DYNAMIC AND FLUID DYNAMIC CHARACTERISTICS OF A FUEL INJECTION NOZZLE

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THESIS

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CHARACTERISTICS OF A FUEL INJECTION NOZZLE

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Dynamic and Fluid Dynamic Characteristics of a Fuel Injection Nozzle

by

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ABSTRACT

Due to the increasing requirements for reduced emissions from internal combustion engines many manufacturers have expressed interest in gasoline fuel injection systems. In order to determine the fluid dynamic and dynamic characteristics of a commercial injector a test chamber was constructed and a laboratory simulation carried out. A proposed method of predicting flow rates and dynamic response has been compared with experimental results.

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ACKNOWLEDGEMENT

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I. INTRODUCTION

> Because of the high density of traffic in some cities on the continent of North America, the United States authorities have found it necessary to lay down regulations concerning the contents of exhaust gases discharged by internal combustion engines.[1]*

The recent public emphasis on environmental protection has provided impetus toward production of cleaner automobile engines. In order to achieve pollution goals, however, certain sacrifices in output (i.e., horsepower) and fuel consumption are required. Additionally, engine temperature has increased (a result of late ignition timing and extremely lean air-fuel ratios at idle), resulting in increased cooling system size. These facts, combined with a worldwide crisis in the supply of refined petroleum products, have engendered feverish attempts by the automotive industry, both in the United States and abroad, to produce an automobile engine which is both economical in size and operating cost and non-polluting.

The severest effects of the pollution restrictions have been felt in the foreign car field, since these traditionally small-engined cars inherently have little or no horsepower reserve to lose to pollution control. As an immediate solution to the problem, several European manufacturers have

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*Numbers in brackets refer to references in the bibliography.

considered fuel injection systems. Among the several systems available the first to reach American highways in significant numbers was the electronically controlled fuel injection system manufactured in Germany by Robert Bosch, GMBH. This system is currently utilized by automobiles manufactured by Volkswagen consortium (Volkswagen, Porsche-Audi, Mercedes Benz) as well as by the Swedish manufacturing firms of Saab and Volvo. Due to the wide application of this system, it was decided to conduct a laboratory investigation of the characteristics of the Bosch injector, with the following objectives:

(1) design and construct a test chamber and fuel supply system

(2) determine operating parameters and obtain oscilloscope photographs of the pulse received by the injector on an operating engine

(3) duplicate the pulse using laboratory equipment and determine injector response

(4) observe and photograph the injector operating in the laboratory environment.

II. BACKGROUND

In order to place the recent introduction of fuel injection in the proper perspective, it may be well to review briefly the history of gasoline injection. During World War II Gasoline injection was used by the Germans in high output aircraft engines. Reasons for the adoption of injection vice carburetor fuel systems were complex, but in no small part were based on the scarcity of knowledge (in Germany) concering high altitude carburetion as opposed to a thorough familiarity with injector theory and application which had been gained through Diesel design studies. [1] In any event, direct injection aircraft engines were in common use by the Luftwaffe by 1945, and the use of gasoline injection is listed as the main factor in the trebling of German aircraft engine performance during the period 1935-1945.

Since the aircraft injection units had all been designed for direct injection two-cycle engines, initial German efforts in the postwar period to adapt the principles of injection to small engines were along these lines. Despite initial successes, however, problems involved with low-load mixtures and the scavenging system indicated that continued experimentation would not yield economically feasible results. Accordingly, research into the direct injection two-stroke cycle engine was abandoned in favor of more promising work.

In the meantime, interest in gasoline injection in England resulted in the formation of the engineering group at Joseph

Lucas, Ltd., which was charged with the responsibility of designing a practical gasoline injection system for automotive application. At the time of formation of this group the hope was expressed that a system could be developed which would offer either substantial gains in performance (output) or improvement in economy to such an extent that savings in fuel cost would offset, at least partially, the anticipated higher cost of an automobile equipped with such a system.

The Lucas effort over the three year period 1946-1949 resulted in the development of the first commercially practical gasoline injection system for small (on the order of 1600 cc), multi-cylinder, high speed automotive engines. The system arrived at by the Lucas group is of interest in that it is designed for indirect, or manifold, injection; that is, the fuel is injected into the induction manifold rather than the combustion chamber. This approach was adopted due to the poor results that had been achieved with direct (cylinder head) injection, principally because of injected fuel contacting the cylinder walls and subsequently being transferred to the crankcase. After experimenting with several locations for the injector valves and directions of injection, it was determined that the ideal solution was to utilize downstream injection in proximity to the intake valve. A significant result of this early work was the discovery that injection timing was not at all critical; that is, there was no requirement for injection to occur coincident

with the time during which the intake valve was open. This fact permits considerable simplification of the electronically controlled system which is of interest here.

The Lucas system consisted of three parts which, in some form, are basic to a fuel injection system. They are:

(1) the constant pressure pump — Experiment had indicated that a fuel pressure of 75 PSI (minimum) was required for proper opening of the injection valve and atomization of the fuel. In this early work a surplus aircraft pump was used to provide a constant 100 PSI regardless of flow rate.

(2) the metering distributor — This is a device so designed that the proper amount of fuel is provided to the proper injector. This device can be either electronic or mechanical. The pioneering Lucas system had a mechanical distributor in the form of a sleeve-in-sleeve device which was engine driven, thus providing direct timing information. Manifold pressure applied in conjunction with throttle position to a movable shuttle regulated the quantity of fuel distributed to each injector.

(3) the injectors — In mechanical systems such as this early Lucas effort the injectors are of the relief valve type. In operation, fuel pressure applied to the mechanical injector overcomes the force of a spring and causes the valve to open; absence of pressure allows the spring to return the injector to its closed position.

As indicated above, this early Lucas system was a mechanically linked, metered and timed system. This type of

system is currently in use in Europe and in fact enjoyed a brief flurry of interest in the United States in the mid-fifties. Its chief disadvantage lies in the increased cost, bulk, and weight added to the engine, and the requirement for high pressure in the fuel system. Nevertheless, certain advantages can be realized from such a system, namely:

(1) minimum modification to existing engines is required
(2) increased power is available
(3) substantial fuel savings can be realized

Additionally, it was found that manifold injection results in significant lowering of cylinder head temperature (up to 45° C lower than achieved with conventional carburetion). The lower temperature permitted leaner mixtures without overheating and the use of higher octane fuel.

A. THE ELECTRONICALLY CONTROLLED INJECTION SYSTEM

In an attempt to reduce the size and number of engine tied components the Bendix Corporation, in 1958, designed an electronically controlled gasoline injection system which was available as an extra cost option on Chrysler Corporation automobiles. This system consisted of:

(1) a positive displacement pump which delivers fuel to a regulator/filter at 25 PSI, from which it is delivered to the injectors at 20 PSI
(2) a solenoid operated injector valve for each cylinder
(3) a modulator
(4) an engine driven trigger selector which delivers the opening pulse to the proper injector and also triggers the modulator.

The distinguishing feature of the Bendix system is the use of electronic timing and metering of the fuel supplied to the engine. As this system is the direct antecedent of the Bosch system, it may be well to describe the principal components of the system in more detail.

The injector is a solenoid operated valve so designed that when lifted from its seat by the electro-magnetic force produced by the trigger pulse six small orifices are opened to permit fuel flow. When the pulse terminates a high rate flat spring returns the valve to its seat.

The modulator, which was referred to as the "brain box", develops electrical pulses whose duration was regulated by inputs received from various sensors indicating throttle position, acceleration, and atmospheric conditions. In eight cylinder configuration the modulator had two separate channels (one each for the left and right cylinder bank) which delivered the pulses to the trigger selector, from which they were routed to the proper injector. It should be noted here that the injection pulses so produced were timed to coincide with the intake stroke for each cylinder; that is, each injector operated independently. Though novel in its approach, the Bendix system suffered from the same lack of public acceptance that the other injection systems of the time received and was no longer available by the end of 1959.

B. THE BOSCH ELECTRONIC INJECTION SYSTEM

When continuing requirements to trim carburetors for minimum emissions improved the economic feasibility of gasoline injection, Robert Bosch GMBH of Stuttgart and the VolksWagen Werke developed an electronically controlled gasoline injection system (ECGI) for the 1600cc engine used in some VolksWagens and in the Porsche 914. Figure 1 shows the typical layout and Figure 2 the block diagram for this system.

The ECGI system developed by Bosch is essentially similar to the earlier Bendix system. The principal difference in the systems is that in the case of the Bosch system the four injectors are collected into two groups, with both injectors in a group opening simultaneously. Since, as noted in the Lucas experiments, injection timing is not critical, increased simplicity is achieved at no penalty to performance. The typical four cylinder injection cycle is shown in Figure 3.

In summary, there are available at the present time two distinct types of timed manifold injection systems

1) Mechanical systems using mechanically timed and metered high pressure fuel injected through poppet type injection valves.
2) ECGI systems using electronically timed and metered injections of moderate pressure fuel through solenoid operated injection valves.

Since the Bosch system is the only one commercially available in the United States in large numbers the Bosch injector was chosen for study. Further information regarding the historic aspects of automotive fuel injection may be found in Reference 1.

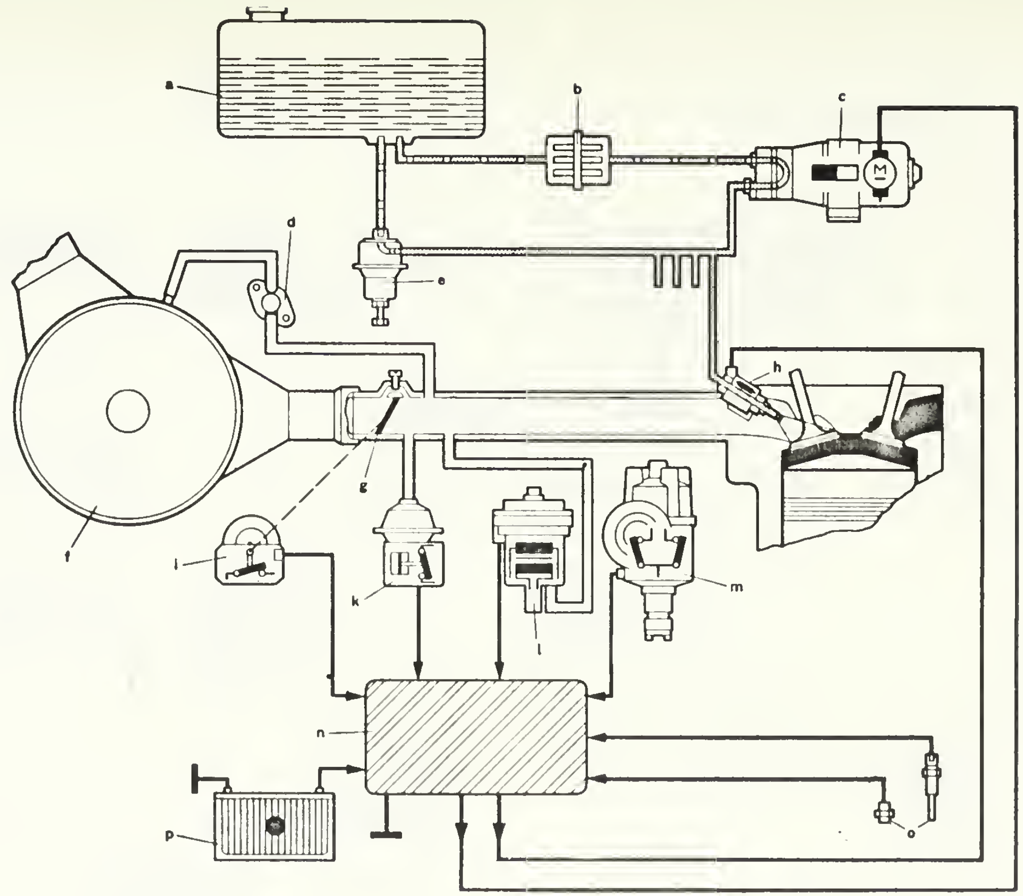


FIG. 1: SCHEMATIC DIAGRAM OF ELECTRONICALLY CONTROLLED FUEL INJECTION SYSTEM [1]

a) – Gasoline Tank
b) – Fuel Filter
c) – Fuel Supply Pump
d) – Auxiliary Idling Air Valve
e) – Fuel Pressure Regulator
f) – Air Filter
g) – Throttle Plate
h) – Fuel Injection Valve
i) – Throttle Position Switch
k) – Full Load Pressure Switch
l) – Manifold Pressure Sensor
m) – Ignition Distributor with Trigger Contacts
n) – Electronic Control Unit
o) – Temperature Sensor
p) – Battery

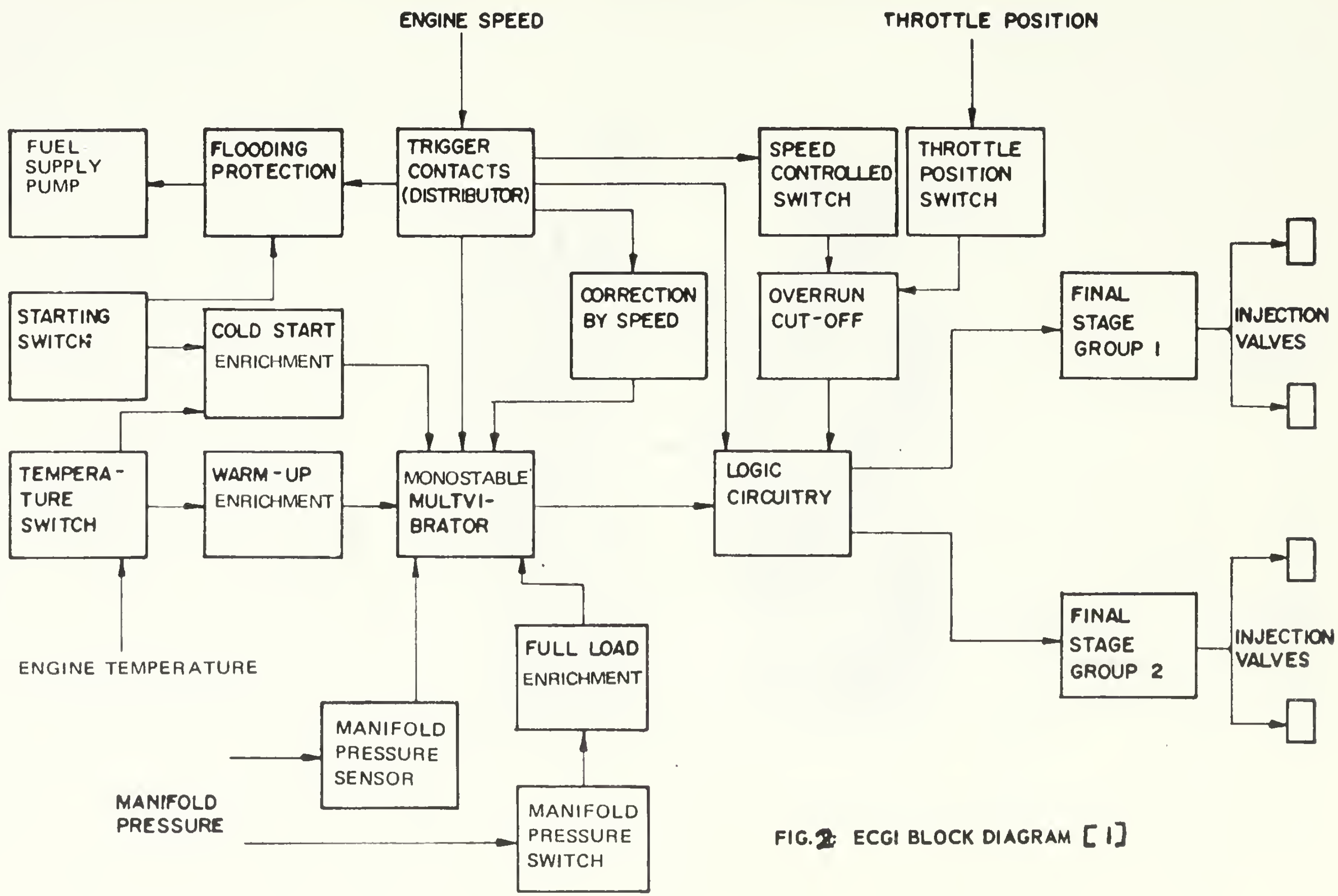


FIG.2: ECGI BLOCK DIAGRAM [1]

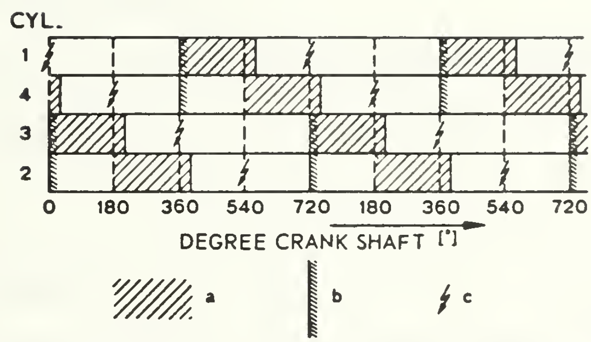


FIG.3: COMMENCEMENT OF FUEL INJECTION VERSUS INTAKE VALVE TIMING. [1]
Shaded line b) indicates commencement of injection; duration of injection depends on speed and load.
Shaded area a) indicates valve opening periods, arrows c) indicate spark plug firing.

III. DESCRIPTION OF APPARATUS

The apparatus for this study consisted of:

(1) an airtight test chamber

(2) a solenoid operated injection valve manufactured by Robert Bosch, GMBH, Stuttgart, Germany

(3) a pressurized fuel system

(4) an air ejector type vacuum pump

(5) instrumentation necessary for generation of the pulse and for making the various measurements associated with the investigation.

Major components of the apparatus will be described in turn. The schematic arrangement of the apparatus is shown in Figure 4.

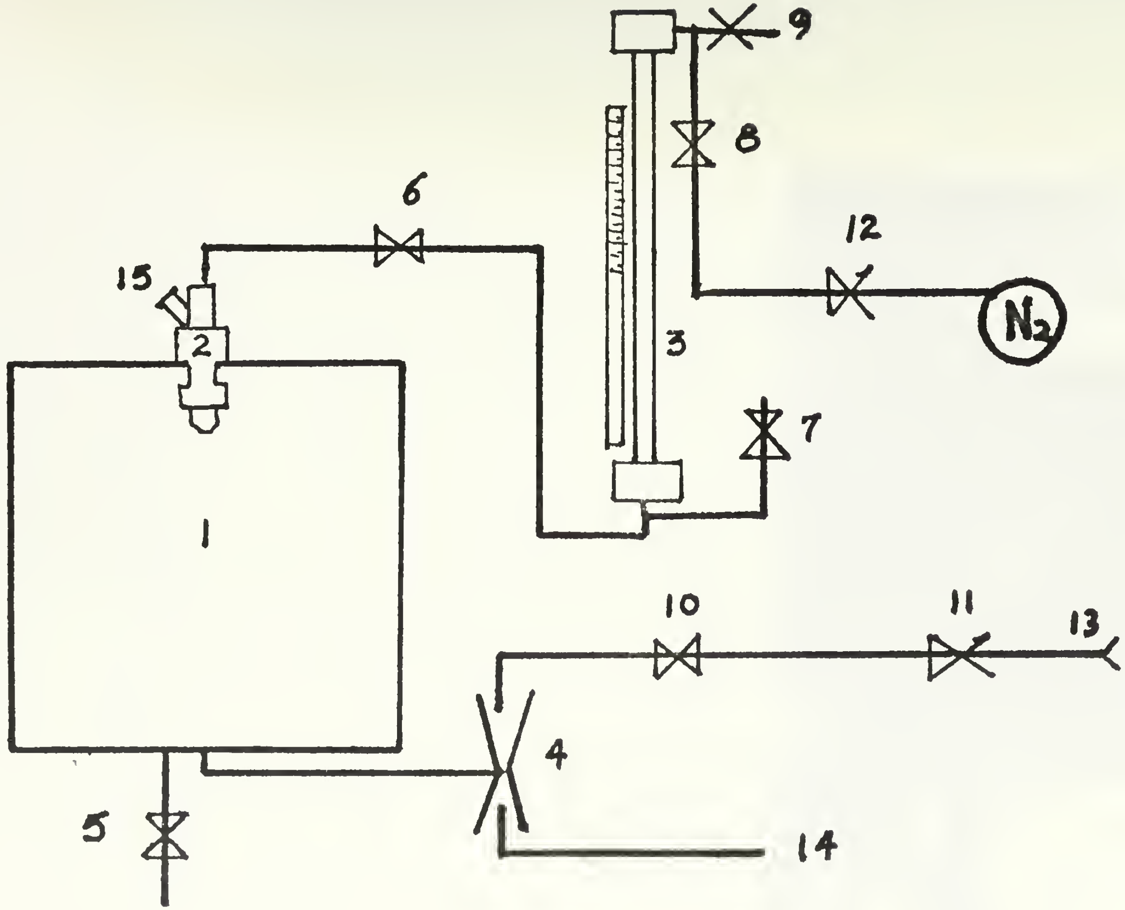
A. THE TEST CHAMBER

The test chamber consists of three parts (Figure 5):

(a) the base, of 3/4 inch aluminum plate grooved to hold an "O-ring"; there is provided a centrally located drain and a suction located toward the rear

(b) the upper section of 3/4 inch aluminum and 3/4 inch Plexiglas; three windows (located at the front and both sides) are provided for observation and photography; the hole for the injector mount is located centrally in the top plate and an additional threaded hole is provided for fitting of a pressure (vacuum) gauge.

(c) the injector mount, machined from aluminum stock and drilled to be a press-fit for the injector seals; the mount is so designed that when inserted into the



1. Test chamber
2. Injector
3. Fuel tube
4. Air ejector
5. Drain
6. Fuel valve
7. Fuel fill
8. Pressure valve
9. Vent
10. Air ejector control valve
11. Air supply regulator
12. Nitrogen regulator
13. Air supply
14. Exhaust
15. Pulse in

Figure 4. Schematic Arrangement of Apparatus

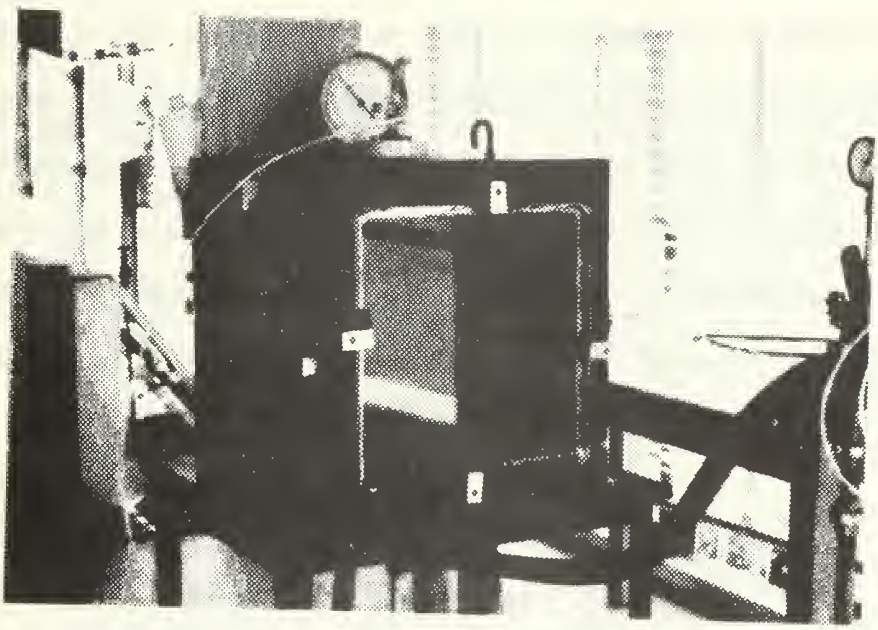


Figure 5. Injector Test Chamber

test chamber the tip of the injector will be flush with the inner face of the chamber top (Figure 6).

Construction and assembly of the chamber is straightforward, consisting of bolting the side, back, and top plate together and sealing the joints with RTV sealant. In assembling the chamber the upper section is placed on the base and the injector mount placed in position in the top of the chamber. "O-rings" are used to provide an air-tight seal between the mount and the chamber and between the upper section and the chamber base. Grooves for proper location of the rings are located in the lower face of the injector mount and in the chamber base.

B. THE FUEL SYSTEM

The fuel supply system is diagrammed in Figure 4. Fuel quantity is determined from the height of the fuel column within the (.430 inch inside diameter) glass tube. Pressure is provided by a Nitrogen bottle providing pressurized gas through a Matheson regulator to the top of the fuel column. In order to promote ease of assembly all fittings are of the "Poly-Flo" type and "Poly-Flo" tubing is used for interconnections throughout.

C. THE INJECTOR

The injectors are commercial units manufactured by Robert Bosch GMBH, Stuttgart, Germany, and intended for use

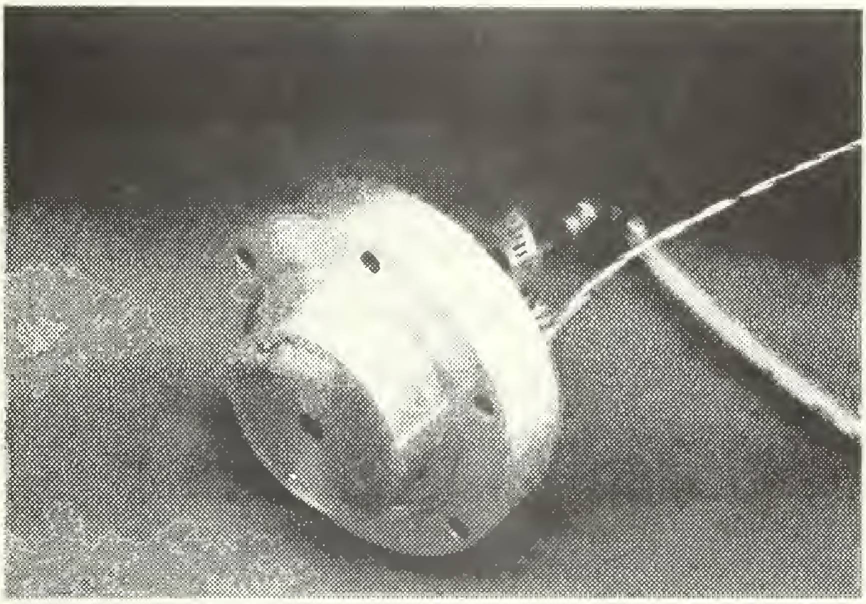


Figure 6. Injector Mount

in the Volvo four cylinder engine. The injectors were obtained locally.

The injector, basically a solenoid operated valve, consists of (Figure 7):

(a) a machined steel housing enclosing the other parts and sealed at the time of manufacture; no repair or adjustment of the injector is therefore possible

(b) a coil wound of 304 turns of number 28 AWG wire on a plastic form; a soft-iron core is centered within the form and the whole assumbly is attached to the plastic end cap which seals the upper end of the housing; the end cap provides an electrical connection and also has a short section of rubberized fabric hose containing an integral fine mesh filter screen which serves as the fuel connection to the injector

(c) a coil spring (rate = 10.7 lbf/in) is partially enclosed within the base of the coil form and is used to close the injection valve at the end of the electrical impulse; the lower end of the return spring rests on the top of the solenoid plunger

(d) the solenoid plunger is permanently attached to the injection valve and is used to transform the magnetic force generated in the coil into movement of the injection valve

(e) the injection valve and solenoid plunger are the only moving parts of the injector; the electrical pulse through the coil causes the valve to move upwards a distance of .005 inch (.127 mm) thus allowing fuel passage

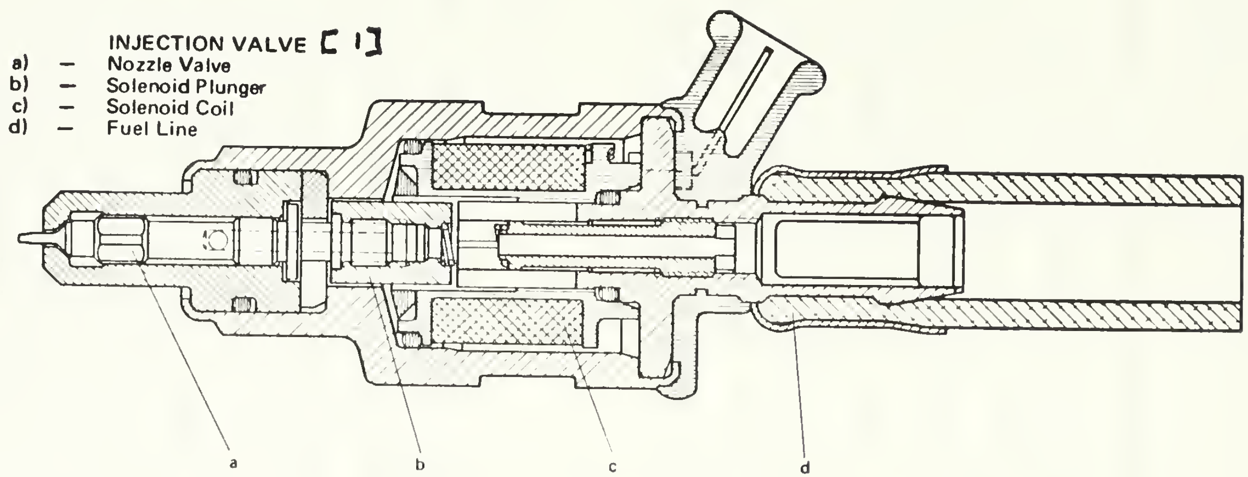


Figure 7. Injector

(f) the stainless steel valve seat forms the bottom closure of the injector housing.

D. FLOW PATH THROUGH THE INJECTOR

Fuel enters the injector through the short section of hose at the top. After passing through the fine mesh filter it flows through the center of the coil core and solenoid plunger, emerging through a transverse port in the injector valve body. The fuel remains in the annular passage between the valve and valve seat until the valve lifts from its seat, at which time it passes through the annular orifice into the manifold. "O-rings" are used within the housing at the top and bottom of the coil to prevent leakage of gasoline to the coil windings and also between the valve seat and injector housing to prevent leakage around the outer periphery of the seat.

E. THE VACUUM PUMP

Due to the high probability of the vacuum pump being required to evacuate a gasoline laden atmosphere from the chamber and the necessity of achieving only moderate vacuum, an air ejector type pump was selected as the ideal choice. This type of pump offered the possibility of infinitely variable and controllable vacuum over the range of interest (0-20" Hg) as well as simplicity of construction and ready availability. Accordingly, a 100 PSI air ejector manufactured by Graham Manufacturing Co. was installed. 100 PSI air was

supplied to the ejector from the compressor through a Shrader air pressure regulator and exhaust air was routed to the exterior of the building via a 2 inch hose.

F. INSTRUMENTATION

The following instrumentation was utilized in conducting this investigation. Specific applications are discussed within the section concerned with experimental procedure.

Hewlett-Packard No. 214a Pulse Generator
Ser. No. 921-05343

Hewlett-Packard Model 141B Oscilloscope with number 1421A time base and delay generator and model 1408a dual trace amplifier installed.
Ser. No. 1225A 00906 applies to the entire unit

Bentley-Nevada Corporation Proximeter Pickup and detector apmlifier
Ser. No. NONE

Monsanto Programmable Counter Timer Model 110B
Ser. No. 013679

Leeds and Northrop Model 86 Precision Millivolt Potentiometer

IV. EXPERIMENTAL PROCEDURE

Although work toward accomplishment of the various objectives proceeded concurrently in most instances, the different areas are discussed in the order presented here for the sake of clarity.

A. DESIGN OF TEST CHAMBER

Initial considerations in the design of the test chamber were rooted in the following considerations:

a) safety
b) air tightness
c) observation and photography
d) cost

The requirement for insuring the safety of operating personnel precipitated the need for a method of relieving internal pressure in the event of inadvertant ignition of the fuel mixture in the chamber. After considering several choices it was determined that the simplest method was to construct the chamber so that it was free of any attachment to the base plate. In order to permit observation and distortion free photographs, windows of 1/2 inch plate glass were installed initially. It was decided, however (after failure of one plate window during initial testing), that the properties of plate glass were too unpredictable for safe use; accordingly, new windows of 3/4 inch Plexiglas were fabricated and installed. Although some image degradation was introduced by the distortion of the Plexiglas windows under load it was

found to be negligible. Airtightness was proven by the chamber's ability to hold a vacuum of 22 inches of Mercury.

Also at this time initial evaluations of the fuel system were carried out. As originally designed and constructed, the fuel supply tube consisted of a length of extruded Plexiglas tubing. Difficulties arose in preventing breakage of the tube by an interaction with gasoline which is not fully understood. After several failures occurred, attempts to use acrylic tubing were abandoned in favor of a glass tube. Though abandonment of the Plexiglas tube invalidated early attempts at data collection, evaluation of the pressure supply system and determination of the best method of filling the fuel system were accomplished. The regulated Nitrogen system proved reliable and able to provide the requisite fuel pressure. It was found that the fuel system could best be filled by connecting the fuel line between the injector and the fuel tube to the spout of a plastic dispenser bottle. By squeezing the bottle it was possible to fill the fuel tube and supply line while at the same time eliminating air from the system.

B. ESTABLISHMENT OF DATA BASE

In order to establish the real world base necessary for construction of the laboratory model, arrangements were made with the local Volvo dealership to instrument an operating vehicle. This effort was facilitated by the availability at the dealership of a Clayton chassis dynomometer. This

instrument allows direct measurement of rear wheel horsepower at various loads (controllable by the operator). Since for the purposes of this project relative load information only was required no further load measurements were attempted.

In addition to engine load it was desired to measure engine speed (revolutions per minute), manifold temperature, injection pulse width and amplitude, and fuel pressure. Attempts to measure engine speed by means of an inductive pickup attached to the distributor and the counter-timer yielded values that were too large by a factor of four. It is felt that this large error was due to the pickup unit receiving extraneous pulses from the distributor, most notably the timing pulse to the ECGI computer as well as the individual cylinder ignition pulses. Since no resolution of this difficulty was immediately available it was decided to use the engine speed information available from the tachometer installed in the vehicle. Although the accuracy of this instrument was questionable, it was felt that is was sufficient for the purposes of the project.

Manifold temperature was measured by means of a thermocouple inserted in the intake manifold in such a way as to position it as close as possible to the intake valve without risking interference with the operation of the valve. Thermocouple readout was provided by the Leeds and Northrop precision potentiometer.

The Hewlett-Packard 141B oscilloscope was utilized to measure the width of the injection pulse. Leads were

connected from the electrical connector at the injector to the differential inputs of the oscilloscope. In order to display a single pulse, the Hewlett-Packard oscilloscope was utilized in the single-sweep mode. With the automobile engine running, the reset button on the instrument was pressed once, generating a single sweep and producing the image of a single pulse.

Since no oscilloscope camera is available to fit the 141B, use was made of the storage feature of the oscilloscope to permit photography of the image using a 35mm camera equipped with a macro-focusing lens. The storage feature of the oscilloscope permits the operator the option of having the scope retain a given image for an indefinite period, a feature which was used throughout the project.

Two data collection runs were made, one at no load condition and throughout the engine speed range and the other at full load conditions throughout the range. Fuel pressure during these runs was monitored by means of a (dealer supplied) pressure gauge attached to the injector side of the fuel line. Other than as previously noted no difficulty was encountered in obtaining the desired data.

C. PULSE SIMULATION

Prior to discussion of the pulse simulation effort, it is necessary to discuss the interpretation of pulse width and amplitude as read from the oscilloscope. Since the inductive load presented by the injector distorts the input pulse as read at the injector, the same pulse was applied

first to a purely resistive load and then to the injector. Relationship between the two displays is as shown in Figure 8. The sharp spikes at the beginning and end of the injector display are visual representation of the coil resisting the instantaneous change in voltage across the terminals. Since the start of this spike occurs exactly at the leading and trailing edges of the input pulse, pulse width and amplitude should always be read as indicated in Figure 8.

In the early stages of the project a search of available literature on the Bosch ECGI indicated that a maximum pulse amplitude of 3.0 volts and a pulse duration of .002 to .010 seconds could be anticipated. Initial attempts at duplication of the pulse were based on these criteria; sufficient control and latitude was incorporated into the system to permit later modification of these values to those actually encountered.

Early attempts to utilize the Hewlett-Packard pulse generator alone to produce the pulse were unsuccessful when it was found that the pulse generator was incapable of producing sufficient power at the parameter values being used. In order to increase the power level of the pulse a pulse amplifier was constructed and incorporated in the pulse circuit. The amplifier proved capable of producing a square wave pulse of sufficient power over the entire range of interest, and of providing excellent simulation when routed through the injector. Oscilloscope photographs of the actual injection pulse (Figure 9a) and the simulated

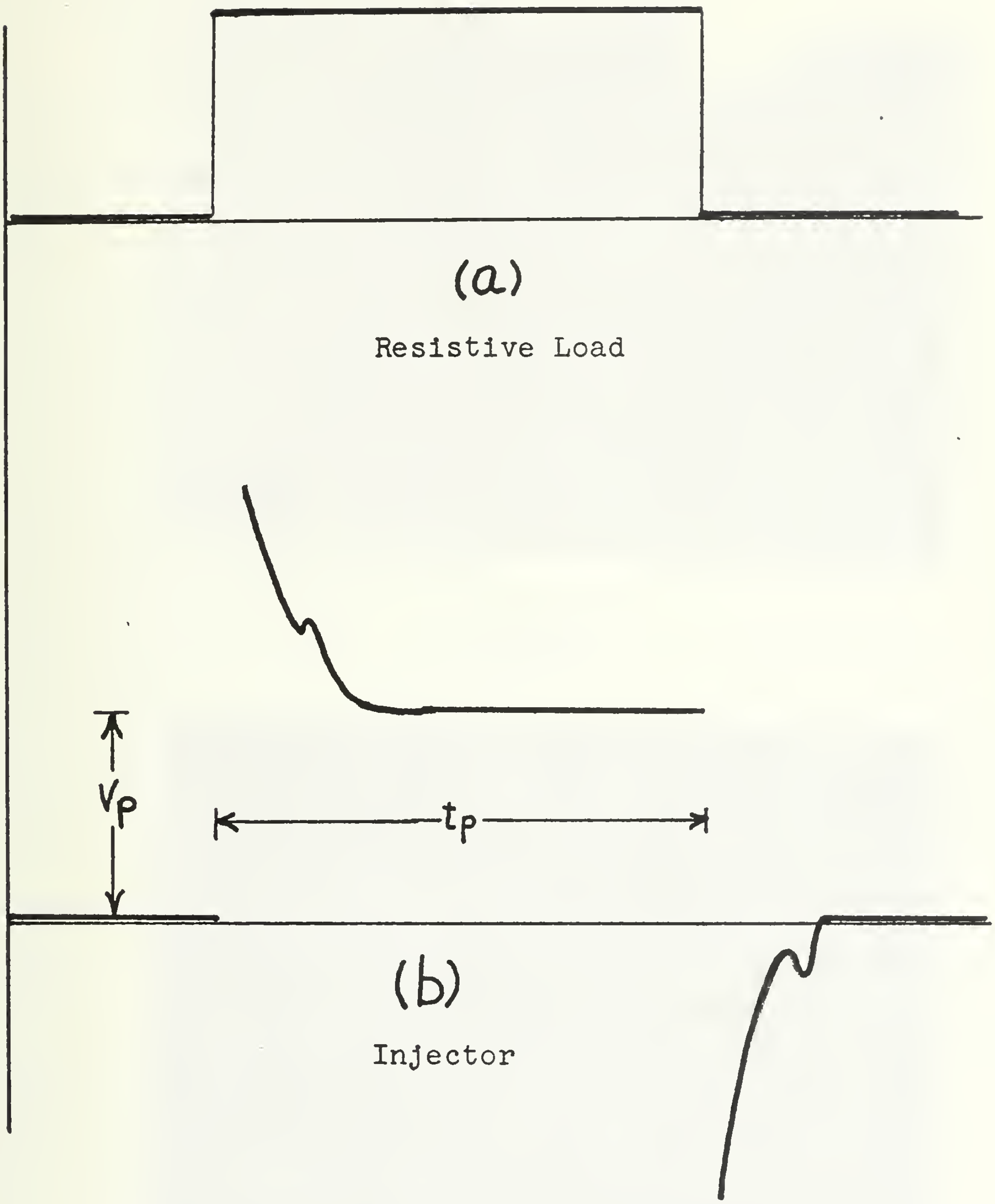
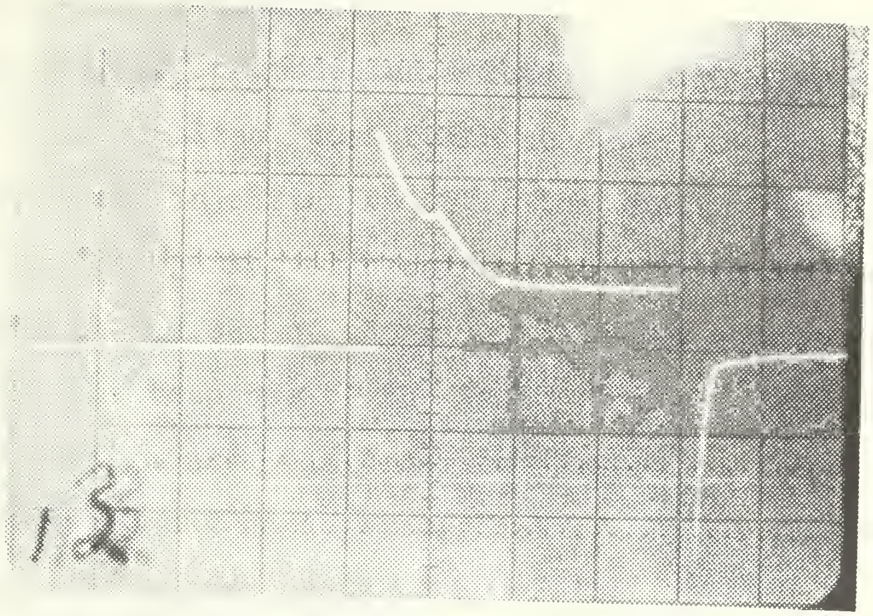
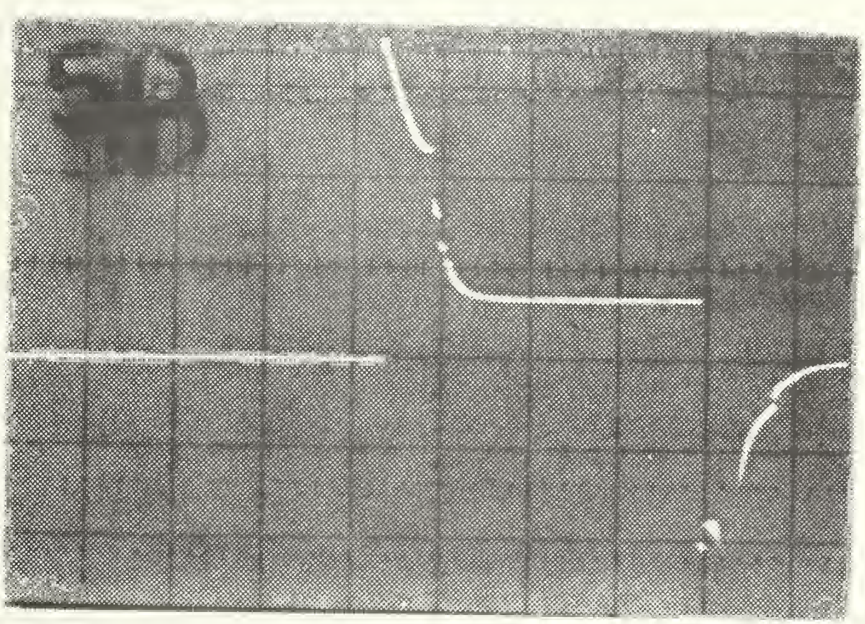


Figure 8. Interpretation of pulse



(a)



(b)

Figure 9. (a) Actual; (b) Simulated Pulse
t_P = 7.2 ms, V_P = 3.5 Volts

injection pulse (Figure 9b) show the high degree of correspondence achieved.

Once success in producing a satisfactory pulse in the lab had been realized, work began to determine the exact response of the injection valve to the input pulse. As any lag in the opening or closing of the injection valve would have the effect of changing the total time of the injection, the response of the valve is a critical factor in valve performance. In order to measure the extremely small (.005 in) and rapid excursions of the injection valve the proximeter system produced by the Bentley-Nevada Corporation was chosen. This system consists of a small inductive pick-up coupled to a power supply/amplifier. The amplifier output signal can be read on any of several devices, and, once calibrated (Appendix A), can be used to measure distance with excellent accuracy. Since, in the case of this project comparison to the input signal was desired, the output signal was led to Channel B of the Hewlett-Packard oscilloscope.

A jig (Figure 10) was constructed which provided for the mounting of the injector, the proximeter pickup, and the dial indicator. The dial indicator was used only as a standard for calibration of the proximeter and to place the proximeter .005 in from the tip of the injector valve. Utilizing this jig the injector was pulsed and response measured in two modes; first using Nitrogen pressure only (dry) and then with pressurized fluid applied. In the second case diesel oil was used as the working fluid in order to

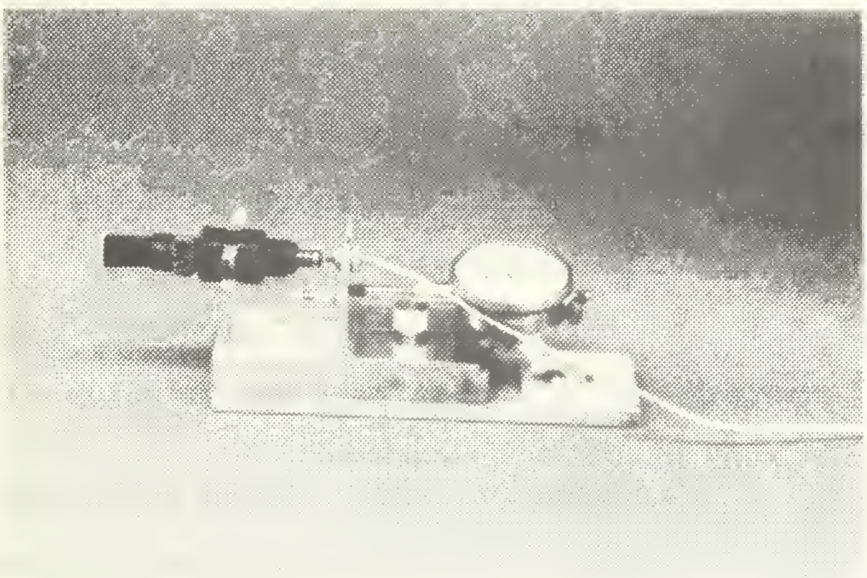


Figure 10

avoid the obvious dangers in spraying gasoline in an open atmosphere. Also, in order to protect the pick-up unit from the oil spray a thin sheet of flexible plastic was interposed between the injector and pick-up. As the plastic was non-conductive, it had no effect on the sensitivity of the proximeter, a fact which was verified experimentally.

A second injector was obtained for use in the event of failure of the test unit. When the second injector was received it was mounted in the jig and its response compared to the first specimen. It was found that within the limits of measurement no discernible difference could be seen in the response of the two injectors. This fact allowed simplification of the final phase of the project.

D. OBSERVATION OF THE OPERATING INJECTOR

The final goal of this project was to simulate, as closely as possible, the operating environment of the injector and to observe the injector in that environment. Although it would perhaps have been of interest to simulate the airstream temperature and velocity as they exist in the induction manifold, overriding considerations of safety dictated that a partially evacuated chamber at ambient temperature be used. The use of such a chamber allowed the use of gasoline as the working fluid, which was necessary for the most accurate model.

In early tests of the chamber it was found that the gasoline would tend to collect in the bottom of the chamber,

from where it would seep out around the bottom seal and pose a hazard. Additionally, there was a problem with the spray splashing onto the windows and obscuring them. To eliminate these problems a piece of ordinary blotter paper was used to cover the bottom of the chamber and the alternate injector, mounted outside the chamber without fuel, was used as a dummy load to set up the pulse before each run. Since continuous pulsing is necessary when setting a particular pulse, this procedure reduced significantly the amount of fuel injected into the chamber.

After establishing the desired pulse the test injector was connected to the output of the pulse amplifier. The chamber pressure was then lowered to the desired value (read on the vacuum gauge located on the chamber top) by means of the valve located on the air ejector. Pressure was then applied to the top of the fuel column and initial fuel column height was recorded. The entire fuel system was checked for leaks or evidence of entrained air. When the system was found to be satisfactory the injector was pulsed manually the desired number of times. At convenient intervals fuel column height was recorded and the totals averaged to arrive at a figure for quantity per injection. Runs were made at a wide range of values for pulse width and amplitude, chamber pressure, and fuel pressure, as noted in tabulated data.

The major difficulty with the use of the test set-up is associated with leakage. Since the amount injected with

each pulse is extremely small, a very small leak can result in significant error. Accordingly, future researchers are cautioned to inspect the system closely for leaks and eliminate all of them prior to taking data. Figures 11, 12, and 13 show the assembled apparatus.

E. PHOTOGRAPHY

When this project was initially considered one of the primary goals was to obtain high quality photographs of the injection spray. At the termination of the project this task had been found much more complex than at first envisioned, and the final results achieved were not entirely satisfactory.

As has been noted, three plexiglas windows were provided in the chamber to permit observation and photography. Initial planning envisioned the use of side lighting to accomplish the desired result. Accordingly, a high intensity stroboscopic light (Strobotac) was obtained and mounted at the left side window of the chamber. The injection pulse was split at the output of the pulse amplifier and part routed to the external trigger input of the Strobotac. It was found, however, that the DC voltage level riding on the Strobotac input terminals caused rapid failure of the pulse amplifier. In addition, the internal delay between the receipt of the pulse and triggering of the flashtube, though short, was of sufficient length to prevent synchronization of the strobe with the injection. Due to the above shortcomings no photographs were obtained using this method.

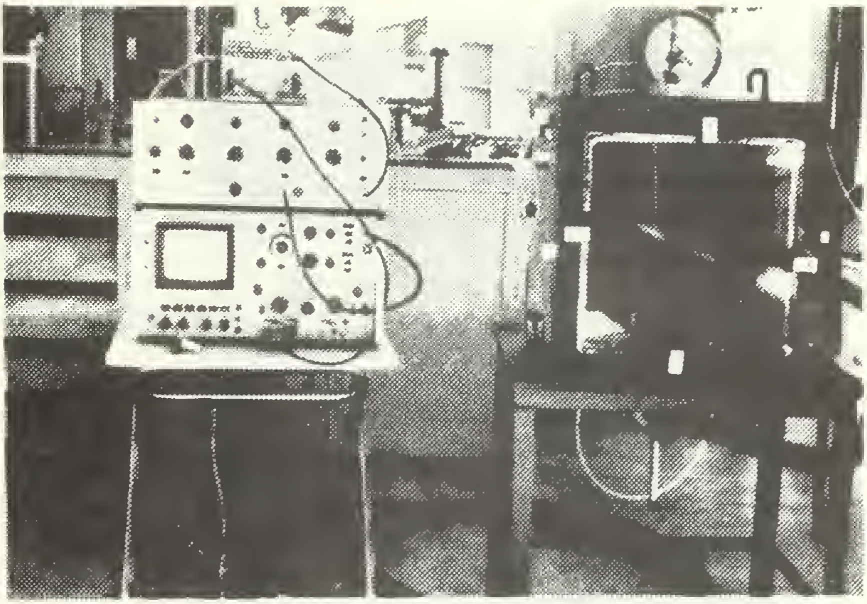


Figure 11. Apparatus

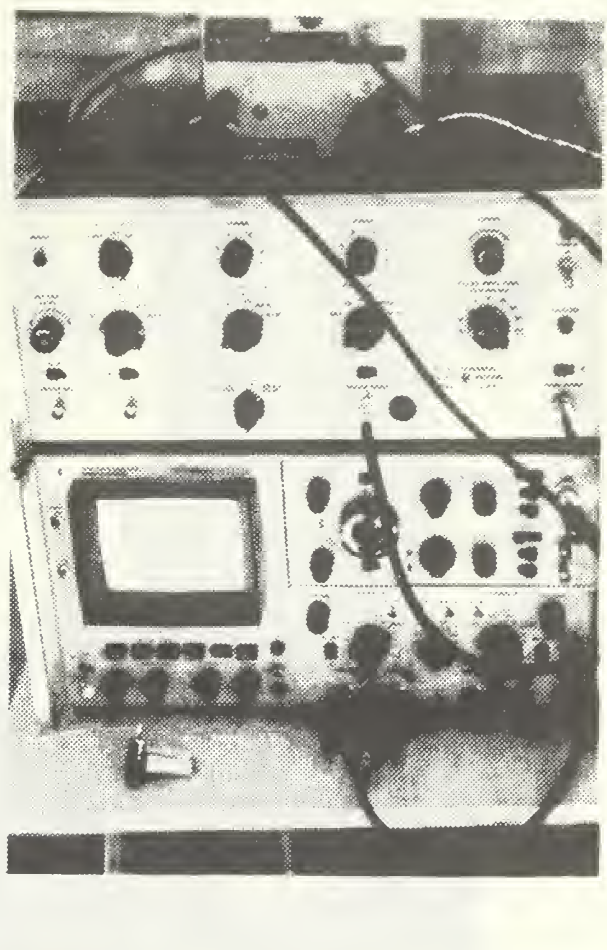


Figure 12. Instrumentation

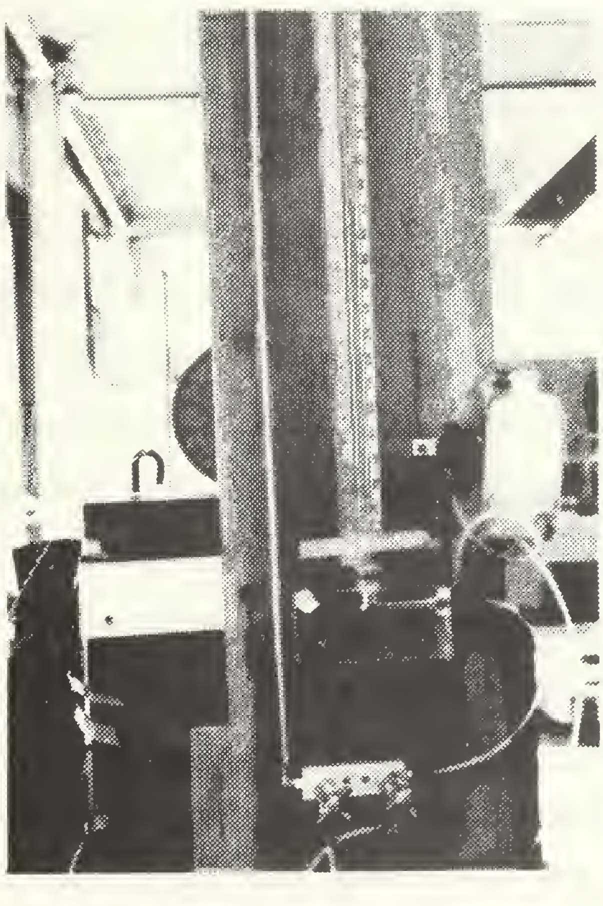


Figure 13. Fuel System

In order to avoid the damage to the pulse amplifier, use of the Strobotac was abandoned in favor of the Honeywell-Wein chronoscope. The chronoscope is a variable stroboscopic flash unit for photographic applications which is triggered, remotely or locally, by a switch. Since remote triggering is also available on the Hewlett-Packard pulse generator this mode was selected and a common switch was connected between the two units. Efforts to obtain photographs of the spray again proved futile, in this case due to a delay in the triggering of the pulse generator after the switch was closed. Since it became obvious at this point that attempts at synchronization were futile without a variable delay of some kind, attempts to synchronize the light with the injection were abandoned.

After abandonment of attempts at synchronization it was decided to attempt to photograph the spray utilizing the open flash technique. In this method, the chamber windows were covered with black paper except for a slit in the covering of the right side window for the lighting and access in the front window for the camera lens. To take a picture, the pulse generator is set for continuous pulsing at maximum repetition rate, the camera shutter is opened, and the chronoscope triggered. The camera shutter is then closed. It was found that the light from the chronoscope was of insufficient intensity to highlight the droplets from a single pulse and that multiple flashes were required to obtain an image on the film. Photographs obtained by

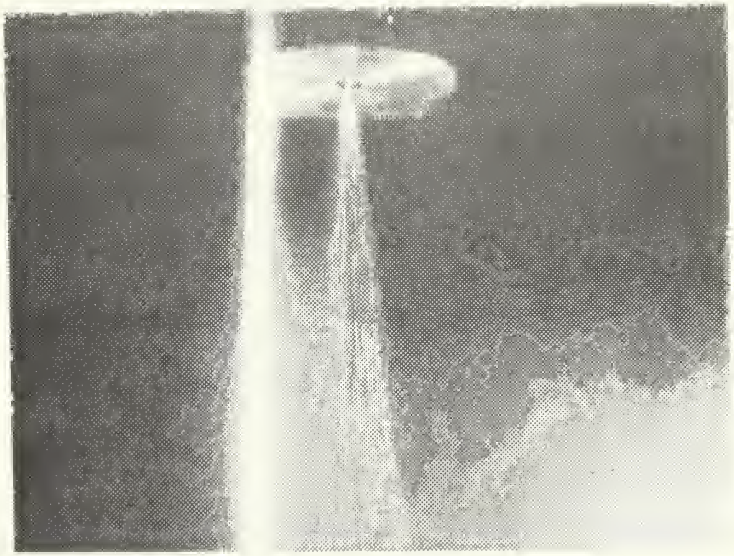
this method were useless except to determine the apex angle of the spray cone which was determined to be 16°-18° for all pulse widths. Figure 14 shows the spray cone for 7.0 and 10.0 ms pulses.

F. EXPERIMENTAL UNCERTAINTY

The principal cause of experimental uncertainty in this project is error incurred due to misreading of the fuel column heights in the fuel sight glass. This error is estimated to be less than ±.02 inches. This results in an uncertainty of less than 0.04 inches/inch, or, in terms of fuel quantity, the uncertainty in determining fuel quantity is no more than 0.00603 inches^3/inch of change in fuel column height.

In order to minimize uncertainty in pulse width and amplitude these measurements were used as basis for comparison only, and were matched to photographs of previous pulses to insure repeatibility.

7.0 ms



10.0 ms

Figure 14

V. THEORY

In order to permit future work with the Bosch injector a formula for predicting the flow through the injector as a function of pulse duration and fuel pressure was sought. The similarity between the flow through the injector and that through the full-periphery ported valves commonly encountered in hydraulic controls suggests that a similar approach be taken. Accordingly, the following was proposed as a theoretical prediction of flow rate.

Merritt, in his text on hydraulic controls [2], gives the equation for flow through a rectangular orifice as

$$Q = C_d A_o \sqrt{\frac{2}{\rho} (P_1 - P_2)\, g_c} \tag{1}$$

where

Q = the volumetric flow rate

A_o = the orifice area

P_1 = the pressure upstream of the orifice

P_2 = the pressure downstream of the orifice

C_d = a constant

The attractiveness of this equation as pertains to the injector problem is that with proper selection of the constant losses within the injector can be neglected. C_d depends on geometry and Reynolds number, but is generally taken to be 0.6 where the flow is turbulent and the orifice

is sharp edged. Lacking better information this value was assumed for the purpose of an initial prediction and experiments were conducted to verify this value or experimentally determine a better coefficient.

The geometry of the injector (Figure 15) shows that when the valve is open the minimum area perpendicular to the flow occurs at section A-A. The flow area through this section is given by

(2) $$A_o = S(\sin\theta)(r + r_o)\pi$$

but,

$$r = r_o - \varepsilon$$

$$= r_o - \delta \sin\theta$$

$$= r_o - S \sin^2\theta$$

Since $\theta = 45°$

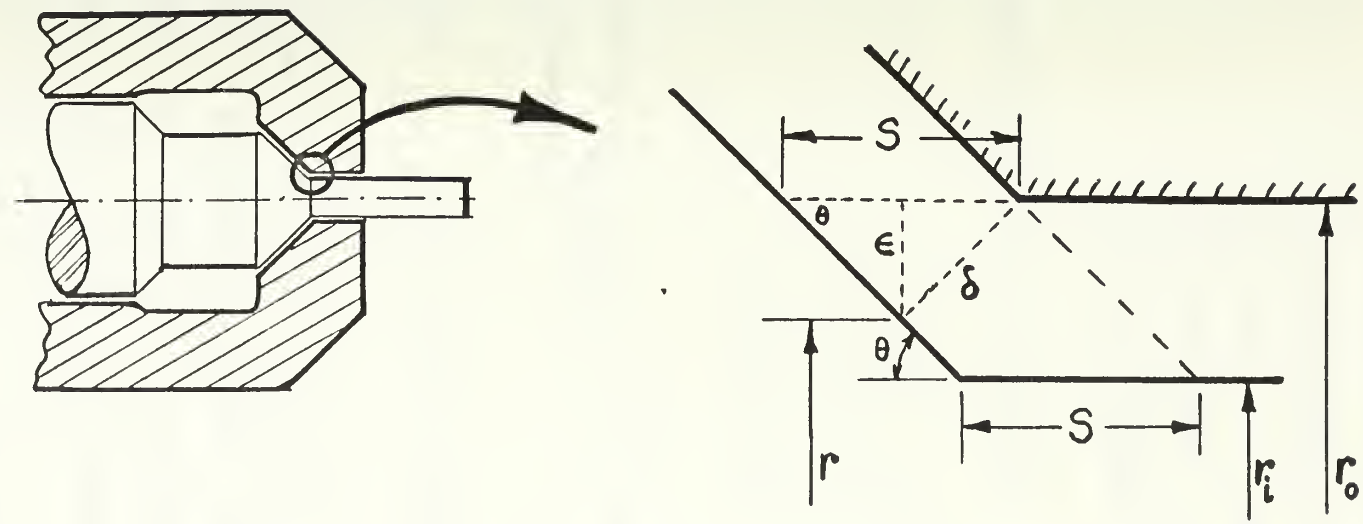
$$r = r_o - \frac{S}{2}$$

$$= .024 - \frac{.005}{2} = .0215$$

Thus

$$A_o = S(\sin\theta)(.0215 + .024)$$

$$= .500 \times 10^{-3} \text{ in}^2$$



radial clearance = $r_o - r_i$ = .004 in

minimum flow clearance = $\delta = S \sin\theta$ = .0035 in

$\theta = 45°$

r_i = .020 in

r_o = .024 in

S = .005 in

FIGURE 15

where

δ = the slant length of the opening

S = the stroke of the injector valve (.005 inch)

r_o = the outer radius of the annulus (.024 inch)

r = the inner radius of the annulus

θ = the shoulder angle (45°)

Substituting in (1) yields

$$(4) \qquad Q = (.500 \times 10^{-3})(.6)\sqrt{\frac{2}{\rho}(P_1 - P_2)g_c}$$

$$= .300 \times 10^{-3} \quad \frac{2}{\rho}(P_1 - P_2)g_c$$

Taking $\rho_{gasoline}$ = .0246 lbm/in^3 and making the necessary conversions

$$(5) \qquad \dot{Q} = 5.32 \times 10^{-2}\sqrt{P_1 - P_2}$$

Substituting various values of $(P_1 - P_2)$ in (5) the values for Q tabulated in Table 1 are obtained. Checking the Reynolds number for the tabulated cases indicates values on the order of 10^3, which gives reasonable expectation of turbulent flow.

TABLE 1

$P_1 - P_2$	28.46	32.46	34.84	38.84
$\sqrt{P_1 - P_2}$	5.335	5.698	5.903	6.232
$\dot{Q}$ (in^3/sec)	.284	.303	.314	.331

A. THEORETICAL RESPONSE

In order to determine the theoretical response time of the injector a free body diagram of the injector valve is drawn (see Figure 16). In the figure,

F_S = force attributable to the return spring

F_P = pressure force

F_C = tractive force of the electromagnet

r_1 = radius of the coil plunger = 0.1375 inches

r_2 = radius of the valve shaft = 0.0485 inches

Since there is pressure on both sides of the solenoid plunger the total static fluid force on the valve is given by

$$F_P = P_f(\pi r_1^2 - \pi(r_1^2 - r_2^2))$$

$$= P_f(\pi r_2^2)$$

where P_f is defined as the difference between fuel and chamber pressure (psia). Using the given value for r_2, and P_f = 38 psia

$$F_P = 0.2736 \text{ lbf}$$

The spring force is given by

$$F_S = kx$$

where k is the spring constant (determined experimentally to

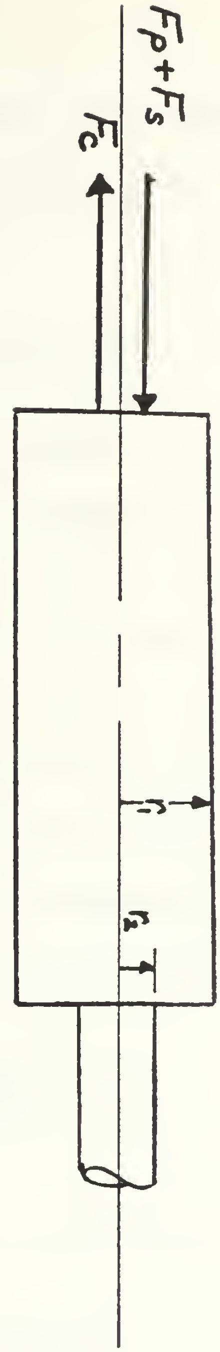


Figure 16

be 10.71 lbf/in) and x the spring deflection. When the valve is seated, the spring is compressed an estimated 0.15 inches; thus,

$$F_S = 0.15(10.7) = 1.601 \text{ lbf}$$

Prior to estimating the tractive force of the magnet, coil parameters were determined as follows:

N = number of turns = 304

E = operating voltage (steady state) = 3.5 volts

R = coil resistance = 2.4 ohms

g = working air gap = 0.0160 inches

Magnet type: flat faced plunger, DC excitation

Rotors, in his exhaustive text on electromagnet design [3], gives the force equation for this type of magnet as

$$F_C = \frac{B^2 {r_1}^2}{22.9} \text{ lbf}$$

where B is the flux in the working air gap. Furthermore

$$NI = \frac{B\ g}{0.002233}$$

and, assuming the transient is short compared to the steady state

$$I = \frac{E}{R}$$

so,

$$\frac{(0.002233)(N\ \frac{E}{R})}{g} = B$$

Substituting into the force equation and including known values yields

$$F = 3.13 \text{ lbf}$$

at the start of the stroke. At the end of the stroke the air gap is decreased to 0.011 inches, and similar calculations give

$$F = 6.61 \text{ lbf}$$

Since the stroke is short, average force (F_{av} = 4.87 lbf) is used to determine acceleration. Referring again to Figure 15 and assuming constant F_S and F_P the force balance is written

$$F_{av} - (F_P + F_S) = Ma$$

Substituting the known mass of the valve and solving for acceleration (a)

$$a = \frac{4.87 - (1.6 + .2736)}{.0089}$$

$$= 336.6741 \text{ ft/sec}^2 = 4040.08 \text{ in/sec}^2$$

Then assuming constant acceleration as an approximation

$$S = \frac{1}{2}\, a\, t^2$$

where S is the injector stroke (0.005 inch).

Solving

$$t^2 = .0247 \times 10^{-4}$$

or $$t = .00157 \text{ sec.}$$

Due to the approximations made in these calculations the foregoing figure is considered to be a maximum, and the actual figure somewhat less.

VI. DISCUSSION OF RESULTS

A. DATA BASE

Figures 17 through 22 show the injection pulse encountered in the operating engine. In the no-load condition pulse width proved independent of engine speed, showing a duration of .0024 seconds and an amplitude of 4.5 volts over the entire speed range. Application of load (Figures 18 through 22) resulted in progressive lengthening of the injection pulse (Table 2) from .006 seconds at 24 indicated rear wheel horsepower to .0076 seconds at 85 indicated horsepower. No significant variation in injector operating voltage (pulse amplitude) was noted over the entire speed range. Data shown in Table 2 is graphically displayed as Figure 23.

B. INJECTOR RESPONSE

Figures 24 through 27 show the results of the experiments to determine the response of the injector to the input pulse. In each case the upper trace is the input signal and the lower trace the proximeter output indicating valve motion. For these figures the oscilloscope time scale is 2.0 ms/cm; vertical scales are: upper - 2 volts/cm; lower - 0.5 volts/cm. The figures show extremely fast response times, with a closing lag of (avg).0009 seconds, slightly longer than the .0008 second opening lag. Since the opening and closing lag are so very nearly equal, the net effect is to produce an injection pulse of duration equivalent to the input pulse but occurring somewhat later.

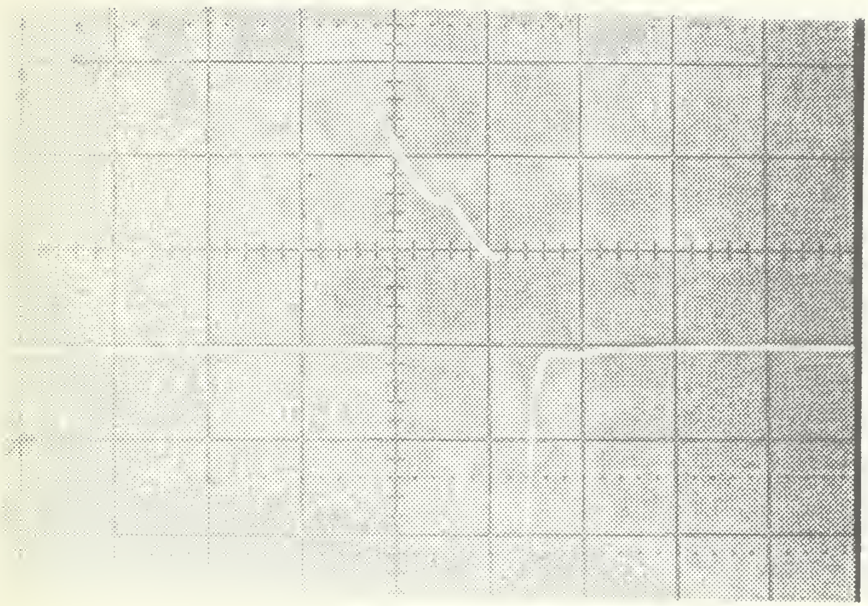


Figure 17. No Load

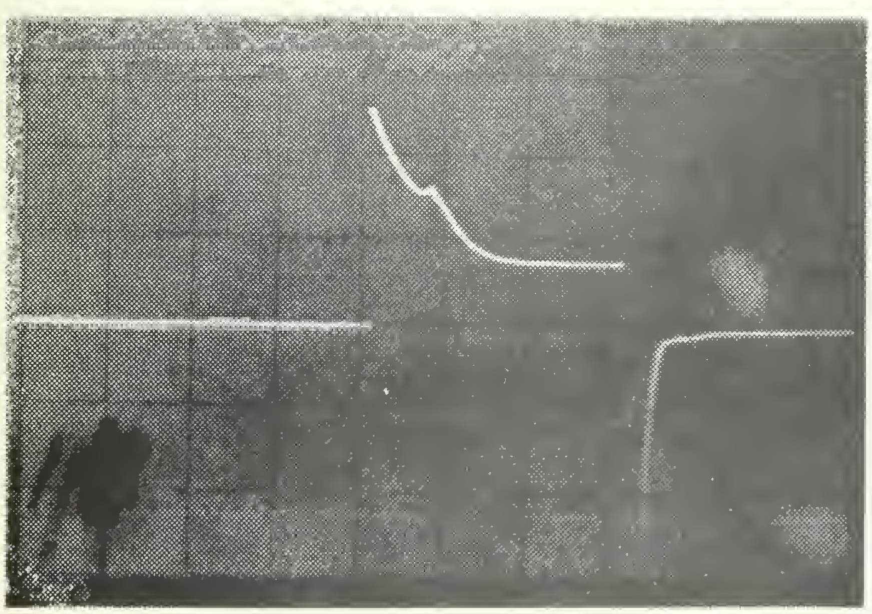


Figure 18. 24 HP

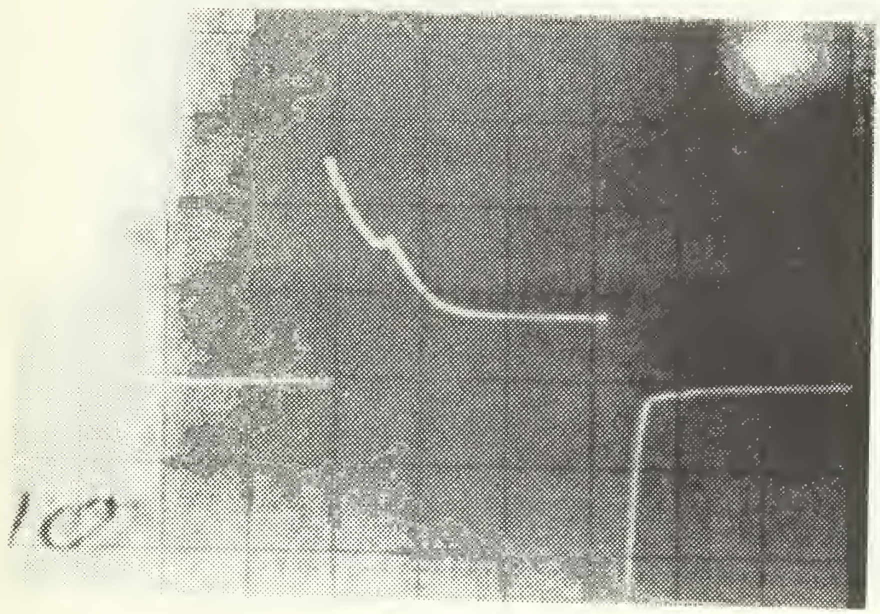


Figure 19. 35 HP

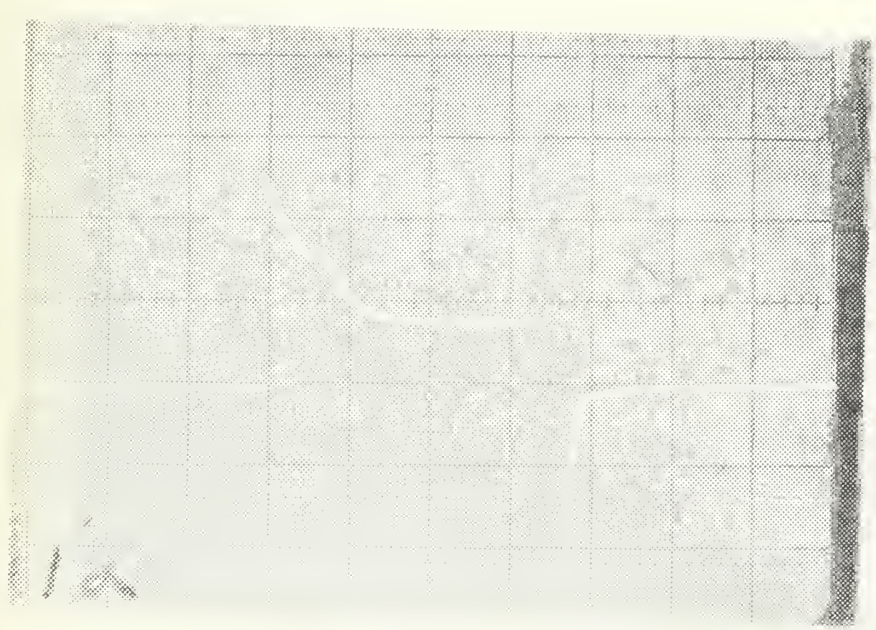


Figure 20. 55 HP

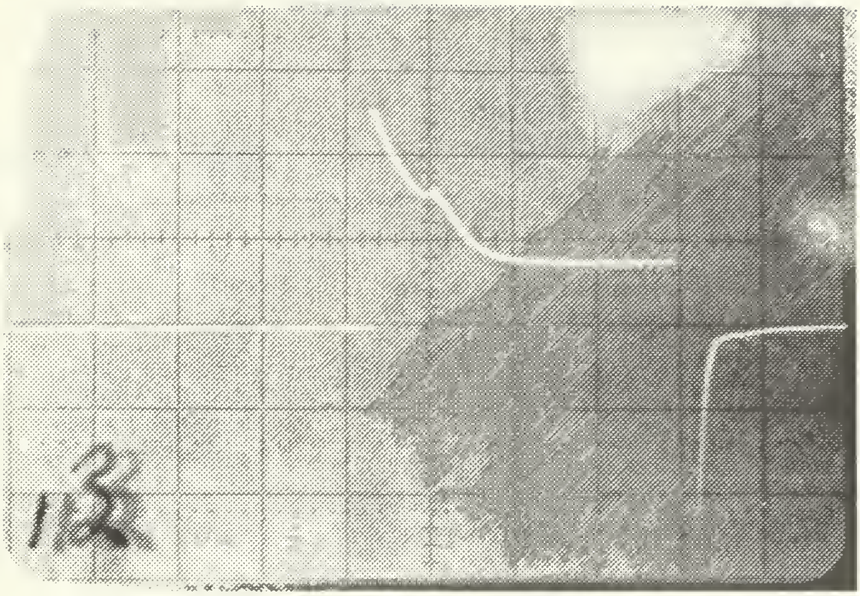


Figure 21. 73 HP

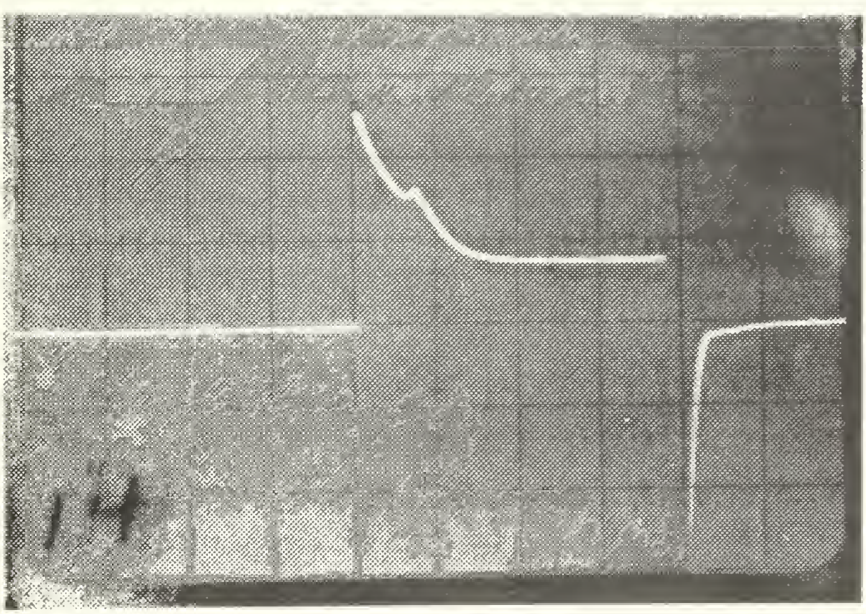


Figure 22. 85 HP

TABLE 2

Engine Speed (RPM)	Load (HP)	Pulse Amplitude (V)	Width (ms)
ALL	NONE	4.5	2.4
1500	24	3.5	6.0
2000	35	3.5	6.4
3000	55	3.8	6.8
4000	73	3.5	7.2
5000	85	3.5	7.6

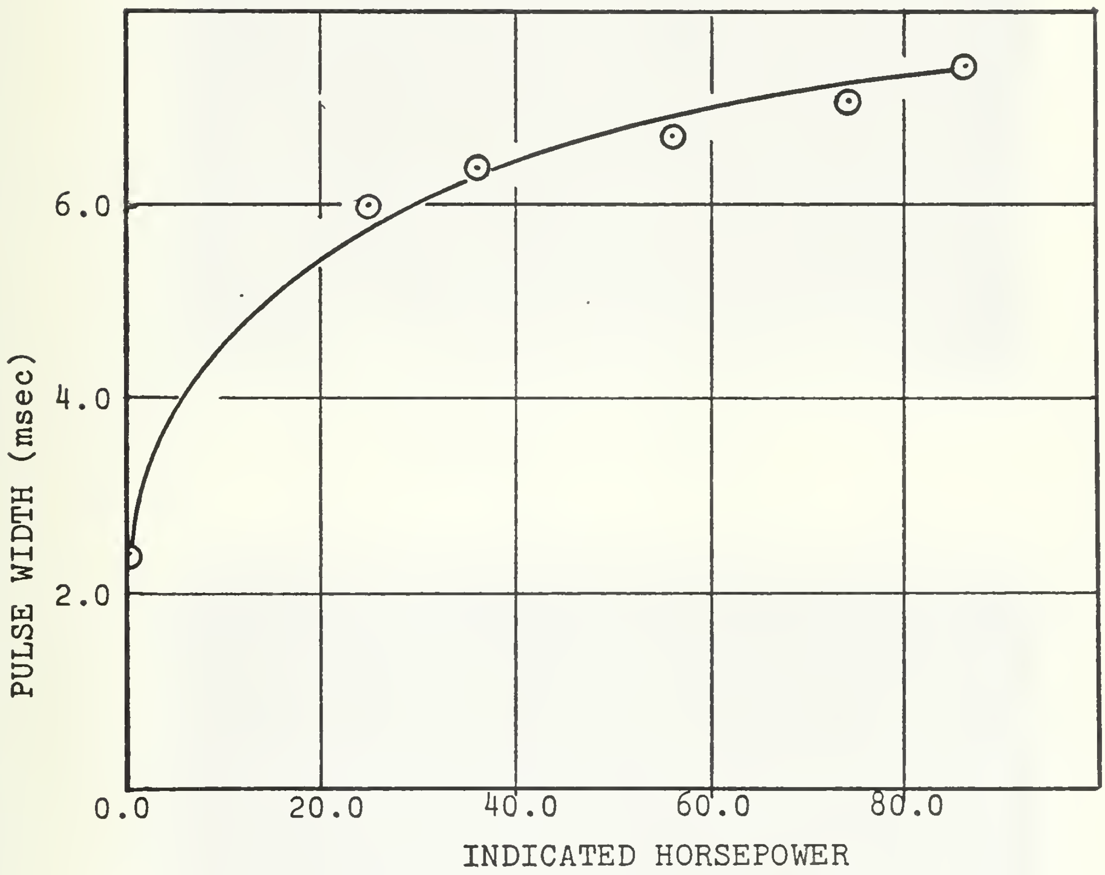


Figure 23. Pulse width vs. rear wheel horsepower

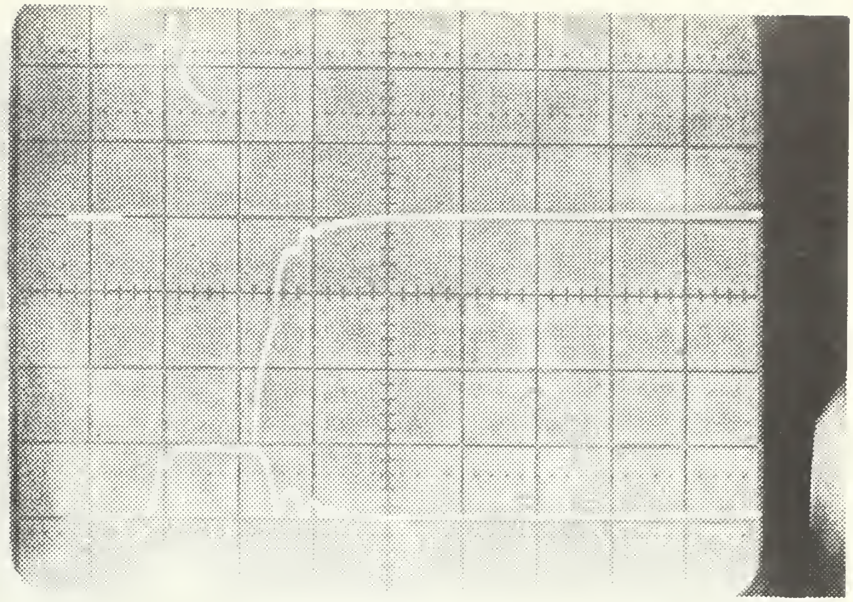


Figure 24. 2 msec

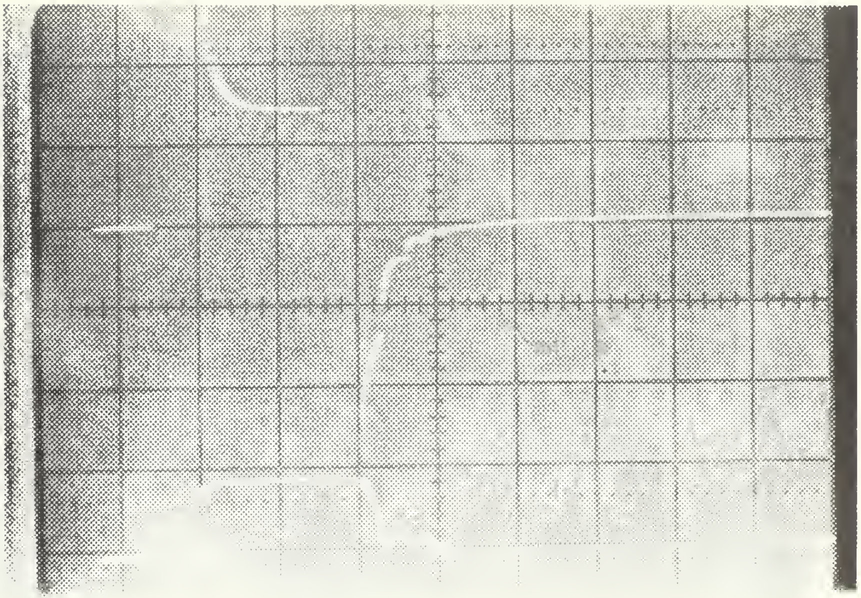


Figure 25. 4 msec

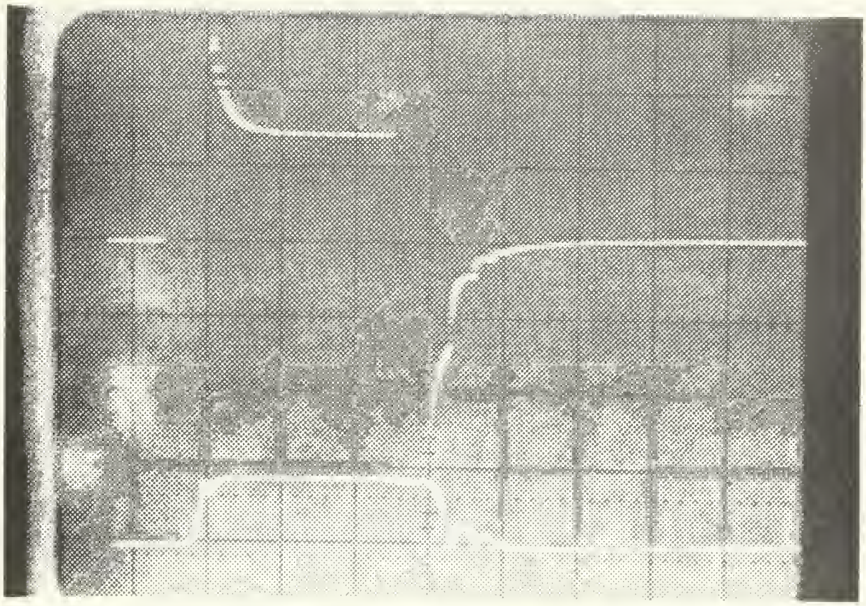


Figure 26. 6 msec

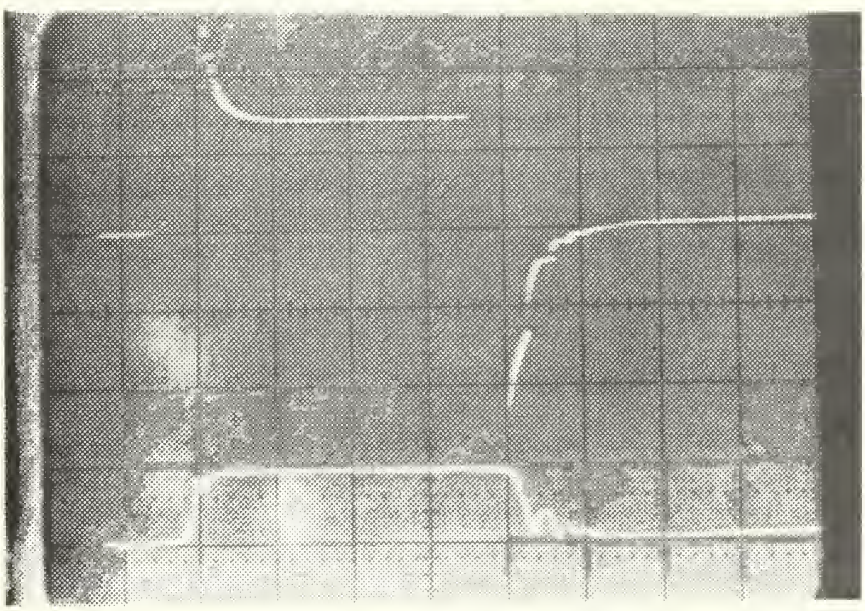


Figure 27. 8 msec

Theoretical predictions of injector response indicated a time of 1.5 ms, while observed times were 0.8 ms. The discrepancy is attributed to

(1) the partly empirical nature of the force equation used in the prediction

(2) the uncertainty of the actual width of the air gap (g), which was estimated as no direct measurement was possible

(3) the non-linearity of the tractive force of the magnet as the gap decreases during valve opening.

C. LABORATORY SIMULATION

Figure 28 displays graphically the results of data taken (Table 3) to determine the dependency of fuel flow on injector operating voltage. As can be seen from the figure there is no significant change in fuel flow for a variation of injector operating voltage from 3.0 to 4.0 volts. It can also be seen that over the entire operating range of 2.0 to 10.0 milliseconds the fuel flow through the injector is a linear function of pulse duration.

The effect of effective fuel pressure, here defined as the difference between the indicated fuel pressure (psia) and the chamber pressure (psia) is shown in Table 4 and graphically displayed in Figure 29. This data indicates a strong interdependence between the quantity of fuel dispensed and the effective fuel pressure. Fuel pressure data taken from the operating engine indicates that fuel pressure is

TABLE 3

Pulse Duration (msec)	Quantity/Injection (in^3) Injector Voltage 3.0	3.5	4.0
2	.00114	.00140	.00124
4	.00196	.00196	.00209
6	.00286	.00286	.00294
8	.00387	.00376	.00366
10	.00455	.00462	.00446

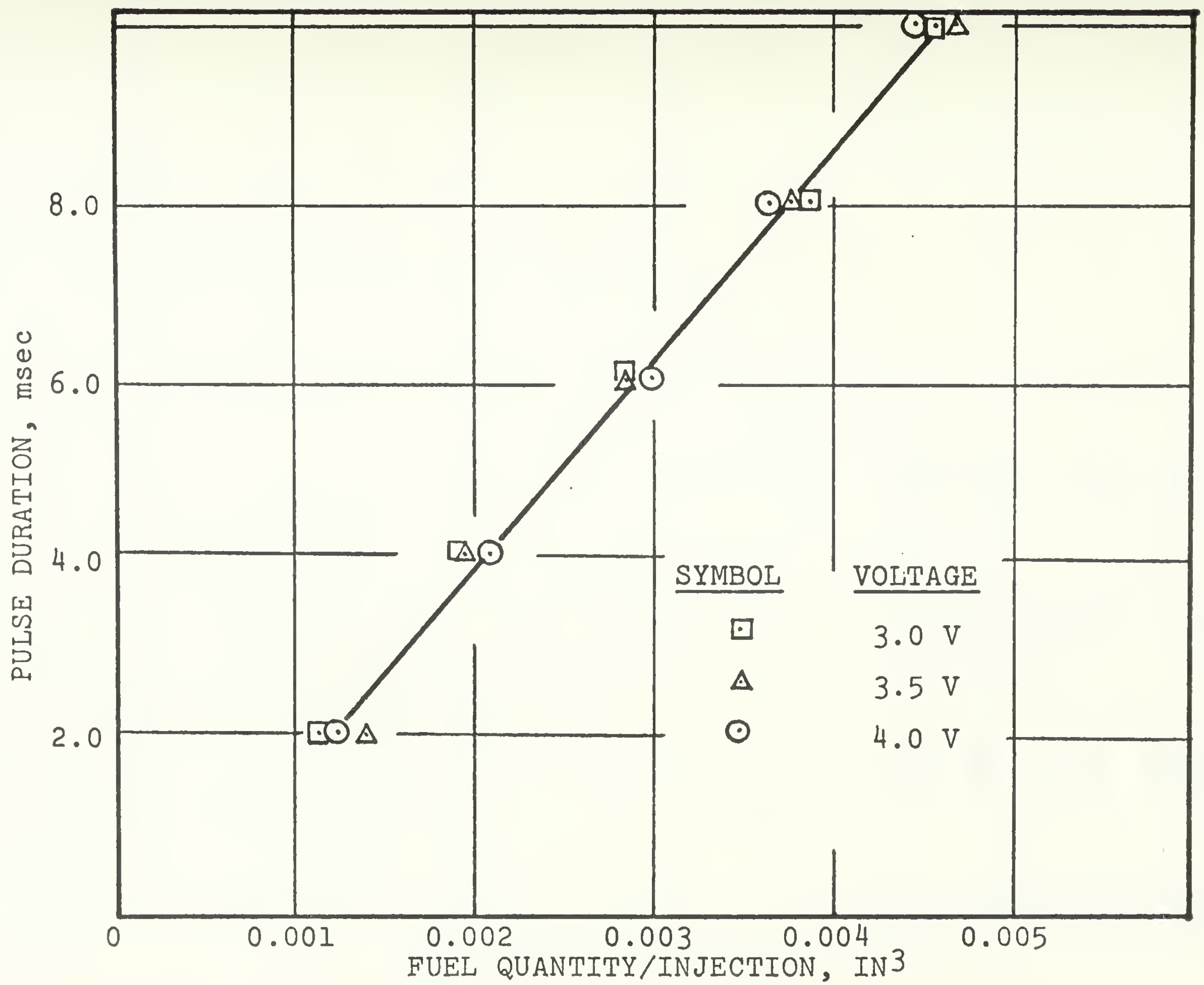


Figure 28. Fuel Quantity/Injection versus Pulse Duration
Various voltages

TABLE 4

Pulse Duration (ms)	Fuel Quantity/Injection (in^3) at $(P_1 - P_2)$ = 28.46	32.46	34.84	38.84
2	.00105	.00148	.00116	.00140
4	.00179	.00233	.00194	.00196
6	.00251	.00286	.00276	.00286
8	.00318	.00344	.00351	.00376
10	.00387	.00422	.00425	.00462

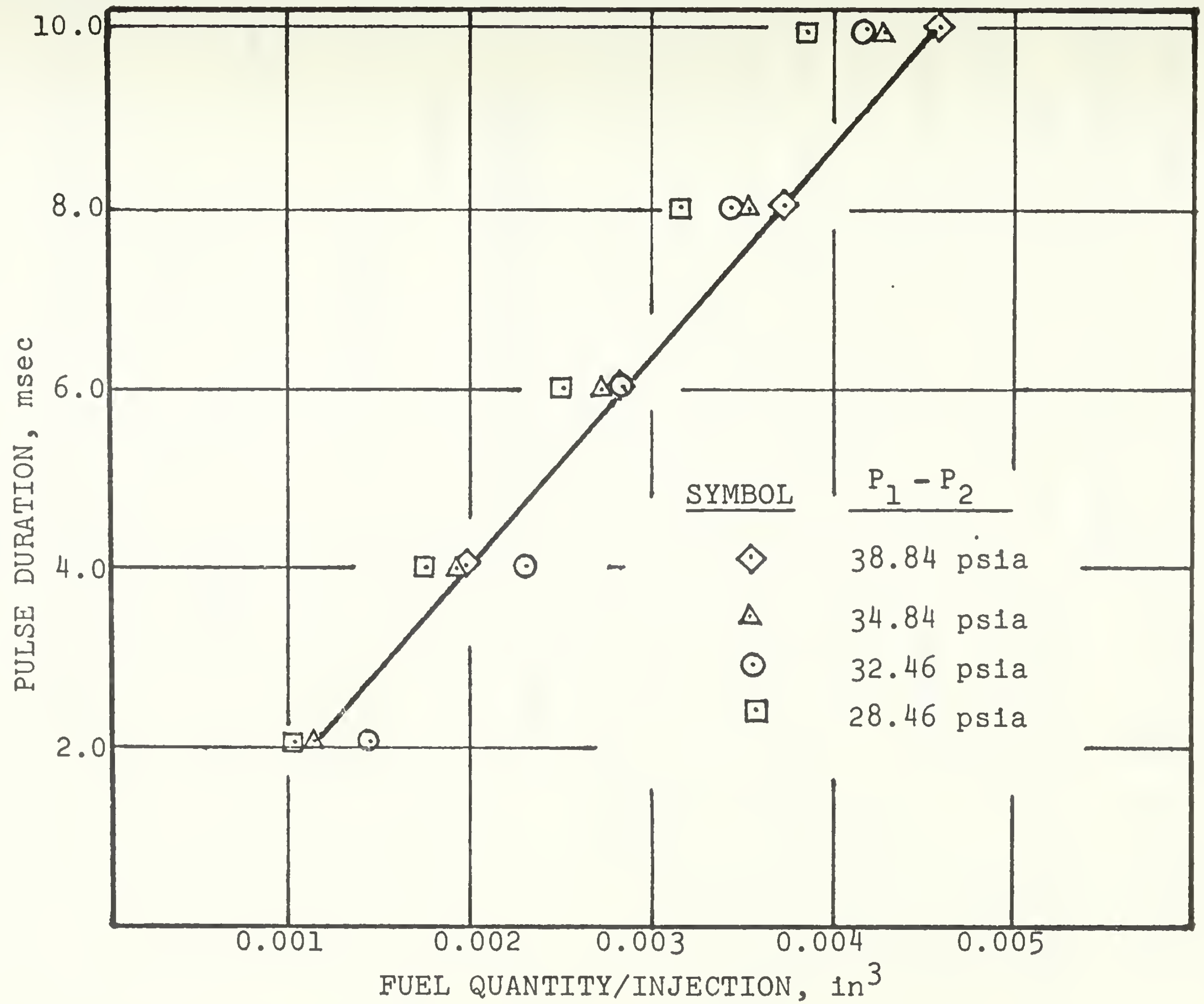


Figure 29. Quantity versus Pulse Duration, various pressures

maintained with a high degree of accuracy. It should be further noted that intake manifold pressure is sensed and provided as an input to the on-board pulse-length computer (modulator). This permits modification of pulse duration to compensate for changes in manifold vacuum, and is the primary method of load sensing.

Duplication of the pulses obtained from the operating engine resulted in the data shown in Table 5. Plotting this data against engine load (normalized to full load) results in the curve shown in Figure 30. It should be noted that while available literature indicates that the computer has a capability of producing pulses with durations of as long as .010 seconds, the longest pulse utilized by the Volvo engine installation was .0076 seconds.

D. EXPERIMENTAL DETERMINATION OF OVERALL DISCHARGE COEFFICIENT, C_D

Since

$$Q_{theory} = C_d \cdot K\sqrt{\Delta P}$$

and

$$Q_{obs} = C_{obs} \cdot K\sqrt{\Delta P}$$

C_{obs} can be found at each data point by

$$C_{obs} = C_d \frac{Q_{obs}}{Q_{theory}}$$

TABLE 5

Manifold (chamber) vacuum 18 in Hg

Fuel Pressure	30 PSI
Load-Normalized to Maximum Load	Quantity/Injection in^3
0	.00168
0.2823	.00309
0.4117	.00304
0.6470	.00310
0.8588	.00318
1.0000	.00367

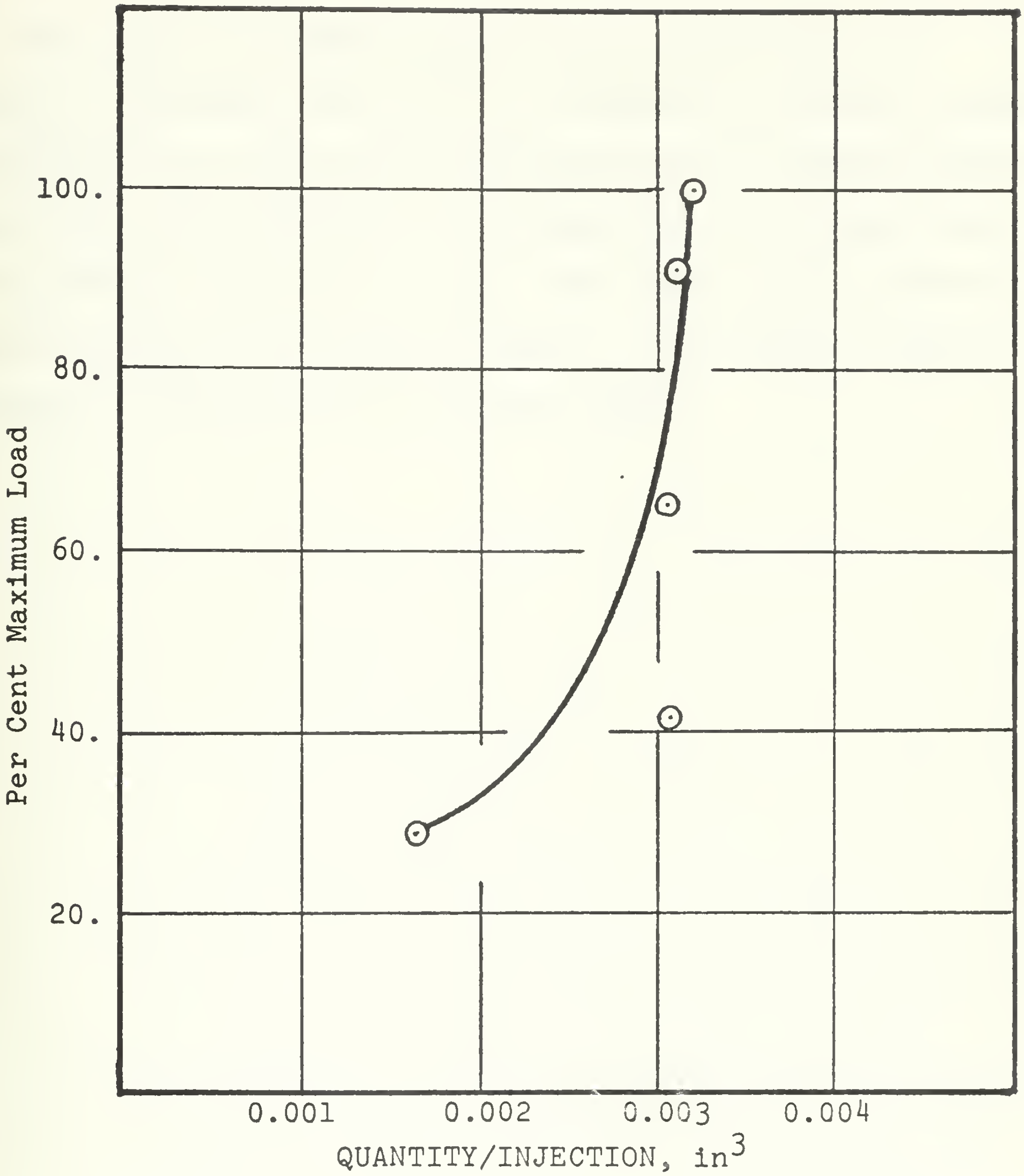


Figure 30. Quantity versus per cent maximum load

The results of these calculations are tabulated in Table 6. Plotting flow rate as a function of pressure across the injector for both the theoretical and the observed cases (see Table 6 and Figure 31) indicates that the average value of C_D for the injector is .738. This value may be rationalized by attributing a finite length to the injector orifice instead of assuming it to be sharp edged.

TABLE 6

$\sqrt{\Delta P}$	5.335	5.698	5.903	6.232
$\dot{Q}_{obs}$ (in^3/sec)	.352	.343	.386	.403
$\dot{Q}_{theory}$	.284	.303	.314	.331
C_{obs}	.743	.680	.738	.730

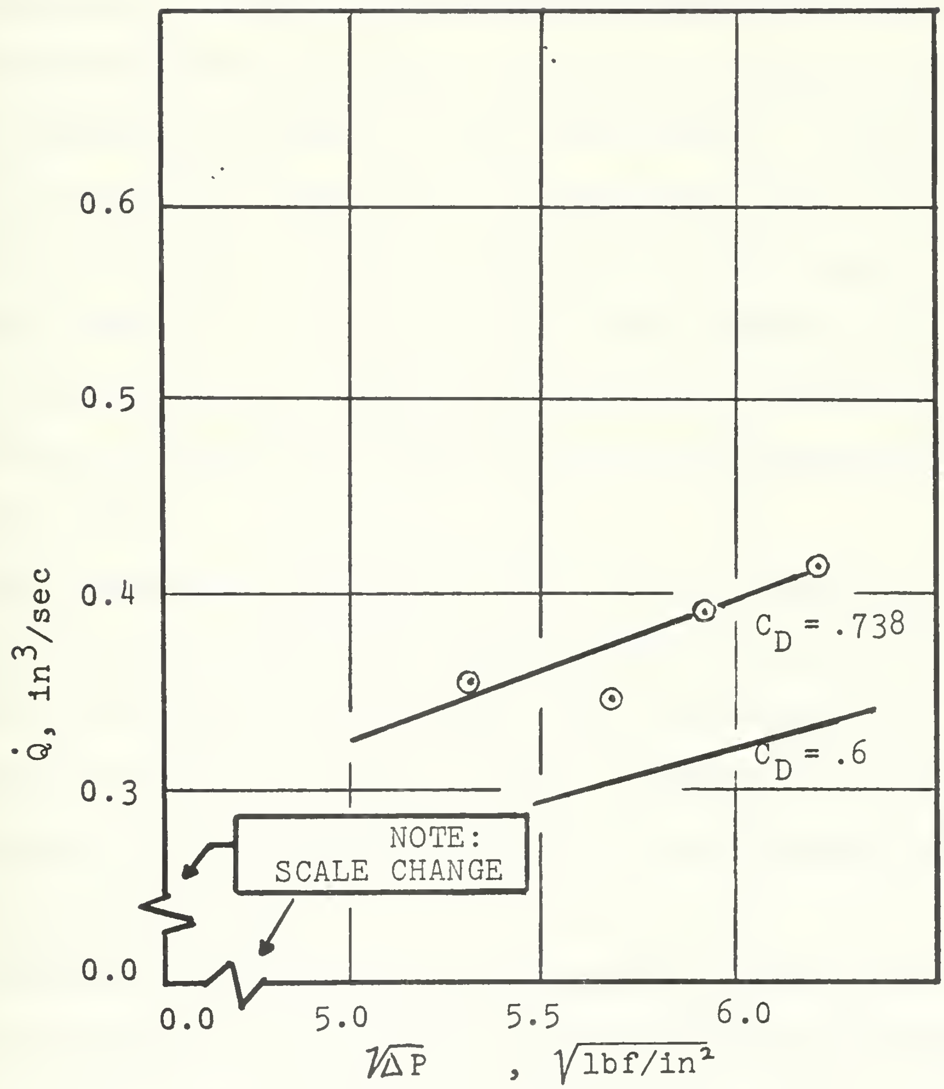


Figure 31. Flow rate versus square root of pressure difference

VII. CONCLUSIONS

The Bosch injector accurately meters a quantity of fuel and dispenses it in a narrow (16-18°) spray cone. The overall discharge coefficient for the injector, based on the total pressure difference across the injector, was determined to be .738. When used in conjunction with the orifice equation this value permits prediction of flow rates. The proposed method for determining the valve opening time will yield times which are longer than the actual times; however, the method is useful for generating a first approximation of response time.

The laboratory simulation of the input pulse as well as the test chamber and vacuum system worked well and proved useful in the collection of data. The fuel system as presently designed is prone to leak. Redesigning of the fuel system will be required prior to accomplishing future work with the system.

Attempts to photograph the spray from a single injection were unsuccessful, due to the minute quantities dispensed and lack of a light source of sufficient intensity. An alternate method of accomplishing this objective, not attempted due to time constraints, is proposed.

VIII. RECOMMENDATIONS FOR FUTURE WORK

The injector and fuel injection systems in general provide an interesting and potentially valuable field of study for the engineer. Particular areas suggested for continued study are:

(1) verification of the proposed value of C_d at higher pressure ratios; accomplishment of this task would require redesign of the fuel system

(2) analysis of the atomization and mixing of the spray, including photographic analysis utilizing a high speed motion picture camera

(3) refinement of the theoretical dynamic response of the injector and experimental verification of the tractive force of the magnet.

It should also prove instructive to attempt conversion of a small engine to fuel injection, comparing its performance before and after conversion. While it is recognized that considerable work would be required to accomplish this, it is not felt to be beyond the in-house capabilities of the mechanical engineering department.

APPENDIX A. CALIBRATION

The basic measurements necessary for completion of this project, with their respective measuring devices, are:

(1) Chamber vacuum - direct reading gauge
(2) Pulse width - Hewlett-Packard oscilloscope
(3) Pulse amplitude - Hewlett-Packard oscilloscope
(4) Fuel flow - sight glass
(5) Injector response - Bentley-Nevada proximeter

Of the above measurements the first three were obtained from direct reading displays previously calibrated and needed no further calibration to meet the requirements of the project. In the case of the last two items, the readout is dependent on the particular configuration. It was thus necessary to produce by experimentation the appropriate calibration curves for the proximeter and the fuel sight glass.

A. THE FUEL SIGHT GLASS

The fuel reservoir is in fact a sight glass of (nominal) 11 mm (.433 inch) I.D. A metal scale calibrated to 1/100 inch was attached to the frame alongside the tube to enable direct reading of the fuel column height. In all cases, the edge of the meniscus, which was sharp and well defined, was used as the reference point for measurements.

In order to calibrate the fuel tube a 100 ml graduated cylinder was obtained. The initial volume of fuel in the graduate was recorded, and then fuel was allowed to flow into the graduate (through the Poly-Flo tube normally

connected to the injector inlet) until the fuel column had fallen 1/2 inch. The volume in the graduate was recorded and the process repeated, generating the data in Table A1. The graph of these data points (Figure A1) is a straight line whose slope is the desired relation between quantity and change in column height. Since some fuel was in the graduate at the start of data collection, the origin of Figure A1 is displaced to the left.

Determination of quantity/inch proceeds as follows:

From Figure A1

$$\Delta h_t = \text{total change in column height}$$

$$= h_2 - h_1 = 4.25 \text{ inches}$$

$$\Delta Q_{total} = Q_2 - Q_1 = 10.5 - 0.0 \text{ ml}$$

$$\text{slope} = \frac{\Delta Q_t}{\Delta h_t} = \frac{10.5}{4.5} = 2.4705 \text{ ml/in}$$

$$= 0.1507 \text{ in}^3\text{/in}$$

B. THE PROXIMETER

Because of the small size of the injector valve tip relative to the surrounding large mass of metal, it was necessary to design a device to calibrate the proximeter which would enable it to sense valve tip motion without a large inductive load caused by the surroundings. Accordingly, the jig designed for the planned injector valve motion study had incorporated into its design provision for proximeter

TABLE A1

TOTAL VOLUME CHANGE	TOTAL CHANGE IN HEIGHT
0.5	0.5
1.9	1.0
3.0	1.5
4.4	2.0
5.5	2.5
6.6	3.0
8.0	3.5
9.4	4.0
10.5	4.5

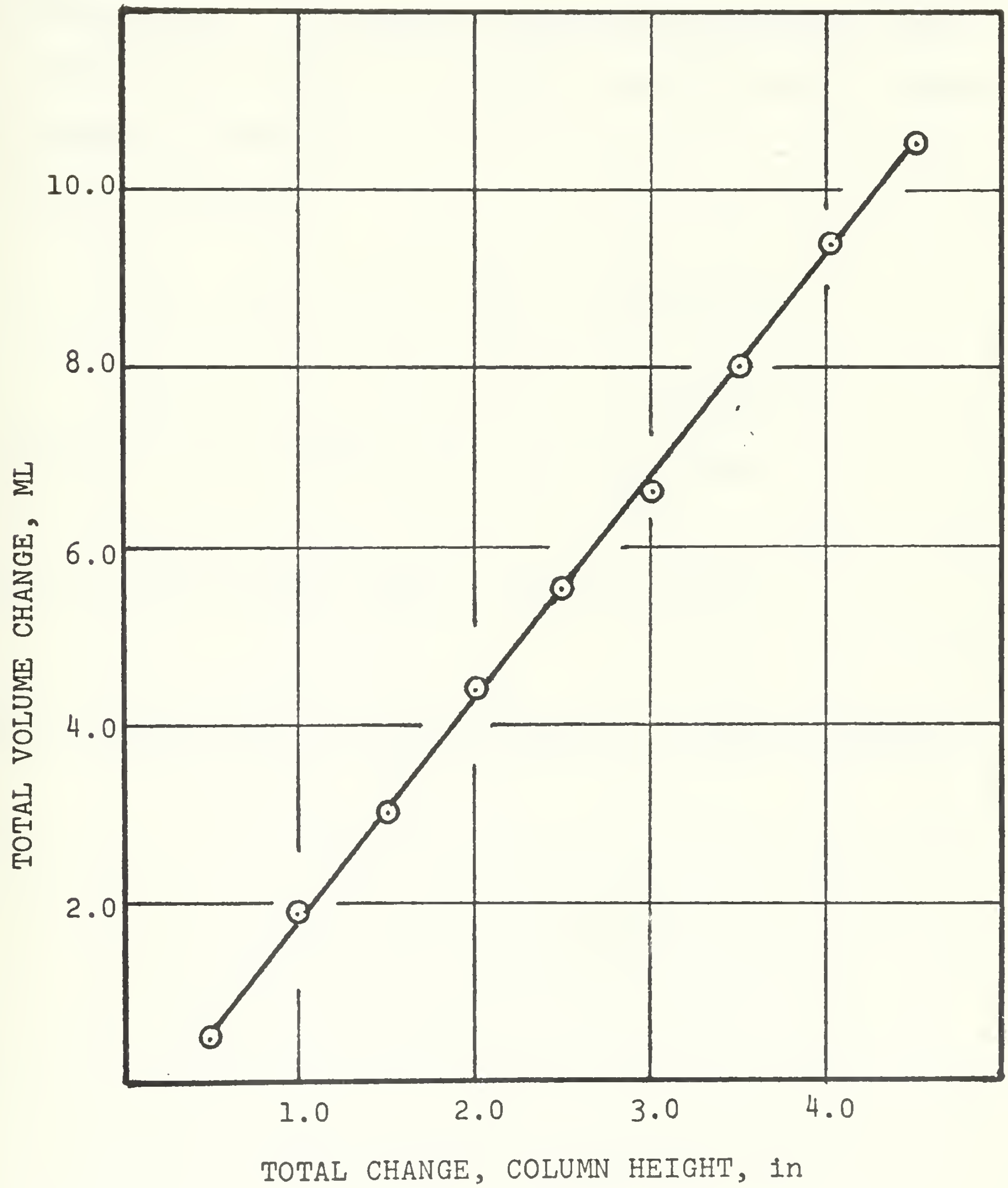


Figure A1. ΔV versus Δh

calibration. A dial indicator with a least count of 0.0005 inches was mounted so that it measured the movement of the proximeter mount. The output of the detector/power supply was connected to a digital voltmeter and the results (Table A2) plotted as Figure A2.

From the figure:

$$X = X_2 - X_1 = .010 - .005 = .005 \text{ inches}$$

$$V = V_2 - V_1 = 1.559 - 1.073 = .4860 \text{ volts}$$

$$\text{slope} = V/X = .486/.005 = 97.2 \text{ volts/inch}$$

or more conventiently, slope = $.0972$ mv/.001 in^2

Figure A2 shows the excellent linearity of the proximeter.

TABLE A2

DEFLECTION	VOLTAGE
.0050	1.073
.0055	1.127
.0060	1.175
.0065	1.222
.0070	1.269
.0075	1.316
.0080	1.365
.0085	1.416
.0090	1.460
.0095	1.512
.012	1.559

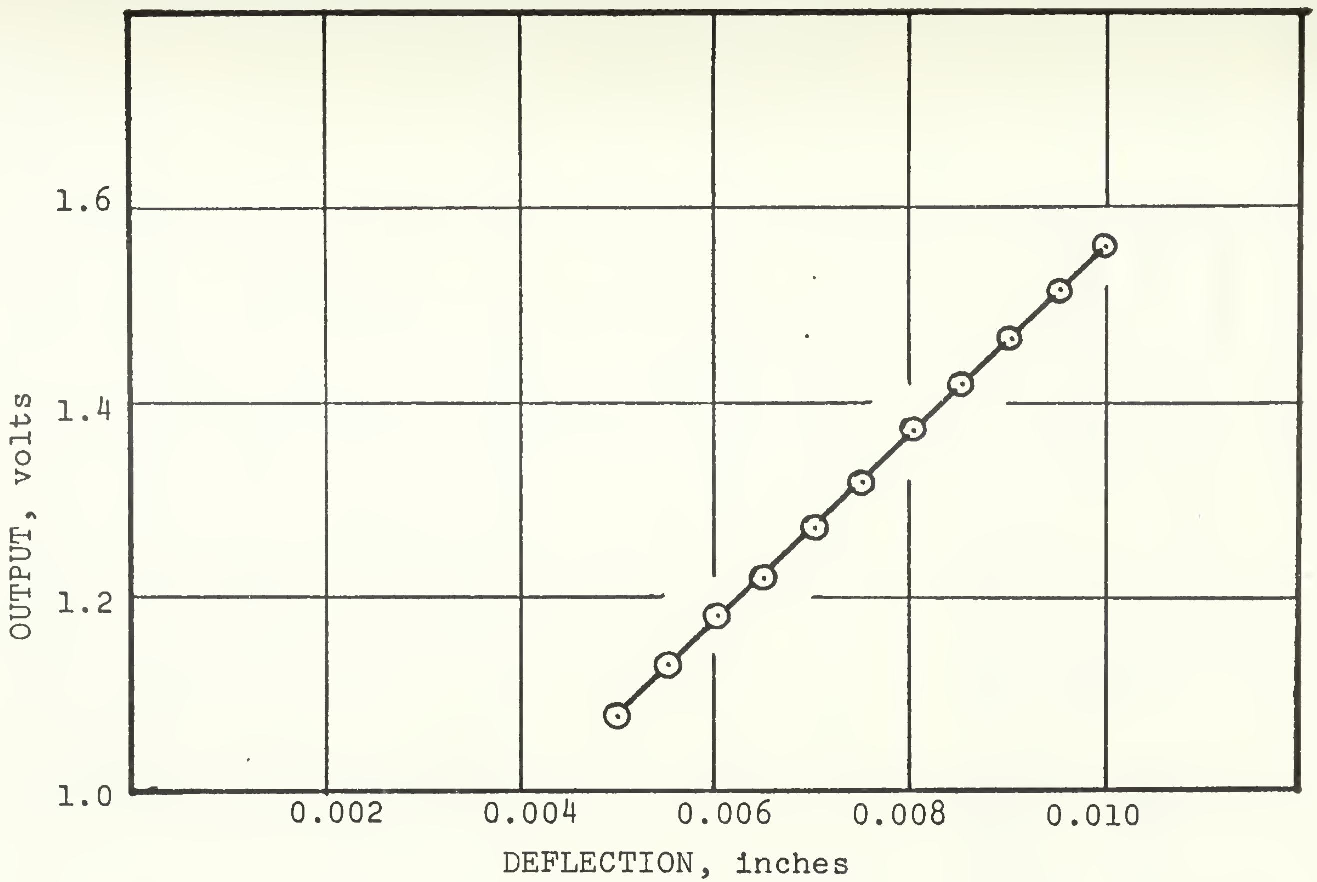


Figure A2. Deflection versus proximeter output

APPENDIX B

Figure B1 is provided for information and indicates chamber vacuum as a function of air ejector inlet pressure when the following apply:

Chamber volume - 3.4 ft^3

Supply pressure to ejector - 100 psi

Chamber exhaust - 3/4 inch

Ejector exhaust-through 25 foot section
2 inch I.D. rubber hose

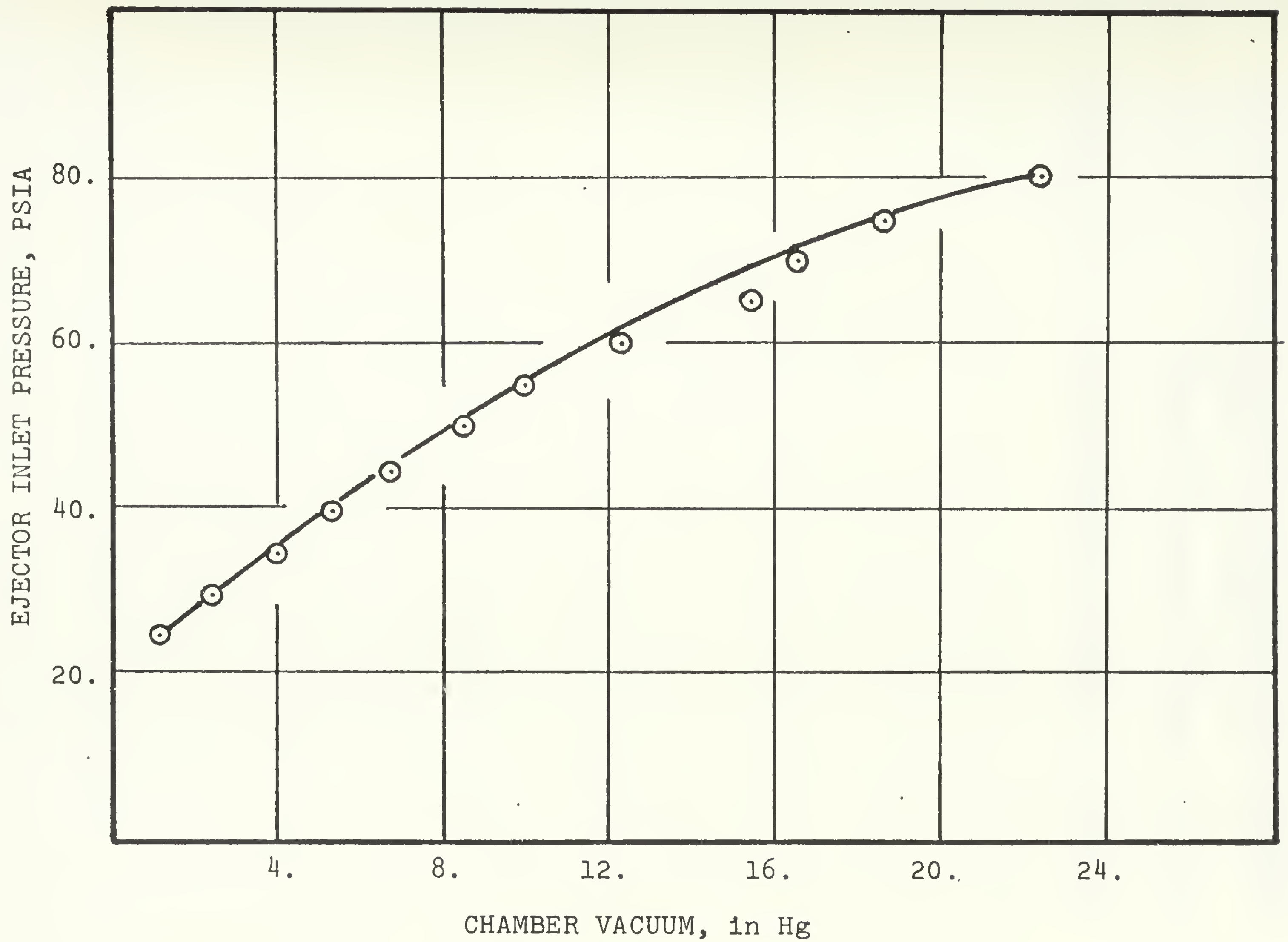


Figure B1. Chamber vacuum versus ejector inlet pressure

LIST OF REFERENCES

1. Croft, B. H., and others, The Petrol Fuel Injection Book for Automobiles, Second edition, Tractor and Mechanical Publications, Brisbane, 1972.

2. Merritt, H. E., Hydraulic Control Systems, p. 39-49, Wiley, 1967.

3. Roters, H. C., Electromagnetic Devices, p. 260-268, Wiley, 1941.

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Due to the increasing requirements for reduced emissions from internal combustion engines many manufacturers have expressed interest in gasoline fuel injection systems. In order to determine the fluid dynamic and dynamic characteristics of a commercial injector a test chamber was constructed and a laboratory simulation carried out. A proposed method of predicting flow rates and dynamic response has been compared with experimental results.

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