INCREASING THE HORSEPOWER OUTPUT OF GAS ENGINES BY MEANS OF A TUNED INTAKE MANIFOLD SYSTEM

BY

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INTRODUCTORY DISCUSSION

In the year 1932 a change was made in the exhaust-stack arrangement for the Diesel engines in the Heat Power Laboratory, located in the Mechanical Engineering building on the campus of the University of Wisconsin. One of the engines that had a new exhaust system installed was a 50-horsepower Fairbanks-Morse Model-Y, two-stroke engine.

This engine had previously delivered its rated power with no trouble. But with the new exhaust system, the engine would not deliver its rated horsepower without exceeding the limiting exhaust-gas temperature. Also it would not deliver the manufacturers guaranteed fuel rate, which it had done with the old exhaust-stack arrangement.

This behavior of the diesel engine was investigated quite thoroughly in the years from 1933 to 1937. The results of these investigations are published in three degree theses (1-3-4) and a fellows report (2).

Since only the exhaust system had been altered in giving the lowering of performance of the Fairbanks-Morse diesel, it was logically concluded that the explanation of this phenomena lay in the exhaust system.

An Article by H. F. Shepard (5), on the use of a correct length of exhaust pipe to improve engine performance gave the workers at the University of Wisconsin a start on the correct reasoning of how the exhaust system can affect engine performance. The first of these workers, Whitefield (1), set up a series of test runs to correlate with a theory on how the length of the exhaust pipe affects performance of engines. The theory in a brief version is as follows: Upon opening of the exhaust valve of the engine a pressure wave was induced in the exhaust pipe due to the high cylinder pressure at the instant of valve opening. The wave passed down the exhaust pipe, and due to the high pressures involved a fairly high vacuum was left in the wake of the pressure wave. After this wave passes out of the exhaust pipe, the atmospheric pressure sets up a pressure wave traveling into the exhaust pipe and returning to the exhaust port. Here the kinetic energy of this wave is converted to pressure and the cycle is repeated. For good operation the pressure on the exhaust port, when it opens, should be negative (as determined from each series of waves induced by the previous opening of the exhaust port). For poor operation there would be a positive pressure on the exhaust port each time it opened. This pressure or lack of pressure on the exhaust port, would hinder or aid scavenging of the cylinder as the case may be. Thus by varying the exhaust-pipe length, variations of operation are to be experienced.

Whitefield set up the series of test runs so that the length of the exhaust-stack was varied in two-foot increments from 22 feet to 86 feet, while holding the engine speed approximately constant at 360 rpm for all runs. With these results Whitefield showed that operation varied from best to worst and back to best again. With this information he also showed that the velocity of the pressure waves was equal to sonic velocity. Thus the old classical pipe-organ-frequency theory of physics is applicable.

Woerner (4) used a like series of test runs but varied the exhaust-pipe length from 24 to 104 feet, and held speed constant at 300 rpm for all runs. Application of wave theory for pipes also was applicable for these runs.

Now it seems logical that the above theory could be used on the intake-manifold system to get a better volumetric efficiency for an engine. This has several advantages which are readily apparent:

(1) More horsepower from a given size of engine.

(2) No supercharger work to lower the engine cycle efficiency.

(3) No moving parts.

(4) A pipe is quite easy to construct.

No literature was found which showed any experimental work had been carried out in an attempt to get a supercharging effect from a proper length of intake pipe. But more study of the differences involved in the intake system lead to the belief that probably the effects desired could not be obtained, primarily because the driving force (in the case of the exhaust systems, the cylinder pressure, and for intake manifolds the vacuum when the intake port opens) is of such different magnitude. The pressure in the cylinder when the exhaust opens may be on the order of a hundred pounds per square inch, while the vacuum pulled when the intake port is opened is on the order of a few inches of mercury at wide open throttle conditions. Thus it seems that the damping effect of a long intake pipe might destroy the desired effects.

In order to get the same effects that are listed as advantages in having an induced high pressure on the intake valve when it is open, a new type of manifold system was tried. The reasoning behind this new system is based on placing a "Helmholz" cavity in the manifold system. The "Helmholz" cavity is an acoustical system consisting of a chamber which opens to the atmosphere through a long narrow air column or neck. This system has been shown to have a very definite resonant frequency property. Adding this cavity would give surges to the air inside, which would pass from one end of the cavity and back at a very precise frequency. The attempt made was to use these surges of air instead of pressure waves to raise the horsepower of the engine.

It is to be noted that the surges induced by the "Helmholtz" cavity are very different in nature from a pressure wave. This might best be shown by the old experiment of throwing a stone into a quiet pond. Both waves and surges are set up in the water. The surges can be told by their large variations in height from crest to trough. These surges radiate from the point where the stone landed. Superimposed on the surges are small ripples in the water. These also radiate from the center and travel much faster than the surges. After the ripples outdistance the surges they lose their energy rapidly and disappear. These ripples correspond to the pressure waves. Remember, the above is an analogy. What was called a water surge above is normally called a wave. This could lead to confusion if not clearly understood.

Thus it was decided that a manifold system containing a "Helmholtz"type resonant cavity and air column should be tried in an attempt to raise the maximum horsepower of the engine. To facilitate matters an engine with a wide speed range was chosen. Thus the manifold could be installed and its effects at various speeds noted.

The following considerations, equations, and calculations led to the design which was used for the manifolds.

The "Helmholz" resonator has the approximate shape shown below. Hereafter the neck of the resonator is referred to as the resonator column and the spherical bulb is referred to as the resonator cavity.

Helmholz Resonator

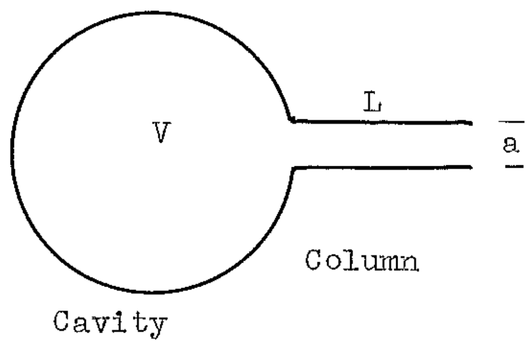


Figure 1

Symbols used in the following derivation are as follows:

V = volume of the cavity

L = length of neck

a = cross-sectional area of neck

d = instantanious density of the gas

d_o = average or static density of the gas

p' = total instantaneous gas pressure

P_s = static or average gas pressure

p = excess gas pressure = $p' - P_s$

s = $(d - d_o)/d_o$ = condensation

k = ratio of specific heat of a gas determined at constant pressure to that determined at constant volume.

Consider that the mass of air enclosed by the column of the flask moves as a unit possessing a mass equal to $(d_o \cdot a \cdot L)$. Because the diameter of the cavity is small compared to a wavelength the variables d, s, and p can be considered to have the same values at all points in the cavity.

From Keenan's "Thermodynamics" (6) it is known that variations in pressure and density in an adiabatic process are related by the equation

$$\frac{p'}{P_s} = \frac{d}{d_o}^{k} \qquad (1)$$

By definition, s is equal to $d = d_o\ (1 - s)$, so that

$$\frac{p'}{P_s} = (1 + s)^k \qquad (2)$$

Expanding the right-hand side of the above equation into a binomial series, and discarding all terms in s higher than the first order,

$$p' = (1 + sk)P_s \qquad (3)$$

Now by definition,

$$p = p' - P_s$$

and by substitution

$$p = k \cdot s \cdot P_s \qquad (4)$$

Because the mass of the gas inside the bulb is constant, the product V-d is constant. Therefore

$$D(V\ d) = 0 = d\ D(V) + V\ D(d), \qquad (5)$$

where D indicates the derivative.

From this

$$-\frac{D(V)}{V} = \frac{D(d)}{d} \simeq \frac{D(d)}{d_o} = s \qquad (6)$$

At the column of the resonator -D(V) is equal to (a · X) where X is the displacement of the gas along the neck. Thus

$$s = \frac{a X}{V} \qquad (7)$$

Also from thermodynamics an acoustical wave travels at a velocity given by the following formula where c denotes the velocity (6)

$$c = \sqrt{k \ P_s/d_o} \qquad (8)$$

Then

$$p = k \ s \ P_s = d_o \ c^2 \ s = \frac{d_o \ c^2 \ a \ X}{V} \qquad (9)$$

The last of the above equations gives the pressure on the air in the column in terms easy to find. Then multiply this by the area of the column to obtain the force on the air in the column. And if X takes on a unit length the resulting equation

$$\frac{d_o \ c^2 \ a^2}{V} \qquad (10)$$

is the stiffness of the air in the cavity.

Now, knowing the mass of air in the column, and knowing the stiffness of the air in the cavity, we can use vibration theory to solve for the frequency of vibration(7).

$$f = \frac{1}{2\pi}\sqrt{K/m} \qquad (11)$$

Where K is the stiffness

$$f = \frac{1}{2\pi} \sqrt{\frac{d_o \ c^2 \ a^2}{V} \Big/ d_o \ a \ L} \qquad (12)$$

$$f = \frac{c}{2\pi} \sqrt{a/VL} \qquad (13)$$

This last equation has the frequency determined in terms of the physical dimensions of the resonator.

No restriction is made on the shape of the cavity in the derivation above. In the experimental work done a cylindrical shape was used.

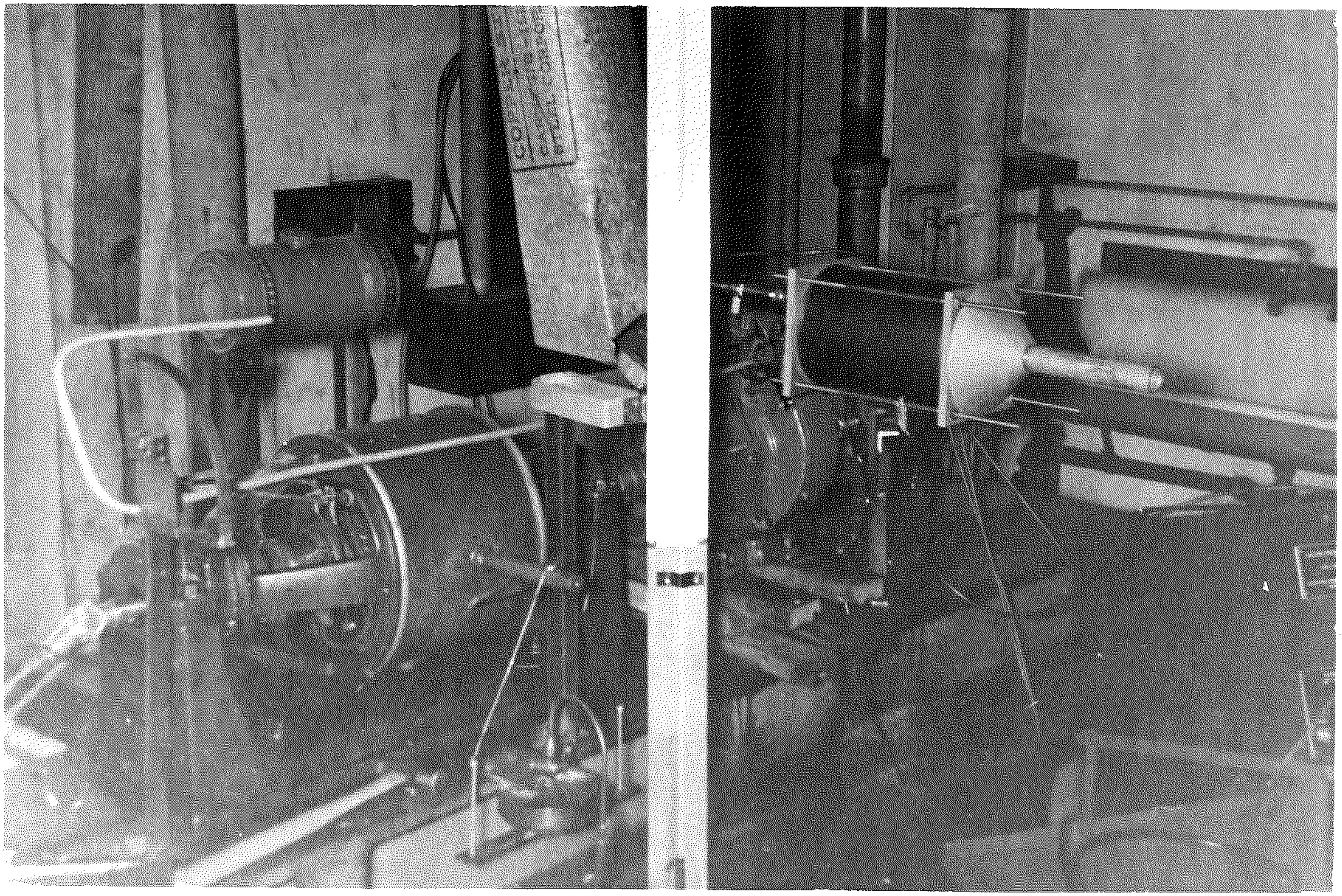
The derivation given above was taken from a set of notes prepared by George Swenson, a graduate student in Electrical Engineering at the University of Wisconsin. Equation (13) is the result obtained by Helmholz when he did his original work in this field.

One of the few articles that can be found which used the H elmholz theory is one by Heath and Elliot (8). In this the unstable operation of centrifugal compressors is discussed. It is found that the formula derived above will give the frequency of oscillation to a good degree of accuracy. This gave considerable encouragement.

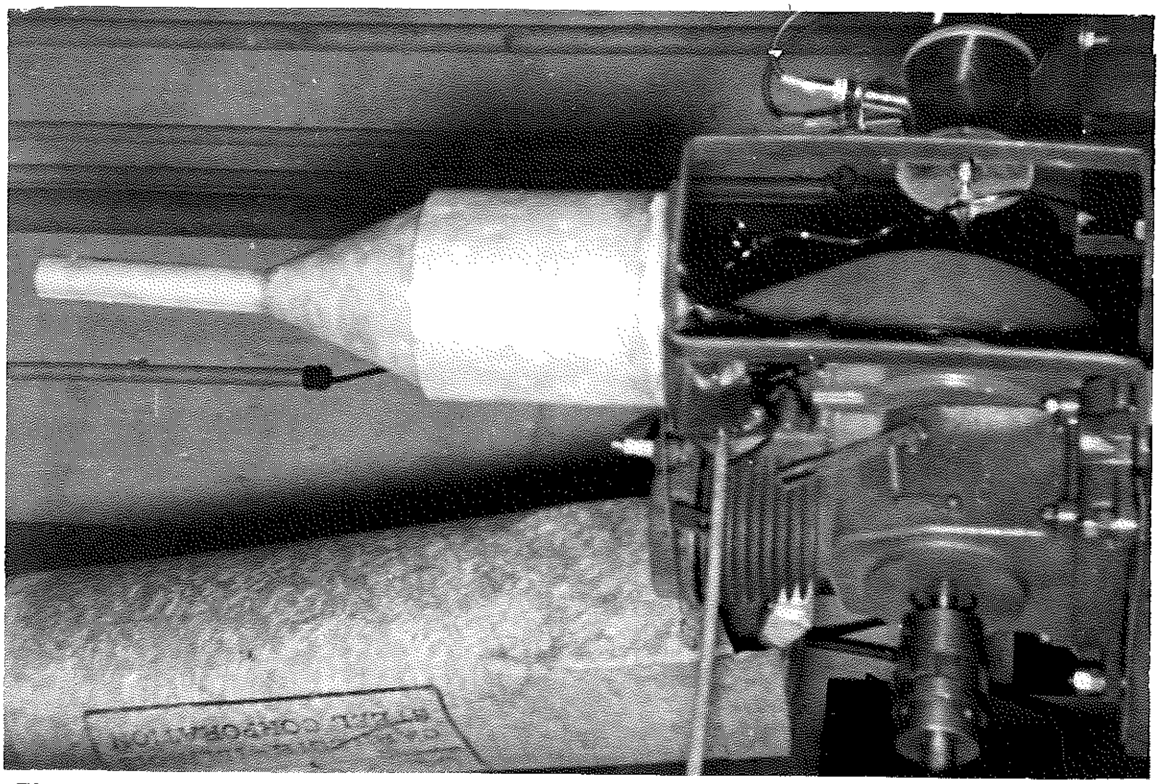
Thus it is to be expected that for a four-cycle single-cylinder engine the resonator should be set to cycle once per revolution to obtain a power increase. This would have the pressure surge returning to the intake port when the intake valve was open.

Also the engine could be expected to give a power increase when the cavity cycled three times for each two engine revolutions. This would be at an engine speed two thirds of the resonator cycling frequency.

No effect could be expected when the engine is operating at half cycling speed. This is due to the fact that the intake valve on gas engines is open about 180 to 240 degrees of crank travel. Thus if the resonator is cycling at twice engine speed a positive and a negative wave would be imposed on the valve port while it is open. From the same reasoning it is to be expected that the engine running at two-thirds resonator frequency would have a negative pressure surge imposed on the intake valve for part of the valve-open period. Thus the effect to be expected should not be as great as for engine speed equal to the resonator frequency.

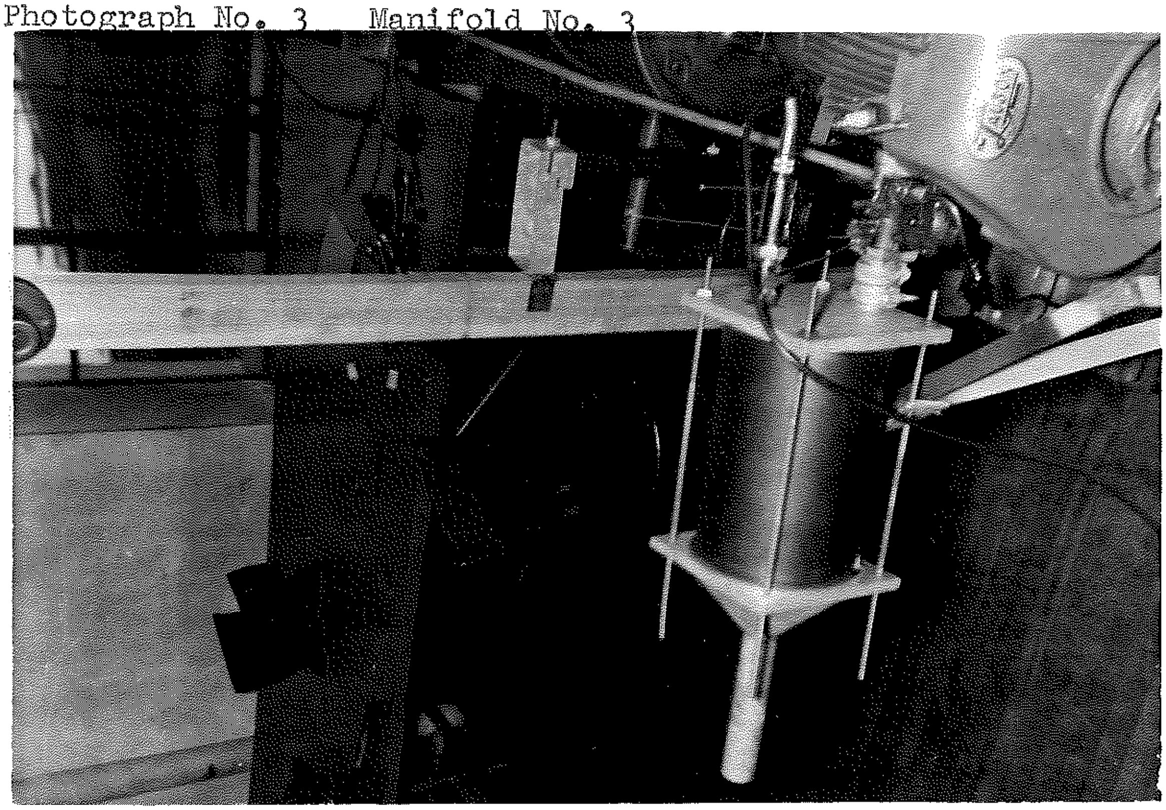


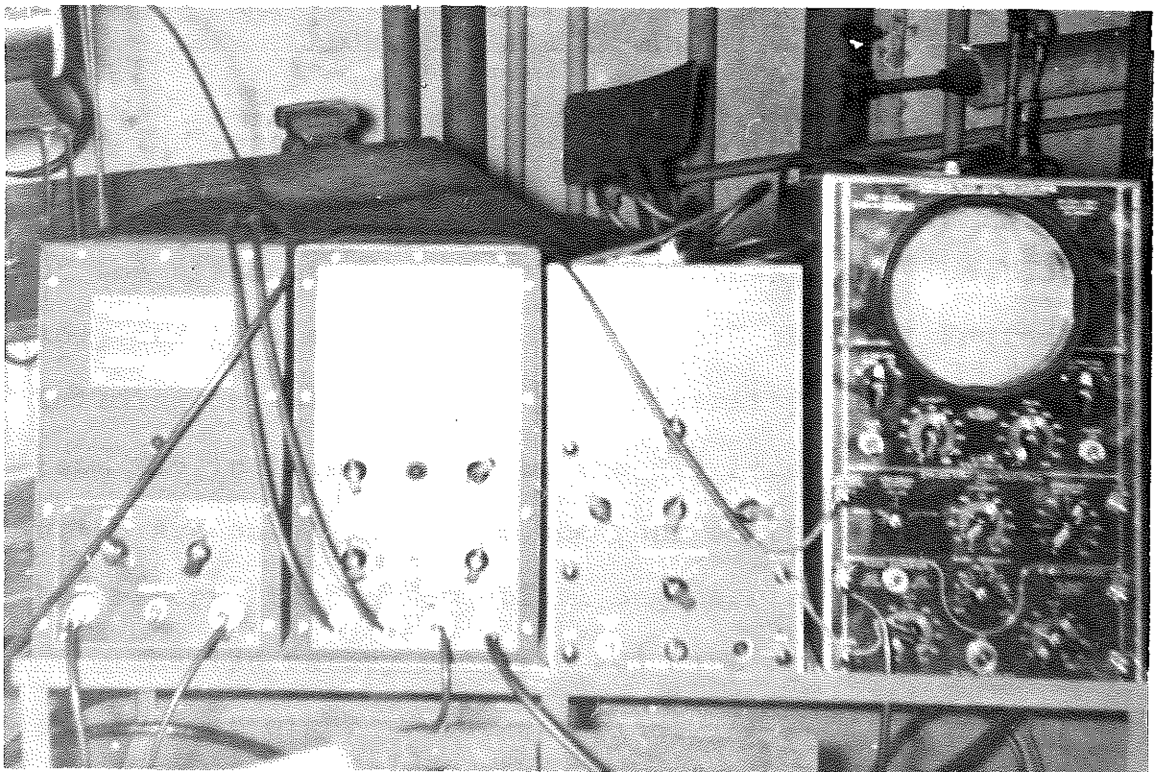
Photograph No. 1 Engine, Torquemeter, and Dynamometer as Mounted on Test Stand



Photograph No. 2   Manifold No. 2

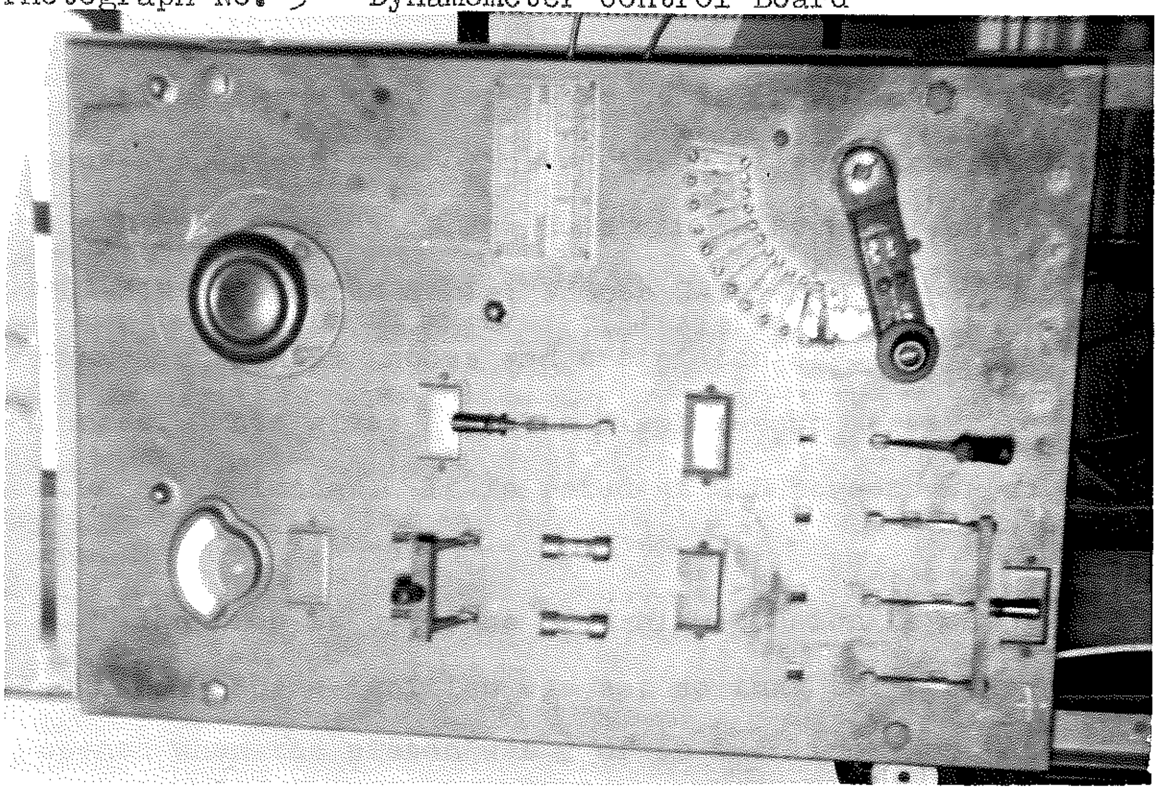
Photograph No. 3   Manifold No. 3





Photograph No.4 Pressuregraph Amplifiers

Photograph No. 5 Dynamometer Control Board



LABORATORY AND APPARATUS

The research work done for this thesis was carried out at the University of Wisconsin under the sponsorship of the Wisconsin Alumni Research Foundation. The laboratory was located in Room 215 of the Mechanical Engineering Building on the campus of the University of Wisconsin. Room 215 is located next to the Oil Laboratory, both of which are on a balcony at the rear of the Heat Power Laboratory, in the Mechanical Engineering Building.

The laboratory room was 20½ feet deep, 18 feet wide and 12 feet high. This gave plenty of room to install all the necessary apparatus. Photograph No. 1 which shows most of the apparatus used was taken facing the right rear corner of the room.

Engine

The engine used for these tests is a single-cylinder, four-stroke-cycle gasoline engine. It was manufactured by the Lauson Company of New Holstein, Wisconsin and is a TLC-349 Model with Serial Number 9-18745. The cylinder diameter is 2.25 inches and the piston stroke is also 2.25 inches. This gives a piston displacement per stroke of 8.94 cubic inches. It is rated at 2.3 horsepower at 3000 rpm.

Ignition is by means of a high-tension flywheel-type magneto. Spark ignition was set so that the breaker points opened when the piston was ¼ of an inch from top face of the cylinder. This point is about 7/64 of an inch of piston travel from top dead center. This times the spark approximately 38 degrees before top dead center. The spark plug used was a 14-mm Champion J-8 with an electrode gap of .025 inches.

The valves, both intake and exhaust, are operated by a single cam. Intake-valve clearance was set at 0.008 inches and exhaust-valve clearance was set at 0.010 inches. The valve-lift-versus-crank-angle diagrams are shown on the following page.

The exhaust muffler provided with the engine was replaced by a straight pipe about 5 inches long, thereby avoiding one source of error by preventing fouling of the muffler. This small exhaust pipe led into a large 8-inch-diameter stove pipe. This can be seen in the upper middle of Photograph No. 1 which is shown at the beginning of this section. This stove pipe leads to a blower driven by an electric motor. The blower forced the exhaust gases outside the building by way of another stove pipe. The fumes from the engine crankcase breather were also led into the stovepipe arrangement by means of a sheet metal attachment designed for this purpose.

An attempt was made to keep the engine alterations to a minimum. One of the changes required was to move the gasoline tank. Originally it was mounted right above the carburetor on the engine. It was removed in order to install the tuned manifolds. It was placed on the opposite side of the engine at a short distance from it, and at a height equal to the height when originally mounted on the engine.

Also a small piece of the cowling, which directs the air flow from the flywheel fins, was removed in order to install the carburetor as close as possible to the engine.

Dynamometer

The dynamometer used in these tests was of the compound-wound type, with a maximum rating of five horsepower. The dynamometer was not wired for motoring the engine. Starting was accomplished by means of a pull rope, which is standard procedure on this type of engine. The dynamometer is shown in Photograph No. 1 in relation to the engine. The dynamometer control panel is shown in Photograph No. 5. The field-sheostat control is the handwheel in the upper right-hand corner of the panel board. For fine field adjustment a small rheostat of the slide-contact type was used. It is not shown on Photograph No. 5 but was located immediately below the bottom of the photograph. For correct operation the knife switches on the panel board should be in the position shown in the photograph.

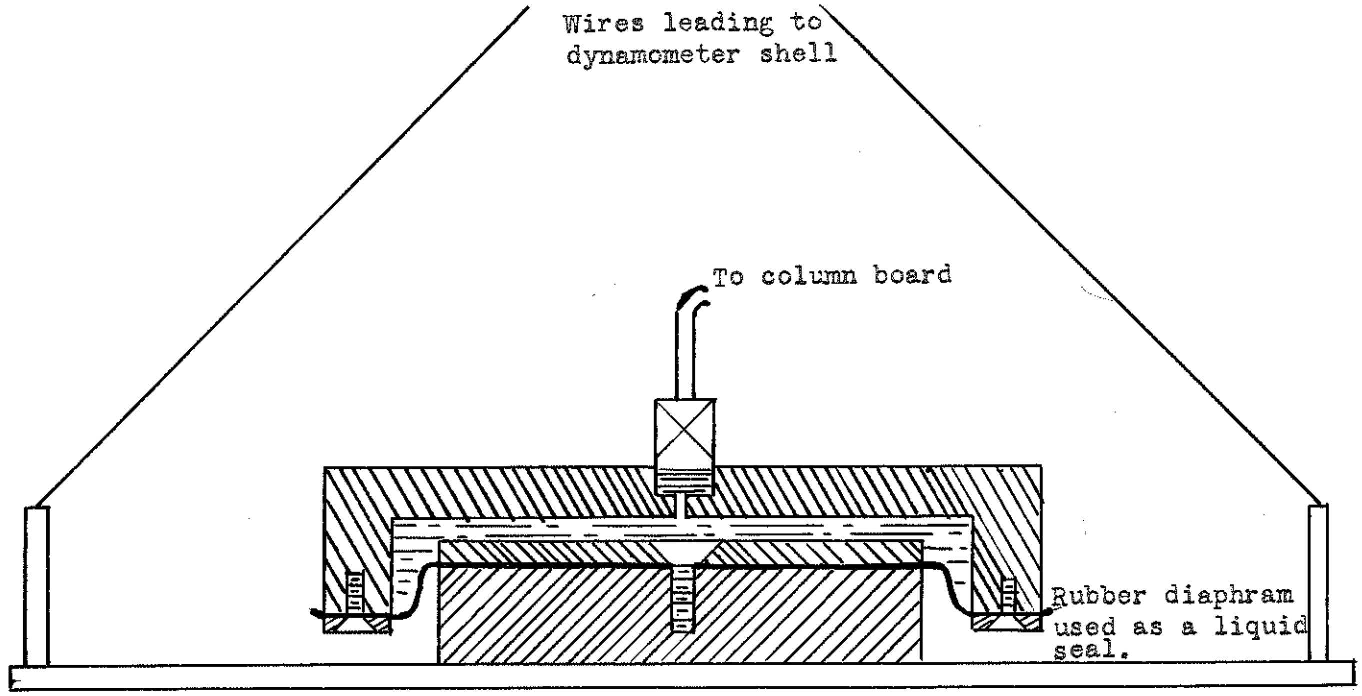
Torquemeter

The torquemeter on the dynamometer when this project was begun was a five-pound scale. This arrangement proved to be extremely unsteady due to the large vibrations produced by the single-cylinder engine. The pointer on the scale was so hard to read that curves plotted from these data were useless. So an entirely different type of torquemeter was installed. It was one of the water-column type widely used by National Advisory Committee for Aeronautics and known as a Hagen type.

The torquemeter used is shown in Photograph No. 1. The board in the immediate foreground of the photograph has the glass water tube mounted on it. To the immediate left of this board is the diaphragm housing. Operation of the torquemeter is as follows, with references to parts of the torquemeter taken from the cross-sectional view of it shown on the following page.

In attempting to rotate, the outer shell of the dynamometer exerts a force on the wire of the torquemeter. This is transmitted to the diaphragm on the bottom of the torquemeter diaphragm housing. Thus the force is converted to a pressure on the water inside the housing. The pressure induced on the water forces the water up the glass tube. The height of the water in the tube gives a direct indication of the torque being produced.

The torquemeter was calibrated by hanging weights on the opposite side of the dynamometer. The torquemeter was preloaded with about 3 foot pounds of load. When the torquemeter was calibrated with the preloading, the major



Schematic Cross-Section View of Torquemeter

Drawing No. 1

divisions were found to be linearly separated. Very good reproducibility was found if a fair amount of vibration was provided. Thus the tenths marks were interpolated in a linear manner between the full foot-pound divisions. With the steadying effect which the one-eighth-inch orfice has on the water column, it was very easy to extrapolate to one hundredths of a foot pound. This was done on all data.

The only disadvantage the new torquemeter showed was a tendency of the water-column board to vibrate laterally. This was overcome by simply placing a two-pound weight at the top of the board. This lowered the natural frequency of the board so that the vibration could not be noticed. With this refinement the torquemeter gave very trouble-free operation.

Pressure Pickup

In the study of pressure variations inside the cavity of the tuned manifold a commercial pressure pickup was used. The unit used was manufactured by the Electro Products Laboratories of Chicago, Illinois. The actual pressure pickup was of the vibrating diaphragm condenser type. The diaphragm used was 0.006 inches thick.

Amplification of the pressures noted by the pickup was by means of an amplifier built by the Electro Products Laboratories, model No. 3700 A. The amplifier is on the extreme left of Photograph No. 5.

A means of applying a reference mark and marks of either 5 degrees of crank rotation or millisecond, was provided. On the oscillograph traces photographed and included in this thesis the reference marker indicates top dead center of the piston. The five-degree and millisecond marks were not used. The reference, time and degree marks were produced by means of condenser pickups on the device shown in Photograph No. 2, to the right of and directly connected to the crankshaft of the engine. This device is called a syncromarker.

The syncromarker is driven at engine speed by direct connection to the engine crankshaft. Inside it contains a 72-tooth gear with a fin attached to the gear. A condenser-type pickup is inserted so that the teeth of the gear pass very close to the pickup. Thus every five degrees of crankshaft rotation will give a pip on the oscillograph trace.

The fin on the gear passes very close to two other condenser pickups. One of these pickups is mounted solidly in the syncromarker housing. This can be set to give a pip on the oscillograph trace at any point desired by means of correct coupling.

The other pickup which is actuated by the fin on the gear is mounted on the rear of the syncromarker and can be rotated to give its pickup to any degree of crank angle. This pickup is connected to the "external sync.

terminal" on the oscilloscope. This gives syncronization with the engine at all speeds.

The amplifier for the reference and degree marks was manufactured by the Electo Products Laboratories and is a Model No. 3851. The millisecond marks were produced by the amplifier from the sixty-cycle current supplied to it.

The cathode-ray oscilloscope used in these studies was manufactured by the Allen B. DuMont Laboratories of Passic, New Jersey. It is type No. 208-B and Serial No. 10049. It is shown on the extreme right in Photograph No, 4.

The amplifier to the left of the cathode-ray oscilloscope is a square-wave generator and is used for detonation study. Thus it was not connected to the system at all.

Speed Indicators

In the early runs a strobotac was used to determine the speed. Since this took a fair amount of the operator's time a tachometer was later installed. This allowed more even spacing of the increments of speed.

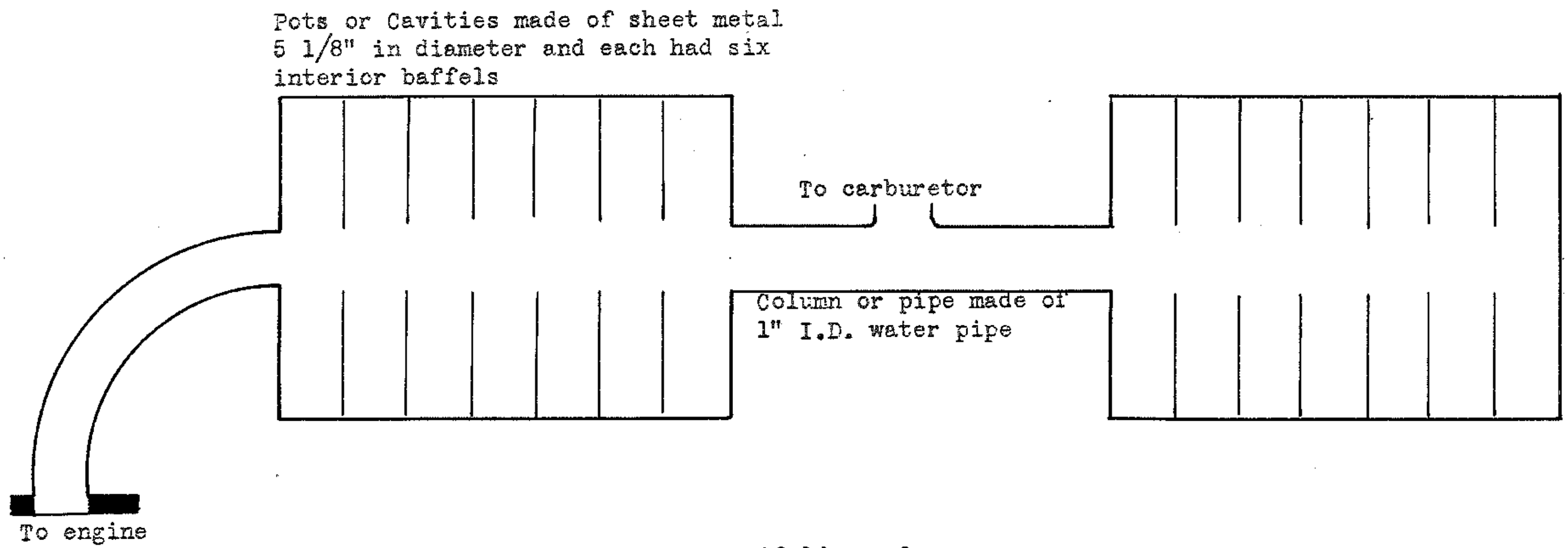
Manifolds

The first manifold built was of the double-cavity type. This was used in an effort to get a maximum of effect. The length of pipe between the two cavities corresponds to the column of air of the Helmholtz resonator. A cross-sectional view of this manifold is shown on the next page. This manifold was unique in that the carburetor was not placed next to the engine as it was for the other manifolds built. Instead it led into the pipe connecting the two cavities. Thus the cavities were full of gas vapor during operation. The two cavities were built with the same dimensions. The baffles were to direct the flow of the gases.

No actual data were ever taken with this manifold. On the first trial the engine back fired and the gases in the cavities exploded. Thus manifold No. 1 as it will be referred to, had a very short life.

In the next manifold built, the air entered the cavity through the resonant column and went from there through the carburetor into the engine. This solved the trouble of explosions. A cross-sectional view of this manifold is shown on a following page. This manifold is referred to as manifold No. 2.

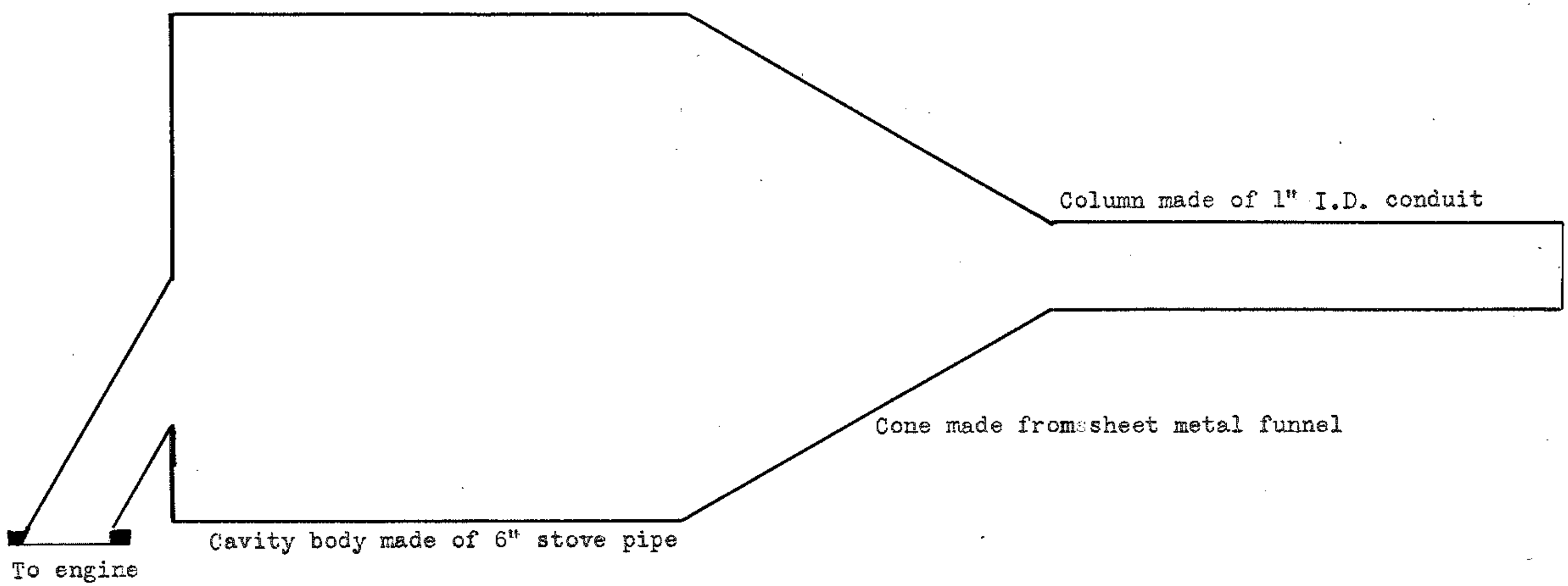
The last manifold was built so the volume of the cavity could be varied along with the length of the column. This was in an effort to get a manifold that would be good for a wide speed range. A cross-sectional view of this manifold is shown on a following page. In sequence, this manifold is referred to as manifold No. 3.



Manifold No. 1

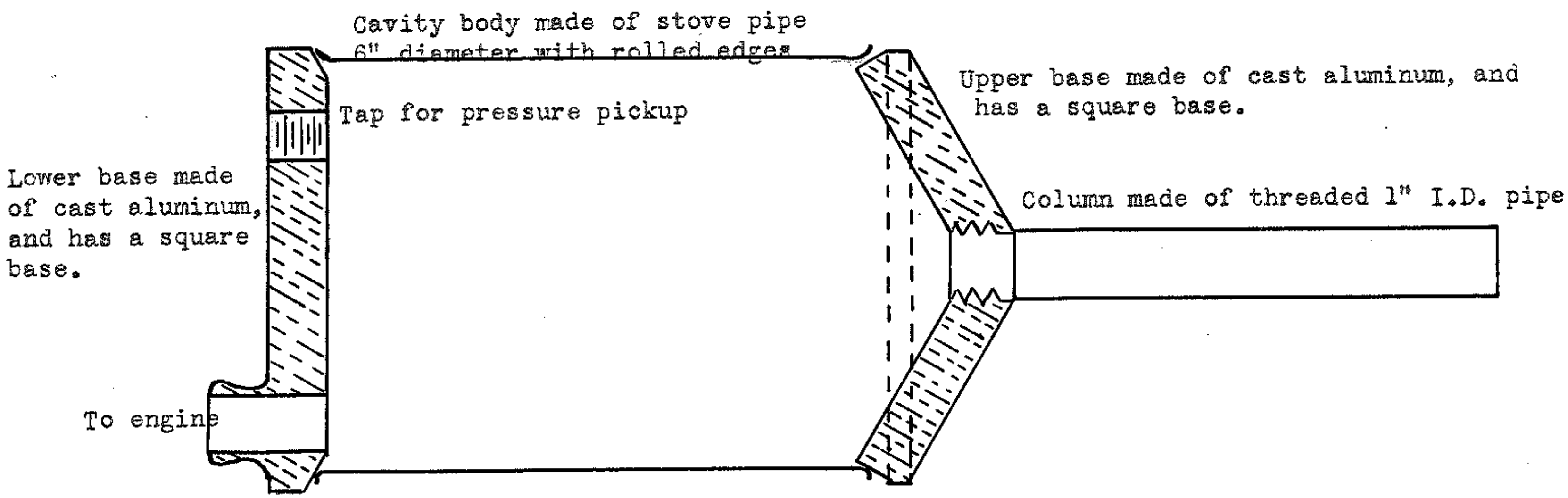
Drawing No. 2

Manifold No. 2



Drawing No. 3

Manifold No. 3



Note: Whole unit held to gether by four long bolts, one in each corner of the bases.

Drawing No. 4

TEST PROCEDURE

All test runs were maximum-horsepower runs. With the throttle wide open, the speed was varied and both speed and torque were noted. With these data both torque and horsepower curves were plotted against speed.

All tests were run after the engine had reached a stable operating condition. The warmup period was of at least 10 minutes duration with the engine running at a speed of 2500 rpm and with wide-open throttle.

The air-fuel ratio of the engine was adjusted for each run. While the engine was warming up, the needle valve on the carburetor was adjusted so that maximum torque was registered on the torquemeter. This of course was for a speed of 2500 rpm. Checks were made at much lower and much higher speeds to see if changing the air-fuel ratio would give an increase in horsepower. It did not, except in the case of the tuned manifold systems, and in this case the effect was slight. Since a needle-valve adjustment for variations in speed is usually not made in actual engine operation, it was decided not to vary the needle-valve adjustment after the warmup period was over for all runs.

Some trouble was had in getting runs that were reproducible. Most of this is attributed to the dynamometer which had a tendency to travel longitudinally and in doing so set up very large vibrations. The noise

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Some trouble was had in getting runs that were reproducible. Most of this is attributed to the dynamometer which had a tendency to travel longitudinally and in doing so set up very large vibrations. The noise made when the armature hit the bearings was comparable to diesel knock.

The traveling of the armature on its bearings was found to be caused by the torsional vibrations of the engine crankshaft. The rubber insert (spider) of the Lovejoy Coupling used was replaced by a leather spider. This improved the situation somewhat. To eliminate this condition it was necessary to wire the coupling halves together so that the engine roller bearings withstood the pull of the dynamometer armature.

Also the torque characteristic was noted to change with time. All of the later tests show the torque as rising to a top value at about 2200 rpm and then slowly falling off. In the earlier tests the torque curve started at a high value and continuously decreased. All of the torque curves are approximately the same above 2200 rpm. And since the test runs which are compared to each other were usually taken the same day, the variation was kept at a minimum.

There were several types of runs taken. First were those with the engine equipped exactly as it came from the manufacturer. Next are those runs with the tuned manifolds. In these the carburetor was mounted directly on the engine and the resonator was coupled directly to the carburetor.

A third set of runs was made with the carburetor mounted directly on the engine. Thus the friction losses and loss of pressure due to the air filter could be judged. As such, this run can be compared to the run with the tuned manifold to determine the change in performance produced by the tuned-manifold system.

TEST DATA

Manifold and air filter as sold.

Warmup: 10 minutes at maximum power and 2500 rpm.

Fuel-air ratio set for a maximum of torque at 2500 rpm.

Speed	Torque	Horsepower
2030	4.14	1.59
2060	4.14	1.62
2130	4.14	1.67
2190	4.15	1.72
2240	4.15	1.76
2290	4.14	1.80
2355	4.13	1.85
2400	4.12	1.88
2450	4.11	1.92
2510	4.11	1.96
2550	4.07	1.98
2630	4.02	2.01
2670	3.98	2.02
2720	3.90	2.02
2810	3.88	2.06
2880	3.85	2.10
2940	3.78	2.11
3010	3.72	2.13
3075	3.67	2.13
3150	3.57	2.13
3230	3.50	2.13
3300	3.40	2.13
3380	3.33	2.13
3410	3.31	2.14
3470	3.27	2.14
3520	3.13	2.09
3560	3.06	2.06

No manifold, carburetor mounted directly on engine.

Warmup: 10 minutes at maximum power and 2500 rpm.

Air-fuel ratio set for maximum torque at 2500 rpm.

Speed	Torque	Horsepower
1800	3.82	1.13
1900	3.85	1.39
2000	3.90	1.47
2100	3.92	1.565
2200	3.96	1.65
2300	3.95	1.73
2400	3.93	1.788
2500	3.92	1.865
2600	3.90	1.925
2700	3.85	1.975
2800	3.76	2.00
2900	3.85	2.12
3000	3.81	2.17
3100	3.74	2.20
3200	3.74	2.27
3300	3.65	2.29

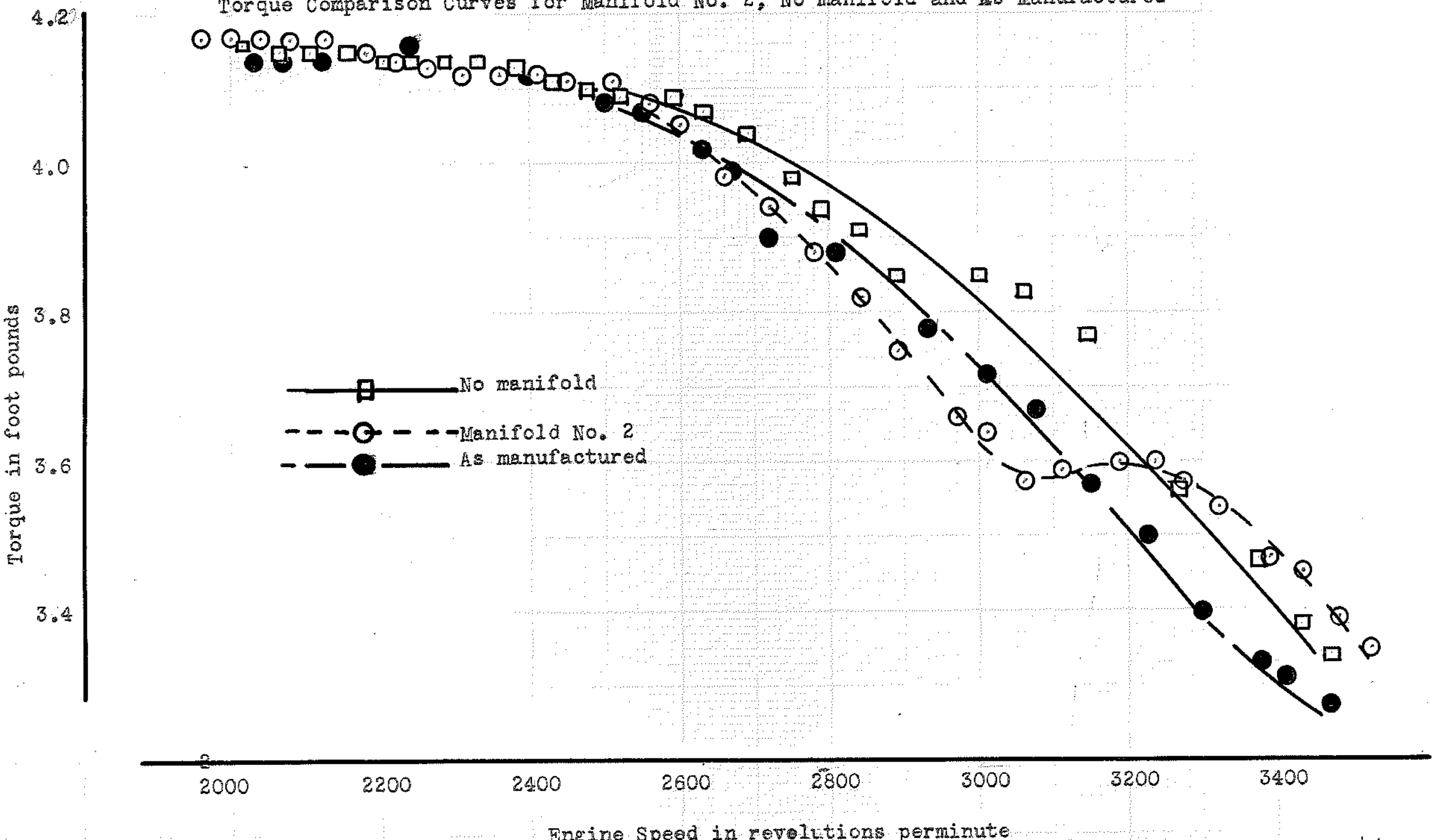
No manifold, carburetor mounted on engine.

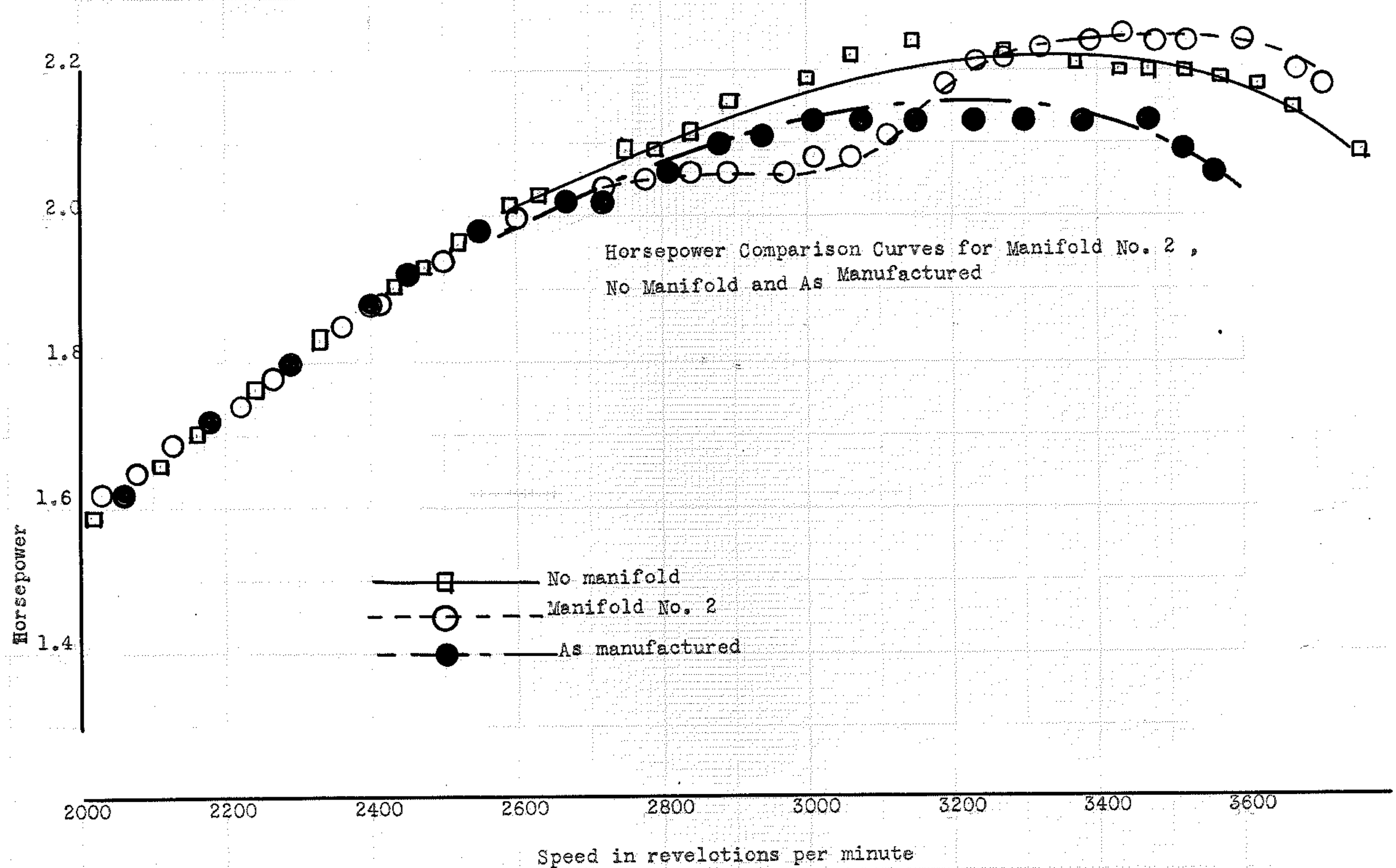
Warmup: 10 minutes at maximum power and 2500 rpm.

Fuel-air ratio set for a maximum of torque at 2500 rpm.

Speed	Torque	Horsepower
2015	4.16	1.59
2065	4.15	1.63
2110	4.15	1.66
2160	4.15	1.70
2205	4.14	1.74
2240	4.14	1.76
2285	4.14	1.79
2330	4.14	1.83
2380	4.13	1.87
2430	4.11	1.90
2475	4.10	1.93
2520	4.09	1.96
2590	4.09	2.01
2630	4.07	2.03
2690	4.04	2.06
2750	3.98	2.09
2790	3.94	2.09
2840	3.91	2.11
2890	3.85	2.15
3000	3.85	2.19
3060	3.83	2.22
3145	3.77	2.24
3270	3.56	2.22
3375	3.47	2.21
3430	3.38	2.20
3470	3.34	2.20
3520	3.29	2.20
3570	3.22	2.19
3620	3.17	2.18
3670	3.10	2.15
3765	2.93	2.09

Torque Comparison Curves for Manifold No. 2, No manifold and As Manufactured




Horsepower Comparison Curves for Manifold No. 2 ,
No Manifold and As Manufactured
No manifold
Manifold No. 2
As manufactured
Horsepower
2.2
2.0
1.8
1.6
1.4
2000
2200
2400
2600
2800
3000
3200
3400
3600
Speed in revelotions per minute

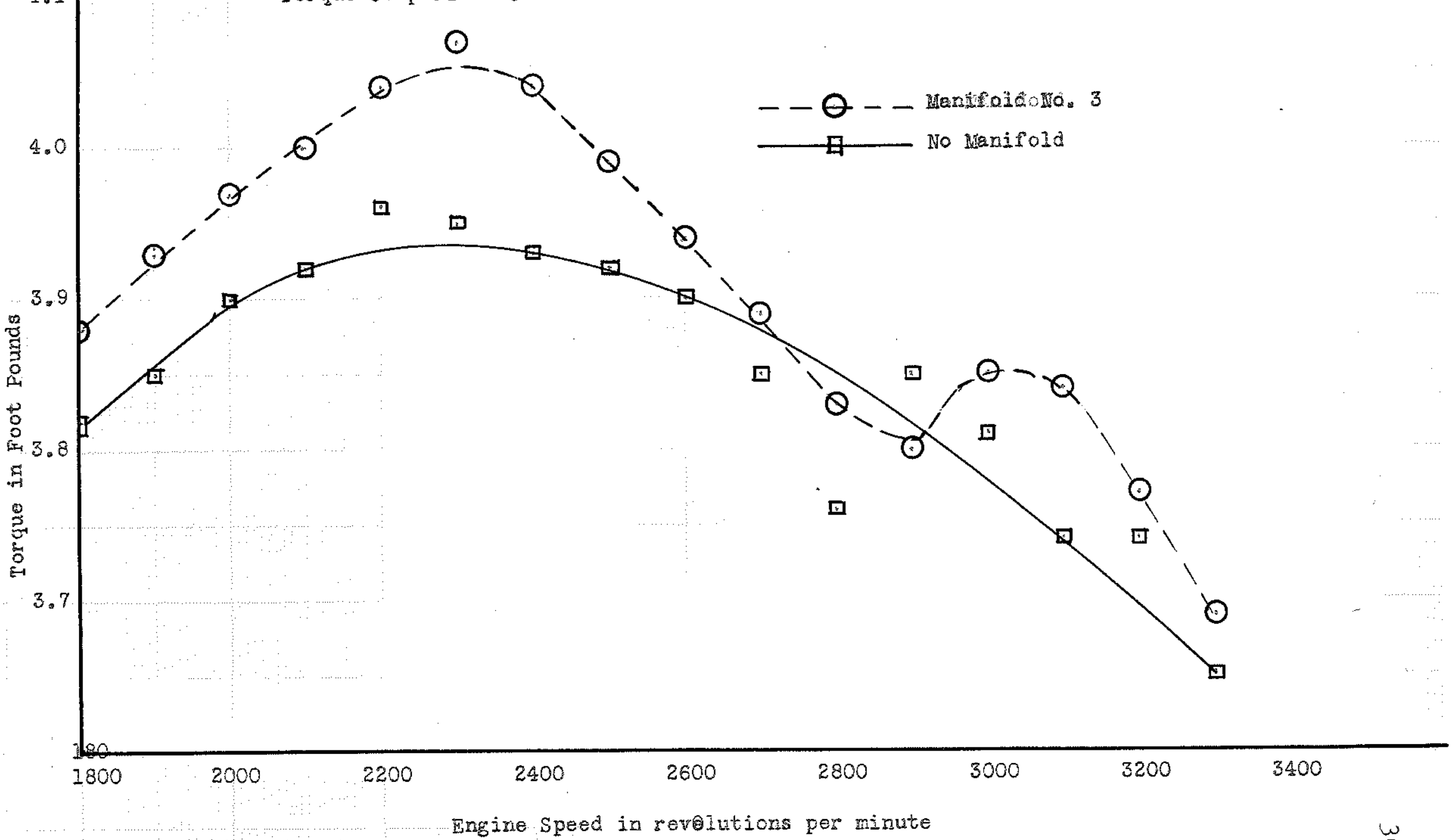
Manifold Number 2.

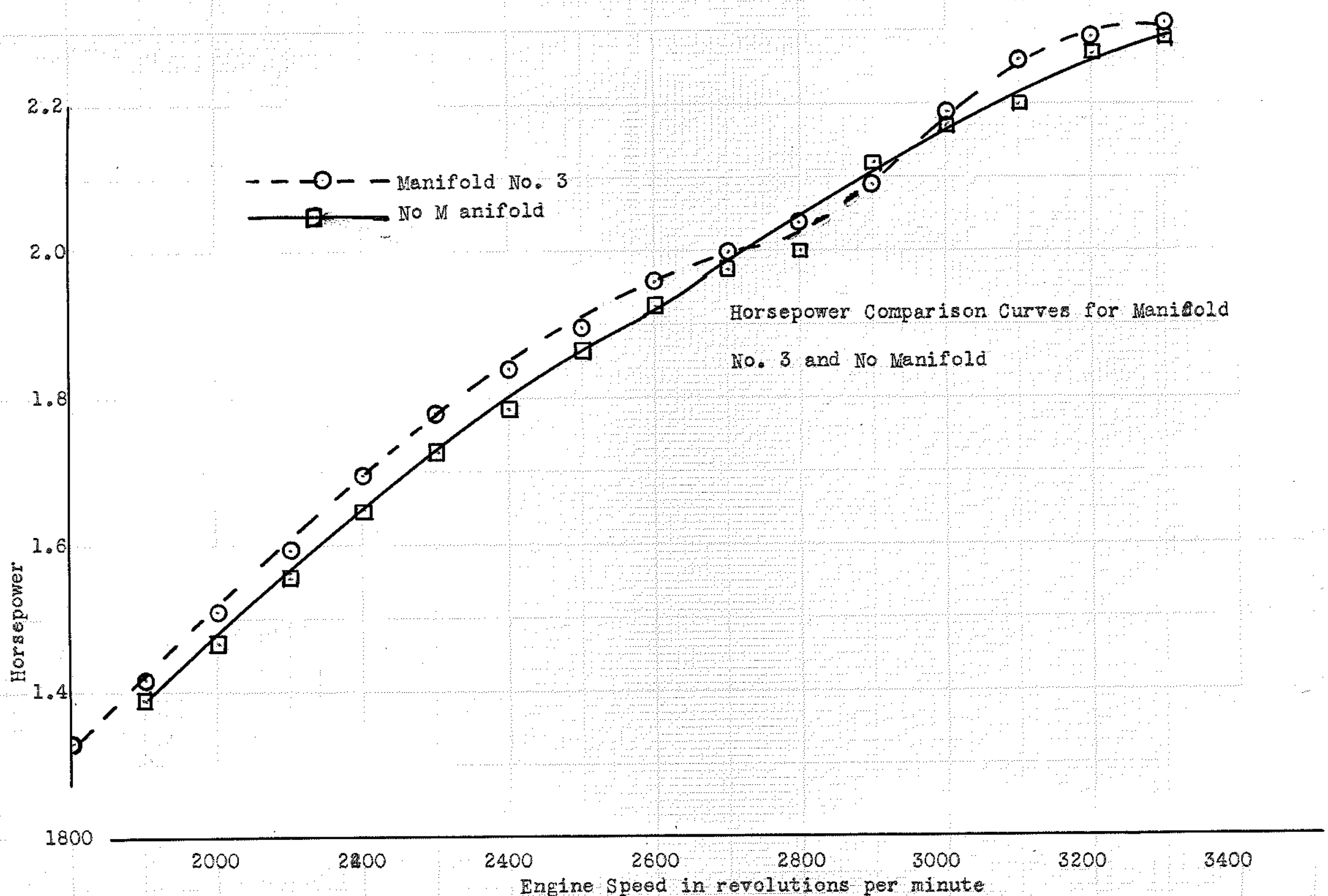
Warmup: 10 minutes at maximum power and 2500 rpm.

Fuel-air ratio set for a maximum of torque at 2500 rpm.

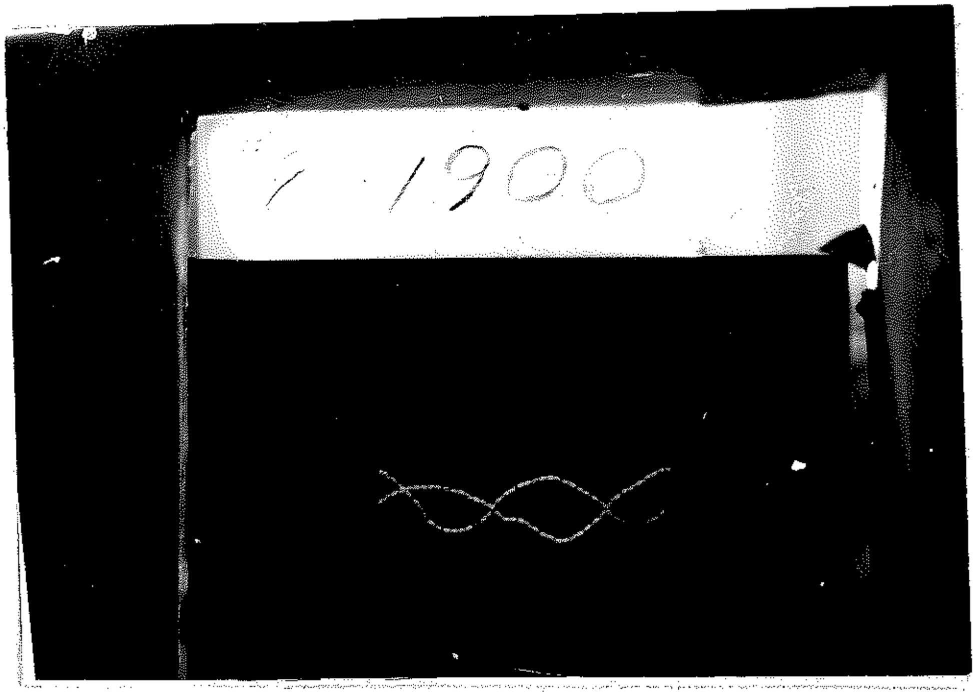
Speed	Torque	Horsepower
1960	4.17	1.55
2000	4.17	1.585
2040	4.17	1.62
2080	4.17	1.65
2130	4.17	1.69
2180	4.15	1.72
2220	4.14	1.74
2265	4.13	1.775
2310	4.12	1.815
2360	4.12	1.85
2410	4.12	1.89
2450	4.11	1.92
2500	4.08	1.94
2560	4.08	1.98
2600	4.05	2.00
2660	3.98	2.01
2720	3.94	2.03
2780	3.88	2.05
2840	3.82	2.06
2890	3.75	2.06
2970	3.66	2.06
3010	3.64	2.08
3060	3.57	2.08
3110	3.59	2.11
3190	3.60	2.18
3235	3.60	2.21
3275	3.56	2.22
3320	3.54	2.23
3390	3.47	2.24
3435	3.45	2.25
3480	3.39	2.24
3525	3.35	2.24
3600	3.28	2.24
3665	3.16	2.20
3710	3.10	2.18

Torque Comparison Curves for Manifold No. 3 and for No Manifold



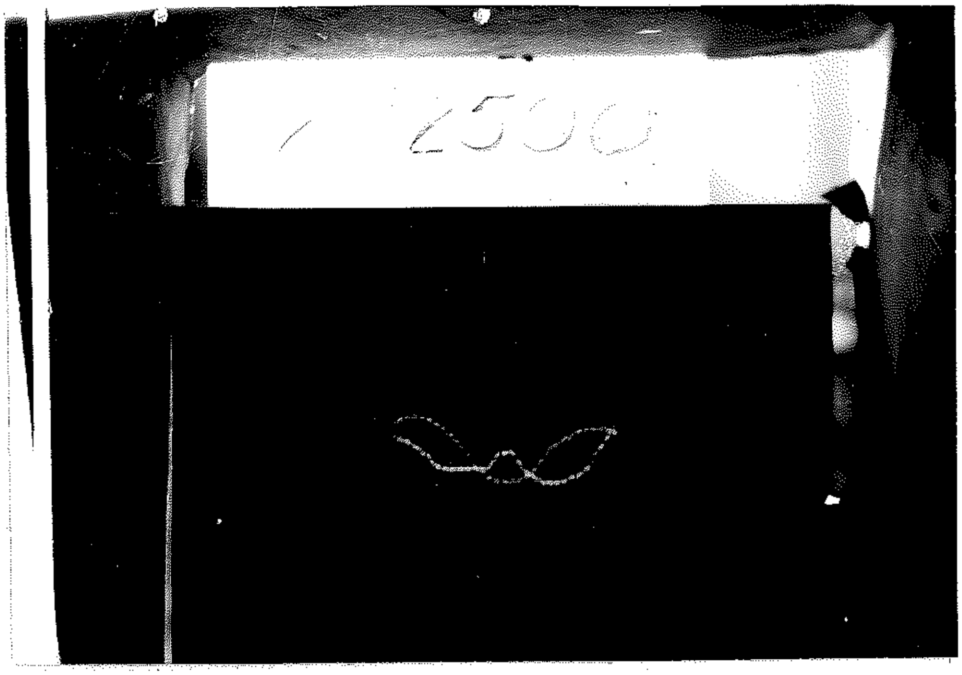


Horsepower Comparison Curves for Manifold No. 3 and No Manifold



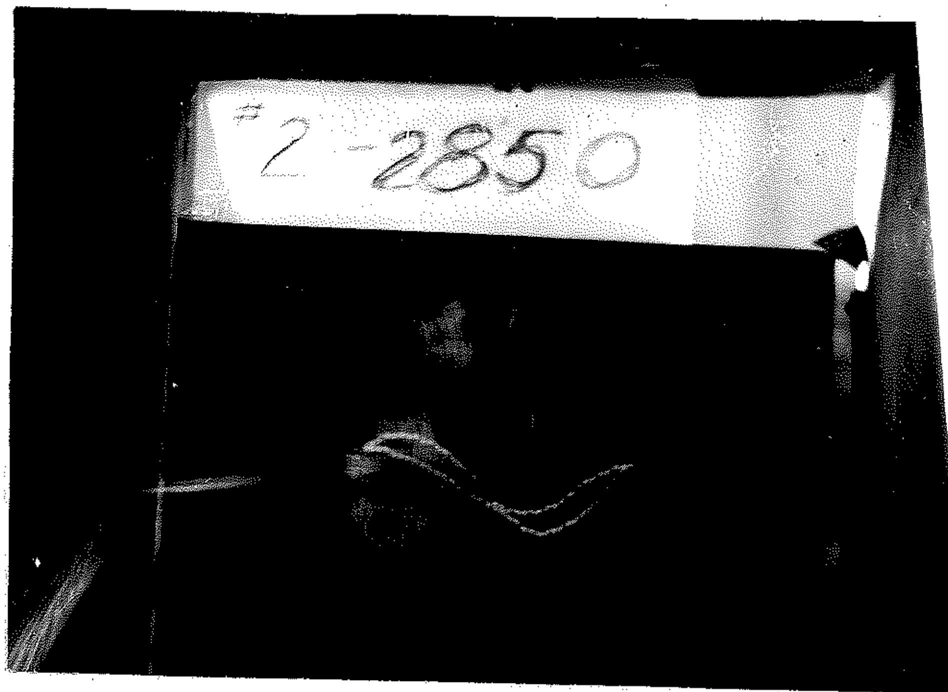
Oscilloscope Trace No. 1
Manifold No. 2. Speed 1900rpm. Torque 3.91ft. lbs.

Oscilloscope Trace No. 2
Manifold No. 2. Speed 2200rpm. Torque 3.91ft. lbs.



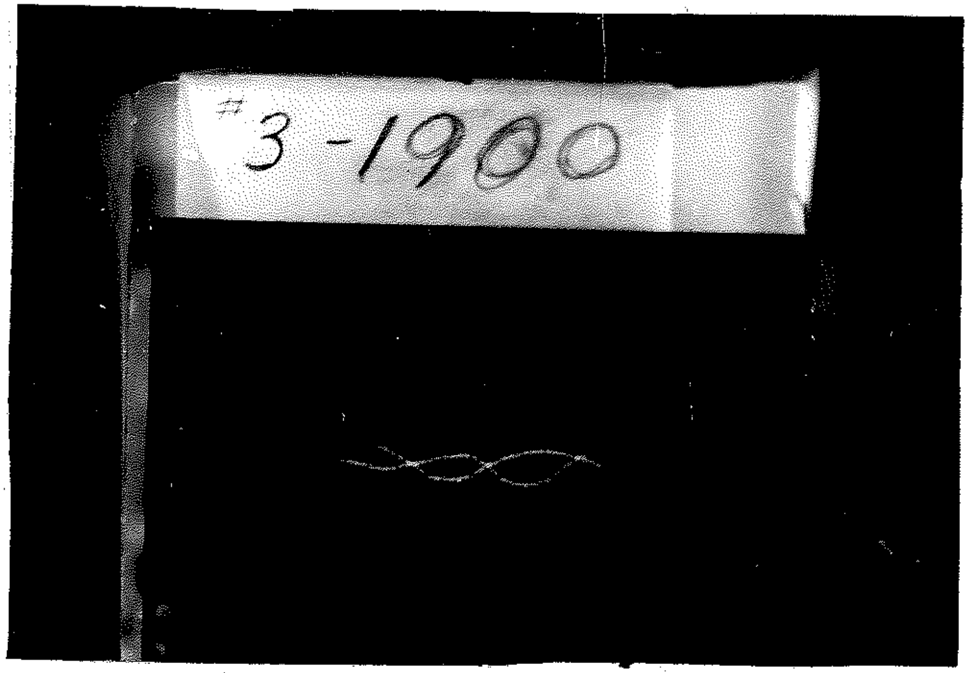
Oscilloscope Trace No. 3
Manifold No. 2. Speed 2500rpm. Torque 3.95ft. lbs.

Oscilloscope Trace No. 4
Manifold No. 2. Speed 2650rpm. Torque 3.96ft. lbs.

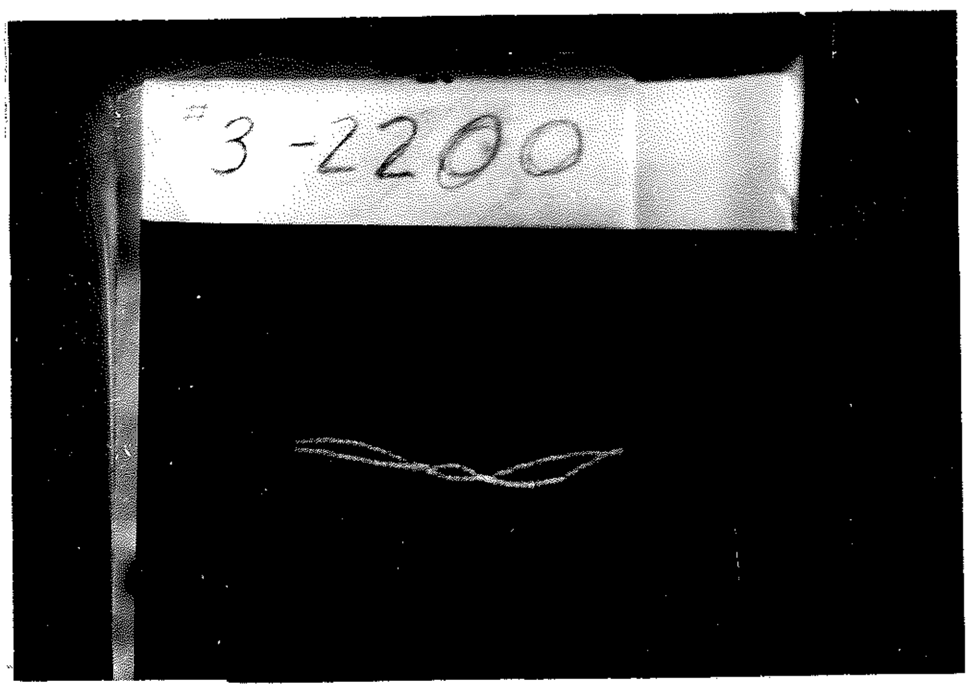


Oscilloscope Trace No. 5
Manifold No. 2. Speed 2850rpm. Torque 3.88ft. lbs.

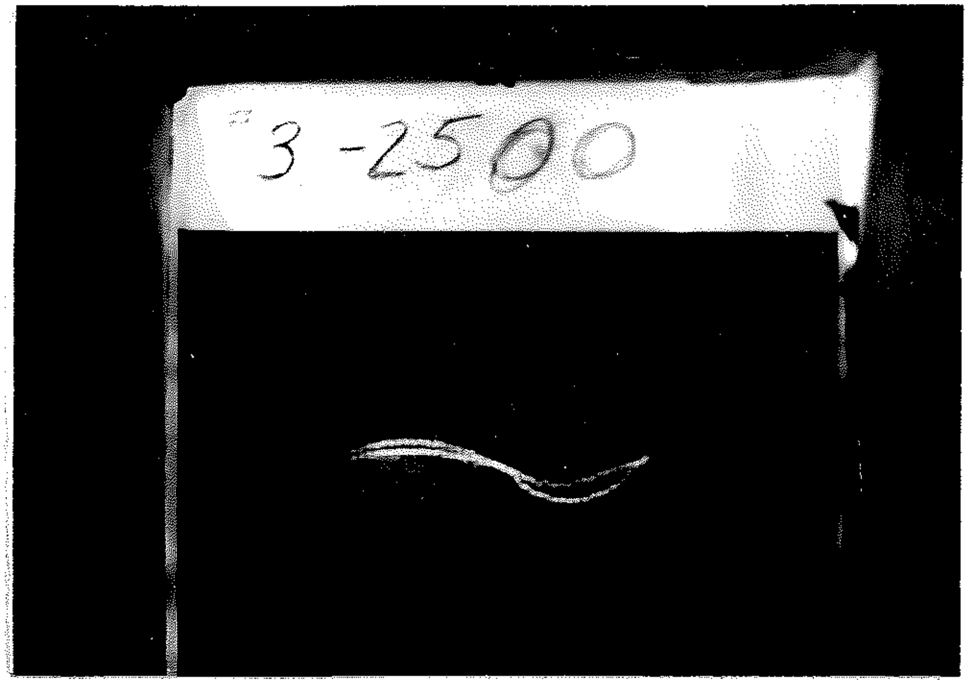
Oscilloscope Trace No. 6
Manifold No. 2. Speed 3000rpm. Torque 3.70ft. lbs.



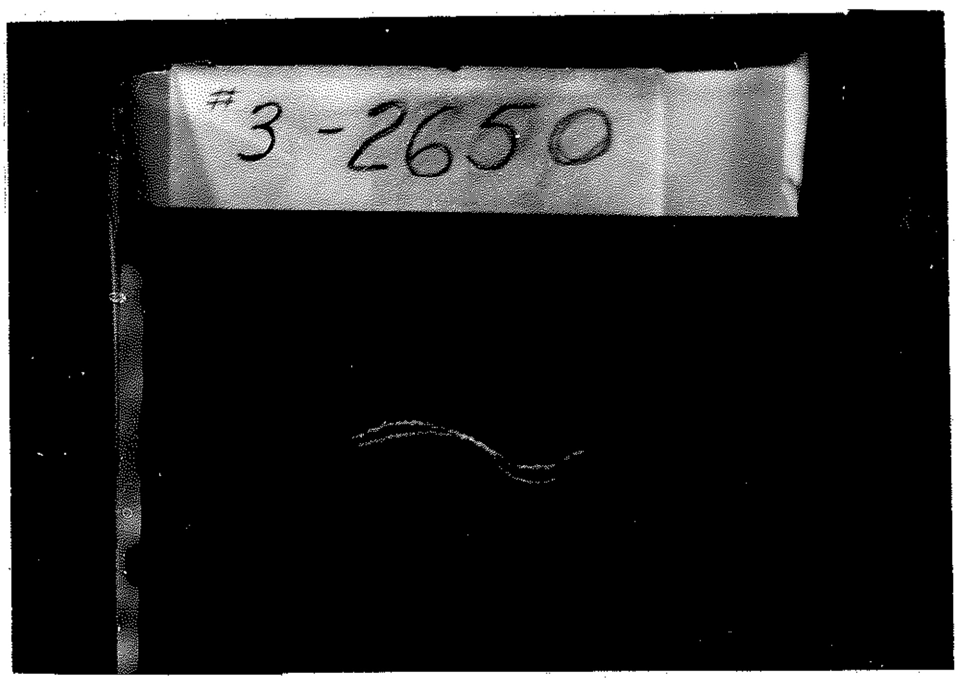
Oscilloscope Trace No. 9
Manifold No. 3. Speed 1900rpm. Torque 3.94ft. lbs.



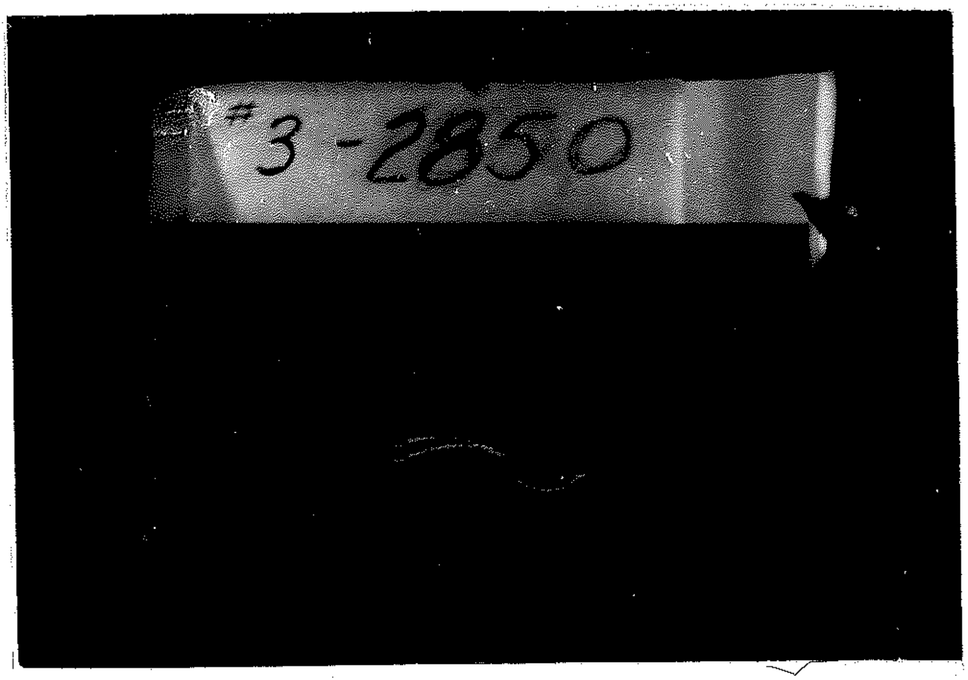
Oscilloscope Trace No. 10
Manifold No. 3. Speed 2200rpm. Torque 4.09ft. lbs.



Oscilloscope Trace No. 11
Manifold No. 3. Speed 2500rpm. Torque 4.07ft. lbs.



Oscilloscope Trace No. 12
Manifold No. 3 Speed 2650rpm. Torque 3.97ft. lbs.



Oscilloscope Trace No. 13
Manifold No. 3. Speed 2850rpm. Torque 3.88ft. lbs.

Oscilloscope Trace No. 14
Manifold No. 3. Speed 3000rpm. Torque 3.91ft. lbs.

STATEMENT OF RESULTS

When manifold No. 2 was completed a series of runs was taken. The data for this and the torque and horsepower curves are shown on pages 29 through 33.

From this, the run with no carburetor has an increase in horsepower at high speeds and did not fall off as fast as did the run with the engine equipped as manufactured. The shape of the two curves is somewhat the same otherwise. The distention of the no-manifold curve is to be expected due to the lack of friction and air-filter effect.

The horsepower curve for the tuned manifold showed a definite change had been wrought in the horsepower curve. It dropped off to low values in the range from 2800-3150 rpm.

This was somewhat puzzling due to the fact that the calculated resonant frequency of this manifold was calculated to be 3120 cpm. Due to uncertainty as to what was happening inside the manifold it was next decided to add the pressuregraph so that a pressure pattern inside the manifold could be observed.

Also since manifold No. 2 had given a power increase at speeds just below the limits of the engine used, it was wondered if lowering the resonant speed would show that the power increase would drop back to normal power at higher than was obtainable speeds. Since manifold No. 2 was a rigid structure, manifold No. 3 was built so that its dimensions could be varied at will. With the 8-inch column used and the cylindrical volume having a length of 8 inches the frequency of this system should be 2540 cpm.

The graphs on page 33 shows the horsepower curves for no manifold and for manifold No. 3 with 8-inch column and 8-inch length of cylinder for the cavity. Power increases are now noted at approximately 2300 rpm and above 3000 rpm for manifold No. 3. Minimum of relative power for manifold No. 3 is from 2500-2900 rpm. This again did not correlate with the calculated frequency of the resonant chamber.

Then the pressuregraph was brought into use and the series of oscillograph traces shown on pages 38 to 45. were studied. These showed that the resonator was giving a cycle per engine revolution for manifold No. 2 at between 3000 and 3200 rpm. Also the manifold was cycling three times per two engine revolutions at between 1900-2200 rpm. These agreed quite closely with the calculated values of the frequency.

In the case of manifold No. 3 the oscillograph traces showed a cycle per engine revolution at about 2650 rpm. At this point the linear distance on the oscilloscope trace of the positive and negative surges is the same. This is a true indication of resonance since it means the time of each surge is the same and therefore the vacuum created by the intake stroke of the engine is not overlapping and affecting the period of one of the other surges. This corresponds to a calculated frequency of 2540 cpm.

The calculated frequency for three pressure surges for every two engine revolutions would be 1700 cpm, and if the value of 2650 cpm for fundamental frequency is taken as correct then 1770 cpm would be the next lower frequency. Looking at the oscilloscope traces with manifold No. 3 at speeds of 1900 and 2200 rpm, it can be seen that at 1900 rpm the resonant frequency had been passed, since the middle loop which is the loop that disappears at higher speeds is already smaller than the other loops. Trying to interpolate at what speed the distance between loops would be equal from the above data is probably useless since the non-linearity of the oscilloscope screen increases at the extremities of the trace. But it seems that at 1900 rpm the trace is not too far from resonance. From that it seems that a frequency of resonance of 1770 cpm is more logical than a frequency of 1700 cpm. Actually it is a hard point to determine.

analog of a Helmholtz resonator is a simple L-C series circuit where the pressure at the end is proportional to the current flowing in the circuit. The current is largest when the circuit is operated at resonant frequencies, since at this frequency of operation the resistance of the circuit is lowest.

The theory proposed previously assumed that by using a driving force of the correct frequency, the actuation desired would be accomplished. Considering the resistance in the resonator to be zero, there would be no decay of oscillations due to the resonator. This is not too far from true as the oscillograph traces do not show much variation of height at the lower frequencies; but this resistance must be present in a small degree. Thus the base current in the resonator can be shown as a simple sine wave, but while the intake valve is open the energy to overcome the energy losses must be imparted to the resonator. As the engine was run for the tests included in this thesis the intake valve opened at 10 degrees after top center. To simplify the above, the following diagram is presented:

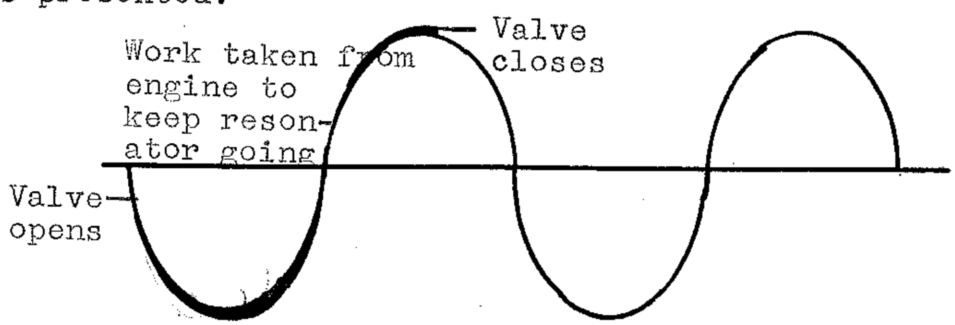


Figure 2

The base line is the pressure present in an ideal resonator with no losses. The shaded area represents the amount of energy required to keep the resonator oscillating. Thus it appears that it is impossible when operating in the manner shown by the diagram above to get any additional air into the cylinder as was proposed, since the resonator as shown in the above diagram simply follows the negative pulse of the engine and uses up energy in oscillation the air in the cavity and at the neck of the column.

Upon realizing this, an alternate method of getting the results desired is proposed. It operates in somewhat the same manner as a class-C amplifier. First, delay the opening of the intake port until a vacuum of appreciable magnitude is present in the cylinder. Now superimpose the results on the same base-current diagram as shown above; thereby obtaining the following diagram:

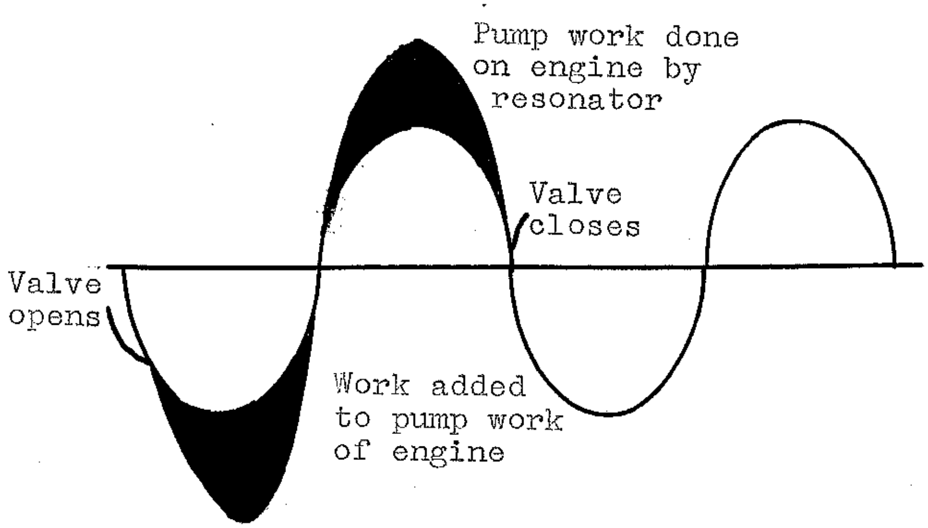


Figure 3

This shows a radical change in operation and needs further discussion. Now with the valve opening to show a large negative pressure, this pressure would soon decrease as air entered. Thus this large depression would show at the bottom of the negative loop, and with valve-opening period of about 240 degrees, as is common, the valve would close with about atmospheric pressure on it. However, all the area above the base line on the pressure surge would be work done by the resonator to pump air into the cylinder.

Now it is granted that the area of the additional suction of the driving pip and energy dissipated in the resonator is probably equal in area to the pump work added to the engine. But the pressure of the air pumped into the cylinder is considerably above atmospheric and would give the supercharging effect desired. Thus with this system it should be possible to increase the horsepower of the engine at resonant speeds. To emphasize the fact that the pressure in the cylinder should be high, is the fact that near closing of the valve the area open for gas to flow out is small. This should force more air into the cylinder than might at first seem evident by looking at the diagram.

The decrease in power experienced at resonant speeds is due to the additional pump work needed to keep the large surges in the manifold. The friction of these surges is probably noticeable at resonant speeds. This decrease in power is not as noticeable at resonance with three resonator cycles per two engine revolutions. This is to be expected since all of the positive pressure wave would be completed while the intake valve is open. This would allow more air at positive pressure to be added to the cylinder.

The reason for having an increase in horsepower at other than resonant speeds was deduced from observation more than recorded data showed. At other than resonant frequencies the pressure trace on the oscillograph screen showed a lowering in height of the pressure surge. This brings to mind a low-pass filter (which the L-C circuit would comprise at low, other than resonant frequencies) and this is probably the explanation. Thus the resonator filters a smooth pressure into the carburetor. This is compared to the uneven gas flow into the carburetor when no manifold was used, as shown by the fact that continually fuel was being sprayed back out of the carburetor in spurts.

Why the calculated frequency for manifold No. 3 is low is probably due to the shape of the cone connecting the column and the cavity. In the case of manifold No. 2 the cone had much more taper than did the cone in manifold No. 3. Thus probably all of the gas in the cone in manifold No. 2 was utilized in the stiffness of the cavity.

As for manifold No. 3, it probably had too much flare to utilize all of the gas present in the conical volume. The error in the calculation is not too great so that the formula can be assumed to be close to correct.

At this point an analysis of the aims of this thesis is proper. The first aim was to study the effect of placing a Helmholz-type resonator in the intake-manifold system. The effects are shown quite well in the oscillograph traces reproduced in this thesis and the horsepower curves on pages 33&37 shown in the section of test data.

The second aim was to show that an increase in horsepower could be obtained at resonant speeds of the engine with a tuned intake manifold. This increase in horsepower was never accomplished in the laboratory. Since, in the long run, it is only the physical results obtained that are positive proof, the second aim of this thesis has not been accomplished. However an analysis of why the anticipated power increase was never obtained, and a discussion on how this power increase should be obtainable is included.

The third aim of this thesis was to present a mathematical method of determining the resonant frequency of a tuned-manifold, and to facilitate the design of a tuned manifold. The formula derived in the INTRODUCTORY DISCUSSION fulfills this objective.

CALCULATION OF RESONANT FREQUENCIES

Due to the shape of the manifolds used some question arises as to how to calculate a value for the volume of the cavity. In the following calculations the volume of the cylindrical section is given exactly and the volume of the conical section is given as for a cone of the height which is measurable. This will give but a small error.

Manifold No. 2.

$$f = \frac{c}{2\pi}\sqrt{\frac{a}{LV}} \times 60$$

$$c = 1130$$

$$a = \frac{\pi}{4}\left(\frac{1}{12}\right)^2$$

$$L = 6/12$$

$$V = \frac{\pi}{4}\left(\frac{6}{12}\right)^2 \cdot \frac{6.5}{12} + 1/3\ \pi/4 \left(\frac{6}{12}\right)^2 \cdot \frac{4.5}{12}$$

$$f = \frac{1130}{6.28}\sqrt{\frac{\pi/4\left(\frac{1}{12}\right)^2}{\left[\frac{\pi}{4}\left(\frac{6}{12}\right)^2\frac{6.5}{12} + \frac{1}{3}\frac{\pi}{4}\left(\frac{6}{12}\right)^2\frac{4.5}{12}\right]\frac{6}{12}}} \cdot 60$$

$$f = 3120 \text{ cpm.}$$

Manifold No. 3

$$f = \frac{1130}{6.28}\sqrt{\frac{\pi/4\left(\frac{1}{12}\right)^2}{\left[\pi/4\left(\frac{6}{12}\right)^2\frac{8}{12} + \frac{1}{3}\frac{\pi}{4}\left(\frac{6}{12}\right)^2 \cdot \frac{3.2}{12}\right] \cdot \frac{8}{12}}} \times 60$$

$$f = 2540 \text{ cpm}$$

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