore, some manifolds were constructed of 3/4" pipe. The first of these is shown in Fig. 6.

The first test of this manifold was run with an end inlet and no inlet pipe attached. The results of this test are shown in Fig. 32. All of the cylinders except cylinder 4 peak at 2800 rpm and fall off slightly at higher speeds. The peak for cylinder 4 occurs at a lower speed. The peak compression pressure for all cylinders except cylinder 4 is about 108 psia. This value compares favorably with the values obtained with larger diameter manifolds. The general characteristics of this manifold are the same as those of the other 4 cylinder manifolds that have been discussed. Cylinder 4 is probably low at the high speeds partly because of the greater inertia effect of the air in the header leading to this cylinder and partly due to the friction of the long path to the cylinder.

Fig. 33 shows the results of a test of this manifold with a 1" x 22" pipe connected to the end inlet. Again the performance of this manifold is similar to that of the other 4 cylinder manifolds that have been discussed. The compression pressures reach a high value of about 105 psia at 1600 rpm. Above this speed the curves are quite flat and constant; cylinder 1 increases slightly while cylinders 3 and 4 show some decrease. At 3000 rpm there is about 10 psi difference between the best and poorest cylinder. This difference probably is due to the larger amount of inertia and friction in the length of pipe leading to cylinders 3 and 4.

Another test was run on this manifold with a 3/4" x 16" inlet pipe. The results of the test are shown in Fig. 34. With the 3/4" inlet pipe the shapes of the curves are almost identical with the shapes of the curves obtained with the 1" pipe; however, cylinders 1 and 2, as well as cylinders 3 and 4, have slightly decreasing compression pressures at the high speeds. Because of this the cylinders are more evenly balanced. The average peak compression pressure is 105 psi. This same value was obtained with the 1" x 22" pipe.

From an examination of the curves for this manifold it is apparent that the masses of air between cylinders in the header have enough inertia effect to produce a noticable difference in the performance of the various cylinders. The larger diameter manifolds discussed previously did not produce a significant unbalance between cylinders apparently because the effective inertia of the air in a large diameter pipe is relatively small.

F. 3/4" Manifold With Balancing Pipe

A 3/4" balancing pipe connecting the two ends of the header on the 3/4" manifold was added as shown by the dashed lines on Fig. 6. The first test on the manifold with a balance pipe was run with an end inlet and no pipe attached. The distance from the open end to inlet pipe of the first cylinder is a little greater than without the balance pipe because a short nipple and a tee were added to the end of the header to provide for the balance pipe. The results of the test are shown in Fig. 35. The speed at the peak is about 2700 rpm and the average peak compression pressure is 107 psia. In comparing these curves with the curves of Fig. 28, it can be noted that the main effect of the balancing pipe was to increase the pressures in cylinder 4 at the high speeds so that better balance was obtained. The general shapes of the curves, the peak speeds, and the peak compression pressures are nearly identical for the two manifolds.

Fig. 36 shows the results of a test run with the inlet at the center of the balancing pipe. No length of inlet pipe was used. The results are nearly the same as with an end inlet. The peak speed and the peak compression pressures remain at about the same values.

Two tests were run with lengths of pipe connected to the center inlet on the balancing pipe. Fig. 37 shows the results of a test with a 1" x 16" inlet pipe. The addition of the pipe caused a flattening of the curves in much the same way that it did when no balance pipe was used. There was also a noticable reduction of the peak compression pressures. The average peak value is about 103 psia. The flattening of the curves is probably due to the same causes that were previously discussed in connection with the 4 cylinder manifolds without balance pipes. The lower peak value may be due to the fact that the vibration of the air in the inlet pipe, when manifold and balancing pipes of comparatively large volume are used, have a lower pressure amplitude. This action is similar to that of a soft spring which will permit considerable motion of a mass attached to it without building up a very great force. A better combination would be a stiff spring (small volume) and a large mass which would give the same natural frequency of vibration. With this combination, the pressures resulting from the vibrations would be greater.

Fig. 38 shows the results of a test run with a 3/4" x 22" pipe connected to the inlct at the center of the balancing pipe. The curves are flat and show a lower peak value (98 psia) than any combination yet used. It seems doubtful that the use of the 3/4" pipe could introduce enough friction to cause the peak to be so low because 3/4" pipe did not produce excessive frictional effects with other manifolds. The excess volume of the manifold is probably an important factor in reducing the peak value of compression pressures to this low value.

G. 3/4" Manifold With Long Sections Between Cylinders

In order to still further emphasize the effects of inertia of the masses of air in the sections of the manifold header between individual cylinder branches, a manifold was constructed with loops of pipe between cylinders as shown in Fig. 7. This manifold was tested with an end inlet, with a center inlet and with a balancing pipe.

The first of these tests was run with no inlet pipe on the end inlet. No balancing pipe was used. The results of the test are shown in Fig. 39. The most noticeable effect of the long pipes between cylinders is the extreme unbalance between cylinders. Cylinders 1 and 2 act in a similar manner and peak at a high speed (2800 rpm). Cylinders 3 and 4 act alike and peak at a low speed (1200 rpm). There is no apparent reason for this pairing of cylinders but the difference in peak speeds is reasonable because the inertia and friction of the air in the lengths of pipe from the inlet to cylinders 3 and 4 is much greater than for cylinders 1 and 2.

The results of the addition of different end inlet pipes are shown in Fig. 40 and 41. The only effect of the addition of an inlet pipe was to increase the peak compression pressure of cylinder 1. The others remainded almost the same. This manifold is extreme in design but it does illustrate the necessity for avoiding high inertia in the pipes between cylinder branches.

By moving the inlet to the center of the manifold the difference in the path lengths to the various cylinders was reduced. This, logically, should give better balance between cylinders. Fig. 42 shows the results of a test with no pipe on the center inlet. The results show that cylinders 1 and 4 peak at a low speed (1700 rpm) while cylinders 2 and 3 peak at a higher speed (2500 rpm). This difference is due to the difference in lengths of the paths from the inlets to the various cylinders. The unbalance between cylinders is less extreme than it was with the end inlet but the cylinders are still rather badly unbalanced except in the middle speed range. The highest average compression pressure is about 103 psia.

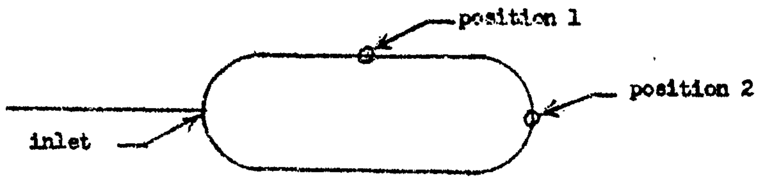
Fig. 43 shows pressure diagrams taken at the intake port for the same manifold arrangement used in obtaining the data for Fig. 42, that is, a short center intake and no balance pipe. Diagrams are shown for cylinders 1 and 2 only, because cylinders 3 and 4 are symmetrical with cylinders 1 and 2 and have similar diagrams. These diagrams indicate that the principle type of vibration is a ramming action much the same as the action of a single cylinder or dual port arrangement. The air in the manifold is apparently accelerated during the first part of each intake stroke and then it rams into the cylinder and builds up pressure at the end of the intake stroke. The amplitudes of the pressures resulting from vibrations during the periods between intake strokes are not very large and have little effect on the next intake process.

A 1" x 15-1/2" center intake pipe was added to the manifold. The results of a test run with this addition are shown in Fig. 44. The addition of the intake pipe did not have a very important effect on the shape of compression pressure curves. There is a slight flattening of the curves and the speed at the peak for cylinder 1 and 4 is shifted to a slightly lower value (1400 rpm instead of 1700 rpm). Pressure diagrams taken at the intake ports are shown in Fig. 45. If these are compared with the diagrams of Fig. 44, it may be noted that during the period of valve opening the diagrams of Fig. 45 are almost identical with those of Fig. 43, while for the period during which the valves on cylinders 1 and 2 are closed, the pressure amplitudes shown in Fig. 45 are greater than those shown in Fig. 43. This indicates that the addition of the intake pipe causes the intake strokes of cylinders 3 and 4 to produce pressure effects at the opposite end of the manifold and also that there may be some pressure effects due to the vibration of the air in the intake pipe. The interaction between cylinders on opposite ends exists to a greater extent because, with a very short intake pipe the pressure at the center of the manifold would always be very close to atmospheric and therefore the two ends of the manifold would be isolated from each other, while with a longer intake pipe the inertia of the air in the intake would prevent atmospheric pressure from being maintained at the center of the manifold. It is apparent from the compression pressure curves that neither interaction between the ends of the manifold nor the vibration of the air in the intake pipe produced a very important effect with this particular manifold. The 4 cylinder manifolds discussed earlier did show important effects as a result of vibrations in the intake pipe.

Diagrams of the pressures at the tees where the branch pipes connect to the header were also obtained. These diagrams are shown in Fig. 46. The shapes of the pressure diagrams obtained at the tees are quite similar to the shapes of the diagrams obtained at the ports. This indicates that the air in the header produces a ramming action much the same as that produced by the air in the branch pipes.

A balancing pipe was added to the manifold as shown by the dashed lines in Fig. 7. The balancing pipe made it possible for the air to flow by two paths to each cylinder. The average length of the two paths was the same for all cylinders so that the cylinders should act more nearly alike than they did without the balancing pipe. Fig. 47 shows compression pressures that were obtained with the balancing pipe added. The 1" x 15-1/2" center inlet that had been used previously was used for this test. The compression pressures show that the cylinders are quite similar in their action except at the high speeds.

At high speed cylinders 2 and 3 have higher compression pressures than cylinders 1 and 4. This difference is perhaps due to a difference in effective inertia and friction of the flow paths to these cylinders. This can be better demonstrated with the air of the following sketch:



Let the total resistance of the ring be equal to 4. If position 1 is one quarter of the way around the ring from the inlet, then the resistance of the two parallel paths from the inlet to position 1 would be 1 and 3. The effective resistance of the paths combined would be

$$R = \frac{1}{\frac{1}{1} + \frac{1}{3}} = \frac{3}{4}$$

For position 2 the effective resistance would be

$$R = \frac{1}{\frac{1}{2} + \frac{1}{2}} = 1$$

The inertia effects for the two positions would be in the same proportion as the resistances so that it is apparent that even though the averages of the two path lengths to each cylinder are the same the inertia and frictional effects are not the same. The balancing pipe, then did not make the cylinders completely similar althogh it did help as can be seen by comparing Fig. 43 with Fig. 40. In addition to making the cylinders more nearly alike, the balancing pipe also made the compression pressure curves flatter and made all of the compression pressures higher at the high speeds. The general character of the curves is similar to that of the curves obtained with the manifolds having short lengths of header between cylinders.

Fig. 48 shows pressure diagrams obtained at the tees where the branch pipes connect to the header. The diagrams for 1000 rpm indicate that there was a vibration in header that was excited by the intake strokes of all of the cylinders. The pressure waves are approximately sinusoidal. The pressure waves are probably the result of the vibration of the mass of air in the intake pipe against the volume of the manifold and cylinders. At 1400 rpm this type of vibration still seems to be the cause of tmost of the pressure variations. At higher speeds this type of vibration shows some decrease in amplitude while a ramming action made evident by the rather sharp peaks near the time of valve closing becomes more important.

Fig. 49 shows pressure diagrams obtained at the ports. At 1000 rpm the sinusoidal wave form is evident. At higher speeds, the ramming action produces the principal pressure effect. The fact that some of the ramming action was also evident in Fig. 48 indicates that the air in the header takes part in the ramming along with the air in the branch pipes.

VI. CONCLUSIONS

Although a rather large quantity of data has been presented, the analysis of the data has not been completed and only the more evident facts can be stated as conclusions in this report. Much of the data indicates that many simplifying assumptions may be made in setting up theories that will make possible the calculation of the

the general characteristics such as speed for peak volumetric efficiency, of rather complicated intake systems.

Conclusions that have been drawn from the data presented in this report are given separtely for each manifold type in the following paragraphs:

<u>Single Cylinder Manifolds</u>

1. Individual pipes for each cylinder gave higher volumetric efficiencies than any of the other manifolds tested.

2. The high volumetric efficiencies were the result of a ramming action rather than a resonant vibration. Only with very long pipes is there an important effect produced by vibrations carried over from one cycle to the next.

3. The speed at which the peak volumetric efficiency will occur can be calculated from Eq. 7 given in this report.

<u>Dual Port Manifolds</u>

1. The action of most dual port arrangements is the same as for single cylinder manifolds. The principal factor causing high volumetric efficiencies is the ramming action of the air in the intake pipe.

2. When the intake strokes of the two cylinders on the dual port are evenly spaced with respect to time, the two cylinders have equal volumetric efficiencies at all speeds.

3. The speed at which the peak volumetric efficiency occurs can be calculated from the same equation used for single cylinder manifolds.

4. When the two cylinders fire in such a sequence that there is an overlap of the valve opening periods, the cylinder with the valve

closing while the other is open will receive a reduced amount of charge. This is due to the fact that the air may ram into both cylinders instead of just one; therefore, the pressure built up due to the ram is reduced.

5. Volumetric efficiencies obtained with a dual port arrangement were about 5% lower than those obtained with single cylinder manifolds due to a greater amount of friction and extra volume at the cylinder end of the pipe.

<u>Four Cylinder Manifolds</u>

1. Two types of vibration produce important effects on the volumetric efficiency. These are: (1) ramming, and (2) a vibration of a system consisting of the mass in the inlet pipe to the manifold header and the spring action of the manifold and cylinder volume.

2. The speed at which ramming produces its maximum effect is determined by the factors given in Eq. 7. Accurate calculations of this speed are difficult to make because the amount of ramming action produced by the air in the manifold is usually difficult to determine.

3. The speed at which maximum benefits are obtained from the second type of vibration given above is usually slightly above the resonant speed for the vibrating system. The resonant speed can be calculated from Eq. 7. The frequency of the driving force for this type of vibration is twice the rpm, since two intake strokes occur each revolution.

4. The combined effects of the two types of vibration may produce a nearly flat volumetric efficiency curve.

5. The highest peak value of volumetric efficiency should be obtained when these two types of vibration are synchronized to

produce maximum effects at the same speed; however, the data is not complete enough to give conclusive evidence of this.

6. On manifolds with short, low inertia sections of header between the branch pipes to the cylinders, the balance between cylinders is good regardless of the position of the inlet to the header. Balance pipes are unnecessary.

7. The addition of a balance pipe connecting the ends of the header generally caused a decrease in volumetric efficiency. This was probably due to a decrease in the pressure amplitude of the second type of vibration which results from an increase in manifold volume.

8. On manifolds with long (high inertia) section of header between the branch pipes, the cylinders were badly out of balance with all inlet positions used. An inlet at the center of the manifold header gave the best results.

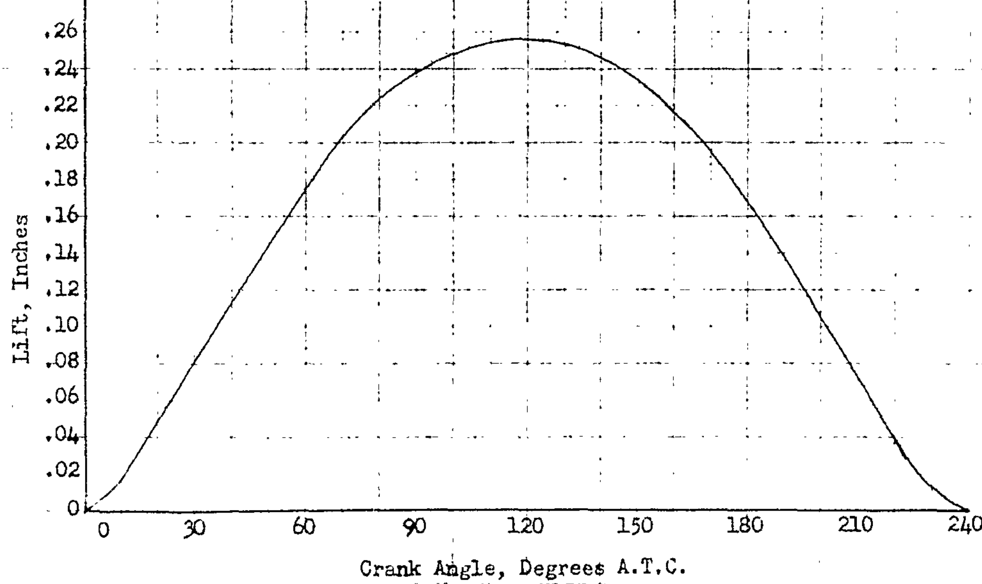
9. A balance pipe added to this manifold improved the balance between cylinders.

10. The balance pipe in this case improved the volumetric efficiency at high speeds, because it reduced the effective friction and inertia of the paths to the cylinders and shifted the speed for peak ramming effect to a higher value.

Fig. 1. Engine Data

Type - 4 cylinder, 4 cycle, air cooled.
Displacement - 20 in.^{3} per cylinder
Intake port diameter - 1 1/8"
Intake port length - 2"
Firing order - 1-3-4-2
Compression ratio - 4.4
Intake opens - T.C.
Intake closes - 60° A.B.C.
Exhaust opens - 40° B.B.C.
Exhaust closes - 15° A.T.C.

Intake Valve Lift Curve



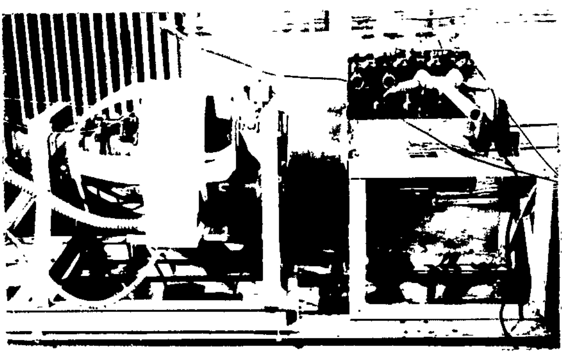


Fig. 2. Engine and Dynamometer

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FIG. 3
MANIFOLD NO. 1

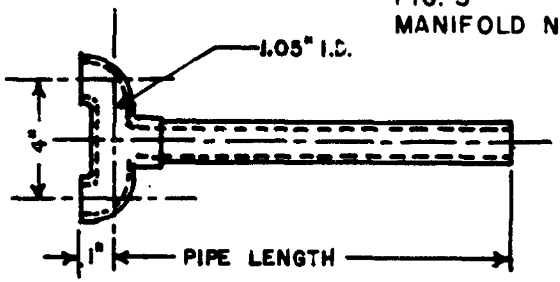


FIG. 4
MANIFOLD NO. 2

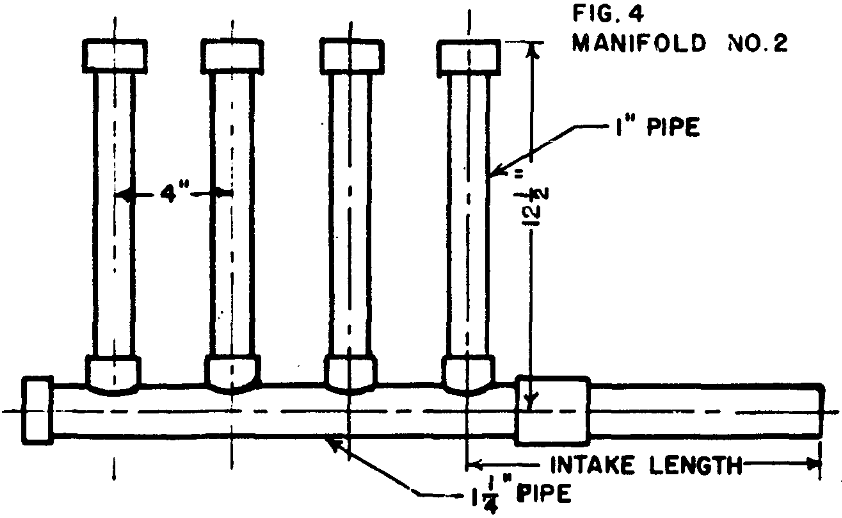


FIG. 5
MANIFOLD NO. 3

FIG. 6
MANIFOLD NO.4

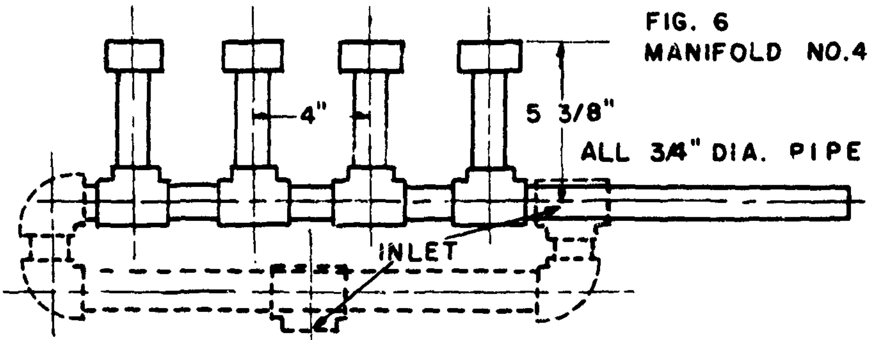
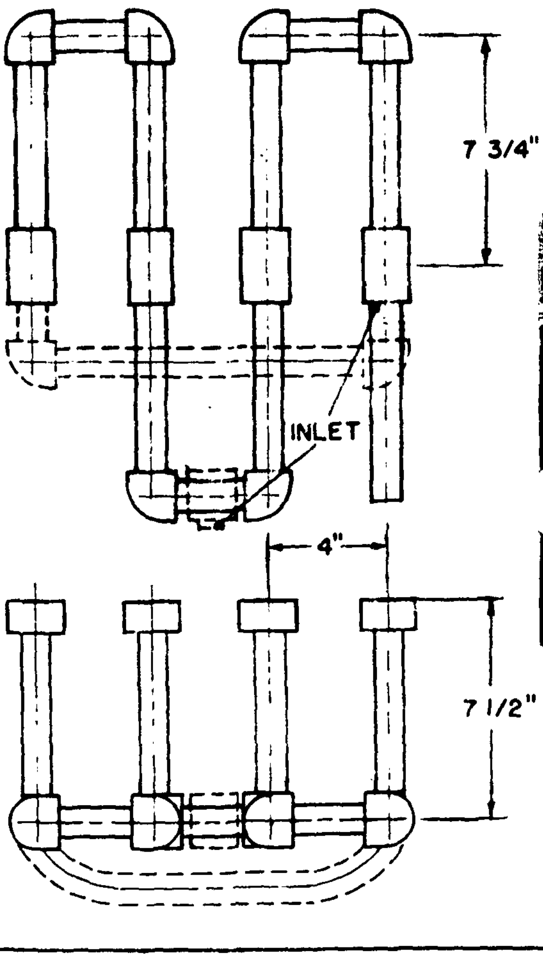
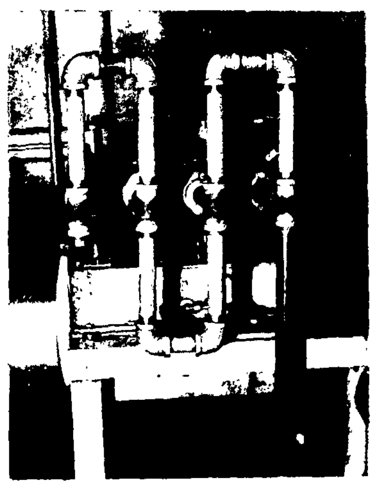


FIG.7
MANIFOLD NO.5





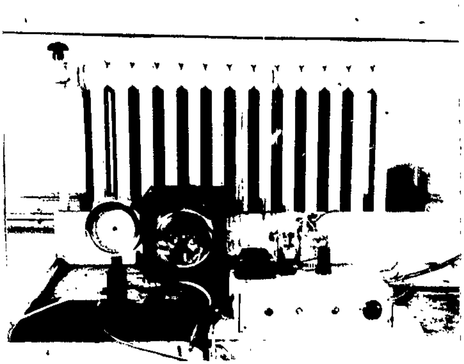


Fig. 8. Balanced Pressure Gage Controls

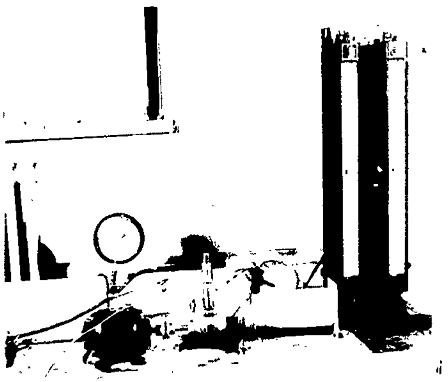


Fig. 9. Apparatus for Pressure Pick-Up Calibration

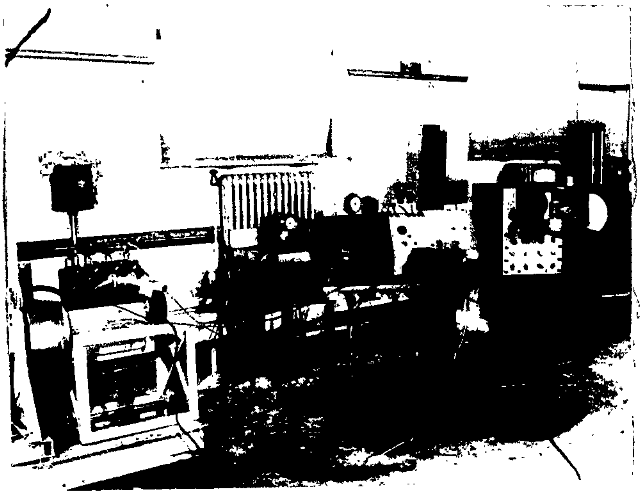


Fig. 10. Experiment Apparatus

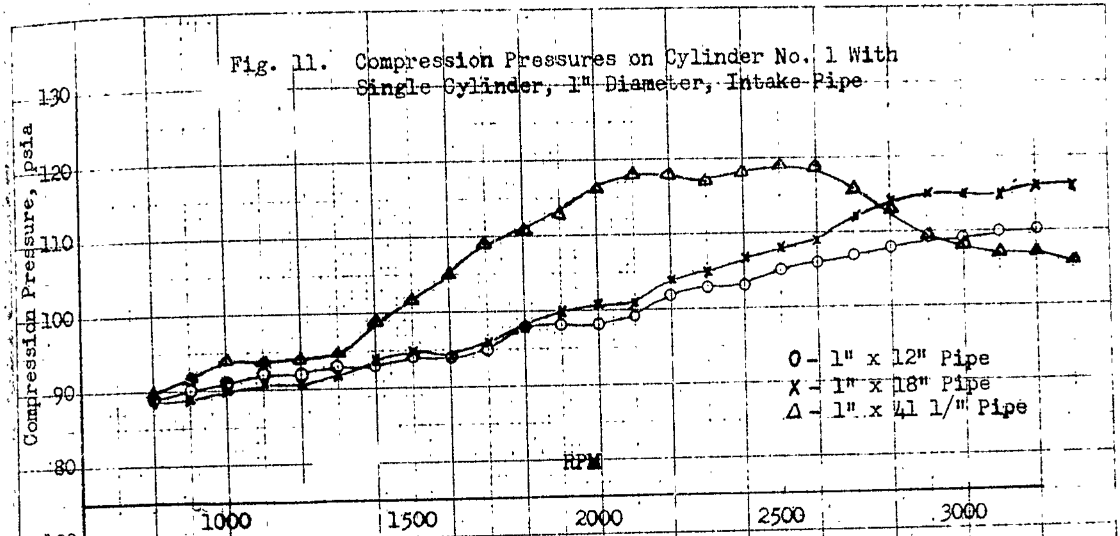


Fig. 11. Compression Pressures on Cylinder No. 1 With Single Cylinder, 1" Diameter, Intake Pipe

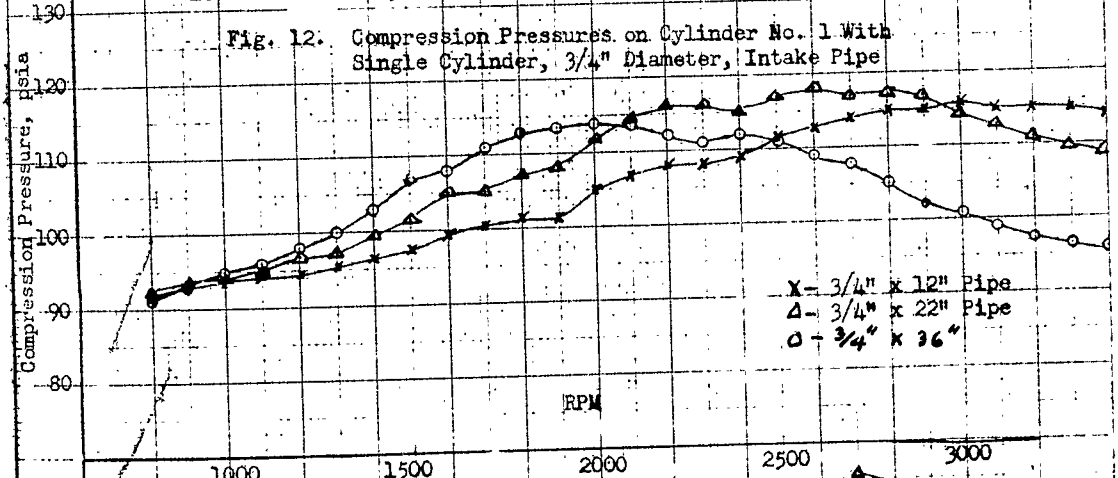


Fig. 12. Compression Pressures on Cylinder No. 1 With Single Cylinder, 3/4" Diameter, Intake Pipe

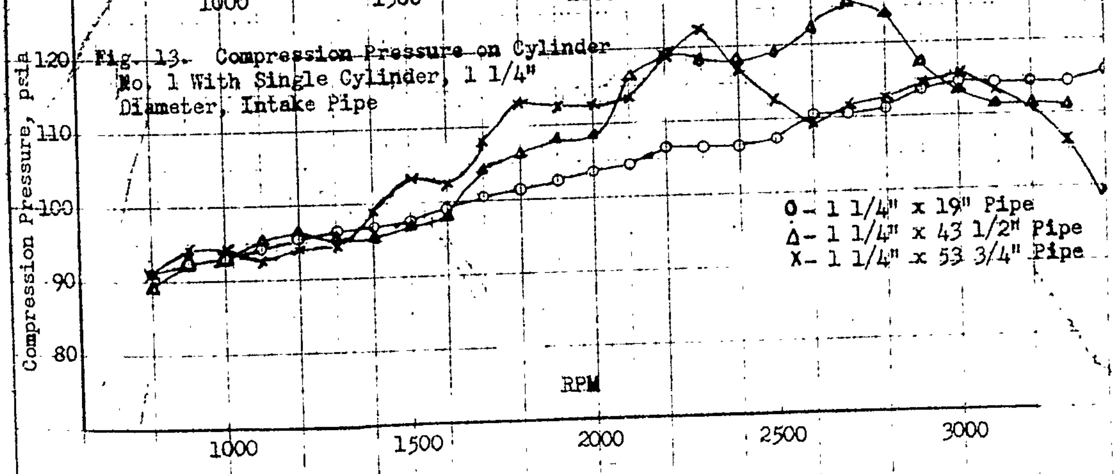


Fig. 13. Compression Pressure on Cylinder No. 1 With Single Cylinder, 1 1/4" Diameter, Intake Pipe

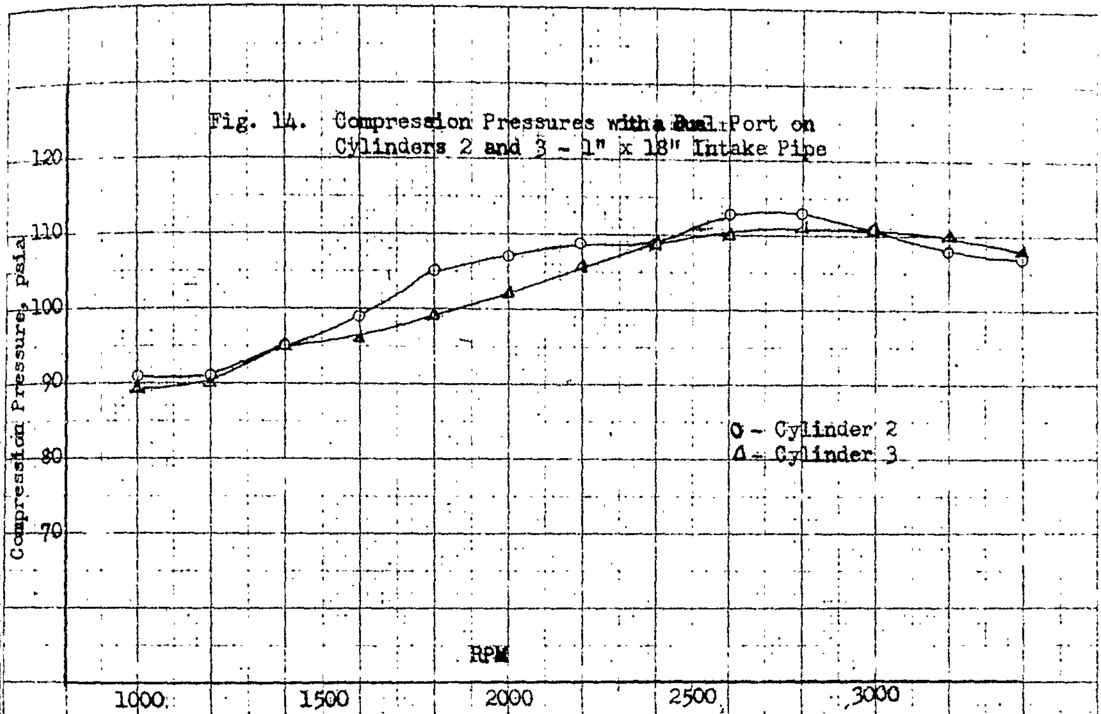


Fig. 14. Compression Pressures with a Dual Port on Cylinders 2 and 3 – 1" x 18" Intake Pipe

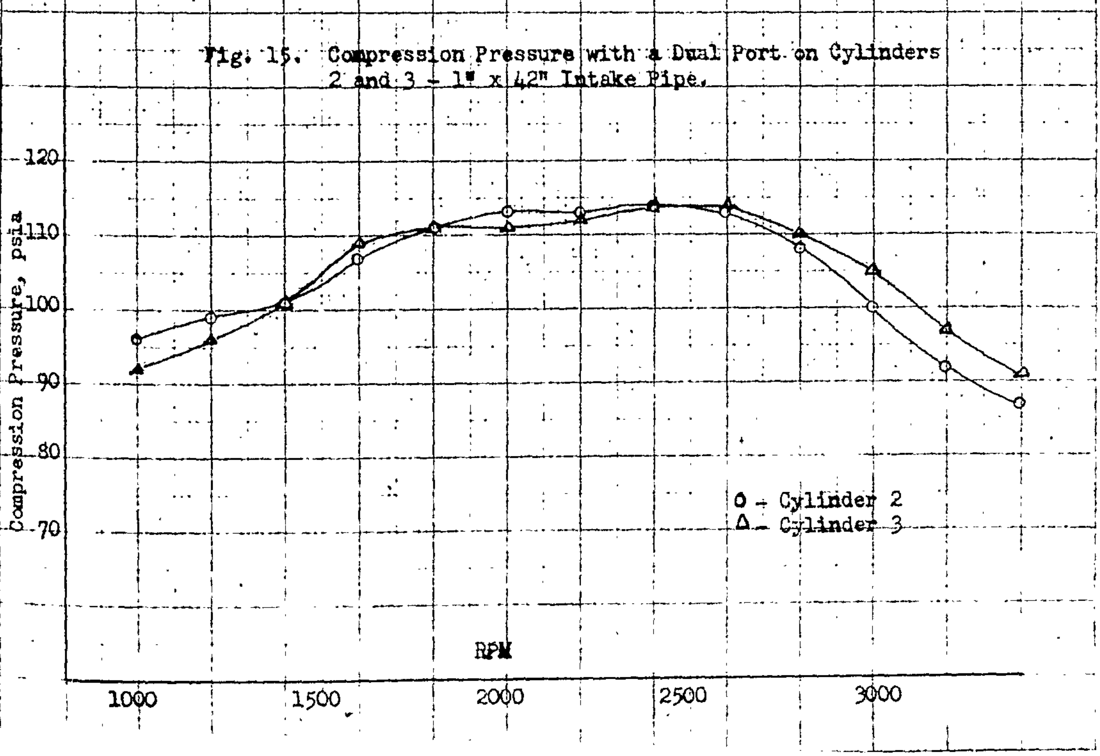


Fig. 15. Compression Pressure with a Dual Port on Cylinders 2 and 3 – 1" x 42" Intake Pipe.

Fig. 16. Compression Pressures with Dual Port on Cylinders 1 and 2 - 1" x 42" Intake Pipe

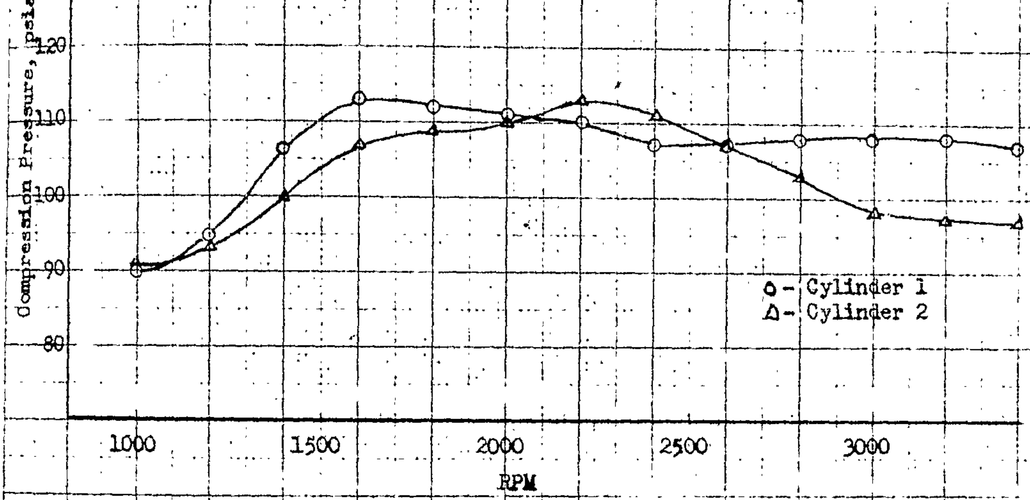
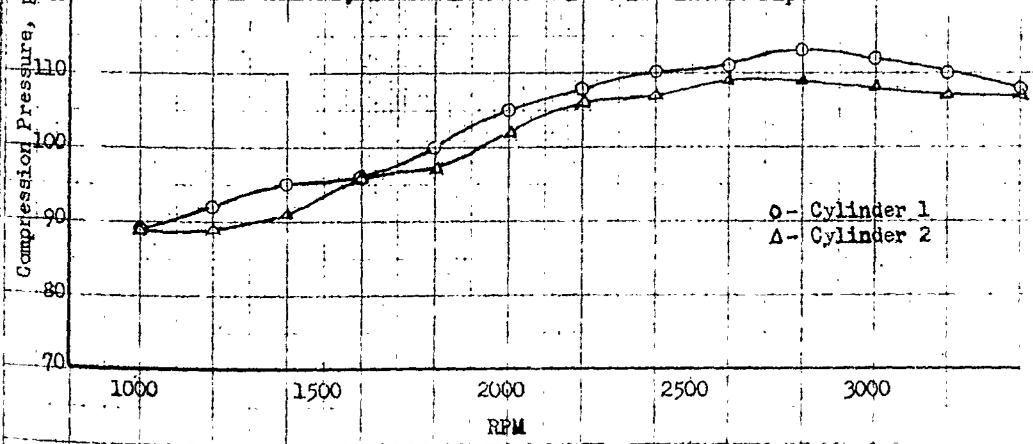


Fig. 17. Compression Pressures with Dual Port on Cylinders 1 and 2 - 1" x 18" Intake Pipe



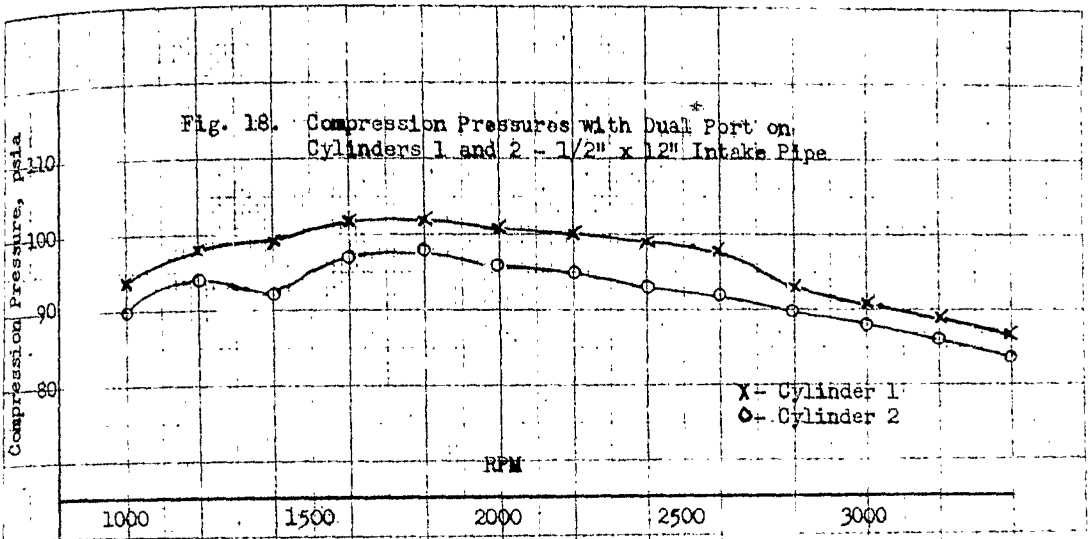


Fig. 18. Compression Pressures with Dual Port on Cylinders 1 and 2 - 1/2" x 12" Intake Pipe

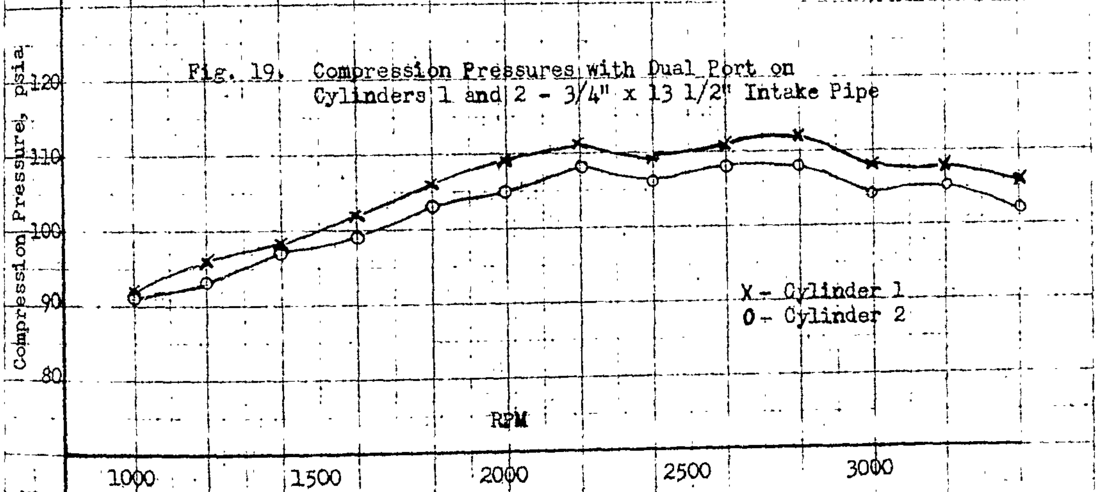


Fig. 19. Compression Pressures with Dual Port on Cylinders 1 and 2 - 3/4" x 13 1/2" Intake Pipe

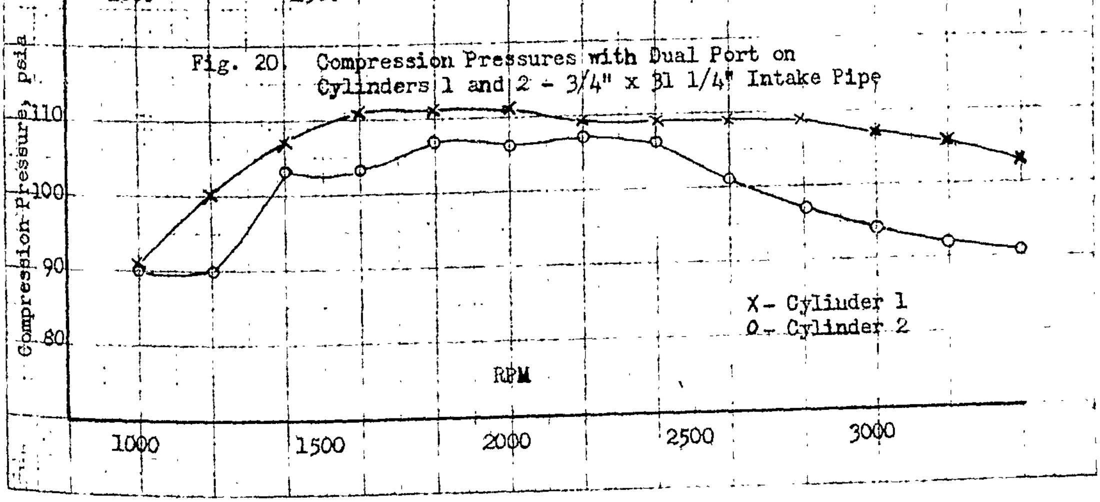


Fig. 20. Compression Pressures with Dual Port on Cylinders 1 and 2 - 3/4" x 31 1/4" Intake Pipe

FIG. 21 PRESSURE VARIATION AT INTAKE PORT WITH MANIFOLD NO. 1 (FIG. 3)
1/2" X 12" INTAKE PIPE

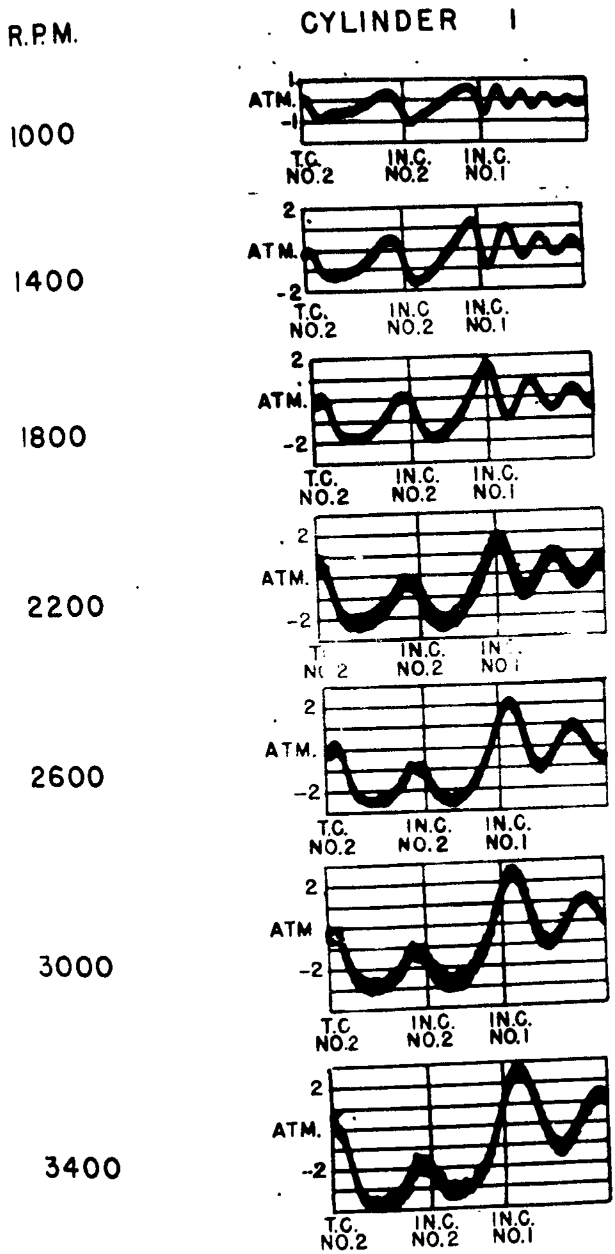


FIG. 22 PRESSURE VARIATION AT INTAKE PORT WITH MANIFOLD NO. 1 (FIG. 3) 3/4" X 13 1/2" INTAKE PIPE

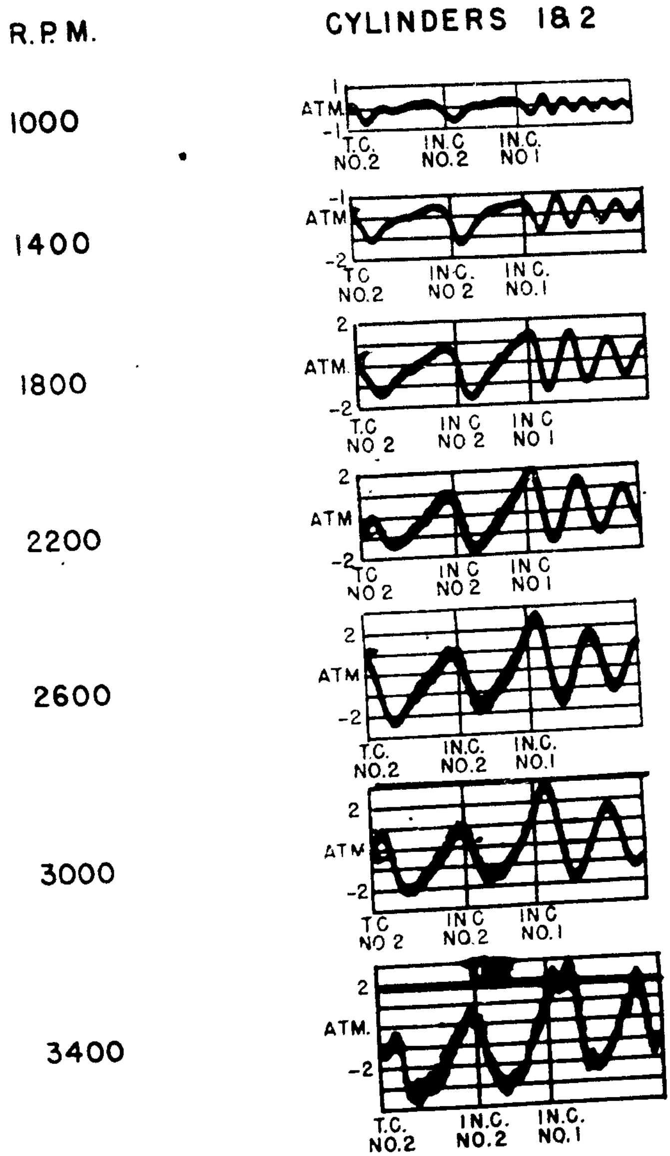
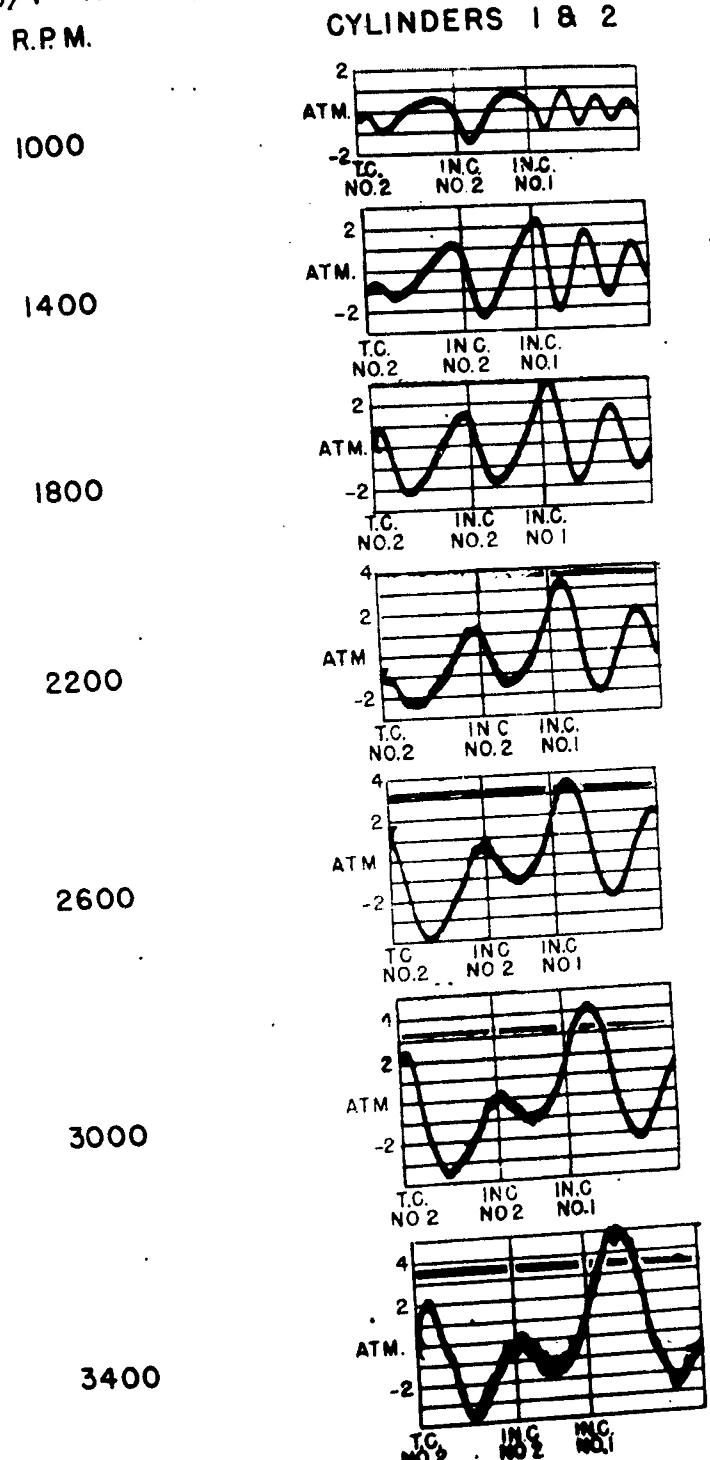


FIG.23 PRESSURE VARIATION AT INTAKE PORT WITH MANIFOLD NO. 1 (FIG. 3)
3/4" X 31 1/4" INTAKE PIPE



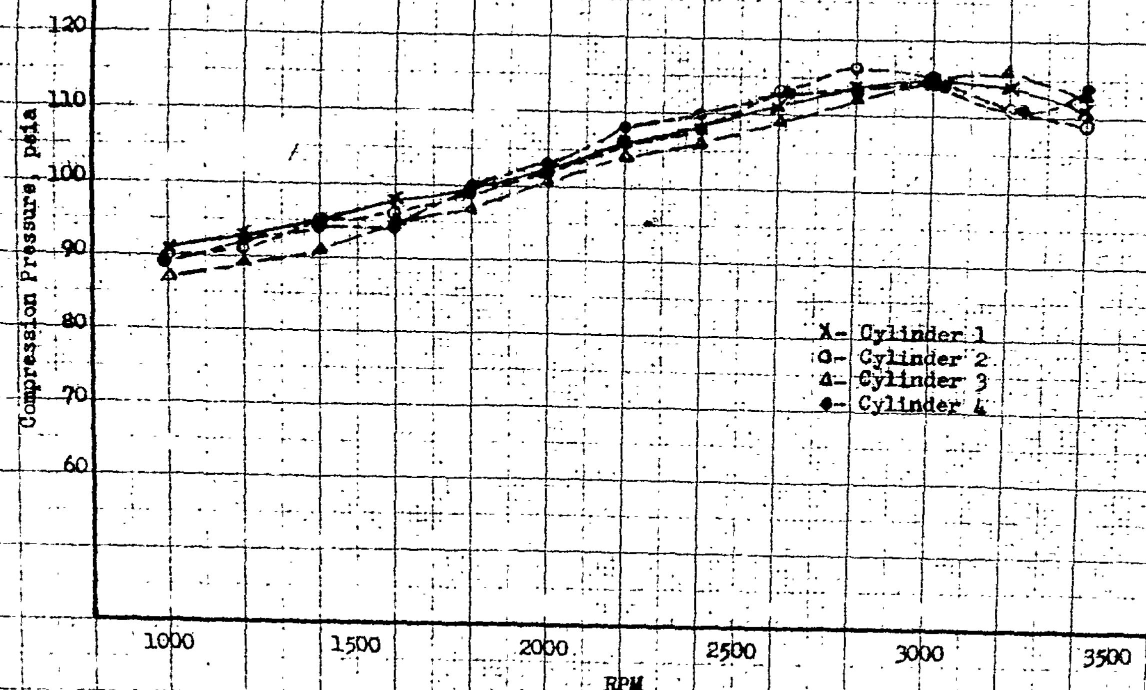
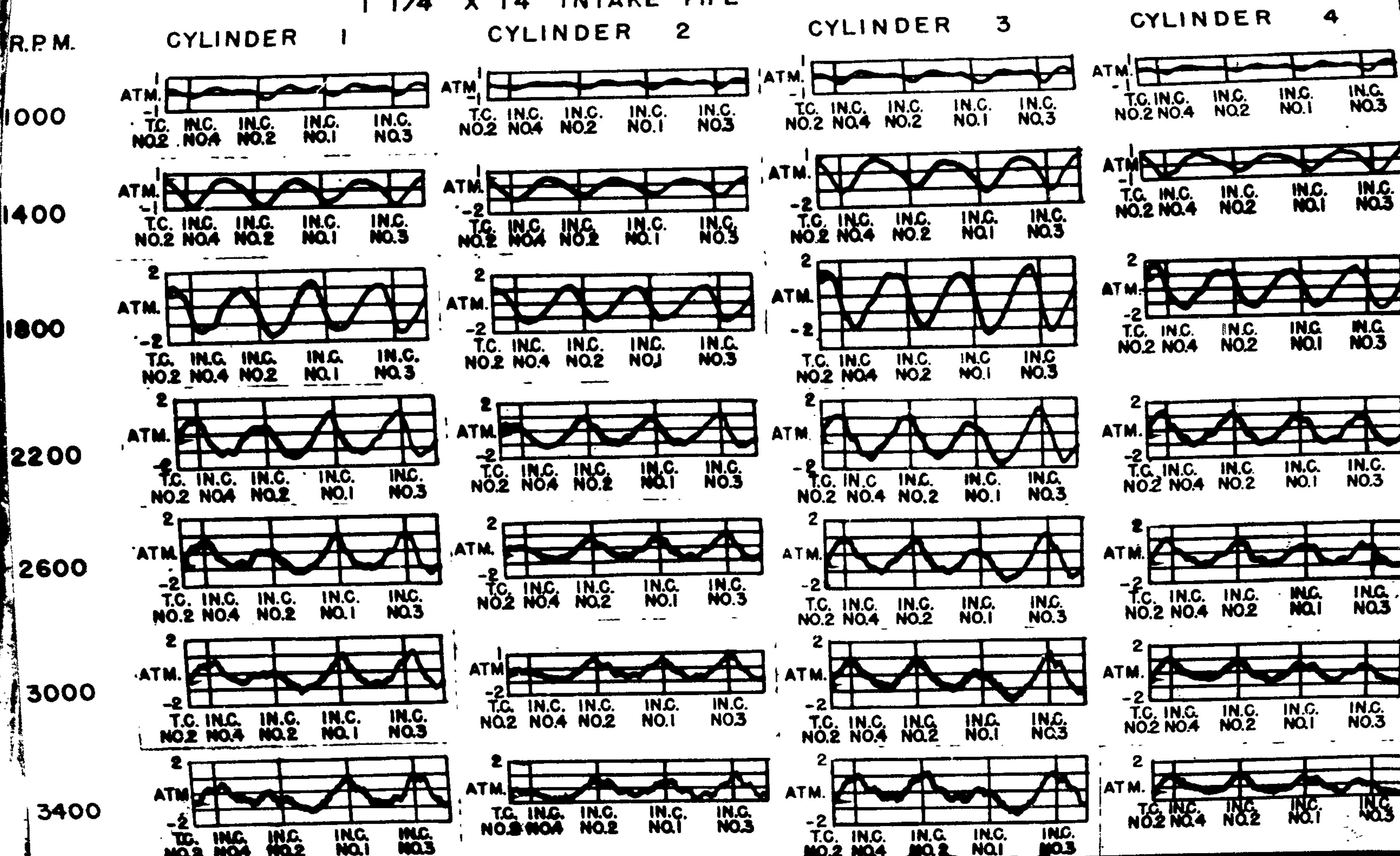


Fig. 24. Compression Pressures with Four Cylinder Large Diameter, Rake Type Manifold End Inlet, No Pipe Attached (See Fig. 4)

FIG. 26 PRESSURE VARIATION AT INTAKE PORT WITH MANIFOLD NO. 2 (FIG. 4) 1 1/4" X 14" INTAKE PIPE



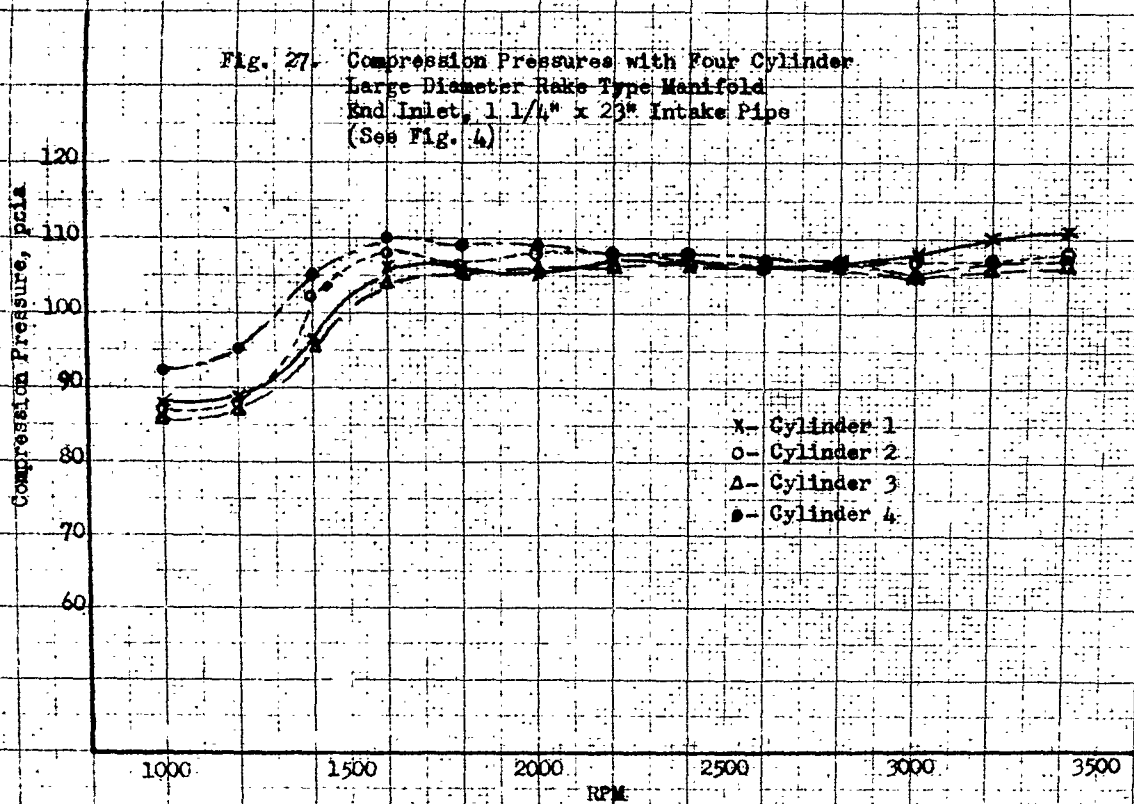


Fig. 27. Compression Pressures with Four Cylinder Large Diameter Rake Type Manifold End Inlet, 1 1/4" x 23" Intake Pipe (See Fig. 4)

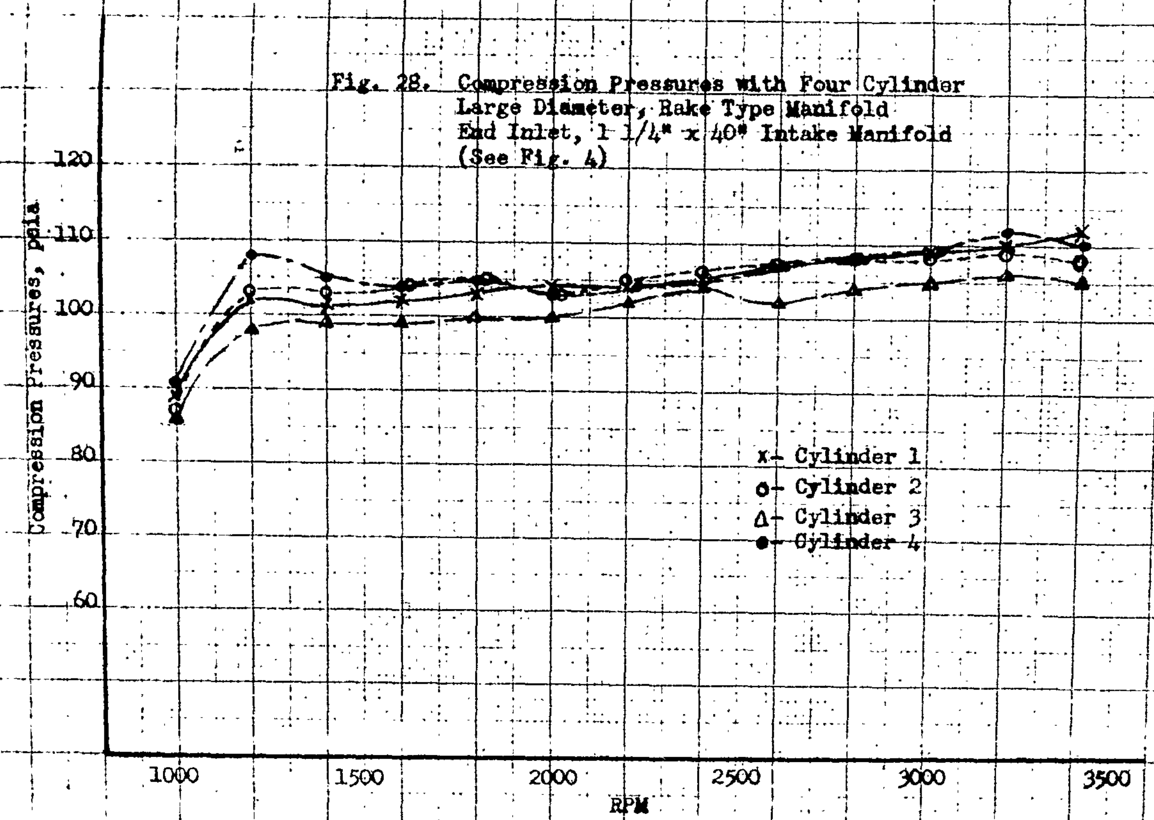


Fig. 28. Compression Pressures with Four Cylinder Large Diameter, Rake Type Manifold End Inlet, 1 1/4" x 40" Intake Manifold (See Fig. 4)

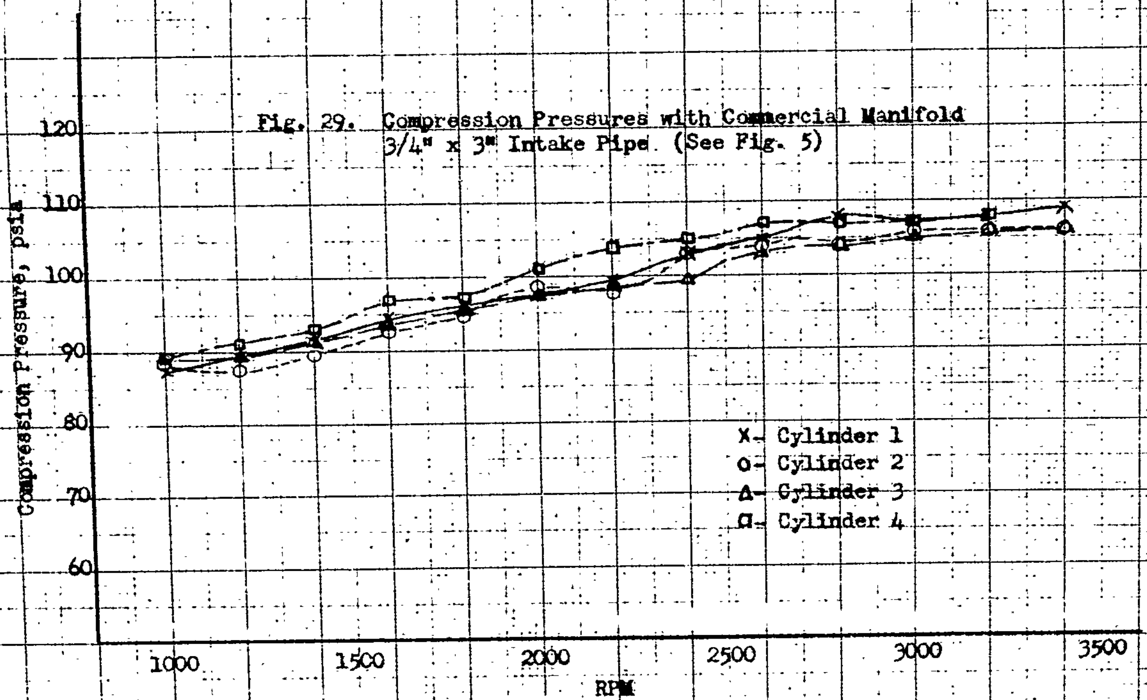


Fig. 29. Compression Pressures with Commercial Manifold 3/4" x 3" Intake Pipe. (See Fig. 5)

Fig. 30. Compression Pressures with Commercial Manifold
1" x 18" Intake Pipe (See Fig. 5)

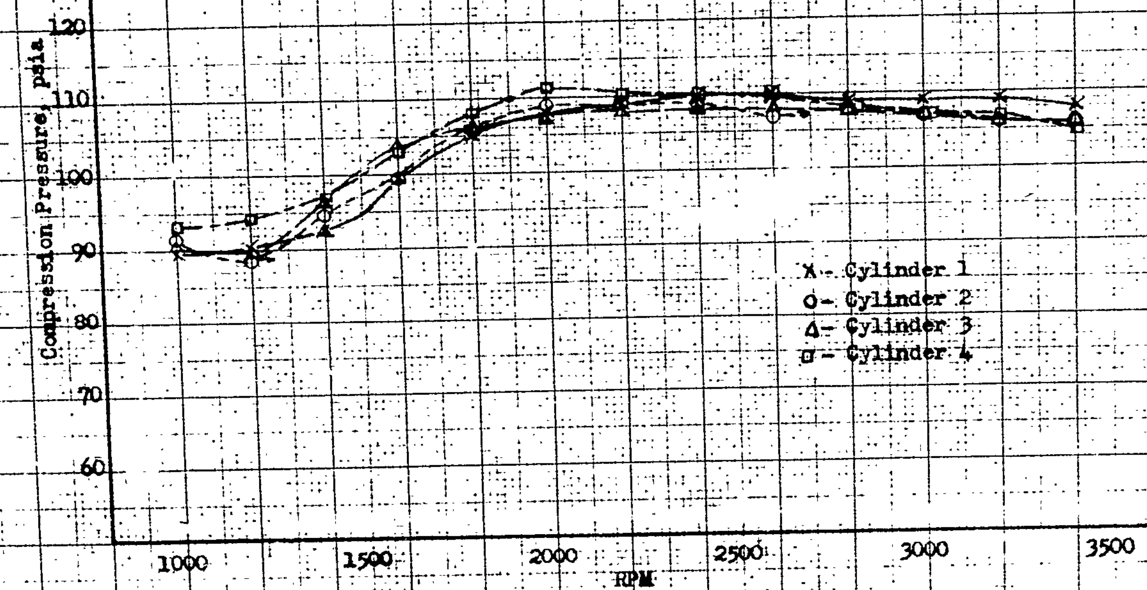
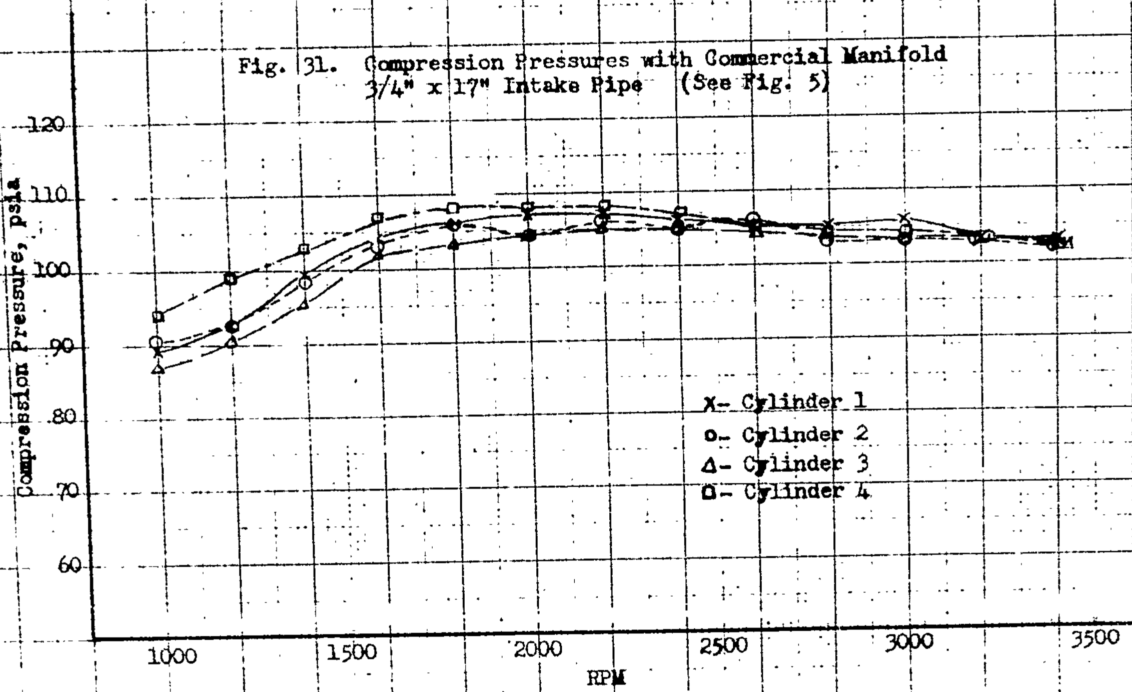


Fig. 31. Compression Pressures with Commercial Manifold
3/4" x 17" Intake Pipe (See Fig. 5)



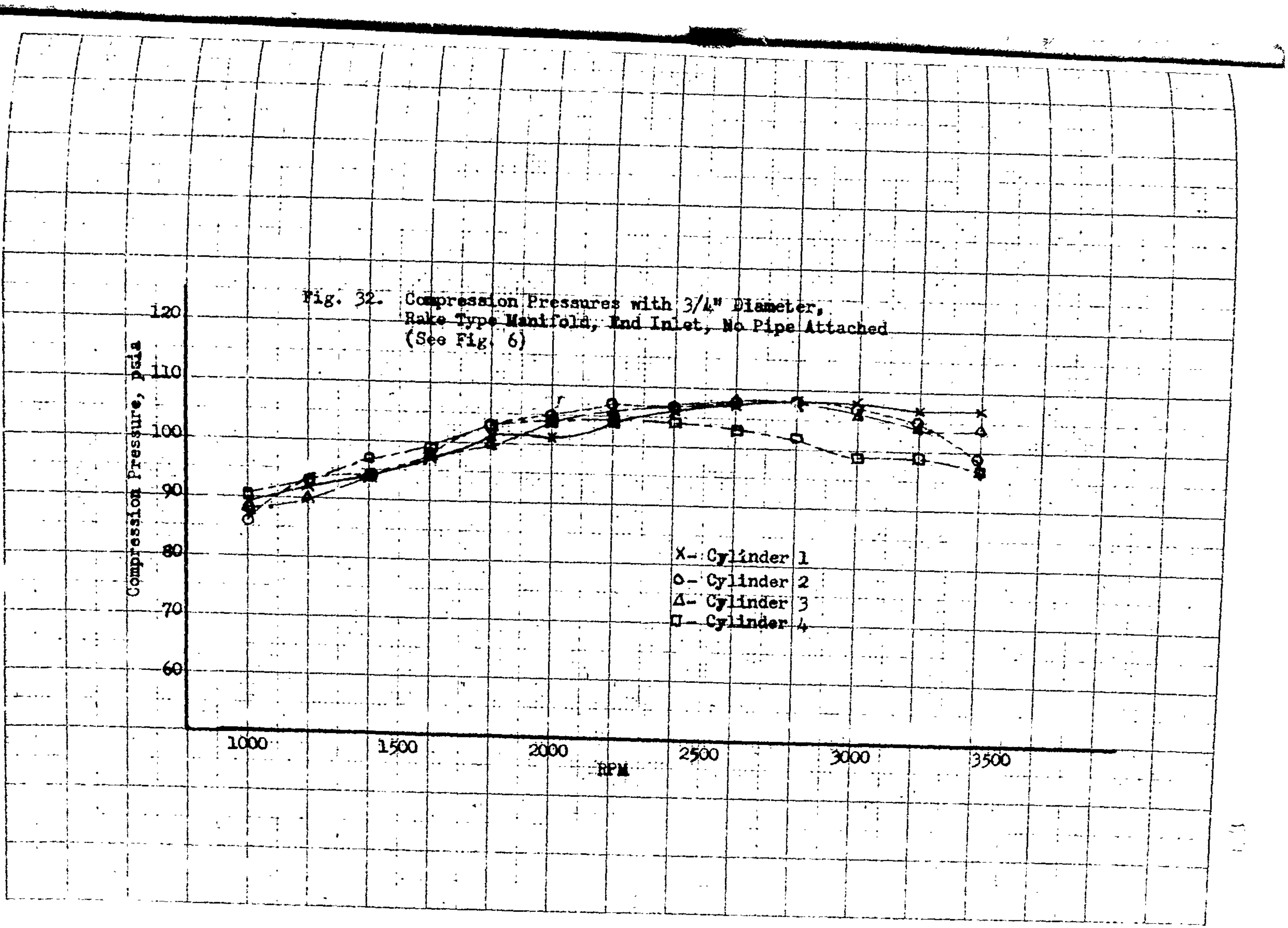


Fig. 32. Compression Pressures with 3/4" Diameter, Rake Type Manifold, End Inlet, No Pipe Attached (See Fig. 6)

Fig. 33. Compression Pressures with 3/4" Diameter, Rake Type Manifold, End Inlet, 1" x 22" Intake Pipe (See Fig. 6)

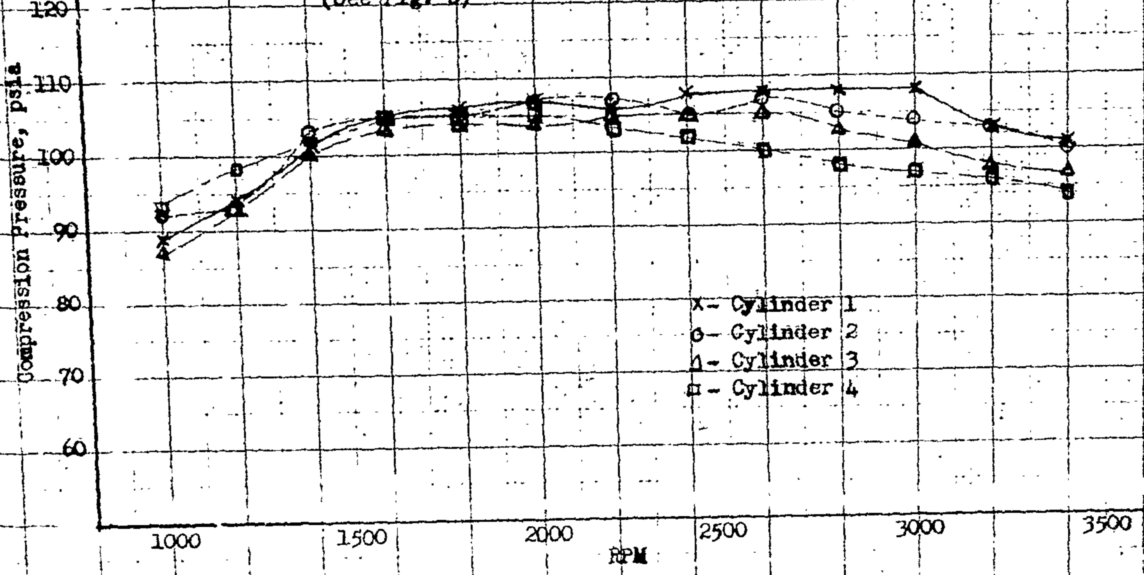
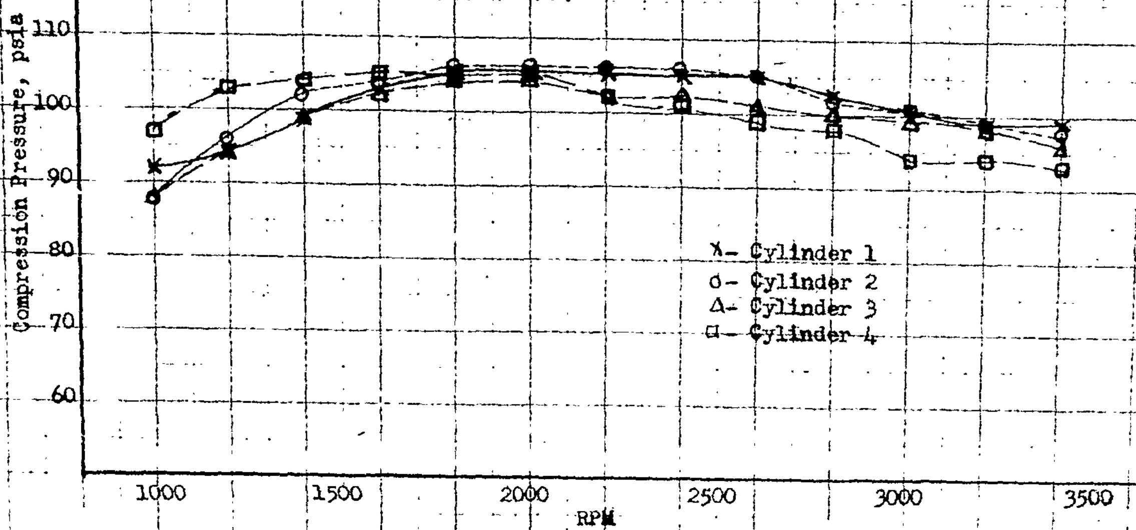


Fig. 34. Compression Pressures with 3/4" Diameter Rake Type Manifold, End Inlet, 3/4" x 16" Intake Pipe — (See Fig. 6)



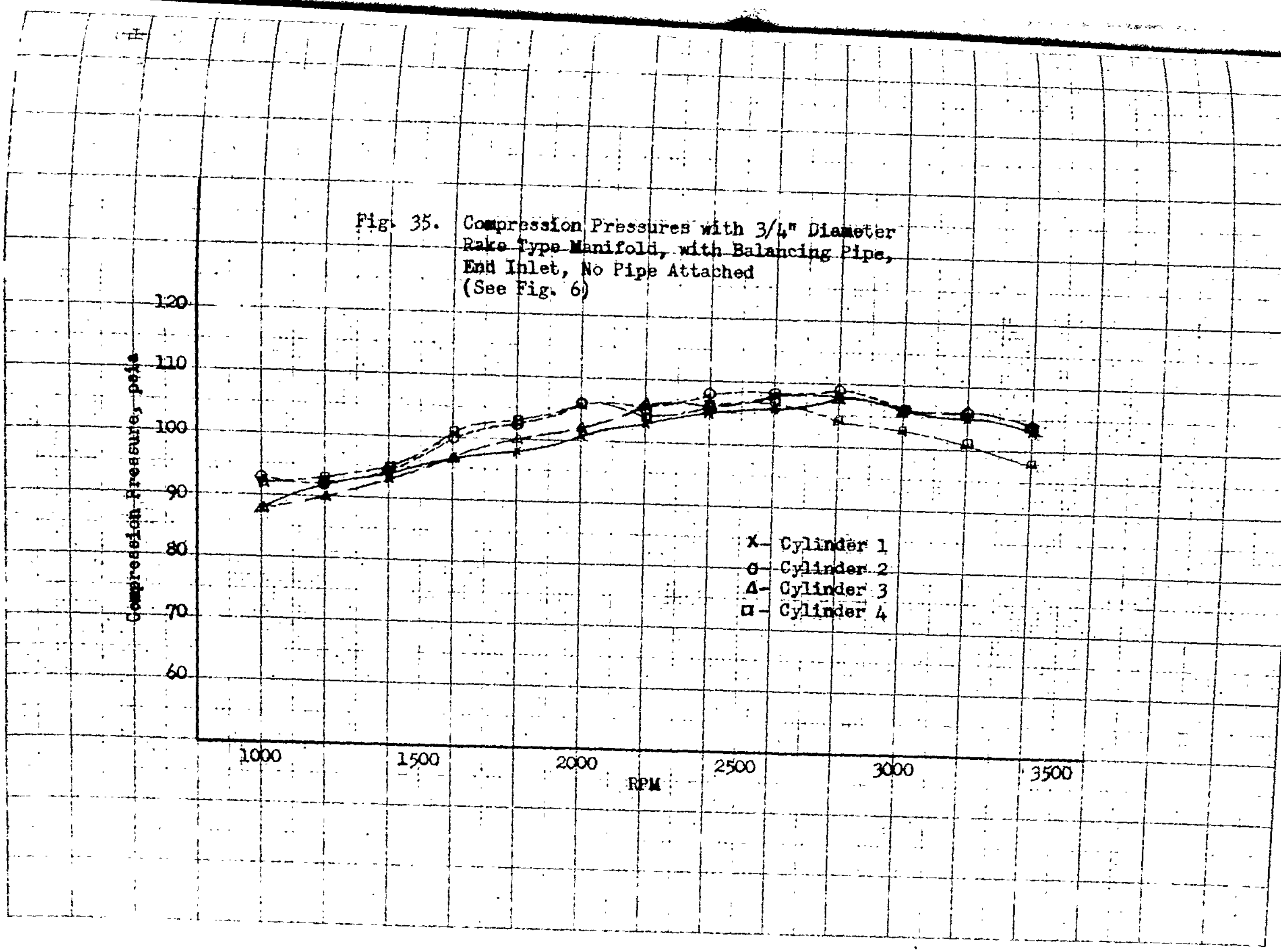


Fig. 35. Compression Pressures with 3/4" Diameter Rake Type Manifold, with Balancing Pipe, End Inlet, No Pipe Attached (See Fig. 6)
120
110
100
90
80
70
60
Compression Pressure, psia
1000
1500
2000
2500
3000
3500
RPM
X – Cylinder 1
O – Cylinder 2
Δ – Cylinder 3
□ – Cylinder 4

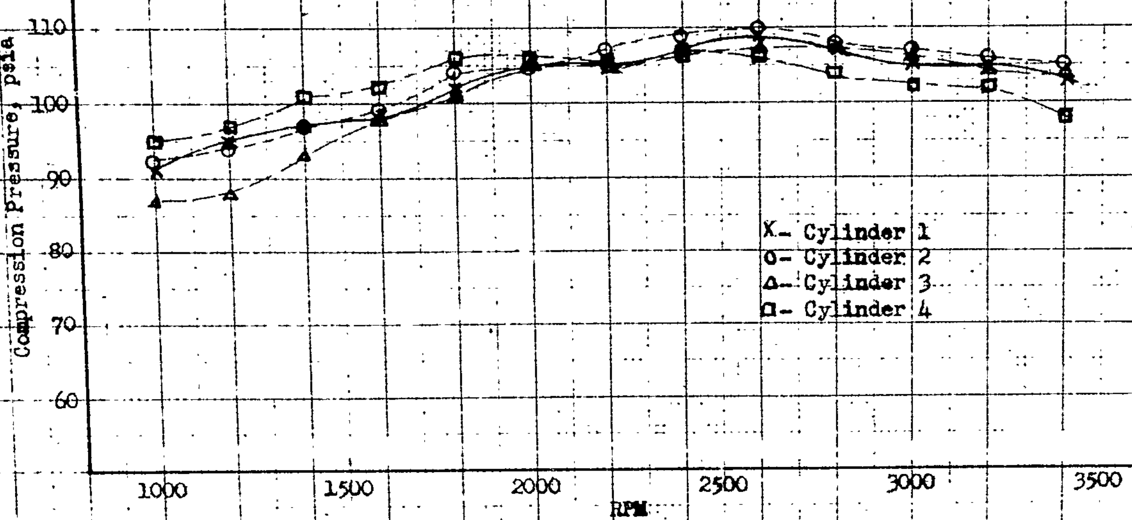
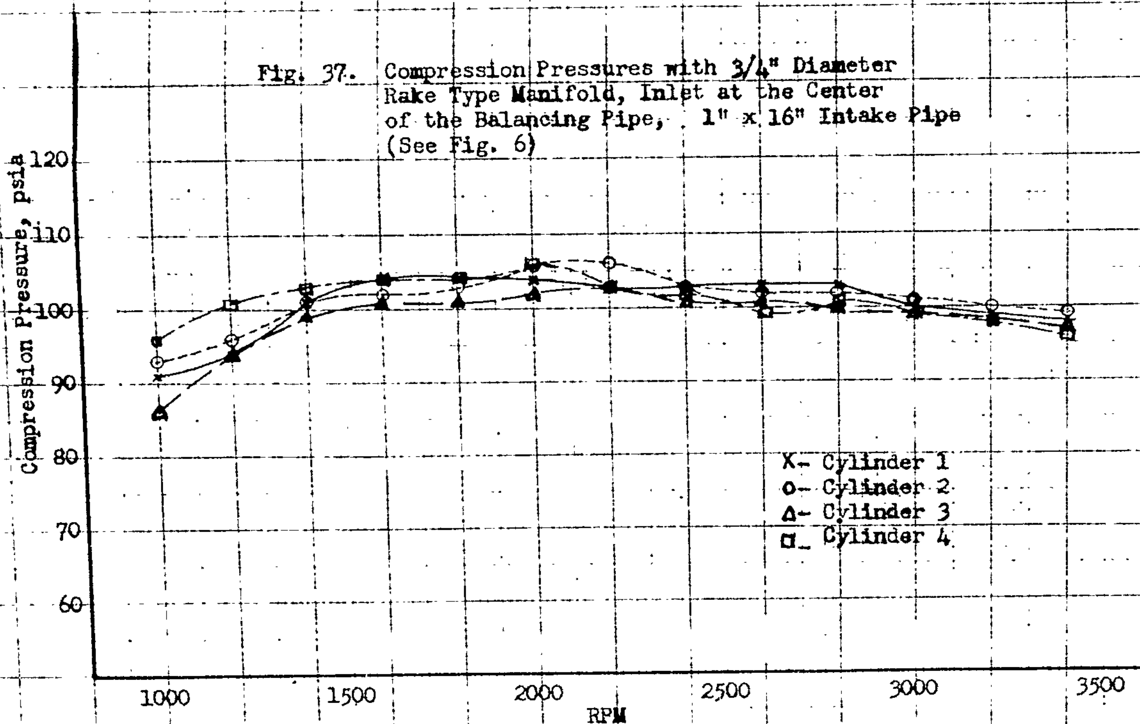


Fig. 36. Compression Pressures with 3/4" Diameter, Rake Type Manifold, with Inlet at the Center of the Balancing Pipe, No Intake Pipe Attached. (See Fig. 6)

Fig. 37. Compression Pressures with 3/4" Diameter Rake Type Manifold, Inlet at the Center of the Balancing Pipe, 1" x 16" Intake Pipe (See Fig. 6)



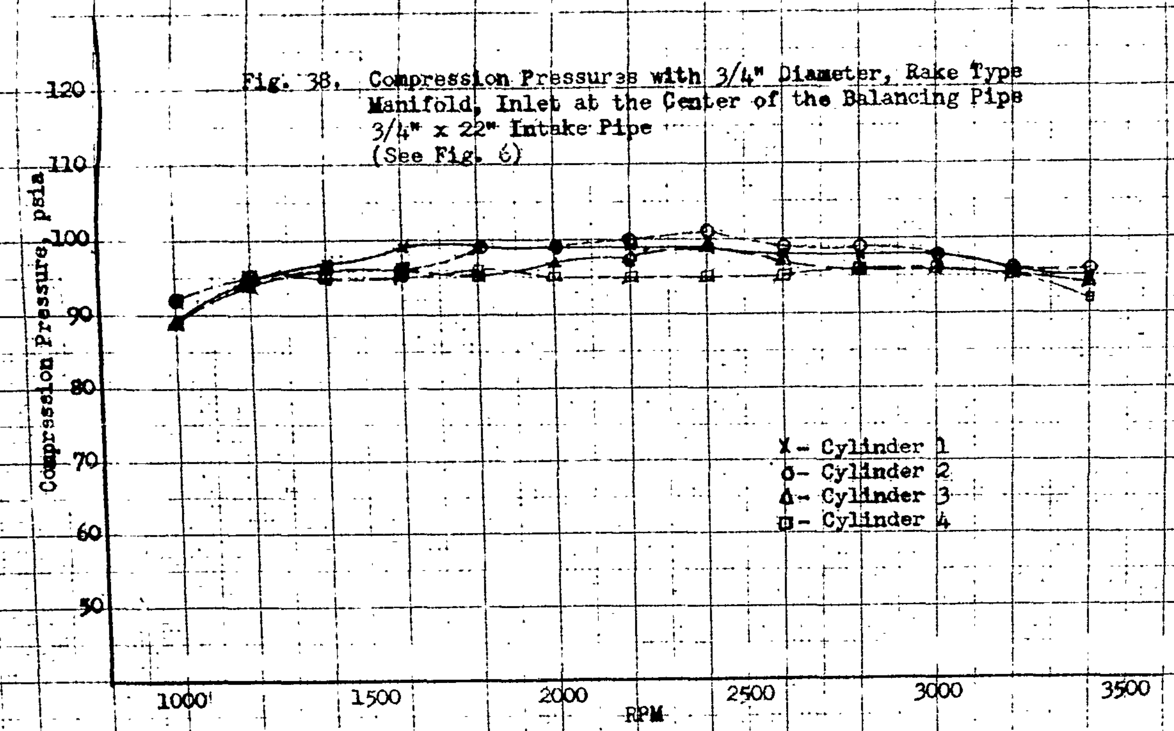


Fig. 38. Compression Pressures with 3/4" Diameter, Rake Type Manifold, Inlet at the Center of the Balancing Pipe 3/4" x 22" Intake Pipe (See Fig. 6)

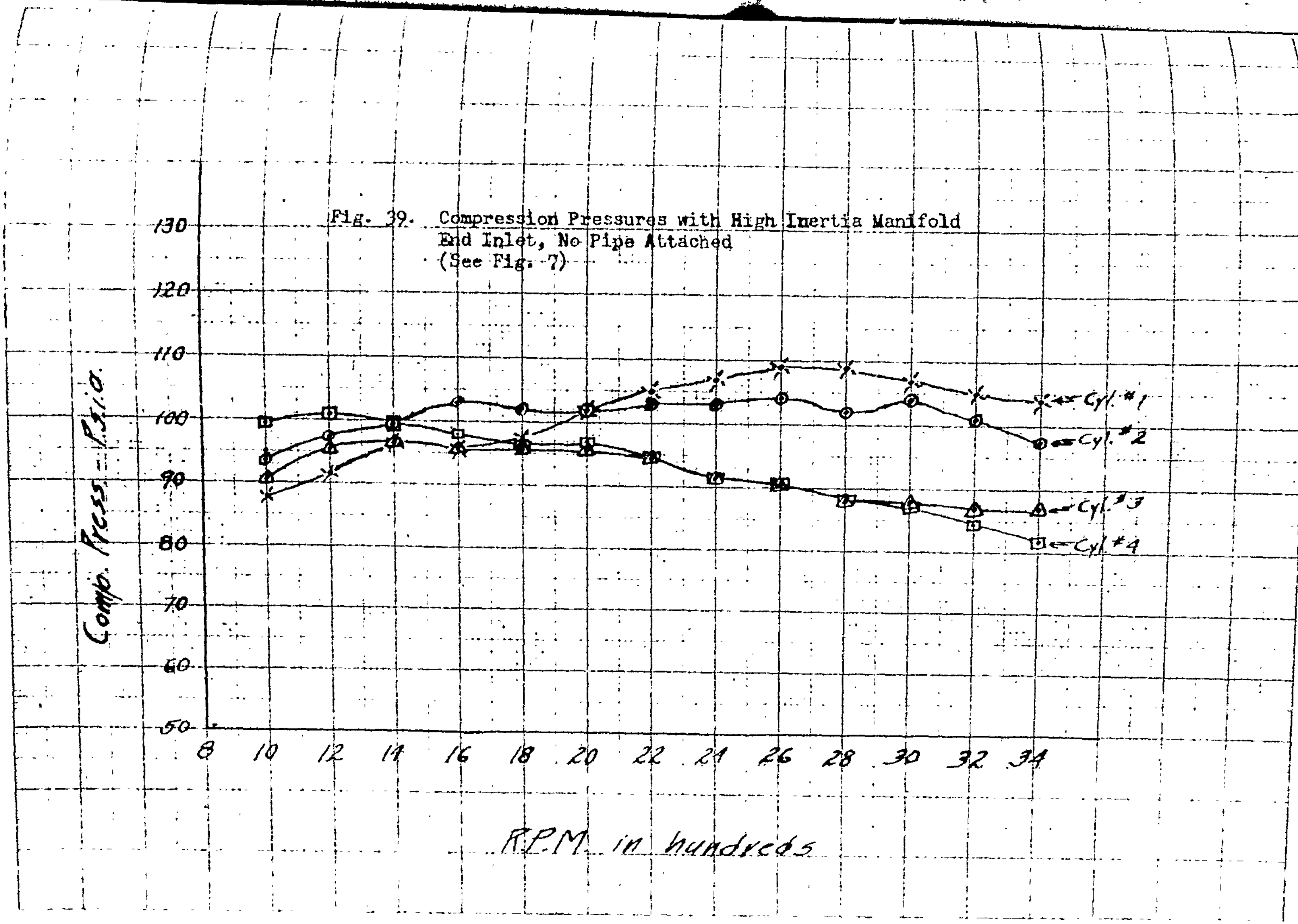
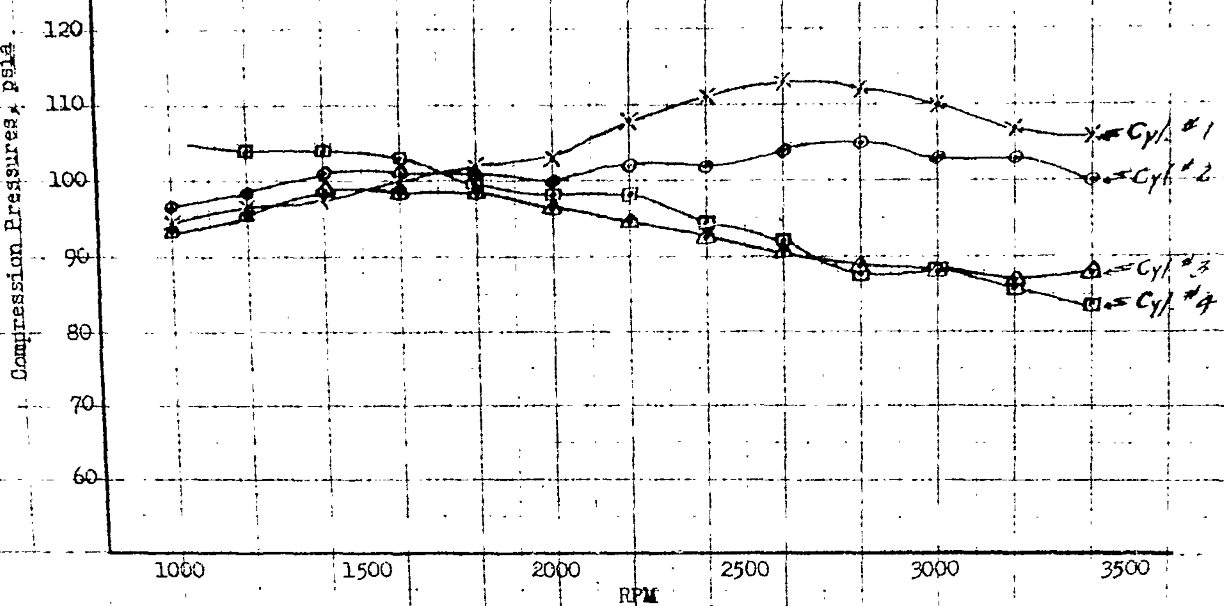


Fig. 39. Compression Pressures with High Inertia Manifold End Inlet, No Pipe Attached (See Fig. 7)

Fig. 40. **Compression Pressures with** High Inertia Manifold End Inlet, 1" x 15" Intake Pipe (See Fig. 7)



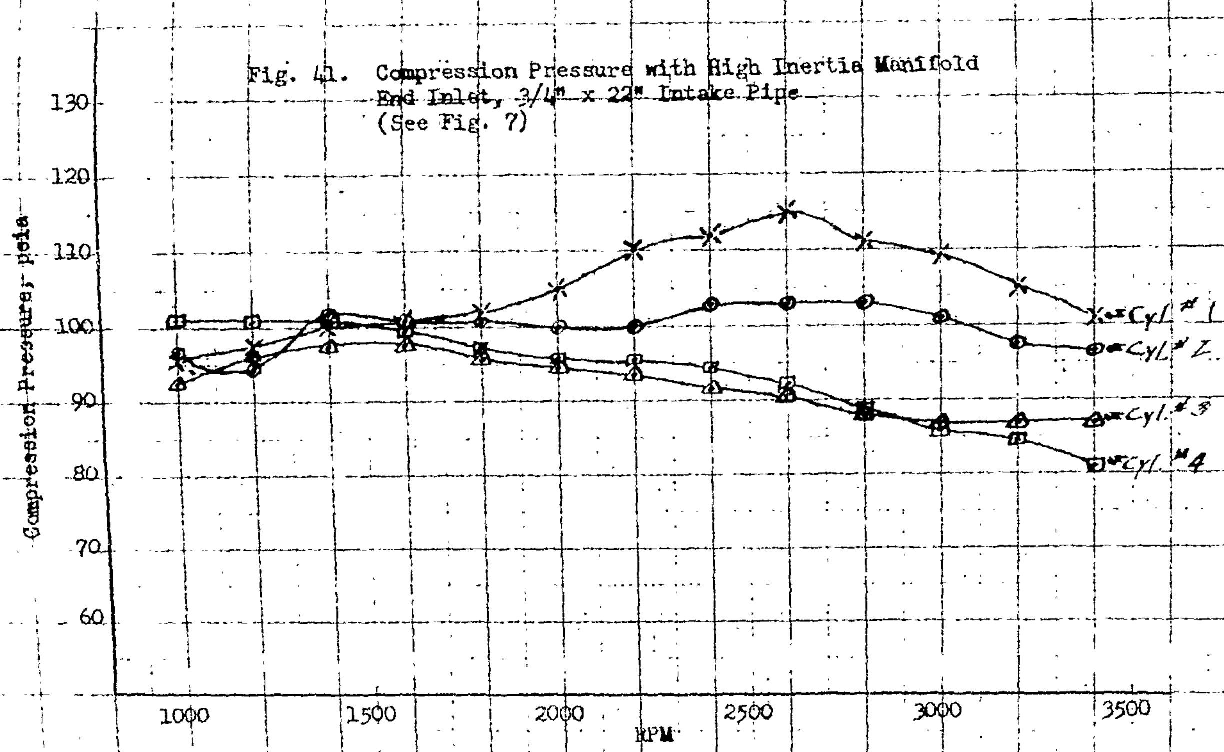


Fig. 41. Compression Pressure with High Inertia Manifold End Inlet, 3/4" x 22" Intake Pipe (See Fig. 7)

Fig. 42. Compression Pressures with High Inertia Manifold Center Inlet, No Pipe Attached (See Fig. 7)

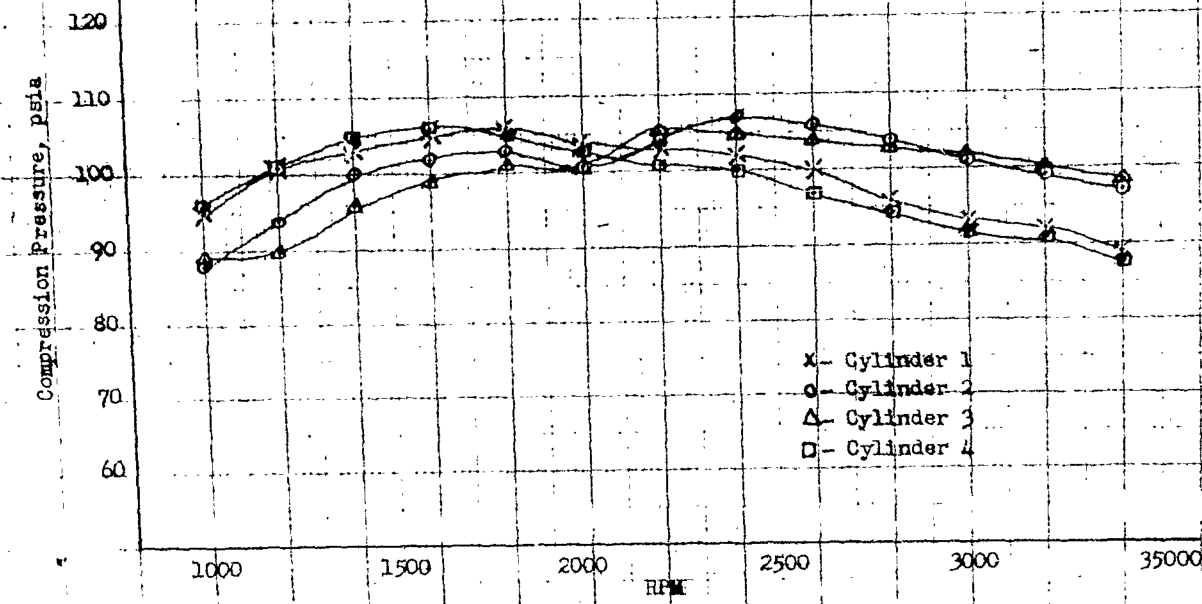


FIG. 43 PRESSURE VARIATION AT INTAKE PORT WITH MANIFOLD NO. 5 (FIG. 7) CENTER INLET, NO PIPE

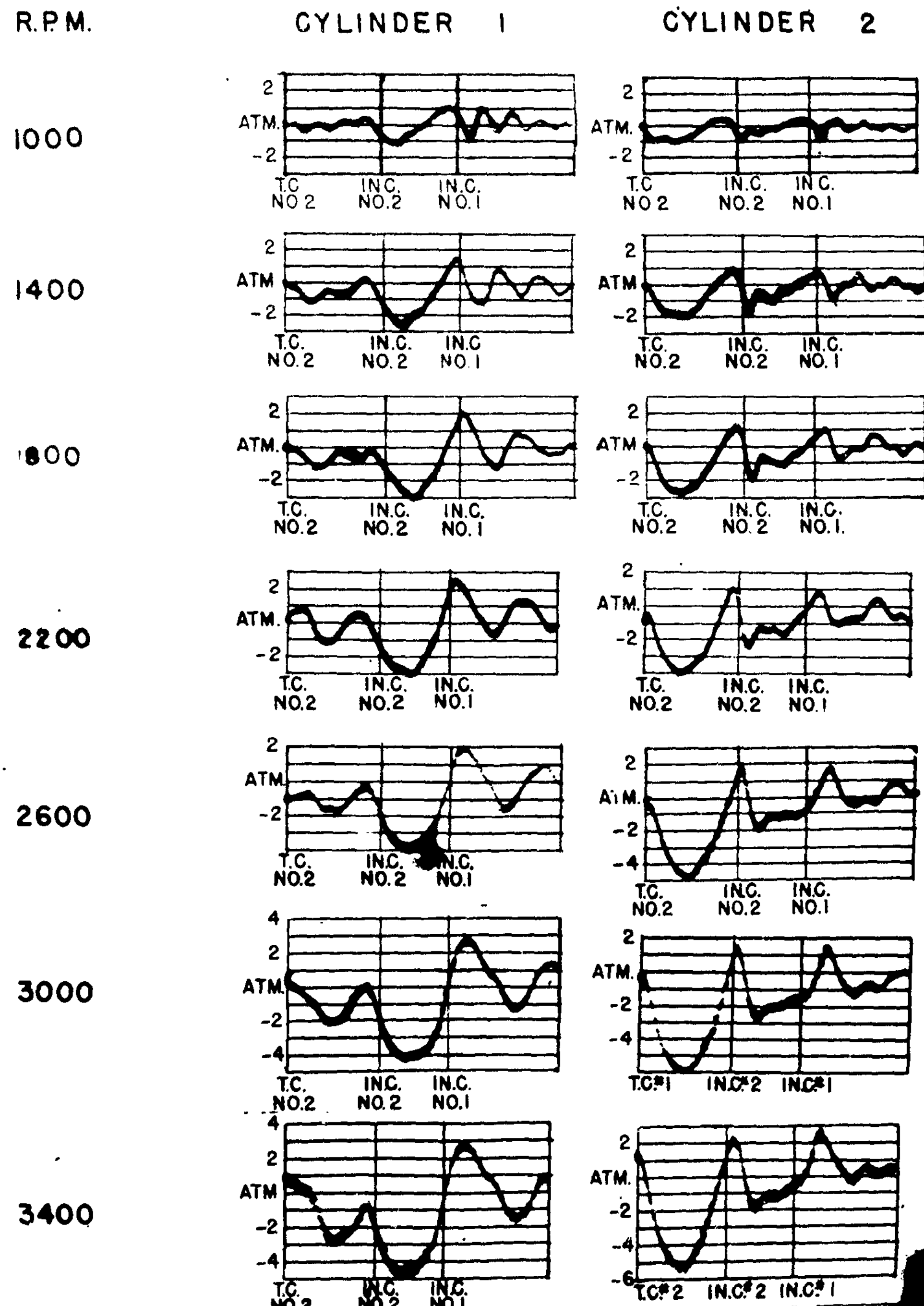


Fig. 44. Compression Pressures with High Inertia Manifold Center Inlet, 1" x 15 1/2" Intake Pipe (See Fig. 7)

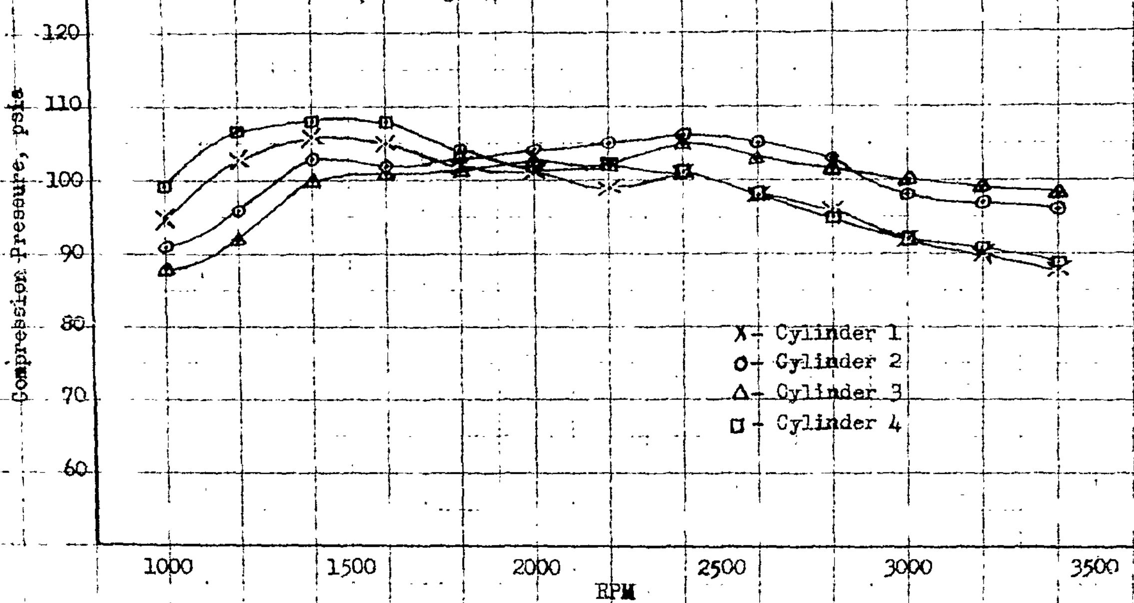


FIG. 45 PRESSURE VARIATION AT INTAKE PORT WITH MANIFOLD NO. 5 (FIG.) 7) CENTER INLET, 1" X 15 1/2" INTAKE PIPE

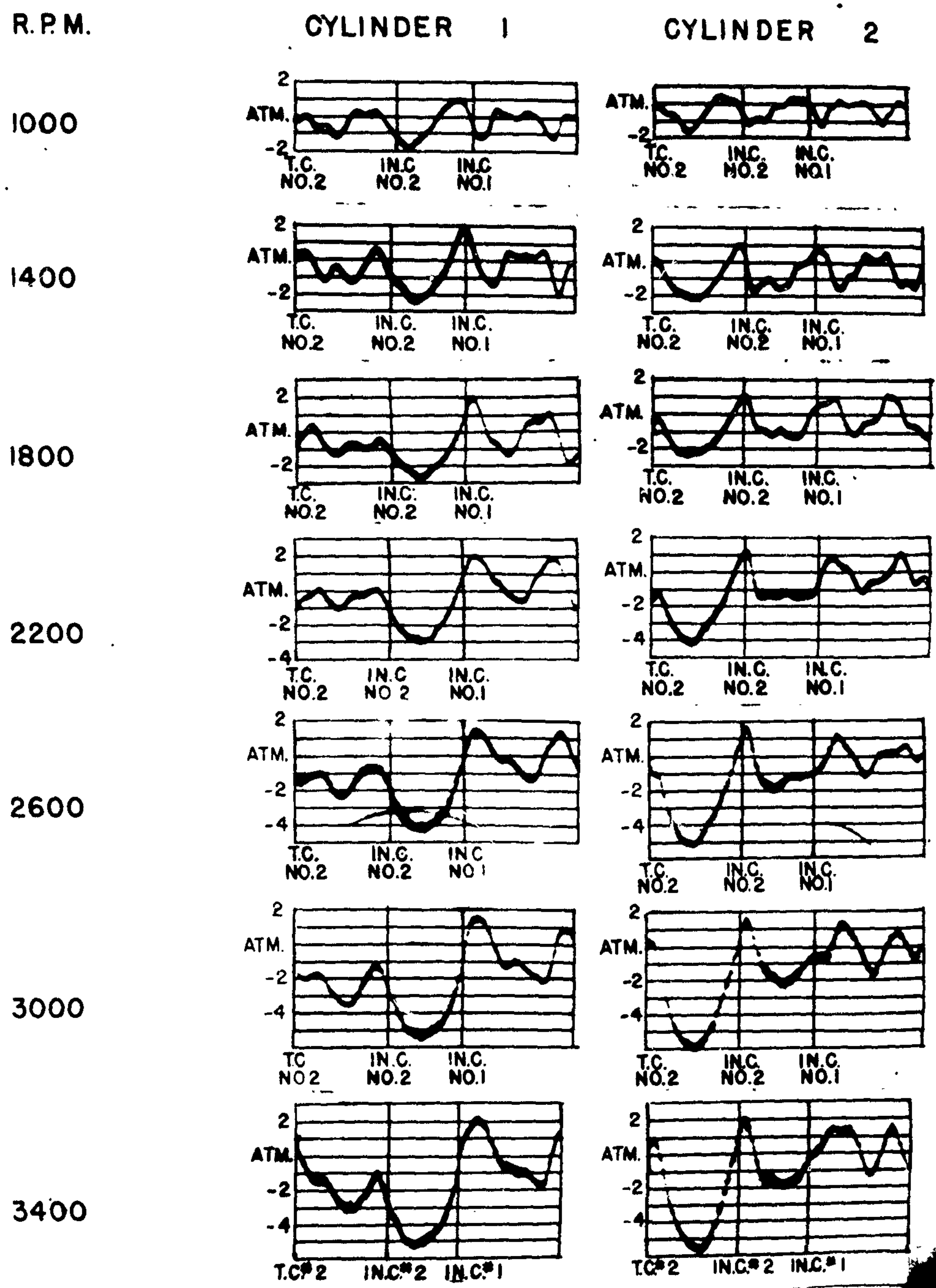


FIG. 46 PRESSURE VARIATION AT INTAKE

TEE WITH MANIFOLD NO. 5 (FIG. 7)
CENTER INLET, 1" X 15 1/2" INTAKE PIPE

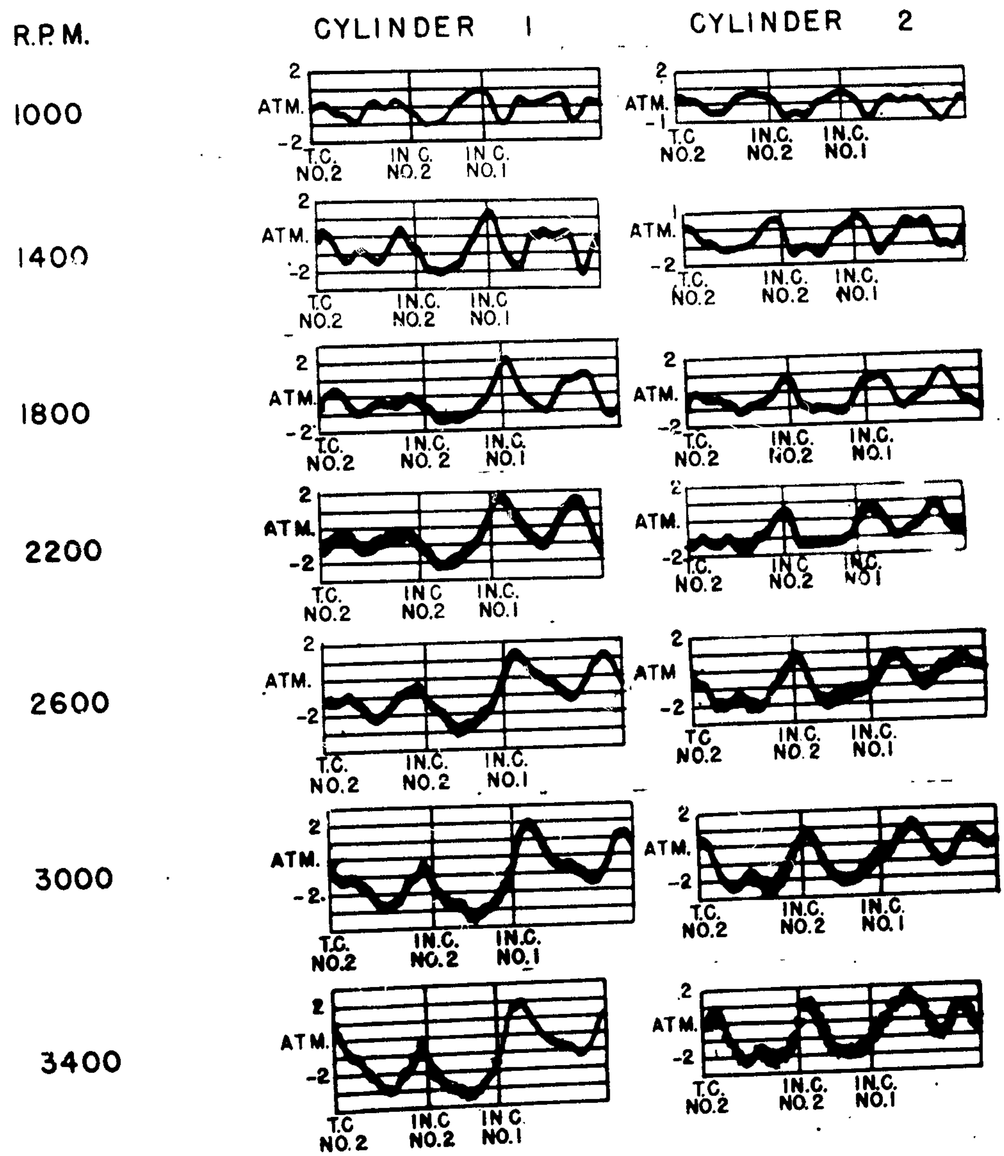


Fig. 47. Compression Pressures with High Inertia Manifold
Center Inlet with Balancing Pipe, 1" x 15 1/2" Intake Pipe
(See Fig. 7)

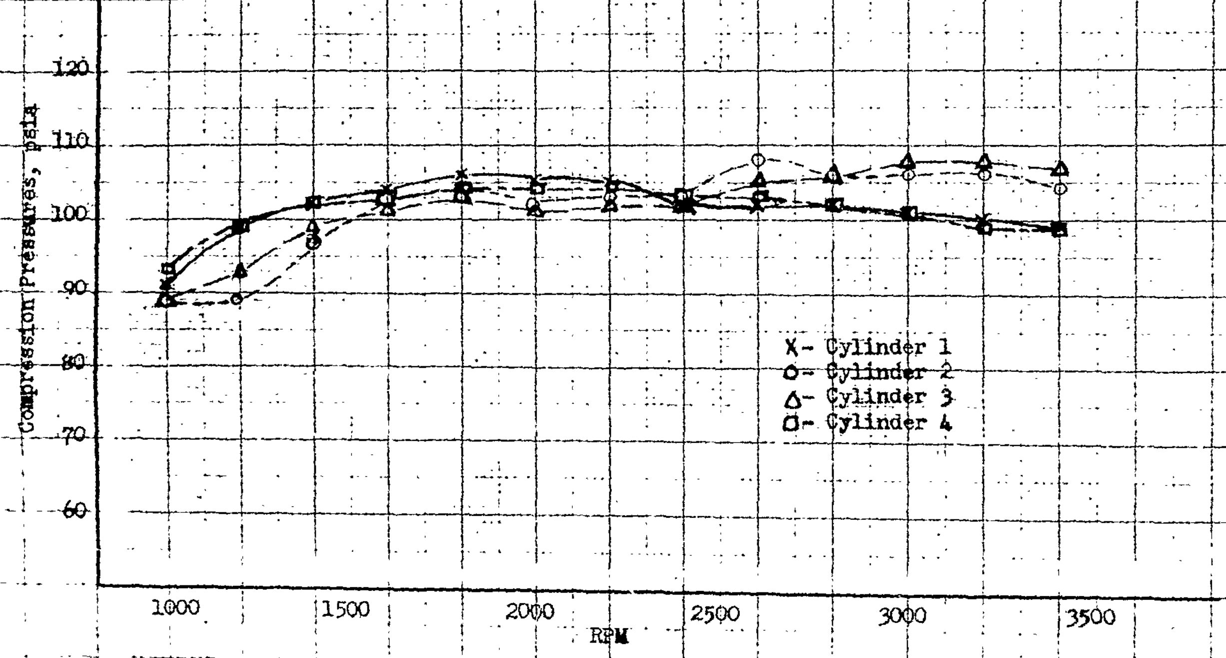


FIG. 48 PRESSURE VARIATION AT INTAKE TEE WITH MANIFOLD NO. 5 (FIG. 7) WITH BALANCING PIPE, 1" X 15" CENTER INTAKE PIPE

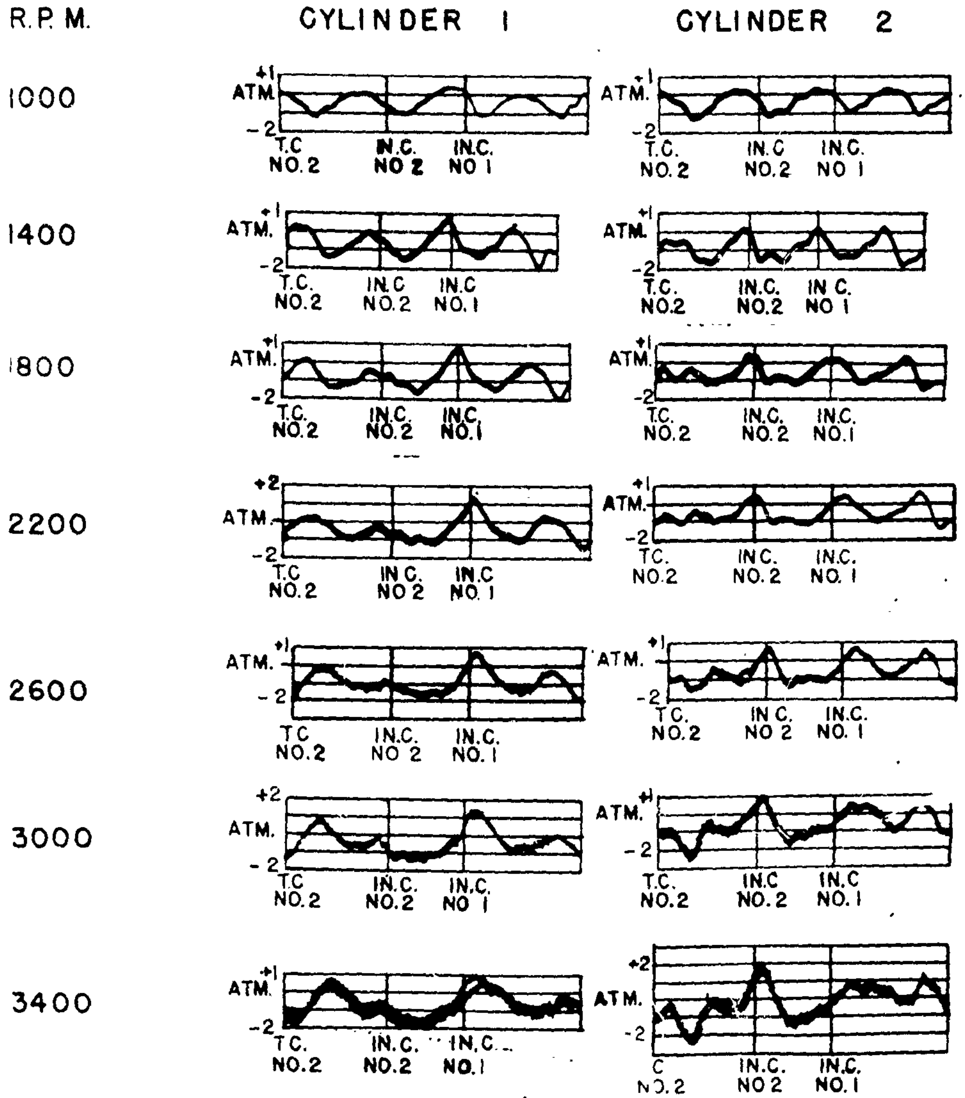


FIG. 49 PRESSURE VARIATION AT INTAKE PORT WITH MANIFOLD NO. 5 (FIG. 7) WITH BALANCING PIPE; 1" X 15 1/2" CENTER INTAKE PIPE.

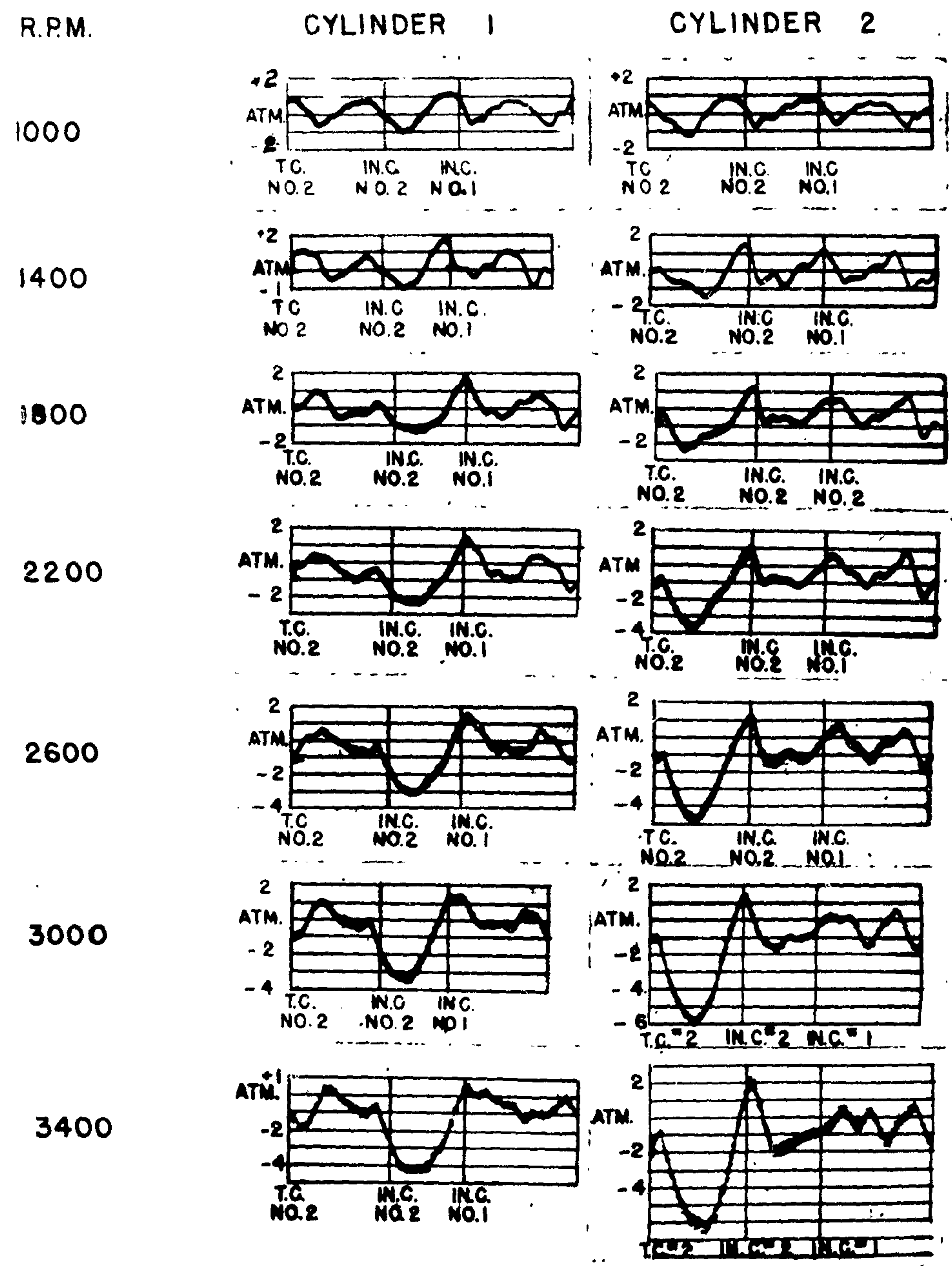


TABLE I

Pipe Size	Speed for Peak Volumetric Efficiency	
	Calculated	Actual
1" x 12"	4080	above 3200
1" x 18"	3280	3300
1" x 41"	1980	2300
3/4" x 12"	3300	3200
3/4" x 22"	2420	2600
3/4" x 36"	1830	2000

NOTE: For calculating the speed for peak volumetric efficiency, 2" was added to the length of each pipe to correct for port length.

TSDIN FORM 69 (13 MAR 47)

Hardy, J. A.
Kemler, E. N.
AUTHOR(S)

DIVISION: Power Plants, Reciprocating (6)
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REVISION

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FORG'N. TITLE: P21/7 Manifolds (Engines)

ORIGINATING AGENCY: Purdue Univ., Engin. Experiment Station, Lafayette, Ind.

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ABSTRACT

Factors affecting the performance of a wide variety of multi-cylinder engine manifolds are presented. To determine relative volumetric efficiencies, compression pressures were measured while engine was run by dynamometer. Two types of vibration occur in multi-cylinder manifolds affecting volumetric efficiency, namely ramming action and vibration of air in intake pipe against spring effect of manifold and cylinder volume. Individual intake pipes gave higher volumetric efficiency. Experimental work was conducted on a four-cylinder motorcycle engine.

T-2, HQ., AIR MATERIEL COMMAND

AD-B805 158

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Robert Storer

Robert Storer
Chief, Records and Declassification Division

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Study of Multi-Cylinder Engine Manifolds

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