NATURAL GAS/DIESEL DUAL FUELING OF A PRECHAMBER TYPE DIESEL ENGINE

by

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NOMENCLATURE

A/D	Analog to Digital Converter
Alagasco	Alabama Gas Company
ASME	American Society of Mechanical Engineers
BMEP	Brake Mean Effective Pressure (lb_f/in^2)
BSFC	Brake Specific Fuel Consumption (lb_m/BHP-hr)
BTU	British Thermal Units
CFR	Cooperative Fuels Research
CJC	Cold Junction Compensation
CNG	Compressed Natural Gas
CO	Carbon Monoxide
CO_2	Carbon Dioxide
EGR	Exhaust Gas Recirculation
EXP-16	MetraByte Model EXP-16 Multiplexer Expansion Board
GVW	Gross Vehicle Weight (lb_f)
HHV	Higher Heating Value (BTU/ft^3)
LHV	Lower Heating Value (BTU/lb_m)
LNG	Liquified Natural Gas
LPG	Liquified Petroleum Gas
NO_x	Nitric Oxides
PVT	Pressure, Volume, and Temperature
RPM	Engine Revolutions per Minute
SO_2	Sulfur Dioxide
STA-U	MetraByte Model STA-U Universal Screw Terminal Board
WOT	Wide Open Throttle
η_{th}	Thermal Efficiency
η_v	Volumetric Efficiency
ϕ	Equivalence Ratio

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CHAPTER I

INTRODUCTION

Background

The diesel engine, or the compression ignition engine, was invented by Dr. Rudolph Diesel, who patented the device in 1892. Diesel's patent covered the use of coal dust as a fuel, but also included other solid, liquid, or gaseous fuels. Some of the fuels tried during the initial development of this engine included coal-tar mixtures, blends of coal dust and oil, and petroleum-based products [1].

The popularity of the diesel engine spread quickly and its continued development has resulted in its use in a wide variety of applications ranging from automotive to marine propulsion. In 1898, Dr. Diesel patented an invention where the primary fuel charge was compressed to slightly below its self-ignition temperature and a charge of a fuel with a lower self-ignition temperature was introduced to initiate the combustion process. This patent covered what is now known as the dual-fuel engine [1].

The use of natural gas as a primary fuel in diesel engines is not a new concept. The first reported use of a "gas-diesel" engine in the United States was in 1926 [2]. Since that time, many researchers have investigated aspects of natural gas-diesel fuel combustion. During and immediately after the Arab Oil Embargos of the 1970s, the motivations for considering natural

gas were using a fuel which was domestically plentiful and to reduce foreign oil imports. These were and still are valid considerations. Joyce [3] discussed benefits, problems, and research and development needs for natural gas fueled vehicles and identified primary reasons to consider natural gas as a replacement for diesel fuel or gasoline to be availability and cleanliness. In 1988, the United States had reported proven reserves of natural gas totaling 795,600 billion cubic feet [4]. Bowen [5] states that, assuming current usage rates, there are proved world reserves of natural gas sufficient to last about 200 years. There are, however, only 1,953 billion barrels of petroleum estimated to be recoverable in the world [6]. Assuming current usage rates reported by Bowen [5], the recoverable reserves of petroleum would be sufficient to last about 85 years.

Statement of Problem

Another, and perhaps more pressing, reason for investigating dual-fuel engines is the increasingly stringent regulation of gaseous and particulate emissions. Natural gas is a clean burning fuel with a high octane rating and broad flammability limits which make it a reasonable candidate for many applications. Because natural gas has only one-half the density of air, it disperses into the atmosphere and thus has safety advantages. Because natural gas has the highest hydrogen content of any hydrocarbon fuel, its combustion also reduces production of CO_2, a principal " greenhouse " gas. The objective of this work was to investigate the performance characteristics of a single-cylinder, prechamber-type diesel engine using dual-fueling and to compare those results with those obtained using diesel fuel alone.

Organization of Thesis

In the next chapter, previous studies on natural gas fueled engines are reviewed. The major results and conclusions from these activities are also presented in Chapter II. Chapter III covers development of the data acquisition apparatus used in this research. This chapter includes development of the test stand, related instrumentation, and computer software used in this research. Chapter IV presents the results obtained from this research along with discussion of the results. Summary and conclusions are developed from the information obtained and are presented in Chapter V. Finally, recommendations for future work are presented in Chapter VI.

CHAPTER II

LITERATURE REVIEW

Experimental Efforts

Direct Injected Diesel Engines

Many researchers have studied different aspects of natural gas-diesel fuel combustion. Acker [7] investigated the performance and emissions characteristics of a dual-fueled Caterpillar 3406 turbocharged direct-injected diesel engine using natural gas as the primary fuel. In this work, the engine was tested in the laboratory using a hydraulic brake type dynamometer. Results showed the power and torque performance remained similar to the published baseline diesel values and CO emissions were higher when the engine was operated in dual-fuel mode as opposed to diesel fueling. Wong et al. [8] converted a Detroit Diesel Allison Model 12V-149T diesel engine to operate in dual-fuel mode, although a natural gas substitution rate higher than 70 % could not be achieved because of engine knock. It was also found that at very high natural gas substitution, this engine developed less than 85 % of its rated power at 1900 RPM. CO and NO_x emissions were reported to decrease when the engine was operated in dual-fuel mode.

Karim [9,10] and Karim and Amoozegar [11] also studied dual-fueling of diesel engines and noted some considerations for using natural gas in diesel engines. Karim and Amoozegar also introduced alcohols, gasoline, benzene, and n-hexane into the intake charge of the engine. They noted two problems

associated with the use of natural gas in dual-fuel diesel engines with relatively small pilot liquid injection. The first is low thermal efficiency when the engine is operated under part load conditions as a result of poor utilization of the gaseous fuel component. The second is the loss of combustion control involving spontaneous ignition of the complex (natural gas + diesel fuel + liquid fuel added) mixture within the cylinder. That is, the knock characteristics of the engine may be adversely affected. It was further reported that NO_x emissions tend to increase with the amount of pilot used.

Karim [10] found that methane is a clean burning fuel having superior exhaust emissions characteristics under most operating conditions. The gaseous nature of the fuel provides better transient operation and engine response together with improved fuel control. The gaseous fuel provides excellent mixing with air as it is introduced into the intake of the engine. The potential advantages of improved mixture control include improved engine performance characteristics. The homogeneous gas/air mixture inducted into the cylinder also provides more continuous energy release, which can improve engine operation. Other potential advantages noted included cleaner and longer-lasting lubricants, superior operation under both cold and warm weather conditions, less fouling of diesel fuel injectors, and longer valve life.

The use of alternate fuels in diesel engines entails many problems that require further research. A major problem with using gaseous fuels in dual-fuel engines is knock. Karim, Klat, and Moore [12] studied the phenomenon of knock in dual-fuel engines. It was determined that the ignition properties of the pilot fuel have relatively little effect in determining the onset of knock

compared with that of the primary fuel gas. It was also reported that as the pilot quantity increased the maximum rate of pressure rise rose sharply, whereas the increase in maximum cylinder pressure was only slightly affected. This assumed that ignition occurs in the envelope enclosed by the pilot fuel spray and then propagates to the rest of the charge. The larger the pilot fuel quantity, the greater will be the initial energy released and the sharper will be the maximum rate of pressure rise.

Lalk and Blacksmith [13] investigated the use of simulated lignite gas in a single-cylinder, direct-injected diesel engine. It was found at 2800 RPM that only 85 % of the baseline power was produced at a gas/air equivalence ratio of approximately 0.5. It was also noted that the engine would intermittently misfire and eventually stall under some operating conditions. Only 70 % of the baseline power could be produced at 3100 RPM at a gas/air equivalence ratio of 0.46 before the engine stalled. The exhaust appeared to be less dense at full load compared with baseline operation. The brake thermal efficiency for dual-fuel operation was 5% to 10% lower than that obtained for straight diesel fueling.

Tesarek [14] discussed the possible harmful side effects of exhaust emissions for diesel and dual-fueling. Natural gas dual-fueling was utilized to investigate exhaust emissions from a single-cylinder, direct-injected diesel engine as a function of speed and load. It was determined that with increasing natural gas utilization, a drastic reduction in soot emissions was achieved. The reduction in soot formation increases with greater load. Exhaust gas recirculation (EGR) and throttling the intake air were also inves-

tigated as possibilities to reduce soot emissions. EGR was found to lower the concentration of harmful exhaust components and influence the course of the combustion reactions, especially reducing the formation of nitric oxides. EGR decreased the amount of carbon monoxide and unburned hydrocarbon emissions and improved the BSFC as the percentage of exhaust gas recirculated was increased. Throttling the intake air produced similar results and had a greater influence with increasing natural gas substitution.

Mitchell and Whitehouse [15] modified a four-cylinder, direct-injected diesel engine to operate on dual-fuels and investigated variation of several parameters for their effect on engine performance. The parameters investigated included variation in pilot fuel quantity, the effect of restricting the intake air, and heating the intake air as a possible alternative to air restriction. It was found that the minimum practical pilot fuel quantity is limited by the characteristics of the fuel injection equipment. The engine was run approximately 1350 hours in dual-fuel mode to investigate the long-term effects upon the injection equipment. No obvious signs of deterioration were noted in the pilot injection equipment. One problem that was discovered as a result of this investigation was the presence of unburned gaseous fuel in the exhaust at light load operating conditions. Considerable improvement in gas consumption could be obtained if the intake air were restricted. Restricting the air flow also provided improvement in the governing characteristics of the engine, producing very little speed change when changing from dual-fuel to diesel operation or diesel to dual-fuel operation. Heating the intake air provided similar results to restricting the intake air, but was limited by detonation or knock.

Lowe and Brandham [16] converted three direct-injected diesel engines to run on a variety of fuels ranging from refinery gas to natural gas. The results gathered from testing these engines were used in the development of components for a dual-fuel diesel engine that would be used for gas compression or electrical power generation. Methods for controlling gas admission to the cylinder, the air-gas ratio, the pilot fuel injection or spark-ignition, and governing systems were under development. It was determined that some sort of air-gas ratio control was needed in order to minimize combustion efficiency losses as compared to diesel fuel operation. Using the same injection timing for diesel in dual-fuel mode resulted in lower maximum cylinder pressures because of the slower burning of the lean methane mixture as compared to diesel fuel.

Karim and Khanna [17] investigated the effect of intake air temperature on performance and exhaust emissions. It was reported that for a given engine speed, the mass of inducted air increases with decreasing temperature. It was also noted that the fueling rate required to maintain a stable speed and load increased rapidly with decreasing air temperature. It was reported that for a fixed compression ratio of 15:1, lowering the air temperature by 1° may lower the air temperature at the end of compression by about 3°. This temperature decrease could cause an increase in the amount of unburned hydrocarbons and CO present in the exhaust due to a reduced reaction rate. The production of NO_x, which is known to be exponentially dependent on temperature, could be decreased.

Indirect Injected Diesel Engines

Much of the prior work using natural gas in dual-fuel engines has been in direct-injected diesel engines. This type of engine is typically used in over-the-road trucks, stationary power generation, and other lower speed applications. Divided-chamber or prechamber-type engines have not received much attention until recently in regard to dual-fueling. Barbour and Lestz [18] studied the use of natural gas and propane as primary fuels in an Oldsmobile 4.3 l V-6 IDI diesel engine. At light loads, natural gas substitution rates in excess of 85 % were obtained, but performance deterioration and increased unburned hydrocarbons, carbon monoxide, and aldehyde emissions were found. At full load, the natural gas substitution rate decreased to 47 %, although similar performance and emissions were found.

Hirako and Ohta [19] evaluated the effect of introducing a portion of the diesel fuel charge for dual-fueling into the intake air stream and changing the shape of the precombustion chamber of the engine. It was determined that the maximum BMEP obtained with intake fuel addition was independent of the prechamber shape. It was also noted that the exhaust smoke was reduced by adding auxiliary fuel, making it possible for the engine to operate at a lower excess air ratio. A study of the components and concentrations of CO, CO_2, and O_2 was also performed using a gas chromatograph. It was determined that intake fuel addition produced essentially no change in the percentage of CO_2 and O_2 and a decrease in the percentage of CO in the exhaust gas for the same amount of diesel fuel injected. It was also reported that improved thermal efficiency occurred at heavy loads.

Ding and Hill [20] studied the emissions and fuel economy of a Caterpillar 3304 prechamber-type turbocharged diesel engine. It was found that the gas/air equivalence ratio was the key variable affecting both emissions and fuel economy. For low loads, the thermal efficiency decreased with increasing utilization of natural gas and at higher loads the thermal efficiency exceeded that for straight diesel operation. The emissions from this engine in dual-fuel operation were strongly dependent on load and diesel fuel proportion. CO emissions at light loads increased with the amount of natural gas used. At higher loads, the CO emissions decreased, but were higher than for 100 % diesel fueling. NO_x emissions decreased at lighter loads as a result of the decrease in the overall cylinder gas temperature as the load was decreased. At heavier loads, the addition of natural gas caused NO_x emissions to increase beyond those obtained for straight diesel fueling.

Gasoline Engines

The following represents a summary of some of the work done with natural gas fueled spark-ignited engines. Wong [21] investigated the relationship between engine performance and fuel quality using a single-cylinder four-stroke engine. The maximum brake power was found to decrease with the quality of the fuel used. The use of a gas having a composition of approximately 65 % methane and 35 % carbon dioxide by volume was found to decrease brake power up to 22 %. The range of satisfactory operating speeds for this engine became narrower mainly in the low speed range when this gas was used. The brake specific fuel consumption, on the other hand, was improved up to 30 % compared to gasoline.

Eghbali [22] operated a cooperative fuels research (CFR) engine on natural gas and compared its performance with gasoline. Several parameters such as spark timing, heating the intake air, and variation in the compression ratio were investigated to determine their role in natural gas combustion. Ignition timing was found to be of primary importance due to the slow laminar flame characteristics of methane as compared to iso-octane. An optimal spark advance of 10° higher than that for gasoline was recommended. Reduced power output was also found due to the slower burning speed of methane as well as lower volumetric efficiencies. Higher intake air temperatures were found to cause knock to occur at lower compression ratios and with a wider range of air/fuel ratios.

Evans, Goharian, and Hill [23] investigated fueling a four-cylinder gasoline engine with natural gas. Variations in the air/fuel ratio were examined for several speeds at wide open throttle (WOT) as well as inducting the natural gas through a natural gas carburetor and baseline tests were conducted with gasoline. The latter two series of tests were conducted to survey the effects of converting a passenger car engine to use methane without major engine modifications. Brake power from this engine was reduced and the BSFC increased when using natural gas. It was reported that an automotive engine can be operated using natural gas more easily with lean mixtures because of the wider flammability limits of natural gas as compared to gasoline.

Theoretical Efforts

Computer Simulation

Mathur, Babu, and Prasad [24] developed a thermodynamic computer simulation of an open-chamber, dual-fuel diesel engine. This simulation included submodels of the compression, combustion, expansion, exhaust, and intake processes. Output from this model was compared to experimental data gathered from a single-cylinder, direct-injected diesel engine operating in diesel and dual-fuel modes. General agreement was found between the computer model and experimental results.

Fundamental Investigations

In order to understand the theoretical aspects of natural gas combustion, one must comprehend the different processes which occur during combustion. Reviews of the combustion processes in the dual-fuel engine were proposed by Karim [25,26]. Two of the main objectives of understanding the combustion processes are to enhance the efficiency of conventional systems and to find effective means to curb the problem of air pollution. Karim [25,26] also stated that normal dual-fuel operation with natural gas at light load may be improved by extending the lower operational limit to lower natural gas concentrations in the intake air charge. Some ways suggested to improve light load dual-fuel performance and emissions included:

1. The use of a larger pilot fuel quantity.
2. The use of the lowest possible nozzle opening pressure.
3. The slight preheating of the intake gas-air mixture.
4. Partial restriction of the intake air.

Karim and Singh [27] investigated numerically the partial oxidation of methane for a temperature range from 1200 K to 2300 K and a pressure range from 1 atmosphere to 100 atmospheres. It was found that the maximum temperature rise normally corresponds to a mixture richer than stoichiometric, which becomes slightly richer with an increase depending upon the extent of preheating. Increasing the pressure generally produces, at a decreasing rate, an increase in the temperature rise following combustion. At high initial temperatures and increasing pressures, this can be very significant because increased pressure allows the recombination of atoms into molecules, which is an exothermic process.

Karim and Khare [28] investigated the laminar flame propagation and quenching characteristics of methane-air mixtures as a function of the inlet mixture temperature. It was found that the maximum value for the laminar burning velocity, defined as the relative velocity perpendicular to the flame front with which the unburned fuel moves, decreases non-linearily as the reactant temperature is decreased. The peak value of the burning velocity shifts in a linear fashion to a more fuel-rich equivalence ratio as the inlet temperature decreases. The minimum quench distance, defined as the largest spacing between two parallel plates which a flame will not propagate through, was found to increase as the inlet reactant temperature was decreased and occurred at a slightly fuel-rich mixture.

Karim, Khan, and Moore [29] utilized pressure-time records to determine the effective rate of heat release from a motored engine. In the process of developing this heat release model, several assumptions were made including:

1. The cylinder charge consists of a single homogeneous zone with no effective temperature or pressure gradients.
2. The extent of conversion of reactants to products was proportional to the energy released by combustion.
3. Ideal gas behavior.
4. The use of a heat transfer correlation first proposed by Annand.

Elliott and Davis [30] investigated dual-fuel combustion in diesel engines in relation to the hazards of operating diesel engines in flammable gas/air mixtures. It was determined that the lower flammability limit of the combustible gas in the compressed gas/air mixture is an important factor to be considered in dual-fuel combustion. If the concentration of the combustible gas is greater than the lower flammability limit, the flame will propagate throughout the gas/air mixture at a high speed. If this occurs and the system is not capable of converting this energy for useful purposes, it is dissipated in thermal, frictional, and vibrational effects, with accompanying knock. Peak pressures within the cylinder were determined to be greater when the engine was operated in dual-fuel mode.

Karim [31] noted that one of the most interesting features displayed during dual-fuel combustion is the very restricted range of air/fuel ratios over which power can be produced efficiently before the onset of knock. The range of air/fuel ratios can vary considerably depending on the gaseous fuel used and operating conditions. The presence of the gaseous fuel was found to influence both the pre-ignition and post-ignition processes in a complex manner. Karim [31] suggests that dual-fuel combustion may be considered

to have two distinct phases. The first phase is associated mainly with the consumption of the pilot fuel and some of the gaseous fuel while the second phase is associated with the burning of the bulk of the charge.

Concerns

Safety Concerns

Karim [32] investigated the safety considerations of compressed natural gas (CNG) as an automotive fuel. It was determined that the following characteristics need to be considered when evaluating alternative fuels:

1. Storage and portability characteristics.
2. The tendency to form a combustible mixture following accidental discharge.
3. Ignition, explosion, and flame spread characteristics.
4. Suppression of fuel combustion characteristics.
5. Environmental consequences of their use and discharge.

The storage and portability characteristics of methane were investigated by Karim [32] and by Joyce [3]. CNG is usually carried in vehicles in high-pressure cylindrical containers. Some alternative fuels, such as liquified petroleum gas (LPG), or propane, are liquid as stored in cylindrical containers on board automobiles. Karim [32] stated that if a tank containing propane were engulfed in a fire, its heating would result in a rapid increase in pressure within the fuel tank. This may require the addition of pressure relief valves to vent the buildup in pressure. Because CNG is a gaseous fuel, it may not require relief valves, and if a CNG tank were engulfed in

flames, its heating would increase the pressure in accordance with the gas laws which are almost directly related to the average temperature. Karim [32] also found that the design of modern cylinders is such that cracks would likely develop releasing CNG more slowly than if rapid fragmentation of the fuel tank had taken place.

The tendency of CNG to form a combustible mixture following an accidental discharge is decreased as compared to propane. Theoretically, a CNG cylinder leaking for any reason into the environment can form a certain maximum volume of combustible mixture. Karim [32] also reported that the release of a unit volume of natural gas into air would produce a maximum combustible mixture that is approximately 40 % that of propane.

The ignition, explosion, and flame spread characteristics of methane are well known. However, the corresponding characteristics for CNG are dependent on the particular gas composition. Karim [32] found that methane combustion characteristics that make it a unique fuel as far as safety is concerned due to the compact chemical structure and its inherent chemical stability. It takes a minimum of over 5 % by volume of methane into air at ambient conditions for flame propagation to occur as compared to approximately 2 % for propane.

Karim [32] also noted several environmental considerations involving CNG applications. Because of the relative small capacity of CNG tanks in automotive applications, a CNG fire would be of short duration. Also, the smoke production will be decreased in methane fires since smoke produc-

tion in flames is largely a function of the carbon content of the fuel. Finally, Karim [32] notes that there is no direct medical evidence suggesting that methane is a toxic fuel.

Vehicle Concerns

The problems with the utilization of natural gas in dual-fuel engines include limited vehicle range, fueling station reliability, and performance. The vehicle range is limited because even when the natural gas is compressed to a relatively high pressure in the vehicle fuel tanks, its density is much less than that for liquid fuels. Joyce [3] indicated that a standard high-pressure CNG tank that would easily fit into the trunk of a mid-size automobile weighs about 130 pounds and holds the equivalent of three gallons of gasoline.

The question of where to refill CNG tanks is a major obstacle to public acceptance of natural gas as an alternate fuel for gasoline or diesel engines. The natural gas would have to be compressed from a few hundred lb_f/in^2, or less, to several thousand lb_f/in^2 pressure. Compression energy can become a significant cost element particularly if the natural gas was taken from a low pressure distribution system. If CNG is to be sold in a similar manner as gasoline or diesel fuel, it must also be accurately measured at the pump.

Wyman [33] reported on the state of natural gas refueling in Canada. Until recently a commercial meter for CNG similar to that used to dispense gasoline or diesel fuel did not exist. It was reported that service station attendants in Italy and New Zealand relied on "fill charts" to estimate the amount of natural gas dispensed. These "fill charts" were based on pressure,

volume, and temperature (PVT) relationships. An accurate mass flow meter for dispensing natural gas was developed in Canada which is accurate to better than 1 %. Wyman [33] concluded that although the ideal gas law can be used to accurately calculate properties at high temperatures and low pressures, it is inaccurate for the high pressures found in CNG installations. When CNG is added to an empty or near empty cylinder during a quick fill, the gas temperature will initially drop and then rise until a temperature difference of approximately 20°F to 40°F above ambient exists. This is due to the non-ideal behavior of the natural gas under the pressures typically involved with CNG. The pressure within the CNG cylinder was found to decrease about 200 lb_f/in^2 to 400 lb_f/in^2 after the gas temperature had equalized.

Natural gas is currently used in two forms: compressed natural gas (CNG), where the natural gas is in a gaseous state, and liquified natural gas (LNG), where the gas is in a liquid state. Constable, Gibson, and Gram [34] investigated the advantages of using LNG in heavy-duty vehicles. The major advantage of utilizing LNG is that it is approximately 600 times denser than the gaseous form of natural gas at a pressure of 1 atmosphere. This makes LNG an economical fuel to store and transport over long distances. It was found that the CNG equivalent of approximately 65 gallons (250 liters) of diesel fuel requires a volume nine times that of diesel and almost five times more weight. In an over-the-road application, this factor becomes very significant. The amount of CNG required to replace 65 gallons of diesel fuel would weigh about 2500 lbs (1150 kg) including tankage as compared to approximately 550 lbs (250 kg) for diesel fuel. Given a maximum gross vehicle

weight (GVW) of 80,000 lbs, an average mileage of 500 miles per day, and diesel fuel usage of about 130 gallons per day, the utilization of CNG would entail a decrease of 5 % by weight in the amount of freight which may be transported. This is of economic concern to trucking companies.

On the other hand, the amount of LNG required to replace 65 gallons of diesel fuel would weigh approximately 640 lbs (290 kg), which is about 90 lbs (40 kg) more than diesel fuel. The utilization of LNG to replace diesel fuel would entail a loss of only 0.2 % in the amount of freight by weight which could be transported given the same mileage range and GVW as discussed for CNG.

Case histories of several fleets of automobiles converted from gasoline to natural gas were compiled by Lafrenz, Glaub, and Jones [35]. Among the operational difficulties encountered in methane-fueled fleets are loss of power, difficulty starting in cold weather, problems with the refueling systems, and improperly trained mechanics. Fleet operators were generally satisfied with natural gas fueled vehicles and the problems listed were not considered to be overly serious nor impossible to overcome.

In this chapter, previous studies on natural gas fueled engines were reviewed. While a great deal of knowledge about dual-fueled engines exists, this work extends the operating expertise of dual-fueled engines to a small, prechamber-type diesel engine. This work examines the thermal and emissions performance of the engine using dual-fuels as compared to diesel fuel alone.

CHAPTER III

DEVELOPMENT OF EXPERIMENTAL FACILITIES

This chapter presents the development of the facilities and the equipment utilized in this work. The items discussed include:

- Development of the test stand and related equipment,
- Development of the data acquisition system, and
- A description of the gas-phase emissions equipment used in this work.

Development of Test Stand and Related Equipment

A test stand was developed to allow the use of natural gas as an alternate fuel in a single-cylinder, prechamber-type, diesel engine. A schematic of the experimental setup is shown in Figure III.1 and a schematic of the instrumentation system is shown in Figure III.2. This stand was unique in that it was designed to be portable. Therefore, the stand had to be self-contained. The items contained on the test stand included the dynamometer, inlet air system, exhaust system, and the water system. Related equipment not included on the test stand included the fueling systems for natural gas and diesel fuel and the data acquisition system. The following paragraphs describe the major systems comprising the test stand.

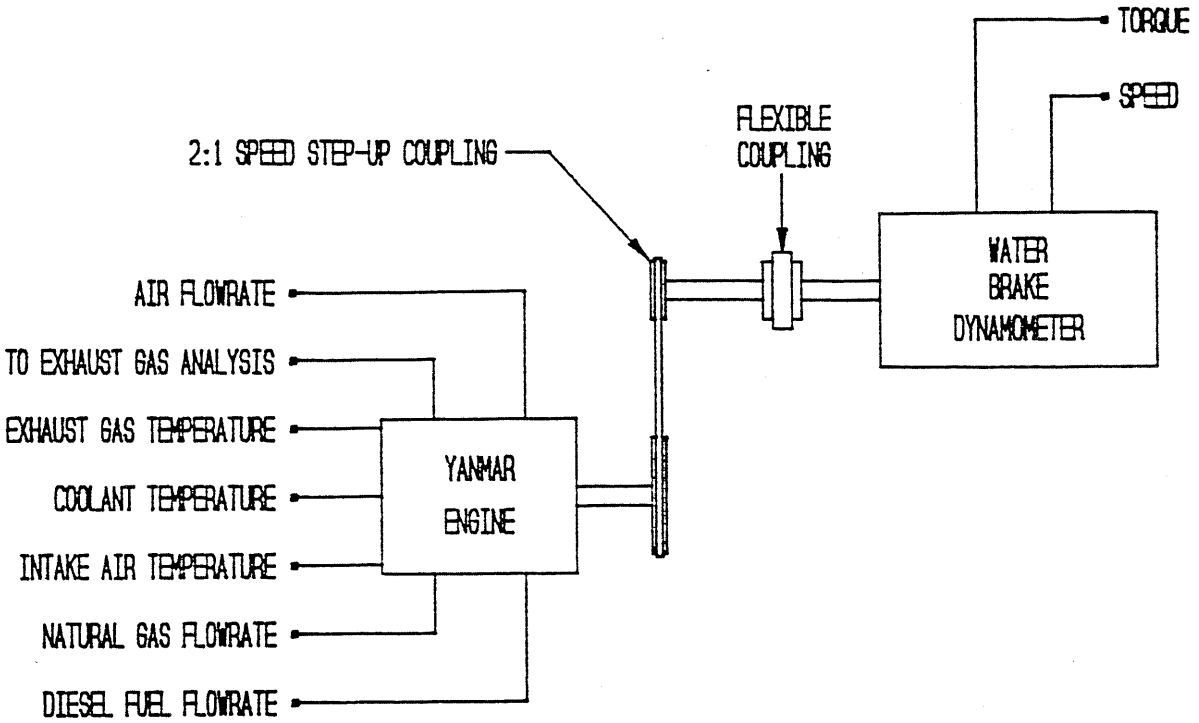

Figure III.1 - Schematic of experimental setup.

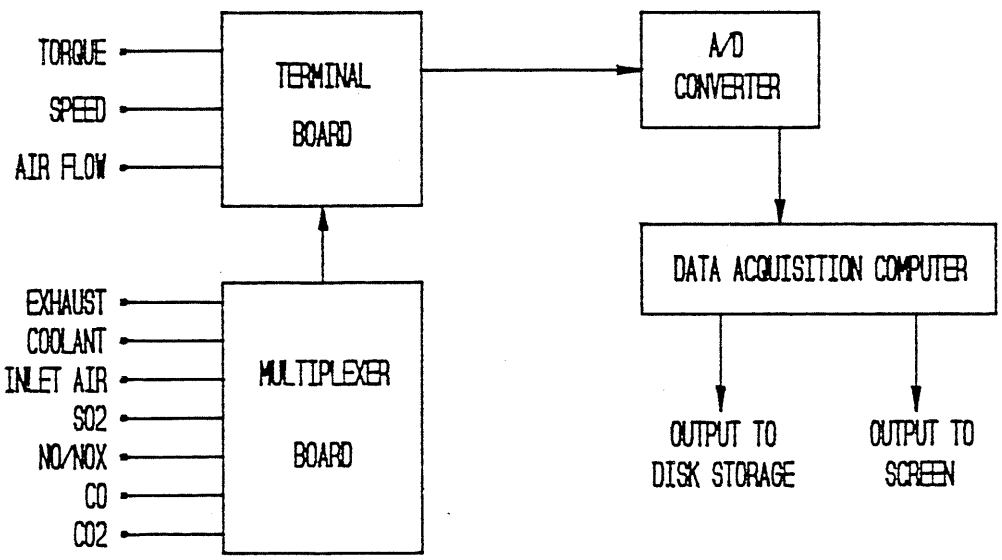

Figure III.2 - Schematic of instrumentation system.

Dynamometer

The engine was coupled to a Go-Power Model D-100 waterbrake dynamometer utilizing a 2:1 speed step up coupling. The speed step up coupling was used to optimize the torque absorption characteristics of the dynamometer. The torque absorbed by the dynamometer was measured using a strain gage load cell and the dynamometer speed was determined using a 60 tooth gear with an inductive sensor. The torque and speed signals were sent to a Go-Power Model C-17 Dynamometer Display. The C-17 was modified to allow analog outputs for the torque absorbed and dynamometer speed to be accessed by a PC-based data acquisition system.

Intake Air System

The intake air system conditions and provides monitoring of the flow rate of air entering the engine. This system consists of a 10 gallon polyethylene tank utilized as a surge drum which dampened out pressure pulsations. The rate of air flow inducted into the engine was determined using an 0.75 inch ASME long radius nozzle. This nozzle was designed such that the pressure difference between the atmosphere and the inside of the surge drum is proportional to the amount of air inducted into the engine. The pressure drop across the ASME nozzle was measured using an Omega Model PX-163 differential pressure transducer. This transducer has a useful range of $\pm$ 5 inches of water, which was adequate for this work.

Exhaust System

The exhaust system utilized in this work provided access to the exhaust by means of 5 sample ports that allow for the exhaust gas temperature to be measured using a thermocouple, exhaust samples taken in order to determine emissions concentrations present in the exhaust gas stream, and conditioning of the exhaust prior to discharge from the test stand. Two exhaust systems, a "wet" exhaust system and a "dry" exhaust system, were designed to handle the discharge from the test stand. A tee with flanged couplings was mounted at the discharge from the test stand that diverted exhaust flow from one exhaust system to the other.

The "wet" exhaust system was utilized during the initial portion of this work when only the performance parameters of the engine were under investigation. The "wet" exhaust system was designed to allow for the portability of the test stand and consisted of a 2 inch diameter marine exhaust hose connected to the tee located at the discharge from the test stand at one end and to the existing exhaust header. In order to utilize the marine exhaust hose, some provision had to be made to cool the exhaust stream. Therefore, a system for the injection of water into the exhaust stream was devised. The exhaust water injection system consisted of a valve to regulate the amount of water being injected, a nozzle that was installed into one of the sample ports that sprayed water into the exhaust stream, and flexible tubing that connected the two.

The "dry" exhaust system was utilized when particulate (smoke) measurements were taken and consisted of steel pipe connecting to the tee located

at the discharge from the test stand and to the existing exhaust header. The reason for the addition of the second exhaust system was that the exhaust water injection system was not used during this phase of the work so that particulates would be less likely to condense to the interior walls of the exhaust line, and the presence of water in the particulate measuring device or sampling valve could damage those components.

Water System

The test stand had a piping system that provided water to the dynamometer to load the engine and to the exhaust water injection system. The piping system consisted of 0.625 inch diameter flexible hose equipped with quick-connecting couplings to allow water to be inducted from existing water lines and 0.5 inch diameter copper tubing for the distribution of water to the dynamometer and exhaust water injection system, and valves to control the amount of water going to the dynamometer and the amount of water utilized in the exhaust water injection system. After the outlet from the dynamometer, there was a gravity drain to discharge water from the test stand into the drain.

Fuel System

The diesel fueling system consisted of two graduated burets. The main fuel supply for the engine consisted of a graduated buret with a capacity of 1000 ml. The fuel supply that was used whenever diesel fueling rates were

desired consisted of a graduated buret with a capacity of 100 ml. The volumetric flow rate of diesel fuel was determined using a stopwatch to measure the time required to consume a specified amount of fuel.

The amount of natural gas aspirated into the intake air stream was determined using an Omega Model FL-224 rotameter. This rotameter allowed direct flow readings in l/min for air and water. Correlations supplied by the manufacturer allowed conversion of the rotameter readings for air to volumetric flow rates for natural gas. During dual-fuel testing, fueling rates for natural gas and diesel fuel were obtained enabling the total injected fuel energy to be determined. This allowed the natural gas substitution rate, thermal efficiency (η_{th}), and brake specific fuel consumption (BSFC) to be determined.

The correlations for the natural gas flow rates based on rotameter readings accounted for the differences in the densities and viscosities between air and natural gas. The chemical composition and the higher heating values (HHV) on a dry and wet basis of the natural gas supplied by the local gas utility, Alagasco, were determined by an independent testing laboratory and the results are summarized in Table III.1. The results obtained enabled a lower heating value (LHV) and the air/fuel ratio for the natural gas to be determined. The LHV determined for the natural gas used was 20,600 BTU/lb_m and the air/fuel ratio was 16.74.

One of the objectives of this research was to perform as few modifications to the engine as necessary to allow the use of natural gas in the engine. A minor modification to the diesel fuel governing system was necessary to allow

Table III.1: Chemical Composition and Analysis of Natural Gas

Component	Formula	Volumetric Analysis (%)
Hexanes +	C_6 +	0.13
Propane	C_3H_8	0.57
Iso-Butane	i-C_4H_{10}	0.11
Normal Butane	n-C_4H_{10}	0.15
Iso-Pentane	i-C_5H_{12}	0.05
Normal Pentane	n-C_5H_{12}	0.06
Carbon Dioxide	CO_2	0.28
Ethane	C_2H_6	3.29
Oxygen	O_2	< 0.01
Nitrogen	N_2	0.96
Methane	CH_4	94.39
Carbon Monoxide	CO	< 0.01

Compressibility Factor Z = 0.99785
Specific Gravity (Air = 1.0) Ideal = 0.5910
HHV @ 60°F and 14.73 psia Dry Gross Ideal = 1047.0 BTU/ft^3

Specific Gravity (Air = 1.0) Saturated = 0.5921
HHV Saturated = 1031.0 BTU/ft^3

extended use of the engine in dual-fuel mode. The governor of this engine consisted of two rotating counterweights attached to the rack adjustment on the fuel injection pump by means of a lever. The position of the lever was determined by the balancing of the centrifugal forces due to the governor counterweights and the spring force from the regulator spring, which the engine operator set at a desired position. If the regulator spring was set at a low engine speed and the engine speed increased for some reason, the purpose of the governor was to reduce the amount of fuel injected into the engine in an attempt to keep the engine from overspeeding. A governor bracket, consisting of two tie rod ends connected by 0.1875 inch diameter all-thread material, was fabricated to "lock" the rack adjustment lever in place. This arrangement is shown in Figure III.3. The desired idle speed could be set by adjusting a bolt, which pivots the entire assembly about an existing stud located on the engine block, enabling the amount of diesel fuel injected to be increased or decreased depending upon the rotation direction of the bolt. Once the desired idle speed was achieved, the bracket was "locked" in place by means of a jam nut.

This arrangement had numerous advantages, chiefly safety. If for some reason the supply of natural gas was interrupted, the engine speed would decrease with decreasing amounts of natural gas used and default to the idle speed which was set before dual-fueling began. Another advantage was that no major modification of engine components was necessary to accomplish the desired task of operating the engine on natural gas as the primary fuel.

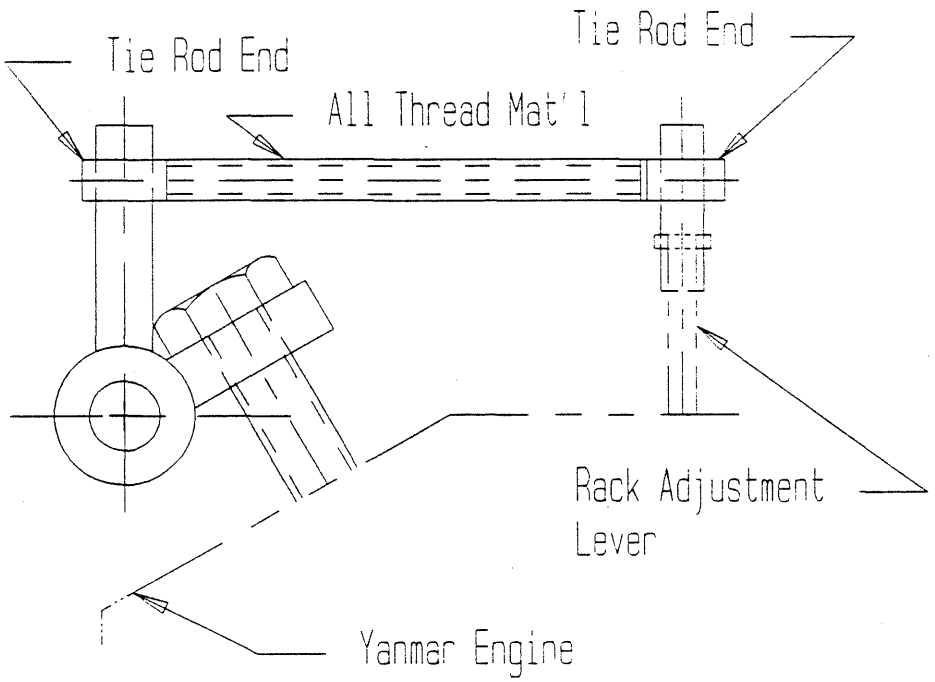

Figure III.3 - Governor bracket arrangement.

Development of the Data Acquisition System

The instrumentation and data acquisition system were built around an IBM XT compatible computer. A high speed analog to digital converter (A/D) was used for data acquisition. The A/D interface board used in this work is a MetraByte Model DAS-16F. The data acquisition system utilized a 12 bit A/D converter and had a throughput of up to 100,000 conversions/sec. Temperatures of the exhaust gas, cooling water, and intake air were measured using a MetraByte Model EXP-16 multiplexer board. This board allows up to 16 differential inputs to be multiplexed as a single input on the DAS-16F. The EXP-16 board has provisions for cold junction compensation (CJC). Outputs from the gas-phase emissions analyzers were also connected to the EXP-16. A description of the emissions equipment utilized will be presented later in this chapter. Outputs for torque and speed from the Go-Power Model C-17 Dynamometer Display and air flow from the Omega Model PX-163 differential pressure transducer were connected to a MetraByte Model STA-U screw terminal board which provides input to the DAS-16F.

The data acquisition system was accessed using a program, YAN2.BAS, written in the QuickBasic programming language. The purpose of this program was to determine parameters such as the speed of the engine, the torque and power absorbed by the dynamometer, and temperatures for the intake air, cooling water, and exhaust. A listing of YAN2.BAS may be found in Appendix A. A second program, FUEL1.BAS, was also written. The purpose of this program was to determine parameters such as the brake specific fuel consumption (BSFC), thermal efficiency (η_{th}), and the equivalence ratio

(ϕ) from the data files produced by the data acquisition program. A listing of FUEL1.BAS is shown in Appendix B.

The software program allowed samples to be taken at regular intervals and includes a screen driver program, which allowed current data to be monitored during testing. Whenever samples were taken, 50 samples of each parameter (torque, speed, etc.) were averaged to produce the value output from the A/D converter. Output data was automatically written to a data file during testing to allow the data to be analyzed at a later time. A detailed description of the data acquisition system and instrumentation may be found in Appendix C.

Description of the Gas-Phase Emissions Equipment

A summary of the gas-phase emissions equipment utilized in this work is shown in Table III.2 and a schematic of the emissions equipment is found in Figure III.4. Samples extracted from the exhaust line to be analyzed for NO/NO_x, SO_2, CO, and CO_2 concentrations were passed through an unheated stainless steel and Teflon line to a sample gas conditioning system where they were dried and filtered and then sent to the analyzers.

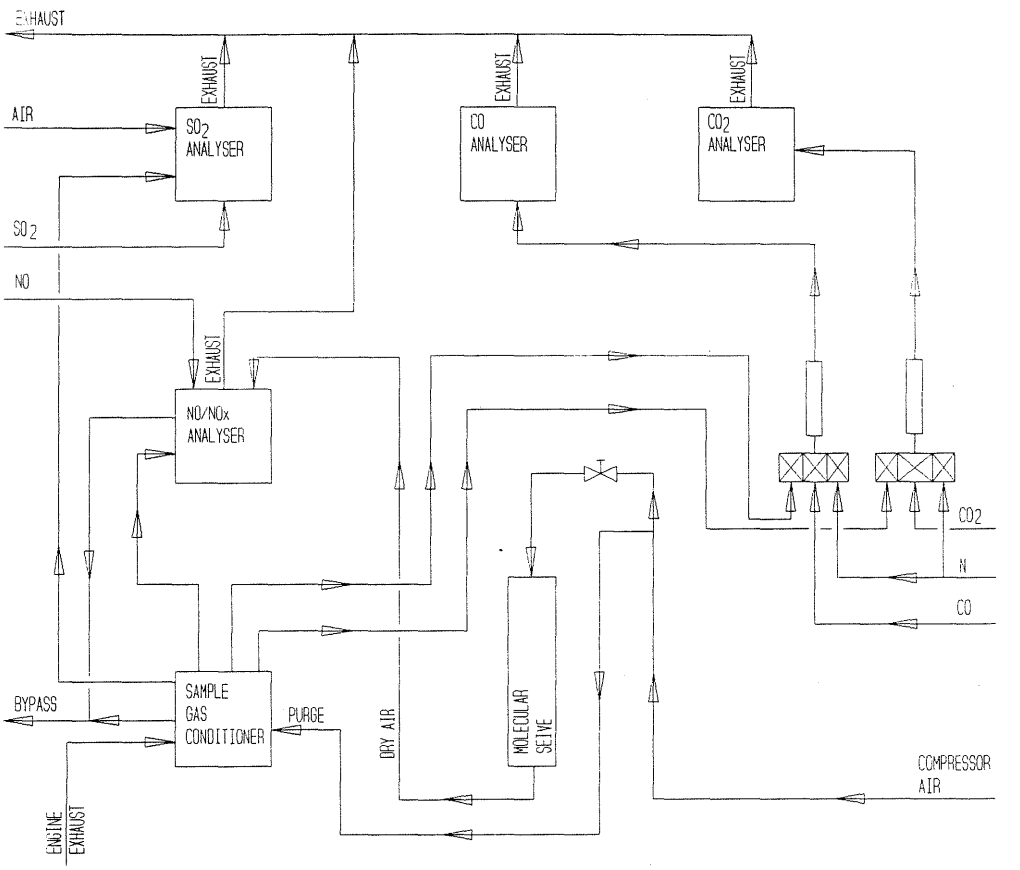

Figure III.4 - Schematic of gas-phase emissions equipment.

Table III.2: Gas-Phase Emissions Equipment Description

Analyzed Gas	*Instrument*	*Description*
CO	Infrared Model 703D	Infrared Analyzer
CO_2	Infrared Model 703	Infrared Analyzer
NO_x	Thermo Electron Model 44	Chemiluminescent Analyzer
SO_2	Thermo Electron Model 40	Pulsed Fluorescent Analyzer

Smoke measurements were taken using a Bosch sampling pump. This device takes a sample of constant volume of the exhaust gas and passes it through filter paper which traps the particulates. The particulate-laden filter paper was analyzed using a Bosch Model EFAW 68A smoke meter. A detailed description of the gas-phase emissions equipment is shown in Appendix D.

In this chapter, a development of the experimental setup was presented. In the next chapter, the results obtained and discussion of those results are presented.

CHAPTER IV

RESULTS AND DISCUSSION

The objective of this work was to investigate the performance characteristics of a single-cylinder, prechamber-type diesel engine using dual-fueling and compare these results with those obtained using diesel fuel alone. The engine used in this work was a Yanmar Model TS180C 4-cycle, horizontal diesel engine. The engine was operated in "as received" condition with only minor modifications to allow for the use of natural gas as the primary fuel. The engine is of the prechamber-type and specifications of this engine are shown in Table IV.1 below. The engine was mounted to the test stand using rubberized engine mounts. This type of mounting was adopted to isolate the test stand from the vibration of the engine.

Table IV.1: Specifications of Test Engine

Bore x Stroke	4.02 in (102 mm) x 4.17 in (106 mm)
Displacement	52.9 in^3 (0.866 l)
Rated Continuous Output	15 HP @ 2200 RPM
Rated 1-Hour Output	18 HP @ 2400 RPM
Specific Fuel Consumption	0.45 lb_m/BHP-hr (205 g/BHP-hr)
Compression Ratio	19.5:1

The figures discussed below represent raw data which was taken using the data acquisition system. Using an interpolating routine, polynomial curves were fitted to the data. The dual-fuel cases shown on these figures represent all points with a specified percentage of natural gas fueling. In

some of the dual-fuel cases, it was not possible to gather data over the entire range of engine speeds subject to baseline testing. The curve fits for each natural gas fueling percentage are shown over the rpm range for which data was taken. Most of the figures shown present only the curve fits in the interest of clarity. For each load shown, several tests were conducted to insure adequate repeatability of the experiments.

The engine was operated at speeds ranging from 1000 RPM to 2000 RPM, in increments of 100 RPM, at full load conditions based on performance data provided by the manufacturer. The range of speeds noted was repeated for three-quarter, one-half, and one-quarter load conditions.

Performance Results

Torque

The torque absorbed by the dynamometer as a function of engine speed and load is shown in Figure IV.1 for the engine being operated on diesel fuel. The figure shows for full load that torque increases from 9.2 ft-lb_f at 1000 RPM to about 16.5 ft-lb_f at about 1600 RPM and remains essentially constant from 1600 RPM to 2000 RPM. This figure shows every 50^{th} data point from the entire data set and curves fit to the data. One of the objectives of this work was to maintain the baseline torque when operating the engine in dual-fuel mode and thus similar torque-speed characteristics were found for dual-fueling.

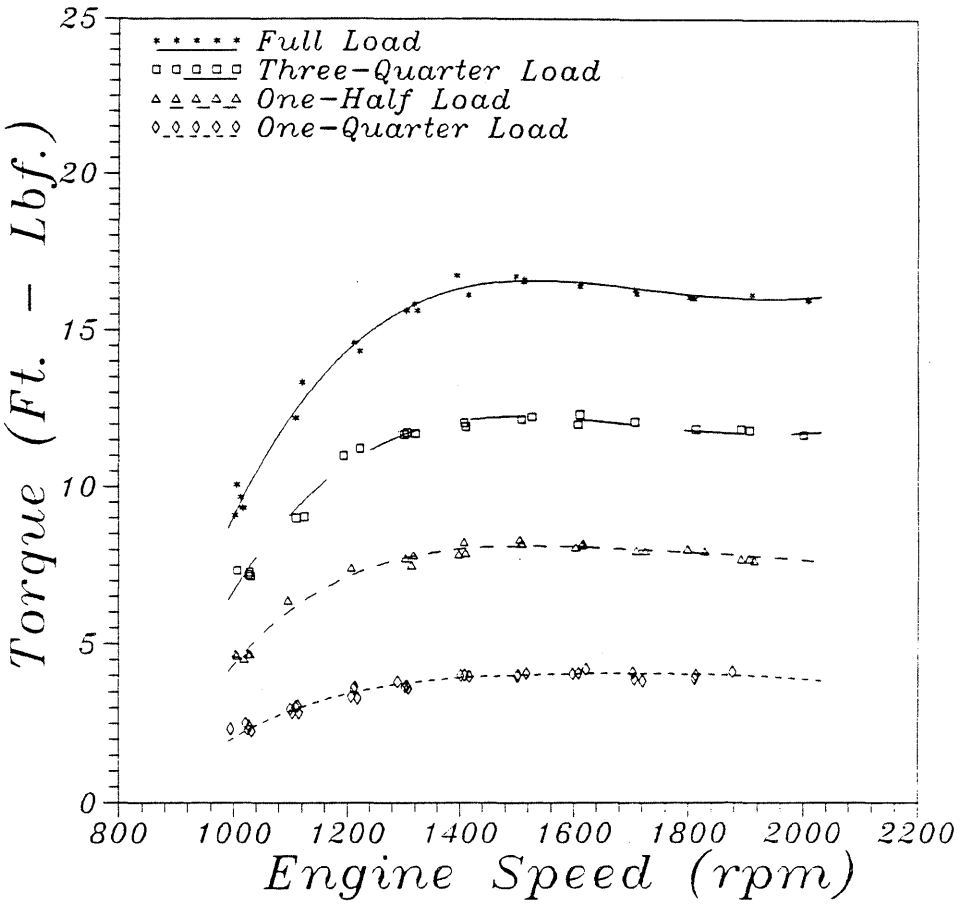

Figure IV.1 - Torque as a function of engine speed and load.

Power

The power absorbed by the dynamometer as a function of engine speed and load is shown in Figure IV.2. Again, this data was taken for 100 % diesel fueling and provided the baseline data for setting the dual-fuel operating conditions. The range of power absorbed at full load ranged from about 3.6 HP at 1000 RPM and increased in a near linear fashion to 12 HP at 2000 RPM. Trends for three-quarter, one-half, and one-quarter load were of a similar nature.

Brake Specific Fuel Consumption and Thermal Efficiency

Figure IV.3 shows the brake specific fuel consumption (BSFC) and the thermal efficiency (η_{th}) as a function of engine speed at one-quarter load for 100 % and 37 % diesel fueling. The percentage of diesel fueling was determined by the percentage of the total injected fuel energy supplied by the diesel fuel. The BSFC for 100 % diesel fueling decreased from 1.1 lb_m/BHP-hr at 1000 RPM to a minimum value of 0.75 lb_m/BHP-hr at about 1575 RPM and increased to approximately 0.9 lb_m/BHP-hr at 1900 RPM. The BSFC for 37 % diesel fueling decreased in a linear fashion from about 1.2 lb_m/BHP-hr at 1100 RPM to 1.0 lb_m/BHP-hr at about 1675 RPM. As the amount of natural gas substitution increased, the optimum operating speed, defined as the speed corresponding to the lowest brake specific fuel consumption, increased from approximately 1550 RPM for 100 % diesel fueling to about 1675 RPM for 37 % diesel fueling.

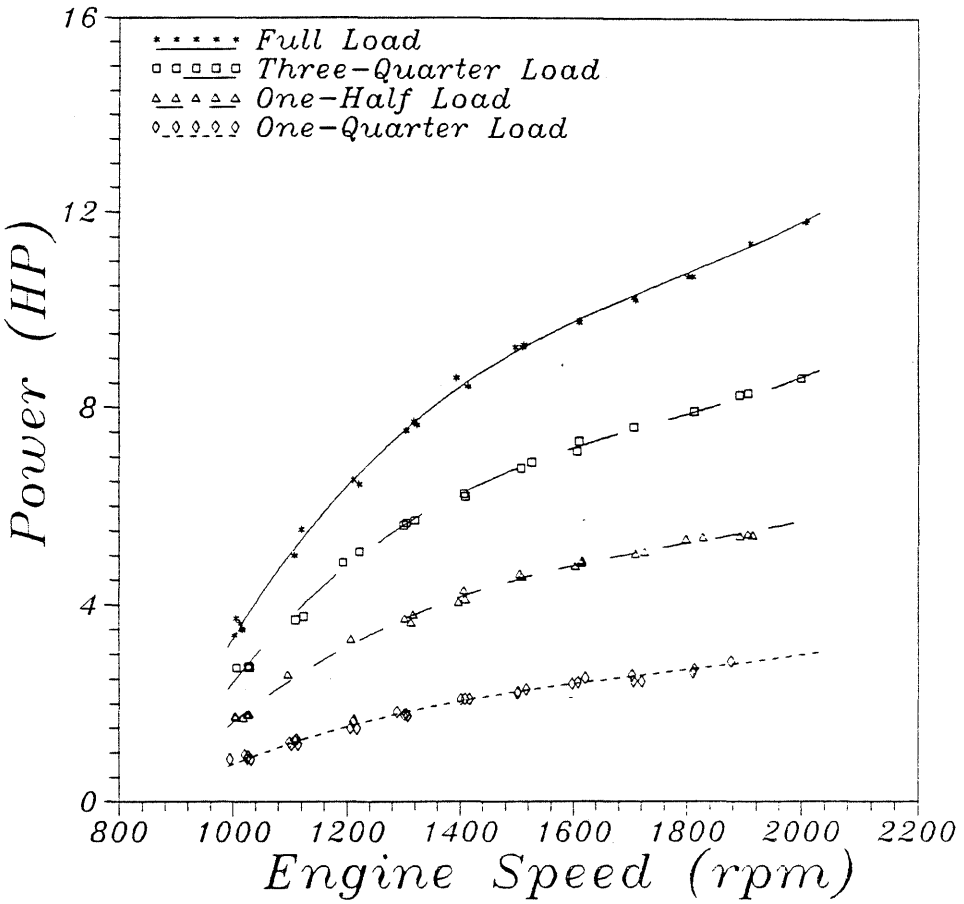

Figure IV.2 - Power as a function of engine speed and load.

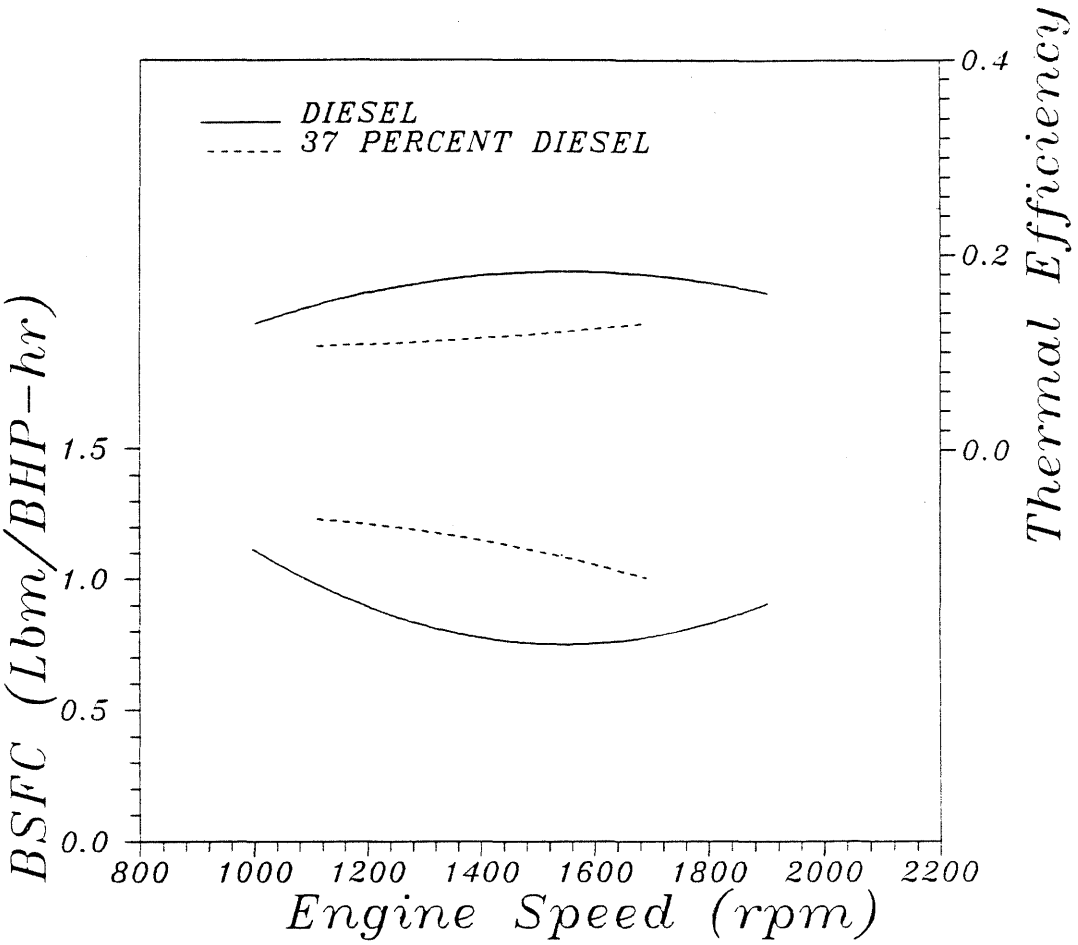

Figure IV.3 - BSFC and η_{th} as a function of engine speed and percentage of diesel fueling, one-quarter load conditions.

The thermal efficiency was determined by dividing the power of the engine by the total injected fuel energy determined from the lower heating values of natural gas and diesel fuel. The thermal efficiency for 100 % diesel fueling increased from 13 % at 1000 RPM to a maximum of 18 % at 1575 RPM and decreased to 16 % at 1900 RPM. The thermal efficiency for 37 % diesel fueling increased from 10 % at 1100 RPM to 13 % at 1675 RPM.

The variations in BSFC and thermal efficiency as a function of engine speed at one-half load for 100 % , 37 % , and 27 % diesel fueling are shown in Figure IV.4. The BSFC for 100 % diesel fueling decreased from 0.7 lb_m/BHP-hr at 1000 RPM to a minimum of approximately 0.55 lb_m/BHP-hr at a speed of 1575 RPM and increased to 0.6 lb_m/BHP-hr at 2000 RPM. The brake specific fuel consumption for 37 % diesel fueling was found to decrease in a linear fashion from 0.6 lb_m/BHP-hr at 1200 RPM to 0.55 lb_m/BHP-hr at a speed of 1450 RPM. The BSFC trend for 27 % diesel fueling was similar to that for 100 % diesel fueling, but was more pronounced. The BSFC trend decreased from approximately 0.9 lb_m/BHP-hr at 1000 RPM to a minimum of 0.55 lb_m/BHP-hr at 1500 RPM and increased to 0.6 lb_m/BHP-hr at a speed of about 1625 RPM.

The thermal efficiency for 100 % diesel fueling at one-half load showed a similar trend as for one-quarter load. The thermal efficiency increased from 22 % at 1000 RPM to a maximum of 26 % at 1575 RPM and then decreased to 22 % at 2000 RPM. The thermal efficiency for 37 % diesel fueling increased linearly with values ranging from 22 % at 1200 RPM to about 25 % at a speed of 1450 RPM. For 27 % diesel fueling, the thermal

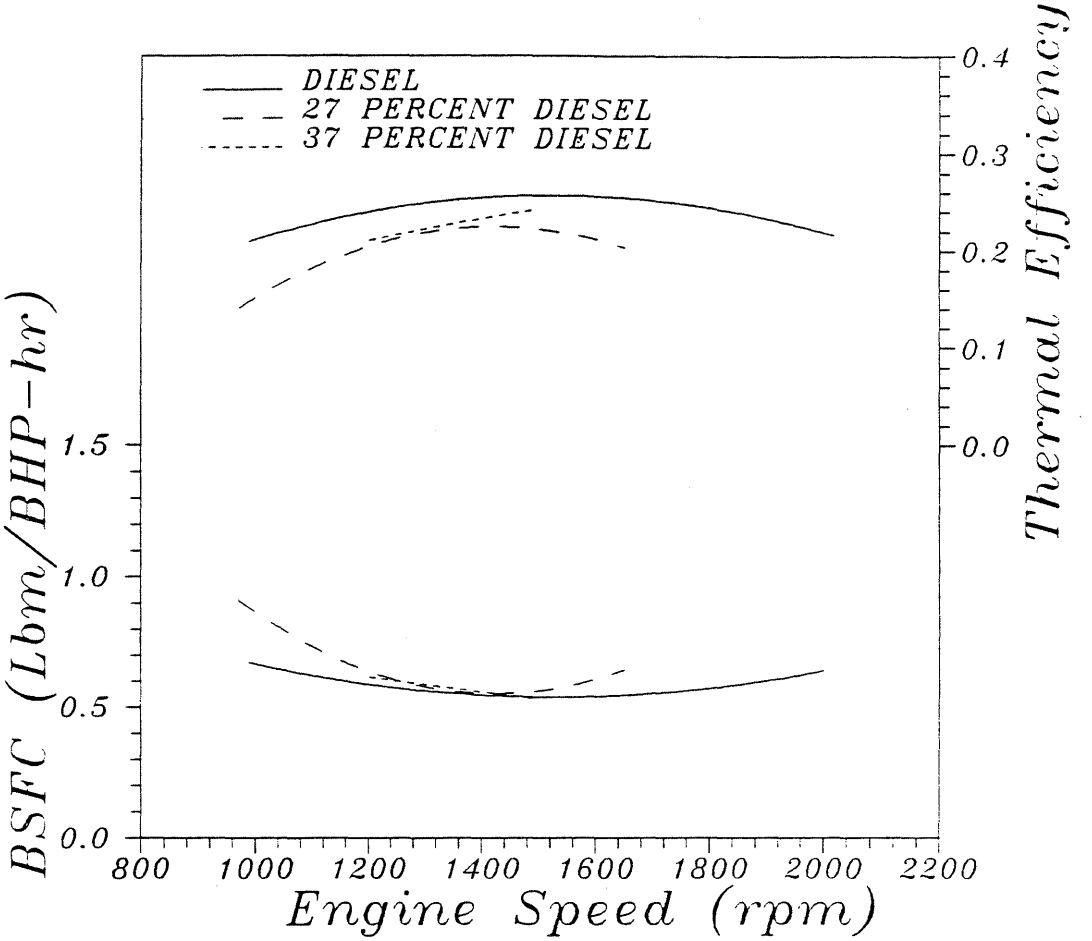

Figure IV.4 - BSFC and η_{th} as a function of engine speed and percentage of diesel fueling, one-half load conditions.

efficiency trend was similar to that for 100 % diesel fueling, but was more pronounced. The values for this case ranged from approximately 15 % at 1000 RPM to a maximum value of 23 % at 1500 RPM and decreased as the engine speed was increased further to 20 % at 1600 RPM.

Figure IV.5 shows the brake specific fuel consumption and thermal efficiency at three-quarter load as a function of engine speed for 100 %, 50 %, and 32 % diesel fueling. The BSFC trend for 100 % diesel fueling was essentially constant regardless of engine speed remaining around 0.55 lb_m/BHP-hr from 1000 RPM to 2000 RPM. The trend for 50 % diesel fueling was virtually the same as for 100 % diesel fueling, but covered the 1000 RPM to 1500 RPM speed range. The brake specific fuel consumption trend for 32 % diesel fueling was higher than that found for 100 % and 50 % diesel fueling from 975 RPM to about 1100 RPM where there was practically no difference among the three trends.

The trend for thermal efficiency for 100 % diesel fueling was similar to the trends exhibited for the other baseline cases discussed previously. The thermal efficiency increased from 23 % at 1000 RPM to a maximum of 28 % at approximately 1525 RPM and decreased as the engine speed was increased further to 25 % at 2000 RPM. The efficiency for 50 % diesel fueling had a similar trend as for 100 % diesel fueling, but covered only a portion of the range compared to the baseline case. The thermal efficiency increased from 20 % at 1000 RPM to 24 % at about 1500 RPM. The trend for 32 % diesel fueling was similar to the trend for 100 % diesel fueling. The efficiency for 32 % diesel fueling increased from 18 % at 1000 RPM to a maximum

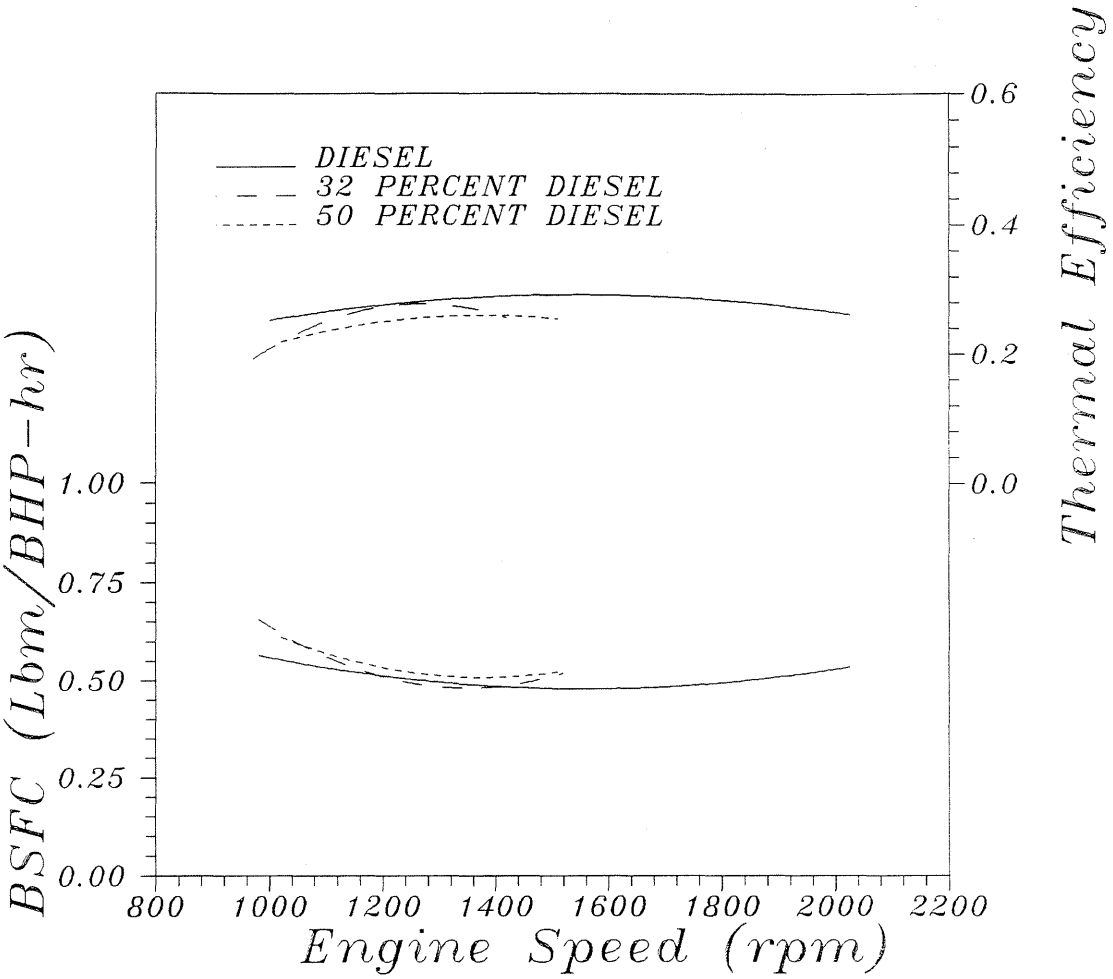

Figure IV.5 - BSFC and η_{th} as a function of engine speed and percentage of diesel fueling, three-quarter load conditions.

of 26 % at a speed of 1250 RPM and decreased with increasing engine speed to 25 % at about 1375 RPM.

Figure IV.6 shows the brake specific fuel consumption and thermal efficiency as a function of engine speed for 100 % and 50 % diesel fueling for full load conditions. For full load conditions, audible knock was encountered with higher natural gas substitution rates. The BSFC and thermal efficiency curves were almost constant regardless of engine speed. This trend is also seen for the dual-fuel case shown. The figure indicates that for 50 % diesel fueling the minimum BSFC was 0.47 lb_m/BHP-hr. The maximum thermal efficiency using 50 % diesel fueling was 28 % as compared to 30 % for 100 % diesel fueling.

The maximum natural gas substitution on an energy basis that allowed repeatability was 50 % for full load, 68 % at three-quarter load, 73 % at one-half load, and 63 % for one-quarter load. Barbour and Lestz achieved in excess of 85 % natural gas substitution at light loads, but were able to achieve only 47 % substitution at full load. Wong et al. were only able to maintain rated power output with less than 65 % substitution of natural gas for diesel fuel.

Some data was obtained with higher percentages of natural gas fueling but not reported due to lack of repeatability. At full load, one operating point was obtained which had 75 % natural gas fueling. There was one set of data points obtained in the 92 % substitution range and two sets at approximately 80 % substitution at three-quarter load. For one-half load, one data set was in the 90 % natural gas fueling range and three sets were

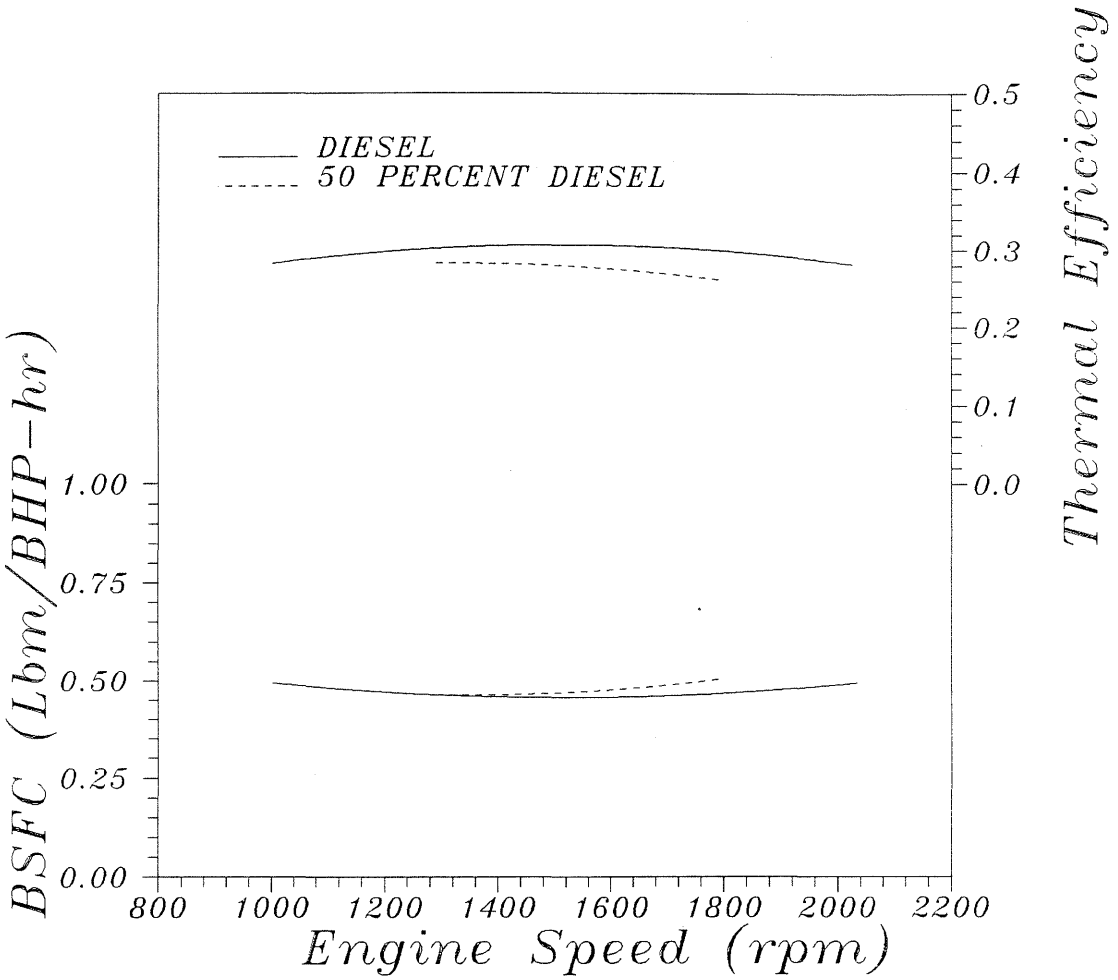

Figure IV.6 - BSFC and η_{th} as a function of engine speed and percentage of diesel fueling, full load conditions.

in the 85 % - 89 % range. There were four sets at one-quarter load in the 70 % - 77 % range spanning 400 RPM. Because of the rather coarse control of the pilot allowed by the current system, repeating cases with low pilot amounts was difficult.

Volumetric Efficiency

Volumetric efficiency as a function of engine speed for 100 % and 37 % diesel fueling at one-quarter load is shown in Figure IV.7. The general trend seen in these figures is that the volumetric efficiency was inversely proportional to the amount of natural gas utilized. The volumetric efficiency for 100 % diesel fueling decreased from approximately 80 % at 1000 RPM to 66 % at 1900 RPM. The volumetric efficiency for 37 % diesel fueling decreased in a near linear fashion from 76 % at 1000 RPM to 64 % at 1700 RPM. There was essentially no change in the volumetric efficiency between the two cases shown at 1000 RPM. However, the utilization of 37 % diesel fueling decreased the volumetric efficiency about 6 % at 1700 RPM.

One-half load volumetric efficiency for 100 %, 37 %, and 27 % diesel fueling as a function of engine speed is shown in Figure IV.8. The volumetric efficiency for 100 % diesel fueling decreased from 79 % at 1000 RPM to 64 % at 2000 RPM. The figure indicates that the volumetric efficiency for 37 % diesel fueling decreased in a near linear fashion from 72 % at 1200 RPM to 66 % at 1500 RPM. There was essentially no change in the volumetric efficiency between 100 % and 37 % diesel fueling at 1200 RPM. However, there was a decrease of about 3 % in the volumetric efficiency between 100 % and 37 % diesel fueling at 1500 RPM. The volumetric efficiency for

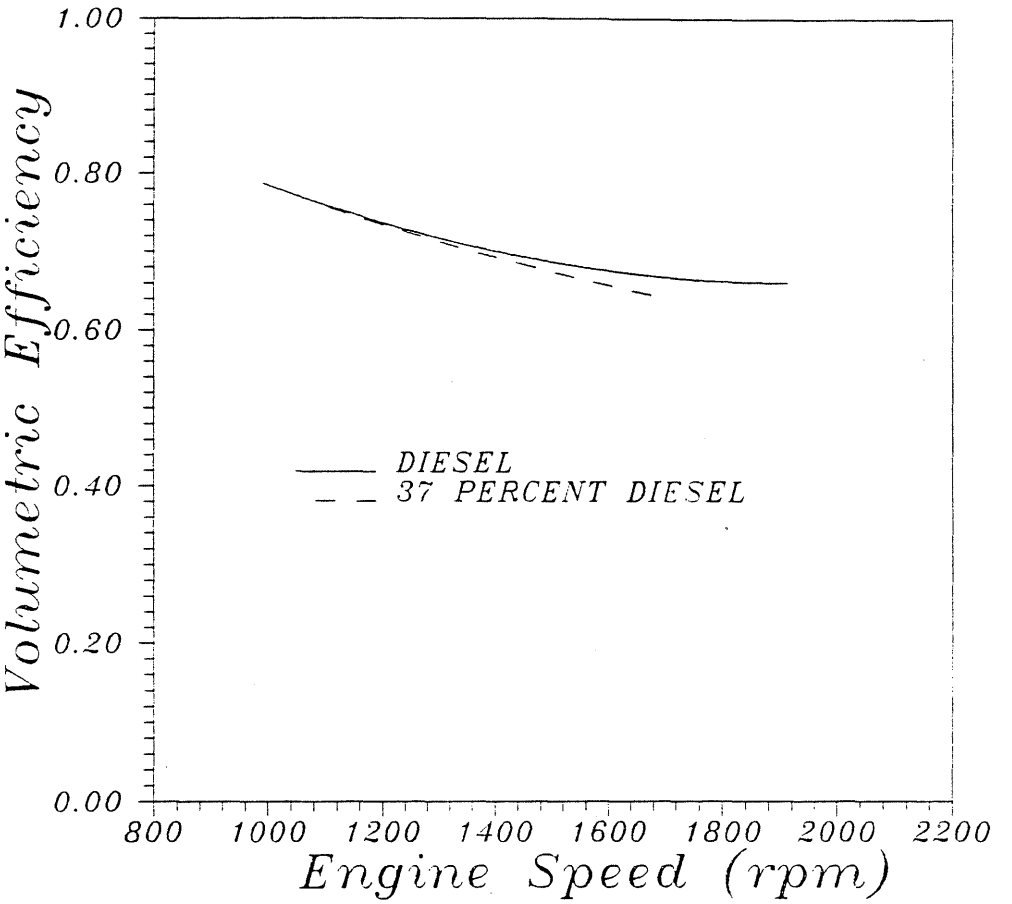

Figure IV.7 - Volumetric efficiency as a function of engine speed and percentage of diesel fueling, one-quarter load conditions.

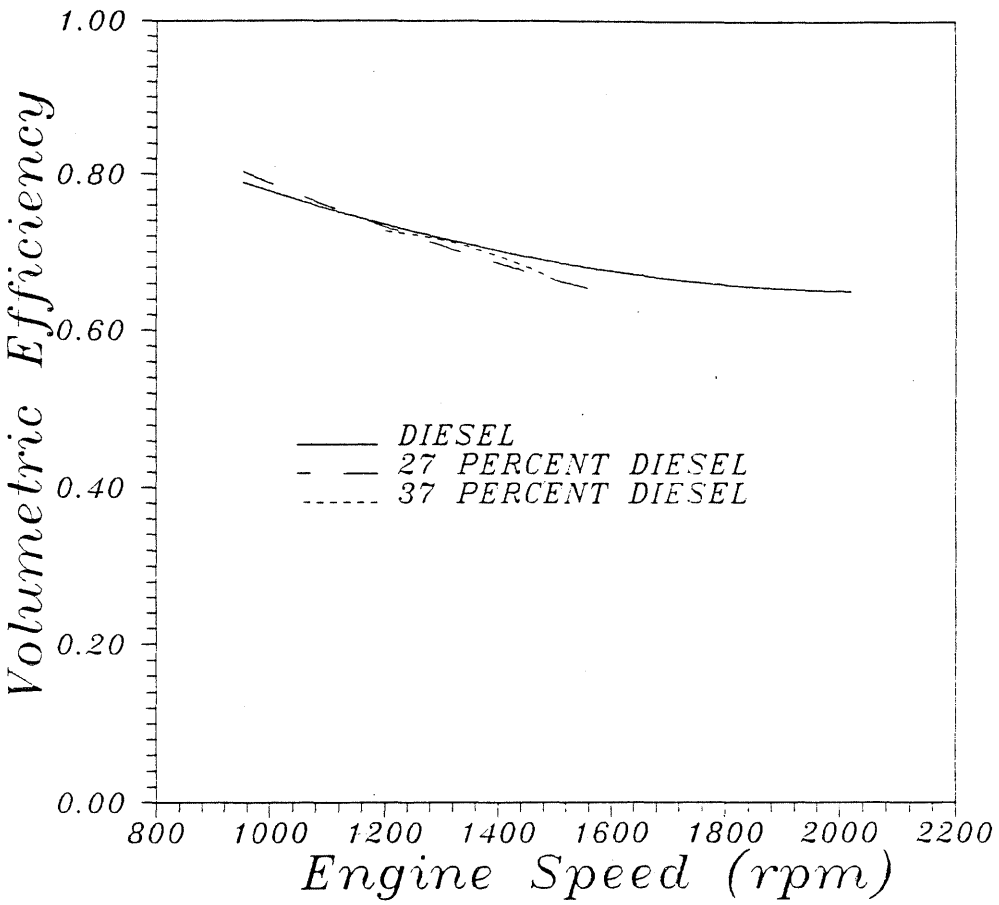

Figure IV.8 - Volumetric efficiency as a function of engine speed and percentage of diesel fueling, one-half load conditions.

27 % diesel fueling ranged from 81 % at 1000 RPM, essentially no difference from the corresponding efficiency for 100 % diesel fueling, to 64 % at approximately 1575 RPM, representing a decrease of about 6 % from the equivalent efficiency for 100 % diesel fueling.

The variation in volumetric efficiency at three-quarter load for 100 %, 50 %, and 32 % diesel fueling is shown in Figure IV.9 as a function of engine speed. The general trends shown are similar to those for the one-half load case discussed earlier. The volumetric efficiency for 100 % diesel fueling ranged from 78 % at 1000 RPM to 64 % at 2000 RPM. The volumetric efficiency for 50 % diesel fueling decreased in a near linear fashion from 79 % at 1000 RPM, no difference from the corresponding efficiency for 100 % diesel fueling, to 66 % at 1500 RPM, which represents a decrease of 4 % from the corresponding efficiency for 100 % diesel fueling. This figure indicates that the volumetric efficiency for 32 % diesel fueling decreased from 80 % at 1000 RPM, no change from the corresponding efficiencies for 100 % and 50 % diesel fueling, to 66 % at approximately 1450 RPM, a decrease of 4 % from the corresponding efficiency for 100 % diesel fueling.

Figure IV.10 shows the variation in volumetric efficiency as a function of engine speed for full load with 100 % and 50 % diesel fueling. The volumetric efficiency for 100 % diesel fueling decreased from 79 % at 1000 RPM to 64 % at 2000 RPM. The figure indicates that the volumetric efficiency for 50 % diesel fueling decreased from 70 % at about 1280 RPM, no change from the corresponding efficiency for 100 % diesel fueling, to 62 % at 1800 RPM, a decrease of 6 % from the corresponding efficiency for 100 % diesel fueling.

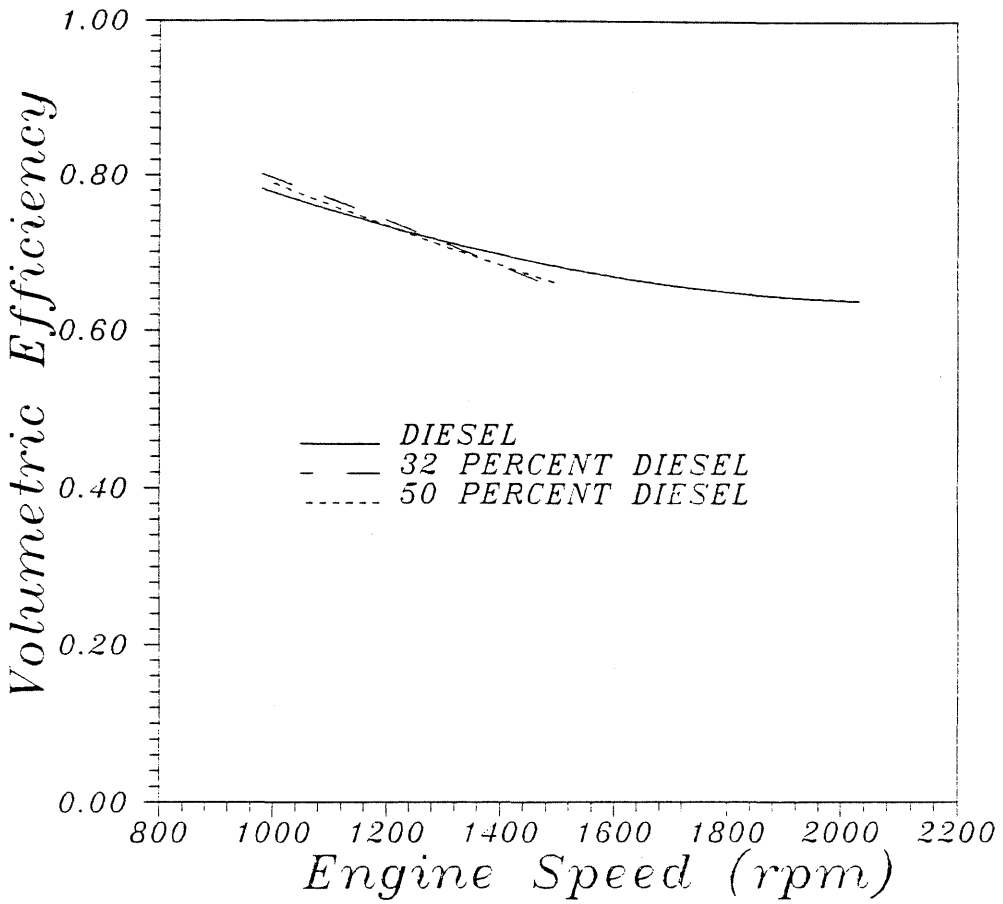

Figure IV.9 - Volumetric efficiency as a function of engine speed and percentage of diesel fueling, three-quarter load conditions.

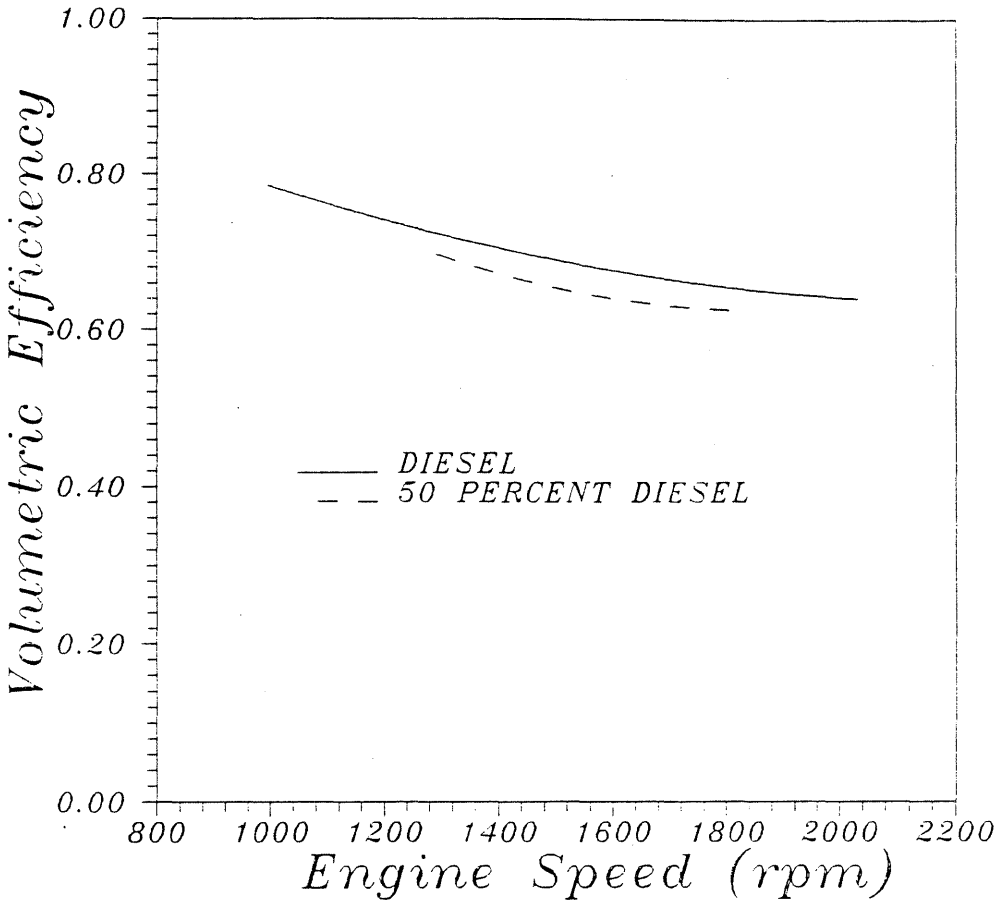

Figure IV.10 - Volumetric efficiency as a function of engine speed and percentage of diesel fueling, full load conditions.

The variation in volumetric efficiency was found to be a function of load and the amount of natural gas utilized. For a given percentage of natural gas fueling, the volumetric efficiency was inversely proportional to the loading of the engine as the aspirated natural gas displaced a portion of the inlet air. The volumetric efficiency was also found to be inversely proportional to the percentage of natural gas used for a given load.

Equivalence Ratio

The variation of the total air/fuel equivalence ratio (ϕ) as a function of engine speed at one-quarter load for 100 % and 37 % diesel fueling is shown in Figure IV.11. The actual air/fuel ratio was determined using the total amount of fuel (natural gas + diesel fuel) on a mass basis and the stoichiometric air/fuel ratio was determined using a "weighted" average of the stoichiometric air/fuel ratio values for natural gas and diesel fuel. The equivalence ratio for 100 % diesel fueling increased from 0.21 at 1000 RPM to 0.32 at 1900 RPM. The figure indicates that the equivalence ratio for 37 % diesel fueling increased from 0.36 at 1000 RPM, an increase of 71 % over the corresponding ϕ for 100 % diesel fueling, to 0.62 at 1500 RPM, which represents an increase of 94 % over the corresponding ϕ for 100 % diesel fueling.

The equivalence ratio at one-half load for 100 % and 27 % diesel fueling as a function of engine speed is shown in Figure IV.12. This figure indicates that the equivalence ratio for 100 % diesel fueling increased from 0.28 at

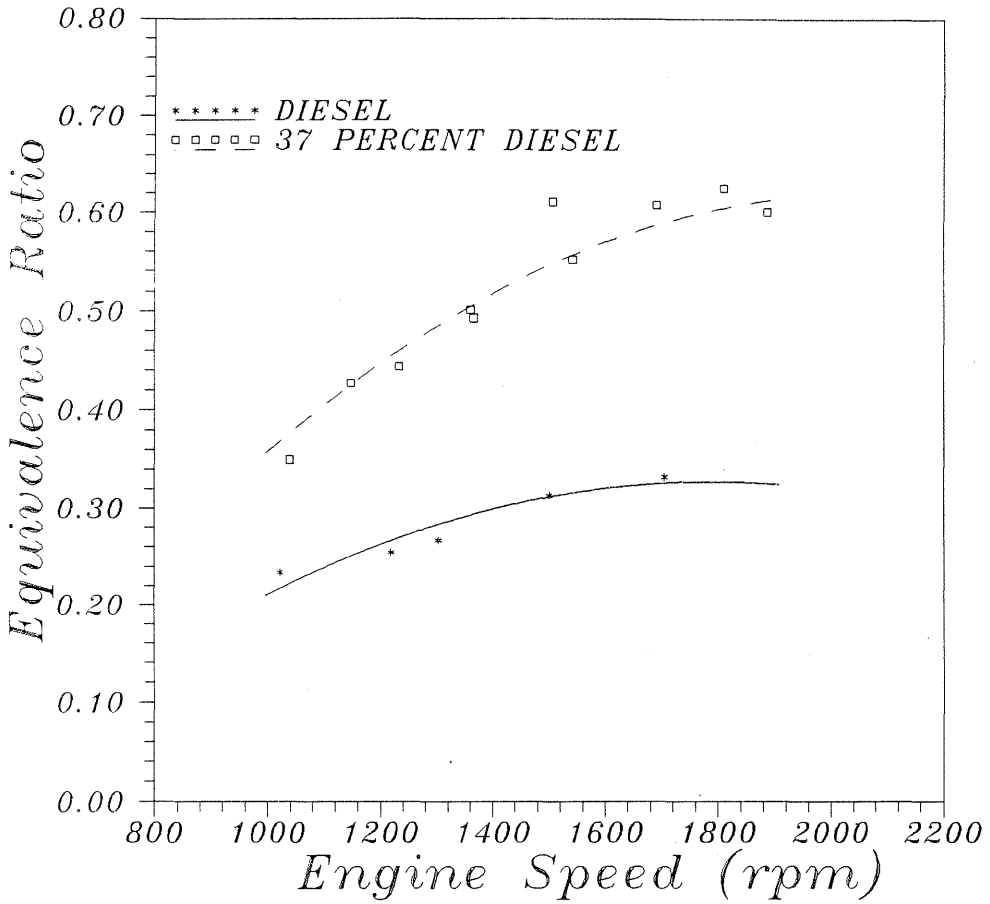

Figure IV.11 - Equivalence ratio as a function of engine speed and percentage of diesel fueling, one-quarter load conditions.

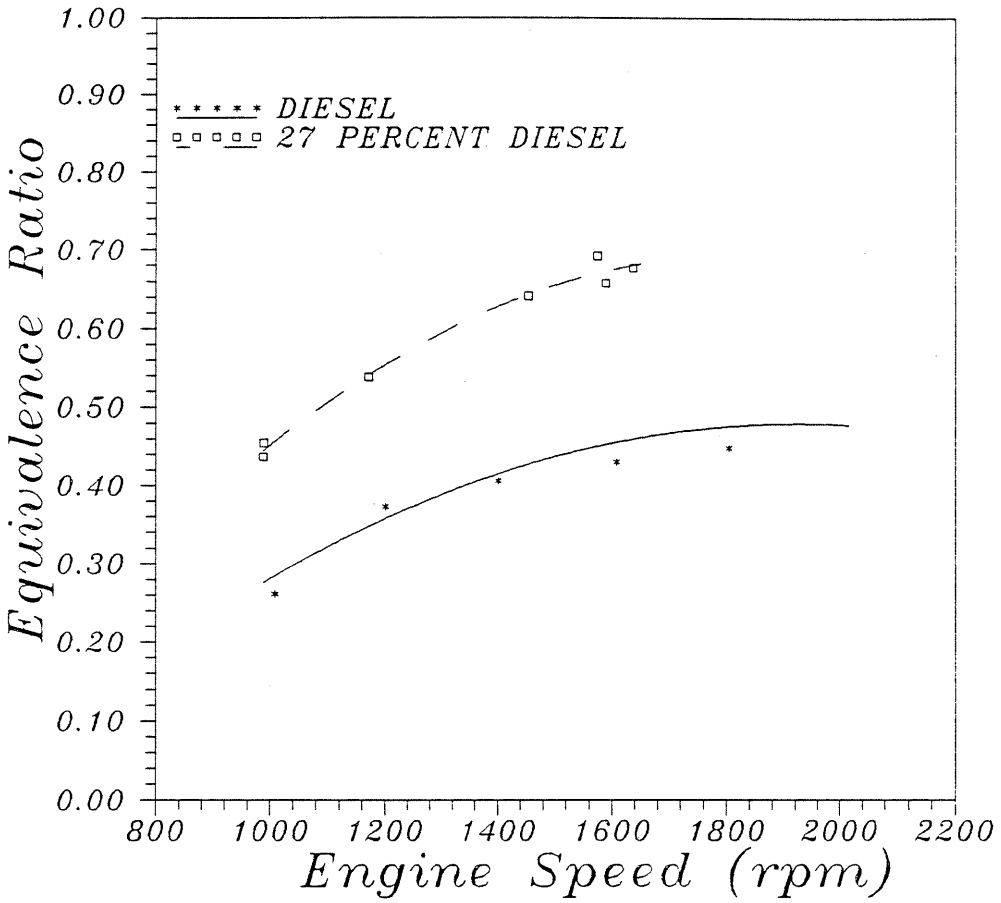

Figure IV.12 - Equivalence ratio as a function of engine speed and percentage of diesel fueling, one-half load conditions.

1000 RPM to 0.48 at 2000 RPM. The equivalence ratio for 27 % diesel fueling increased from 0.44 at 1000 RPM, an increase of 57 % over the corresponding ϕ for 100 % diesel fueling, to 0.68 at about 1650 RPM, representing an increase of 48 % over the corresponding ϕ for 100 % diesel fueling.

Figure IV.13 shows the variation of equivalence ratio for 100 %, 50 %, and 32 % diesel fueling as a function of engine speed at three-quarter load. The general trends for the previous loads discussed are also seen here. The equivalence ratio for 100 % diesel fueling increased from 0.36 at 1000 RPM to 0.64 at 2000 RPM. The equivalence ratio for 50 % diesel fueling ranged from 0.48 at 1000 RPM, representing an increase of 33 % over the corresponding ϕ for baseline conditions, to 0.80 at about 1450 RPM, an increase of approximately 43 % over the analogous ϕ for 100 % diesel fueling. The equivalence ratio for 32 % diesel fueling climbed from 0.52 at 1000 RPM, an increase of 8 % over the related equivalence ratio for 50 % diesel fueling and 44 % over the ϕ for baseline conditions, to 0.81 at 1500 RPM, representing an increase of 40 % above the corresponding equivalence ratio for 100 % diesel fueling.

The full load equivalence ratio is shown for 100 % and 50 % diesel fueling as a function of engine speed in Figure IV.14. The equivalence ratio for 100 % diesel fueling increased from 0.44 at 1000 RPM to 0.80 at 2000 RPM. The equivalence ratio for 50 % diesel fueling ranged from 0.57 at 1000 RPM, representing an increase of 30 % over the related equivalence ratio for 100 % diesel fueling, to approximately 1.10 at 1800 RPM, an increase of 39 % over the ϕ obtained at 1800 RPM for baseline conditions.

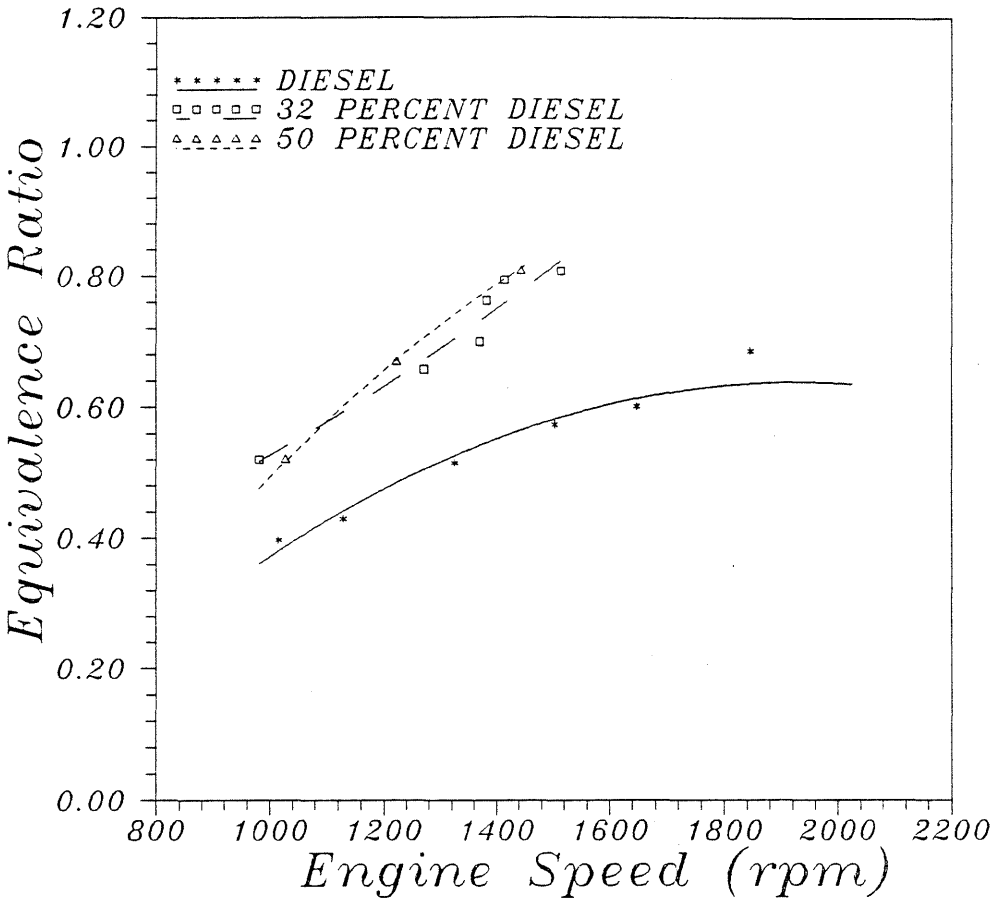

Figure IV.13 - Equivalence ratio as a function of engine speed and percentage of diesel fueling, three-quarter load conditions.

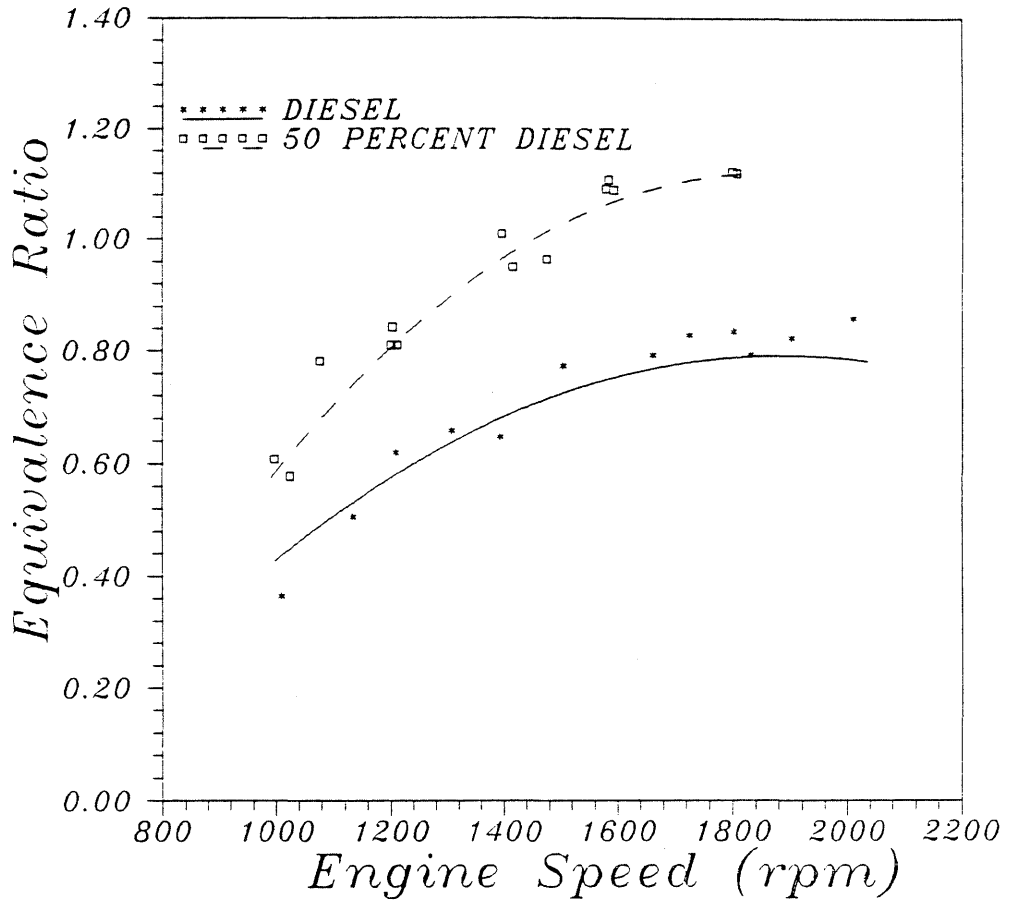

Figure IV.14 - Equivalence ratio as a function of engine speed and percentage of diesel fueling, full load conditions.

As demonstrated in the figures cited above, the equivalence ratios obtained during dual-fuel testing were significantly richer than those obtained during baseline testing. There are two primary reasons for the increased dual-fuel equivalence ratios. The decrease in the dual-fuel volumetric efficiency (η_v) indicates that the introduction of natural gas into the intake air stream displaced some of the air which normally would have been inducted into the engine during the intake stroke, decreasing the actual air/fuel ratio, which in turn increased the equivalence ratio. Secondly, the decrease in the thermal efficiencies (η_{th}) obtained during dual-fuel testing indicates that the engine required more fuel to maintain the baseline torque and power characteristics at the specified load, which also increased the equivalence ratio. The increase in the dual-fuel equivalence ratio was found to be inversely proportional to the engine loading. For a given percentage of natural gas utilized, increasing the load resulted in a smaller increase in ϕ above the corresponding equivalence ratio obtained for baseline conditions. It was also found that ϕ was proportional to the amount of natural gas utilized. For a given load, increasing the amount of natural gas used resulted in a richer equivalence ratio, as expected. The increase in the equivalence ratio for dual-fueling above that acquired during baseline testing, with two exceptions, was also found to be proportional to engine speed. The two exceptions to this trend were the 27 % diesel fueling case at one-half load and the 32 % diesel fueling case at three-quarter load. The equivalence ratio for baseline as well as dual-fuel operating conditions also increased due to the flow restriction of inlet air by the ASME nozzle. That is, the engine was not receiving adequate air for complete combustion to occur at higher speeds and loads.

Exhaust Gas Temperature

The variation of exhaust gas temperature in degrees Fahrenheit at one-quarter load for 100 % and 37 % diesel fueling as a function of engine speed is shown in Figure IV.15. This case was unique in that it was the only load where the exhaust gas temperature for dual-fueling was higher than for baseline conditions. This probably occurred due to the increased ignition delay caused by the introduction of natural gas and led to a greater portion of the total injected fuel energy to be rejected to the exhaust gas. This result is also confirmed by the low thermal efficiency for the one-quarter load dual-fuel case. That is, the lower thermal efficiency would be accompanied by higher cooling and exhaust gas energy rejection. It should be noted that the injection timing was not altered from normal diesel operation. The exhaust gas temperature for baseline conditions increased in a linear fashion from 210°F at 1000 RPM to 320°F at 1900 RPM. The corresponding trend for the dual-fuel case shown increased from about 220°F at 1100 RPM to 300°F at about 1650 RPM. This represents a modest 3 % increase over the corresponding exhaust temperature for 100 % diesel fueling at 1650 RPM.

Exhaust gas temperatures at one-half load as a function of engine speed for 100 %, 37 %, and 27 % diesel fueling are shown in Figure IV.16. This figure shows that the exhaust gas temperature is inversely proportional to the amount of natural gas utilized. The exhaust gas temperature for 100 % diesel fueling increased from about 210°F at 960 RPM to 400°F at 2000 RPM. The temperature for 37 % diesel fueling increased from 270°F at 1200 RPM, a 7 % decrease from the corresponding temperature for 100 %

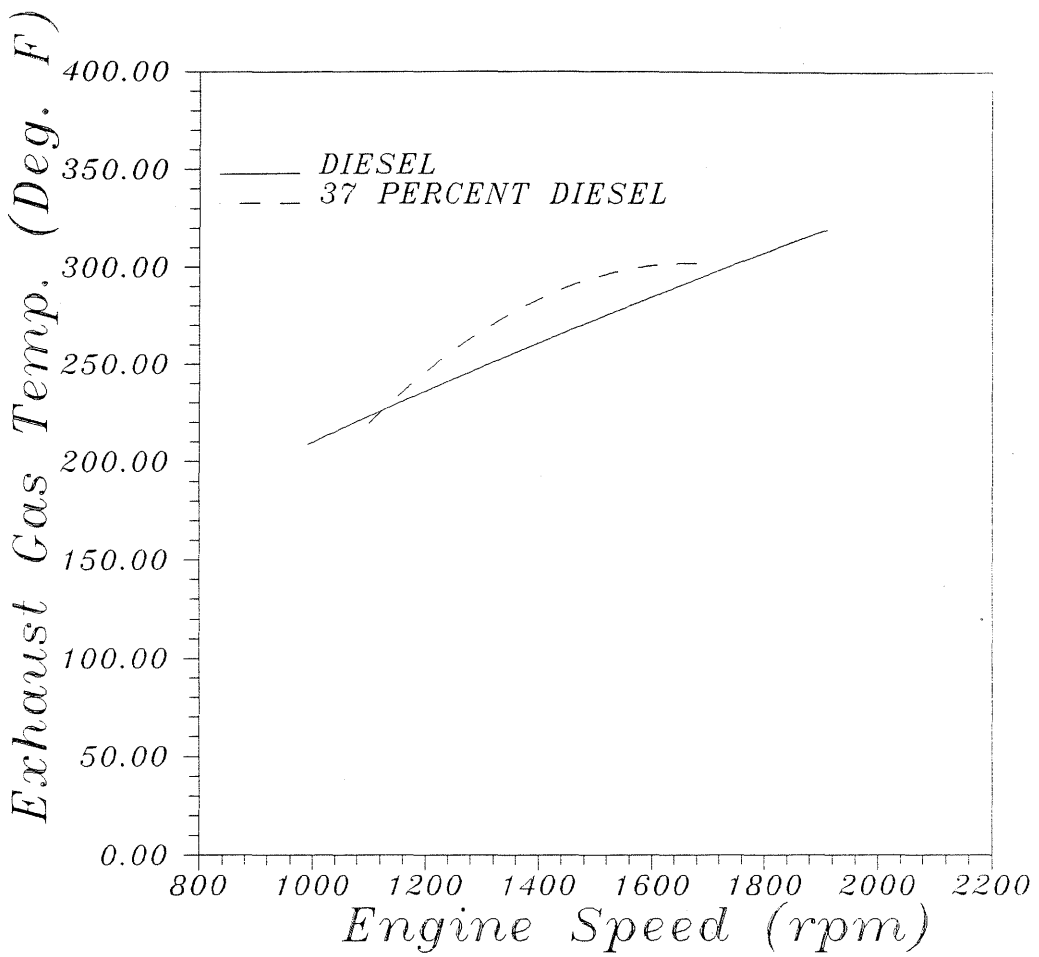

Figure IV.15 - Exhaust gas temperature as a function of engine speed and percentage of diesel fueling, one-quarter load conditions.

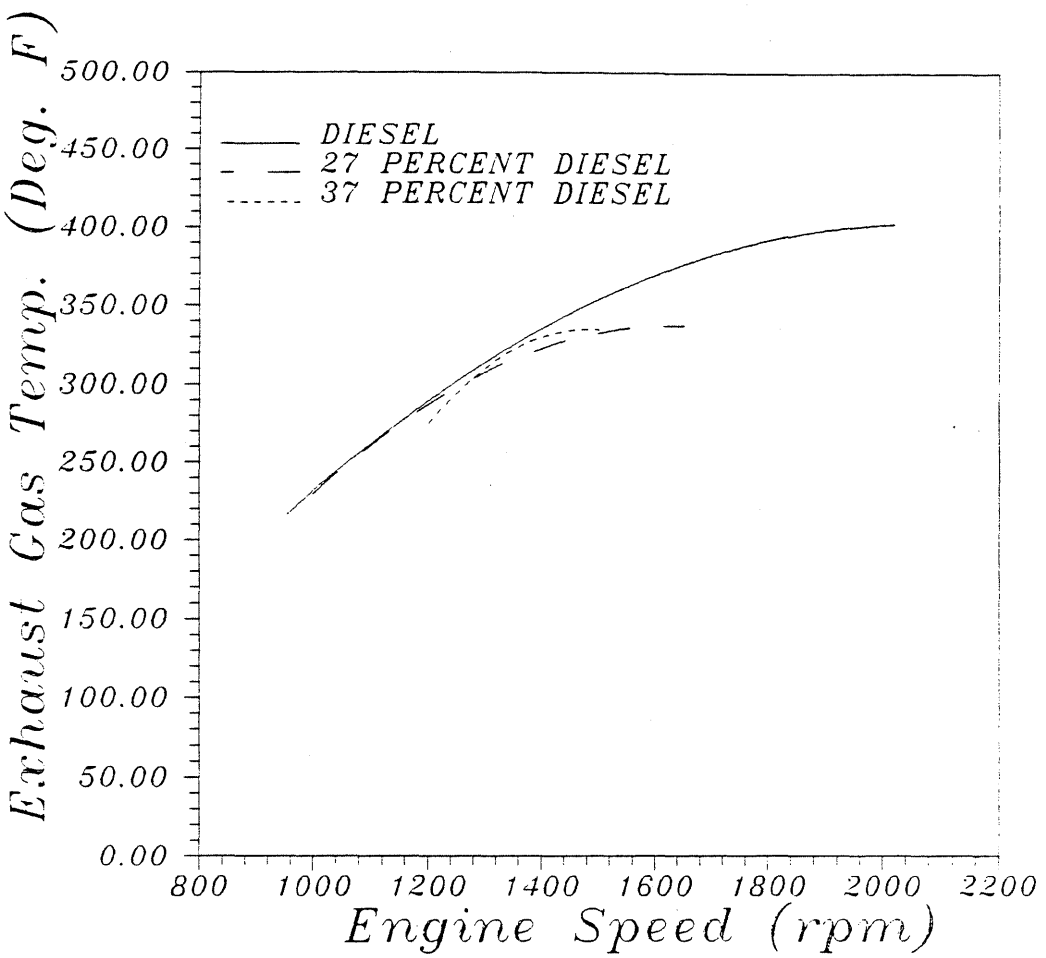

Figure IV.16 - Exhaust gas temperature as a function of engine speed and percentage of diesel fueling, one-half load conditions.

diesel fueling, to 330°F at 1500 RPM, representing a decrease of 6 % from the corresponding baseline exhaust gas temperature. The figure indicates that the exhaust gas temperature for 27 % diesel fueling increased from 230°F at 1000 RPM to about 340°F at 1640 RPM, which represents essentially no change in the exhaust temperatures for the 100 % and 27 % diesel fueling cases at 1000 RPM and a 9 % decrease from the baseline temperature at 1640 RPM.

Figure IV.17 shows variation in the exhaust gas temperature for 100 %, 50 %, and 32 % diesel fueling at three-quarter load as a function of engine speed. The trends for the two dual-fuel cases are similar to those found for the one-half load case discussed earlier. The exhaust temperature for baseline conditions increased from about 260°F at 1000 RPM to approximately 480°F at 2000 RPM. The figure indicates that the exhaust gas temperature for 50 % diesel fueling increased from about 260°F at 1000 RPM to 440°F at about 1550 RPM. There was essentially no change in the exhaust gas temperature between baseline conditions and 50 % diesel fueling at 1000 RPM and at 1550 RPM. The exhaust temperature for 32 % diesel fueling increased from about 285°F at 1000 RPM to approximately 390°F at 1450 RPM. There was an increase of 10 % in the temperature between 32 % diesel fueling and the temperatures obtained for 100 % and 50 % diesel fueling at 1000 RPM. However, there was a 7 % decrease in the exhaust gas temperature between the 50 % and 32 % diesel fueling cases and an 11 % decrease in the temperatures obtained from the baseline and 32 % diesel fueling cases shown at 1450 RPM.

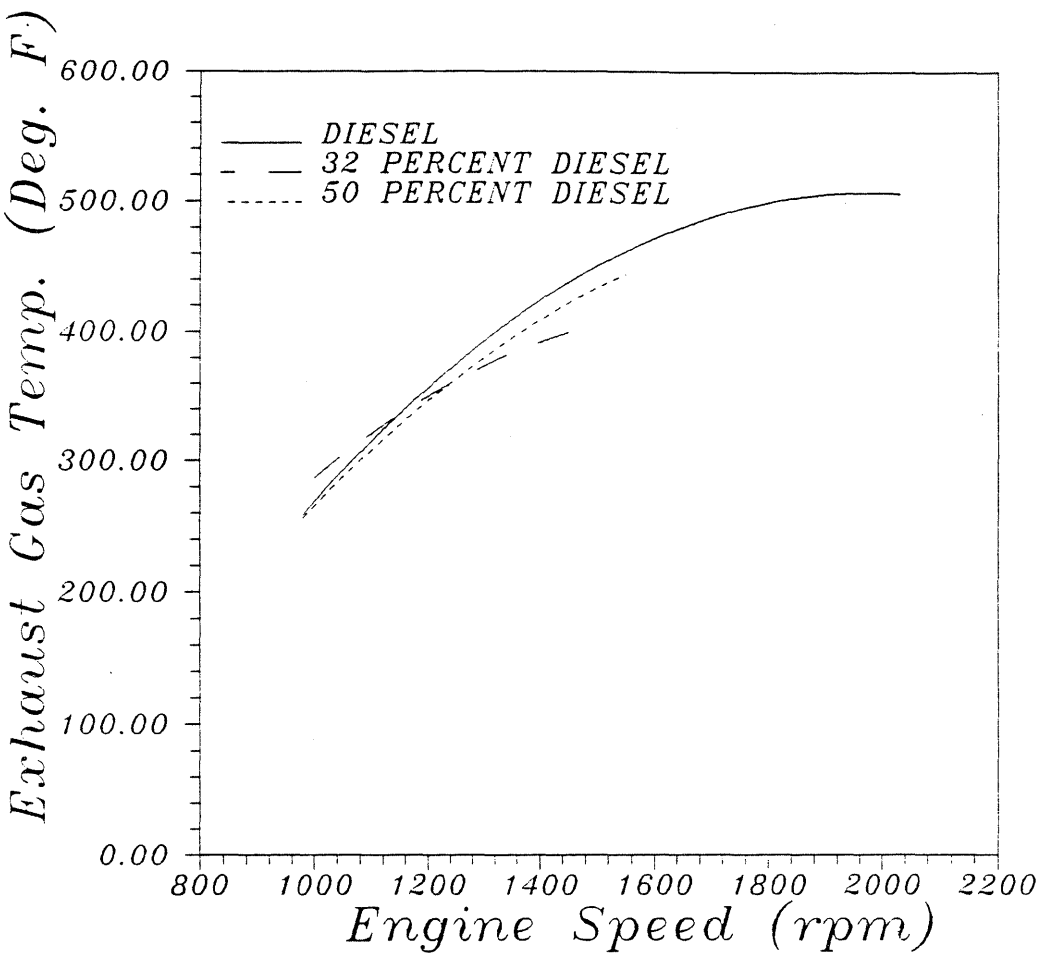

Figure IV.17 - Exhaust gas temperature as a function of engine speed and percentage of diesel fueling, three-quarter load conditions.

The variation in exhaust gas temperature for 100 % and 50 % diesel fueling at full load is shown as a function of engine speed in Figure IV.18. The trends shown in this figure are similar to those found for the one-half and three-quarter load cases discussed earlier. The figure indicates that the exhaust gas temperature for 100 % diesel fueling increased from 280°F at 1000 RPM to 640°F at 2000 RPM. The temperature for 50 % diesel fueling increased from 440°F at about 1280 RPM to a maximum temperature of approximately 570°F occurring at 1640 RPM, and then decreased to 540°F at 1800 RPM. The results for the dual-fuel case shown indicate that the temperature decrease with respect to baseline conditions increases with engine speed. There was essentially no change at 1000 RPM, a 5 % decrease at 1640 RPM, and a decrease of 16 % at 1800 RPM between the exhaust temperatures of the two cases obtained.

The results at hand indicate that the exhaust gas temperature is inversely proportional to load and the percentage of natural gas utilized. For a given load, increasing the amount of natural gas utilized produced cooler exhaust temperatures than those obtained for baseline conditions.

Emissions Results

Nitric Oxides

Figure IV.19 shows full load NO_x concentration in parts per million (ppm) as a function of engine speed for 100 % and 50 % diesel fueling. This figure indicates that NO_x levels increase with increasing natural gas substitution. The NO_x concentrations for 100 % diesel fueling decreased

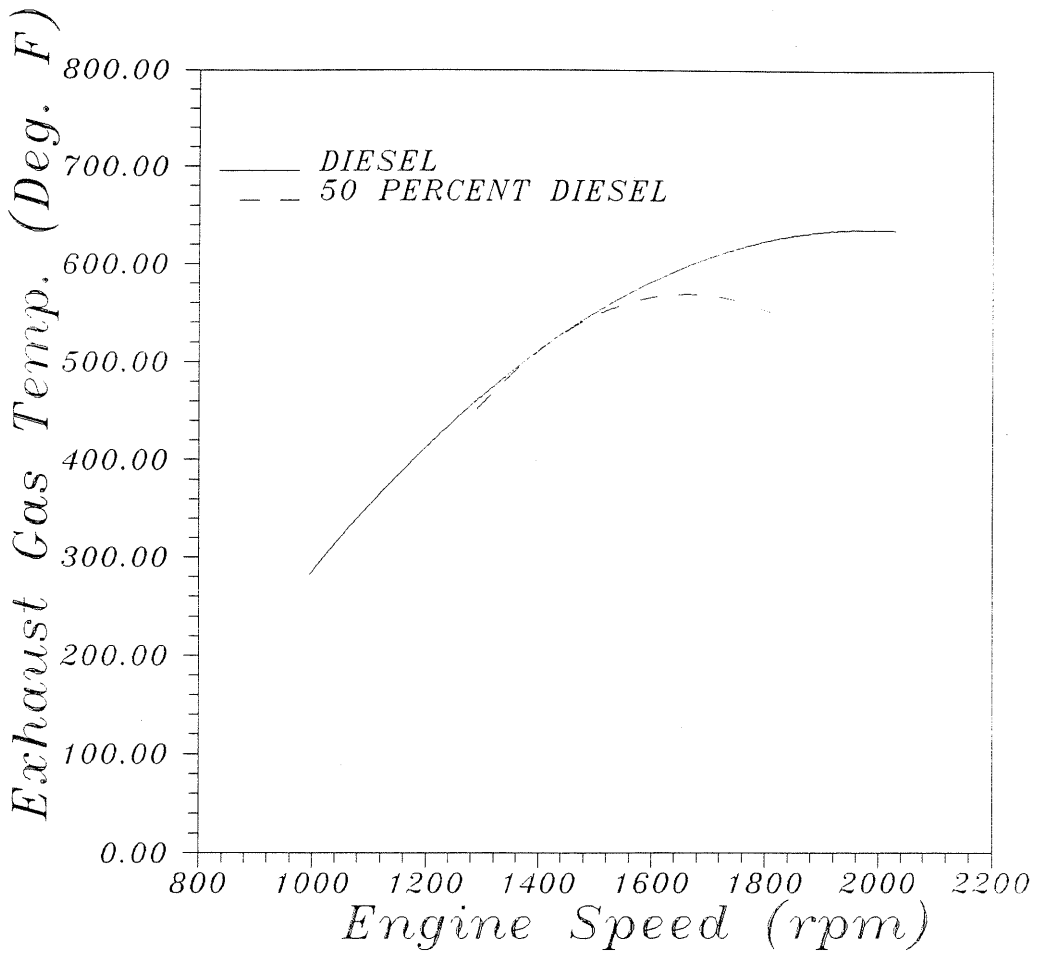

Figure IV.18 - Exhaust gas temperature as a function of engine speed and percentage of diesel fueling, full load conditions.

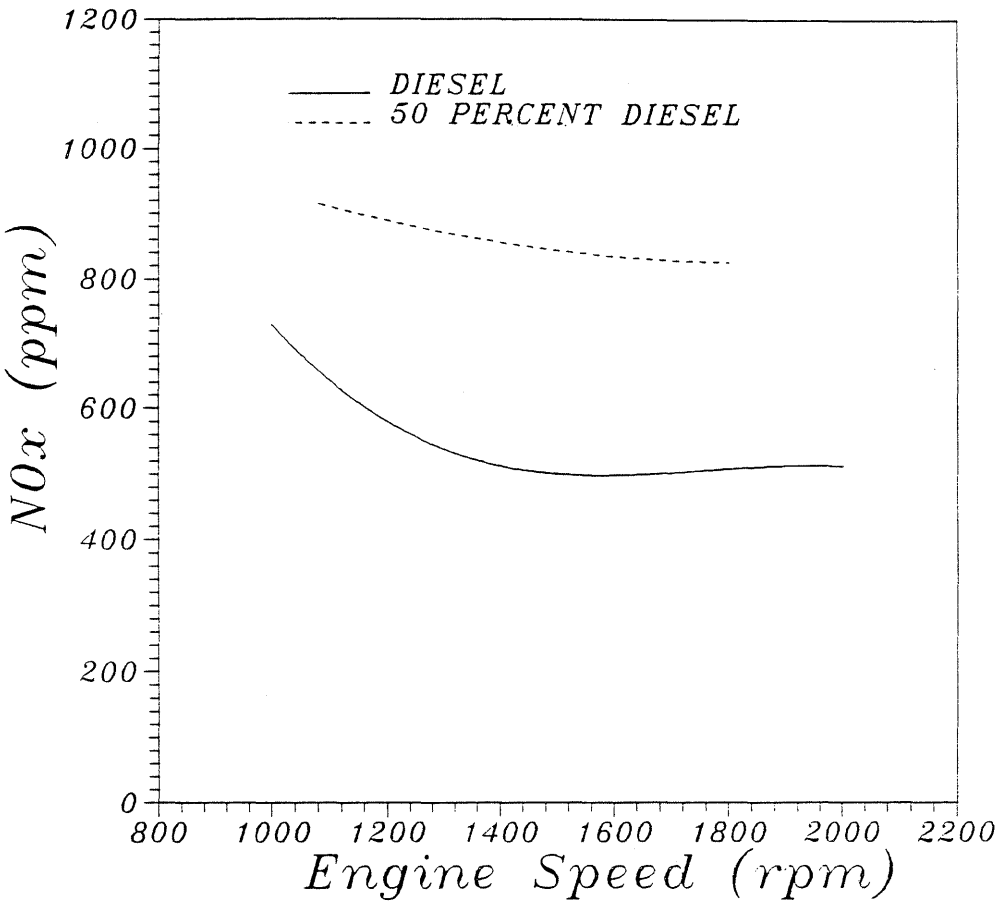

Figure IV.19 - NO_x concentration as a function of engine speed and percentage of diesel fueling, full load conditions.

from 720 ppm at 1000 RPM to a value of 500 ppm at 1520 RPM. The baseline NO_x level remained around 500 ppm as the speed was increased from 1520 RPM to 2000 RPM. The NO_x levels for 50 % diesel fueling were higher than those obtained for 100 % diesel fueling. The NO_x concentration for 50 % diesel fueling decreased from about 900 ppm at 1080 RPM to 820 ppm at 1800 RPM.

The NO_x concentrations encountered during dual-fuel testing show a dependence on the load and percentage of diesel fueling. This phenomenon is seen in other work referred to previously. Heywood [36] found that for indirect injected diesel engines that most of the NO forms in the prechamber and is then transported into the main combustion chamber where the higher than equilibrium levels are frozen as rapid mixing with "cool" air occurs. Ding and Hill [20] found at higher loads that the higher rate of energy release by the nearly homogeneous combustion raises the peak temperature and thus increases the amount of NO_x formed.

The results at hand show that the NO_x concentrations were increased for dual-fueling. For full load conditions, the equivalence ratios necessary for achieving the straight diesel fueling loads were found to be higher than the diesel cases. The more near stoichiometric fueling required likely resulted in higher combustion temperatures leading to the increased NO_x levels. The possibility that the ASME nozzle restricted the intake air flow was also investigated. Preliminary results of this investigation show that NO_x concentrations are decreased sharply when the engine is operated without an intake air restriction in dual-fuel mode. That is, the elevated NO_x concentrations

associated with higher loads and speeds appear to be partially a result of the restricted air flow of the inlet testing system, which resulted in higher equivalence ratios for dual-fueling.

Carbon Monoxide

Full load carbon monoxide concentrations for 100 % and 50 % diesel fueling are shown in Figure IV.20 as a function of engine speed. From the figure, the CO levels for 100 % diesel fueling increased from approximately 200 ppm at 1000 RPM to a maximum value of about 550 ppm at 1600 RPM. The CO levels decreased as the engine speed was increased further to about 400 ppm at 2000 RPM. CO concentrations for 50 % diesel fueling were higher than those obtained for baseline and remained essentially constant at approximately 1300 ppm from 1100 RPM to 1800 RPM.

The CO concentrations encountered during dual-fuel testing were significantly higher than those obtained during baseline testing. Acker [7], Barbour and Lestz [18], Karim [9], and Ding and Hill [20] encountered similar phenomena. The richer equivalence ratios encountered in dual-fuel testing would increase the CO concentration measured.

Sulfur Dioxide

The sulfur dioxide levels at full load for 100 % and 50 % diesel fueling are shown as a function of engine speed in Figure IV.21. The SO_2 concentrations for 100 % diesel fueling increase with engine speed from approximately 130 ppm at 1000 RPM to 210 ppm at a corresponding speed of 2000 RPM. The

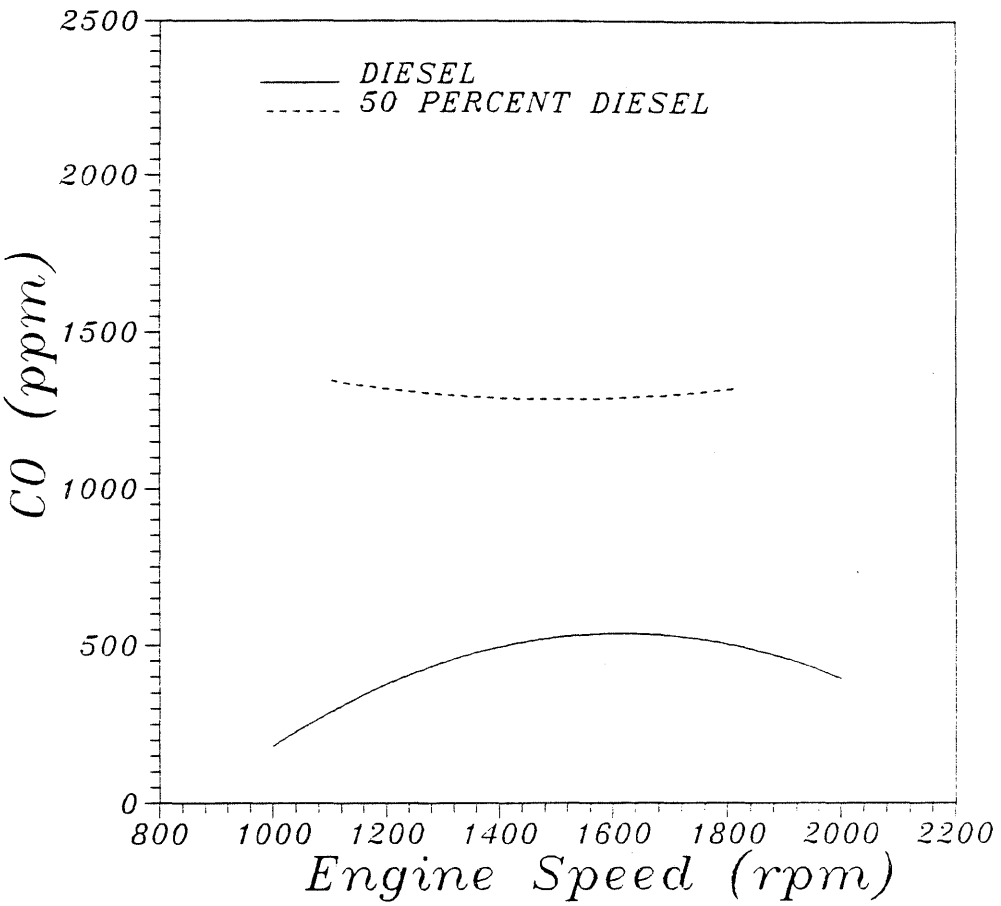

Figure IV.20 - CO concentration as a function of engine speed and percentage of diesel fueling, full load conditions.

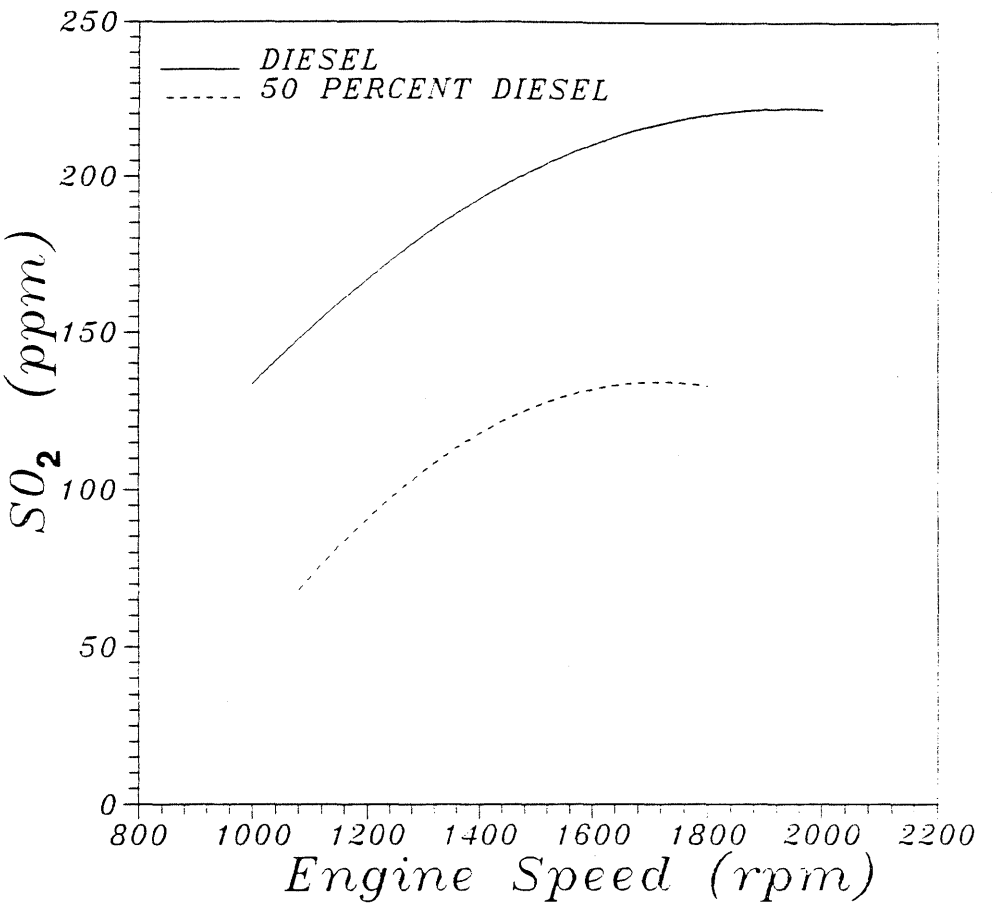

Figure IV.21 - SO_2 concentration as a function of engine speed and percentage of diesel fueling, full load conditions.

concentrations for 50 % diesel fueling followed a trend similar to that for baseline fueling, but were decreased in magnitude. As expected, this figure indicates that SO_2 levels were generally reduced by increased substitution of natural gas, which has a much lower sulfur content than diesel fuel.

Bosch Smoke Units

The variation in Bosch smoke number as a function of engine speed for full load is shown in Figure IV.22 for 100 % and 50 % diesel fueling. This figure indicates that decreases in the Bosch number were proportional to the amount of natural gas utilized. The figure indicates that the Bosch smoke number trend for 100 % diesel fueling increases from 1.1 at 1000 RPM to a maximum value of 3.3 occurring at a speed of 1680 RPM. This figure shows that the Bosch smoke number trend decreased from the maximum value as the engine speed was further increased to a value of 2.9 at 2000 RPM. The Bosch smoke number trend for 50 % diesel fueling was similar to that seen for 100 % diesel fueling, but reduced in magnitude. The Bosch smoke number for 50 % diesel fueling increased from 0.7 at 1200 RPM to a maximum value of 2.1 occurring at approximately 1600 RPM. The Bosch number then decreased with increasing speed to 2.0 at about 1700 RPM.

In this chapter, the results and discussion of results were presented. In the next chapter, a summary of the results and conclusions based on these results are presented.

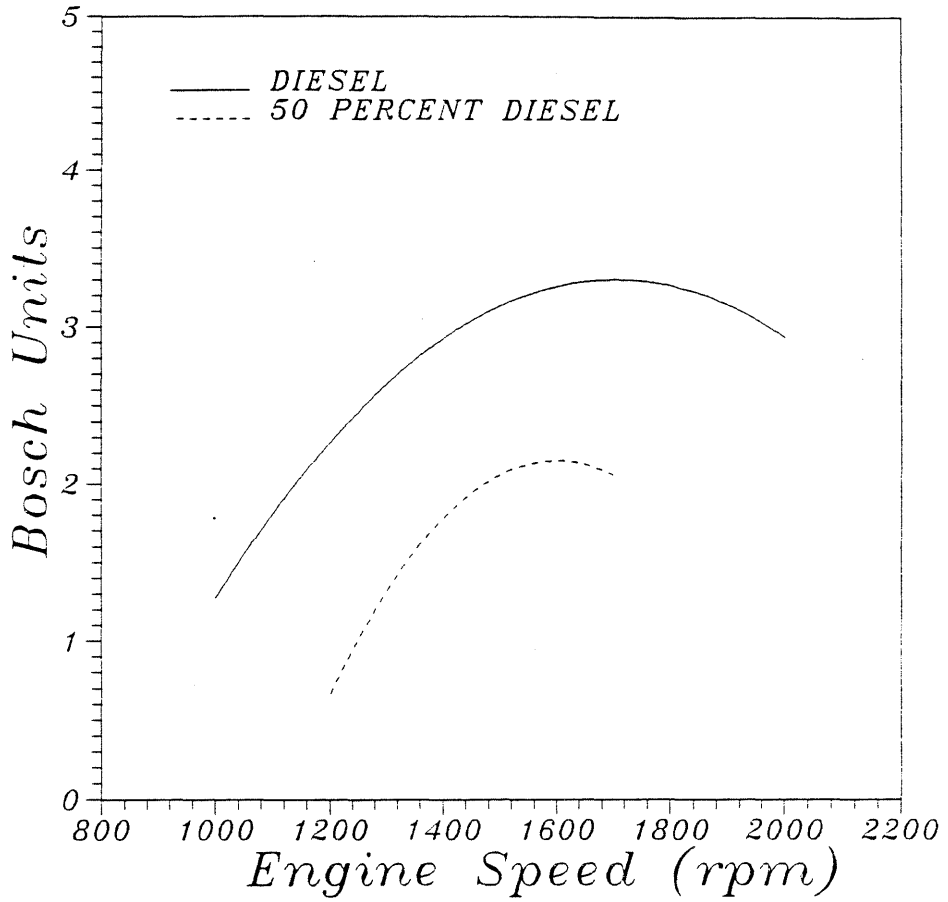

Figure IV.22 - Bosch smoke units as a function of engine speed and percentage of diesel fueling, full load conditions.

CHAPTER V

SUMMARY AND CONCLUSIONS

The purpose of this work was to take a "stock" single-cylinder, prechamber-type diesel engine, perform minor modifications as needed to allow extended use of natural gas as the primary fuel, investigate the performance characteristics of the engine in dual-fuel mode, and compare those results with results obtained during baseline testing. One of the objectives of this work was to maintain the baseline torque and power characteristics while operating in dual-fuel mode. The results of this research are summarized below.

Brake Specific Fuel Consumption and Thermal Efficiency

The brake specific fuel consumption for the dual-fuel cases, with the exception of the one-quarter load case, follow similar trends to those obtained during baseline testing. The one-quarter load case demonstrated that natural gas dual-fueling at light loads resulted in poor engine performance, which was consistent with the literature. While the dual-fuel BSFC obtained at higher loads generally followed the BSFC trends for 100 % diesel fueling, the dual-fuel trends were more sensitive to engine speed. This indicates that as the engine speed is varied from its optimum operating speed, the BSFC increases more dramatically in dual-fuel mode than for baseline diesel.

The thermal efficiency for dual-fueling was significantly decreased as compared to baseline at one-quarter and one-half load. This effect was also found in the literature. At higher loads, less penalty was found for natural gas dual-fueling.

Volumetric Efficiency

The volumetric efficiency was found to be dependent on load and the percentage of natural gas utilized. For a given percentage of natural gas fueling, the volumetric efficiency is inversely proportional to load. The volumetric efficiency was also found to be inversely proportional to the percentage of natural gas utilized for a given load.

Equivalence Ratio

The equivalence ratios obtained during dual-fueling were considerably richer than those obtained during baseline testing. The increase in the dual-fuel equivalence ratio was found to be inversely proportional to the loading of the engine. For a given percentage of natural gas used, increasing the load resulted in a smaller increase in ϕ above that obtained for baseline conditions. It was also found that ϕ was proportional to the amount of natural gas used. For a given load, increasing the amount of natural gas aspirated into the engine resulted in a richer ϕ. The increase in the equivalence ratio above that obtained during baseline testing, with two exceptions, was found to be proportional to engine speed. The two exceptions to this trend were 27 % diesel fueling at one-half load and 32 % diesel fueling at three-quarter load.

Exhaust Gas Temperature

The results at hand indicate that the exhaust gas temperature is inversely proportional to load and the percentage of natural gas utilized. For a given load, increasing the amount of natural gas produced cooler exhaust gas temperatures than was obtained for 100 % diesel fueling.

Conclusions

The following conclusions may be reached as a result of this work:

1. Natural gas dual-fueling yields similar power performance compared to baseline diesel operation near full load conditions. However, this may be at the expense of increased CO and NO_x concentrations.
2. The thermal efficiency (η_{th}) is adversely affected by the introduction of natural gas at light loads.
3. The dual-fuel brake specific fuel consumption (BSFC) is similar to that found for baseline diesel fueling, except for one-quarter load. The dual-fuel BSFC was also more sensitive to speed variation.
4. The total air/fuel equivalence ratio (ϕ) was enriched and the volumetric efficiency (η_v) was decreased by the aspiration of natural gas into the engine.
5. Decreased particulates, as indicated by significant reductions in the Bosch smoke number, and decreased sulfur dioxide emissions are possible with natural gas dual-fueling.

CHAPTER VI

RECOMMENDATIONS FOR FUTURE WORK

The objectives of this work have been successfully completed. The performance characteristics of a single-cylinder, prechamber-type diesel engine operated in dual-fuel mode with natural gas as the primary fuel have been investigated and compared to those obtained during baseline testing. However, there is more which could be done. The recommendations for future work are summarized below:

1. The primary focus of this research was the determination of performance characteristics. However, emissions are going to be of utmost concern due to the more stringent Environmental Protection Agency emissions regulations for diesel engines which will be coming into effect within the next five years. The logical progression of this work would be to continue this work, but concentrating on the emissions from this engine.

2. One of the concerns with natural gas fueled engines is the presence of unburned methane in the exhaust. The determination of the amount of unburned methane utilizing a gas chromatograph, which was recently acquired, would be another logical progression of this work.

3. The soluble organic fraction (SOF) has become the subject of much recent research. The recent completion of a dilution tunnel will allow the determination of this type of pollutant from internal combustion engines. The third recommendation of future work would be to utilize the dilution tunnel to investigate the effect of dual-fueling on the SOF of particulate emissions.

4. One parameter which was not investigated in this work was the cylinder pressure characteristics of the engine for baseline and dual-fuel modes. The cylinder pressure is a useful parameter on its own, but it is also useful in the determination of the fuel heat release rate of the engine. The preliminary development of an optical-based switching system which would enable direct correlation of cylinder pressure data with crank angle position has been completed. The fourth recommendation for this work is to complete the optical switching system and integrate it with a cylinder pressure transducer to obtain cylinder pressure and fuel heat release rate data.

5. The apparatus used to fix the amount of pilot fuel injected into the engine during dual-fuel testing was far from perfect. The present arrangement made repeatability of test runs with low percentages of diesel fueling very difficult. The fifth recommendation for continuing this work would be to either refine the present apparatus to allow for better control of the amount of diesel fuel being injected into the engine, or to devise a new method for fixing the amount of pilot being injected.

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APPENDIX A

PROGRAM LISTING OF

YAN2.BAS

```
'======================   YAN2.BAS   ================================'
' This is the data acquisition file written for the Yanmar TS180C   '
' using QuickBasic.                                                 '
'====================================================================='
'                    Written by Charles Ramsey                      '
'              Version 2.13     Dated 06/27/89   Time 11:00 am      '
'====================================================================='
'
'====================================================================='
'                 Channel Assignments for DAS-16                     '
'====================================================================='
'           0  =  Cylinder Pressure Transducer                       '
'           1  =  Needle Lift Sensor                                 '
'           2  =  RPM                                                '
'           3  =  Torque                                             '
'           4  =  Air Flow Transducer                                '
'           5  =                                                     '
'           6  =  CJC Compensator                                    '
'           7  =  EXP-16 Output Channel                              '
'====================================================================='
'
'====================================================================='
'                 Channel Assignments for EXP-16                     '
'====================================================================='
'           0  =  Exhaust Gas Thermocouple                           '
'           1  =  Coolant Water Thermocouple                         '
'           2  =  Intake Air Thermocouple                            '
'           3  =  SO2 Analyser                                       '
'           4  =  CO Analyser                                        '
'           5  =  NOX Analyser                                       '
'           6  =  CO2 Analyser                                       '
'           7  =  HC Analyser                                        '
'====================================================================='
'
'       Declare DAS16 QuickLibrary as an external subroutine.
'
                DECLARE SUB DAS16 (MODE%, BYVAL DIO%, FLAG%)
'
'       Dimension Data for use with DASH-16
'
                DIM DIO%(4), LT%(2), D%(8), D2%(8)
                COMMON SHARED DIO%(), LT%(), D%(), D2%()
'
'       Clear 40K for stack space
'
                CLEAR , , 40000
'
'       Define General Constants
'
        CONST PI = 3.14159, GAIN = 50
'
'       Other Constants.
'
                X = 1
```

```
                Z = 1
                TESTF$ = "C:\YANTSTXX.DAT"
                ETSTF$ = "C:\EYANXX.DAT"
'
'
INIT:
'
        COLOR 0, 7
        PRINT " INVERSE INPUTS ENTER WITH RETURN KEY "
        COLOR 7, 0
'
                PRINT " PLEASE INPUT ENGINE TEST NUMBER (two digits 01-99)"
                TST$ = INPUT$(2)
                PRINT USING " TEST NUMBER IS  \  \     CORRECT (Y/N)"; TST$
                Q1$ = INPUT$(1)
                IF (Q1$ = "N") OR (Q1$ = "n") THEN GOTO INIT
'
        COLOR 0, 7
                INPUT " USE Yanmar BASE Setup (Y/N)", Q2$
                IF (Q2$ = "Y") OR (Q2$ = "y") THEN GOTO BAS
                INPUT " USE Yanmar NATURAL GAS Setup (Y/N)", Q6$
                IF (Q6$ = "Y") OR (Q6$ = "y") THEN GOTO NAT
                INPUT "USE Yanmar COAL-WATER SLURRY Setup (Y/N)", Q7$
                IF (Q7$ = "Y") OR (Q7$ = "y") THEN GOTO CWS
'
TEST:
                PRINT " INPUT TEST NAME:   (8 Characters) "
                AMES$ = INPUT$(8)
                PRINT USING " TEST NAME IS  \        \     CORRECT (Y/N)"; AMES$
                Q3$ = INPUT$(1)
                IF (Q3$ = "N") OR (Q3$ = "n") THEN GOTO TEST
'
TYP:
                PRINT " INPUT ENGINE TYPE:   (13 Characters) "
                TYPE$ = INPUT$(13)
                PRINT USING " ENGINE TYPE IS  \             \     CORRECT (Y/N)"; TYPE$
                Q4$ = INPUT$(1)
                IF (Q4$ = "N") OR (Q4$ = "n") THEN GOTO TYP
'
FUEL:
                PRINT " INPUT FUEL NAME:   (17 Characters) "
                FUELL$ = INPUT$(17)
                PRINT USING " FUEL IS  \                \     CORRECT (Y/N)"; FUELL$
                Q5$ = INPUT$(1)
                IF (Q5$ = "N") OR (Q5$ = "n") THEN GOTO FUEL
'
        COLOR 0, 7
                INPUT " INPUT ENGINE BORE (in inches) ", BORE
                INPUT " INPUT ENGINE STROKE (in inches) ", STROKE
                INPUT " INPUT ENGINE CYCLE (2/4) ", CYCLE
                INPUT " INPUT COMPRESSION RATIO ", CR
                INPUT " INPUT ATMOSPHERIC PRESSURE (in psia) ", PR
                GOTO START
'
```

```
BAS:
'
'       Yanmar TS180C Base Data
'
                COLOR 7, 0
                AMES$ = "         "
                TYPE$ = "Yanmar TS180C"
                FUELL$ = "DIESEL     "
                BORE = 4.02
                STROKE = 4.17
                CR = 19.5
                CYCLE = 4
                PR = 14.7
                GOTO START
'
NAT:
'
'       Yanmar TS180C Natural Gas Data
'
                COLOR 7, 0
                AMES$ = "         "
                TYPE$ = "Yanmar TS180C"
                FUELL$ = "NATURAL GAS"
                BORE = 4.02
                STROKE = 4.17
                CR = 19.5
                CYCLE = 4
                PR = 14.7
                GOTO START
'
CWS:
'
'       Yanmar TS180C Coal Water Slurry Data
'
                COLOR 7, 0
                AMES$ = "         "
                TYPE$ = "Yanmar TS180C"
                FUELL$ = "COAL WATER SLURRY"
                BORE = 4.02
                STROKE = 4.17
                CR = 19.5
                CYCLE = 4
                PR = 14.7
                GOTO START
'
START:
                COLOR 0, 7
                INPUT " INPUT TOTAL TEST TIME (in minutes) :  ", TIME
                INPUT " INPUT SAMPLE RATE (in seconds)  :  ", SEC
                COLOR 7, 0
                TIME = TIME * 60
                R% = FIX(TIME / SEC)
'
'       Add test identification number to data files
```

```
'
              MID$(TESTF$, 10) = TST$
              MID$(ETSTF$, 8) = TST$
'
'       Open Data Files TESTF$ and ETSTF$
'
              OPEN TESTF$ FOR OUTPUT AS #1 LEN = 1000
              PRINT #1, "PASS   TIME   DYNO     ENGINE    TORQUE      HP   AIR FLOW  BMEP  VOLEFF
INTAKE AIR  COOLANT  EXHAUST", DATE$, TIME$
              CLOSE #1
              OPEN ETSTF$ FOR OUTPUT AS #2 LEN = 1000
              PRINT #2, "              PPM    PPM    PPM     %      PPM"
              PRINT #2, "PASS   TIME   SO2    CO    NOX    CO2    HC", DATE$, TIME$
              CLOSE #2
              COLOR 7, 0
              PRINT
              LOCATE 24, 1
              COLOR 0, 7
              PRINT "  -  -  -  PLEASE WAIT  -  -  -  "
              COLOR 7, 0
              LOCATE 25, 1
              PRINT "     Loading DASH 16 I/O address and thermocouple lookup data"
              OPEN "C:\DAS16.ADR" FOR INPUT AS #1
              INPUT #1, DIO%(0)       ' DAS16 base address
              CLOSE #1
              DIO%(1) = 2
              DIO%(2) = 1
              FLAG% = 0
'
'       Set MD%=0 to initialize DAS 16 system
'
              MODE% = 0
              CALL DAS16(MODE%, VARPTR(DIO%(0)), FLAG%)
              IF FLAG% <> 0 THEN GOTO CRASH
'
'---------------------------------------------------------------
' -  -  -  -  Table lookup data for K type thermocouple
'       This routine should only be run in the initialization
'       section of the program.  The DATA statements read in
'       are:    1) The Number of Data Points,
'               2) The Voltage Step Interval (mV),
'               3) The Starting Voltage (mV),
'               4) Temperatures in Degree C.
              DATA  309 ,  .2 , -6.6
              READ NK, SIK, SVK
'Temperature at -6.6mv, -6.4mV, -6.2mV etc.
              DATA -353.5,-249.3,-224.0,-207.6,-194.3,-182.8,-172.3,-162.8,-153.8,-145.4
              DATA -137.3,-129.6,-122.3,-115.2,-108.3,-101.6, -95.1, -88.7, -82.5, -76.4
              DATA  -70.4, -64.6, -58.8, -53.1, -47.5, -42.0, -36.6, -31.2, -25.9, -20.6
              DATA  -15.4, -10.2,  -5.1,  -0.0,   5.0,  10.1,  15.1,  20.0,  25.0,  29.9
              DATA   34.8,  39.7,  44.6,  49.5,  54.3,  59.1,  64.0,  68.8,  73.6,  78.4
              DATA   83.2,  88.0,  92.9,  97.7, 102.5, 107.4, 112.2, 117.1, 122.0, 126.9
              DATA  131.8, 136.7, 141.7, 146.6, 151.6, 156.5, 161.5, 166.5, 171.5, 176.5
              DATA  181.6, 186.6, 191.6, 196.6, 201.6, 206.6, 211.6, 216.6, 221.5, 226.5
```

```
                DATA  231.5, 236.4, 241.4, 246.3, 251.2, 256.1, 261.0, 265.9, 270.8, 275.6
                DATA  280.5, 285.3, 290.2, 295.0, 299.8, 304.6, 309.4, 314.3, 319.1, 323.9
                DATA  328.7, 333.4, 338.2, 343.0, 347.8, 352.6, 357.3, 362.1, 366.9, 371.6
                DATA  376.4, 381.1, 385.9, 390.6, 395.4, 400.1, 404.8, 409.6, 414.3, 419.0
                DATA  423.8, 428.5, 433.2, 437.9, 442.6, 447.3, 452.0, 456.8, 461.5, 466.2
                DATA  470.9, 475.6, 480.3, 485.0, 489.7, 494.4, 499.1, 503.8, 508.5, 513.1
                DATA  517.8, 522.5, 527.2, 531.9, 536.6, 541.3, 546.0, 550.7, 555.4, 560.0
                DATA  564.7, 569.4, 574.1, 578.8, 583.5, 588.2, 592.9, 597.6, 602.3, 607.0
                DATA  611.7, 616.4, 621.2, 625.9, 630.6, 635.3, 640.0, 644.8, 649.5, 654.2
                DATA  658.9, 663.7, 668.4, 673.2, 677.9, 682.7, 687.4, 692.2, 696.9, 701.7
                DATA  706.5, 711.3, 716.1, 720.8, 725.6, 730.4, 735.2, 740.0, 744.8, 749.7
                DATA  754.5, 759.3, 764.1, 769.0, 773.8, 778.7, 783.5, 788.4, 793.3, 798.1
                DATA  803.0, 807.9, 812.8, 817.7, 822.6, 827.5, 832.4, 837.3, 842.2, 847.2
                DATA  852.1, 857.1, 862.0, 867.0, 872.0, 876.9, 881.9, 886.9, 891.9, 896.9
                DATA  901.9, 906.9, 911.9, 916.9, 922.0, 927.0, 932.0, 937.1, 942.2, 947.2
                DATA  952.3, 957.4, 962.5, 967.6, 972.7, 977.8, 982.9, 988.0, 993.1, 998.2
                DATA 1003.4,1008.5,1013.7,1018.8,1024.0,1029.2,1034.4,1039.6,1044.8,1050.0
                DATA 1055.2,1060.4,1065.6,1070.8,1076.1,1081.3,1086.6,1091.9,1097.2,1102.4
                DATA 1107.7,1113.0,1118.3,1123.7,1129.0,1134.3,1139.7,1145.0,1150.4,1155.8
                DATA 1161.2,1166.6,1172.0,1177.4,1182.9,1188.3,1193.8,1199.2,1204.7,1210.2
                DATA 1215.7,1221.2,1226.8,1232.3,1237.9,1243.5,1249.1,1254.7,1260.3,1265.9
                DATA 1271.6,1277.3,1282.9,1288.6,1294.3,1300.1,1305.8,1311.5,1317.3,1323.1
                DATA 1328.9,1334.7,1340.5,1346.4,1352.2,1358.1,1363.9,1369.8,1375.7
'
                DIM TK(NK - 1)
'
                FOR I = 0 TO NK - 1
                        READ TK(I)
                NEXT I
'-----------------------------------------------------------------
                CLS
'
'       Dimension data for performance parameters
'
                DIM TEMP(R%, 3), AIR(R%), BMEP(R%), POWER(R%)
                DIM VOLEFF(R%), CJC(R%), RPM(R%), ERPM(R%), TORQUE(R%)
'
'       Dimension data for emissions parameters
'
                DIM SO2(R%), CO(R%), NOX(R%), CO2(R%), HC(R%)
'
'-----------------------------------------------------------------
'       Start of data collection process
'
                STIME = TIMER
                CLS
                ON TIMER(SEC) GOSUB DASH
LABEL1:
'
'       Turn Timer On
'
                TIMER ON
'
LOOP1:
```

```
                PTIME = TIMER - STIME
                HH = FIX(PTIME / 3600)
                TIME1 = PTIME - (HH * 3600)
                MM = FIX(TIME1 / 60)
                SS = FIX(TIME1 - (MM * 60))
'
'       Print Out Elapsed Time
'
                LOCATE 2, 45
                PRINT USING "ELAPSED TIME IS ##:##:##"; HH; MM; SS
'
'       If test loop number is less than the number of iterations
'       needed, repeat loop.  Else quit.
'
                IF X > R% THEN GOTO QUIT ELSE GOTO LOOP1
'
'       End of data collection process.
'-----------------------------------------------------------------
LABEL3:
'
'   Now determine which portion of the output screen is going to
'   be updated.
'
                IF Z = 0 THEN GOTO SCRN1
'
'-----------------------------------------------------------------
'       Update entire screen (SCRN: & SCRN1:)
'
SCRN:
                REM IF X=1 AND Z=1 THEN CLS
                LOCATE 2, 1
                COLOR 7, 0
                PRINT USING "        TEST NAME: \         \"; AMES$
                PRINT USING "        TEST NUMBER \    \"; TST$
                PRINT USING "        ENGINE TYPE : \              \          FUEL : \           \";
TYPE$; FUELL$
                PRINT USING "        CYCLE : ## cycle                        BORE : ###.## inches";
CYCLE; BORE
                PRINT USING "        ATM. PRESSURE : ##.### psia          STROKE : ###.## inches";
PR; STROKE
                PRINT USING "        COMPRESSION RATIO : ##.# to 1  "; CR
'
'       Update Performance Parameters Only (SCRN1:)
'
SCRN1:
                Z = 0
                LOCATE 9, 1
                      PRINT "  - - - - - - - - - Engine Performance Data  - - - - - - - -"
                      PRINT "           Dyno RPM    Engine RPM    Torque (ft-lbf)    Air (lbm/hr)"
                PRINT USING "           ####.###     ####.###        ###.###          ####.###";
RPM(X); ERPM(X); TORQUE(X); AIR(X)
                      PRINT "           Power (HP)     Volumetric Efficiency          BMEP (psi)"
                PRINT USING "             ##.#                #.###                    ###.###";
POWER(X); VOLEFF(X); BMEP(X)
```

```
                PRINT
                     PRINT "  - - - - - - - -       Emissions Data       - - - - - - - -"
                     PRINT "            SO2          CO          NOx          CO2          HC"
                     PRINT "            PPM          PPM          PPM          %           PPM"
                PRINT USING "           ###.##       ###.##       ###.##       ###.##       ###.##";
SO2(X), CO(X), NOX(X), CO2(X), HC(X)
                PRINT
                     PRINT "  - - - - - -  Thermocouple Temperatures in Degree F  - - - - - -"
                     PRINT "                 INTAKE           COOLANT           EXHAUST"
                PRINT USING "                  ###              ###               ####"; TEMP(X, 2);
TEMP(X, 1); TEMP(X, 0)
                PRINT USING " PASS = #####"; X
'---------------------------------------------------------------
'
'  Now store data in data files
'
                OPEN TESTF$ FOR APPEND AS #1 LEN = 1000
                PRINT #1, USING "### ##### ####.### ####.### ###.### ###.## ###.### ###.###
#.### ##### ##### #####"; X; PTIME; RPM(X); ERPM(X); TORQUE(X); POWER(X); AIR(X); BMEP(X);
VOLEFF(X); TEMP(X, 2); TEMP(X, 1); TEMP(X, 0)
                CLOSE #1
                OPEN ETSTF$ FOR APPEND AS #2 LEN = 1000
                PRINT #2, USING "### ####  ####.### ####.### ####.### ####.### ####.###"; X;
PTIME; SO2(X); CO(X); NOX(X); CO2(X); HC(X)
                CLOSE #2
                COLOR 7, 0
                LOCATE 2, 30
'
'       Increment counter X and return for another set of data.
'
                X = X + 1
                GOTO LABEL1
'---------------------------------------------------------------
'       An error has occured in the data acquisition process.
'       This section of the program tells which mode the error
'       occured, and prints out the error flag.
'---------------------------------------------------------------
CRASH:
                PRINT USING "ERROR IN MODE ## : FLAG% = ##"; MODE%; FLAG%
                END
'---------------------------------------------------------------
'       Data run is complete.  Program will now end.
'---------------------------------------------------------------
QUIT:
                LOCATE 24, 10
                COLOR 0, 7
                PRINT "ENGINE TEST IS COMPLETED"
                COLOR 7, 0
                END

DASH:
'---------------------------------------------------------------
'       This subroutine performs the data acquisition for the
'       engine test.
```

```
'
'
'       Get fifty values to average
'
              FOR FIVE = 1 TO 50
'-------------------------------------------------------------
'       Get data from thermocouples and emissions equipment attached to EXP16
'
                      GOSUB EXP16
'-------------------------------------------------------------
LABEL4:
                      MODE% = 1
'
'       Set STA-U Channels 2 - 5 for data acquisition.
'
                      LT%(0) = 2
                      LT%(1) = 5
                      CALL DAS16(MODE%, VARPTR(LT%(0)), FLAG%)
                      IF FLAG% <> 0 THEN GOTO CRASH
'
'       This section of the routine gathers RPM, TORQUE,
'       AIR FLOW, FUEL FLOW, and CJC Temperature Data.
'
                      FOR J% = 2 TO 5
                              MODE% = 3' Do 1 A/D Conversion
                              CALL DAS16(MODE%, VARPTR(DIO%(0)), FLAG%)
                              D%(J%) = DIO%(0)
                              IF FLAG% <> 0 THEN GOTO CRASH
                      NEXT J%
'
'       Lock DAS-16 To Channel #6 using Mode 1
'
                      LT%(0) = 6
                      LT%(1) = 6
                      MODE% = 1
                      CALL DAS16(MODE%, VARPTR(LT%(0)), FLAG%)
                      IF FLAG% <> 0 THEN GOTO CRASH
'
'       Get CJC Data from Channel #6 using Mode 3
'
                      MODE% = 3
                      DIO%(0) = 7      ' DAS-16 DATA RETURN VAR.
                      DIO%(1) = 6      ' DAS-16 DATA CONVERSION CHANNEL
                      CALL DAS16(MODE%, VARPTR(DIO%(0)), FLAG%)
                      IF FLAG% <> 0 THEN GOTO CRASH
                      D%(6) = DIO%(0)
'
'       Sum Digital Values to be averaged.
'
                      RPM(X) = D%(2) + RPM(X)
                      TORQUE(X) = D%(3) + TORQUE(X)
                      AIR(X) = D%(4) + AIR(X)
                      CJC(X) = D%(6) + CJC(X)
              NEXT FIVE
```

```
'
'       Average data and convert to analog voltage
'
                RPM(X) = (RPM(X) * 5) / 2048
                RPM(X) = RPM(X) / 50
                TORQUE(X) = (TORQUE(X) * 5) / 2048
                TORQUE(X) = TORQUE(X) / 50
                AIR(X) = (AIR(X) * 5) / 2048
                AIR(X) = AIR(X) / 50
                CJC(X) = (CJC(X) / 50) / 10
'
'       Average emissions data and convert to analog voltage
'
                SO2(X) = (SO2(X) * 5) / 2048
                SO2(X) = SO2(X) / (GAIN * 50)
                CO(X) = (CO(X) * 5) / 2048
                CO(X) = CO(X) / (GAIN * 50)
                CO2(X) = (CO2(X) * 5) / 2048
                CO2(X) = CO2(X) / (GAIN * 50)
                NOX(X) = (NOX(X) * 5) / 2048
                NOX(X) = NOX(X) / (GAIN * 50)
                HC(X) = (HC(X) * 5) / 2048
                HC(X) = HC(X) / (GAIN * 50)
'
'       Average Temperature data and convert to Degrees F.
'
                FOR K = 0 TO 2
                        V = (TEMP(X, K) * 5) / 2048
                        V = V / (GAIN * 50)
                        GOSUB INTERP
LABEL2:
'
'       Temperature has been determined.
'
                        TEMP(X, K) = TF
                NEXT K
'
'       Determine Performance and Emissions Parameters.
'
                GOSUB PARAM

EXP16:
'---------------------------------------------------------------
'       Get thermocouple data from EXP-16
'
'
'       STEP 1: Lock STA-U on the channel that the EXP-16
'               is connected to.
'
                        MODE% = 1
                        LT%(0) = 7
                        LT%(1) = 7
                        CALL DAS16(MODE%, VARPTR(LT%(0)), FLAG%)
                        IF FLAG% <> 0 THEN GOTO CRASH
```

```
'
'       STEP 2: Digital Outputs OP0 - OP3 drive the EXP-16
'               sub-multiplexer address, so use Mode 13 to
'               set up the sub-multiplexer channel.
'
                        FOR D1% = 0 TO 2
                                MODE% = 13
                                CALL DAS16(MODE%, VARPTR(D1%), FLAG%)
                                IF FLAG% <> 0 THEN GOTO CRASH
'
'       STEP 3: Gather data from the EXP-16 channel selected.
'
                                MODE% = 3
                                CALL DAS16(MODE%, VARPTR(D2%(D1%)), FLAG%)
                                IF FLAG% <> 0 THEN GOTO CRASH
                                TEMP(X, D1%) = D2%(D1%) + TEMP(X, D1%)
                        NEXT D1%
'
'       STEP 4: Gather data from emissions equipment.
'
                        FOR D1% = 3 TO 7
                                MODE% = 13
                                CALL DAS16(MODE%, VARPTR(D1%), FLAG%)
                                IF FLAG% <> 0 THEN GOTO CRASH
'
'       STEP 5: Gather data from the EXP-16 channel selected.
'
                                MODE% = 3
                                CALL DAS16(MODE%, VARPTR(DIO%(0)), FLAG%)
                                D2%(D1%) = DIO%(0)
                                IF FLAG% <> 0 THEN GOTO CRASH
                        NEXT D1%
                        SO2(X) = D2%(3) + SO2(X)
                        CO(X) = D2%(4) + CO(X)
                        CO2(X) = D2%(6) + CO2(X)
                        NOX(X) = D2%(5) + NOX(X)
                        HC(X) = D2%(7) + HC(X)
'
'       All done. Return to gather other engine data.
'
        RETURN LABEL4

INTERP:
'-----------------------------------------------------------------
'       This subroutine is an interpolation routine to determine
'       the temperature of K-type thermocouples in Degrees F.
'
'Entry variables:-
'    CJC = cold junction compensator temperature in deg. C.
'      V = thermocouple voltage in volts
'
'Exit variables:-
'     TC = temperature in degrees Centigrade
'     TF = temperature in degrees Fahrenheit
```

```
'
'       Determine thermocouple voltage in mV.
'
'Execution time on std. IBM P.C. = 46 milliseconds
'Perform CJC compensation for K type
                VK = 1000 * V + 1! + (CJC(X) - 25) * .0405'VK in mV
'
'       Find look up element
'
                EK = INT((VK - SVK) / SIK)
'       If Temp is below lower limit, round Temp to lower linit.
                IF EK < 0 THEN TC = TK(0): GOTO TF
'       If Temperature is above limit, round Temp to upper limit.
                IF EK > NK - 2 THEN TC = TK(NK - 1): GOTO TF
'
'       Do interpolation and determine temperature in Degrees C.
'
                TC = TK(EK) + (TK(EK + 1) - TK(EK)) * (VK - EK * SIK - SVK) / SIK'Centigrade
'
'       Convert from Degrees C. to Degrees F.
'
TF:
                TF = TC * 9 / 5 + 32  'Fahrenheit
        RETURN LABEL2

PARAM:
'---------------------------------------------------------------
'       This subroutine changes voltages to actual values.
'       These equations also calculate performance and emissions parameters.
'
'
'       The following equation converts from voltage to RPM.
'
                VRPM = RPM(X)
                RPM(X) = 25.6717 + (10014.9 * VRPM)
'
'       The following equation determines the engine RPM.
'
                ERPM(X) = 6.40213 + (5180.28 * VRPM)
'
'       The following equation converts from voltage to Torque (ft-lbf).
'
                TORQUE(X) = -.121357 + (103.442 * TORQUE(X))
                IF (TORQUE(X) < 0!) THEN TORQUE(X) = ABS(TORQUE(X)) - .46978
'
'       The following equation determines the power of the engine
'       in Horsepower.
'
                POWER(X) = RPM(X) * TORQUE(X) / 5252'hp
'
'       The following equation converts from voltage
'       to diff. press. (inches of Water).
'
                AIR(X) = (.364454 * AIR(X) ^ 2) - (4.79659 * AIR(X)) + 14.4122
```

```
'
'       The following equation converts from diff.
'       press. in inches of water to lbm/hr.
'
            AIR(X) = ((-3.58539 * AIR(X) ^ 4) + (45.6461 * AIR(X) ^ 3) - (220.322 * AIR(X) ^ 2)
+ (662.622 * AIR(X)) + 163.771) * .07408
'
'       This equation determines brake mean effective pressure (lbf/in^2).
'
            BMEP(X) = (CYCLE * .5 * POWER(X) * 12 * 33000) / (.25 * PI * BORE ^ 2 * STROKE *
ERPM(X))
'
'       This equation determines volumetric efficiency.
'
            VOLEFF(X) = (AIR(X) * CYCLE * 53.34 * (TEMP(X, 2) + 460) * 12) / (ERPM(X) * 2 * PI *
.25 * BORE ^ 2 * STROKE * PR * 60)
'
'       The following equations convert from voltage to emissions.
'
        SO2(X) = 116.2 * SO2(X) * 100
        CO(X) = 29.7 * CO(X) * 1000
        NOX(X) = 98.2 * NOX(X) * 100
        CO2(X) = ((.007891 + .082887 * CO2(X) + .001084 * CO2(X) ^ 2) * 1000) / 2
        HC(X) = 4761.9 * HC(X)
        RETURN LABEL3
```

APPENDIX B

PROGRAM LISTING OF

FUEL1.BAS

```
'------------------------------ FUEL.BAS ----------------------------'
'       Written by: Charles Ramsey     Date: 09/08/89 11:00 pm       '
'                             Version 2.11                           '
'--------------------------------------------------------------------'
'       The purpose of this program is to identify engine parameters  '
'       dealing with fuel consumption which are not already computed  '
'       in the main data acquisition program YAN2.BAS.  The parameters '
'       calculated in this program include:                           '
'               Brake Specific Fuel Consumption (BSFC),               '
'               Thermal Efficiency (THEFF),                           '
'         and   Equivalence Ratio (PHI)                               '
'--------------------------------------------------------------------'
'                                                                     '
'       This program will prompt the user for the engine test number, '
'       the number of fuel samples taken, the sample rate for the main '
'       test, the fuel consumption rate (sec/20 ml of fuel), and the  '
'       time when the fuel consumption rate was taken in the main test. '
'                                                                     '
'--------------------------------------------------------------------'
'
'       Set aside the data files for output data.
'
        OUTF$ = "C:\GRAPHER\WONG\THERMXX.DAT"
'
'       Initialization of the program.
'
INIT:
        PRINT "PLEASE INPUT ENINE TEST NUMBER (two digits : 01 - 99)"
        TST$ = INPUT$(2)
        PRINT USING "TEST NUMBER IS \    \   CORRECT (Y/N)"; TST$
        Q1$ = INPUT$(1)
        IF (Q1$ = "N") OR (Q1$ = "n") THEN GOTO INIT
'
'       Add test identification number to data files.
'
        MID$(OUTF$, 22) = TST$
'
'       Input the number of fuel consumption rates taken
'       during the main test and the sample rate of the main test.
'
        INPUT "INPUT NUMBER OF FUEL SAMPLES TAKEN : ", N%
        INPUT "INPUT SAMPLE RATE FOR TEST (in seconds) ", RATE
'
'       Dimension fuel related parameters.
'
        DIM BSFC(N%), THEFF(N%), FUEL(N%), ERPM(N%), HP(N%), TEMP(N%)
        DIM PHI(N%), FUEL1(N%), AIR(N%), METH(N%), METH1(N%)
'
        PRINT "Enter 'N' for Natural Gas Test, or"
        PRINT "Enter 'B' for Baseline Diesel Test"
        INPUT DECIS$
        IF (DECIS$ = "N") OR (DECIS$ = "n") THEN GOTO CH4
        OPEN OUTF$ FOR OUTPUT AS #1
```

```
        PRINT #1, " I     PASS   TIME   ENGINE       HP        FUEL      AIR      A/F ACT    PHI
BSFC       THEFF"
        CLOSE #1
'
'       Main loop of the file.  Here the fuel consumption rates
'       for the individual samples are input as well as the time
'       when it was taken.  The program will prompt the user for
'       the corresponding engine speed, horsepower, and air consumption.
'
        FOR I = 1 TO N%
                OPEN OUTF$ FOR APPEND AS #1
                PRINT USING "INPUT FUEL COMSUMPTION RATE NO. ##"; I
                INPUT FUEL(I)
                FUEL(I) = 1 / (FUEL(I) / 20)            'ml/s
                FUEL2 = FUEL(I) * 3.6                   'l/hr
                PRINT FUEL2
                FUEL1(I) = FUEL(I) * 3.6 * 1.50728      'lbm/hr
                INPUT "INPUT TEST TIME OF FUEL SAMPLE (in minutes) : ", TIME
                INPUT "INPUT TEST TIME OF FUEL SAMPLE (in seconds) : ", SEC
                TIME = (TIME * 60) + SEC
                R% = FIX(TIME / RATE)
                PRINT USING "INPUT ENGINE RPM FOR PASS ####"; R%
                INPUT ERPM(I)
                PRINT USING "INPUT HORSEPOWER FOR PASS ####"; R%
                INPUT HP(I)
                PRINT USING "INPUT AIR CONSUMPTION RATE FOR PASS ####"; R%
                INPUT AIR(I)
                BSFC(I) = (FUEL(I) * 6.83744) / HP(I)
                THEFF(I) = 2545 / (BSFC(I) * 18276.87)
                AFA = AIR(I) / FUEL1(I)
                AFS = 15.2
                PHI(I) = AFS / AFA
                PRINT #1, USING "####   ####   ####   ####.###   ##.####   ##.####   ###.####
###.###   #.####   ##.#####   #.####"; I; R%; TIME; ERPM(I); HP(I); FUEL1(I); AIR(I); AFA; PHI(I);
BSFC(I); THEFF(I)
                CLOSE #1
        NEXT I
CH4:
        OPEN OUTF$ FOR OUTPUT AS #1
        PRINT #1, OUTF$
        PRINT #1, " I     PASS   TIME   ENGINE       HP        FUEL      AIR      A/F ACT    PHI
BSFC       THEFF       PCT.PILOT       PCT.CH4"
        CLOSE #1
        FOR I = 1 TO N%
                OPEN OUTF$ FOR APPEND AS #1
                PRINT USING "ENTER NATURAL GAS CONSUMPTION RATE NO. ##"; I
                INPUT METH(I)
                METH(I) = 1.00586 + (2.39505 * METH(I)) + (.00292109# * METH(I) ^ 2)   'from air to
CH4
                METH(I) = METH(I) * (1000 / 60)                 'ml/s
                METH1(I) = METH(I) * .001 * 61.02375 * .000578 * 3600 * .0443 'lbm/hr
                PRINT METH1(I)
                METH2 = (METH1(I) / .0443) * 929     'BTU/hr
                INPUT "ENTER AMOUNT OF FUEL MEASURED : ", AMT
```

```
                PRINT USING "INPUT DIESEL FUEL CONSUMPTION RATE NO. ##"; I
                INPUT FUEL(I)
                FUEL(I) = 1 / (FUEL(I) / AMT)            'ml/s
                FUEL1(I) = FUEL(I) * .001 * 61.02375 * .000578 * 3600 * .876 * 62.4'lbm/hr
                PRINT FUEL1(I)
                FUEL2 = (FUEL1(I) / (.876 * 62.4)) * 999010'BTU/hr
                TOTF = METH1(I) + FUEL1(I)
                TOTE = METH2 + FUEL2     'Total BTU/hr
                PPLT = (FUEL2 / TOTE) * 100
                PCH4 = 100 - PPLT
                INPUT "INPUT TEST TIME OF FUEL SAMPLE (in minutes) : ", TIME
                INPUT "INPUT TEST TIME OF FUEL SAMPLE (in seconds) : ", SEC
                TIME = (TIME * 60) + SEC
                R% = FIX(TIME / RATE)
                PRINT USING "INPUT ENGINE RPM FOR PASS ####"; R%
                INPUT ERPM(I)
                PRINT USING "INPUT HORSEPOWER FOR PASS ####"; R%
                INPUT HP(I)
                PRINT USING "INPUT AIR CONSUMPTION RATE FOR PASS ####"; R%
                INPUT AIR(I)
                AFSM = 16.74
                AFSD = 14.5
                AFA = AIR(I) / TOTF
                AFS = (METH1(I) / TOTF) * AFSM + (FUEL1(I) / TOTF) * AFSD
                PHI(I) = AFS / AFA
                BSFC(I) = TOTF / HP(I)   'lbm/BHP-hr
                THF(I) = (HP(I) * 2545) / ((FUEL(I) * 18276.87) + (METH(I) * 20600))

                THEFF(I) = (HP(I) * 2545) / ((FUEL1(I) * 18276.87) + (METH1(I) * 20600))
                PCT(I) = ((THF(I) - THEFF(I)) / THF(I)) * 100

                PRINT #1, USING "####   ####   ####   ####.###   ##.####   ##.####   ###.####
###.###   #.####   ##.#####   #.####     ###.####     ###.####"; I; R%; TIME; ERPM(I); HP(I); TOTF;
AIR(I); AFA; PHI(I); BSFC(I); THEFF(I); PPLT; PCH4
                CLOSE #1
        NEXT I
```

APPENDIX C

DESCRIPTION OF DATA ACQUISITION SYSTEM AND RELATED INSTRUMENTATION

Temperature Measurement

Chromel-Alumel Thermocouples (Type K)

Differential Pressure Transducer

Manufacturer:	Omega
Model:	PX-163-005D
Range:	± 5 inches of H_2O differential
Linearity:	± 0.5 % Full Scale
Hysteresis and Repeatability:	± 0.25 % Full Scale

0.75 *Inch ASME Long Radius Nozzle*

Fabricated In-House

Dynamometer

Manufacturer:	Go-Power Corporation
Model:	D-100
Type of Absorption:	Waterbrake
Rotation Direction:	CW or CCW
Maximum Rated Horsepower Absorption:	100 HP @ 8000 RPM
Maximum Rated Torque Absorption:	65 ft-lb_f @ 5000 RPM
Speed Pickup:	60 tooth gear with inductive sensor
Torque Transducer:	Strain gage load cell

Dynamometer Instrumentation

Manufacturer:	Go-Power Corporation
Model:	C-17
Speed Accuracy:	±10 RPM from 100 RPM to 14,000 RPM
Torque Accuracy:	±0.3 ft-lb_f from 1 to 65 ft-lb_f

Natural Gas Rotameter

Manufacturer:	Omega
Model:	FL-224
Range:	1,000 to 36,000 l/min (Air)
Accuracy:	±5 % Full Scale
Repeatability:	±1 % Full Scale

Diesel Fuel Delivery System

1.	Main Fuel Supply		
	Buret:	Capacity -	1000 ml ±5 ml
2.	Testing Fuel Supply		
	Buret:	Capacity -	100 ml ±0.2ml

Data Acquisition System

1. Computer

Manufacturer:	Tussey Computer Products
Model:	Swan XT-10
CPU (Max. Rated Speed):	Intel 8088-1 (10 MHz)
Co-Processor:	Intel 8087-1

Data Acquisition System (Continued)

2. Analog to Digital Interface

Manufacturer:	MetraByte Corporation
Conversions/Sec.:	100,000 with direct memory access
Channels:	16 Single-Ended
Converter Type:	12 Bit successive approximation
Conversion Time:	8.5 μsec
Accuracy:	0.01 % of reading $\pm$1 bit

3. Screw Terminal Board

Manufacturer:	MetraByte Corporation
Model:	STA-U

4. Multiplexer

Manufacturer:	MetraByte Corporation
Model:	EXP-16
Nonlinearity:	0.006 %
Settling Time:	14 μsec

APPENDIX D

DESCRIPTION OF GAS-PHASE EMISSIONS EQUIPMENT

1. CO, CO_2, NO/NO_x Sample Gas Conditioner

Manufacturer:	Thermo-Electron Corporation
Model:	600

For particulate and condensate removal

2. CO Analyzer

Manufacturer:	Infrared Industries, Inc.
Model:	IR-703D
Range:	0 to 6000 ppm
Accuracy:	± 1 % Full Scale
Repeatability:	± 1 % Full Scale
Linearity:	± 1 % Full Scale
Type:	Non-Dispersive Infrared Analyzer

3. CO_2 Analyzer

Manufacturer:	Infrared Industries, Inc.
Model:	IR-703
Range:	0 to 20 % in air (by Volume)
Accuracy:	± 1 % Full Scale
Type:	Non-Dispersive Infrared Analyzer

4. SO_2 Analyzer

Manufacturer:	Thermo-Electron Corporation
Model:	40
Ranges:	0 to 5 ppm
	0 to 10 ppm
Range Multipliers:	x 10, 100, 1000
Accuracy:	$\pm$1 ppm
Type:	Pulsed Fluorescent Analyzer

5. NO/NO_x Analyzer

Manufacturer:	Thermo-Electron Corporation
Model:	44
Ranges:	0 to 2 ppm
	0 to 5 ppm
	0 to 10 ppm
Range Multipliers:	x 1, 10, 100, 10,000
Accuracy:	$\pm$1 % Full Scale
Linearity:	$\pm$1 % Full Scale
Type:	Chemilumiscent Analyzer

6. Smoke Meter

Manufacturer:	Robert Bosch GMBH
Model:	EFAW 65A (Sampler)
	EFAW 68A (Analyzer)