AN EXPERIMENTAL ANALYSIS OF INTAKE MANIFOLD TUNING OF FOUR-STROKE CYCLE MULTI-CYLINDER DIESEL ENGINES

by

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Thesis submitted in partial fulfilment

of the requirements for the degree of

Master of Science

The Faculty of Graduate Studies and Research

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Winnipeg

March 1966

ABSTRACT

This thesis presents the results and method used to investiage 'intake manifold tuning', or 'ram induction', in a four-stroke cycle, six cylinder diesel engine. It is mainly concerned with the determination of the fundamental dynamics of a tuned intake process and the application of the principles determined thereon to the tuning of multi-cylinder engines. The wave motion in the intake tubes was analysed by studying the pressure variation in the intake port during a complete cycle for various tube lengths at constant engine speeds.

The increased volumetric efficiency was found to be mainly due to the action of the rarefaction wave, initiated in the inlet port, at the beginning of the inlet process. The wave traveled to the manifold end of the tube where it was reflected and returned as a compression wave to produce a positive pressure at the ending of the same inlet process to force a greater amount of air into the cylinder. The timing of the return of the wave was found to be the important factor and from it a correlation was developed between the optimum tube length, engine speed, and average wave velocity. The timing of the standing wave produced in the inlet tube by the preceding cycle was also found to be important in producing a reinforced wave action during the inlet process.

The effect of the volume of the manifold joining the tubes and the use of bell-mouthed tube ends was also studied. The practicability of tuning is considered in a preliminary study of curved tubes. A gain as high as 15% in the volumetric efficiency of a multi-cylinder engine can be obtained over a wide range of speeds by a suitable choice of inlet pipes and manifold.

ACKNOWLEDGEMENTS

The author wishes to acknowledge the generous assistance of the technical staff of the Mechanical Engineering Department of the University of Manitoba in the construction and operation of the apparatus. Appreciation is also expressed to those who provided guidance and assistance in many aspects of the study.

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NOMENCLATURE

The main symbols used are listed below.

a - velocity of sound, ft. per second

L - total length of pipe, ft.

N - speed of engine, rpm.

P - port pressure, psig.

g - acceleration of gravity, ft/sec^2

K - ratio of constant pressure specific heat of air to constant volume specific heat

R - gas constant, $ft\text{-}lb/lb_m\text{-}^{\circ}R$

T - temperature of air, $^{\circ}R$

Greek Symbols

ϕ - delay angle, degrees of crankshaft rotation

I INTRODUCTION

It is well known that the volumetric efficiency of diesel engines and similar reciprocating machines can be markedly increased by tuning the intake system. This is accomplished by proportioning the intake manifold tube length for the operating conditions so that the air vibrations and waves set up in them increase the air flow rate into the engine. The volumetric efficiency can be increased by more than 15 percent at no expense of mechanical power to the engine and the power output can be correspondingly increased.

Although many engines have been successfully tuned and the subject has been extensively investigated the entire mechanism of intake manifold tuning is not fully agreed upon. Many papers have been written on the subject which attribute the gain in volumetric efficiency to different factors. There is much agreement with the theory that the increase is due to vibrations of the air particles in the tube much the same as the vibrations of an air column in an organ pipe. However they do not agree on exactly how the vibrations cause the increase in flow and an accurate correlation between manifold tube length and the influencing factors is lacking. Most authors have limited their work to the analysis of single-cylinder engines and little work has been done on the tuning of multi-cylinder engines.

Experiments with single-cylinder engines have shown that by varying the length of the intake tube at a fixed engine speed an

optimum length can be found which produces pressure pulsations that augment the air flow into the engine producing a maximum volumetric efficiency. It is of interest to know if the tubes of optimum length of a multi-cylinder engine can be joined by a manifold and if the volume of the manifold has an effect on the tuning process. It is also of particular interest to know the effect such an arrangement would have on the volumetric efficiency at speeds other than the fixed speed at which the tubes were an optimum.

A general relationship between the optimum tube length, and engine speed and other influencing factors would be useful for tuning the intake systems of other engines.

The use of modern pressure measuring devices such as piezoelectric crystal pressure transducers make it possible to measure rapidly changing pressures with great accuracy and precision. By installing such a device in the intake port near the intake valve it is possible to obtain a graph of the pressure during the process from which the wave motion in the tube can be analysed. The necessary pressure conditions in the port for maximum volumetric efficiency can be determined. A six-cylinder four-stroke diesel engine was equipped with such a device to undertake an investigation with the following objectives.

(1) To do an experimental analysis of intake tuning to determine the factors responsible for the gain in volumetric efficiency of a tuned intake system.

(2) To develop a general relationship between tube length and engine speed and other influencing factors responsible for maximum volumetric efficiency.

(3) To determine if a multi-cylinder engine can be successfully tuned.

(4) To determine if the size of manifold joining the tubes has an influence on the process.

(5) To determine the effect of tuning over the speed range of the engine when the manifold is tuned for a particular speed.

II GENERAL

2.1 Volumetric Efficiency and Engine Performance

A measure of an engine's efficiency or ability to pump air is the volumetric efficiency which is defined as the ratio of the mass of air actually inducted into the engine per cycle to the mass of air that would be inducted if the cylinders were filled to atmospheric pressure at the end of the intake process. Due to the effects of fluid friction and the dynamics of the intake process the volumetric efficiency is always less than 100% for a naturally aspirated engine. Fluid friction reduces the volumetric efficiency by reducing the pressure of the air entering the cylinder. The drag on the air as it is drawn through the manifold and valves causes a pressure loss which reduces the amount entering the cylinder. The dynamics of the intake system also has an important effect on the volumetric efficiency. The pressure in the cylinder must be lower than that in the intake port in order for air to flow into the cylinder during the short period of time for the intake process. Thus at the end of the process the pressure in the cylinder must necessarily be lower than atmospheric and the volumetric efficiency less than the optimum of 100%.

Good engine performance is consistent with a high volumetric efficiency. The power output is almost directly dependent on the volumetric efficiency. The greater the amount of air an engine can induct the greater the amount of fuel that can be burned in the combustion process. It is the energy

produced by the combustion process that determines the power of the engine. Work is expended during the scavenging loop which is the portion of the cycle during which exhaust gases are expelled and fresh air is inducted into the engine. When a greater amount of air is inducted the average pressure during the intake process is higher and the work lost is lower. The decrease in work lost during the scavenging process increases the brake thermal efficiency by making available more shaft work at the same rate of fuel consumption.

The volumetric efficiency of naturally aspirated engines can be improved substantially by reducing the adverse effects of friction and by improving the dynamics of the intake process. Fluid friction can be minimized by making the valves and passages as large as possible and the passages as smooth as possible and by the use of relatively large radii where curves in the manifold are necessary. The dynamics may be improved by the use of an intake manifold which creates pressure waves in it that favour the flow into the engine. The waves produce a high positive pressure in the port during a portion of the process which increases the flow.

Improving the dynamics of the intake process is called intake manifold tuning, inertia supercharging, or induction ramming, and it has been shown by experiments that properly tuned systems can increase the volumetric efficiency by more than 15% of that obtained with a simple short tube type of manifold. The intake system is tuned by the use of a manifold having long tubes leading to the cylinders. Usually, but not always, one tube supplies air to one cylinder. The length and diameter of the tubes are determined in part by the speed of the engine or speed range at which maximum

volumetric efficiency is desired and the thermodynamic properties of the air. The intake process creates pressure waves and air vibrations in the long tubes which augment the air flow into the cylinder. The function of the tubes is to control and time the wave action and the vibrations of the surging air column within it during the intake process and to act as a storage device for some of the kinetic energy developed which can be used in the following cycle.

2.2 Preview and Scope

An accurate and direct correlation between tube length and operating conditions has not been developed for the four-stroke cycle engine and little work has been done on the tuning of multi-cylinder engines. Methods of tuning multi-cylinder engines have not evolved readily from the work done on single-cylinder engines although experiments have shown they can be tuned. This is perhaps an indication of a lack of fundamental understanding of the tuned intake process.

This thesis is concerned mainly with developing a fundamental understanding of the tuned process and finding an explanation for the method by which the waves in the tuned manifold produce the gain in volumetric efficiency. The experimental approach was to analyse the wave motion in the tubes as indicated by the graph of the pressure in the intake port and to measure the volumetric efficiency for a range of tubes (including the optimum length) at a constant engine speed. A relationship between tube length and operating conditions was sought from the explanation and these principles were then used to tune a multi-cylinder engine

and determine its performance over the entire speed range of the engine. The process will be explained in terms of wave motion.

The scope of this thesis is limited to the above mentioned objectives and a theoretical mathematical analysis is not included in the study. To perform an accurate mathematical analysis a fundamental understanding of the process must first be developed in order to describe the process in mathematical terms. It is often necessary to make assumptions and approximations in such an analysis and the process must be well understood to assess the validity of each and the limits with which they apply.

A literature review is presented by which the development of the subject and different theories are described. The different theories on the subject are discussed with a view to finding an accurate fundamental explanation of the process and to determine the present status of the subject. This is followed by an explanation of the experimental apparatus and procedure followed in the experiments. The theory of the process, based on experimental investigation and on the review of literature, is explained before the discussion of results in order to provide the reader with an insight to the subject before going into the details of the study.

III REVIEW OF LITERATURE

A review of the published material on the subject has revealed that intake manifold tuning has been extensively investigated and that the actual mechanism of the tuned process is not fully agreed upon. An accurate direct correlation between the optimum tube length and engine operating conditions has not been developed for the four stroke cycle engine. Theoretical analyses of the process have produced methods of calculating the volumetric efficiency of an engine with a given intake system at particular operating conditions but are unwieldy and do not provide a direct relationship between intake tube length and engine operating conditions. Many of these are based on simplifying assumptions and assumed conditions which do not accurately describe the process. Considerable experimental data has been compiled mostly for the purpose of determining the potential available by tuning and developing a basic understanding of the phenomena. Some of the publications of a more basic nature (including those which present accepted theories) will be discussed in this chapter.

3.1 Resonance Theory

A theory on the process presented early in the history of investigations of tuning was that of resonance. If the intake tube is long the periodic opening of the intake valve and consequent induction of air through it can set up an oscillation of the air column in its natural frequency. Theoretically maximum gain is attained when the natural frequency of the intake tube is an exact

multiple of the engine speed or frequency of the intake valve. Capetti (3) claimed that maximum gain would be attained when the period of free oscillation of the column is equal to the time required for the crankshaft to turn through 180 degrees for a single cylinder engine. Based on this theory he gives a formula for the optimum length of intake pipe.

$$L = 7.5 \frac{a}{N}$$

in which

L = Length of pipe, ft.

a = Velocity of sound, ft. per sec.

N = RPM of the engine

This equation is equivalent to saying that the frequency of the tube is 4 times the engine speed or equal to the fourth harmonic of the engine. Tests conducted by Capetti indicated that the optimum lengths were less than those given by the formula.

It has also been shown that a gain will be realized when the frequency ratio is in the range of 3, 4 and 5 and larger gains occur with tubes slightly shorter than when the frequency ratio is an interger. Morse et al (5) argue that the amplitude of the standing wave at the inlet port will be large if the frequency ratio is an interger, and if the pressure waves produce a positive pressure during the portion of the intake process when the valve is closing, then, and only then, will a gain be realized. Using these basic assumptions they developed a theoretical analysis for the pressure waves in the tube which enabled them to predict when a supercharging effect will occur. Their theory is in accordance with experiment

in that it predicts that a gain could only occur with frequency ratios of 3, 4 and 5.

Although the resonance theory predicts when a gain will occur it does not explain the process nor does it account for several results. It does not explain the gain obtainable with a long pipe over that with no pipe when the frequency ratio is not an interger, or well between intergers, and that a larger gain is always obtained with pipes shorter than when the frequency ratio is an interger. An examination of the cycle shows that resonance is not an exact description of the tuned process because a free standing wave does not occur until the valve has closed. During the period when the valve is open the intake process produces pressure waves which dominate the wave action in the tube until the valve closes. It is only possible for a standing wave (true resonance condition) to contribute to the gain at the beginning of the next process. It can produce a positive pressure in the port at the beginning of the next process but this has never been found to be solely responsible for the gain since large gains can be obtained when a negative pressure wave is timed to arrive at the port during valve opening.

3.2 Inertial Ramming Theory

In the inertial ramming theory the inlet pipe is considered to be a device for storing and regulating kinetic energy in the intake air. Dennison (3) noted that considerable gain could be obtained by pipes much shorter than those anticipated from the resonance theory. He theorized that the gain was produced by accelerating the air column to a high velocity in the first half of the suction stroke; the kinetic energy so developed is then

expended in the latter part of the process to force a surplus of air into the cylinder.

He reasoned that as the air in the intake tube flows toward the cylinder during the beginning of the intake process kinetic energy is stored in the air column by virtue of its velocity. Some of this energy can be recovered at the ending of the inlet process by the ramming effect of the air column as the air is decelerated at the valve end by the increasing cylinder pressure. The ramming effect builds up the pressure in the port forcing more air into the cylinder as the valve is closing. By this theory the gain is also produced by a positive pressure at the ending of the intake process.

Based on this theory Dennison developed a method of calculating the pressure at the valve port at any time during the cycle, from which knowing the flow resistance characteristics of the intake valve, the volumetric efficiency for a given system can be calculated by a step by step integration process. This theoretical analysis has not proved to be in good agreement with experimental data (Taylor) and the relationship of the ramming effect to engine design and operating conditions was not developed.

The basic theory of the process is sound but the theoretical analysis has not been accurately developed. The theoretical solution was based on the simplifying assumptions of incompressible and uniform flow along the length of the pipe. Since the pressure fluctuations at the intake port have been measured to be as much as half an atmosphere and resonance conditions are known to be set up in the pipe during portions of the engine cycle, these assumptions appear to be oversimplifications.

3.3 Recent Experimental Investigations

Thus far, two theories have been presented on the process; that of resonance and that of inertial ramming effects, both of which have been for the most part independently credited with being responsible for the gain. Noting these two theories Taylor et al (6) conducted extensive tests, and have compiled what may be the most extensive collection of data on the subject available, to ascertain the validity of each theory and the extent to which each may be responsible for the gain. He concluded that the gain produced by the long tubes is the result of two distinct causes; the kinetic energy inertial ramming effect and the standing wave produced after the valve closes. He also concluded that the inertial ramming effect was mainly responsible for the gain and the standing wave had minor effects only at particular conditions.

Although in his tests he attributed the gain to inertial ramming effects he was unable to find a direct method of correlating the intake tube length with the engine operating conditions. The experimental procedure employed was to vary the engine speed for a fixed intake system and observe the air flow rate for marked change. It is difficult to find the optimum length of tube for a given engine speed by this method because the standing wave is capable of producing pronounced effects on the air flow at several different speeds and the air flow rate changes with speed. The characteristics of the engine itself which change with speed also introduce complications.

In performing a theoretical analysis Taylor considered the flow to be compressible and nonuniform along the length of the tube. He was able to calculate the pressure in the port for any instant

and plot the results by the use of a digital computer program of his mathematical analysis. However in his explanation of the process and experimental results he bases much of the results on the effects of inertia with little regard for the effects of elasticity on the motion of the air in the tube.

3.4 The Reflected Wave Theory

Since compressibility must be considered with inertia in describing the motion of the air in the tube the motion is probably best described in terms of waves. Due to compressibility and inertia the air in the port nearest the valve will be accelerated first and the pressure will decrease as the air begins to flow into the cylinder at the beginning of the process. The inertia of the air will permit only air adjacent to the low pressure area to be accelerated toward the valve and the low pressure area will travel outward toward the end of the tube as the air moves toward the low pressure area. The combined effects of elasticity and inertia create a travelling rarefaction wave at the beginning of the intake process.

In a more recent investigation of intake tuning Bannister (1) conducted tests on a single cylinder air compressor and did a theoretical analysis of the process explaining it in terms of wave motion. He found that the pulsating inflow through the intake pipe of the compressor was due to the combined effect of two superimposed oppositely moving wave trains. The wave motion originates as a rarefaction created by the motion of the piston away from the head end at the beginning of the intake process. This rarefaction wave

travels at approximately sonic velocity toward the open end of the tube where it is reflected in the form of a compression wave and returns to the port while the piston is still in motion. Upon returning to the port the compression wave creates a high pressure during the ending of the intake process which forces more air into the cylinder. The pipe also contains a residual wave system in the form of a standing wave created by preceding cycles and the new pulse is added to and merges with the outgoing wave train of this system. It is the first reflection of the new rarefaction which returns to the valve port as a compression which is mainly responsible for the gain in volumetric efficiency.

Bannister found that the reflected wave must return to the port approximately 80 degrees of crankshaft rotation after generation to produce a maximum gain for the compressor tested. The angle through which the crankshaft rotates while the new wave pulse makes a double traverse of the tube is referred to as the delay angle and the theory of the process will be referred to henceforth as the reflected wave theory. This theory takes into account the effects of compressibility and inertia of the air column and provides a simple correlation between operating conditions and intake tube length. The effect produced by the waves is similar to the inertial ramming effect described by Dennison. A high pressure is created by the high velocity inflow of air in the tube at the end of the process. The development of the ramming effect is more accurately described by the reflected wave theory and takes into account all of the major factors which affect the air flow during the process.

The cycle for a four-stroke cycle engine differs from that of the air compressor in that the compressor intakes every revolution whereas the four-stroke cycle engine intakes every second revolution. The high delivery pressure causes a delay in the actual beginning of the intake process due to the re-expansion of the air in the head space, hence the timing of the new rarefaction is delayed in the case of the compressor. In both cases, wave theories appear to be the most applicable method of analysing and explaining the tuned intake process, and in view of the advantages of the wave theory approach it appears to be the most suitable.

IV EXPERIMENTAL ARRANGEMENTS

The experiments were performed in the Mechanical Engineering laboratory of the University of Manitoba using a 6 cylinder four-stroke cycle Caterpillar diesel engine having a displacement of 743 cu. in., a bore of 5 1/8 in., and a 6 in. stroke. The engine had originally been a supercharged model with a compression ratio of 17.4:1, an intake valve opening timing of 25°BTC and closing at 64°ABC. The supercharger was removed for all tests, but the stock manifold was used for initial tests to obtain a comparison for following tests.

4.1 Inlet System

The engine had individual intake ports to which were fitted tubes of approximately the same diameter for the tuning tests. The tubes were 2.15 in., in diameter and the lengths could be changed in 1.5 in. increments, but 3 in. increments were used for preliminary tests. The tubes were connected to a variable volume manifold which was connected by a 4 in. diameter pipe to a large plenum as shown schematically in Fig. 1a.

The air flow into the large plenum chamber was measured by a calibrated 3 in. flow nozzle. The nozzle was made of wood and was oiled to prevent drying and cracking. It was calibrated by placing it in the exit of a low velocity wind tunnel and measuring the static pressure drop across it and the corresponding velocity pressure profiles. The flows were calculated and the mass rate of flow and pressure drop were plotted in a graph of the dimensionless parameters affecting the flow which is shown in Appendix I.

The purpose of the large plenum chamber was to dampen out pressure pulsations from the intake to obtain a steady flow through the flow nozzle. The chamber was 70.5 in. long, 27 in. wide and 20.5 in. high, and contained a volume approximately 52.5 times the engine displacement. The flow nozzle was mounted in the top approximately 19 in. from one end and the outlet to the manifold was also located in the top about 17 in. from the other end.

Fig. 1 b shows the variable volume manifold which was constructed of plywood with a plexiglass top. All joints were sealed with plastic wood to prevent leakage. The air inlet was located in the top at the center of the manifold as near the side in which the outlet pipes were located as possible. The volume of the manifold was varied by sliding a movable panel opposite the outlet side, back or forth, and could be set at any volume and as large as 5.75 times engine displacement. A second manifold, shown in Fig. 1 c, was built and tested consisting of a plywood box with plexiglass top which was of fixed volume. It was 39 in. long, 6 in. high and 4 in. wide, containing a volume approximately 1.26 times that of engine displacement. The outlets to the intake pipes consisted of bellmouths. The inlet was a 4 in. pipe located at the center of the top of the manifold.

4.2 Pressure Measurement

The pressure at the intake valve was measured by a Kistler model 601 quartz crystal pressure transducer mounted in the intake port near the valve. To get the probe as near the valve as possible a small frost plug was removed from the top of the head directly

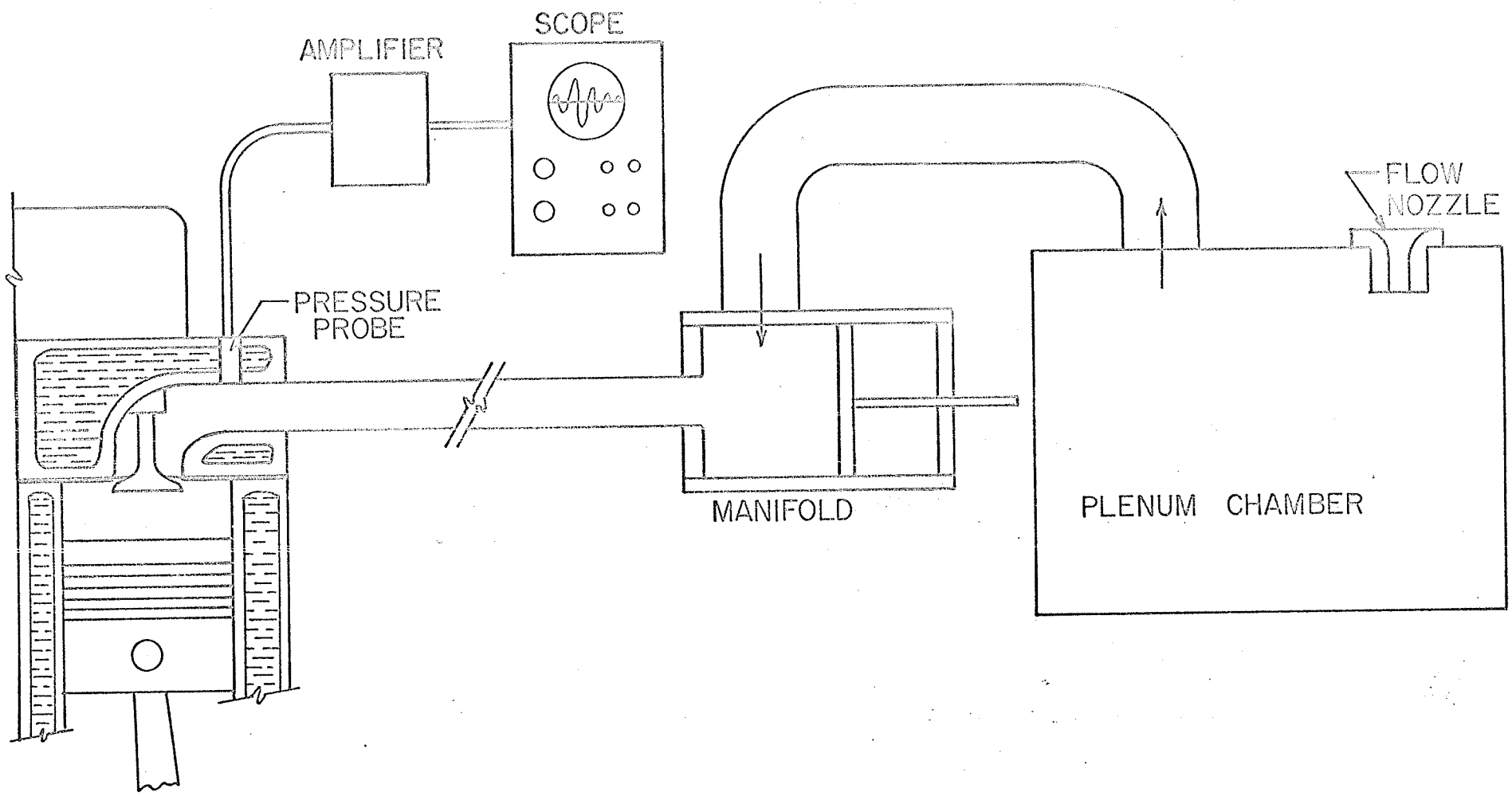


FIG. 1a

SCHEMATIC OF TEST APPARATUS.

above the port and a hole was drilled through the port wall. The hole was tapped with 1/4 in. pipe thread and an adaptor was made which would screw into the hole and seal the frost plug hole in the top. A Kistler model 624 probe adaptor sleeve was supported in the adaptor by two rubber rings so that the crystal was flush with the inner surface of the port. This permitted the static pressure in the port to be measured and the rubber mounting reduced the vibrations from the engine which may have affected the signal.

The signal produced by the pressure on the quartz crystal was transmitted by a Kistler model 671 cable to a model 566 charge amplifier which transmitted the signal to an oscilloscope. The signal produced a trace of the varying pressure on the oscilloscope. Top dead centre near the beginning of the intake process was indicated on the oscilloscope by the signal from a synchromarker driven by an auxillary shaft of the engine at half engine speed. This arrangement made it possible to obtain a graph of the pressure in the intake port during a complete cycle of the intake valve. A polaroid camera was used to photograph the trace. For further details see Appendix IV.

4.3 Ancillary Equipment

The power output of the engine was absorbed and measured by a Froude type hydraulic dynamometer. Speed measurements were made by a stroboscope (see Appendix II). The fuel consumption rate was determined by measuring the time required to consume a fixed weight of fuel. The coolant water temperature was maintained nearly constant for all tests by mixing with water from the building water supply in a mixing tank. A water cooled heat exchanger was used to maintain the temperature of the lubricating oil nearly constant.

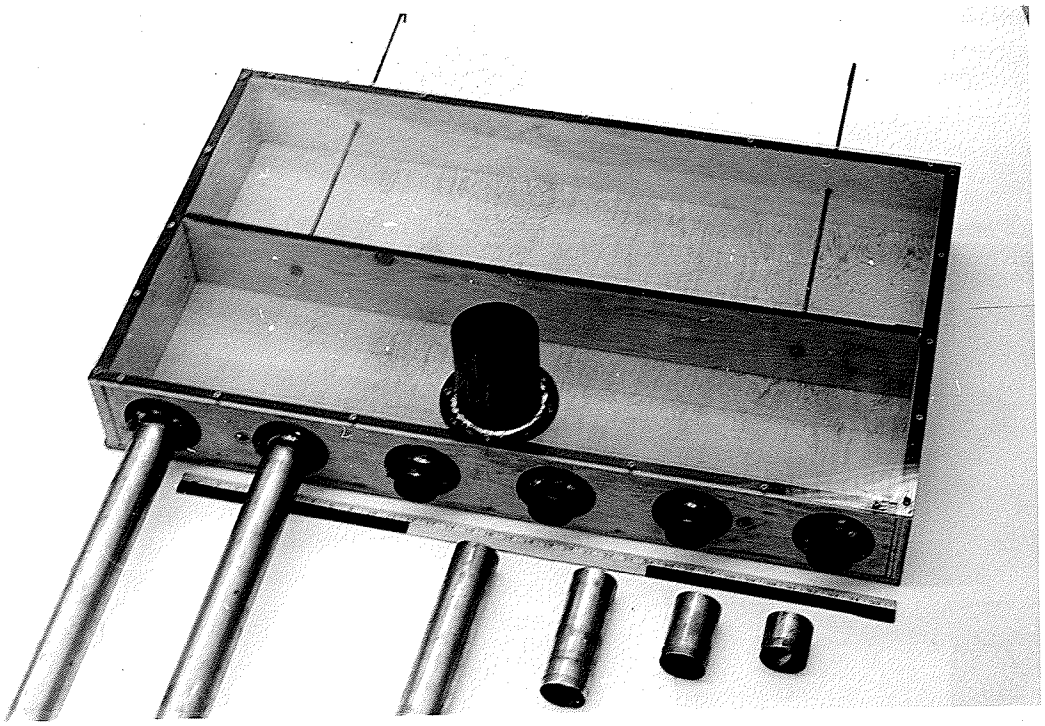


Figure 1b

Variable Volume Manifold

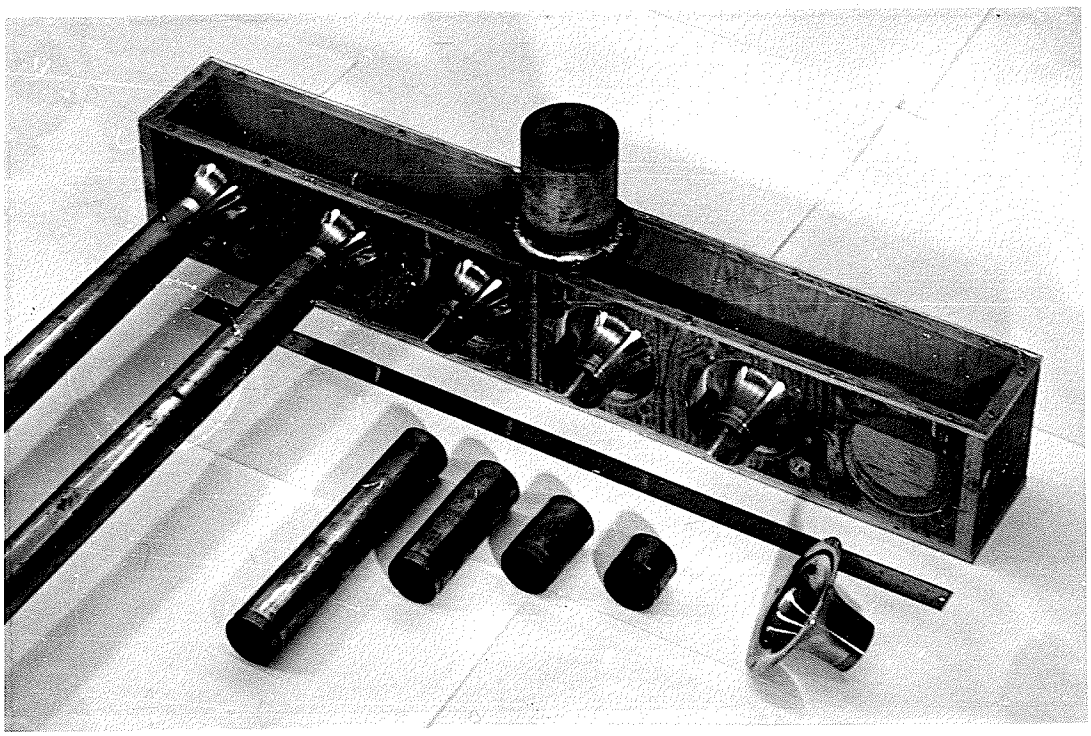


Figure 1c

Bell-Mouthed Outlet Manifold

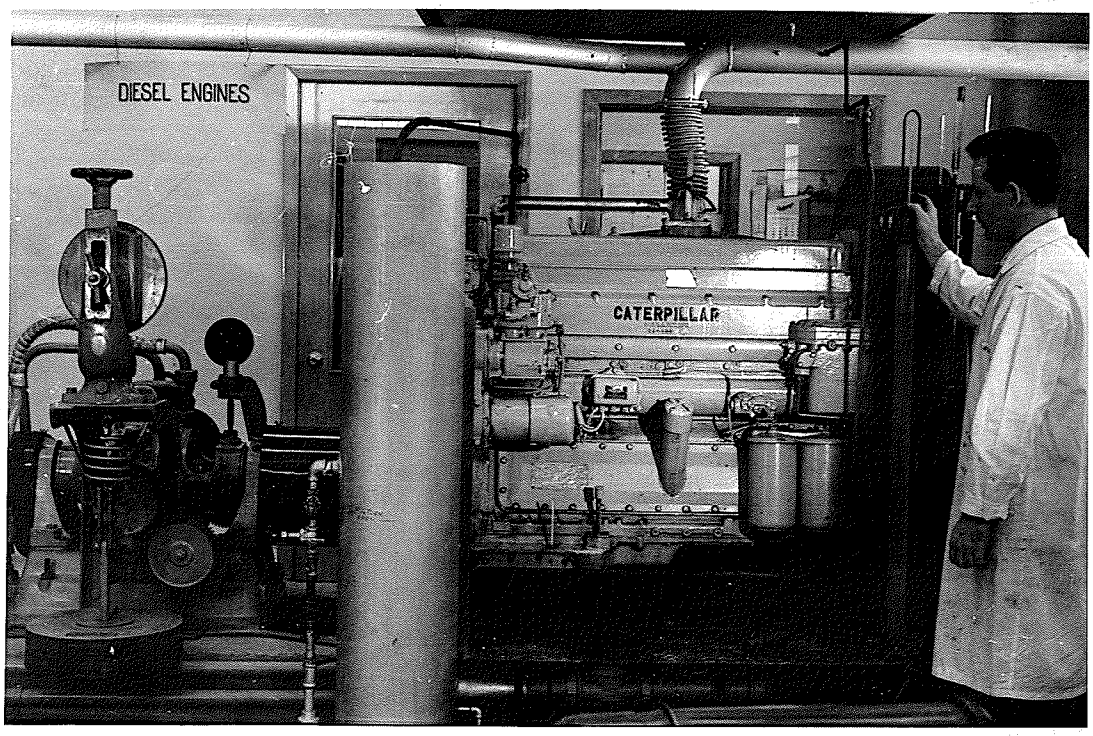


Figure 1d

Engine Before Tests - Right View

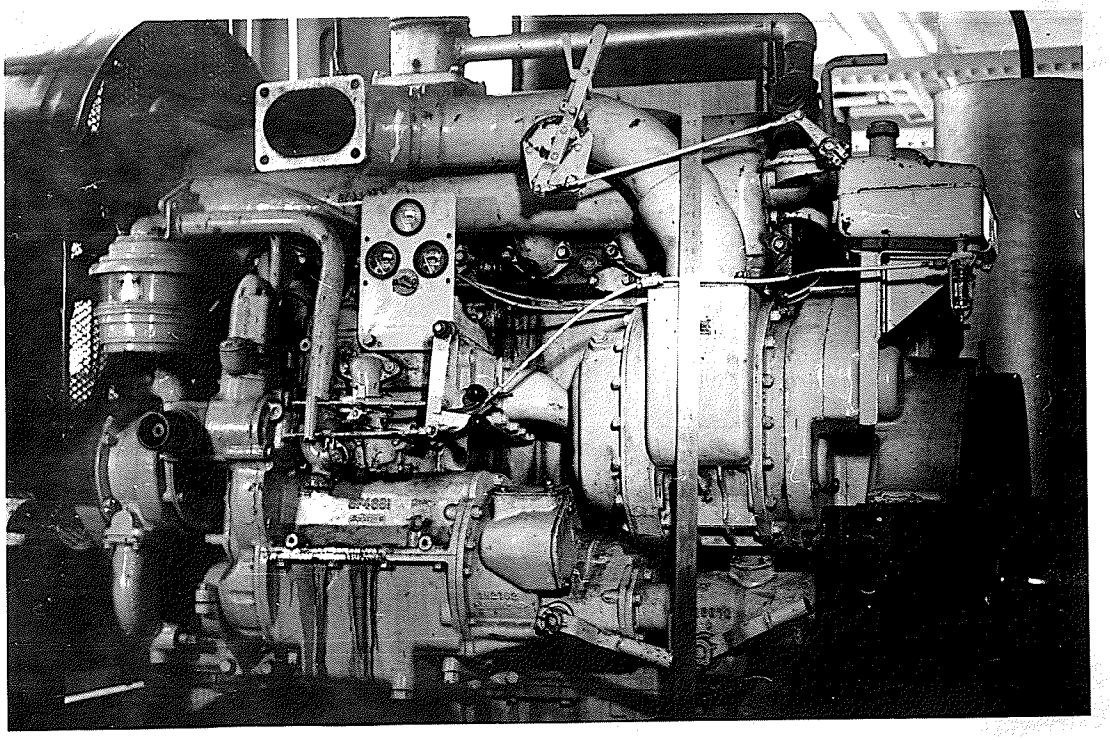


Figure 1e

Engine Before Tests - Left View

V EXPERIMENTAL PROCEDURE

5.1 Preliminary Tests

The engine was first tested with the stock manifold to obtain a comparison for further tests. The stock manifold had very short tubes leading from the main gallery to the ports and it was assumed that negligible ramming was being produced. The stock manifold was connected to the large plenum chamber by a 4 in. pipe to enable intake air flow rate measurements.

Tests were run over the speed range of the engine from 1200 to 1900 rpm. Readings taken included manometer readings of the pressure drop across the flow nozzle, brake load, temperature of inlet air, time to consume a fixed weight of fuel and barometric pressure. The temperature of the cooling water and oil were maintained nearly constant at 175°F and 240°F respectively. A sample of exhaust gas was taken during each test which was analysed by an Orsat apparatus to determine the carbon dioxide and oxygen content from which could be determined the air-fuel ratio as a check on the air metering system. A photograph of the pressure graph traced on the oscilloscope and the blip indicating valve opening was taken.

From the data obtained it was possible to calculate the brake horsepower, torque, brake specific fuel consumption, and the volumetric efficiency for each test, all of which were plotted on a graph versus speed. All tests were performed with the fuel injection pumps set at a fixed constant displacement by eliminating the governing mechanism so that the amount of fuel injected per cycle was approximately

constant for each speed. The air-fuel ratio used was approximately 19:1 which was the ratio obtained when the engine was equipped with the supercharger.

5.2 Tuning the Engine

It was decided to tune the intake system for a speed of 1800 rpm. This speed was chosen for two reasons; it is well in the speed range of nearly all diesel engines and was near the upper limit for the engine tested; it is the speed at which alternating current machinery is driven in this country for which diesel engines find great use. With the fuel injection pumps at a constant setting the speed was adjusted by varying the load imposed by the dynamometer. A stroboscope set at line frequency indicated when the engine speed stabilized at 1800 rpm.

In attempting to tune the intake system the stock manifold was removed and tubes were fitted to the intake port in which the pressure probe was located. The end of the tube was left open, no attempt being made at this time to measure the air flow into the engine, and the pressure trace on the oscilloscope was observed as the length of the tube was increased by adding short lengths. The addition of the long lengths of tubing to the port produced a pressuregraph having an entirely different shape from that obtained with the stock manifold. The trace took on the shape of approximately a sine curve beginning with the pressure decreasing at the opening of the intake valve, reaching a minimum, and then rising to a maximum and decreasing to oscillate in the form of a sine curve of decreasing amplitude.

The pipe length was varied until the pressure oscillations produced a trace indicating the pressure was at a maximum when the

intake valve was beginning to open. At this length the frequency of vibrations in the tube was almost 4 times the frequency of the valve opening. The variable volume manifold was installed and the pipes which connected it to the other cylinders. The manifold inlet was connected to the large plenum chamber enabling flow measurements to be made. The installation of the manifold at its largest volume setting was found to have no effect upon the pressure trace.

A series of test were then performed in which the intake tube length was varied in 3 in. increments in a range 6 in. longer and shorter than the length which produced a pressure maximum at the beginning of the process. The air flow for each pipe length was measured and the length which produced the highest vol. eff. was then used in a test in which the volume of the manifold was varied. The engine speed was held constant and the manometer was observed as the volume was decreased. The air flow and trace did not change as the volume was decreased until the manifold was less than 3 in. wide. At widths less than this, the movable wall began to restrict the air flow into the manifold and the manometer indicated a decrease in air flow. It was concluded that the volume of the manifold was not an important factor in the process and a width of 4 in. was chosen for the remaining tests. This was a practical size of manifold and the volume was 1.05 times the displacement of the engine.

With the manifold fixed at a width of 4 in. and the engine speed held constant at 1800 rpm, a complete set of tests were performed in which the tube length was varied in 3 in. increments from 3.25 feet to 6.25 feet. This range of lengths included the lengths at which the resonant frequencies of the tubes were 3, 4 and 5 times

the frequency of the intake process. Where necessary, the tube length was changed in increments of 1.5 in. to produce special pressure conditions at the beginning and end of the intake process. After each test was completed the volume of the manifold was varied and the manometer and pressure graph observed for changes to check for any effects the volume may have at other lengths. Photographs of the pressure trace were taken for each tube length tested.

The tubes were then fixed at the length which produced a maximum volumetric efficiency at 1800 rpm and tests were performed to determine their effect over the entire speed range of the engine. The engine speed was varied from 1400 to 1900 by varying the load in successive tests in which a complete set of data were collected with pictures of the pressure trace at each speed.

To improve the wave action by increasing the efficiency of wave reflection at the manifold the variable volume manifold was replaced by the fixed volume manifold with bellmouthed outlets. A series of tests were then run with lengths varying about that at which maximum volumetric efficiency was obtained with the plain end tubes.

To determine the effect of the tube diameter on the wave action a 1.875 in. diameter tube was flared at one end to fit into the port. The tube was fitted into the port as close to the valve as possible to minimize the effects that may be produced by the change in diameter from port to tube upon the wave action. The engine was held at a constant speed and the pressure trace observed as the tube length was varied in 3 in. increments over the range previously tested. The increment was reduced to 1.5 in. wherever pressure trace forms

of special interest occurred. No attempt was made to measure the air flow into the engine.

A tube slightly shorter than optimum length was bent in a 90 degree curve at a 20 in. radius close to the valve end to determine the effects of curves on the wave action. The radius of curvature to tube diameter ratio was 9.3. The tube was fitted to the engine with the open end turned down. The speed was held constant at 1800 rpm and the pressure trace observed as the length of the curved tube was varied in 1.5 in. increments.

VI THEORY

6.1 The Intake Process Without Tuning

The untuned intake process is described first to show the contrast between the dynamics of tuned and untuned systems and to point out the shortcomings and inefficiencies of the untuned process. The ways in which the process is inefficient will be emphasized with a view as to how they can be remedied or improved by tuning.

The intake process begins by the opening of the valve and the motion of the piston away from the head end. Air is drawn into the cylinder through the valve by the low pressure in the cylinder created by the volume displaced by the moving piston. The air in the port near the valve and in the manifold is accelerated to a high velocity during the process and the pressure is lowered corresponding to the energy conversion from pressure energy to kinetic. The maximum pressure decrease in the port occurs approximately when the piston is at its highest velocity which is near midstroke.

The low pressure in the port is maintained throughout the process and the flow in the manifold experiences disorderly pulsation of low amplitude. At the end of the process the air is brought to a halt by the rise in pressure in the cylinder as the piston reaches the end of its stroke and by the closing valve. The velocity energy of the air is dissipated in the form of turbulence and high frequency vibrations due to the rapid deceleration.

Throughout the process the pressure in the port is maintained well below atmospheric and the pressure differential across

the valve is reduced. The air entering the cylinder is consequently at a low pressure which reduces the amount inducted and accounts for a less than optimum volumetric efficiency. The dynamics of the process dictate, therefore, that the air in the cylinder must be at a lower pressure than atmospheric at the end of the process and an amount of energy at the end of the process be unavailable to that process or the next.

6.2 The Tuned Intake Process

A tuned intake manifold having long tubes has a major effect on the dynamics of the intake process. The effect of the long tubes is mainly on the pressure in the port during the process and is due to their ability to transmit and reflect pressure waves. At the end of the process the tubes store kinetic energy in the form of a standing wave which can be used in the following process. One of the major functions of the tubes is to control and time the wave action so that the pressure pulsations they produce in the intake port augment the air flow into the engine.

The wave motion will be described in the following paragraphs with the aid of Figures 2 a - d, which show the relationship between the longitudinal waves and the port pressure. The figures are a plot of the pressure in the port at various crank angles and the pressure along the length of the intake tube. The pressure changes along the length of the tube due to the individual waves are shown rather than the resultant pressure along the tube in order to show the progress of the waves.

The wave motion is initiated by the opening of the intake valve and the motion of the piston away from the head end. The low

pressure in the cylinder accelerates air through the valve and subjects the tube to a low pressure at the valve end. Due to the elasticity and inertia of the air it is accelerated progressively along the length of the tube and a rarefaction wave is created which travels toward the manifold at approximately accoustic velocity. Figure 2 a, shows the development of the wave at the stage at which the leading edge has reached the manifold end of the tube. The amplitude of the wave increases as the motion of the piston increases and reaches a minimum rarefaction when the piston is at maximum velocity.

Upon reaching the manifold end (which is essentially an open end) the outward moving rarefaction causes a rapid surge of air into the tube creating an inward moving compression wave. That is, the rarefaction wave is reflected point by point as a compression wave at the open end. Figure 2 b, shows the inward travelling compression wave where the leading edge has reached the port. At this stage the still outward travelling rarefaction has reached a minimum pressure at the port. As the point of minimum rarefaction reaches the open end it is reflected as a point of maximum compression as shown in Figure 2 c.

The compression wave travels back toward the port (which due to the high resistance of the valve approximates a closed end) where it is reflected as an outward moving compression wave. The outward moving compression wave superimposes itself on the inward moving compression to create a positive pressure in the port as shown in Figure 2 c. The pressure reaches a peak when the maximum compression arrives at the port just before the closing of the valve (Fig. 2 d).

RELATIONSHIP BETWEEN WAVE MOTION AND PORT PRESSURE

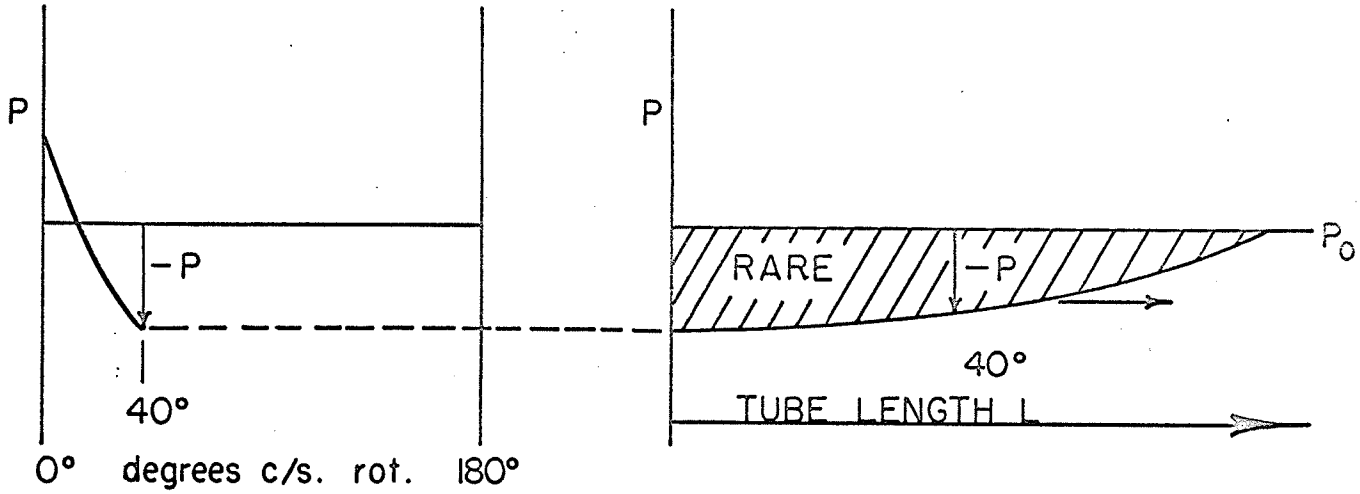


FIG. 2a

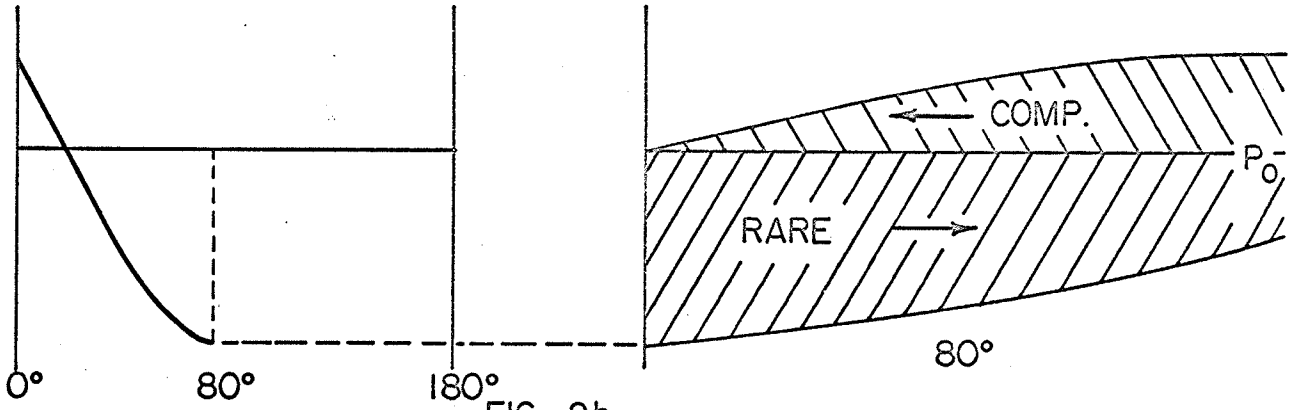


FIG. 2b

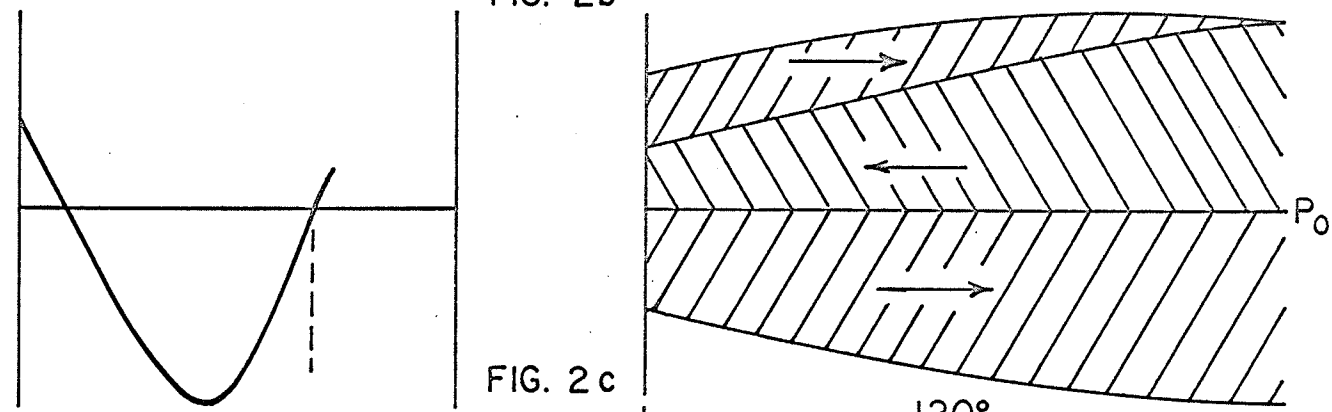


FIG. 2c

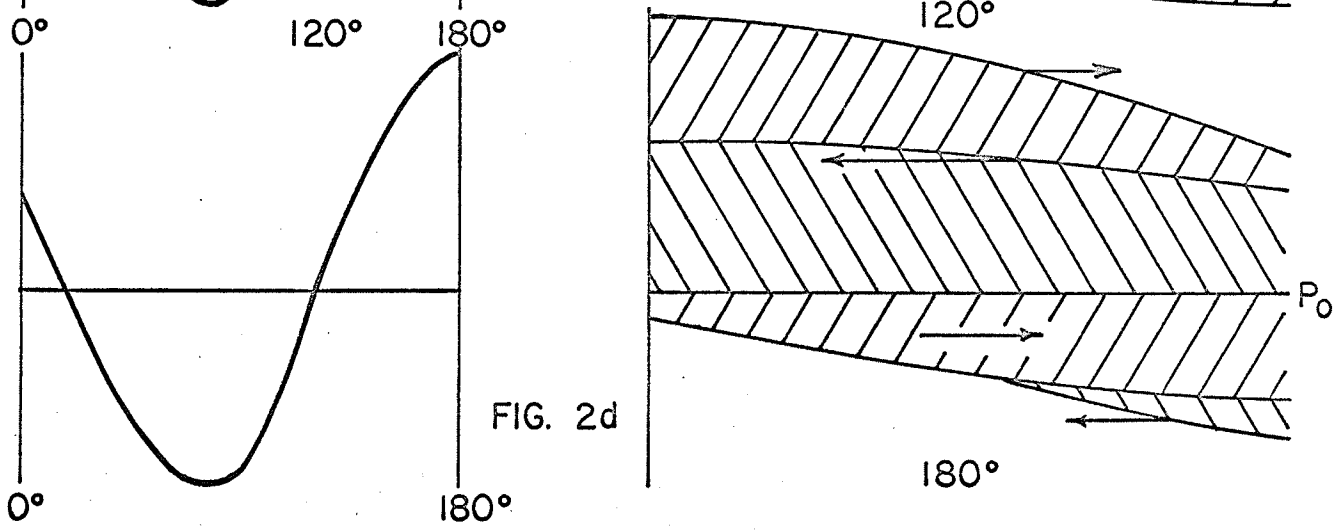


FIG. 2d

The superimpositioning of the two oppositely moving waves produces a standing wave or vibrating air column in the tube at the end of the process.

Immediately before the generation of each new rarefaction the intake tube contains a residual system of waves consisting of the superimposed multiple reflections of the waves generated in the preceding cycles. At the beginning of the intake process the new rarefaction adds to and merges with the outward moving rarefaction of this system. Propagation and reflection losses in the residual wave system reduce their magnitude and the new rarefaction becomes dominant. Reinforced by its merger with the standing wave, the rarefaction travels toward the manifold at approximately acoustic velocity as it is generated. It is the timely return of its first reflection to the port as a compression which is mainly responsible for the increase in volumetric efficiency by tuning.

After generation, the minimum rarefaction point of the wave must travel to the end of the tube and return to the port as a maximum compression in time to raise the pressure in the port to a peak just before the valve closes. The wave is then reflected and the combined effect of the two oppositely moving wave trains creates the standing wave which oscillates through three complete cycles before merging with the new rarefaction at the beginning of the next process. The merger of the standing wave with the new rarefaction increases the amplitude of the resulting rarefaction and the amplitude of the compression produced by its first reflection is correspondingly increased. The return of the wave to the port as a compression produces a high pressure in the port at the end of

the process which forces more air into the cylinder.

6.3 Correlation of Parameters

The angle through which the crankshaft rotates while the wave makes a double traverse of the tube is referred to as the delay angle. When the optimum angle is known and the wave velocity can be calculated, the optimum length of tube can be related to the engine speed. In calculating the time required for the rarefaction wave to travel to the end of the tube and back, the velocity of the wave was assumed to be acoustic. Although the waves are of the finite amplitude type and in calculating propagation velocities the finite amplitude wave laws should be used, the latter approximation is justified since with open end reflections changing rarefaction pulses into compressions and vice versa, the changes in wave profile and velocity occurring during outward travel are cancelled by those during inward travel. The average velocity of the wave during a double traverse of the tube is determined by the thermodynamic properties of the air from the small wave equation.

$$a = \sqrt{KgRT}$$

The time required for each wave point to make a double traverse of a tube length L would be $\frac{2L}{a}$ seconds. For an engine speed of N rpm, the angular speed of rotation is 6N degrees per second. The number of degrees of crank rotation required for the waves to make a double traverse of the tube would be

$$\phi = \frac{12NL}{a} \text{ degrees}$$

where ϕ is the delay angle. When $\phi = 83$ degrees the tube is an optimum length for the particular speed and acoustic velocity.

Rearranging the expression and substituting the optimum delay angle gives the equation for the optimum tube length.

$$L = 6.9 \frac{a}{N}$$

6.4 Tuning of Multi-Cylinder Engines

The intake systems of multi-cylinder diesel engines with individual intake ports can be successfully tuned provided there is no appreciable distortion of the wave action by the interference of waves amongst neighboring tubes, and the pressure in the manifold is not appreciably affected by the waves themselves. This can be accomplished by joining the tubes of optimum length by a rake type of manifold having approximately the same volume as the engine displacement. Manifolds having volumes as small as 0.75 times the engine displacement are equally as effective and are a thoroughly practical size.

The volume of the manifold has no influence upon the tuning process and serves only as a plenum chamber at which the waves in the tubes are reflected. The wave action in the tubes may be affected if the volume of the manifold was small enough to allow the waves to cause pressure fluctuations in the manifold similar to the case of a Helmholtz resonator. The manifold must also be sufficiently large enough not to propagate large pressure waves past the ends of the tubes and cause interference amongst the waves.

Since the reflected wave is mainly responsible for the increase in volumetric efficiency by tuning and acts only during the intake process, only 180 degrees of crank rotation per cylinder is required to achieve a substantial gain by tuning. Two cylinders

may be tuned by a common pipe provided the timing of the intake processes is not such that the standing wave produced at the end of the intake process of one cylinder does not nulify or greatly reduce the rarefaction at the beginning of the other. Two cylinders should therefore obtain a substantial gain from one pipe provided the crank angle between the beginnings of the processes is not less than 180 degrees.

6.5 Varying the Speed of the Engine

If the intake system is tuned for a speed near the upper limit of the engine's speed range, the tubes will perform as if shorter than optimum at lower speeds. This means that the reflected wave will return to the port in a smaller delay angle than the optimum of 83 degrees.

The return of the reflected wave at smaller delay angles can produce a gain provided the wave does not return too soon to produce a pressure above atmospheric at the end of the process. At delay angles less than optimum the timing of the standing wave may produce adverse effects. An angle is reached at which the merger of the standing wave with the new rarefaction reduces the amplitude of the rarefaction instead of reinforcing it. The resulting reflected wave is correspondingly reduced in amplitude and the pressure produced in the port by its return is lowered. The gain, though substantial, is slightly reduced over the range of delay angles at which this unfavourable merger occurs.

At higher speeds the tubes are longer than optimum and the delay angle is greater. The reflected wave cannot sufficiently

return before the closing of the valve to raise the pressure in the port high enough to produce as large a gain. At larger delay angles, a range is again reached at which the standing wave is not timed to reinforce the outgoing rarefaction at the beginning of the process, but rather produces a cancelling effect which reduces the rarefaction resulting from the merger. These two effects and increased friction losses cause a sharp decrease in volumetric efficiency at speeds slightly above the tuned speed.

VII EXPERIMENTAL RESULTS

7.1 Effects of Long Tubes

The addition of the long tubes to the intake ports completely changed the shape of the graph of the pressure in the port. The graph with the stock manifold indicated a sharp pressure drop during valve opening followed by a series of high frequency pressure fluctuations indicating high frequency air vibrations in the port and manifold. There seemed to be no orderly pattern followed by the vibrations and multiple traces produced by successive cycles produced a blurred line on the oscilloscope indicating the process was not precisely repetitive. The pressure trace obtained with the long tubes approximated the shape of a sine curve of diminishing amplitude beginning with the pressure decreasing at the opening of the valve, reaching a minimum, and then rising to a maximum and decreasing to oscillate with diminishing amplitude until the next cycle. At the beginning of each cycle the new rarefaction completely dominated the pressure in the port and the graph was repeated. The graph was completely repetitive and the multiple traces produced by successive cycles on the oscilloscope produced a clear curve indicating a high degree of orderliness in the pressure fluctuation produced by the long tubes.

One of the most conclusive results obtained early in the experiment was the apparent complete independence of intake tuning of the volume of the manifold joining the long tubes. When the tube was at a length having a natural frequency 4 times the

frequency of the valve and produced a pressure peak at the beginning of the process, the installation of the manifold had no effect on the pressuregraph indicating the tubes could be joined by a manifold without in any way affecting the wave motion. The tube length which produced a maximum volumetric efficiency in the range in which the frequency ratio was 4 with the manifold at its largest volume was used to determine the effect of decreasing the volume. The decrease in volume did not affect the air flow or pressuregraph until the manifold was less than 3 in. wide. The air flow decreased and the pressuregraph began to change. However at these narrow widths the sliding panel began to restrict the air inlet at the top of the manifold which would account for the decrease in flow. The remaining tests were performed with the volume set at a width of 4 in. and varying the volume at the end of each test was found to have no effect on the air flow or pressuregraph for the range of tubes tested. At widths of 3 in. and 4 in. the ratio of the volume to engine displacement was 0.78 and 1.05 respectively. These sizes are practical and are comparable to manifolds in common use.

Figure 3a shows the effect of tube length on the volumetric efficiency at a speed of 1800 rpm. The curve rises above the untuned value as the tube length increases and peaks 3 times before decreasing toward values approaching the untuned at very long tube lengths. The curve peaks first at 89% at a tube length of 3.5 feet after which there is a sudden but small decline at 3.75 feet to 87.7%. The curve continues to rise from this point to peak a second time at the maximum value of 92% between 4.25 and 4.5 feet. Between 4.5 and 4.75

feet the curve drops sharply to 87% but rises slightly at 5.5 feet to peak again at 87.25%. The curve then continues to decrease with increased tube lengths. The three peaks occurred at lengths slightly less than 3.8, 4.75 and 6.33 feet respectively, which are the lengths at which the resonant natural frequency of the tubes are even multiples of 5, 4 and 3 times the intake process respectively.

Figure 3b is the corresponding curve of the volumetric efficiency and the delay angle, $\phi = 12NL/a$, calculated from the engine speed, tube length and acoustic velocity of the air. The curve has the same shape as that of Figure 3a due to the linear relationship between tube length and delay angle. It shows the relationship of the volumetric efficiency to the delay angle which is essentially an indication of the timing of the return of the reflected wave. The curve shows the influence of the timing of the reflected wave on the volumetric efficiency and the affect of the timing of the standing wave on the performance of the reflected wave.

7.2 Pressuregraphs

Figures 4a to 4j are the pictures of the graph of the pressure in the intake port for a complete cycle for tube lengths from 4 feet to 6.25 feet. The pictures show the variation in port pressure during the intake process and the remainder of the cycle. Each figure contains two pictures of the same pressuregraph but on different time scales. One picture is on a time scale which includes the entire cycle and the other is on an expanded scale, double the rate of the first, so as to obtain a clear, more precise graph of the beginning and the intake portion of the cycle. A small blip, or wave in the horizontal line, indicates top dead centre after the beginning of the opening of the valve and near the beginning of the

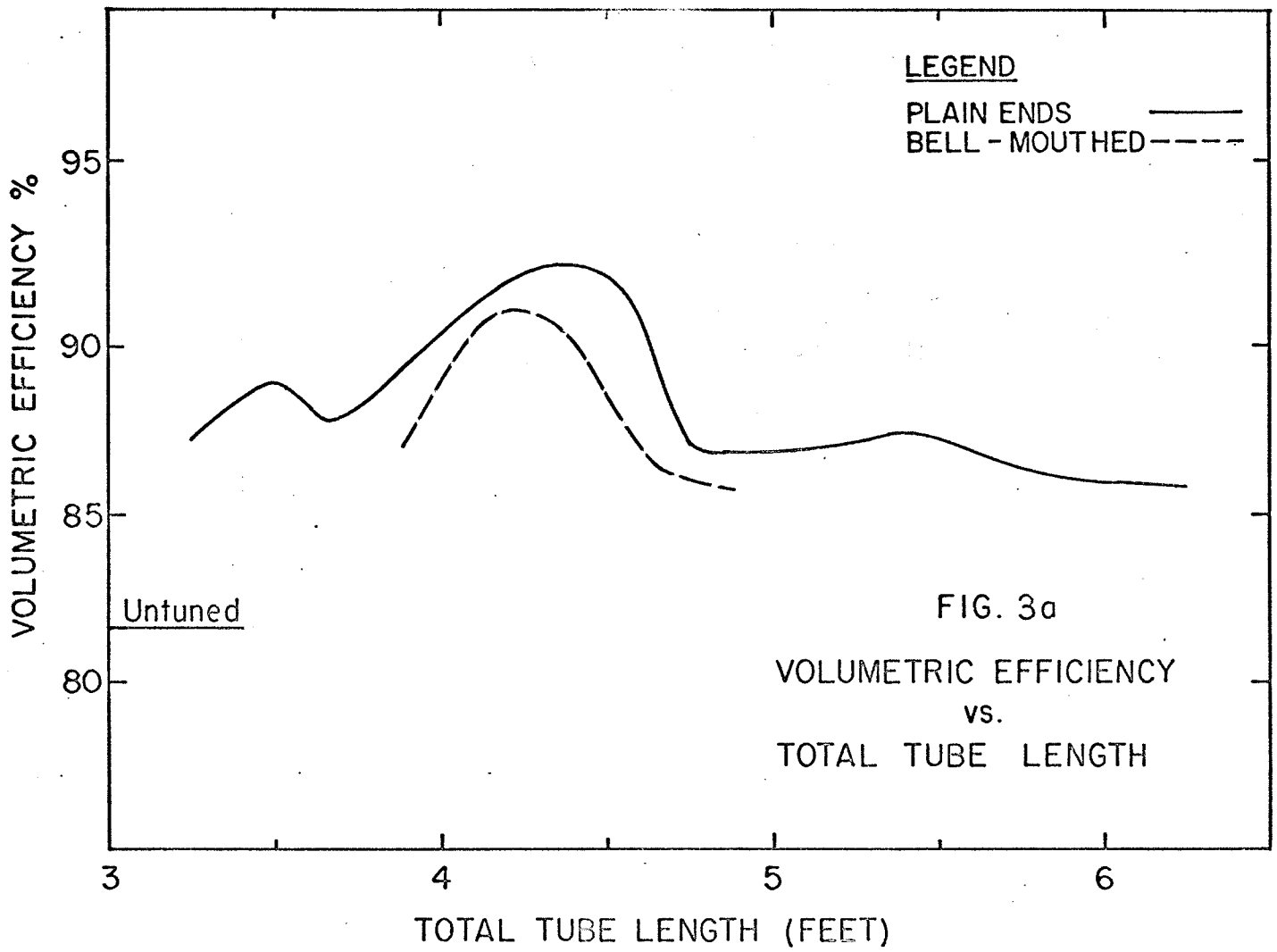


FIG. 3a

VOLUMETRIC EFFICIENCY
vs.
TOTAL TUBE LENGTH

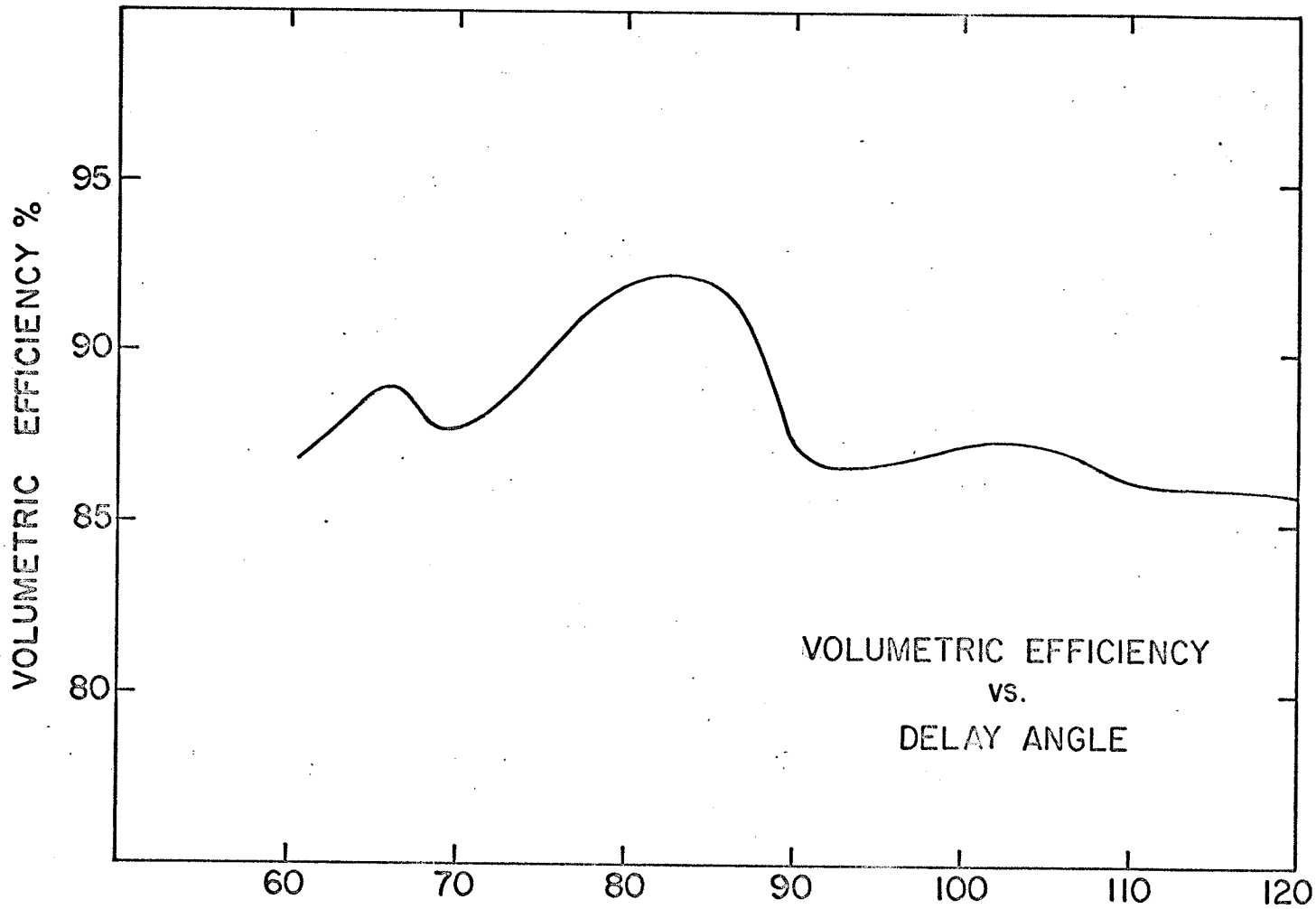


FIG. 3b DELAY ANGLE OF WAVE $(\phi) = \frac{12\,NL}{a}$

intake process. The valve lift chart is shown in Appendix III.

Each vertical division on the pressuregraphs represents 1.25 psi. and each horizontal division represents 100 degrees of crankshaft rotation. The graphs all exhibit a rapid decrease in pressure at the beginning of the intake process followed by a rise to a high peak above atmospheric near the end. The pressure then oscillates with decreasing amplitude until the beginning of the next process. However there are pronounced differences in pressure conditions at valve opening, differences in rarefaction shape and in the timing and shape of the compression.

The pressure conditions at the beginning of the intake process indicated by Figures 4a to 4j, show the affect of the standing wave on the reflected wave and account for the shape of the volumetric efficiency curves. Consider Figures 4b, c and d, which are the graphs for the pressure in the port for the tube lengths 4.25, 4.38 and 4.5 feet respectively, which produced maximum gain. The pressure is at a peak compression and is starting to decrease when the valve begins to open. The rarefaction which follows this peak is timed to be coincident with the new rarefaction generated by the beginning of the next process and is superimposed on it. The superpositioning of the rarefaction of the standing wave and the new rarefaction produces a resultant rarefaction of greater amplitude and the compression produced by its first reflection is correspondingly increased. Upon its return to the port the compression produces a high pressure during the latter part of the process to force more air into the cylinder.

Figure 4e is the pressuregraph for the 4.75 foot length of

tube which is slightly longer than those which produced the maximum gain. This length produced a sharp decrease in volumetric efficiency and is the first of a series of tubes which caused the second depression in the volumetric efficiency curves. The standing wave is so timed by this tube that the pressure is rising from a rarefaction and at nearly neutral pressure when the new rarefaction begins. The compression peak of the standing wave is coincident with the beginning of the new rarefaction and is superimposed on it. This unfavorable merger of the two waves has a nulifying effect on the new rarefaction which reduces its amplitude and changes its profile. Its leading edge is so reduced that a delay occurs before pressures of significant amplitude began to leave the port as the leading edge of the resultant rarefaction. The leading edge of the compression produced by its first reflection is correspondingly delayed and the rise in pressure produced by its arrival occurs later in the process. Due to the greater delay angle of the long tube, the maximum point of the reflected wave returns later in the process to produce a pressure peak when the flow resistance of the valve is high.

The first depression in the curves is also caused by an unfavorable merger between the standing wave and the new rarefaction due to the timing of the standing wave produced by the associated tubes. The first peak in the curves is the result of a favorable merger between the two waves but the return of the compression is not well timed. The small delay angle of the shorter tubes permits a rapid return of the leading edge of the new rarefaction as a compression which is superimposed on the tail of the rarefaction

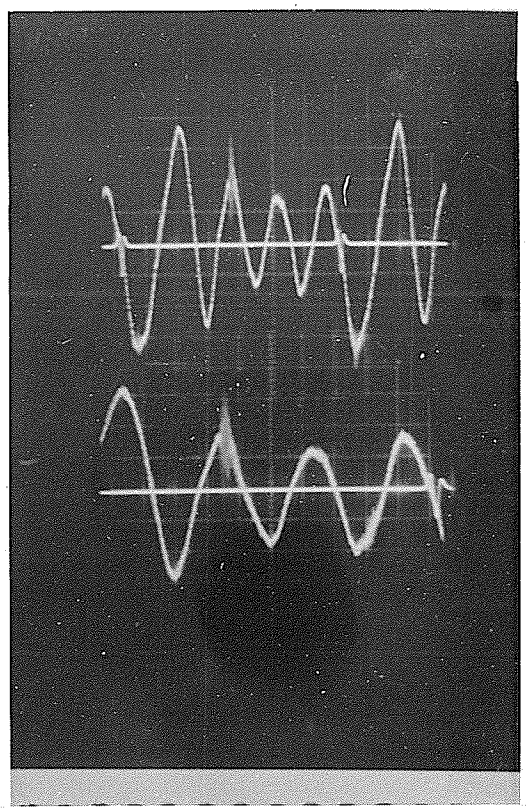


Fig. 4a
L = 4.00 ft.

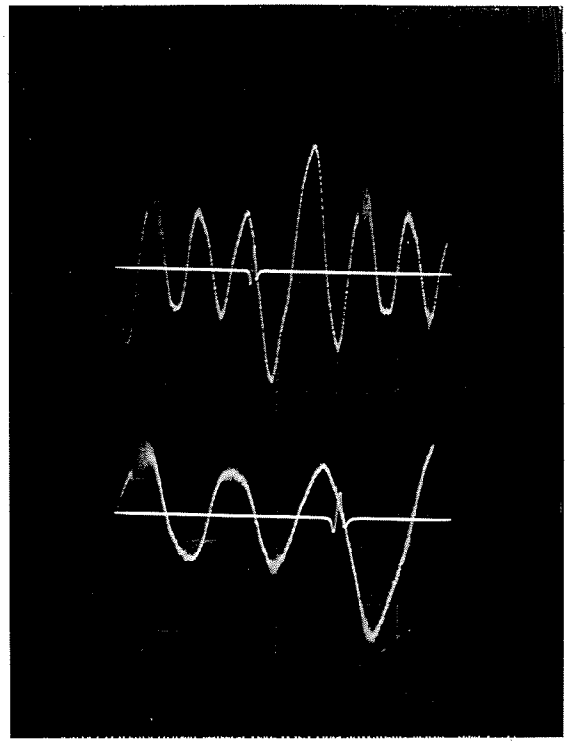


Fig. 4b
L = 4.25 ft.

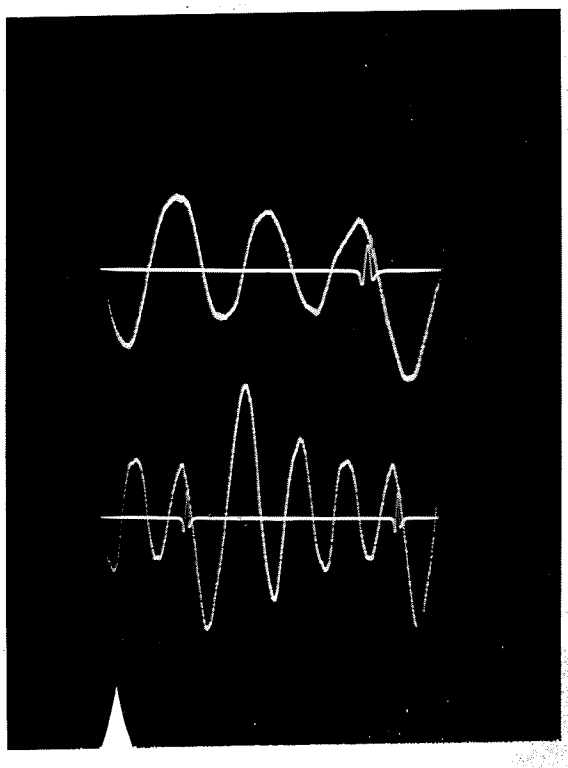


Fig. 4c
L = 4.37 ft.

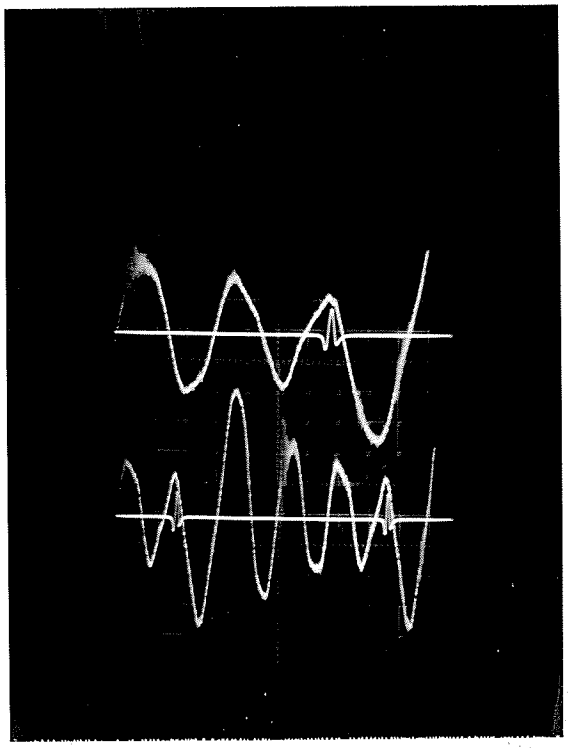


Fig. 4d
L = 4.50 ft.

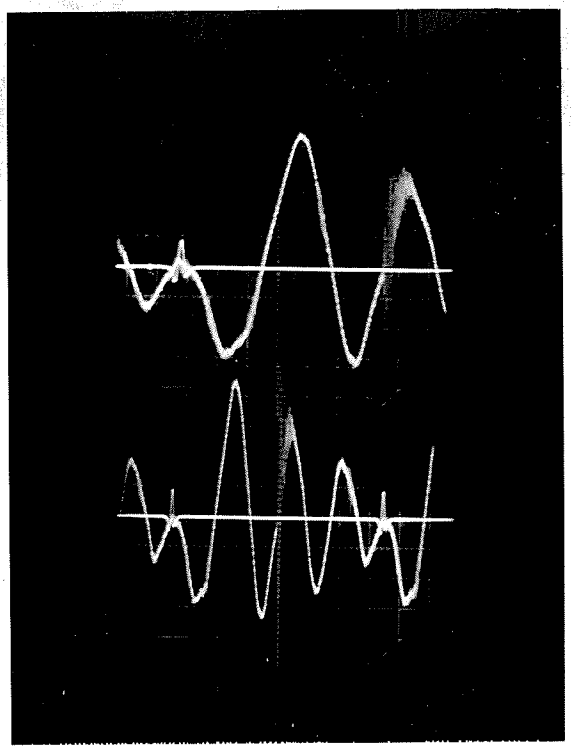


Fig. 4e
L = 4.75 ft.

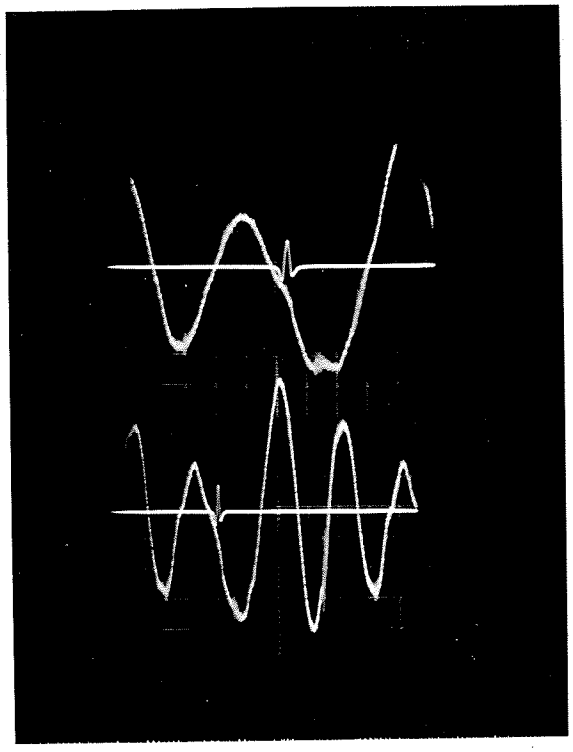


Fig. 4f
L = 5.25 ft.

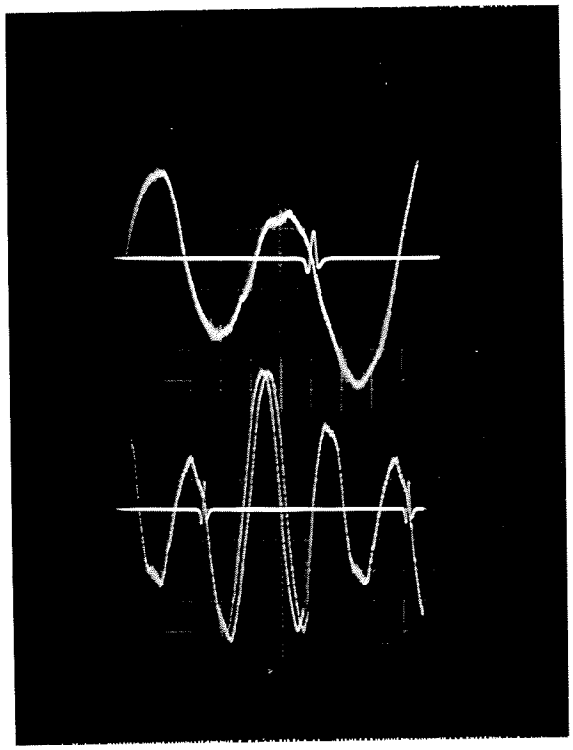


Fig. 4g
L = 5.50 ft.

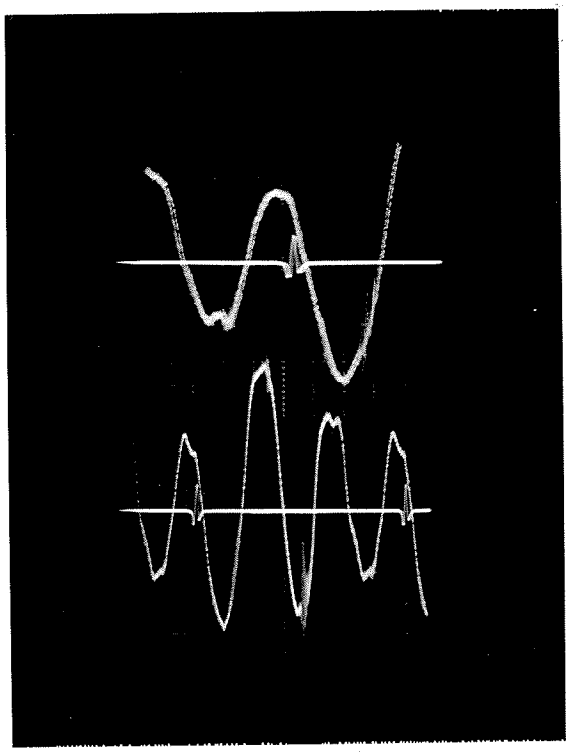


Fig. 4h
L = 5.75 ft.

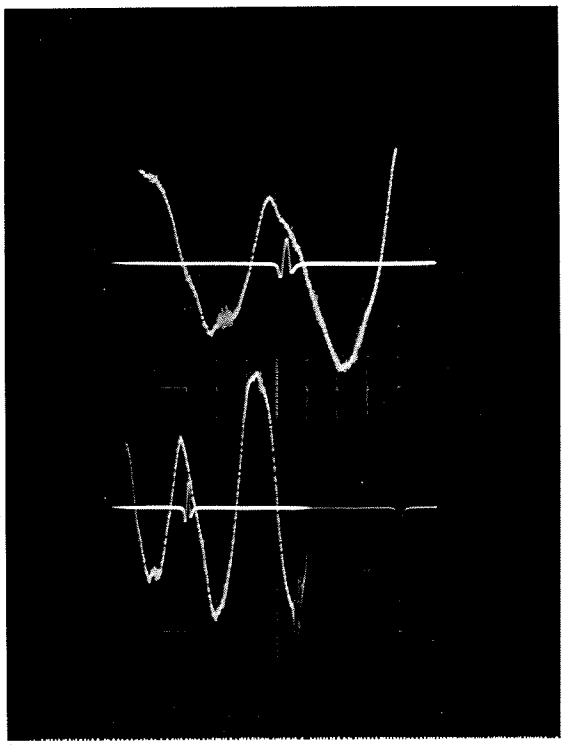


Fig. 4i
L = 6.00 ft.

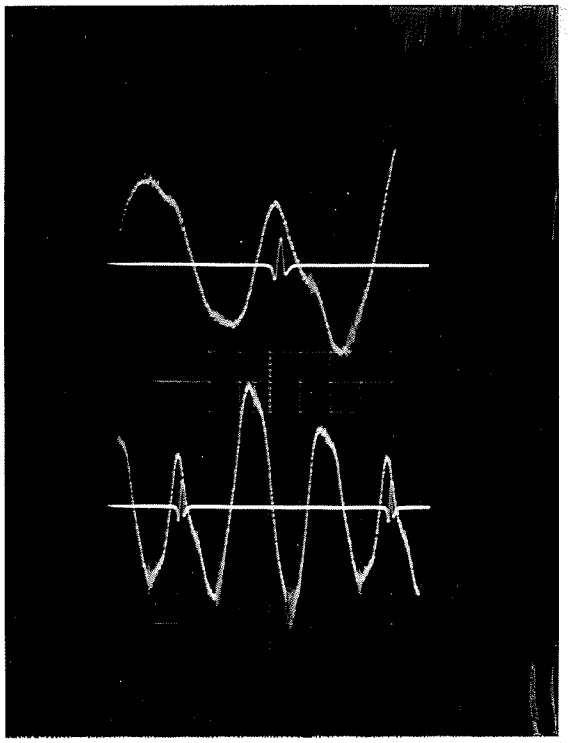


Fig. 4j
L = 6.25 ft.

which is still being generated. The resultant wave is a compression of reduced amplitude which produces a lower pressure peak in the port during the later part of the intake process. Consequently the first peak in the volumetric efficiency curves is lower than the second.

7.3 Varying the Speed of the Engine

Figure 5 is a graph of the results of varying the speed of the engine from 1400 to 1900 rpm, using the optimum length of intake tube for 1800 rpm. The results obtained with the stock manifold are included for comparison purposes. The brake horsepower and torque curves are higher and the brake specific fuel consumption curve is lower than those for the stock manifold for all speeds. The volumetric efficiency curve for the tuned system was higher for all speeds and the difference between the two curves increased as the speed decreased below 1800 rpm, and decreased at speeds above 1800 rpm.

The higher power and torque curves and lower fuel consumption curve may be attributed to two possible effects caused by the increased flow. The first being the reduction in pumping losses produced by more efficient induction and, the second, the effects on the combustion process obtained with the higher air-fuel ratios. Due to the larger amount of air inducted, the average pressure in the cylinder during the induction process is higher which reduces the area of the scavenging loop of the cycle and the pumping losses. More work is available as shaft horsepower at the same rate of fuel consumption. The increased air-fuel ratio improves combustion

efficiency by making available more air for combustion and for more thorough mixing. Combustion temperatures are lowered by the excess air resulting in a lower rate of heat transfer to the cooling system.

Figure 5 shows that the volumetric efficiency for the tuned system decreased only slightly from 91% to 90.6% as the engine speed decreased from 1800 to 1600 rpm. The curve rose to above 92% at the lower speeds. The speed of 1600 rpm corresponded to a delay angle of 73.7 degrees. This delay angle is in the range of the first depression in the volumetric efficiency curve of Figure 3b, which is caused by an unfavorable merger between the standing wave and the new rarefaction. Figures 6a to 6e are the pressuregraphs for the speeds from 1900 to 1400 rpm, respectively. Consider Figure 6d, which is the pressuregraph for 1600 rpm. The pressure has passed the neutral value and is decreasing toward a minimum rarefaction when the new rarefaction begins. The rarefaction of the standing wave is timed too soon to be coincident with the new rarefaction and the following compression maximum is superimposed on the end of the new rarefaction reducing its amplitude. Its first reflection as a compression is correspondingly reduced producing a lower pressure peak in the port during the latter part of the process. However, despite the unfavorable effects of the standing wave, the volumetric efficiency is not seriously reduced in this range.

The pressuregraphs for 1400 and 1500 rpm are shown in Fig. 6e. Both exhibit a deviation from the usual pressure pattern obtained in all the other long tube tests and there also appears to be considerable cancellation of the standing wave at about 540 degrees

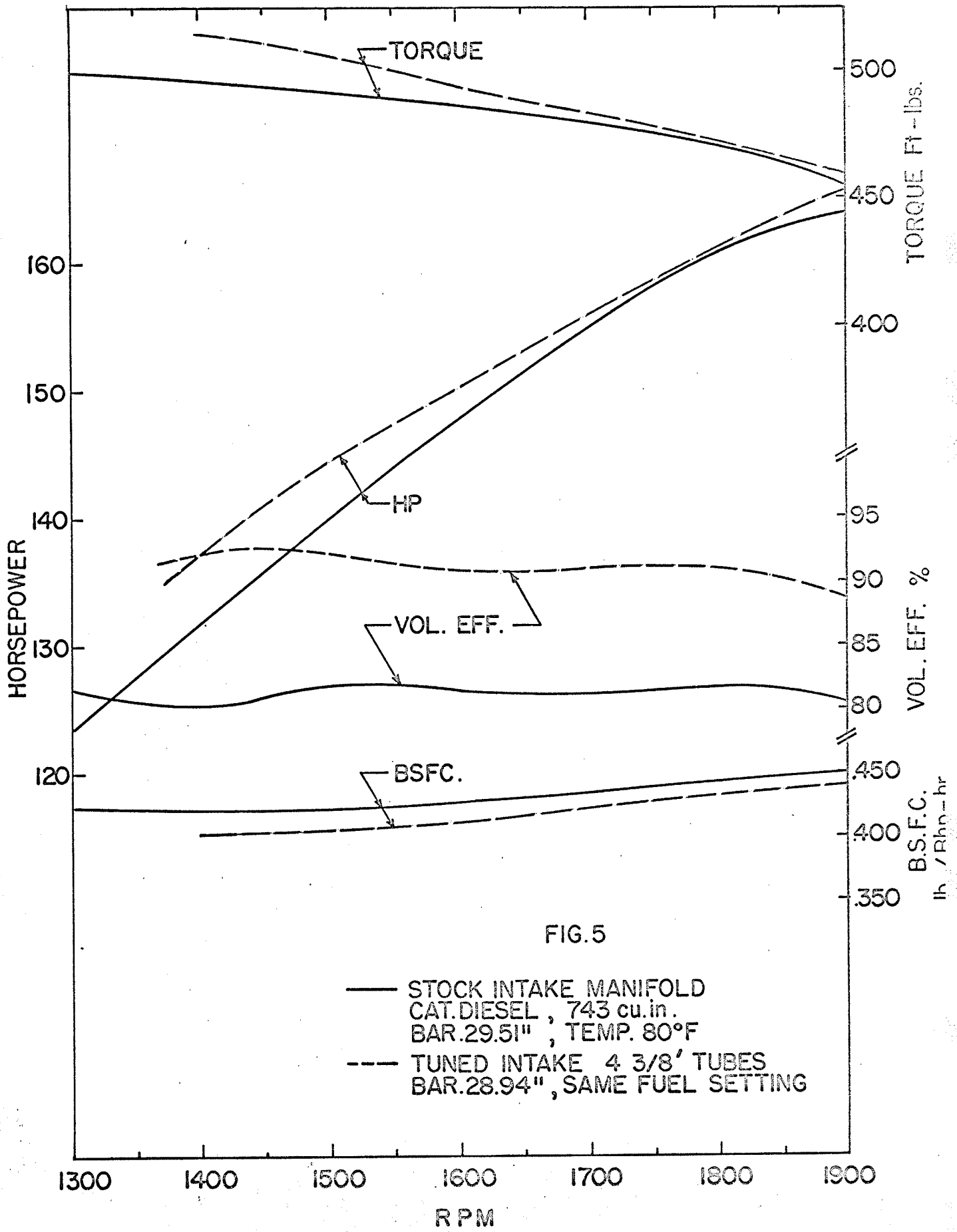


FIG. 5

—— STOCK INTAKE MANIFOLD
CAT. DIESEL, 743 cu. in.
BAR. 29.51", TEMP. 80°F

--- TUNED INTAKE 4 3/8' TUBES
BAR. 28.94", SAME FUEL SETTING

C.L.G. 1962

of crankshaft rotation after valve opening. Immediately after the cancellation a pressure peak occurs just before the opening of the valve followed by a rarefaction. The rarefaction is almost coincident with the new rarefaction and a favorable merger results. At 1500 rpm the delay angle is 69 degrees, which is at the minimum point of the first depression in the volumetric efficiency curve. At 1400 rpm the delay angle is 64.5 degrees which is near the maximum of the first peak of the curve. The volumetric efficiency for both these speeds is above 92%, which seems to indicate the dominant influence of the timing of the reflected wave. The delay angles at all the speeds tested below 1800 are in a range where the reflected wave can return to raise the pressure in the port before the closing of the valve to effect a substantial increase in flow.

Although the timing of the reflected wave appears to be the dominant influence on the tuning of the intake process, Figure 6e indicates the new rarefaction has lost a substantial degree of dominance of the waves in the system at the reduced speed. This explains the deviation in shape of Figure 6e, and introduces another important aspect in intake tuning of tube diameter. At lower speeds the flow rate is lower. The velocity is reduced and the new rarefaction is correspondingly reduced. The small delay angle permits the leading edge of the new rarefaction to return to the port as a compression so early as to reduce the amplitude of the resultant rarefaction as it is being generated. The two effects produce a resultant wave of low amplitude. The new rarefaction is dominant during the intake process but the amplitude of the standing wave produced after valve closure is so low it can be cancelled by the

standing wave of the preceding cycles when opposite wave points are coincident. The standing wave of the preceding cycle has not been greatly reduced by damping due to its low amplitude and is of comparable amplitude to that produced by the new rarefaction. Had the tube diameter been smaller the new rarefaction wave would have been greater resulting in a greater amplitude standing wave which would have dominated the damped wave of the preceding cycle. There is an indication here that a large tube producing low amplitude rarefactions would perform badly at delay angles at which the wave merger is unfavorable due to the more comparable amplitudes of the standing wave to the new rarefaction.

At 1900 rpm the volumetric efficiency fell sharply to 88.5%. The delay angle at this speed was 87.6 degrees, which is in the range where the volumetric efficiency is decreasing sharply with increasing delay angle from the maximum peak to the second depression. The pressuregraph, Fig. 6a, is very similar to Fig. 4e, which is the pressuregraph for a 4.75 ft. tube at 1800 rpm, having a delay angle of 90 degrees. Both have the same general shape and pressure conditions at the beginning of the intake process. The unfavorable merger of the standing wave has the same adverse effects on the reflected wave and hence on the volumetric efficiency. Increasing the speed further would increase the delay angle putting it in the range of the second depression in the volumetric efficiency curves, caused by the unfavorable timing of the standing wave. The higher speeds also increase the delay angle until the return of the reflected wave occurs too late in the process to effect a high gain.

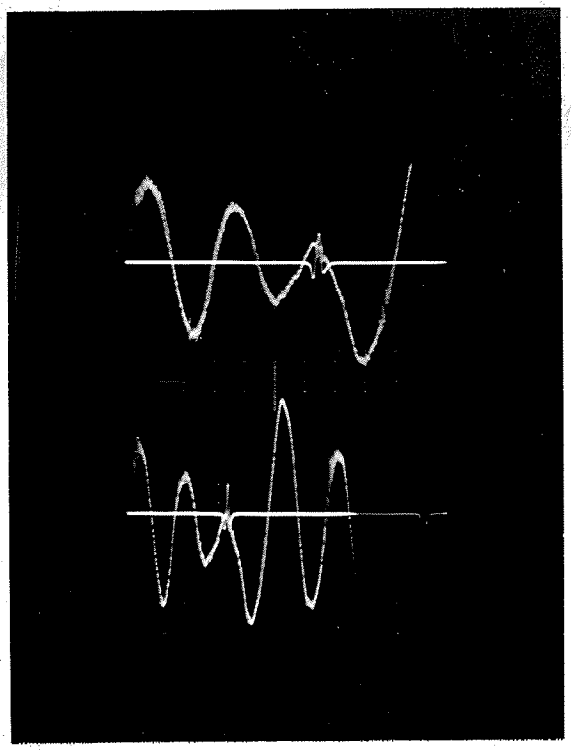


Fig. 6a
N = 1900 rpm

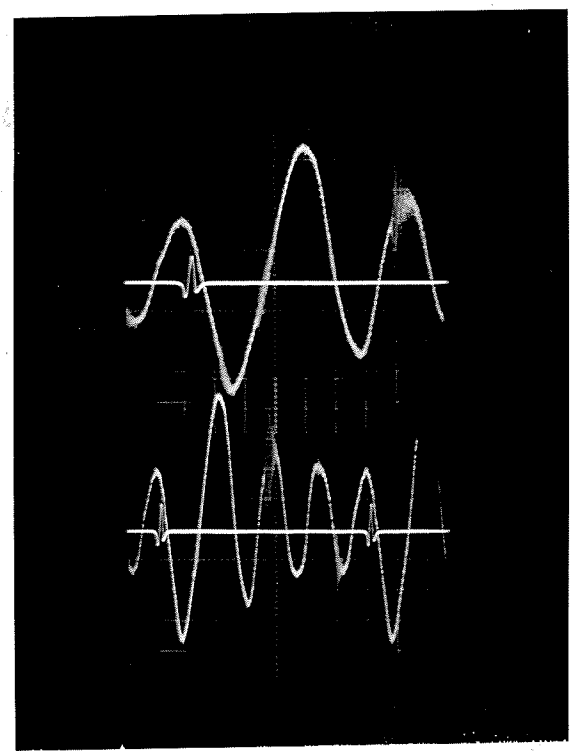


Fig. 6b
N = 1800 rpm

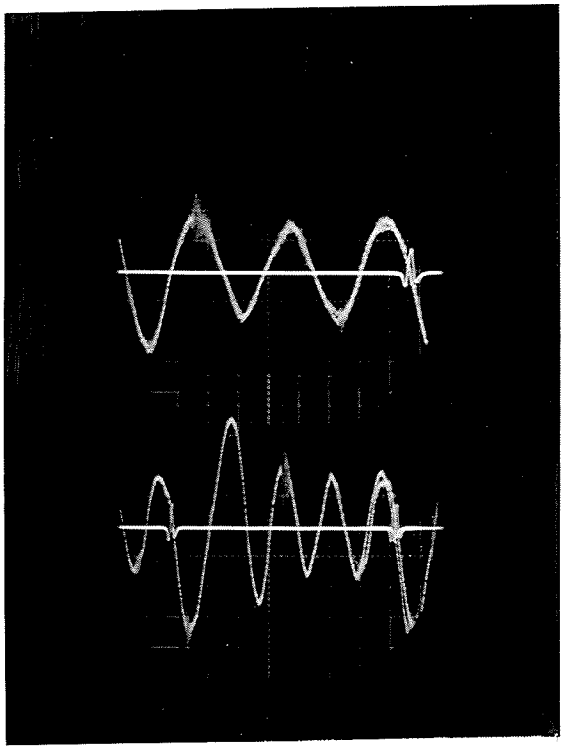


Fig. 6c
N = 1700 rpm

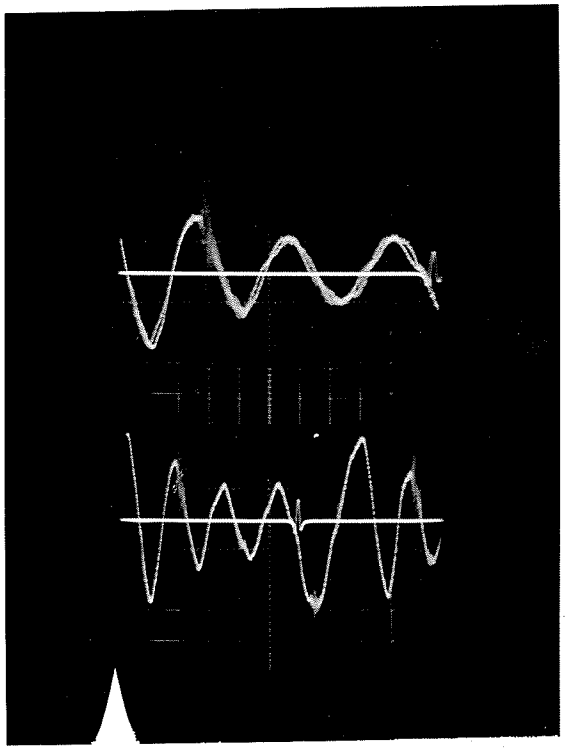


Fig. 6d
N = 1600 rpm

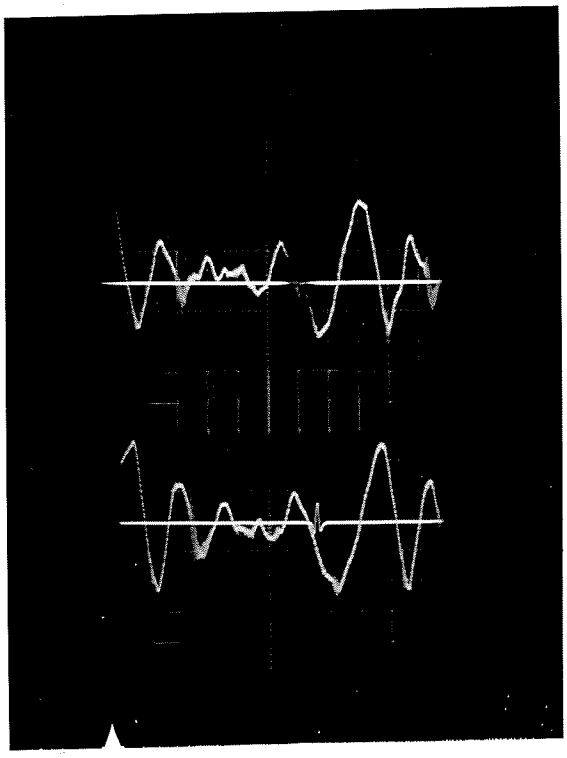


Fig. 6e

N = 1400 rpm

N = 1500 rpm

The displacement of the fuel pumps was not changed after the tests at varying speeds in order to ensure the same setting for the following tests with the bellmouthed tubes. There was not sufficient testing time remaining with the apparatus to perform tests with the fuel pump displacement increased to produce the same air-fuel ratio as obtained with the stock manifold at the tuned speed. Such a test would have determined the actual power increase available with the tuned intake manifold and its variation with speed.

The performance curves obtained with the same fuel pump setting are somewhat misleading as they do not indicate the true gain in power obtainable by tuning. Also, they show a higher gain in power obtained at all speeds other than the tuned speed. This is due in part to the characteristics of the engine and some added effects due to tuning. The lesser gain at the tuned speed may be due to the fact that the speed was that for which the engine was designed for optimum operation. Such factors as fuel injection timing, valve timing and combustion chamber design which are fixed and do not vary with speed may have produced the higher power outputs with the stock manifold at the tuned speed.

The influence of these factors on the performance of the engine may have been overcome by the tuned manifold. The effect of the valve timing would certainly be changed by the change in dynamics of the inlet process. With the stock manifold, the late timing of the closing of the valve may have prevented the achievement of higher volumetric efficiencies at the lower speeds by allowing some air to escape at the beginning of the compression process, and by closing too soon at the higher speeds to allow more air to enter. The high

pressure at the end of the process due to the reflected wave may have been sufficient to overcome these effects. The effect of the fuel injection timing and combustion chamber characteristics may have been changed by the higher air-fuel ratios and high percentage of excess air. The injection timing and combustion system may have been designed to produce maximum mixing, combustion efficiency and a smooth combustion process in the range near the tuned speed. The higher volumetric efficiency produced by the tuned manifold would ensure a higher temperature at the end of the compression process and a higher air-fuel ratio. These would improve the mixing, produce a higher combustion efficiency and a smoother combustion process at the other speeds.

The engine is a precombustion chamber design and factors such as high compression temperatures and increased turbulence influence the performance substantially at different speeds. Air-fuel ratio, characteristics of the fuel, volume of the precombustion chamber and size of the shape of the precombustion chamber entrance are among the many variables, because of which, the combustion system is usually designed for optimum operation at a particular rated load and speed.

Although the curves are somewhat misleading in some ways as mentioned earlier they indicate an important point. That is, with the original fuel setting, the performance, at speeds other than the rated speed at which the design was optimized, can be significantly improved by tuning.

The air-fuel ratio selected for the stock manifold tests was approximately 19:1. This was the ratio obtained from the tests at 1800 rpm under full load when the engine was supercharged. When

the intake system was tuned the air-fuel ratio rose to about 21.5:1. Generally there was good agreement between ratios obtained with the Orsat analysis and the air flow measurements. The Orsat analysis produced ratios averaging within 3% of those produced by the air flow measurements. It was found that a good Orsat analysis could only be obtained by fixing an impact tube in the centre of the exhaust pipe. Using a simple static pressure type of bleed off produced apparent air-fuel ratios about twice as high as the actual.

7.4 Bellmouthed Tube Ends

Figure 3a shows the variation in volumetric efficiency with total tube length (including the length of the bellmouths) for the tests with the fixed volume manifold having bellmouthed outlets. The bellmouthed ends produced a lower maximum volumetric efficiency than the plain end tubes and the peak in the curve occurred at slightly shorter tube lengths. The peak occurred over a narrower range of lengths than that of the plain end tubes. This may be due to the more efficient wave reflection produced by the bellmouths. The amplitude of the standing wave was not as greatly reduced during reflections and remained high enough to more adversely affect the new rarefaction when it was unfavourably timed. Thus the first and second depressions in the volumetric efficiency curve caused by the adverse merger of the two waves are lower and broader due to the greater effect of the higher amplitude of the standing wave.

The more efficient wave reflection reduces the total damping experienced by the wave, producing a higher velocity wave after reflection and a higher frequency of oscillation of the standing wave for the same length of tube. The timing of the reflected wave is not

appreciably affected by the reduced damping since the velocity is only affected during a single traverse from the tube end to the port, but the timing of the standing wave may be significantly affected by the higher velocity resulting after the first reflection. The standing wave experiences four reflections and approximately a dozen traverses of the pipe before merging with the new rarefaction of the next cycle. The higher frequency of oscillation of the standing wave during the many oscillations may greatly change the timing of the wave during the merger, producing an adverse effect on the new rarefaction.

By varying the tube length the standing wave may be timed to reinforce a reflected wave which does not return at an optimum time to effect as high as gain as the plain end tubes. If the length is adjusted to obtain optimum return timing of the reflected wave the standing wave may be timed to have adverse effects. Thus the bellmouths may produce a standing wave and reflected wave which cannot be synchronized to produce a well timed reinforced reflected wave. Further investigation with the use of clear enlarged pressuregraphs is necessary to establish the exact manner in which the wave action is affected by the bellmouths.

7.5 Effect of Tube Diameter

The effects of tube diameter can only be dealt with briefly here due to the limited investigation of this aspect of the subject. The pressuregraph for the 1.875 in. diameter tube exhibited the same general shape as that produced by the 2.15 in. tube at the optimum length which indicates that the diameter of the tube has little effect on wave timing. The amplitude of the pressure fluctuations produced

by the smaller tube was slightly larger at the beginning of the cycle. The new rarefaction decreased the pressure to a lower minimum at the beginning of the cycle before the return of its first reflection which raised the pressure to a higher maximum at the end of the intake process. The standing wave began with a high amplitude but was damped to a much lower amplitude at the end of the cycle. The difference between the amplitude at the end of the cycle and at the beginning was greater for the smaller tube.

The major differences between the effects of the two tubes are the degree of damping and the amplitude of the new rarefaction produced in each. The smaller tube produces a rarefaction of larger amplitude and a higher degree of damping of the reflected and standing waves. Since the smaller tube has a lower flow area the air velocity at the beginning of the process is higher and the amplitude of the new rarefaction is greater corresponding to the energy conversion from pressure to kinetic. Higher amplitudes involve higher particle velocities resulting in greater friction losses during the reflected and the standing wave which accounts for the large decrease in amplitude during the standing wave.

The difference in amplitude at the beginning and end of the cycle indicates an important effect of tube diameter. Due to the lower amplitude of the standing wave at the end of the cycle and higher amplitude of the new wave at the beginning of the next the standing wave has less adverse affects on the new rarefaction when an unfavorable merger occurs between the two. A smaller tube can therefore be expected to perform better at delay angles at which the nullifying effects of an adversely timed standing wave

tend to reduce the gain. The volumetric efficiency curve would be more uniform over a wider range of delay angles making the smaller tube more suitable for an application where operation over a wide range of speeds is necessary.

The average flow velocity during the intake process in the 2.15 in. diameter tube at 1800 rpm was 170 ft. per second, and the corresponding Mach number was 0.149 using the sonic velocity of 1140 ft. per second for the 80°F inlet air. For the 1.875 in. diameter tube the velocity was 224 ft. per second and the Mach number 0.196. Further investigation to determine the effects of tube diameter and air velocity on the shape of the volumetric efficiency versus delay angle curve is necessary in determining the size of tube most suitable for particular applications. For example, an engine operating at constant speed may perform well on a large tube whereas an engine which requires good characteristics over a range of speeds may require a smaller diameter of tube to offset the effects of an adversely timed standing wave.

7.6 Performance of the Curved Tube

The curved tube produced exactly the same shape of pressuregraph for the optimum length as the straight tube indicating the wave action was not affected by the 20 in. radius curve in the tube. By the use of similar curved tubes turned downward along the side of the engine and joined by a manifold the aspect of intake manifold tuning becomes more practical. The space problem, posed earlier by the long tubes projecting out from the side of the engine as much as 5 feet, is not so prohibitive by using a curved tube

manifold occupying a 2 foot space along the side of the engine. Further investigation into determining the minimum radius of curve for a given size of tube would be of great value in determining the minimum radius that could be used to tune engines by means of curved tube manifolds.

VIII CONCLUSIONS

The conclusions arrived at directly from the investigation are presented in the following point form.

(1) The gain obtainable by tuning was found to be due mainly to the timing of the return of the first reflection, as a compression, of the rarefaction wave created at the beginning of the intake process. The timing of the return of the wave can be related to the crankshaft rotation by the delay angle which is the angle through which the crank rotates while the wave makes a double traverse of the tube. The optimum length of tube is therefore determined by the speed of the engine, the average velocity of the wave, and the optimum delay angle. The optimum delay angle for the four stroke cycle diesel engine was 83 degrees from which can be derived the equation for the optimum length of tube.

$$L = 6.9 \frac{a}{N}$$

(2) Although the reflected wave is mainly responsible for the gain it is subject to favorable or unfavorable effects from the residual wave system of preceding cycles depending upon its timing. The optimum delay angle of 83 degrees is the angle at which a favorable merger between the residual wave system and the new rarefaction occurs resulting in a reinforced resultant wave. At other delay angles the reflected wave remains dominant and a substantial gain is realized despite adverse effects of an unfavorably timed residual wave system. Delay angles ranging from 60 degrees

to 90 degrees can cause a substantial increase in flow.

(3) A multicylinder engine can be tuned by joining the tubes of optimum length by a rake type manifold having a volume at least 0.75 times the engine displacement. The presence of the manifold has no effect on the tuning process or the wave action in the tubes.

(4) The volume of the manifold has no effect on the tuning process and is not a factor in determining the optimum length of tube.

(5) Intake tuning can produce a substantial gain over a broad range of engine speeds provided the manifold is tuned for a speed near the upper limit of operation. At speeds below the tuned speed the delay angle decreases and a high gain is maintained in the range from 83 degrees to 60 degrees. At lower speeds and smaller delay angles the gain can only diminish to short tube manifold performance. At speeds above the tuned speed the gain diminishes rapidly.

(6) Tube diameter has negligible effect on wave timing and diameters smaller than usual port size produce similar pressure conditions in the intake port.

(7) In general smaller tubes produce waves of larger amplitude and a higher degree of damping of the residual wave system following the intake process. This is an indication that a relatively smaller tube may produce a high gain over a broader range of delay angles due to the less adverse effects of the damped residual wave system at angles at which unfavorable timing occurs.

(8) The use of bellmouthed entrances at the manifold end of the tube produces a slightly lower maximum gain which occurs over

a narrower range of delay angles than plain end tubes.

(9) An intake tube having a 90 degree curve at a 20 in. radius near the valve end has no effect on wave timing and produces identical pressure conditions in the intake port.

(10) Intake manifold tuning of multicylinder four stroke cycle diesel engines is a thoroughly practical and effective means of increasing the volumetric efficiency of the engine by as much as 15% over the entire speed range of operation.

APPENDIX I

FLOW NOZZLE DESIGN AND CALIBRATION

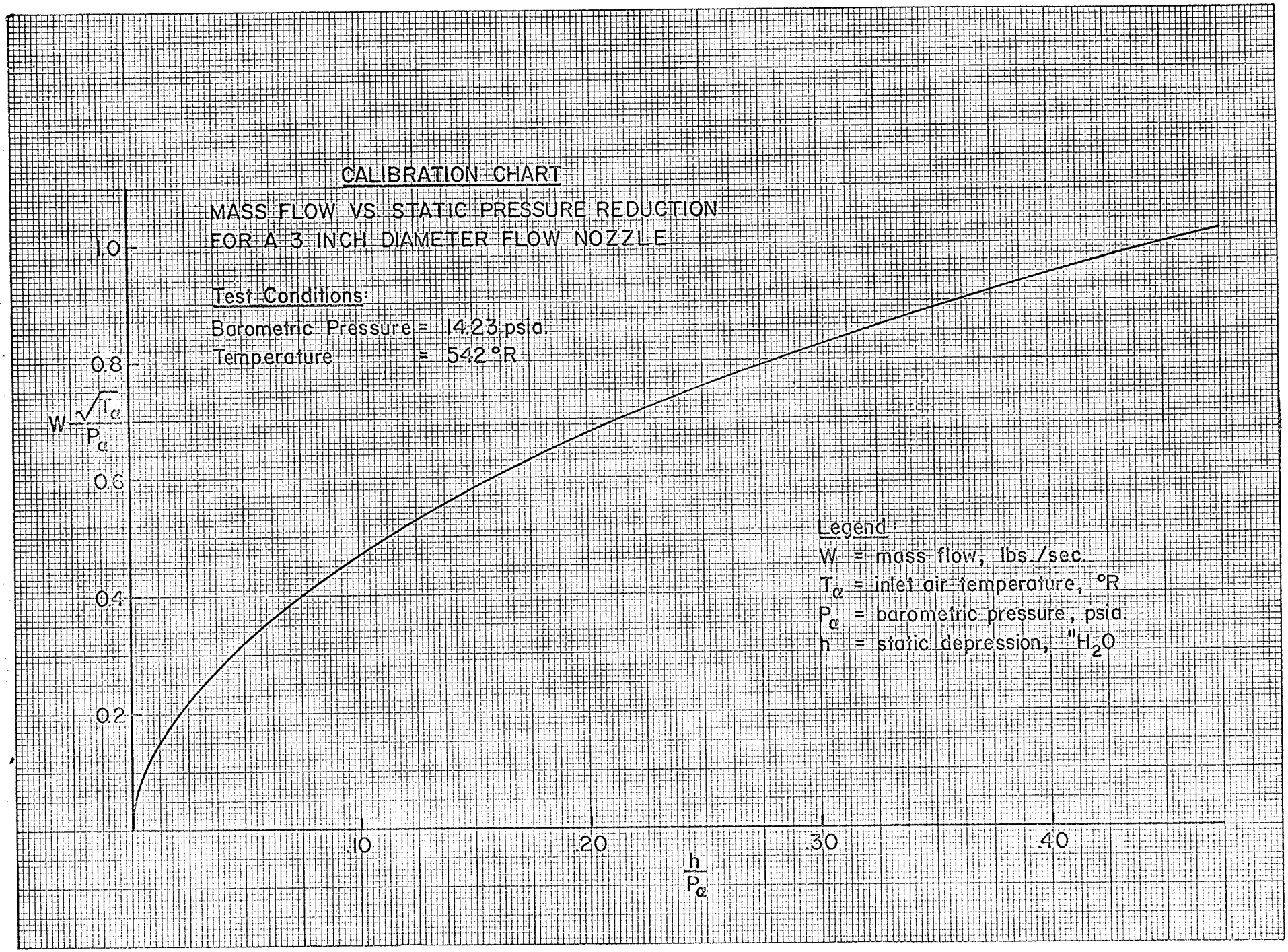
The flow nozzle was of laminated wood construction and was designed to conform to specifications outlined in the ASME Power Test Codes. The laminations were perpendicular to the axis of the nozzle and consisted of maple planks glued together with the grain of each piece orientated at right angles to the adjacent piece. The nozzle was turned on a lathe with the entrance shaped in accordance to a sheet metal template designed from the ASME specifications.

The approach to the nozzle throat was the quadrant of an ellipse with the semi-major axis parallel to the flow axis and equal to the throat diameter. The semi-minor axis was 2/3 of the throat diameter and the length of the throat was 0.60 times the diameter. After manufacture the nozzle was oiled and waxed to prevent drying and shrinkage and to provide a smooth, highly polished surface.

The nozzle was placed in the exit of a low velocity wind tunnel for calibration. At each calibration point the static pressure in the tunnel upstream was measured by a water filled manometer and the velocity pressure profile of the escaping jet was measured by an impact tube connected to a water filled manometer. The velocity profile was found to be uniform for all flow rates. Data was collected for a range of static pressures from zero to approximately six inches for a total of twenty different flow rates. The flows were calculated and plotted in a chart of the dimensionless parameters affecting the

flow as shown on the following page. The static depression produced by the engine at 1800 rpm was slightly less than 3 inches of water.

The absolute values of the volumetric efficiency and air flow were not the main objectives of the experiments, but rather, a consistant indication of a change in volumetric efficiency and air flow. Hence the accuracy of the values obtained is not as important as the ability of the apparatus to reproduce the values and clearly indicate the changes that occur as other variables are changed.



APPENDIX II

SPEED MEASUREMENT

The exact speed of the engine was an important factor in the experiment and had to be determined directly to conduct the tests and correlate the results. Accurate speed determination was therefore essential and the most accurate method available was used. A stroboscope operating at line frequency was used at all speeds to indicate when the desired speed was attained. Thus, the accuracy of measurement approximated the accuracy of the frequency of the alternating current power supply.

The engine was connected to the brake by a chain and sprocket type of coupling having 18 cogs. The top surface of the cogs were painted white. This provided 18 similar equally spaced marks 20 degrees apart on the periphery of a circle. A single white chalk mark was drawn on the hub of the sprocket for use when operating the stroboscope at a selected frequency. The coupling was covered with a shield having a plexiglass window on one side through which the flashing light of the stroboscope could be shone.

At 1800 rpm, with the stroboscope flashing at line frequency there were two flashes per engine revolution and 180 degrees of rotation between flashes. Since there are 20 degrees of rotation between cogs, 9 cogs will have moved past any fixed point between flashes and the tenth will have occupied the position of the first during the previous flash. Since the cogs were similar, no motion

was apparent in the light of the stroboscope and the cogs appeared to be stationary.

At 1700 rpm there were 2.12 flashes per revolution and 170 degrees of rotation between flashes. Since there were 20 degrees of rotation between cogs the second flash occurred when the original positions of the cogs were occupied by spaces and the original spaces were occupied by cogs. This produced the effect of twice as many cogs. Since the marks are similar no motion was apparent and the cogs appeared stationary.

At 1600 rpm, there were 160 degrees between flashes and the positions of the cogs during the first flash were exactly occupied by other cogs during the next flash. Whenever the engine was operating at an integral multiple of 100 rpm the cogs on the coupling appeared stationary. At even multiples the cogs appeared to have the actual spacing. At odd multiples there appeared to be twice as many cogs at half the actual spacing. This method was used for all speeds from 1200 to 1600 rpm. Since it is not readily apparent which multiple the engine is operating at when the cogs appear stationary, the stroboscope can be switched to flash at a varied frequency on the singular white chalk mark to determine the speed. The tachometer on the brake also provided a good reference in determing the speed.

APPENDIX III

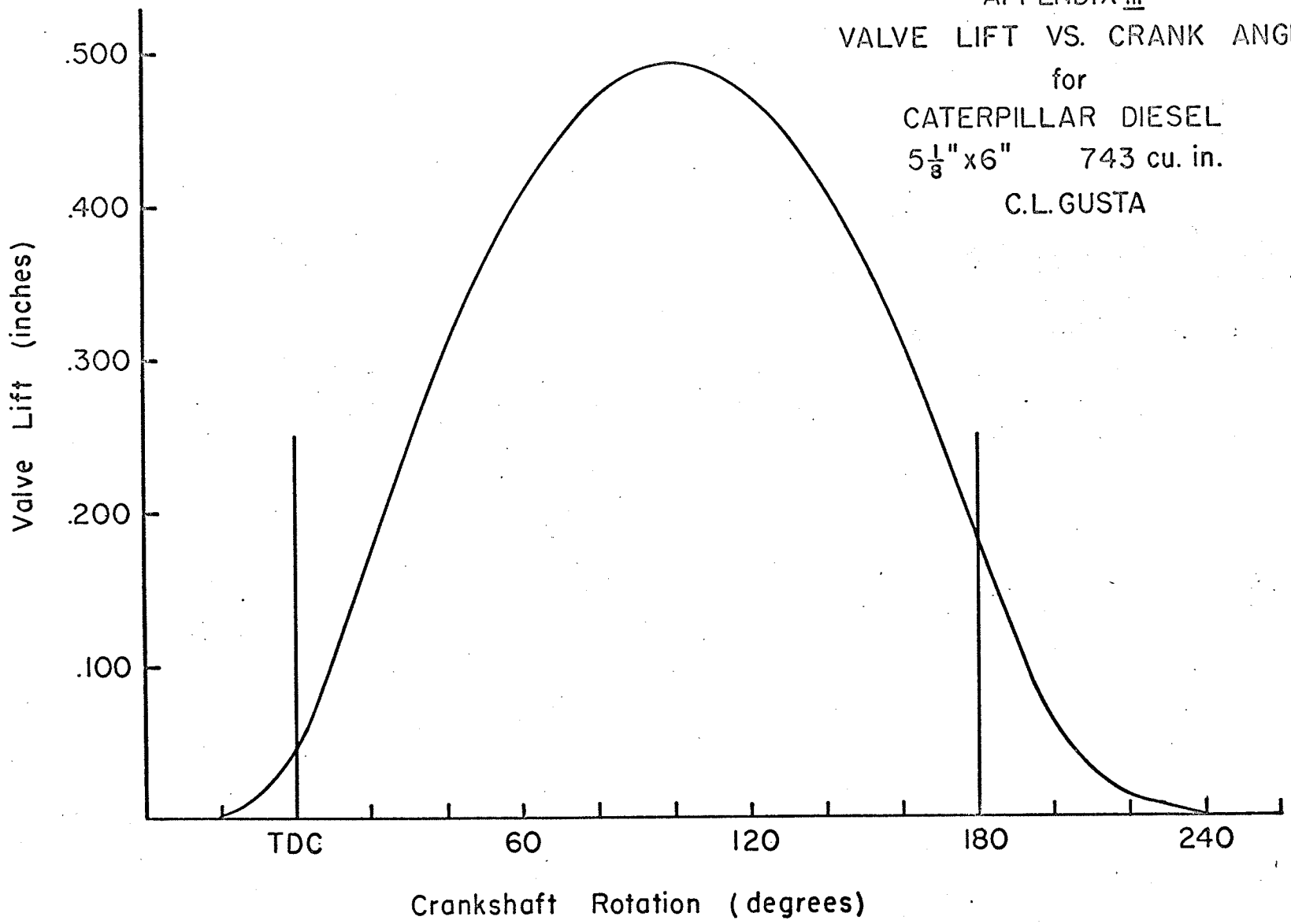
VALVE LIFT VS. CRANK ANGLE

for

CATERPILLAR DIESEL

$5\frac{1}{8}$"x6" 743 cu. in.

C.L. GUSTA



APPENDIX IV

INTAKE PORT PRESSURE MEASUREMENT

The pressure in the intake port was measured by a piezoelectric quartz crystal pressure transducer. These transducers operate on the principle that when a pressure is applied to them an electrostatic charge of proportional magnitude is created. The crystal is connected to a charge amplifier which contains a condenser and amplifier unit. The charge from the crystal is transmitted to the condenser where the combined electrical capacitance of the crystal, cable and condenser determines the voltage generated. The voltage is then amplified and transmitted to a scope where the magnitude of the signal is indicated by the displacement on the screen.

These crystals have the feature of being able to measure and indicate rapidly changing pressures without any measurable lag or delay which may cause a distorted signal. Hence inertial effects common to most pressure measuring devices are eliminated for signals varying at frequencies as high as 100,000 cycles per second. However due to the leakage of electrical charge they will not indicate a sustained static pressure for long periods of time and accurate calibration with a dead weight tester is difficult. Rapidly varying pressures can be measured and indicated without serious error due to charge leakage and the timing of any change in pressure is precisely indicated.

The object of the pressure measurements was not to measure the absolute pressure in the port, but rather, to determine the exact manner in which it varied. Hence, the absolute values of pressure indicated are not as important as the exact timing of any change in pressure indicated and the accuracy with which the relative magnitudes of variation in pressure are indicated. A clear graph of the variation accurately showing the smallest features in the shape of the pressuregraphs are of major importance in analysing the wave motion in the tubes and effects of pressure on the air flow.

The crystal sensitivity was 0.5 pCb/psi (10^{-12} coulombs per psi). The charge amplifier was operated at a magnification of 50 mv/pCb (millivolts per pCb) producing a signal equal to 25 mv/psi. With the oscilloscope sensitivity set at 20 mv/cm. each centimeter represented 1.25 psi on the vertical scale. The pressuregraphs are pictures of the pressure trace on the oscilloscope. Each vertical division represents 1.25 psi and each horizontal division represents 100 degrees for the graph of the full cycle and 50 degrees for the graph showing the beginning of the process.

APPENDIX V

PRESSURE WAVES IN GASES IN TUBES

A brief attempt will be made here to explain the action of pressure waves in tubes for the reader who is not familiar with the subject but wishes to obtain a greater knowledge to more fully understand the manner in which the waves produce the tuned intake process as claimed in the theory of this thesis. Only moderate-amplitude waves will be considered in the following discussion as this is the basic type of wave in the tubes. For a more thorough treatment of wave motion the reader is referred to the references listed in the bibliography.

Pressure waves in tubes are longitudinal waves, that is, they are travelling pressure disturbances in the tubes which are characterized by a particle displacement of the medium parallel to the direction of travel of the wave. This is in opposition to transverse waves which are characterized by particle displacement at right angles to the direction of travel as is the case of surface waves and vibrating strings. If the direction of travel of a longitudinal wave is the same as the direction of the particle displacement, the wave is a compression or positive pressure wave. If the particle velocity is in the opposite direction the wave is termed a rarefaction or negative pressure wave. The form of the wave is determined by the initiating cause.

A wave may be initiated in a tube by forcing a piston at

one end rapidly into the tube. The force of the piston against the adjacent air, as it is being accelerated into motion, creates a high pressure near the piston. Since the air is compressible and the individual particles have inertia, the high pressure is not uniformly distributed along the pipe but developes only at the piston. The high pressure compresses the air and causes a flow in the direction of motion of the piston. Due to its momentum the flow will create a high pressure against the air particles immediately downstream and the high pressure region will continue downstream after the motion of the piston has ceased. The air particles adjacent to the piston will return to their original pressure when the piston stops but they will have been displaced an amount equal to the displacement of the piston. In this way, a compression wave pulse is generated, its shape being dependent on the motion of the piston. If the piston were to be rapidly returned to its original position a low pressure would be created adjacent to it in the tube. The low pressure would cause air downstream to flow toward the piston creating low pressure areas further downstream and a rarefaction wave would be created which would move down the tube away from the piston.

A rarefaction wave can also be created by drawing some air out of the end of the tube. Figure 2a shows the case of air being drawn out of the end of a tube by an engine cylinder. The figure shows the graph of the pressure created at the end of the tube during a period of crankshaft rotation and the corresponding pressure deviation from atmospheric along the length of the tube. The low pressure in the cylinder causes air to flow out the end of the tube creating a low pressure area in the tube. The inertia of the air particles and

the pressure gradient causes the low pressure area to move long the tube at a high velocity to form a traveling rarefaction wave. The cross-hatched area in the figure represents the length along the tube at which the pressure has been reduced below ambient by the rarefaction wave. Upon reaching the open end of the tube the low pressure area causes an inrush of air and a compression wave is created which travels toward the cylinder and superimposes itself on the outgoing rarefaction. That is, the portion of the rarefaction which has reached the open end is reflected as a compression wave. During an open end reflection of moderate-amplitude waves the pressure at the tube end remains closely atmospheric and the reflected portion of the wave is superimposed on the original wave producing a net pressure equal to atmospheric at the tube end.

Figure 2b shows the superimposition of the inward travelling compression on the outward travelling rarefaction and it should be noted that the amplitude of both waves are equal and opposite at the tube end producing the necessary cancellation to maintain atmospheric pressure at the open end. During the reflection of the rarefaction however the inward open-end flow creates a vena contracta with some turbulence and back flow occurring just inside the open end resulting in some restriction, loss of flow energy and wave amplitude.

In general the reflection of a wave at an open end produces a reflected wave of opposite amplitude. The reflection of a wave at a closed end produces a wave of a similar amplitude. Figure 2c shows the reflection of the inward travelling compression at the valve. Due to the high resistance of the valve the wave is partially reflected as a compression wave which travels back toward the open tube end. The

superimpositioning of the two oppositely moving compression waves produces a high pressure at the valve.

Standing waves are a particular form of progressive waves resulting from the superimpositioning of oppositely moving waves of approximately sinusoidal form in such a manner that the pressure variation in the tube at any point is approximately sinusoidal. Such a situation can develop from the reflection of a wave in a tube at a closed end. The amplitude of variation would be greatest at the closed end where reflection is occurring and zero at the open end which are referred to as a velocity node and a pressure node respectively. The net resulting motion of the air in the tube is an air column surging back and forth into and out of the tube. Such a standing wave is characterized by the fact that the approximated sine wave has a wave length equal to four times the length of the tube.

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