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FUEL SYSTEMS FOR HEAT BALANCED INTERNAL COMBUSTION ENGINES.(U)
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A TRIDENT SCHOLAR PROJECT REPORT

NO. 94

"FUEL SYSTEMS FOR HEAT BALANCED INTERNAL COMBUSTION ENGINES"



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U.S.N.A. - TRIDENT SCHOLAR PROJECT REPORT; NO. 94 (1978)

"Fuel Systems for Heat Balanced Internal Combustion Engines"

A Trident Scholar Project Report

by

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ABSTRACT

The development of the heat balanced engine (HBE), as proposed by Blaser, has resulted in a new class of internal combustion engines with unique fuel delivery requirements. In this study the characteristics and requirements of fuel metering and control techniques required for internal combustion engines in general are investigated. A general approach to the design and selection of fuel delivery systems for heat balanced engines has been developed. Using this approach, a particular fuel delivery system compatible with a specific two-stroke heat balanced engine has been selected, fabricated, and installed. An engine testing facility and simple model formulated for the engine have been established and a complete test program outlined.



Preface

As men, we sometimes have a tendency to neglect the forest as we focus our attention on the trees.

Researchers are plagued with this problem no less than anyone else.

It requires a lot of thought and a tremendous amount of effort to bridge the gaps and tie together fields in which others are truly expert. It is indeed fortunate that those experts had the patience and understanding to allow their advisee to "do his own thing."

As always, the path I walked I did not walk alone; the names of those who assisted me in my work could fill a small book. A word of thanks to my "honorary" advisor, Mr. Richard F. Blaser, and to Professors Andrew A. Pouring and Bruce H. Rankin, advisors with whom I spent far too little time. The assistance of Mr. Ervin Leshner and and his son, Michael, of Fuel Injection Development Corporation, was indispensible; they freely shared with me their experience as I was busily engaged in adapting their Vaporized Fuel Injector to the Heat Balanced Engine. Kevin McDonald, the motorcycle maniac of East Coast Racing, was an ardent supporter, as were the personnel of the Technical Support Division as they bent over backwards to assist me in any way they could. Hats off to Bernie Diechgraber, the machinist who has tolerated me for so long, and to all the lab technicians who were so helpful. Mike Bailey was (and is) a champ and a true craftsman. Thanks to Associate Professor Keith for his advice, and to Mrs. Doris Keating for her superb typing job. My classmates George DeMarco, Larry McCracken, and Bill Yeager helped me more than they realize. And last, but by no means least, a

special word of thanks to my parents and my principal advisor, Assistant Professor Eugene L. Keating.

I have begun to discover the forest.

Annapolis
Maryland
17 May 1978

S. W. Petri

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To my brother, Robert John Petri:

Love in your heart wasn't put
there to stay;
Love isn't love
'til you give it away.

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1.0. INTRODUCTION

The feasibility of the heat balanced engine (HBE) has been amply demonstrated by Blaser on several occasions, both in single- and multi-cylinder engine configurations.[1]*

The unique fuel distribution requirements of the heat balanced cycle discussed in Appendix A, however, cannot be adequately met by existing conventional fuel delivery systems. Additionally, a general approach to the design of fuel systems for internal combustion engines is nonexistent.

2.0. OBJECTIVES

This study investigates fuel delivery techniques compatible with internal combustion engines, develops an operational preliminary design and model, and outlines a comprehensive test program to validate the proposed design.

3.0. FUEL DELIVERY SYSTEMS

Modern fuel delivery systems used in conjunction with internal combustion engines are briefly discussed in Appendix C. While many fuel delivery systems are far more complex than the simple systems diagrammed in the appendix, the principles of operation for all remain essentially the same.

The fuel systems in existence today are geared towards producing either a homogeneous or a heterogeneous air/fuel mixture, not the stratified mixture required by the heat balanced engine. Existing fuel

*Footnotes appear at the end of the paper.

systems can be modified so as to be compatible with the heat balanced engine, but these modifications introduce undue complexity to the fuel system and compromise its performance.

What is needed is a simple fuel system designed specifically to deliver a stratified mixture to an internal combustion engine.

4.0. FUEL METERING AND CONTROL TECHNIQUES

4.1. DESIGN APPROACH

The lack of the requisite fuel system for the heat balanced engine forces the designer to either develop or select one of his own. Due to the complexity of the systems and engines involved, most research in the fuel delivery field depends heavily upon empirical techniques such as modeling and testing.

Generally, a model of the system under consideration is formulated, used to predict the effects of design decisions, and refined through continued testing. Before commencing this modeling/testing sequence, it is advantageous to first consider fuel metering and control techniques in general.

4.2. DESIGN CONSIDERATIONS

The principal task of fuel delivery systems is proper control of the air-fuel mixture. As indicated in Figure 1, the options initially open to the designer are to meter air or meter fuel. Since the mass of air involved in combustion is, on the average, about fifteen times greater than the mass of fuel, the approach normally taken in fuel system design

is to meter the fuel. Accordingly, the metering of air will not be considered in detail any further.

After selecting which fluid will be metered, the next decision to be made involves how the metering will be effected. Either discrete quantities of mass can be measured (mass metering) or appropriate flows can be controlled (flow metering) so that the desired amount of fuel is isolated.

From this point, the options open are restricted by the decisions already made. Mass metering, usually used in conjunction with the Diesel engine, can be achieved by controlling the volume of fuel delivered by the fuel system to the engine. Flow metering, on the other hand, can be accomplished by flow of fuel through a known orifice, fixing either the differential pressure across the orifice, the interval during which flow occurs, or neither.

After metering, the mixture delivered to the engine can be either a liquid, a homogeneous vapor, or a heterogeneous mixture of liquid and vapor. This metered mixture can then be delivered by the fuel system to the engine for combustion either directly or indirectly, via a duct or manifold.

The final consideration is that of control technique. Metering can be controlled either electronically, mechanically, or fluidically: electronic control is realized by electronic sensing and control elements; mechanical control utilizes linkages, gears, etc.; and fluidic control depends upon the physical and dynamic characteristics of the fluids themselves.

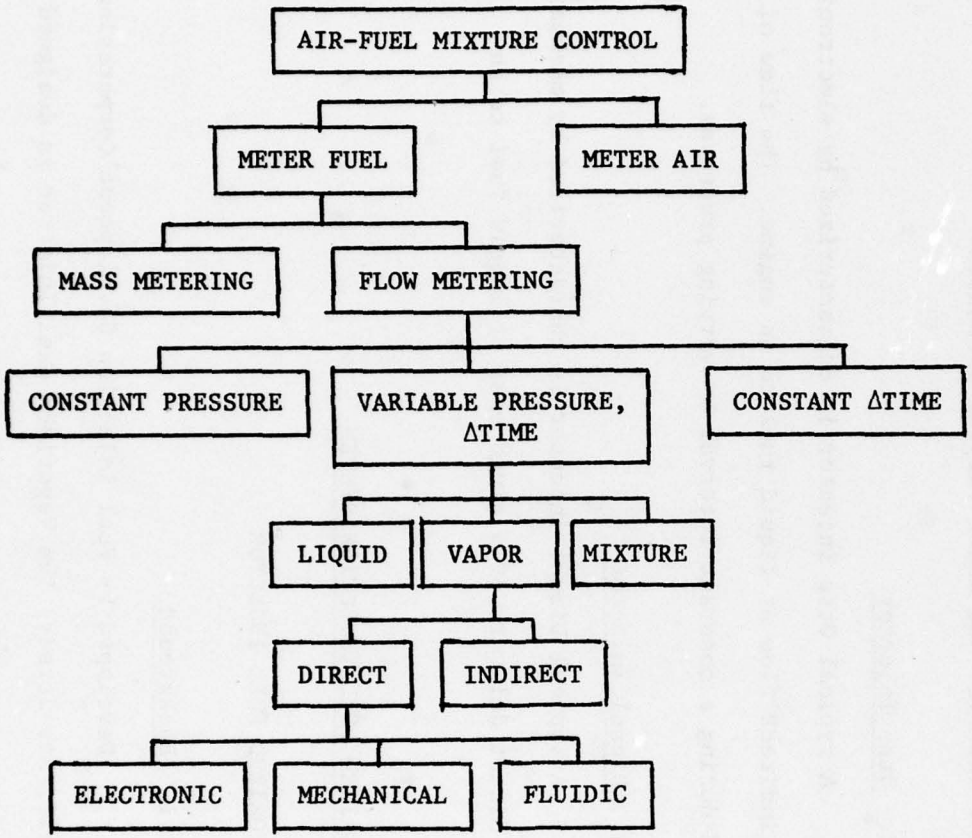


Figure 1. Fuel Metering and Control Techniques

4.3. EXAMPLES

4.3.1. Carburetor

A carburetor is an example of a mechanically and fluidically controlled device which indirectly delivers to an engine a heterogeneous mixture of fine fuel droplets suspended in air. This mixture is produced by the flow metering of fuel through fixed orifices during a constant interval and under varying pressures.

4.3.2. Otto Injector

A typical Otto injector is characterized by electronic control of the indirect flow of liquid fuel to an engine. The flow of fuel is metered during a constant interval by varying pressures.

4.3.3. Diesel Injector

A typical Diesel injector is characterized by mechanical control of the direct delivery of a given mass of liquid fuel to an engine.

5.0. MK 1 VAPORIZED FUEL INJECTION SYSTEM

5.1. VAPORIZED FUEL INJECTOR

5.1.1. Background

Developed by Fuel Injection Development Corporation, Inc., of Bellmawr, New Jersey, the Vaporized Fuel Injector is designed to deliver a stratified mixture to a two-stroke cycle engine. The Vaporized Fuel Injector fluidically controls the direct delivery of a vaporized, homogeneous fuel mixture to the engine under variable

differential pressure and during a variable flow interval. The proper mixture is achieved by the flow metering of fuel.

5.1.2. Operation

A schematic of the Vapor Fuel Injector is shown in Figure 2. Liquid fuel flows from the pressurized fuel tank through the fuel line (I) to the check valve (II), where back-flow from cylinder to fuel tank is prevented by a ball-check arrangement. During the combustion phase of engine operation, hot combustion gases travel up the injector (III) from the cylinder, turbulently mixing, preheating and vaporizing the fuel. When the cylinder pressure drops below fuel tank pressure, the check valve opens and allows a fresh charge of fuel to enter the injector as the previously vaporized fuel mixture already in the injector is drawn into the cylinder (IV).

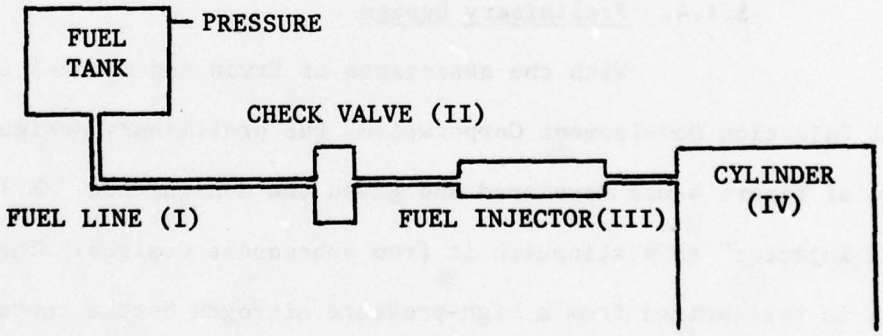


Figure 2. Schematic - MK 1 Vapor Fuel Injector

5.1.3. Applicability

The operating characteristics of the Vaporized Fuel Injector are compatible with the combustion characteristics and fuel

distribution requirements of a heat balanced engine operating on a two-stroke cycle. Consequently, the Vaporized Fuel Injector has been selected for use with the heat balanced engine.

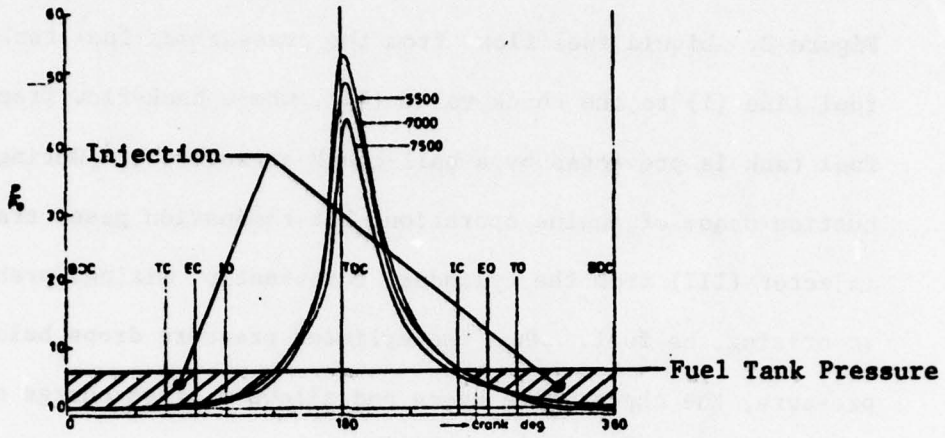


Figure 3. Typical Cylinder Pressure of Two-Cycle Engine (From Blair and Cahoon[2])

5.1.4. Preliminary Design

With the assistance of Ervin and Michael Leshner of Fuel Injection Development Corporation, the preliminary design configuration of Figure 4 was developed and given the designation "MK 1 Vaporized Fuel Injector" to distinguish it from subsequent designs. The fuel tank is pressurized from a high-pressure nitrogen bottle controlled by a pressure regulator. The needle valve is a Nupro precision needle valve used to control the flow metering of the fuel. The injector is installed on the 1-2-75 Heat Balanced Engine under concurrent development.

An assembly drawing illustrating the details of manufacture is included in Appendix D.

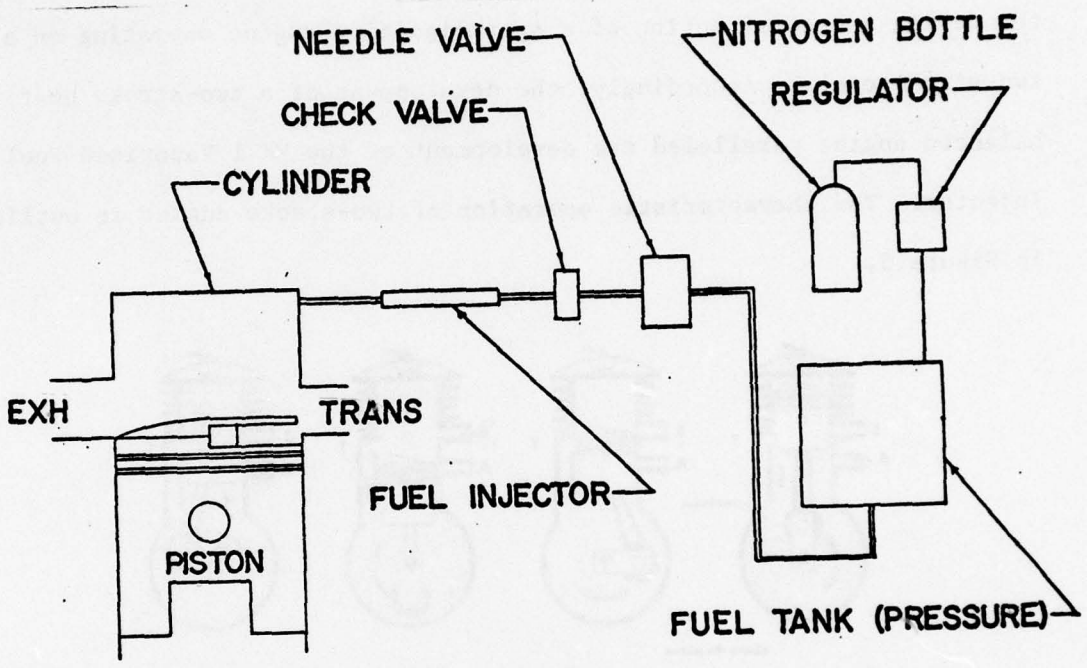


Figure 4. Preliminary Design

5.1.5. Model

A simple model for the MK 1 Vaporized Fuel Injection System was developed to establish basic guidelines for the selection of

initial operating conditions. This model is presented in Appendix B.

5.2. 1-2-75 HEAT BALANCED ENGINE

The selection of the MK 1 Vaporized Fuel Injector as the fuel delivery system to be used with the heat balanced engine necessitated the design and construction of a spark ignition engine operating on a two-stroke cycle. Accordingly, the development of a two-stroke heat balanced engine paralleled the development of the MK 1 Vaporized Fuel Injector. The characteristic operation of two-stroke engine is outlined in Figure 5.

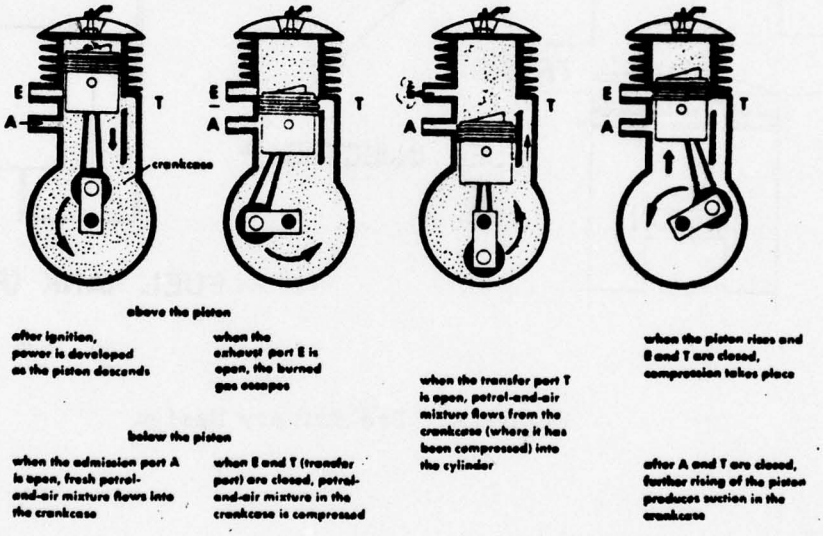


Figure 5. Crankcase Scavenged Two-Cycle Engine[3]

5.2.1. Suzuki TM 75 Engine

The Suzuki TM 75 motorbike engine was the engine chosen for conversion to a heat balanced configuration. The TM 75 is a small, late model two-cycle engine, with readily available repair

parts. Crank-case scavenged, the TM 75 possesses a rotary disc induction valve which fixes the crank angle during which air can flow into the crankcase. Specifications for the TM 75 are listed in Table 1.

Engine:	
Type	2 Cycle
Cylinder Arrangement	Single
Displacement	4.4 cu. in. (72 cc)
Bore x Stroke	1.85 x 1.65 in. (47 x 42 mm)
Compression Ratio	6.8:1 (theoretical) 4.8:1 (trapped)
Scavenging Pump	Crankcase
Induction Valve	Rotary disc
Port Timing:	
Transfer Closes	135° BTDC
Exhaust Closes	105° BTDC
Exhaust Opens	105° ATDC
Transfer Opens	135° ATDC
Lubrication System	Oil Inject
Ignition System	Flywheel Magneto

Table 1. Suzuki TM 75 Engine Specifications[4]

5.2.2. Conversion to Heat Balanced Configuration

With the assistance of Mr. Richard Blaser and Professor Andrew Pouring, preliminary considerations in the conversion of the TM 75 to a heat balanced configuration commenced. After the necessary volumes for both balancing chamber and combustion chamber were established, Dr. Pouring, drawing upon his experience from similar previous conversions, designed the piston cap shown in Figure 4 and detailed in Appendix D. An additional complication introduced by this two-cycle design was the maintenance of the proper port timing.

5.2.3. Nomenclature

The converted TM 75 was dubbed the 1-2-75 MK 1 Heat Balanced Engine (*1* cylinder, *2* stroke-cycle, *75* cc per cylinder). The 1-2-75 MK 1 HBE is the first two cycle, spark ignition, fuel injected heat balanced engine.

5.3. TEST FACILITY

A test facility was developed for the 1-2-75 HBE. The test facility is discussed in Appendix E.

5.4. TEST PROGRAM

A comprehensive test program for the 1-2-75 HBE is outlined in Appendix F.

6.0. CONCLUSIONS

A general approach to the design and selection of fuel delivery systems for internal combustion engines was developed, an operational fuel delivery system for the heat balanced engine manufactured, and a simple computer model formulated. In addition, an experimental facility was assembled and a detailed test program outlined.

The problem of fuel delivery is a four-dimensional one involving not only three-dimensional space but time as well. The characteristics of combustion influence heavily any fuel delivery system; combustion, an energy process, is highly time dependent. Any fuel delivery system must deliver to an engine the fuel it requires, both *where* and *when* it is required.

7.0. RECOMMENDATIONS

Further research into two-cycle engines is warranted. Its large specific output and the potentially more efficient and less polluting heat balanced configuration make the two-cycle engine an attractive possibility.

Better modeling of existing engines and fuel systems is necessary. The model developed by the author in the short time available is painfully inadequate.

Improved technological capability in the area of fuel delivery and control techniques is encouraged. The development, for example, of an electronic injector capable of operating at high engine speeds will expand tremendously our capabilities in the fuel injection field.

Finally, continued testing of the heat balanced engine is imperative. Conducting detailed tests is the only way that the requirements for the heat balanced engine's optimum performance can be determined.

FOOTNOTES

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APPENDIX A: THE HEAT BALANCED ENGINE

A.1. Introduction

A.2. Heat Balanced Engine Geometry

A.3. Combustion Characteristics

A.4. Conclusions

APPENDIX A: THE HEAT BALANCED ENGINE

A.1. INTRODUCTION

This section briefly discusses the general characteristics of the heat balanced engine (HBE) and its associated fuel requirements. More detailed information relating to the heat balanced engine is presented in several prior papers, references 14, 15, and 16.

A.2. HEAT BALANCED ENGINE GEOMETRY

The heat balanced engine at this stage in its development is a reciprocating piston engine whose combustion chamber is separated by a pressure exchange cap integral with the piston into two distinct volumes. The volume above the pressure exchange cap is termed the combustion chamber, while that below the pressure exchange cap is called the balancing chamber (see Figure A.1). Prior to combustion, the balancing chamber is charged with air while fuel is delivered to the combustion chamber.

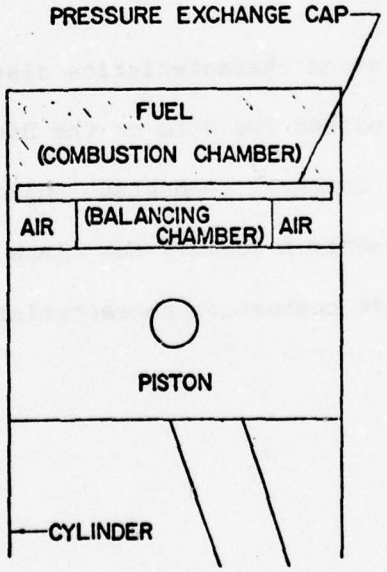


Figure A.1. Heat Balanced Engine Geometry

A.3. COMBUSTION CHARACTERISTICS

The pressure exchange cap atop the piston induces considerable turbulence in the upper cylinder during the compression stroke, producing a scrubbing effect which minimizes the "quenching" of the combustion reaction against the relatively cool cylinder walls. Additionally, the large surface area of the pressure exchange cap (hot from the previous cycle's combustion) preheats the air/fuel mixture and further reduces wall quenching, owing to its high surface temperature. Most important, the stratification of the air/fuel mixture between combustion chamber (fuel rich) and balancing chamber (fuel lean) allows for a locally rich mixture in the combustion chamber (well within the flammability limits of the air/fuel mixture) to be balanced by a large volume of air, the net effect of which is a lean mixture overall. Thus, a combustion reaction initiated in a sufficient concentration of fuel is able to continue to completion because it is provided with both the air and the time required for complete combustion.

A.4. CONCLUSIONS

The above combustion characteristics distinguish the heat balanced engine from either the Otto or the Diesel. Far from being a further refinement in internal combustion engine technology, the heat balanced engine represents a totally new class of internal combustion engine with distinctive combustion characteristics, operating conditions, and fuel requirements.

APPENDIX B: MODELING OF MK 1 VAPOR FUEL INJECTION SYSTEM

B.1. Summary

B.2. Background

B.2.1. MK 1 Vapor Fuel Injector

B.2.2. Scope

B.3. Model

B.3.1. Simplifying Assumptions

B.3.1.1. Fuel Line

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B.3.1.4. Cylinder

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B.3.3. Analog Simulation

B.4. Results

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B.6. Computer Programs

APPENDIX B: MODELING OF MK 1 VAPOR FUEL INJECTOR SYSTEM

B.1. SUMMARY

An analog computer model of the MK 1 Vapor Fuel Injection system installed on the 1-2-75 Heat Balanced Engine was developed using a MiniAC Analog/Hybrid Computer. As an order-of-magnitude approximation of actual fuel system performance, the model provides basic guidelines for the selection of initial operating conditions prior to engine testing. The model is idealized as describing fuel delivery system performance at wide open throttle, independent of load as a function of engine speed. The model is applicable only to the vapor fuel injector operating in conjunction with a crank case scavenged, two stroke cycle internal combustion engine.

B.2. BACKGROUND

B.2.1. MK 1 Vapor Fuel Injector

A schematic of the MK 1 Vapor Fuel Injection System is illustrated in Figure B.1. Liquid fuel flows under pressure from the fuel tank through the fuel line (I) to the check valve (II), where back-flow from cylinder to fuel tank is prevented by a ball-check arrangement. After being metered by the check valve into the fuel injector itself (III), the fuel is mixed and preheated by combustion gases as it flows into the cylinder (IV) when cylinder pressure drops below fuel injector pressure.

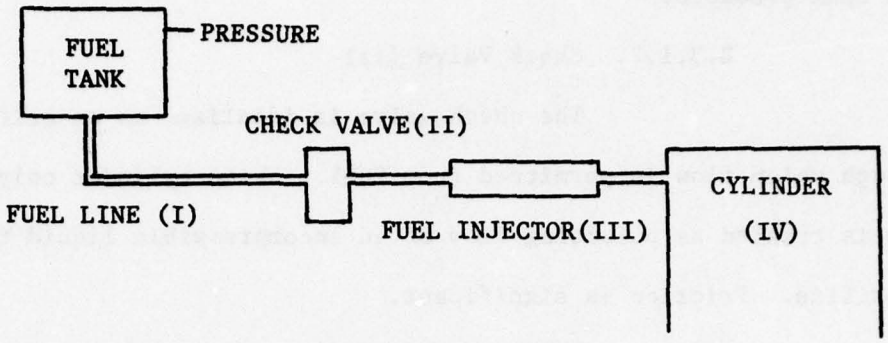


Figure B.1. Schematic - MK 1 Vapor Fuel Injector System

B.2.2. Scope

The flow mechanism of the vapor fuel injector depicted in Figure B.1 is complex at best. Compounding the problem of pulsating, unsteady flow are several factors such as friction, heat transfer, and two-phase, multi-dimensional flow. Additionally, flow through small diameter tubing and orifices may be choked. It is difficult, if not impossible, to develop a single model to describe this mechanism. Although they introduce inaccuracies to the model's solution, simplifying assumptions must be made in an effort to reduce the modeling problem to a more workable form.

B.3. MODEL

B.3.1. Simplifying Assumptions

The following assumptions refer to the schematic of Figure B.1.

B.3.1.1. Fuel Line (I)

Since mass flow rate of fuel and, thus, flow velocity are small, the flow of liquid fuel through the fuel line is

idealized as being incompressible and frictionless. The pressure of the fuel at the check valve is assumed to be constant, identical to fuel tank pressure.

B.3.1.2. Check Valve (II)

The check valve is idealized as an orifice through which flow is permitted from fuel tank to cylinder only. The flow is treated as pulsating flow of an incompressible liquid through an orifice. Friction is significant.

B.3.1.3. Fuel Injector (III)

The effects of heat transfer, friction, and two-phase flow are neglected due to the inordinate complications they introduce. The pressure throughout the fuel injector is assumed to be identical to cylinder pressure, and flow is considered to be unchoked. It is anticipated that considerable revision and refinement of the model will be required to correct the inaccuracies caused by these assumptions, which are in all likelihood over-simplifying.

B.3.1.4. Cylinder (IV)

From Figure B.2, cylinder pressure is relatively independent of engine speed in the vicinity ($\pm 90^{o}$) of bottom dead center (BDC). Cylinder pressure near bottom dead center is assumed to be independent of load and engine speed.

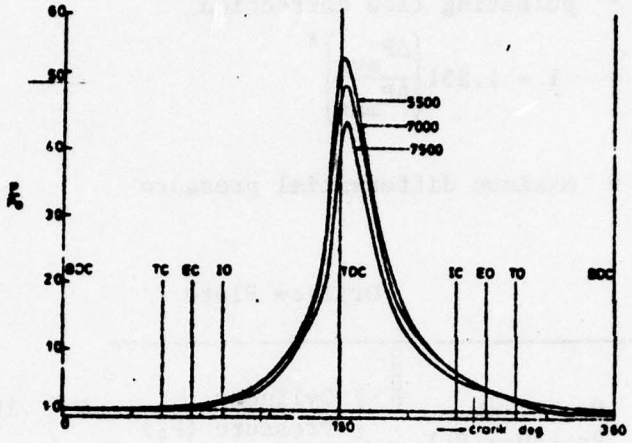


Figure B.2. Comparison of Experimental Cylinder Pressure for 380 cc Two-Cycle Engine at 5500, 7000, and 7500 rpm (From Blair and Cahoon[5])

B.3.2. Formulation

After making the assumptions of B.3.1., the modeling problem becomes one of pulsating flow of an incompressible liquid through an orifice (Figure B.3). Dr. Richard Bajura of West Virginia University[6] has determined that the mass flow rate of such a flow is given by the following relationship:

$$\dot{m} = C_D A Y \sqrt{2\rho \Delta P_{avg}} \qquad \text{(B.1)}$$

where $\dot{m}$ = mass flow rate

C_D = orifice discharge coefficient

A = orifice area

ρ = fluid density

ΔP = differential pressure, $P_1 - P_2$

P_1 = fuel tank pressure

P_2 = cylinder pressure

ΔP_{avg} = average differential pressure

Y = pulsating flow correction

$$= 1 - 1.551\left(\frac{\Delta P_{avg}}{\Delta P_{max}}\right)^2$$

ΔP_{max} = maximum differential pressure

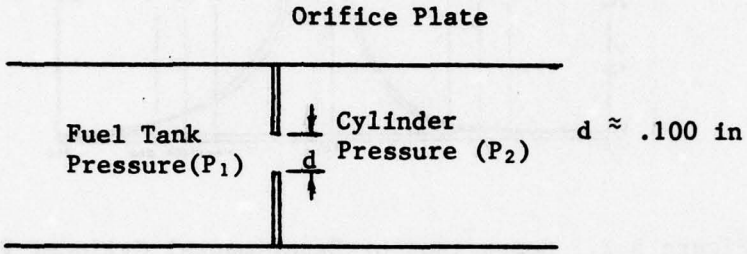


Figure B.3. Model

All that is required to be able to evaluate the mass flow rate is to integrate the differential pressure so that an average differential pressure may be calculated. The analog computer is ideally suited to this task.

B.3.3. Analog Simulation

Using a MiniAC Analog/Hybrid computer, the analog computer program of Figure B.4 was developed.

From Figure B.2, the approximate cylinder pressure of Figure B.5 was generated; since the compression ratios of the two engines are comparable (about 6:1), the pressures can be assumed as being the same. This pressure model was then normalized and input into an eleven point Variable Function Generator (VFG). This pressure was

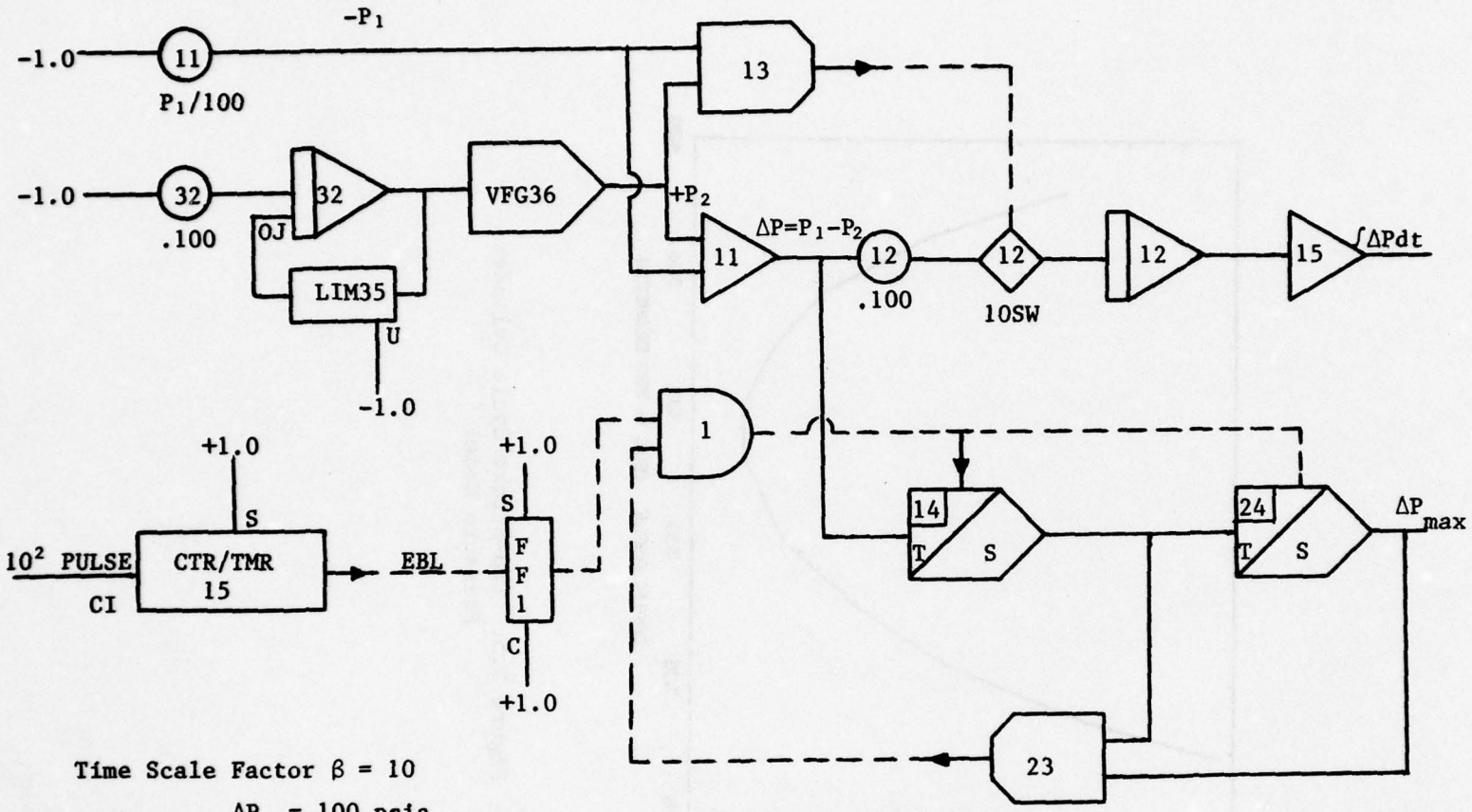


Figure B.4. Flow Diagram - Computer Model

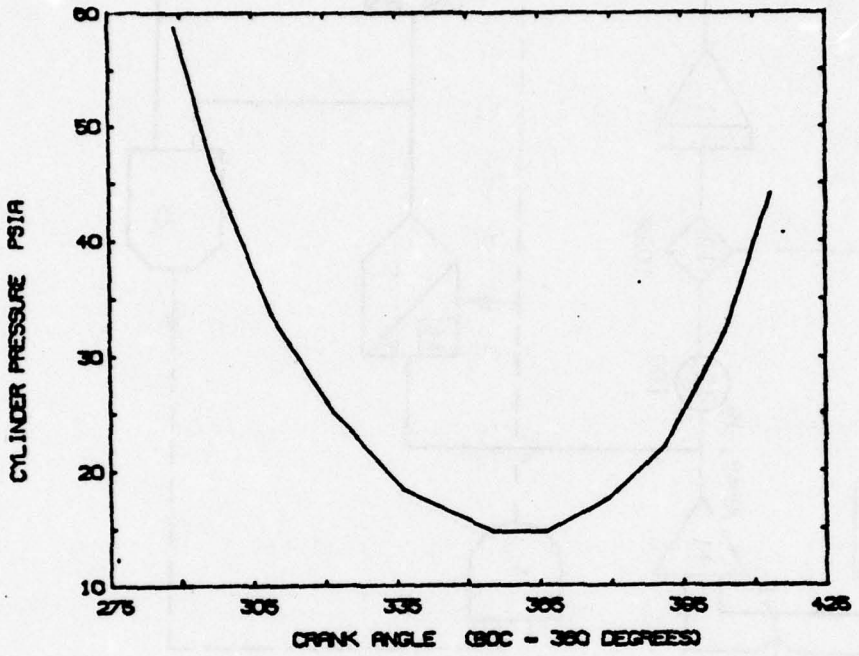


Figure B.5. Two-Stroke Cycle Cylinder Pressure Model

then subtracted from the fuel tank pressure to yield the differential pressure, ΔP.

A comparator (#13) senses the differential pressure, ΔP, and controls the integration of ΔP. When $\Delta P > 0$ (i.e., $P_1 > P_2$), fuel flows through the orifice and integration proceeds. When $\Delta P < 0$ (i.e., $P_1 < P_2$), fuel flow ceases and the comparator halts integration. At the conclusion of the cycle, $\int \Delta P dt$ is displayed and ΔP_{avg} can be determined according to the relationship

$$\Delta P_{avg} = \frac{1}{T} \int_0^T \Delta P dt \qquad (B.2)$$

where T is the computer time for 360° of crankshaft revolution (2.890 computer seconds).

Another comparator (#23) used in conjunction with two cascaded track/store summer-amplifiers and a flip-flop toggle controls the alternate storage and monitoring of the differential pressure so that the largest differential pressure, ΔP_{max}, is recorded.

B.4. RESULTS

B.4.1. Data Reduction

Using the MiniAC computer program outlined in B.3.3., the data of Table B.2 was generated. Since the computer requires one computer second to sweep the cylinder pressure model of Figure B.5, which also takes about 125° of crankshaft revolution, there are $360^\circ/125^\circ = 2.890$ computer seconds per crankshaft revolution, and

$$T = 2.890 \text{ computer seconds.}$$

Thus, ΔP_{avg} can be calculated from equation (B.2).

	x	y
0	.240	.588
1	.288	.463
2	.361	.338
3	.442	.250
4	.529	.184
5	.644	.147
6	.712	.147
7	.793	.176
8	.865	.221
9	.942	.323
10	1.000	.441

Table B.1. Cylinder Pressure - Variable Function Generator Set Up

Equation (B.1) yields both the mass flow rate of fuel and, by dividing by engine speed, the mass of fuel injected during each cycle. From equation (B.1):

$$\dot{m} = 129.280\ d^2 \sqrt{\left[1 - 1.551 \left(\frac{\Delta P_{avg}}{\Delta P_{max}}\right)^2\right] \Delta P_{avg}} \quad \frac{lbm}{min} \qquad (B.3)$$

where d = orifice diameter (inches)

using $C_D = 0.62$ and $\rho_{gasoline} = 43.8\ lbm/ft^3$ (from Taylor[7])

Further, from equation (B.3):

$$m = \frac{\dot{m}}{N} \frac{lbm}{cycle} \qquad (B.4)$$

where $\dot{m}$ = fuel per cycle (lbm)

N = engine speed (rpm)

P_1(psia)	$\int\Delta P dt$(psia-sec)	ΔP_{avg}(psia)	ΔP_{max}(psia)
14.7	0.010	0.003	0.8
20.0	1.210	0.419	6.2
22.5	2.110	0.730	8.6
25.0	3.180	1.100	11.2
27.5	4.370	1.512	13.7
30.0	5.630	1.948	16.1
32.5	7.000	2.422	18.6
35.0	8.480	2.934	21.1
37.5	10.040	3.474	23.7
40.0	11.700	4.048	26.1
42.5	13.440	4.650	28.6

Table B.2. Model Data

B.4.2. Data Plots

Figures B.6, B.7 and B.8 are plots of equation (B.4) for orifice diameters of 0.050 inches, 0.075 inches, and 0.100 inches, respectively, as engine speed varies from 1000 RPM to 7000 RPM. Figure B.9 is a plot of equation (B.3) for orifice diameters ranging from 0.025 inches to 0.150 inches.

Data points for Figures B.6, B.7, and B.8 were generated using the BASIC program "HBE1", while data for Figure B.9 was generated by "HBE2". All plots were made on a Tektronix 4051 computer terminal using "TEKGRAF2", an interactive graphics computer plotting package.

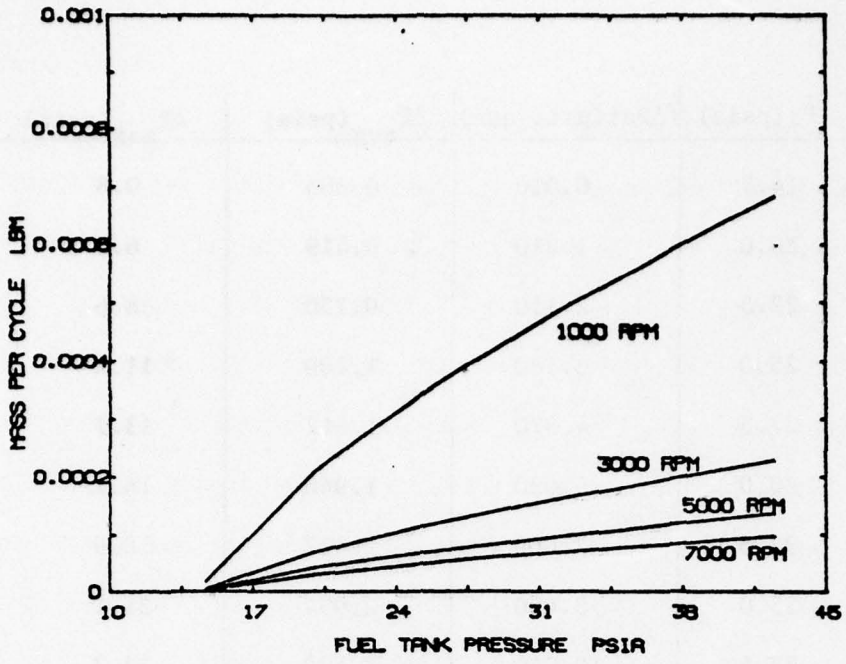


Figure B.6. Fuel per Cycle Orifice Diameter 0.050 inches

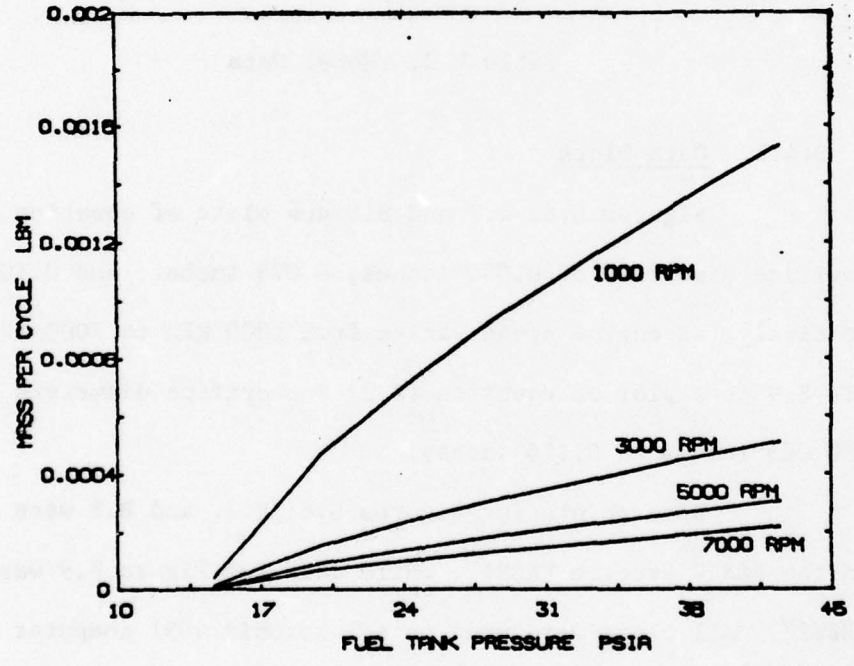


Figure B.7. Fuel per Cycle Orifice Diameter 0.075 inches

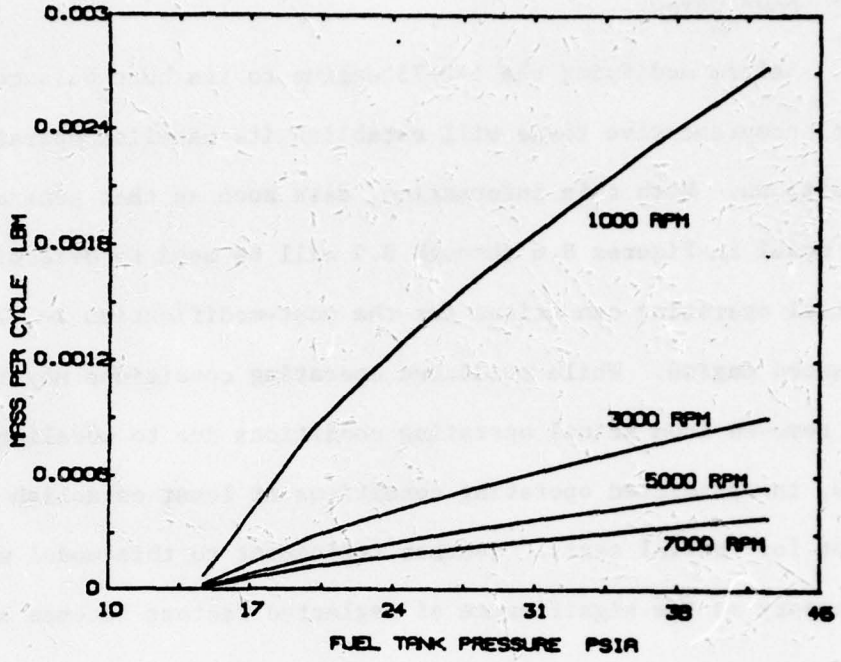


Figure B.8. Fuel per Cycle Orifice Diameter 0.100 inches

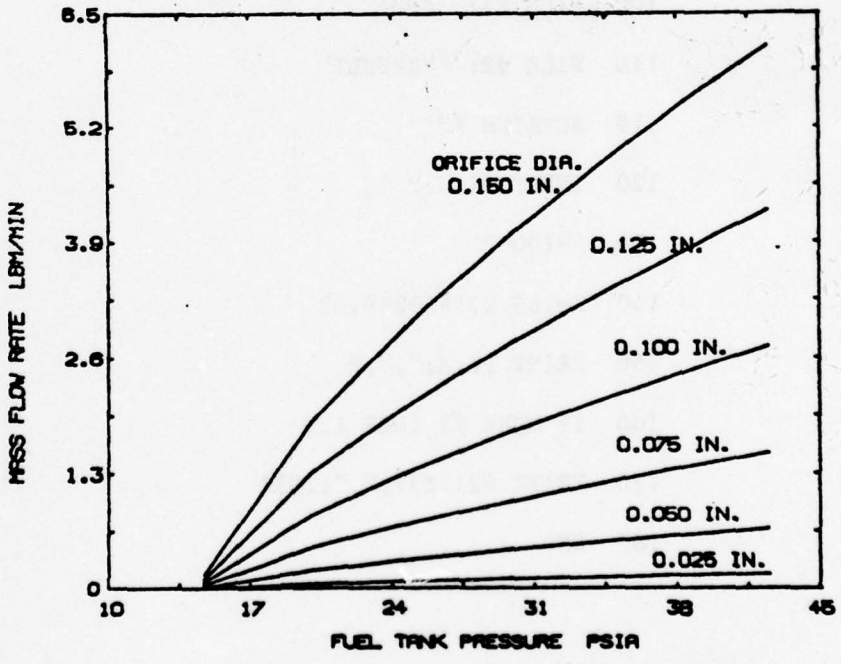


Figure B.9. Fuel Flow Rate for Different Orifice Sizes

B.5. CONCLUSIONS

Before modifying the 1-2-75 engine to its heat balanced configuration, comprehensive tests will establish its baseline operati ng conditions. With this information, data such as that generated by the model in Figures B.6 through B.9 will be used to determine the initial operating conditions for the post-modification 1-2-75 Heat Balanced Engine. While predicted operating conditions may in fact be far removed from actual operating conditions due to modeling inaccuracies, the predicted operating conditions at least establish a starting point for initial tests. Further refinement to this model will become necessary as the significance of neglected factors becomes known.

B.6. COMPUTER PROGRAMS

B.6.1. HBE - Generator Pressure Waveform of Figure B.5

```
100  FILE #1:  "PDATA"
110  FILE #2:  "RESULT"
115  SCRATCH #2
120  INPUT #1:X,P
130  P=100*P
140  X=163.977#X+249.63
150  PRINT #2:X;",";P
160  IF MORE #1 THEN 120
170  PRINT #2:1E37;",";1E37
180  END
```

B.6.2. HBE1 - Generator Fuel/Cycle Data of Figures B.6, B.7, and B.8

```
100  FILE #1:  "MDATA"
110  FILE #2:  "RESULT"
120  SCRATCH #2
130  PRINT "INPUT ORIFICE DIAMETER (INCHES)"
140  INPUT D
150  FOR N = 1000 TO 7000 STEP 2000
160     INPUT #1: P1,P2,P3
170     R=(1-1.551*(P2/P3)*(P2/P3))*P2
180     M=129.280*D*D/N*SQR(R)
190     PRINT #2:P1;",";M
200     IF MORE #1 THEN 160
210     PRINT #2:1E37;",";1E37
220     RESET #1
230  NEXT N
240  END
```

B.6.3. HBE2 - Generator Fuel Flow Rate Data of Figure B.9

```
100  FILE #1:"MDATA"
110  FILE #2:"RESULT"
120  SCRATCH #2
130  FOR D=0.025 to 0.150 STEP 0.025
160     INPUT #1:P1,P2,P3
170     R=(1-1.551*(P2/P3)*(P2/P3))*P2
180     M=129.280*D*D*SQR(R)
190     PRINT #2:P1;",";M
200     IF MORE #1 THEN 160
```

```
210     PRINT #2:1E37;",";1E37
220     RESET #1
230  NEXT D
240  END
```

APPENDIX C: MODERN FUEL SYSTEMS AND COMBUSTION SCIENCE

C.1. Introduction

C.2. Combustion Characteristics

C.2.1. Otto Combustion

C.2.2. Diesel Combustion

C.3. Fuel Systems

C.3.1. Otto (Spark Ignition) Engines

C.3.1.1. Carburetion

C.3.1.2. Fuel Injection

C.3.2. Diesel (Compression Ignition) Engine

C.3.2.1. Fuel Injection

C.4. Summary

APPENDIX C: MODERN FUEL SYSTEMS AND COMBUSTION SCIENCE

C.1. INTRODUCTION

Any fuel system used in conjunction with an internal combustion engine must deliver fuel to that engine in a manner determined by the combustion characteristics of the particular engine under consideration. Thus, the nature of the combustion occurring within the engine specifies which types of fuel systems may be used. Before discussing contemporary fuel systems, then, it is advantageous to first discuss the features of combustion in general.

C.2. COMBUSTION

Combustion is nothing more than a chemical reaction, in which fuel and air rapidly react at high temperature to produce heat and reaction products. The relative amounts of fuel and air present, as well as the efficiency and thoroughness of the combustion reaction determine how much heat is released and what combustion products, both desirable and undesirable, are produced. General characteristics of the combustion reaction dictate that neither too much nor too little fuel be present during the reaction and that the combustion be as turbulent as possible. Turbulent motion of combustion gases "scrubs" stagnant layers of reacting gases from the relatively cool walls of the combustion chamber prevents a local decrease in the rate of reaction. Specific combustion characteristics vary between two general categories of internal combustion engines: Otto (spark ignition) engines, and Diesel (compression ignition engines).

C.2.1. Otto Combustion

Otto combustion is characterized as a homogeneous combustion process in which air and fuel are thoroughly mixed prior to combustion. Combustion occurs very rapidly and is idealized as occurring at constant volume.

C.2.2. Diesel Combustion

Diesel combustion is characterized as a heterogeneous combustion process in which droplets of fuel react locally with air. Combustion occurs relatively slowly and is idealized as occurring at constant pressure.

C.3. FUEL SYSTEMS

C.3.1. Otto (Spark Ignition) Engine

Fuel systems as used in conjunction with Otto engines are geared to achieve as homogeneous a mixture of air and fuel as possible. This mixture is generally effected in one of two manners: carburetion or fuel injection.

C.3.1.1. Carburetion

A simple carburetor is diagrammed in Figure C.1. Air flowing through the venturi section (section 2) causes a decrease in pressure and draws fuel into the carburetor throat to the engine. Properly sized orifices meter the fuel so that an appropriate amount of fuel is drawn into the air stream.

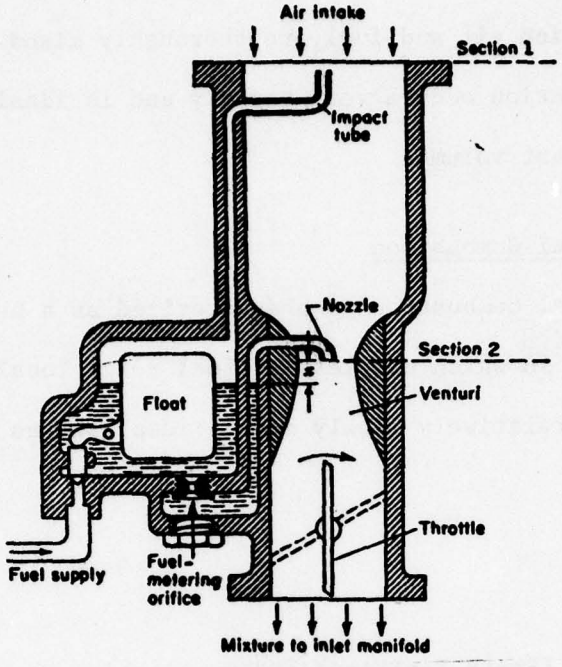


Figure C.1. Basic Open Passage Carburetor[8]

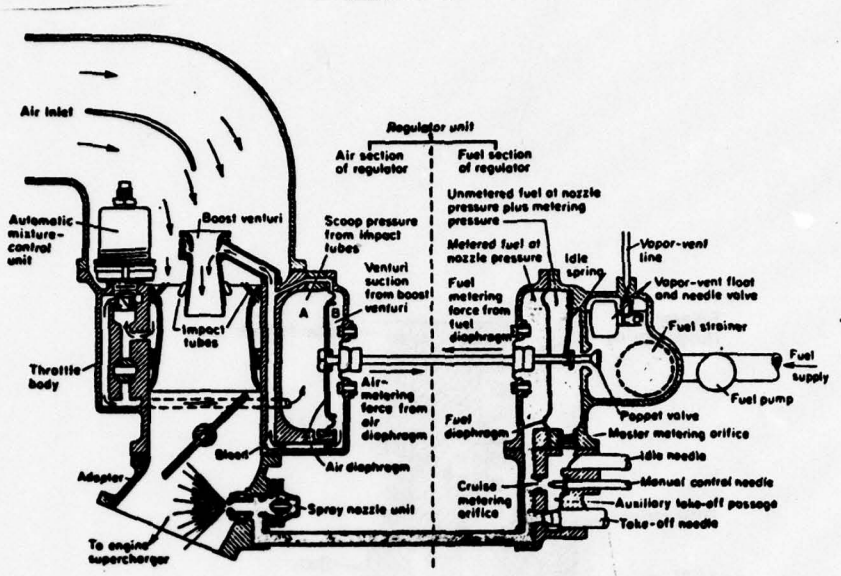


Figure C.2. Continuous Injection[9]

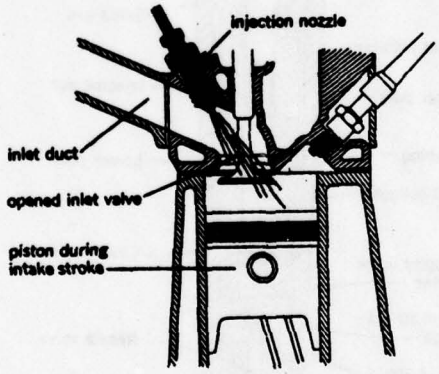


Figure C.3. Manifold Injection[10]

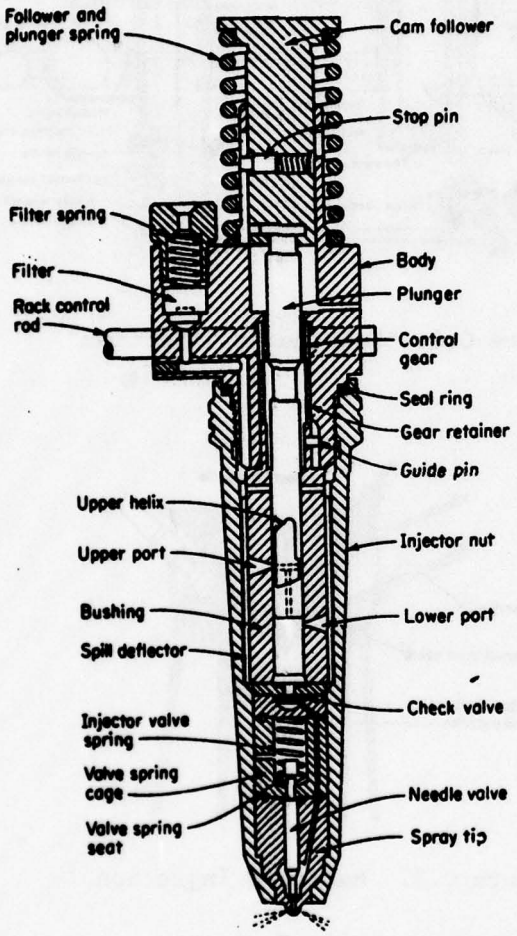


Figure C.4. Diesel Fuel Injector[11]

C.3.1.2. Fuel Injection

Fuel injection for Otto engines consists essentially of fuel forced under pressure to flow through spray nozzles located in the engine. Fuel injectors in Otto engines are either centrally located, continuous operation injectors such as that of Figure C.2, or locally positioned, controlled operation injectors such as that of Figure C.3. Very few, if any, Otto injectors inject directly into the cylinder owing to the difficulty of their attaining a homogeneous mixture.

C.3.2. Diesel (Compression Ignition) Engines

Fuel systems for Diesel engines must deliver fuel to the engine after the cylinder air charge has been compressed to a high temperature so that premature combustion does not occur. The pressures involved virtually dictate that direct injection to the cylinder be utilized.

C.3.2.1. Fuel Injection

Fuel injectors for Diesel engines resemble hypodermic syringes into which a precise quantity of fuel is metered and subsequently injected directly into the cylinder (Figures C.4 and C.5). Droplets of fuel injected by the injector provide the requisite heterogeneous mixture of the Diesel engine.

C.4. SUMMARY

From the preceding discussion, it is apparent that fuel systems for Otto and Diesel engines serve different, though similar, requirements and are not interchangeable despite the fact that both Otto and Diesel

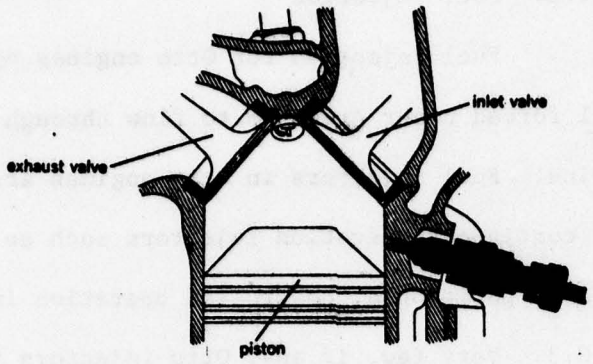


Figure C.5. Direct Cylinder Injection[12]

are internal combustion engines. Considering that the combustion characteristics of either engine are mutually incompatible, this is not a surprising conclusion. This suggests that *any fuel delivery system developed for an internal combustion engine must consider in detail the combustion characteristics of that engine.*

APPENDIX D: ASSEMBLY DRAWINGS

D.1. Piston Cap, 1-2-75 MK 1 Heat Balanced Engine

D.2. MK 1 Vaporized Fuel Injector

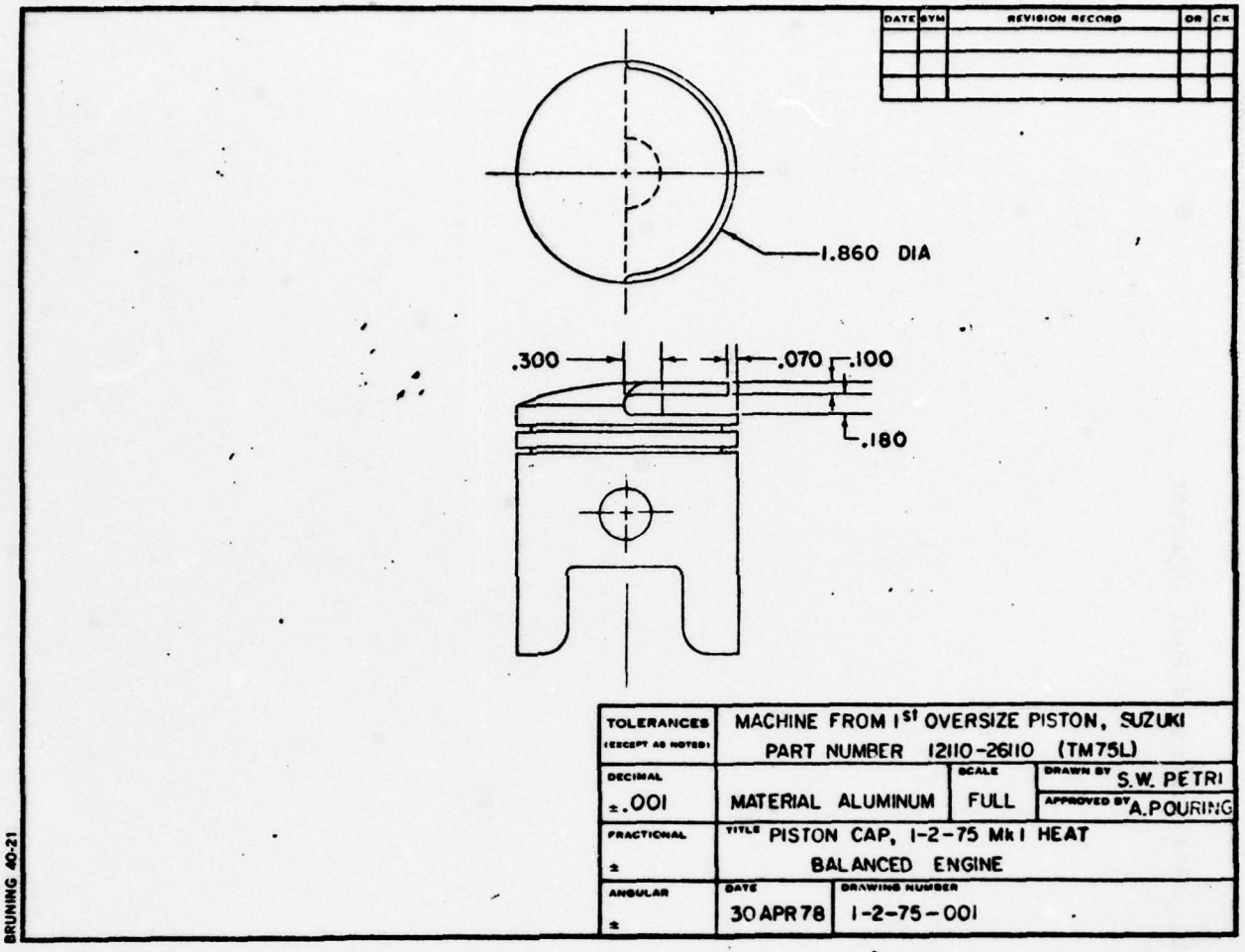


Figure D.1. Piston Cap, 1-2-75 MK 1 Heat Balanced Engine

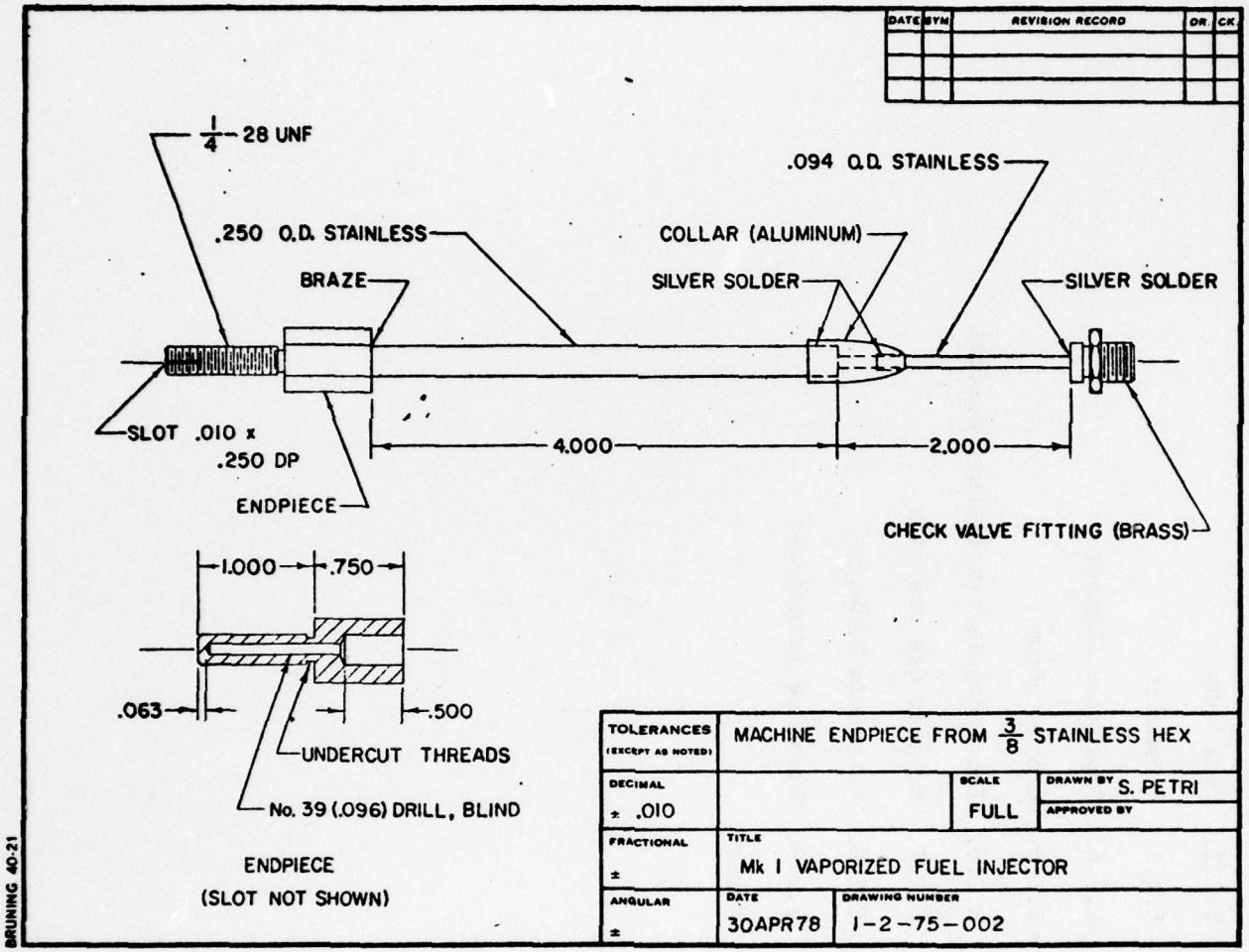


Figure D.2. MK 1 Vaporized Fuel Injector

APPENDIX E. EXPERIMENTAL APPARATUS

E.1. Introduction

E.2. Equipment

E.2.1. Megatech Dynamometer - Generator

E.2.2. Fuel Tank

E.2.3. Flowmeter

E.2.4. Expansion Chamber

E.2.5. Exhaust Drum

E.2.6. Thermocouple/Temperature Readout

E.2.7. Exhaust Gas Analyzer

E.2.8. Tektronix Package

E.2.9. Pressure Transducer

E.3. Future

APPENDIX E. EXPERIMENTAL APPARATUS

E.1. INTRODUCTION

The test facility as configured for initial tests is diagrammed in Figure E.1. Not shown is a precision mercury barometer for measuring atmospheric pressure and a sling psychrometer for determining specific humidity.

E.2. EQUIPMENT

E.2.1. Megatech Dynamometer-Generator

The 1-2-75 MK 1 Heat Balanced Engine is mounted on a test stand and loaded into a Megatech Model DG-8 Electric Dynamometer-Generator. The Dynamometer unit allows for the control of engine load so that a specified operating condition can be achieved. The dynamometer is gimbal mounted so that torque can be measured directly.

E.2.2. Fuel Tank

The fuel tank is a Sachs fuel tank of approximately one gallon capacity. It is pressurized through a pressure regulator from a high-pressure nitrogen bottle.

E.2.3. Flowmeter

The flowmeter is a Brooks E/C variable area flowmeter.

E.2.4. Expansion Chamber

A stainless-steel cylinder six inches long by five inches in diameter, the expansion chamber is roughly tuned to the test engine, the 1-2-75 HBE.

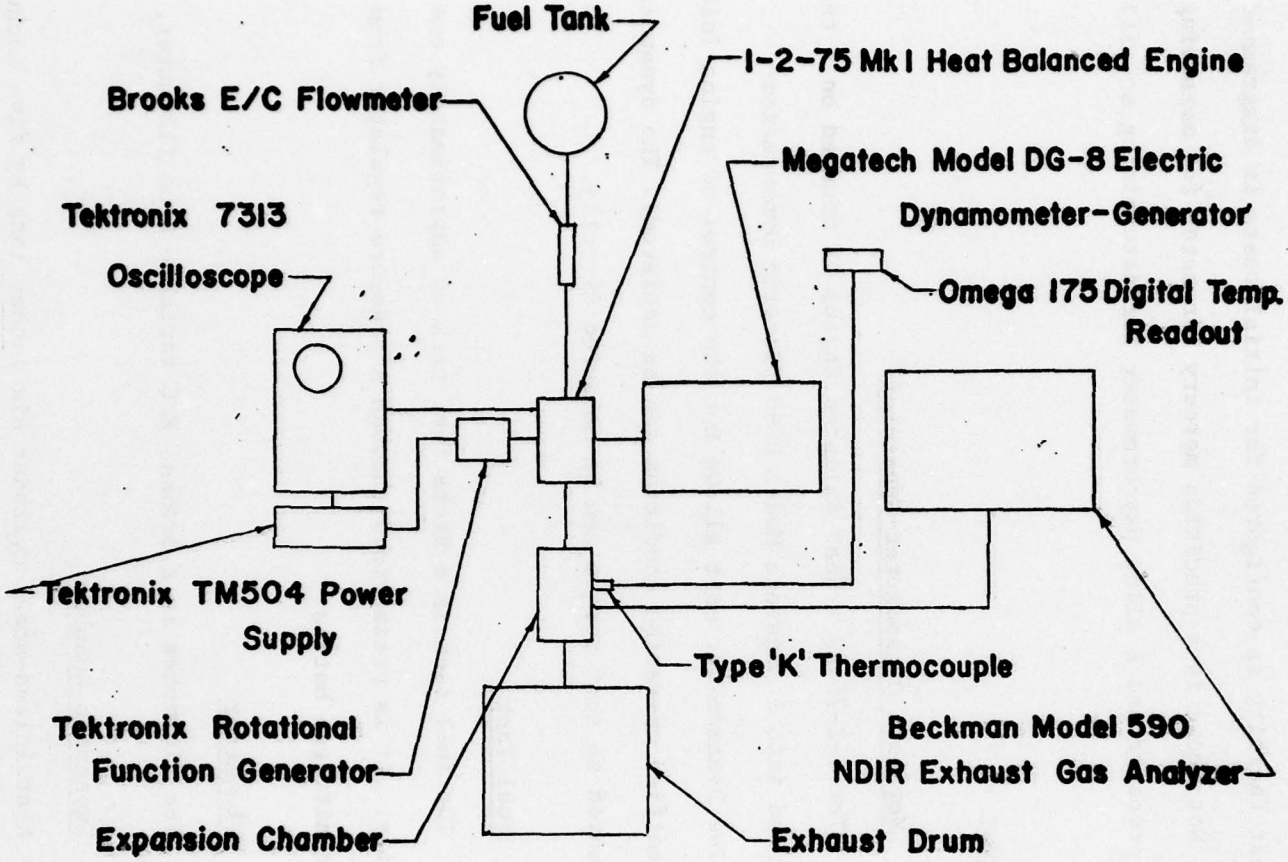


Figure E.1. Schematic of Test Facility for Initial Tests

E.2.5. Exhaust Drum

The exhaust drum is a large diameter drum to which the test engine exhausts at nearly atmospheric pressure.

E.2.6. Thermocouple/Temperature Readout

A type "K" Chromel-Alumel thermocouple measures the exhaust gas temperature, which is displayed on an Omega 175 Digital Temperature Readout device. The temperature is monitored primarily to prevent engine overheating.

E.2.7. Exhaust Gas Analyzer

The Beckman model 590 non-dispersive infra-red exhaust gas analyzer measures both carbon monoxide (CO) and unburned hydrocarbons (hexane equivalent) content of the test engine's exhaust gas.

E.2.8. Tektronix Package

Coupled directly to the test engine's crankshaft, the Tektronix Rotational Function Generator generates time, volume, and crank angle markers which are processed by the TM504 Power Supply for display on the 7313 oscilloscope. Ignition, vibration, and pressure displays may be selected and viewed versus either cylinder volume, crank angle, or time.

E.2.9. Pressure Transducer

Cylinder pressure is sensed by a PCB Piezotronics Model 111A24 piezoelectric pressure transducer.

E.3. FUTURE

Future plans call for the transfer of the test equipment to Melville Hall, where Mr. Richard Blaser's NAHBE Research Labs have more sophisticated exhaust gas analyzing equipment capable of measuring oxides of nitrogen. Also anticipated is the addition of an air box to the test equipment, allowing for the determination of air flow rate through the test engine.

APPENDIX F: TEST PROGRAM

F.1. Guidelines

F.2. Data

F.3. Program

F.3.1. Baseline Data

F.3.2. MK 1 Vaporized Fuel Injection System

F.3.3. Modification to Heat Balanced Engine

F.4. Goals

APPENDIX F. TEST PROGRAM

F.1. GUIDELINES

The scheduled test series on the 1-2-75 MK 1 Heat Balanced Engine will adhere to the guidelines established by the Small Spark Ignition Test Code, SAE J607a (reference 58).

F.2. DATA

The following data will, as a minimum, be determined:

Brake Horsepower (BHP)

Brake Mean Effective Pressure (BMEP)

Brake Specific Fuel Consumption (BSFC)

Brake Torque

Fuel per Cycle

Pollutants

Scavenging Ratio

F.3. PROGRAM

F.3.1. Baseline Data

Initially, a comprehensive test series will be run on the unmodified 1-2-75 to establish baseline data for the stock engine.

F.3.2. MK 1 Vaporized Fuel Injection System

After completing tests on the unmodified engine, the MK 1 Vapor Fuel Injector will be installed in the engine and a similar test series run. It is anticipated that at this point the optimum design parameters for the fuel injector will be determined.

F.3.3. Modification to Heat Balanced Engine

Finally, the Test engine will be converted to a heat balanced engine and a concluding test series run, building upon all previous tests. Alterations to the heat balanced engine configuration will be made in the search for the best piston geometry.

F.4. GOALS

It is anticipated that the feasibility and viability of the two-stroke cycle heat balanced engine will be demonstrated. A direct result of this work will be the submission of several papers to the Society of Automotive Engineers for publication.

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Accepted by the chairman of the Trident Scholar Committee.

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20. ABSTRACT *(Continue on reverse side if necessary and identify by block number)*

The Heat Balanced Engine (HBE) is a new class of internal combustion engines with unique fuel delivery requirements. In this study the characteristics and requirements of fuel metering and control techniques required for internal combustion engines in general are investigated. A general approach to the design and selection of fuel delivery systems for heat balanced engines has been developed. Using this approach, a particular

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fuel delivery system compatible with a specific two-stroke heat balanced engine has been selected, fabricated, and installed. An engine testing facility and simple model formulated for the engine have been established and a complete test program outlined.

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