THE UNIVERSITY OF CALGARY

EXAMINATION OF THE EFFECTS OF PILOT FUEL QUALITY ON THE PERFORMANCE OF GAS FUELED DIESEL ENGINES

by

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ABSTRACT

The dual fuel engine is a diesel engine that operates on gaseous fuels while maintaining some diesel fuel injection to provide a deliberate source for ignition. Such a system attempts usually to minimize the use of the diesel fuel by its replacement with normally somehow cheaper gaseous fuels while maintaining satisfactory engine performance. There are some problems associated with the conversion of a conventional diesel engine to dual fuel operation. At light load, the dual fuel engine tends to exhibit inferior fuel utilization and power production deficiencies with higher unburnt gaseous fuel and carbon monoxide emissions, relative to the corresponding diesel performance. Operation at light load is also associated with a greater degree of cyclic variations in performance parameters, such as peak cylinder pressure, torque and ignition delay, which have narrowed the effective working range for dual fuel applications in the past.

A literature search revealed that much research has been expanded towards providing effective measures for the improvement of dual fuel engine operation at light load. However, very little information exists about the influence of the diesel fuel quality on dual fuel engine performance.

The present work is an experimental study of the effects of changes in the cetane number (CN) of liquid pilot fuels on the ignition delay period derived from pressure - time development records and of the exhaust emissions characteristics, with special reference to light load application.

Various pilot fuel quantities were employed with methane, propane and low heating value gaseous fuel mixtures, represented by methane-nitrogen and methane-carbon dioxide mixtures, over a wide range of engine load. The ignition delay variations with increased gaseous fuel admission showed a strong dependence on both the quantity and the quality of the pilot fuel used. It was found that the use of high cetane number (CN) pilot liquid fuels permitted smaller pilot quantities to be used satisfactorily. Engine operation with higher cetane number diesel fuel pilots on propane and low heating value gaseous fuel mixtures improved in comparison with the corresponding dual fuel engine operation employing common diesel fuels.

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CHAPTER ONE

INTRODUCTION

1.1 Background

In the last decade, energy efficiency, economic competitiveness, energy security and environmental protection have moved to the center of the international stage. Research in the field of the internal combustion engines has gained a new momentum. With the growth in number of diesel engine operated vehicles and with the relatively high rate of diesel fuel usage, diesel fuel engine emissions have been shown to be one of the major sources of current air pollution [54]. In many industrialized countries, legislation presently in effect attempts to limit the exhaust emissions of internal combustion engines operating according to the diesel principle.

Increased environmental awareness, stimulated fuel diversification and the high prices of conventional fuels created incentives for the promotion and further evaluation of alternative fuel sources and for the conversion of conventional internal combustion engines to alternative fuel operation. One example is the compression ignition engine of the dual fuel type which can be operated interchangeably on gaseous fuel while maintaining some diesel pilot injection to provide a deliberate source of ignition for the fuel gas-air charge, or on 100% diesel fuel.

1.2 Dual Fuel Engine

Historical Perspective and Applications

Early in the development of the diesel engine, the concept of operating on gaseous fuels with pilot oil ignition was recognized. Dr. Rudolf Diesel was granted a patent in 1898 for his method consisting of the injection of a liquid fuel to start the combustion of the working mixture of a gaseous fuel and air in the engine cylinder. However, the dual fuel commercial development only goes back to the late 1930's when a stationary dual fuel engine fuelled with town gas was produced and operated in England [44].

The interest in the application of the dual fuel engine principle rose in several countries during the Second World War due to the shortage of conventional liquid fuel supplies. Many conversions of conventional diesel engines for military or civil applications employed natural gas, coal gas and blast furnace gas. The chief components of these gases were hydrogen, carbon monoxide, methane, ethane, propane and other hydrocarbons, often in the presence of diluent gases such as nitrogen, carbon dioxide and water vapour [5].

In the United States, Worthington Pump and Machinery Co., Buffalo, N.Y., was the first to build and test the naturally aspirated dual fuel engine to operate with low pressure gas induction and pilot oil ignition in 1943. Reliable availability of natural gas, particularly in North America at the time, promoted research and development of high output dual fuel engines.

Presently, multicylinder large size, highly turbocharged engines, are manufactured that produce power outputs exceeding 7500 kW [36].

Diesel fuel ranks second after gasoline as a fuel for automotive propulsion and has been largely used for stationary applications. Dual fuel technology could make possible the widespread substitution of natural gas for diesel fuel, with the potential to reduce cost and emissions in a number of applications such as heavy-duty trucks, locomotives, marine vessels, industrial and farm equipment, power generation, etc.

The potential advantages offered by the dual fuel technology include diesel-like efficiency and power with much lower emissions of oxides of nitrogen (NO_x) and particulate matter. However, recent conversions of conventional diesel engines to dual fuel operation were affected by major increases in carbon monoxide (CO) and unburnt gaseous fuel emissions during operation at light loads, which is characteristic for many diesel engines applications, especially vehicular.

Another problem is the sensitivity of the combustion process to the type and composition of the gaseous fuel and the quality of the diesel fuel being used [24].

Natural gas is widely used in dual fuel engines and considered to be an alternative fuel for transportation that can offer cost, technological and environmental advantages [14,15,18,55] over gasoline and diesel fuels. Some examples of these advantages include the absence of evaporative emissions, lower carbon dioxide (CO_2) production and cold start CO emissions, high knock resistance and lower sulfur and particulate matter content. However, methane present in the exhaust gas is a powerful "greenhouse" gas that is considered to have a potential contribution to the global warming.

The cetane number (CN) is a measure of the ignition quality of diesel fuel based on the ignition delay in an engine. The higher the cetane number, the shorter the ignition delay and the better the ignition quality.

The quality of the diesel fuel available in different countries varies widely. For example, the cetane number is around 58 for common diesel fuels in Japan, 45 in Canada and the USA and approximately 50 in Europe. It is forecasted that with the depletion of good quality petroleum crudes, diesel fuel quality (CN and distillation properties) will deteriorate continuously in the future, which will greatly affect the performance of diesel engines and will conflict with environmental goals to lower pollution emissions [19,32,34].

It is generally agreed that exhaust emissions are affected by changes in fuel properties and in most studies it is admitted that the emission measurement results are device dependent [40,51]. Therefore, it is necessary to investigate the influence of the diesel fuel quality on the operational behaviour of the engines. This knowledge is needed to facilitate a better adaptation of fuels and engines towards satisfying fuel consumption, engine performance and exhaust emissions expectations.

1.3 Objectives of the Present Work

This work represents a part of a continuing research effort carried out over the years at the Mechanical Engineering Department to provide improved knowledge of the combustion phenomena in gas fueled internal combustion engines in general and the dual fuel engine in particular.

Much information is available in the literature regarding the relationship between diesel fuel properties and diesel engine performance. However, with regard to dual fuel applications, there appears to be only a very limited amount of information available in the open literature.

The present contribution describes the results of an experimental testing program set out mainly to examine the effects of the quality (especially the CN) and quantity of the pilot liquid fuel injection on the performance and the associated ignition processes of a single cylinder, direct injection dual fuel engine, with special reference to the possible improvement of light load performance. Engine operation with very lean mixtures was, perhaps for the first time, extended well into the partial motoring range, representing negative brake power output.

Of special interest was the analysis of the measured variations in the ignition delay period of the pilot diesel fuel under dual fuel conditions derived from pressure - time development records and of the observation of the corresponding changes in exhaust emissions characteristics under these conditions.

The research aim was to determine whether through the use of a pilot liquid fuel having a high CN, smaller pilot quantities can be employed equally well when operating on natural gas and the influence of such changes on the extent of exhaust emissions. Moreover, it was aimed to establish whether dual fuel engine performance at light load then can be improved further when operating on propane or low heating value gas mixtures which usually display longer ignition delay periods than those observed with methane.

CHAPTER TWO

LITERATURE SURVEY

2.1 Dual Fuel Engine Combustion and Ignition Delay

Much research effort has been expended so far towards providing effective measures to the further improvement of dual fuel operation of compression ignition engines [22,42,49]. The effects of changes in the initial charge temperature, oxygen and diluent concentrations, pilot liquid fuel quantity and injection characteristics on the length of the ignition delay period of the pilot, when a range of different gaseous fuels were admitted with the intake air, were investigated by Karim and his co-workers [8,25,26]. Their studies indicated for example that changes in the mean charge temperature and pressure at the end of compression resulting from the variations in the physical properties of the charge, preignition energy release due to chemical reactions, external heat transfer and effects of residual gas mixing are major factors that control the extent of variations in the ignition delay period, cyclic variations and exhaust emissions of dual fuel engines in general and at light load in particular. In a recent study, Belardini et al. [4] presented results of three dimensional computational correlation with the experimental measurements of pressure and heat release in a diesel engine using liquid fuels having different cetane number. The simple correlation used to compute the ignition delay period allowed a correct scaling of tests performed with fuels having different cetane number and with various engine settings.

2.2 Dual Fuel Engine Emissions

Many investigations regarding combustion phenomena and exhaust emissions in gas fuelled diesel engines have been conducted. Elliott and Davis [13], in an early study on a direct injection, naturally aspirated CFR engine, indicated that the amount of gas reacted depended on the gas-air mixture strength and on diesel quantity employed. They suggested on these bases that there is an apparent lower limit of flammability of the mixture. They also found that at light load, a part of the gaseous fuel was left to pass unburnt through the engine cylinder, probably due to the poor flame propagation. This resulted in poor fuel utilization, reduced power output and lower efficiency.

The operational problems due to the poor utilization of gaseous fuel at low loads have been widely recognized and have drawn considerable attention in dual fuel research over the years. The findings of Lewis [35], Felt et al.[16] and Karim [23] confirmed that with very lean gaseous fuel-air ratios, the flames initiated from the multitude of the ignition centres of the liquid diesel pilot spray will fail to propagate throughout the whole combustion chamber, leaving various amounts of the gaseous fuel-air charge unconverted.

The work of Azzouz [2] represents an experimental study of methane fuelled dual fuel engine based on measurements of exhaust emissions over an extended range of pilot injection timings, injection pressures and intake temperatures, carried out in a single cylinder direct injection engine of similar size and design to the one being employed here. He reported that the concentration of CO and unburnt CH_4 peaked around a mixture strength of an equivalence ratio of 0.4 to 0.5. Any increase or decrease in the gaseous fuel concentration could reduce the presence of CO and unreacted CH_4 in the exhaust gas.

Studies by Clark and Bunch [10], Karim and Rogers [30] and Karim and Klat [29] established that the ignition and combustion processes depended not only on the injection and ignition characteristics of the diesel pilot, but also depended strongly on the type and concentration of the gaseous fuel employed. Tesarek [49], who fumigated natural gas in a single cylinder, direct injection diesel engine and replaced up to 80% (on an energy basis) of the liquid fuel by gas, mainly to reduce particulate emissions, observed very large associated increases in unburnt hydrocarbons (HC) and CO emissions.

Bro and Pedersen [6], who investigated the use of methane and other alternative fuels (such as methanol, ethanol and ammonia) in dual fuel engines, found methane to be the most advantageous from an overall efficiency, power output and exhaust emissions point of view. However, when operating near stoichiometric fuel-air ratios, they recorded a large increase in NO_x emissions.

In a comparative study of methane and propane, Karim and Wierzba [31] attributed the increased HC and CO emissions found with propane fumigation at light load to the fact that the ignition delay is much longer during operation with propane than it is with methane. The admission of methane and propane produced lower concentrations of NO_x, particularly at light loads, in comparison to diesel operation.

Burn [7] extended the work of Karim and Khanna [27] to dual fuel engines to study the effect of cold intake mixture temperature on the combustion characteristics such as when using liquified natural gas as a fuel. He found that the lowering of the intake mixture temperature, even when the water jacket temperature was maintained constant, has a detrimental effect on the already poor combustion efficiency associated with very lean mixtures, due to the narrowing of the effective lower flammability limit of the gaseous fuel-air charge under the conditions at the end of compression. He also suggested that an increased pilot quantity has a two fold effect in enhancing the extent of combustion efficiency with lean mixture operation. Apart from widening the flammability limits of the charge due to the increased heat release following the pilot fuel ignition and combustion [26,28], the increased pilot quantity will spread the combustion zone to a larger volume of the gaseous fuel-air charge.

The work by Amoozegar [1] is also of interest to the present contribution because he examined the combustion process and engine performance when yet an additional auxiliary fuel (H_2), inert gas (N_2, CO_2) or a liquid (water, alcohols, gasoline, benzene, n-hexane) was introduced with the main intake charge. He reported that water or alcohol addition appeared to slow down the combustion rates. As a consequence, CO and unburnt CH_4 concentrations in the exhaust increased, while NO_x concentrations decreased. The carbon dioxide addition increased the exhaust CO concentration throughout the whole load range, while with the N_2 addition, the CO concentration decreased at high loads. An increase in the amount of inerts resulted in more unburnt gaseous fuel. He also found that the CO concentration decreased with an increase in the pilot size at low and moderate loads. With the addition of H_2 to the main gaseous fuel CH_4, the CO emissions experienced a marked decrease, while NO_x concentrations increased significantly.

Varde [52] investigated the use of propane and natural gas as alternative fuels. With relatively small pilots, a reduction in the exhaust smoke concentration (especially for natural gas operation) and NO_x at all engine loads was observed. For all of the pilot rates and load conditions, he found that the HC (mostly unburnt fuel) levels for the gaseous fuels in the exhaust gas were increased.

In a survey of the dual fuel engine performance at light load, Karim [22] concluded that any of the measures which increase the size of the combustion zone in the vicinity of the pilot or enhance the effective flammability limits of the charge at around the pilot fuel, could reduce significantly the unburnt gaseous fuel in the exhaust gas.

The experimental work by Tao [47] showed that NO_x emissions in a dual fuel engine fuelled with natural gas were lower than the values observed during full diesel operation of the engine. He suggested that NO_x emissions decrease mainly due to the lower combustion temperature and burning velocity when compared to diesel operation.

Liu [37] developed a multi-zone analytical model to predict the features of combustion processes in a dual fuel engine. His results were consistent with the findings of the previous experimental work and showed that for lean mixtures, most of the CO is produced from the partial oxidation of the gaseous fuel [39]. This was due to the low charge temperature and relatively slow reaction rates of the gaseous fuel in the reacting zone. For higher gaseous fuel concentrations in the mixture, no partial oxidation was evident in dual fuel engine operation and most of the CO was produced primarily from the preignition reactions of the gaseous fuel within the unburnt zone.

Most of the above mentioned investigations were performed on single cylinder, direct injection, normally aspirated research engines that provide better flexibility of control and economy of testing. Some examples of studies performed on larger industrial engines types are also available such as those of Ding and Hill [12] and Barbour et al. [3]. Ding and Hill used a four cylinder, turbocharged, prechamber, 7 litre diesel engine fuelled with natural gas to explore the combustion phenomena over a wide range of engine speed and load. It was found that the gas-air equivalence ratio is the key variable affecting both emissions and fuel economy. They reported very high unburnt hydrocarbons and CO concentrations at very light loads when using relatively small pilot diesel fuel input. They suggested that, especially at light loads, emission improvements may be realized by means of throttling (except NO_x emissions) and by advancing the injection timing. Barbour et al. [3] examined the performance and emission characteristics of a six cylinder, prechamber, 4.3 litre diesel engine fuelled with natural gas and propane. At full load, no important change in either engine performance or exhaust emissions was found with the introduction of gaseous fumigants. As load was decreased, gaseous fuel emissions (except NO_x) increased considerably. The percent of the unburnt gas was found to be independent of the amount of the particular gaseous fuel used. It was suggested that the main factor of influence concerning the unburnt fuel is the overall combustion temperature. However, it can be suggested that this can be viewed as the dominance of the pilot fuel combustion on the overall engine behaviour and emissions.

2.3 Diesel Fuel Properties and Exhaust Emissions

There is a growing evidence that the influence of diesel fuel composition on exhaust emissions is as important as the combustion parameters and engine design features [9]. Numerous attempts have been made to describe the emission behaviour of diesel fuels based on aromatic content and cetane number using a broad range of diesel engine types [41,45,53]. Because the diesel fuel properties are usually strongly interrelated, it is not uncommon to find conflicting results reported in the literature relative to which of the fuel properties actually bring about the reduction in emissions observed.

Henein [20] expressed the opinion that it is impossible to develop a single combustion parameter which would be able to rate the emissions behaviour of fuels in diesel engines of different designs. The Cetane Scale, which characterizes the autoignition behaviour (ignition delay) of a fuel, was not intended to rate any other combustion, performance or emissions parameter.

In a study to determine the influence of poor fuel quality on combustion and emissions in both a naturally aspirated and an externally supercharged single cylinder, air cooled DI diesel engine, Lepperhoff et al. [34] reported that, at low load, a change in combustion and a high increase in CO, HC and particle emissions were found with decreasing the cetane number of the fuel from 50 to 39. In the medium and high load ranges, the fuels examined have practically no influence on the emission characteristics.

In similar studies, Ullman et al. [50,51] concluded that the value of the fuel CN was the key to reducing HC and CO emissions. Both the CN and aromatics content of the fuel affected NO_x emissions. NO_x increased as the CN is decreased and as the aromatics content is decreased. However, they showed that there is no simple answer to how the fuel properties affect emissions because different engines used in the investigation did not show similar effects. Cowley et al. [11] also reported that the main controlling factor for emissions in diesel engines is the cetane number of the fuel, although, depending on the engine type, the density of the fuel can also have some effects. Similar trends have been observed by Gabele et al. [17] when recording exhaust emissions from a diesel passenger car. The fuels tested were a "high quality" fuel (CN 46.8 and low aromatic content) and a "low quality" fuel (CN 32.0 and a high aromatic content). Their results showed a decrease of up to 40% in HC, CO and NO_x when using the "high quality" diesel fuel.

Shell and Mercedes-Benz companies have joined efforts to investigate the effects of diesel fuel properties (density, distillation range, cetane number and aromatics content) on exhaust emissions in an advanced European indirect injection (IDI) passenger car and a modern commercial vehicle direct injection (DI) engine [33]. Their results indicated that the CN and not the total aromatics content accounted for the variation in NOx emissions. By increasing the CN, the NO_x emissions were reduced, particularly when raising CN from levels of 45 to 55. Above a CN 55 the reductions became rather small.

Contradicting results were found by Rantanen et al. [43]. Emissions were measured from four turbocharged direct injection engines chosen as representative types of existing heavy duty engines. The cetane number was found not to be important in reducing NO_x emissions.

A study was conducted by Tamanouchi and Akasaka [46] using a commercial direct injection (DI) diesel and a prototype diesel engine with fuel jet impingement (OSKA-DH). They reported that the decrease in CN caused an increase in NO_x and a decrease in particulate matters (PM) for the DI engine because of the long ignition delay. However, in the case of the OSKA-DH engine, a decrease in the cetane number seldom caused an increase in PM emissions. From the evaluation of oxygenates, it became clear that the oxygen in the fuel can reduce emissions for both engines with and without exhaust gas recirculation (EGR).

Accordingly, it is evident that although there is general information about the role of changes in the cetane number of the liquid fuel on diesel engine performance, conflicting findings between some of the investigators can be found. It can also be seen that the role of changes in the cetane number of the pilot fuel in dual fuel engine applications is clouded with even more uncertainty and lack of information.

CHAPTER THREE

EXPERIMENTAL SETUP

3.1 The Engine and Test Bed Instrumentation

The engine used to investigate dual fuel combustion in the present work was a single cylinder, four stroke, direct injection, naturally aspirated, water cooled, open chamber, research type Gardner diesel engine. It was capable of operating either with entirely diesel fuel or with gaseous fuel/diesel pilot fueling combination as a dual fuel engine.

Some of the main particulars of the engine are the following:

- Bore: 107.45 mm
- Stroke: 152.40 mm
- Displacement: 1395 cm^3
- Compression Ratio: 14.2
- Maximum Speed: 1600 rev / min
- Brake Power (Maximum): 11.93 kW at 1600 rev / min
- Injector: 4-holes: 13.3 MPa injection pressure
- Motoring Brake Mean Effective Pressure (BMEP): 30.78 kPa

The engine test bed is represented schematically in Fig. 3.1.1.

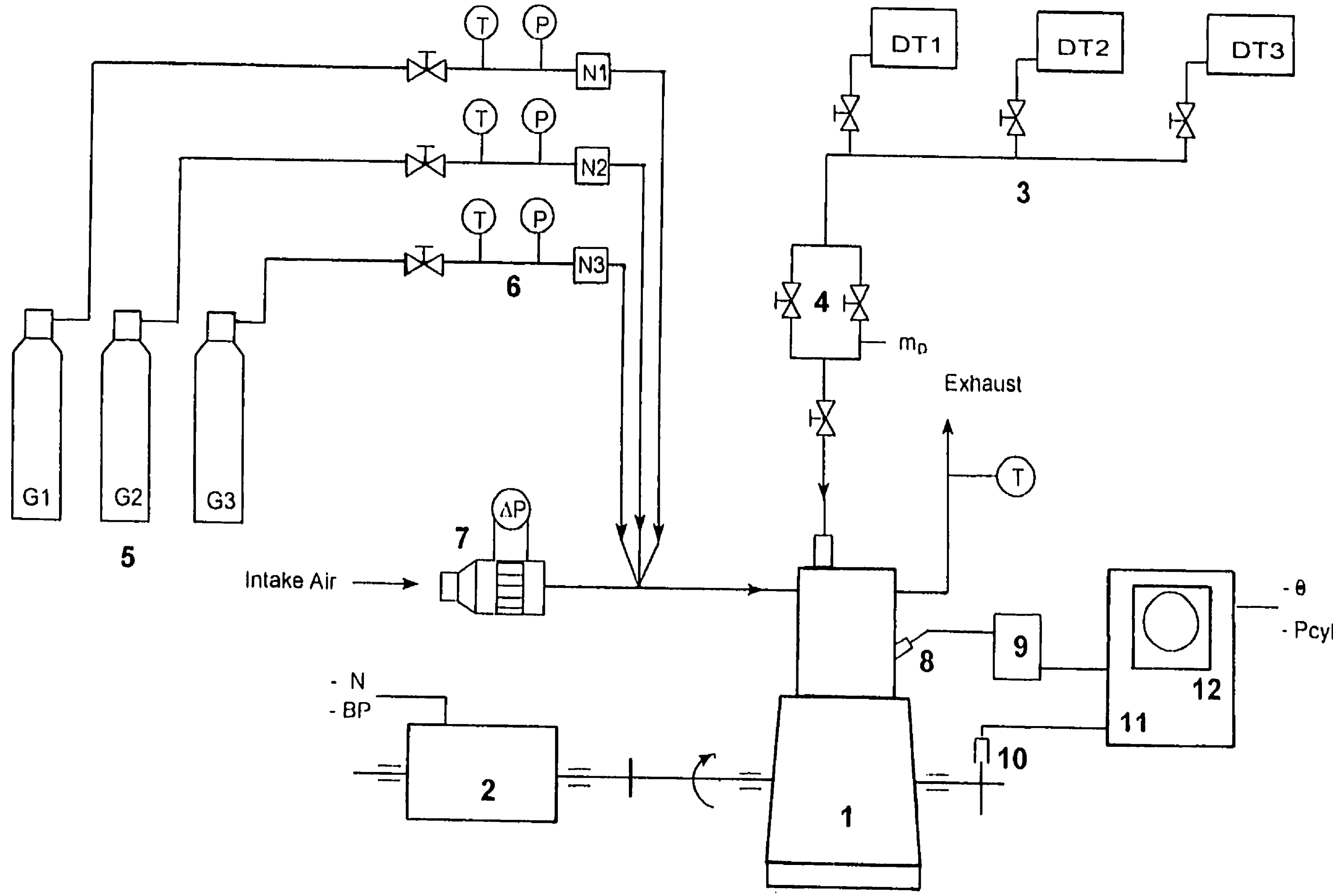


Fig. 1 Schematic Diagram of the Engine Test Bed

Legend for Figure 1

1	- Diesel engine
2	- Motor-generator dynamometer
3	- Diesel fuel supply
4	- Diesel fuel metering system
5	- Gaseous fuel supply
6	- Gaseous fuel metering system
7	- Air filter and viscous flow air meter
8	- Piezo-electric pressure transducer
9	- Charge amplifier
10	- Crank angle degree pick up
11	- Oscilloscope
12	- Oscilloscope camera
T	- Thermocouples

P, ΔP - Pressure gauges

DT1, DT2, DT3 - Diesel fuel tanks

G1, G2, G3 - Gaseous fuel bottles

N1, N2, N3 - Gaseous fuel nozzles

m_D	- Mass flow rate of diesel fuel
N	- Engine speed
BP	- Brake power
θ	- Crank angle

The engine was coupled directly to an electric swinging field dynamometer of the motor-generator type. By means of suitable switching arrangements from generating to motoring circuit (without interrupting engine operation), it was possible to operate the engine under partial motoring conditions, representing negative brake power output. The engine torque was transmitted to the framework of the dynamometer by the drag of the magnetic field and was indicated on a dial type spring balance attached to the load carrying arm of the dynamometer frame. This permitted the metering of the engine brake torque and hence through the knowledge of the engine rotational speed, the power output.

Adjusting the resistance in the armature of the dynamometer circuit through switches enabled to vary the engine load in steps. The engine power was absorbed by generating electrical energy in the armature circuit (of the dynamometer) and was dissipated through immersion heaters.

The capacity of the dynamometer load was found suitable for diesel operation, but was exceeded when engine power occasionally increased due to dual fueling. This imposed a limitation on the extent of gaseous fuel intake that could be employed. However, since the present investigation was centered around a load range well below the peak output, this set up was found to be adequate for the current investigation.

The engine speed was indicated on the engine control panel. In order to control and maintain the operating conditions constant, special engine coolant and lubricating systems were provided.

The engine closed coolant system included two heat exchangers of the shell and tube type. A water heater was provided to increase the coolant water temperature (up to the required value) during the engine warm-up. Cooling water circulation was maintained by an electrically driven pump. Throughout the entire engine operation the water jacket temperature was kept at $56° \pm 3°$ C by means of a thermostat.

The oil pressure through the engine lubricating system was maintained by an oil pump. The oil pressure values were displayed also on the control panel. Oil temperature was controlled thermostatically with an electric heater provided for heating if and when required.

Several thermocouples were installed at various locations and were used to measure the temperature of:

- the intake air
- the gaseous fuels at the metering points
- coolant water
- engine oil
- exhaust gas

The intake mixture temperature was measured using a shielded copper-constantan thermocouple installed in the engine manifold just outside the cylinder head. The measured values of temperature were displayed on a digital millivoltmeter mounted on the engine control panel.

The observation of cylinder events was limited to the recording of the temporal cylinder gas pressure variation, crank angle timing and diesel fuel injector needle lift which were considered essential for the examining of some of the main features of engine combustion phenomena. Variations of these parameters were displayed on a Tektronix type 502A dual beam oscilloscope screen. Permanent photographic records were obtained on 35 mm black and white Kodak film using a Cossor model 1428 Mk II B oscilloscope camera. The film could either be exposed to continuously record a few consecutive cycles or just a single one. This procedure though tends to be laborious was necessary so that a continuous record of a number of consecutive cycles can be obtained reliably in spite of the presence of cyclic variations, especially at light load with gaseous fuel applications.

To measure the cylinder gas pressure, a water cooled Kistler piezo-electric transducer, range 0-5000 psi, was located at the end of a short passage in the cylinder head leading to the combustion chamber. Such an arrangement tends to have an insignificant effect on the response of the pressure signal. The pressure transducer output was amplified in a Kistler dual mode charge amplifier before being displayed on the oscilloscope. The same oscilloscope beam could alternatively display the injector needle lift while the second beam was permanently used for crank angle display to provide a reference to the corresponding cylinder volume..

The injector needle lift which was measured by means of a magnetic pick-up transducer was used to provide information about the injection timing. The start of the injection was considered as the point of the first noticeable lifting movement of the injector needle.

The engine crank shaft rotation was considered as the time base for cylinder gas pressure and needle lift variations. Slots were cut 5 degrees apart as far as 30 degrees on either side of top dead center (TDC) position and 10 degrees apart on the remaining circumference of the circular plate rigidly mounted on the engine crank shaft. The TDC position was identified by a slightly larger slot. Light passing through these narrow slots triggered a photo transistor signal. The timing signal was further amplified and displayed on the oscilloscope screen as vertical marks, permitting the identification of the location of the cylinder events with respect to time.

3.2 Fuel Supply System

3.2.1 Liquid Fuel Supply

In the present investigation four commercial diesel fuels were used having the following cetane numbers: 41.5, 46.5, 53.0 and 58.0. Information on the diesel fuels is provided in Appendix E.

The diesel fuels were alternatively supplied to the engine injection pump from separate tanks, under gravity feed. Fuel consumption measurements were performed on a volumetric basis using two graduated glass burettes and converted to gravimetric basis by measuring the fuel density. The burettes could be filled just prior to metering from the tanks through a set of three-way valves.

A constant pressure in the fuel line was necessary to be maintained to ensure the accurate checking of the pilot quantity required. Accordingly, the experiments were carried out with the burettes supplying the fuel under a constant head. The pilot fuel quantity could be varied directly onto the fuel pump by adjusting the injector pump plunger stroke against a rack. In the present study, in keeping with normal diesel engine practice and to reduce the number of testing variables, the injection timing of the pilot fuel was kept constant.

3.2.2 Gaseous Fuel Supply

The gaseous fuels (natural gas and propane) and the inert gases (N_2, CO_2) employed in the present experimental work were commercially pure. The composition of a typical sample of natural gas is presented in Appendix E.

The gases were introduced to the engine intake manifold through a mixing venturi, just upstream of the cylinder inlet valve.

The gaseous fuels and inert gases were supplied from the high pressure storage bottles through regulators to a metering system employing precalibrated choked nozzles. These permitted the mass flow rate of the gas to be metered merely by taking readings of the upstream pressure and temperature since under chocked conditions, the gas flow is independent of the downstream pressure and unaffected by the intake manifold pressure pulsations. The calibration of the chocked nozzles and the underlying theory is presented briefly in Appendix A.

A regulating valve together with coarse and fine control valves were fitted upstream of the chocked nozzles. Two chocked nozzles, which were used to measure the low and high flow rates of the gaseous fuel, were connected in parallel such that either one or both nozzles could be used. A third chocked nozzle was reserved for the metering of inert gases when used.

3.3 Air Intake System

The air flow to the engine was metered by passing the intake air through a precalibrated viscous flow air meter which had the advantage of not being adversely affected by the pulsating flow, nor by the down stream addition of a gaseous fuel. The meter consisted of an air filter element and a flame-trap matrix of a very large number of narrow triangular passages whose total inside surface area exposed to the flow was so chosen that the air flow through these passages to the engine was laminar.

The pressure differential created across the matrix was therefore directly proportional to the air flow velocity and was recorded using an inclined liquid manometer mounted on the engine control panel.

The viscous flow meter was calibrated against a standard orifice plate using a large capacity tank interposed between the orifice meter and the engine so as to minimize the amplitude of the air pulsation at the orifice. The calibration of the meter and the calculation of the air mass flow rate entering the engine are included in Appendix B.

3.4 Exhaust Emissions Analysis System

Gas samples were drawn while the engine was operating steadily under a specified set of operating conditions. These samples were taken from a point on the exhaust manifold just beyond the exhaust valve using a multi-holed water cooled sampling tube and introduced in a sampler / control unit (module). The samples were cooled and passed through a particulate filter, a water drop-out collector drier section before being admitted to a series of analyzers. These were mainly of the non-dispersive infrared type. Calibration of the analyzers was carried out twice a day using standard calibrating gases. The analyzers were allowed to reach a thermal stability before use. Beckman Model 864 infrared analyzers were employed to detect CH_4, CO and CO_2 while a Beckman model 755 paramagnetic type analyzer was used for O_2 metering. No nitrogen oxide analyzer was employed during this experimental work since no accurate analyzer was available at the time and due to the fact that the investigation was confined to light load regions were it has been shown that extremely low concentrations of NO_x are produced, particularly when gaseous fuels are employed.

Appendix D illustrates the procedure adopted for calculating the percent of intake methane unreacted when using the direct readings of the CH_4, CO, CO_2 and O_2 concentration in dry exhaust.

3.5 Experimental Procedure

3.5.1. Preliminary Work

Before conducting the actual engines tests, the air, fuel and pressure measuring devices were calibrated. Based on initial trials, the following reference engine operating parameters were established for all subsequent diesel and dual fuel tests:

Engine speed: 1000 ± 10 rev/min

Coolant water temperature: 56 ± 3° C

Engine oil temperature: 47 ± 3° C

Injection timing: 18° BTDC

Diesel pilot quantity: 0.4; 0.3; 0.2 kg/h

The atmospheric pressure in the engine room ranged from 82.16 kPa to 89.7 kPa and the atmospheric temperature varied between 16° and 23° C. To account for these daily variations, appropriate corrections were made in all calculations.

In order to ensure the consistency of the experimental observations and to establish a basis for comparison of the results, preliminary performance tests were carried out with the engine operating on diesel fuel only.

After warming up the engine and reaching stable operating conditions (i.e. running the engine on diesel fuel for about one hour on half load), experimental data regarding fuel and air consumption, exhaust temperature, brake power and cylinder pressure-time events were collected, for every step in engine loading. Plots of ignition delay period, power output, exhaust temperature, average maximum cylinder pressure, specific fuel consumption were then made, based on the dimensionless parameter "equivalence ratio" which is defined in Appendix C.

Before testing any other diesel fuel with a different CN, the fuel lines were purged and the fuel filters were changed. The fuel supply line was later connected to the tank and the next diesel fuel to be tested was introduced in the system. The engine was then run for a sufficient period of time in order to ensure that the last amount of the previously used fuel which could possibly still remain in the system was consumed.

3.5.2. Experimental Procedure for Dual Fuel Tests

The examination of dual fuel operation was carried out using pilot quantities of 0.40 kg/h, 0.30 kg/h and 0.20 kg/h for each of the four different CN diesel fuels. Pilot quantities less than 0.20 kg/h were found to be unsuitable for steady engine operation. Commercial methane was used extensively as the gaseous fuel. The investigation was then extended to include commercial propane as well as low heating value gaseous fuels represented by CH_4-N_2 and CH_4-CO_2 mixtures.

Before introducing any gaseous fuel, the engine was first run on diesel fuel only for about one hour on half load. After reaching stable operating conditions, the gaseous fuel supply valves were opened slowly and the diesel fueling rate was adjusted to keep the engine speed at 1000 rev/min. When the desired diesel pilot quantity was reached, the fuel pump rack was locked into its position. To stabilize the engine operation, the engine was then run for another fifteen minutes. For any change in the engine loading, the gaseous fuel supply was adjusted to keep the engine running at constant speed. Each experiment was then extended to partial motoring conditions by reducing the amount of the gaseous fuel while keeping constant the diesel pilot quantity. The motoring speed of 1000 rev/min was maintained constant by means of the dynamometer controls.

Readings were taken for fuel and air consumption, exhaust temperature, brake load, upstream pressure and temperature of the gaseous fuel, exhaust emissions and cylinder pressure-time events were recorded on photographic film. From these observations, calculations were made for the total equivalence ratio, brake power, specific energy consumption, etc. The photographic films on which cylinder charge pressure variations with time were recorded, were developed in the photo lab and analyzed to obtain information on the ignition delay period and average maximum cycle pressure. The average values obtained from several consecutive cycles were used. This analysis was very laborious and time consuming.

CHAPTER FOUR

EXPERIMENTAL RESULTS AND DISCUSSIONS

4.1 Overall Engine Performance as a Dual Fuel Engine

In the present work, engine performance with constant injection timing and engine speed was observed mainly in terms of brake power output, specific energy consumption, average cylinder peak pressure and exhaust temperature. The variations of the ignition delay period and exhaust emissions will be discussed in the next sections of Chapter 4.

As shown in Fig. 4.1.1, the increase in the cetane number of the diesel fuel when operating the engine on the diesel mode did not bring about a significant change in the engine brake power output. The effect is sufficiently small, however to fall within the range of possible experimental errors. Fig. 4.1.2 represents the variations of brake specific energy consumption with total equivalence ratio when operating with different cetane number diesel fuels. No changes were found in the values of the exhaust temperature, as shown in Fig. 4.1.3.

Operation with higher cetane numbers generally resulted in lower cycle pressures due to the reduced ignition delay periods.

The engine was operated in the dual fuel mode on commercial methane with different amounts of diesel pilot. Fig. 4.1.4 shows typically the brake power output

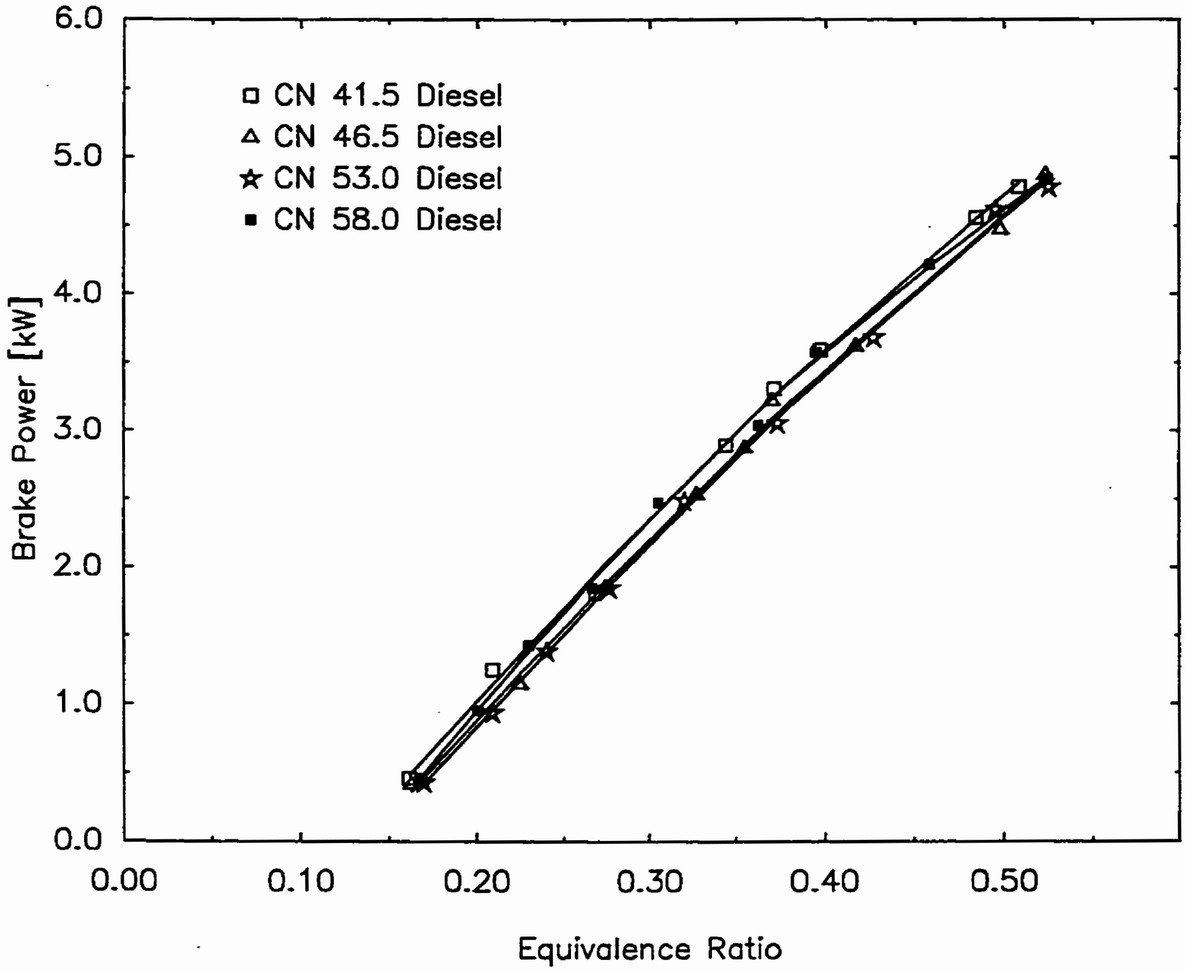


Figure 4.1.1. Variations of Brake Power with Equivalence Ratio for Diesel Operation when Using Different Cetane Number Fuels

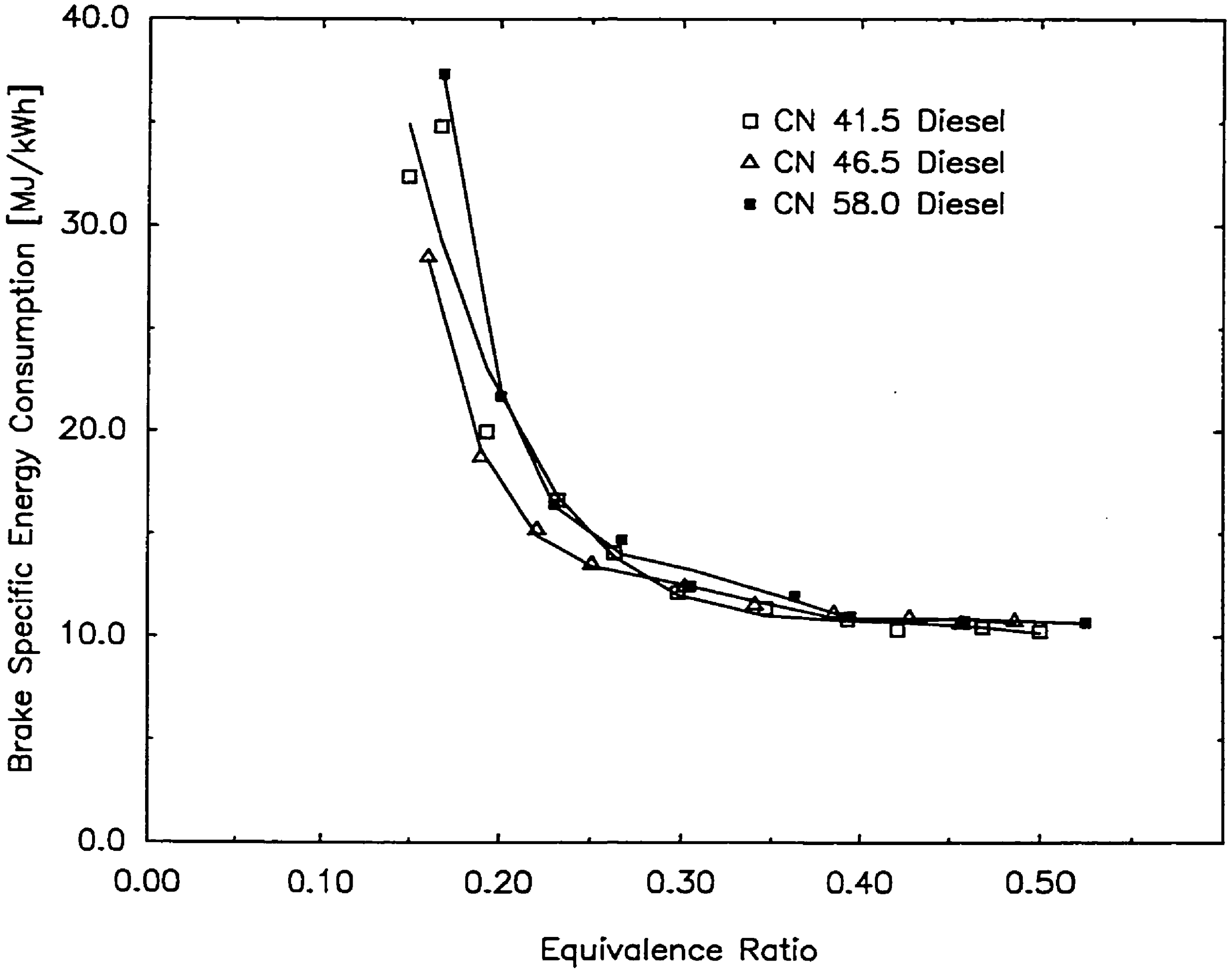


Figure 4.1.2. Variations of Brake Specific Energy Consumption with Equivalence Ratio for Diesel Operation when Using Different Cetane Number Fuels

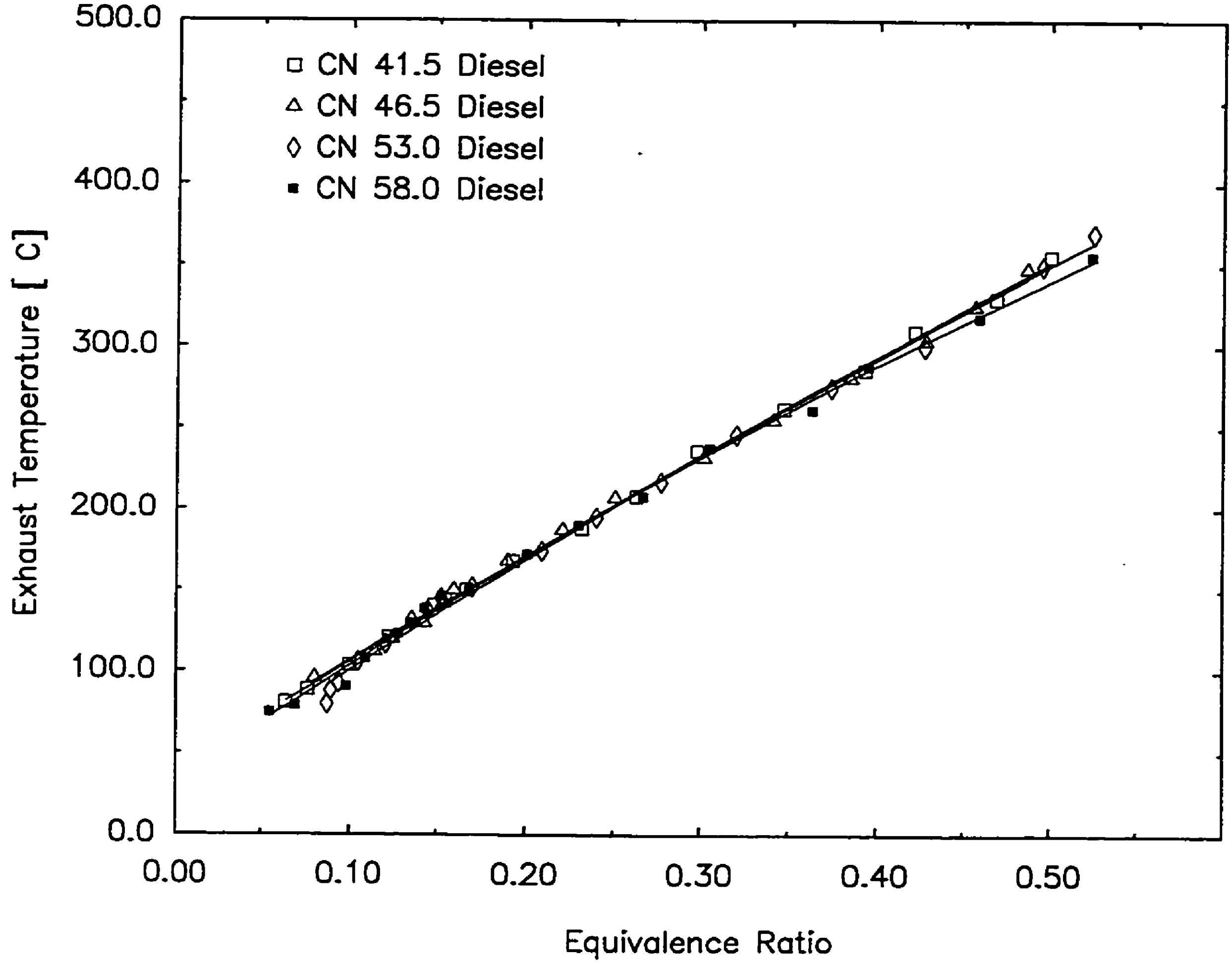


Figure 4.1.3. Variations of Exhaust Temperature with Equivalence Ratio for Diesel Operation when Using Different Cetane Number Fuels

variation with total equivalence ratio.

It can be seen that at light loads dual fuel operation is inferior to that of the normal diesel, producing less power output. However, as the amount of gaseous fuel is increased, more power is produced, even surpassing the corresponding values obtained with normal diesel operation. As the size of the pilot is reduced, there is a limiting minimum power output which is associated with richer overall fuel-air ratios. This limit corresponds with the effective limit for flame propagation throughout the gaseous fuel-air mixture. Variations of the corresponding brake specific energy consumption with total equivalence ratio when operating on methane with different cetane number diesel fuels and different pilot quantities are represented in Fig. 4.1.5. It can be seen that dual fuel operation at light load is less efficient than the corresponding diesel operation, irrespective of what diesel fuel is used. The efficiency of dual fuel operation improves beyond half load and surpasses that of the diesel.

A typical variation of the cylinder charge pressure with time for diesel operation with CN 58 diesel fuel is shown in Fig 4.1.6. Analysis of the recorded pressure-time diagrams showed that the maximum average cylinder pressure is increased with gaseous fuel addition for dual fuel operation, as shown typically in Fig.4.1.7. The corresponding rates of pressure rise are much lower, contributing towards quieter engine running than that with diesel fuel only. As compared to diesel operation, dual fuel operation displayed a considerable decrease in maximum cycle pressure, except for high load.

Variations of cylinder charge pressure with time when operating with smaller pilot amounts are represented in Figs 4.1.8 and 4.1.9.

Typical variations of the maximum average cycle pressure under dual fuel operating conditions are shown in Fig. 4.1.10. The decrease in the pilot fuel quantity lowered the average maximum cycle pressure; this effect was greater in the low and moderate load regions.

A typical variation of the exhaust temperature with total equivalence ratio shown in Fig. 4.1.11 indicates that for the lean mixtures used, higher exhaust temperatures are produced when larger pilots and increasingly higher equivalence ratios are employed, both a reflection of the increase in the proportion of gaseous fuel undergoing combustion and the associated increased energy release rates that continue further down the expansion stroke. Due to the increased delay periods and late combustion under dual fuel operation, higher temperatures at the end of the expansion stroke are encountered than the corresponding values for normal diesel operation.

Engine performance is influenced also by the type of the gaseous fuel used. The admission of propane brought an increase in the engine brake power output as compared to methane operation, as shown in Fig.4.1.12.

The average maximum cycle pressures and brake power showed appreciable reductions as low heating value gas fuel mixtures, represented typically by a mixture of 63% by volume methane and 37% by volume nitrogen (i.e. 50% by mass methane), simulating the behavior of a natural gas containing significant amounts of diluents, were introduced into the engine cylinder. Similar gaseous mixtures of

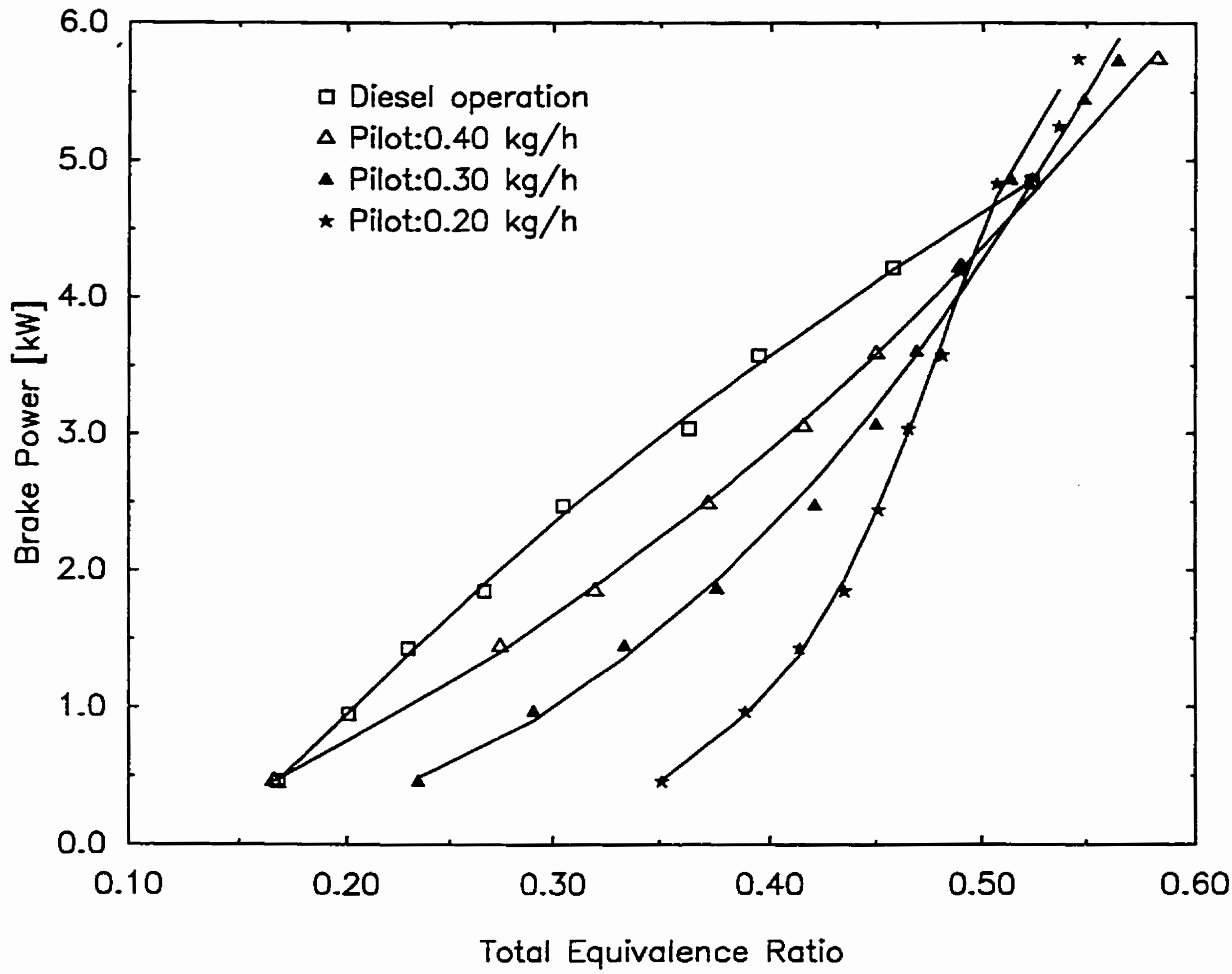


Figure 4.1.4. Variations of Brake Power with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using a 58.0 Cetane Number Diesel Fuel and Different Pilot Quantities. (ϕ_D=0.166 for 0.40 kg/h Pilot; ϕ_D=0.123 for 0.30 kg/h Pilot; ϕ_D=0.081 for 0.20 kg/h Pilot)

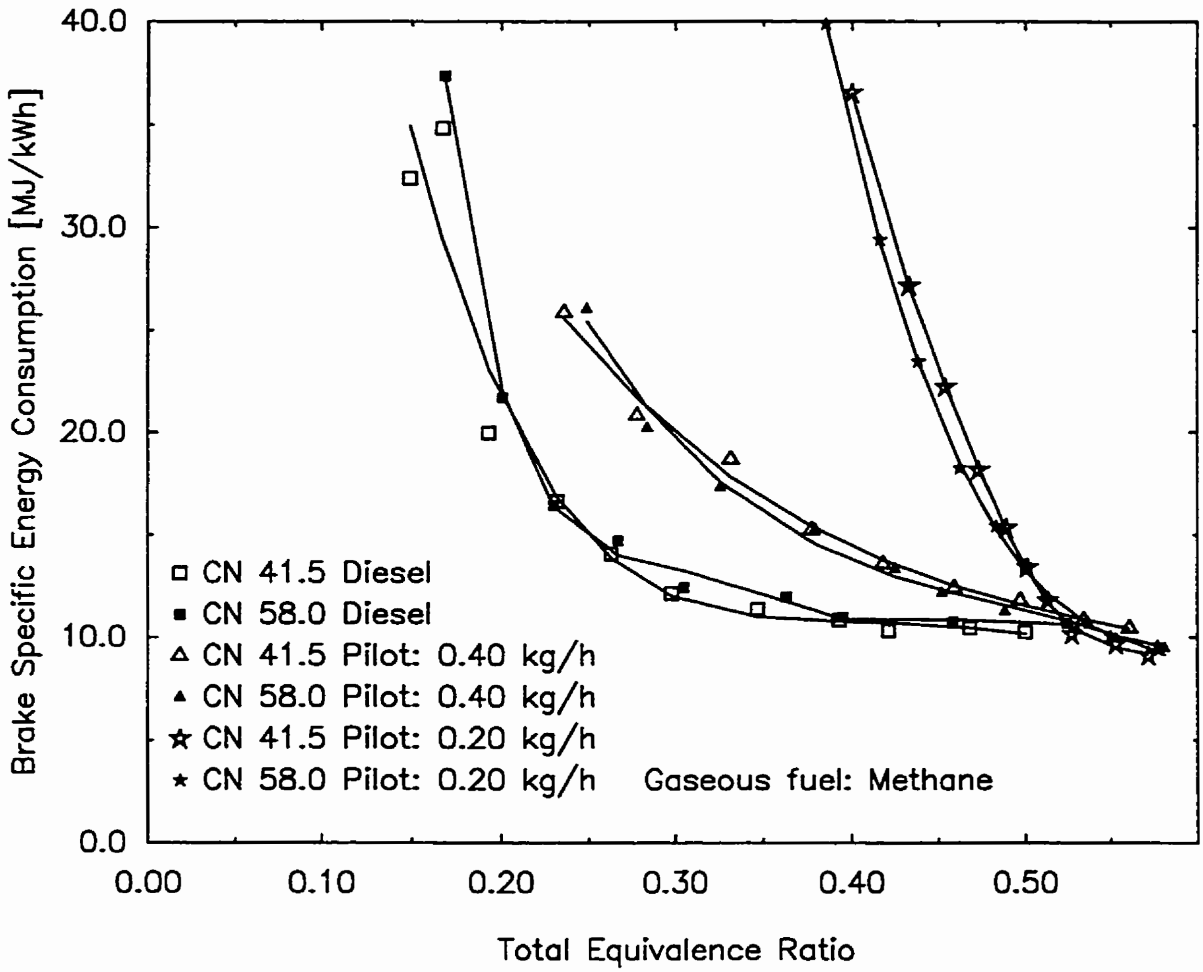
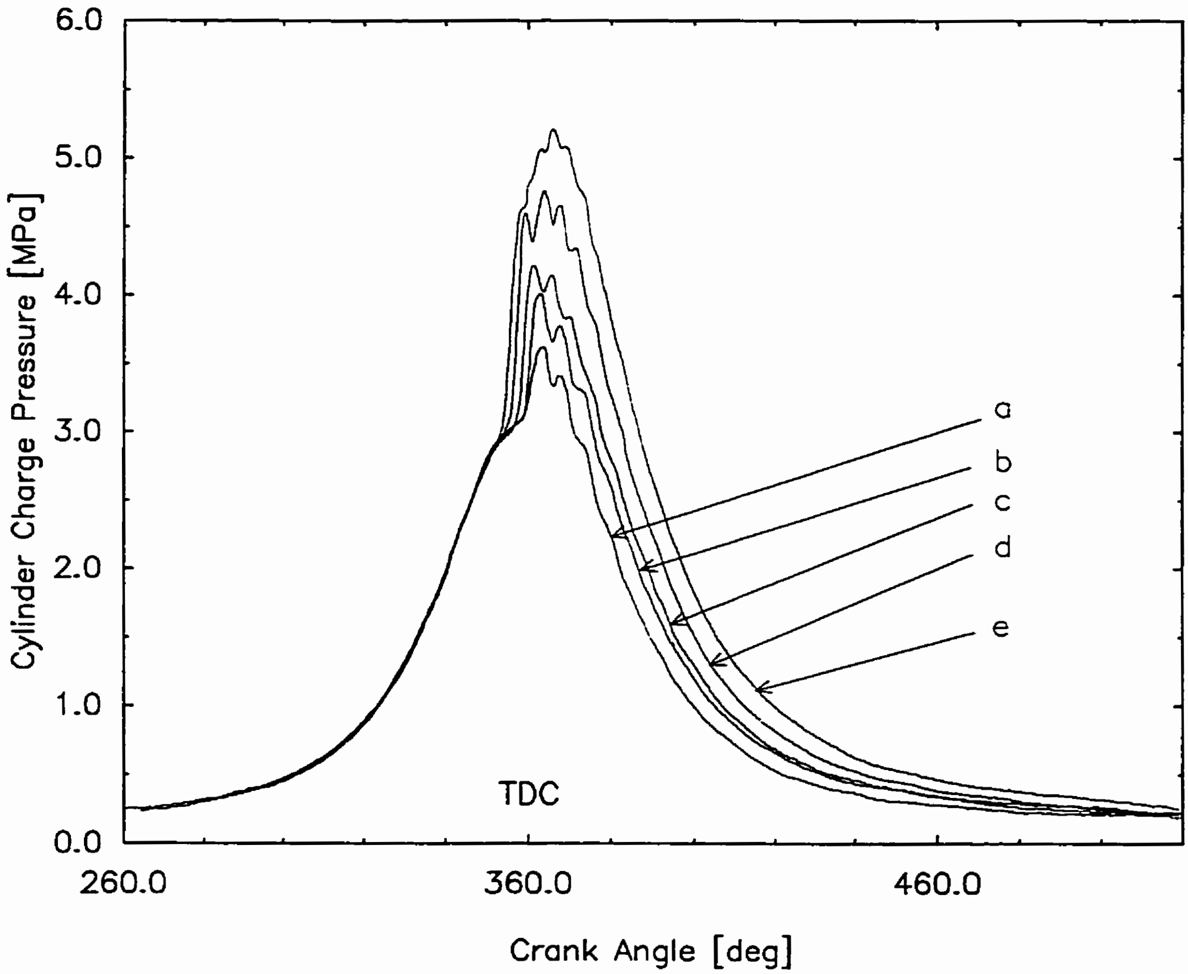


Figure 4.1.5. Variations of Brake Specific Energy Consumption with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Diesel Fuels and Different Pilot Quantities. (ϕ_D=0.166 for 0.40 kg/h Pilot; ϕ_D=0.081 for 0.20 kg/h Pilot)

methane and carbon dioxide were also used. Generally, it was observed that for a given inert gas concentration and pilot quantity, lower power outputs and cycle pressures are produced with CO2 than with N2 addition.

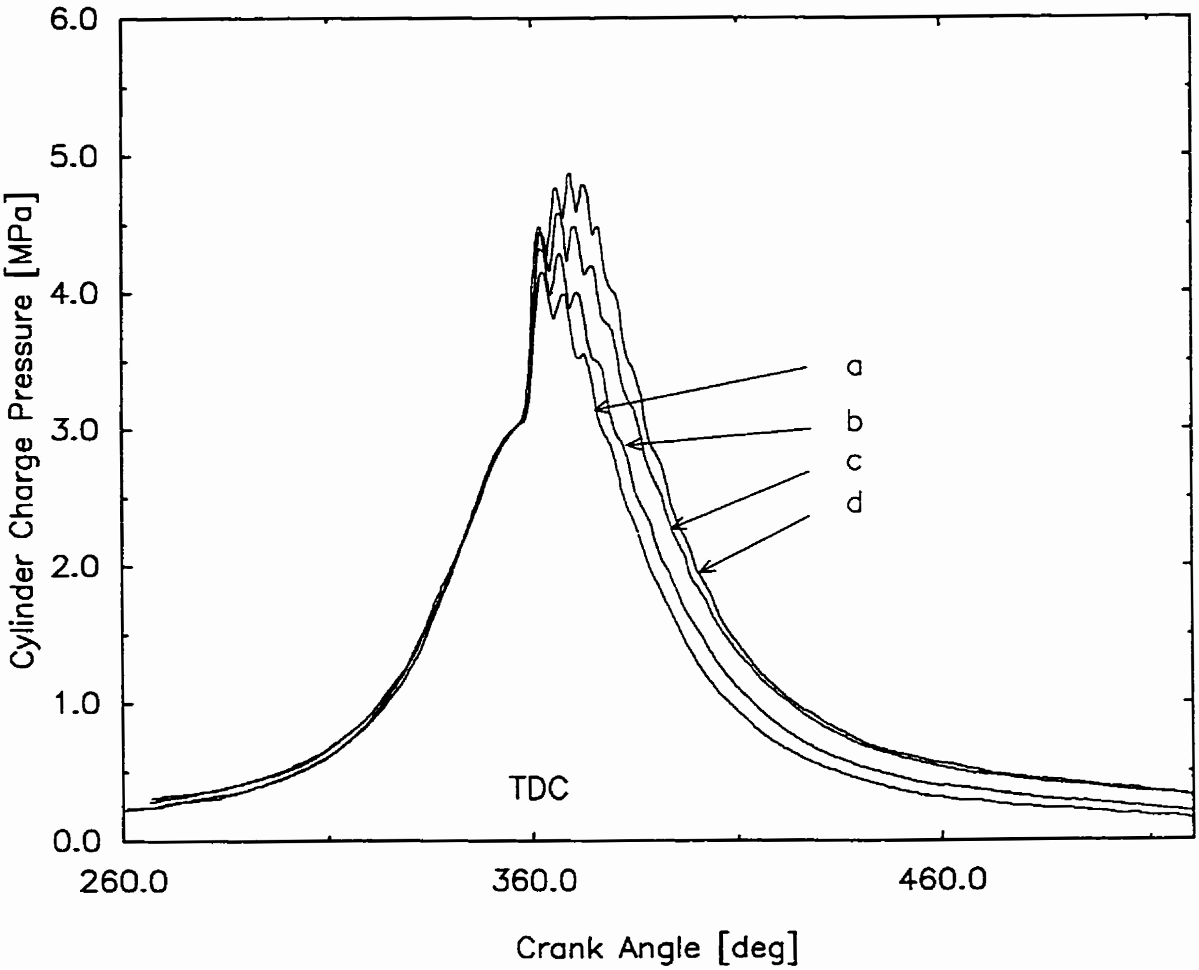
Fig. 4.1.13 represents a comparison of the brake power variation when using a 0.20 kg/h pilot with various gaseous fuel mixtures.

The increase in the cetane number of the pilot when operating the engine on methane has an insignificant effect on the engine power output, as shown in Fig. 4.1.14, for a 0.40 kg/h pilot and in Fig. 4.1.15, for a 0.20 kg/h pilot. A similar trend was observed for the exhaust temperature values, as indicated in Figs. 4.1.16 and 4.1.17. The maximum average cycle pressure values showed appreciable reductions when the cetane number of the pilot was increased. This was particularly the case at higher loads, as shown in Fig. 4.1.18. The operation of the engine on propane when using a higher cetane number pilot displayed no significant variation in the engine power output, as represented in Fig. 4.1.12. The variations of brake specific energy consumption with total equivalence ratio when operating on propane with different cetane number diesel fuels are shown in Fig. 4.1.19.



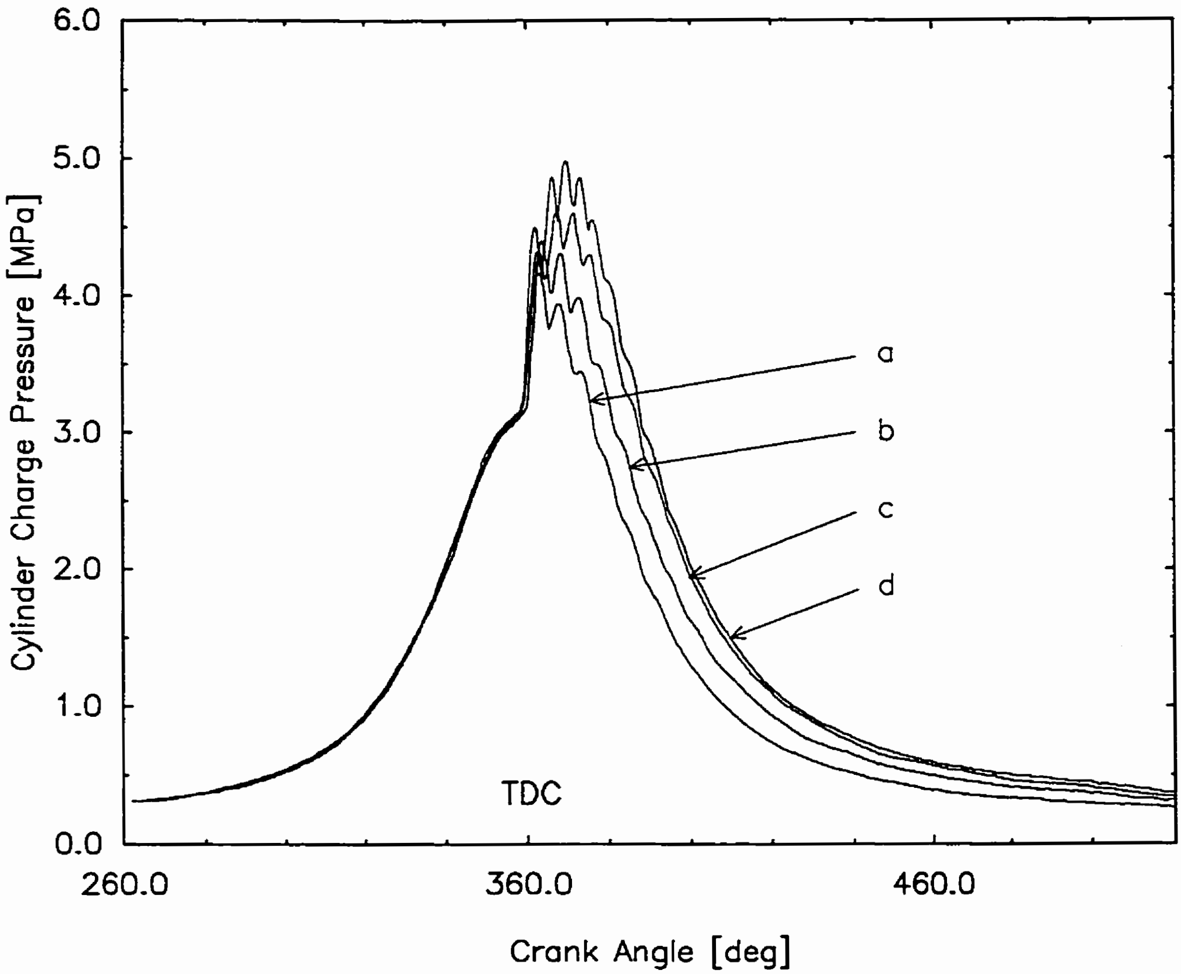
a: $\phi = 0.109$; $m_D = 0.262$ kg/h
b: $\phi = 0.153$; $m_D = 0.365$ kg/h
c: $\phi = 0.267$; $m_D = 0.634$ kg/h
d: $\phi = 0.363$; $m_D = 0.849$ kg/h
e: $\phi = 0.524$; $m_D = 1.205$ kg/h

Figure 4.1.6 Variations of Cylinder Charge Pressure with Time for Diesel Operation with a 58.0 Cetane Number Fuel. (m_D is the mass flow rate of diesel)



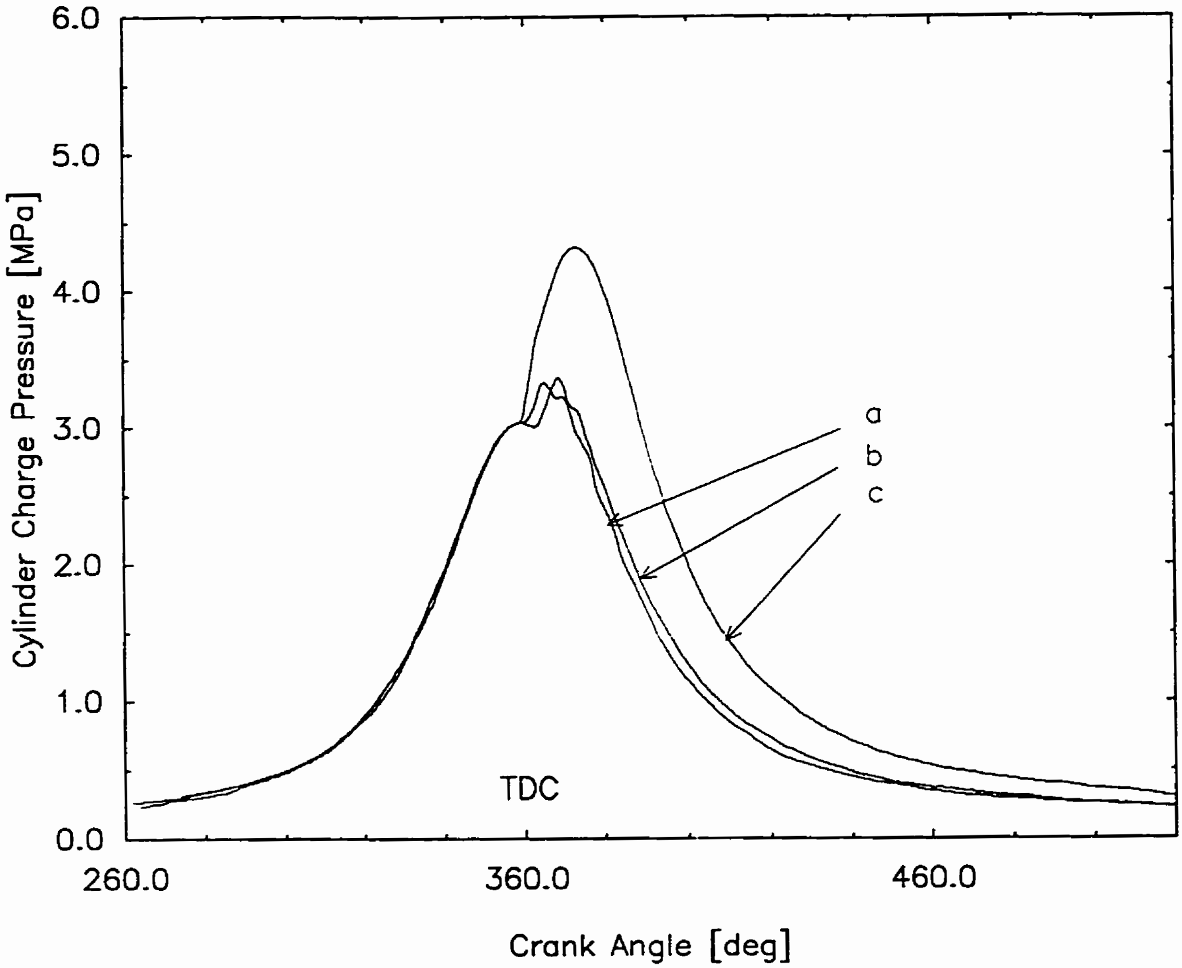
a: $\phi t = 0.275$; $\phi g = 0.125$; $m_{CH4} = 0.214$ kg/h
b: $\phi t = 0.416$; $\phi g = 0.292$; $m_{CH4} = 0.480$ kg/h
c: $\phi t = 0.524$; $\phi g = 0.420$; $m_{CH4} = 0.673$ kg/h
d: $\phi t = 0.583$; $\phi g = 0.490$; $m_{CH4} = 0.770$ kg/h

Figure 4.1.7 Variations of Cylinder Charge Pressure with Time for Dual Fuel Operation with Methane when Using a 58.0 Cetane Number Diesel Fuel with a 0.40 kg/h Pilot. (ϕ_t is the total equivalence ratio; ϕ_g is the gas equivalence ratio)



a: $\phi = 0.123$; $\phi g = 0.000$; $m_{CH4} = 0.000$ kg/h
b: $\phi = 0.235$; $\phi g = 0.126$; $m_{CH4} = 0.231$ kg/h
c: $\phi = 0.421$; $\phi g = 0.335$; $m_{CH4} = 0.587$ kg/h
d: $\phi = 0.514$; $\phi g = 0.440$; $m_{CH4} = 0.755$ kg/h

Figure 4.1.8 Variations of Cylinder Charge Pressure with Time for Dual Fuel Operation with Methane when Using a 58.0 Cetane Number Diesel Fuel with a 0.30 kg/h Pilot.



a: $\phi = 0.288$; $\phi g = 0.224$; $m_{CH4} = 0.420$ kg/h
b: $\phi = 0.481$; $\phi g = 0.432$; $m_{CH4} = 0.775$ kg/h
c: $\phi = 0.546$; $\phi g = 0.503$; $m_{CH4} = 0.883$ kg/h

Figure 4.1.9 Variations of Cylinder Charge Pressure with Time for Dual Fuel Operation with Methane when Using a 58.0 Cetane Number Diesel Fuel with a 0.20 kg/h Pilot.

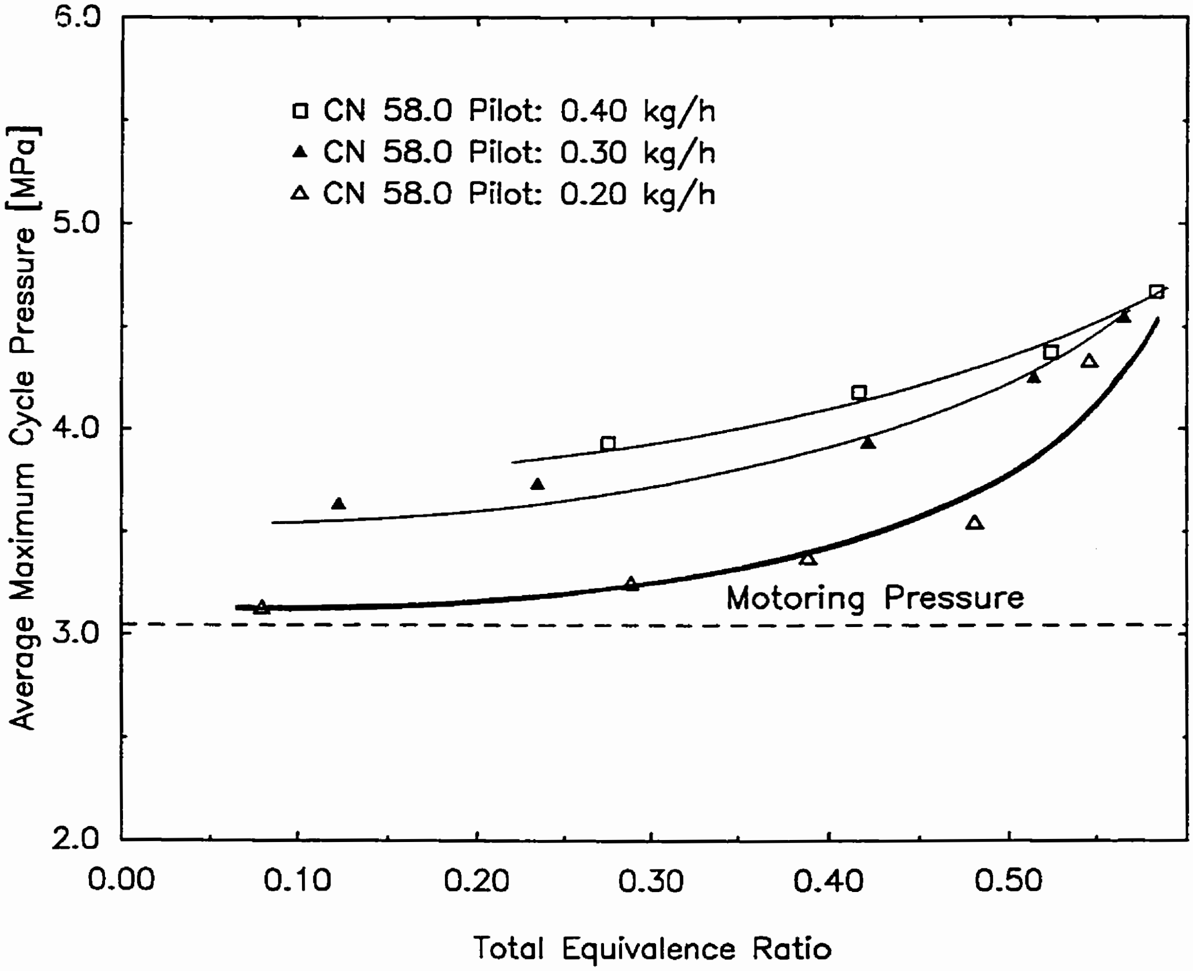


Figure 4.1.10 Variations of Average Maximum Cycle Pressure with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using a 58.0 Cetane Number Fuel with Different Pilot Quantities

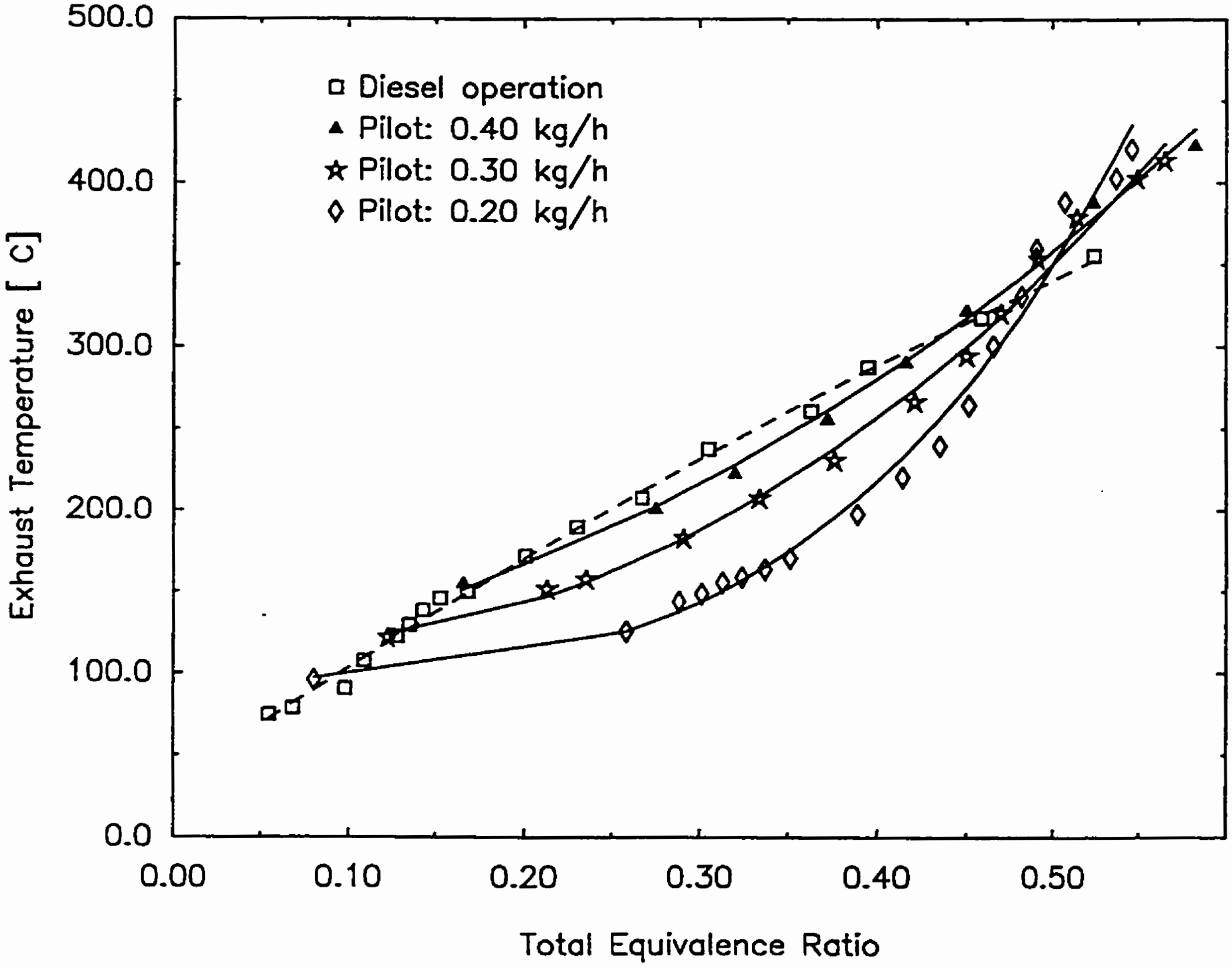


Figure 4.1.11 Variations of Exhaust Temperature with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using a 58.0 Cetane Number Fuel with Different Pilot Quantities

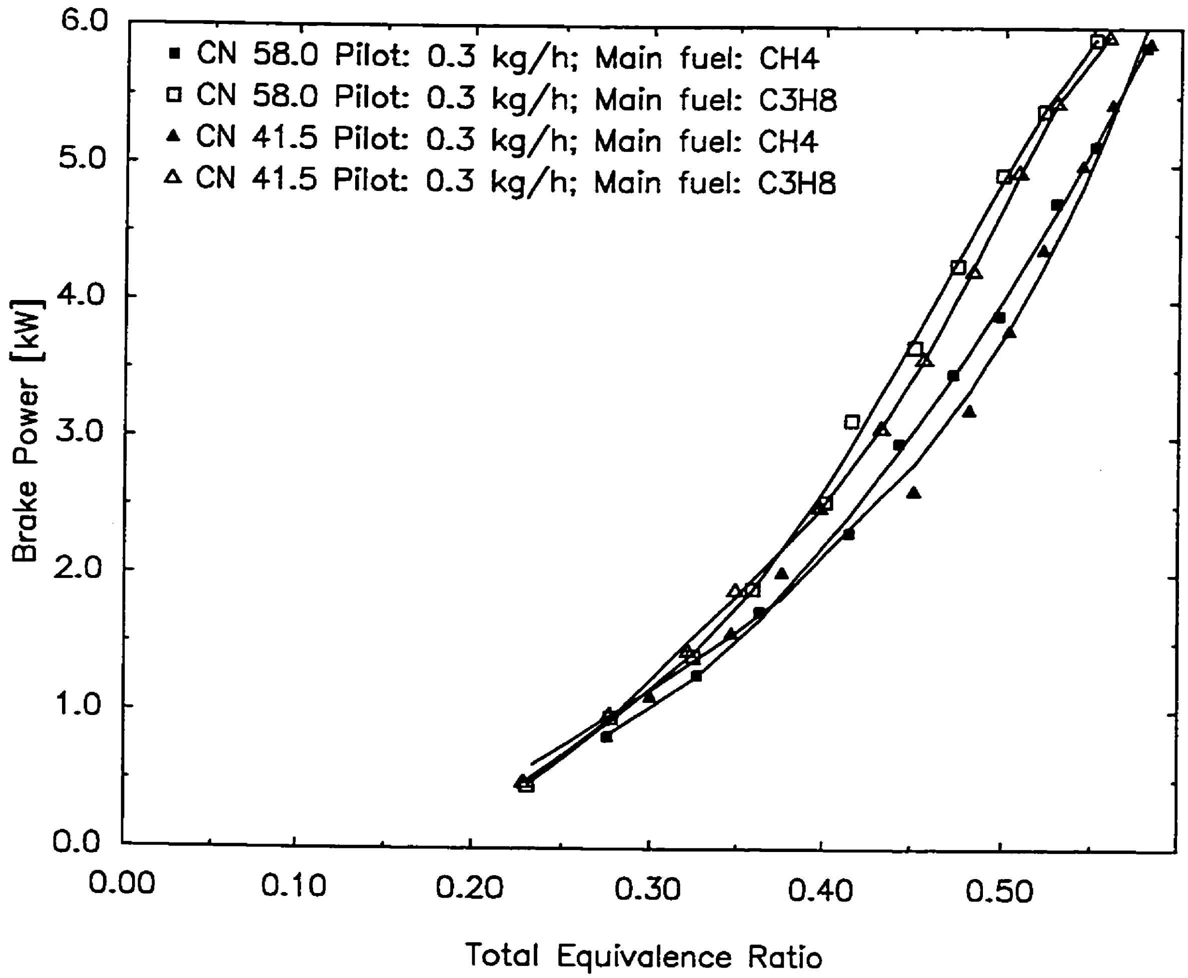


Figure 4.1.12 Variations of Brake Power with Total Equivalence Ratio for Dual Fuel Operation with a 0.30 kg/h Pilot when Using Different Cetane Number Fuels with Propane as Compared to Corresponding Values Obtained when Operating with Methane

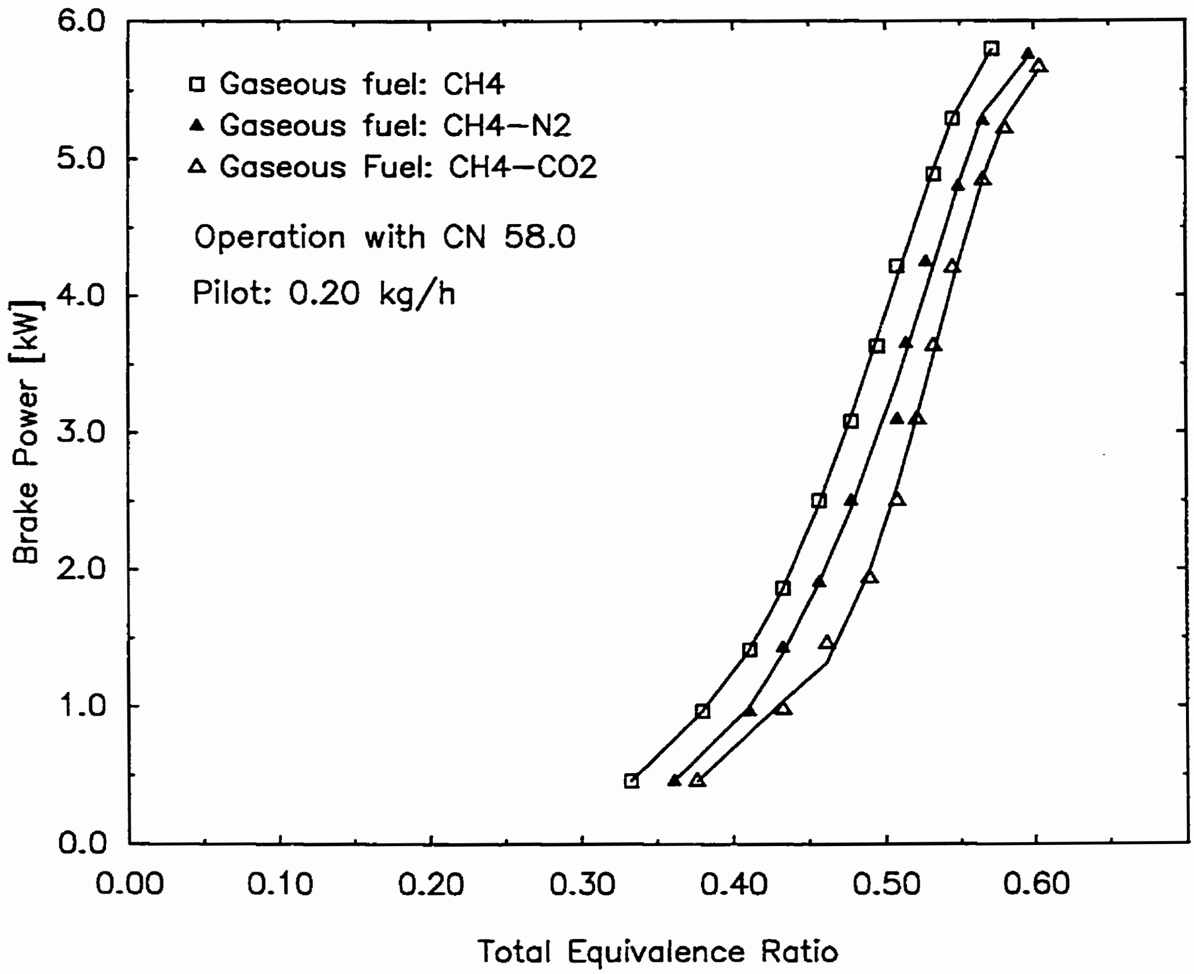


Figure 4.1.13 Variations of Brake Power with Total Equivalence Ratio for Dual Fuel Operation when Using a 58.0 Cetane Number Fuel with a 0.20 kg/h Pilot and Different Gaseous Fuels

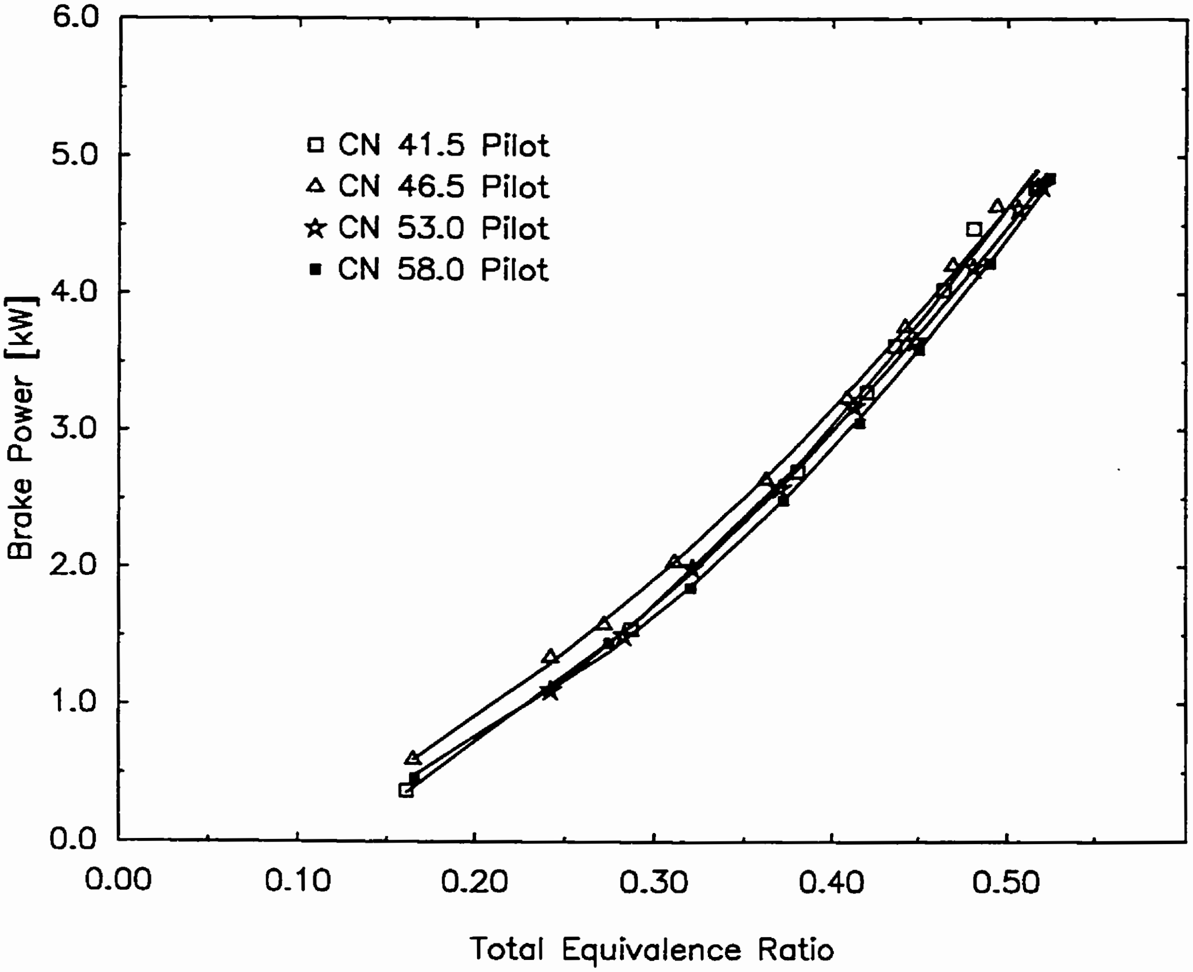


Figure 4.1.14 Variations of Brake Power with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.40 kg/h Pilot

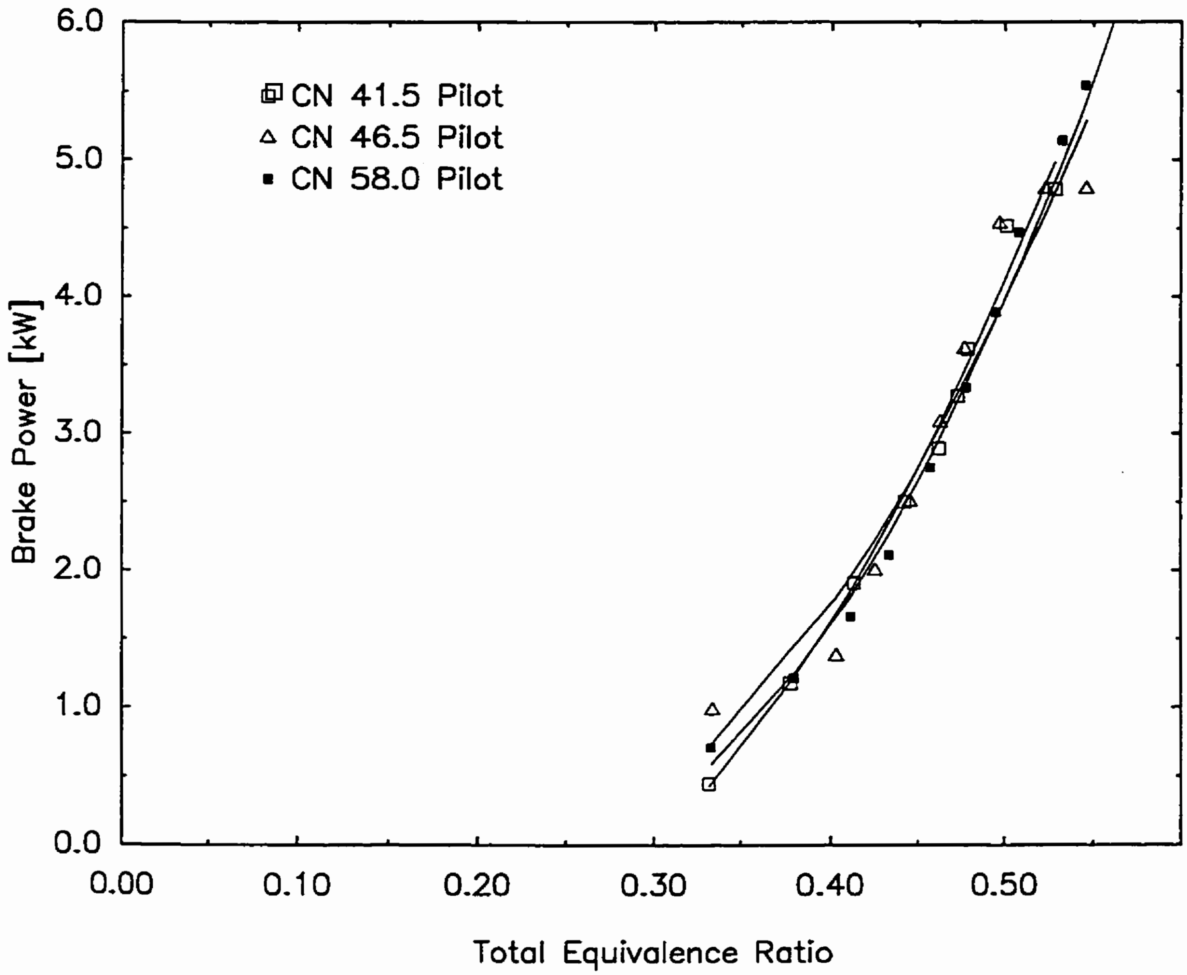


Figure 4.1.15 Variations of Brake Power with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

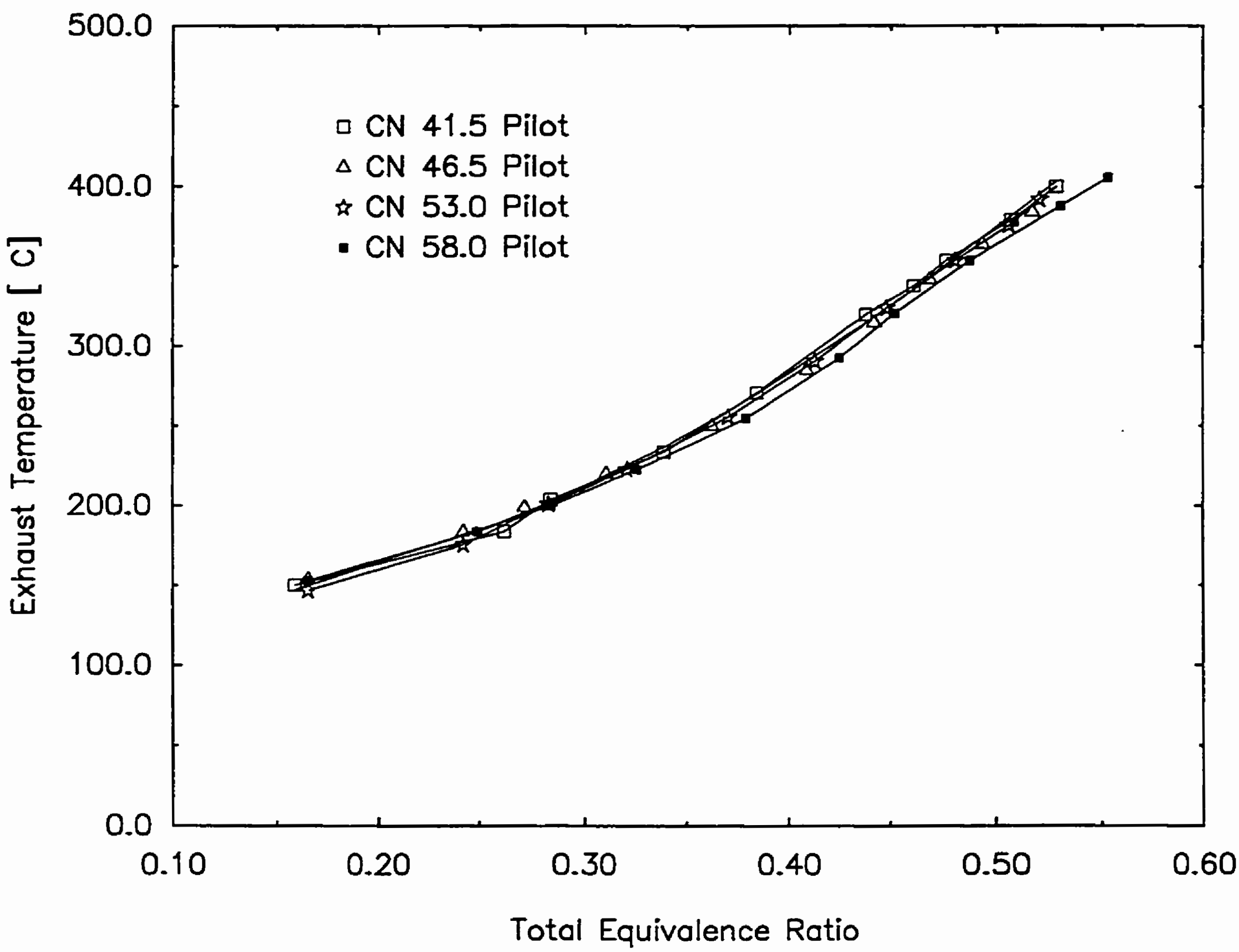


Figure 4.1.16 Variations of Exhaust Temperature with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.40 kg/h Pilot

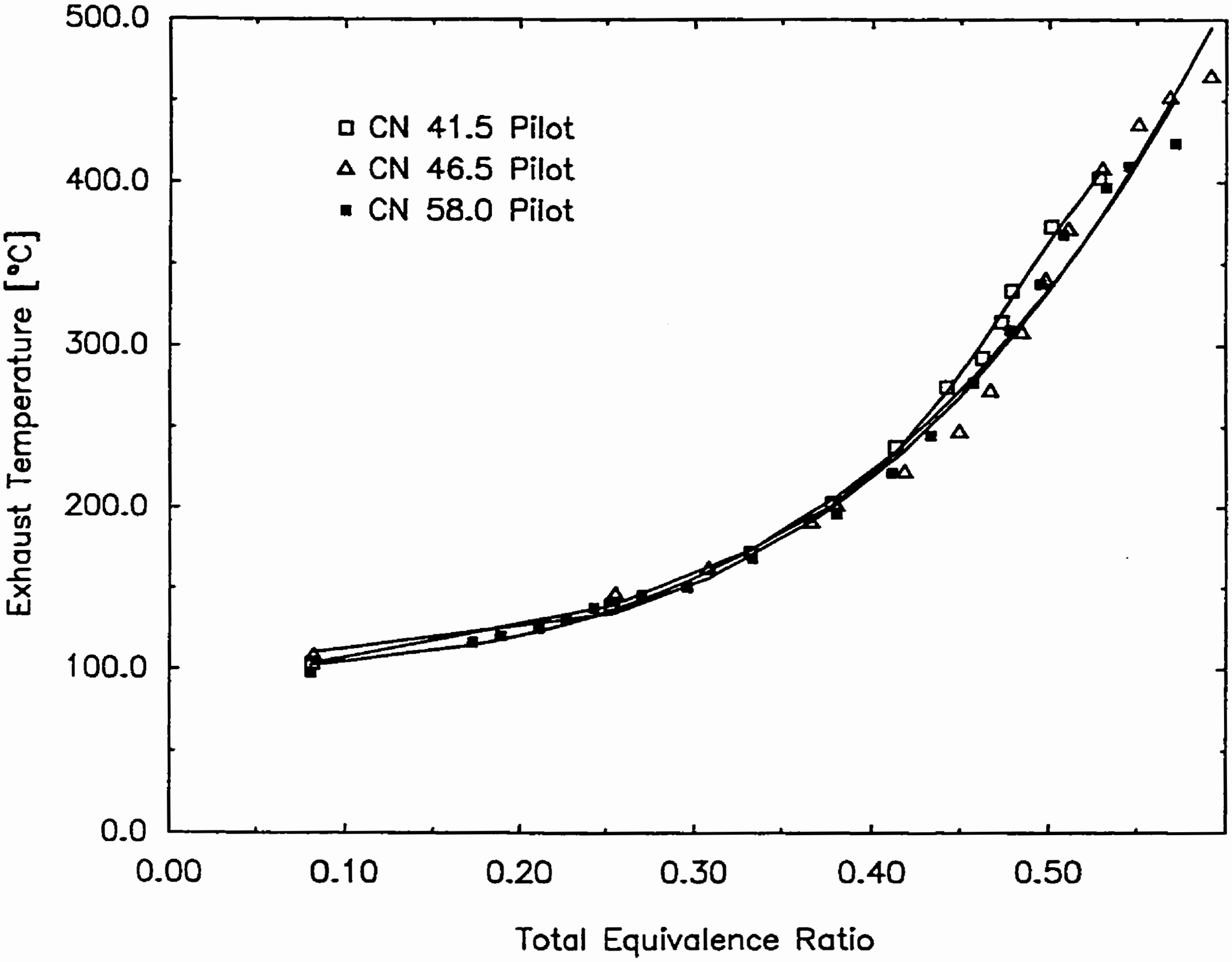


Figure 4.1.17 Variations of Exhaust Temperature with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

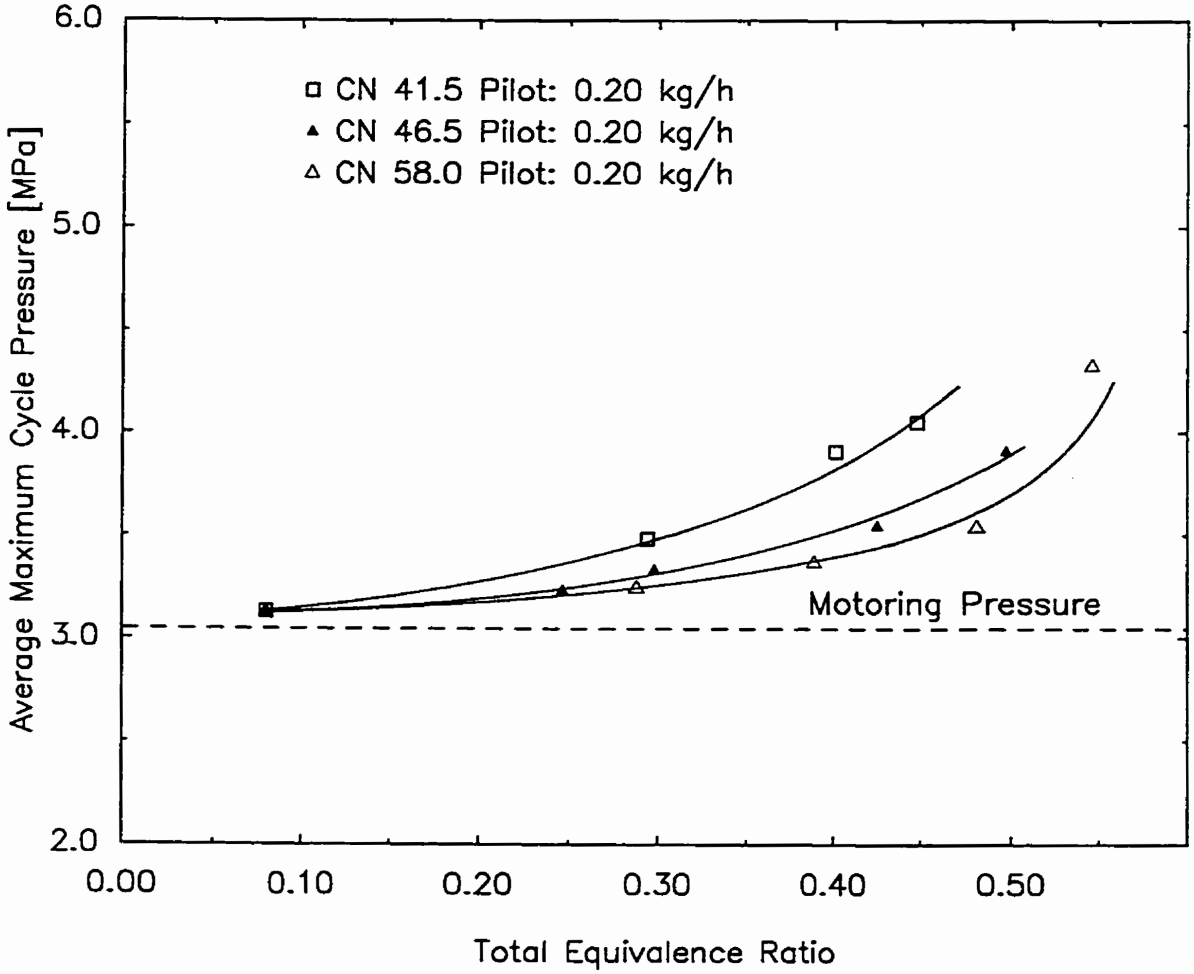


Figure 4.1.18 Variations of Average Maximum Cycle Pressure with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

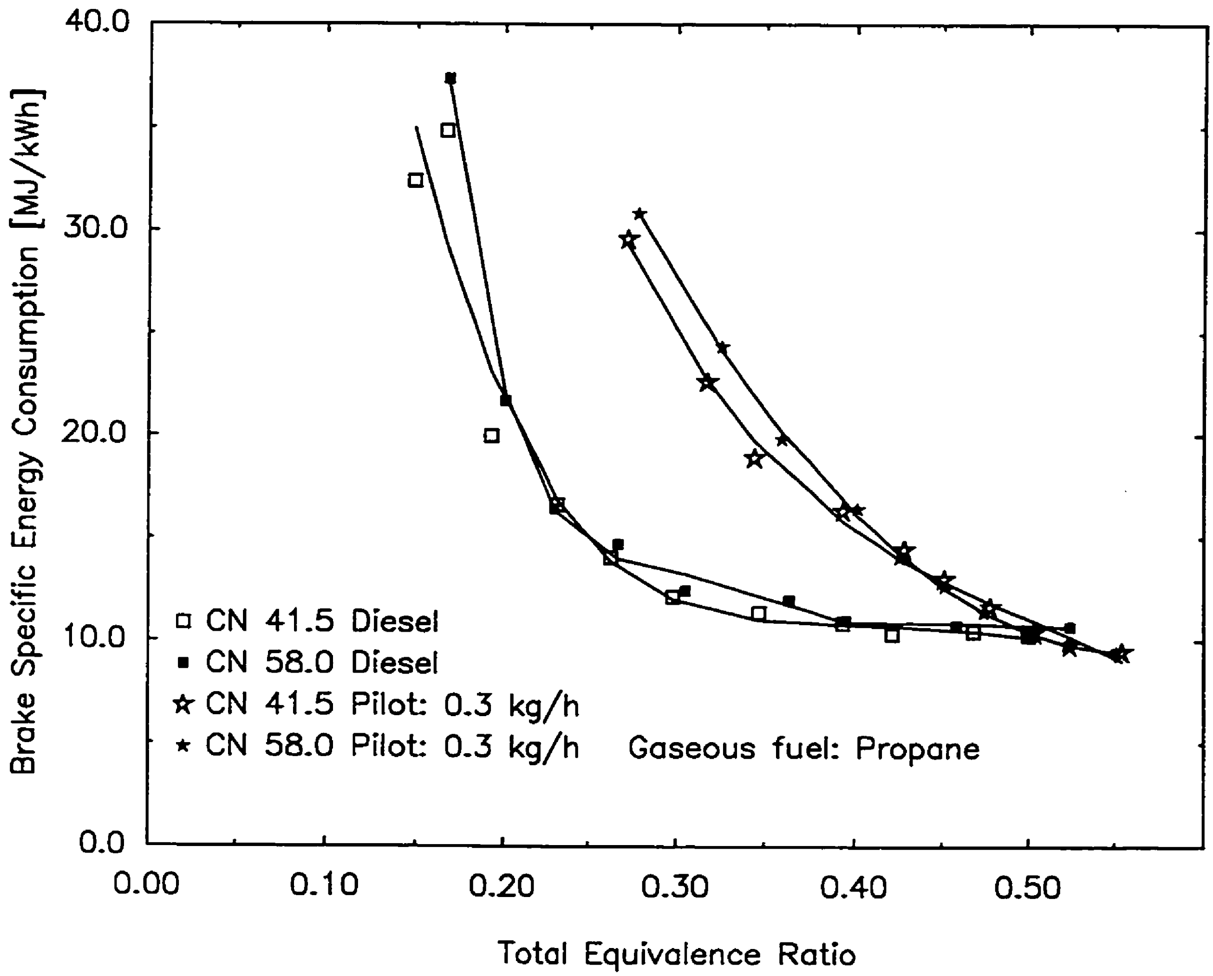


Figure 4.1.19 Variations of Brake Specific Energy Consumption with Total Equivalence Ratio for Dual Fuel Operation with Propane when Using Different Cetane Number Fuels with a 0.30 kg/h Pilot

4.2 Ignition Delay During Dual Fuel Operation with Commercial (CN 41.5) Diesel Fuel Pilots

The variations in the length of the ignition delay in compression ignition engines have a profound and controlling effect on the subsequent combustion processes and hence on almost every feature of engine performance. With a constant diesel fuel injection timing, the crank angle at ignition was used to represent the ignition delay variation. The top dead centre (TDC) position was used as the reference point (0.0) for the representation of the variations of the crank angle at ignition. Ignition was established from the measured pressure-time records on the basis that the first departure of the pressure curve from the normal trend during compression is indicative of the onset of ignition. As shown typically in Fig. 4.2.1, the point of ignition and hence the ignition delay in diesel engine operation with constant fuel injection timing decreases significantly for all the fuels tested as the amount of diesel fuel injected is increased. The plots are made on the basis of the equivalence ratio of the total amount of fuels used. Regular ignition could not be maintained unaided below a certain level of pilot quantity. For diesel operation with small amounts of injected liquid fuel consistent with those used as pilots in dual fuel engine applications often produce insufficiently small power outputs to maintain the engine speed at the constant prescribed value and will be within the partial motoring region (i.e. equivalence ratio values below ϕ=0.12). The ignition delay values are then increased and become subject to significant levels of cyclic variations. Through the injection of larger amounts of liquid fuel per cycle, the ignition delay is reduced and tends to approach a constant value.

Different amounts of pilot of commercial CN 41.5 diesel fuel were initially employed in dual fuel operation with methane. It can be seen from Fig. 4.2.2 that, for any fixed pilot quantity and injection timing, as the amount of the gaseous fuel is increased under dual fuel operation, producing a higher total equivalence ratio (i.e. the equivalence ratio based on both the gaseous fuel and liquid pilot), the ignition delay increases markedly to reach a maximum value. The ignition delay then begins to decrease later on with the continued addition of the gaseous fuel. These changes in the delay period depend very strongly on the pilot quantity used, as illustrated in Figs. 4.2.3, 4.2.4 and 4.2.5. Larger pilots tend to display smaller changes in the delay, while relatively small pilots bring about a larger extension of the ignition delay. With a continued decrease in the pilot fuel size, the ignition delay period increases to such an extent that regular ignition cannot be maintained below a certain level of pilot quantity.

The extent of the ignition delay observed depends also on the type of the gaseous fuel used. The admission of a fuel such as propane brings an increase in the ignition delay period to an extent much longer than the corresponding values observed with methane, as shown typically in Fig. 4.2.6, when operating the engine on CN 41.5 diesel fuel. This extension to the delay with propane operation is mainly due to the lower temperature and pressure at around the top dead centre region where the pilot fuel is injected, resulting from the lowering of the effective specific heat ratio of the cylinder charge. The decrease in the mean charge temperature and pressure during the delay, lowers the preignition reaction activity and the associated postignition energy release rates [38] and brings the extension to the delay observed.

Similarly, the operation with low heating value gas fuel mixtures, represented typically by a mixture of 63% by vol. methane and 37% by vol. nitrogen (i.e. 50% by mass methane), simulating the behavior of a natural gas containing significant amounts of diluents, shows changes in the ignition delay that tend to be essentially linearly related to the concentration of the nitrogen admitted, as shown in Fig. 4.2.7. This increase in the delay is a reflection of the combined effects of the corresponding reduction in the partial pressure of oxygen of the intake charge, due to the increased displacement of the air with the diluent, the associated reduced reaction activity, any changes in the extent of heat transfer and any corresponding changes in the effective temperature level during combustion. Similar mixtures that contained carbon dioxide instead of nitrogen, as shown in Fig. 4.2.8, produced yet much greater increase in the delay mainly due to the further lowering of the mean temperature and pressure during the delay as a result of the reduced value of the effective polytropic index of compression and also changes to the heat transfer. Moreover, the presence of substantial amount of carbon dioxide in the charge will also affect both the physical and chemical factors that bring about the ignition delay of the pilot.

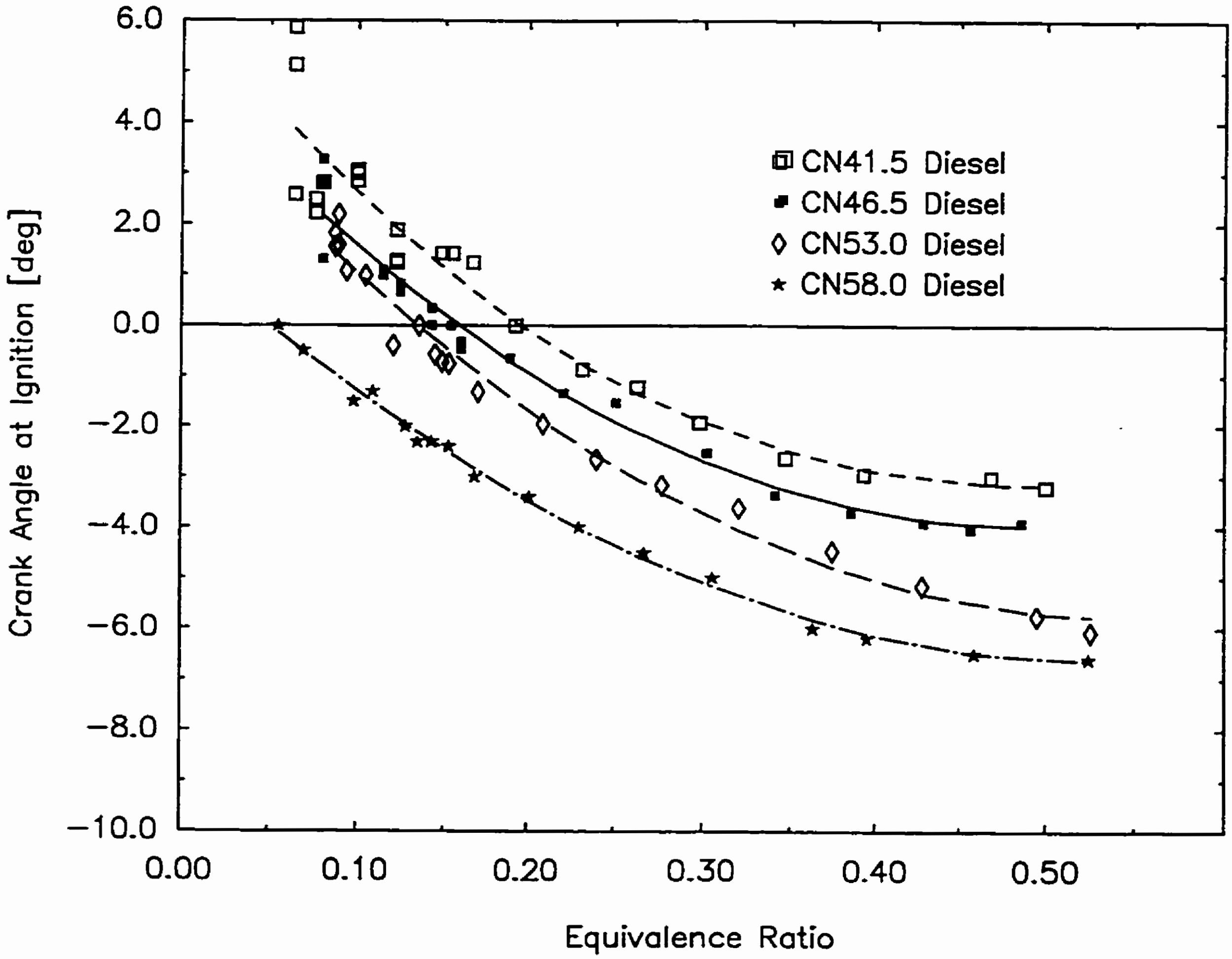


Figure 4.2.1 Variations of the Point of Ignition with Equivalence Ratio for Diesel Operation when Using Different Cetane Number Fuels

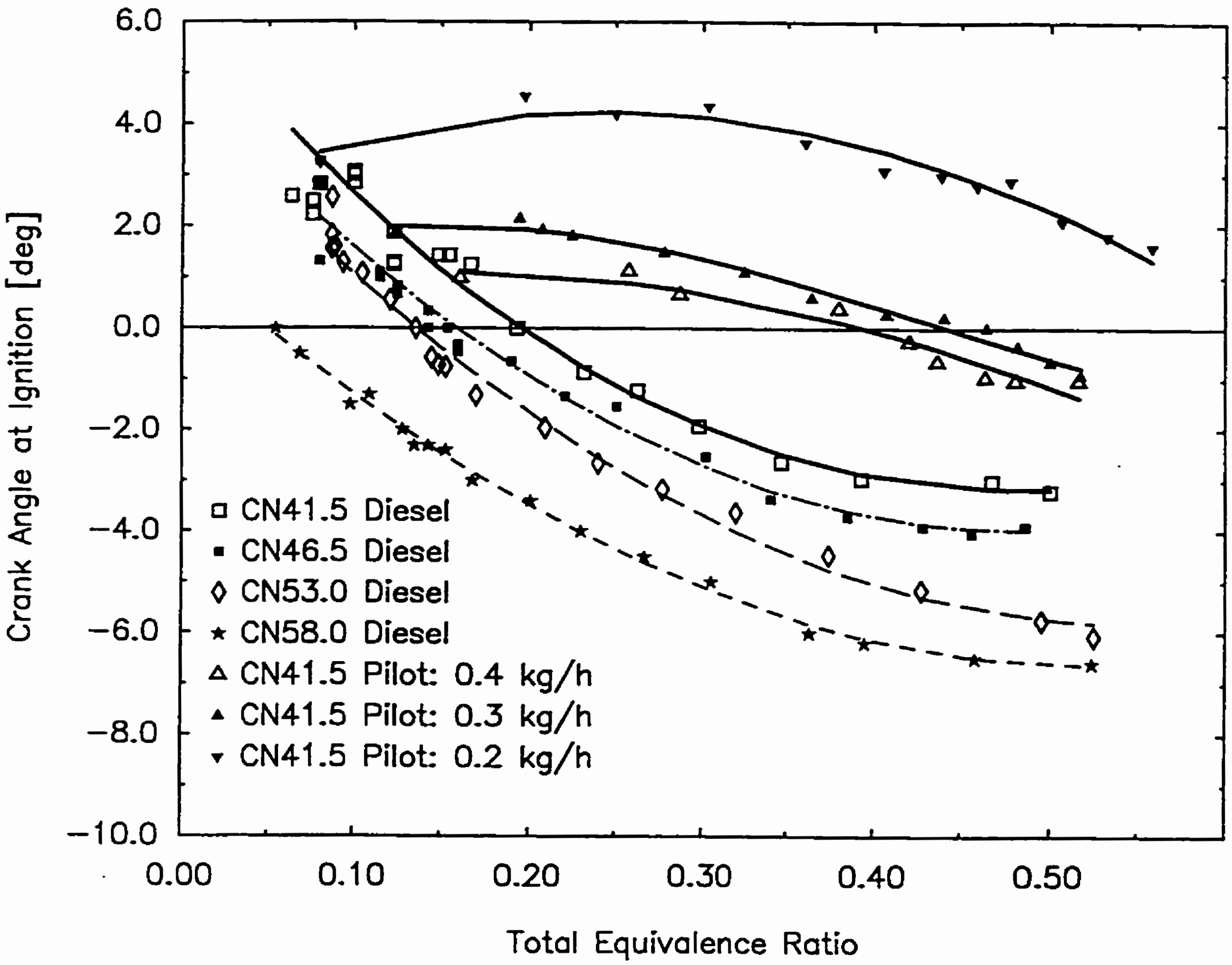


Figure 4.2.2 Variations of the Point of Ignition with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using a 41.5 Cetane Number Fuel with Different Pilot Quantities

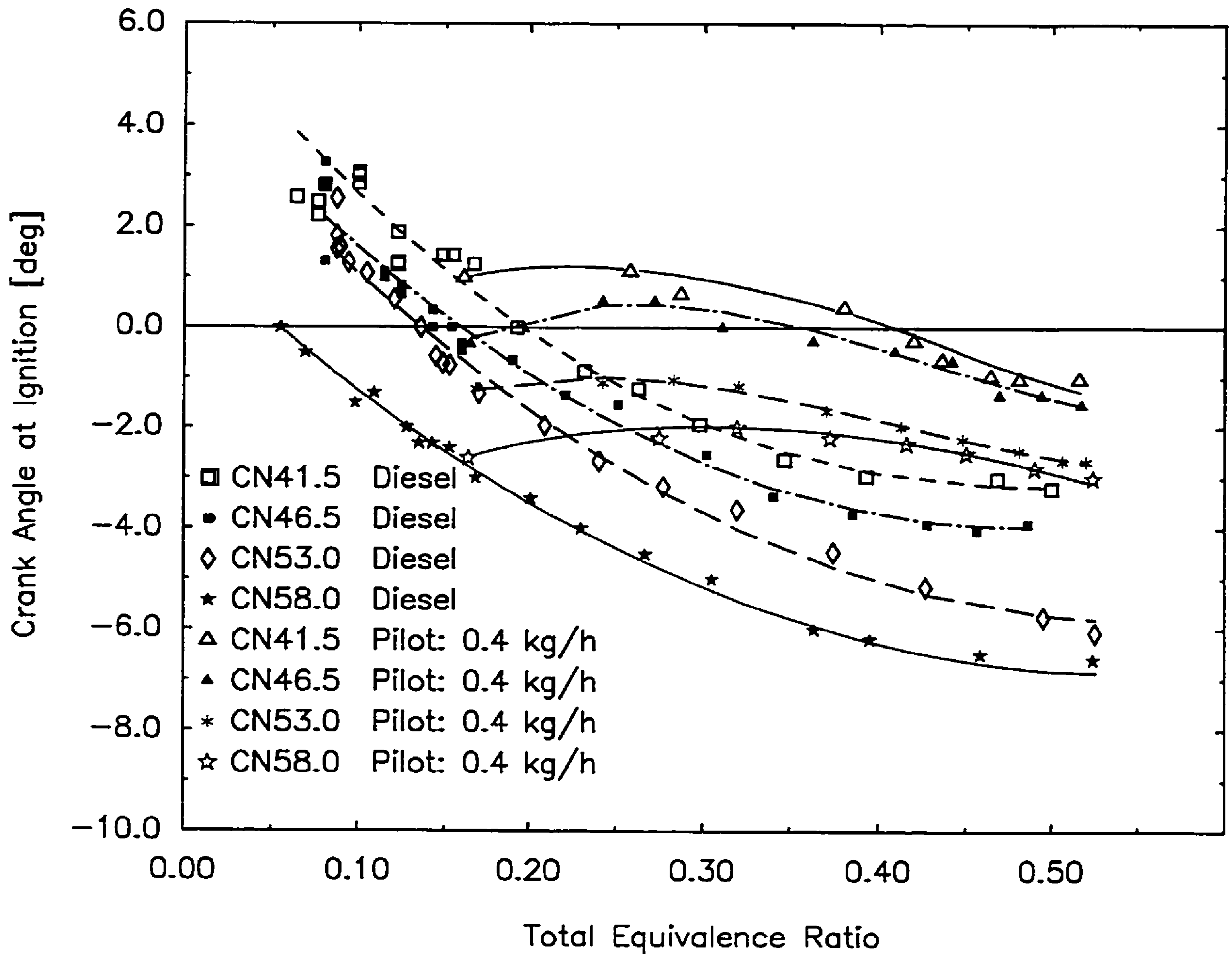


Figure 4.2.3 Variations of the Point of Ignition with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.40 kg/h Pilot

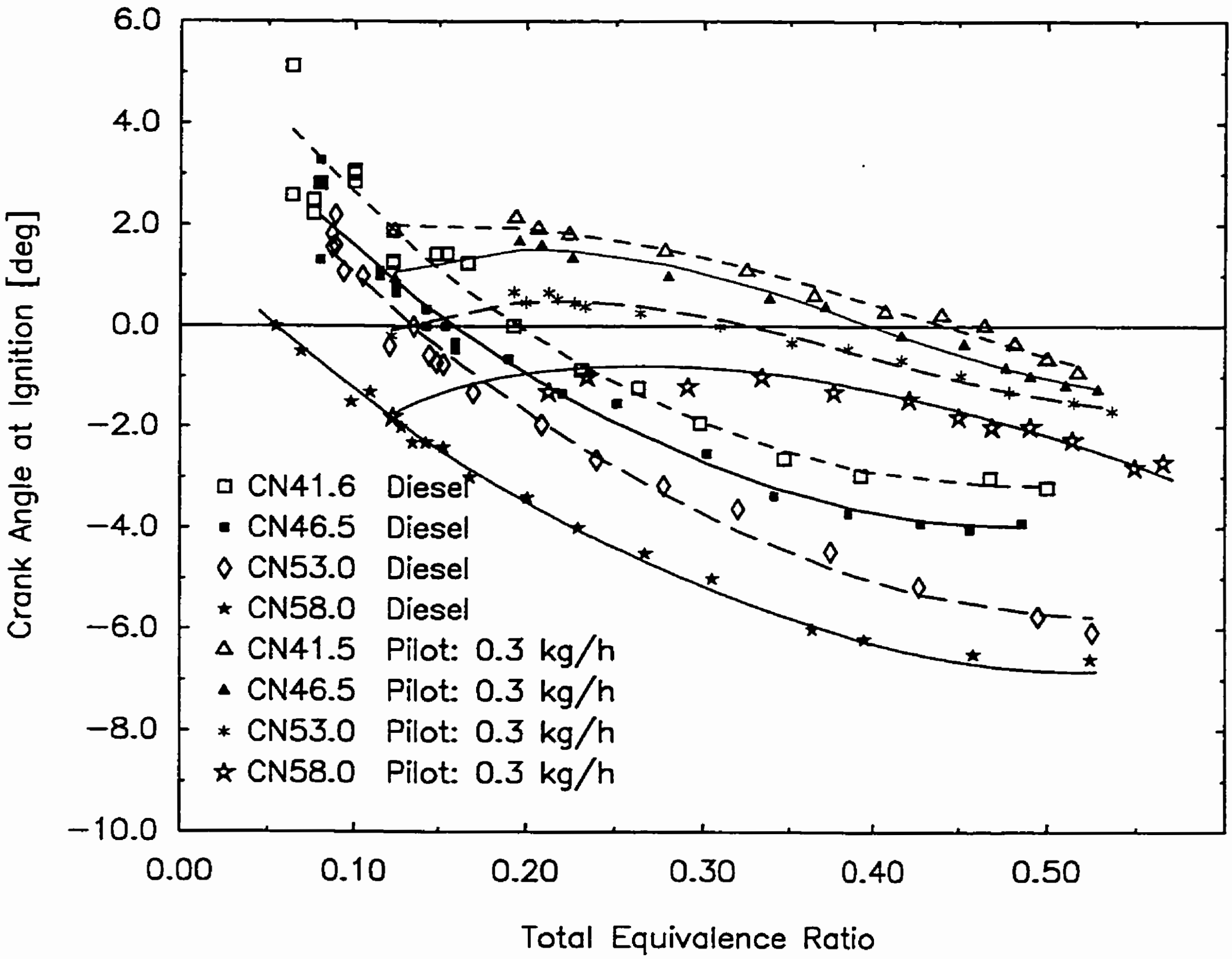


Figure 4.2.4 Variations of the Point of Ignition with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.30 kg/h Pilot

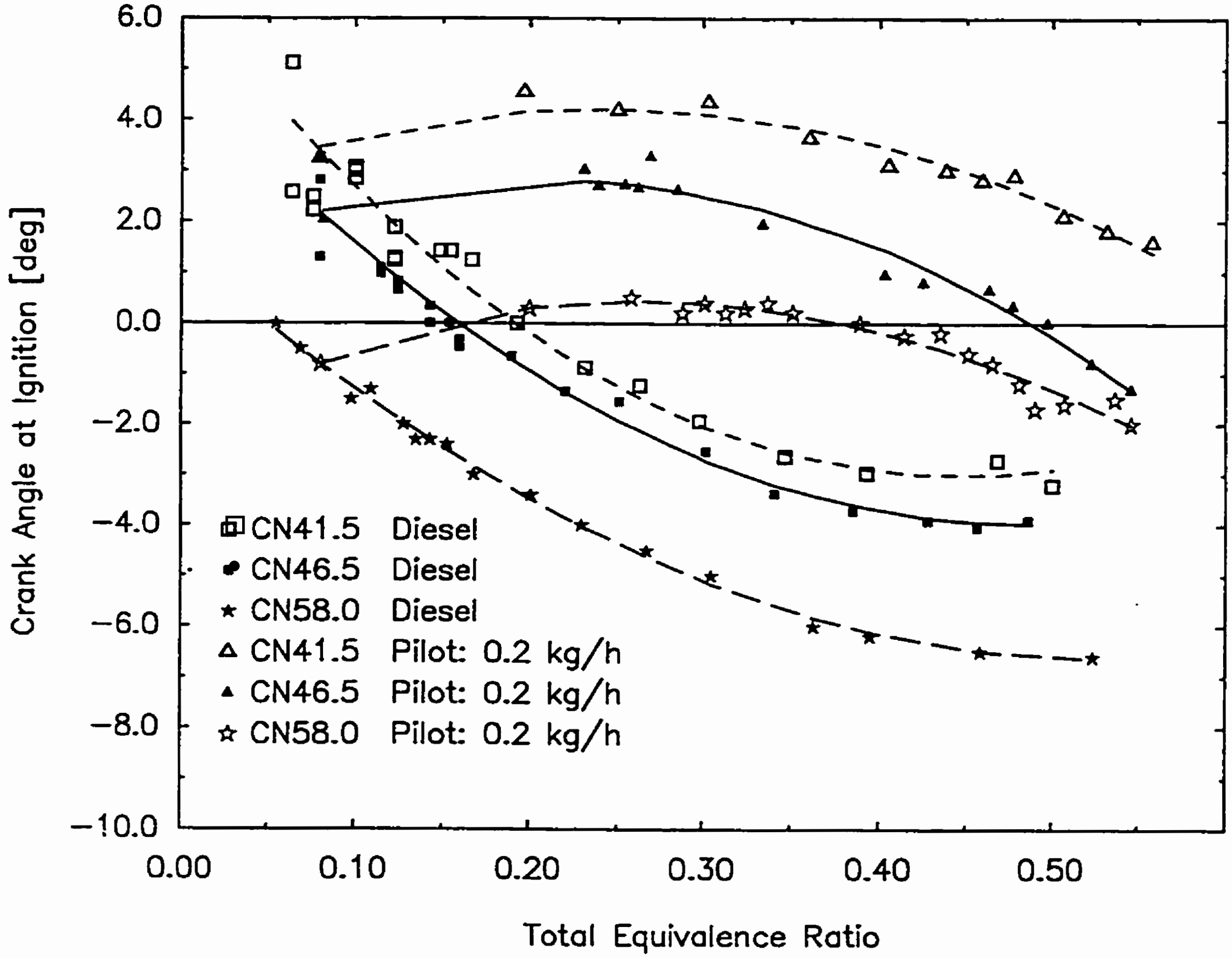


Figure 4.2.5 Variations of the Point of Ignition with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

4.3 Ignition Delay During Dual Fuel Engine Operation with Different Cetane Number Diesel Fuel Pilots

Expectedly, by definition, the ignition delay decreases with an increase in the diesel fuel cetane number. Each of the four diesel fuels tested, having different cetane numbers tends to display almost the same incremental decrease in the ignition delay over the range of the charge overall equivalence ratio, as represented in Fig. 4.2.1.

An increase in the cetane number of the pilot fuel to 58.0 when operating the engine on methane with different pilot quantities, produces significant earlier ignition, while maintaining its characteristic variations with the increased admission of the gaseous fuel, as shown in Fig. 4.2.9. In Fig. F.1.3 (Appendix F) are represented the corresponding variations of the point of ignition based on methane equivalence ratio only, when using different pilot quantities.

Same trends in the variations in the point of ignition during dual fuel operation with methane were found when using different cetane number diesel fuels and different pilot quantities as shown in Figs. 4.2.3, 4.2.4 and 4.2.5 (For the operation with a 0.20 kg/h pilot, the corresponding values based on methane equivalence ratio are shown in Fig. F.1.1). The change in the point of ignition is relatively less marked when operating with a relatively high pilot liquid fuel quantity (0.40 kg/h) when a high cetane number fuel is employed. Moreover, as shown in Fig. 4.2.5, the use of a relatively small pilot quantity (0.20 kg/h) with a higher cetane number fuel can lead to a substantial improvement in the delay, lower rates of pressure rise, improved combustion at light load and smoother operation. In fact, dual fuel operation at very light loads (total equivalence ratio below 0.25) was impractical with the 41.5 cetane number fuel and the small pilot quantity of 0.2 kg/h due to the long delays involved and the associated cyclic variations.

The changes in the length of the ignition delay depend not only on the chemical interactions between the fuel jet vapor and the surrounding gaseous fuel-air mixture but also on the characteristics of the pilot fuel injection, spray penetration, atomization, vaporization and mixing processes within the cylinder charge. For a heavy diesel fuel with a high viscosity and low volatility, the extension to the physical part of the delay may be the controlling factor [21]. This was the case when using a diesel fuel having a nominal cetane number of 53 and displaying relatively inferior physical properties compared to the other three commercial diesel fuels tested. It was found that with such a fuel, the use of smaller pilot quantities than 0.30 kg/h could not maintain satisfactory engine operation as it led to very late ignition and erratic engine load and speed fluctuations. Special modifications to the injection system of the engine, including the pre-heating of the fuel, would be required to make the engine operation possible when using such a fuel.

In Fig. 4.2.6. are represented the variations of the point of ignition with total equivalence ratio for dual fuel operation with propane when using a CN 58.8 fuel, as compared to corresponding values obtained when operating with a CN 41.5 fuel. It can be seen that the employment of a relatively high cetane number pilot fuel with propane can reduce significantly the ignition delay normally observed with this fuel at light load, to values comparable to those obtained with methane operation when pilots of a regular commercial diesel fuel (CN 41.5) are used. In Fig. F.1.2 (Appendix F) are represented the corresponding variations of the point of ignition based on gas equivalence ratio only, when using different pilot quantities.

The use of high cetane number pilots also enhanced dual fuel operation with CH_4-N_2 fuel gas mixture. In Fig. 4.2.7 is shown that a significant reduction in the ignition delay can be obtained throughout the whole range of engine load, but especially when very lean mixtures are used with relatively small pilots. Similarly, the engine operation improved when using a higher CN pilot with a CH_4-CO_2 mixture, as shown in Fig. 4.2.8.

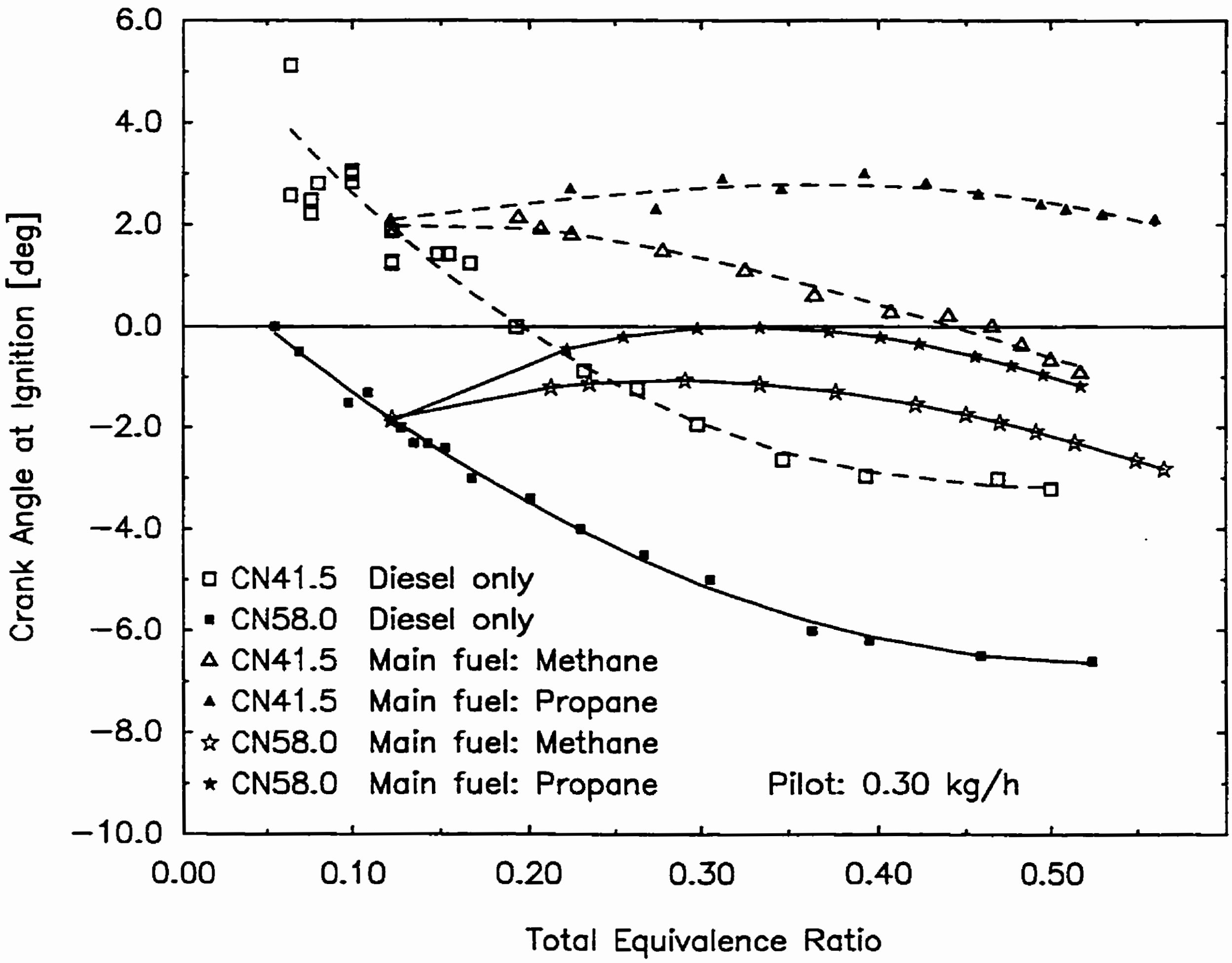


Figure 4.2.6 Variations of the Point of Ignition with Total Equivalence Ratio for Dual Fuel Operation with Propane when Using Different Cetane Number Fuels as Compared to the Corresponding Values Obtained when Operating with Methane

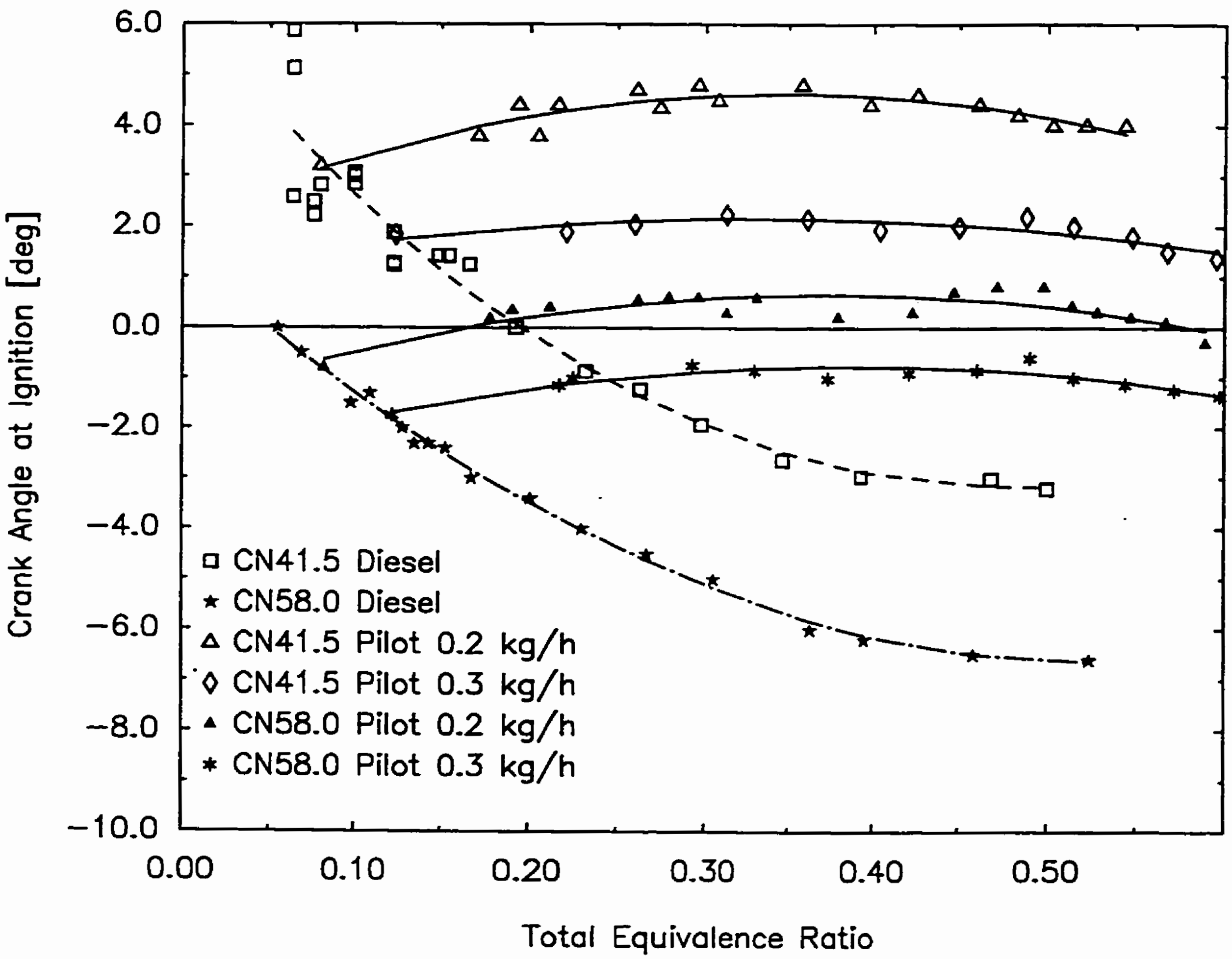


Figure 4.2.7 Variations of the Point of Ignition with Total Equivalence Ratio for Dual Fuel Operation when Using Different Cetane Number Fuels and Different Pilot Quantities with a CH4-N2 Mixture (50% by mass CH4)

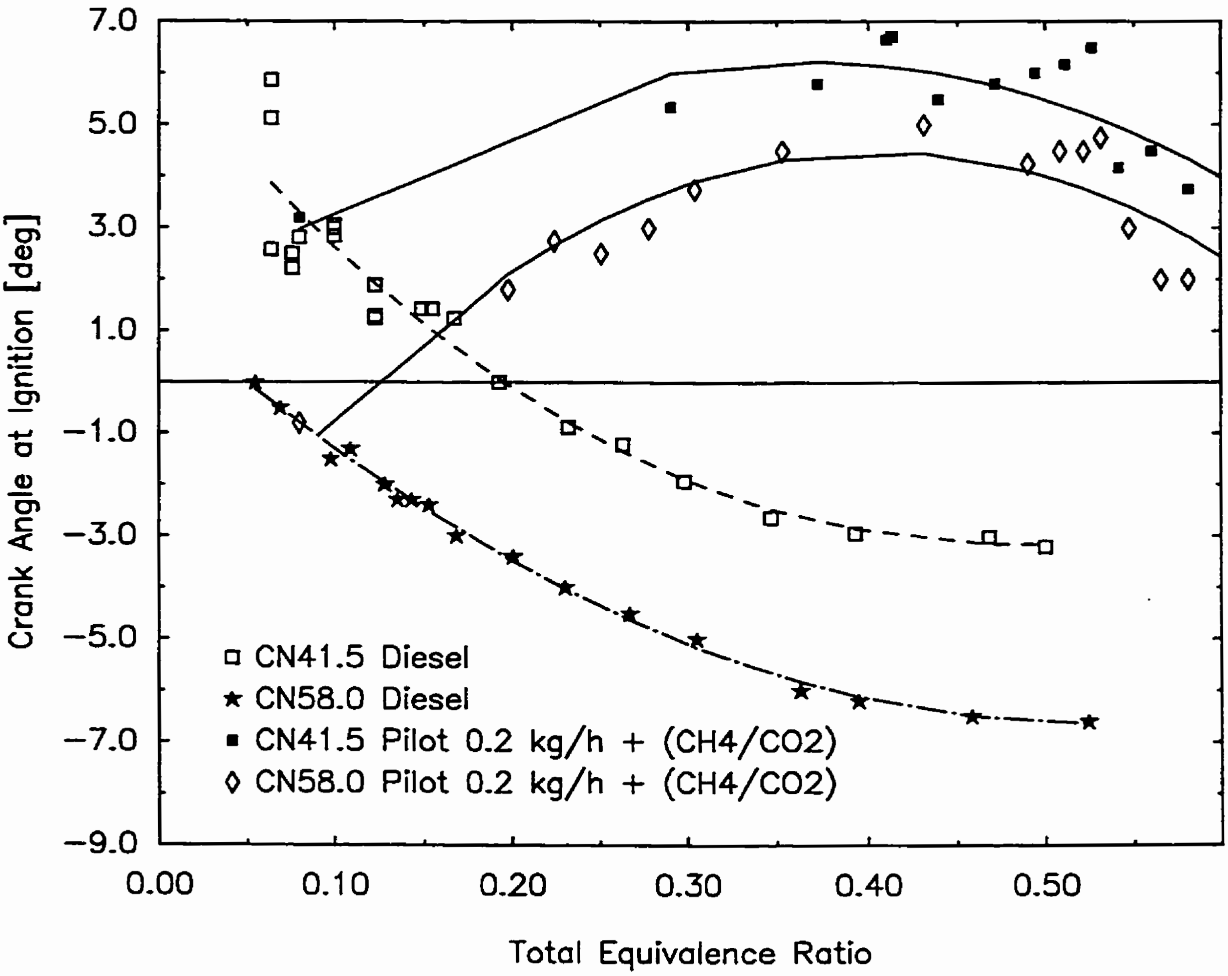


Figure 4.2.8 Variations of the Point of Ignition with Total Equivalence Ratio for Dual Fuel Operation when Using Different Cetane Number Fuels and Different Pilot Quantities with a CH_4-CO_2 Mixture (50% by mass CH_4)

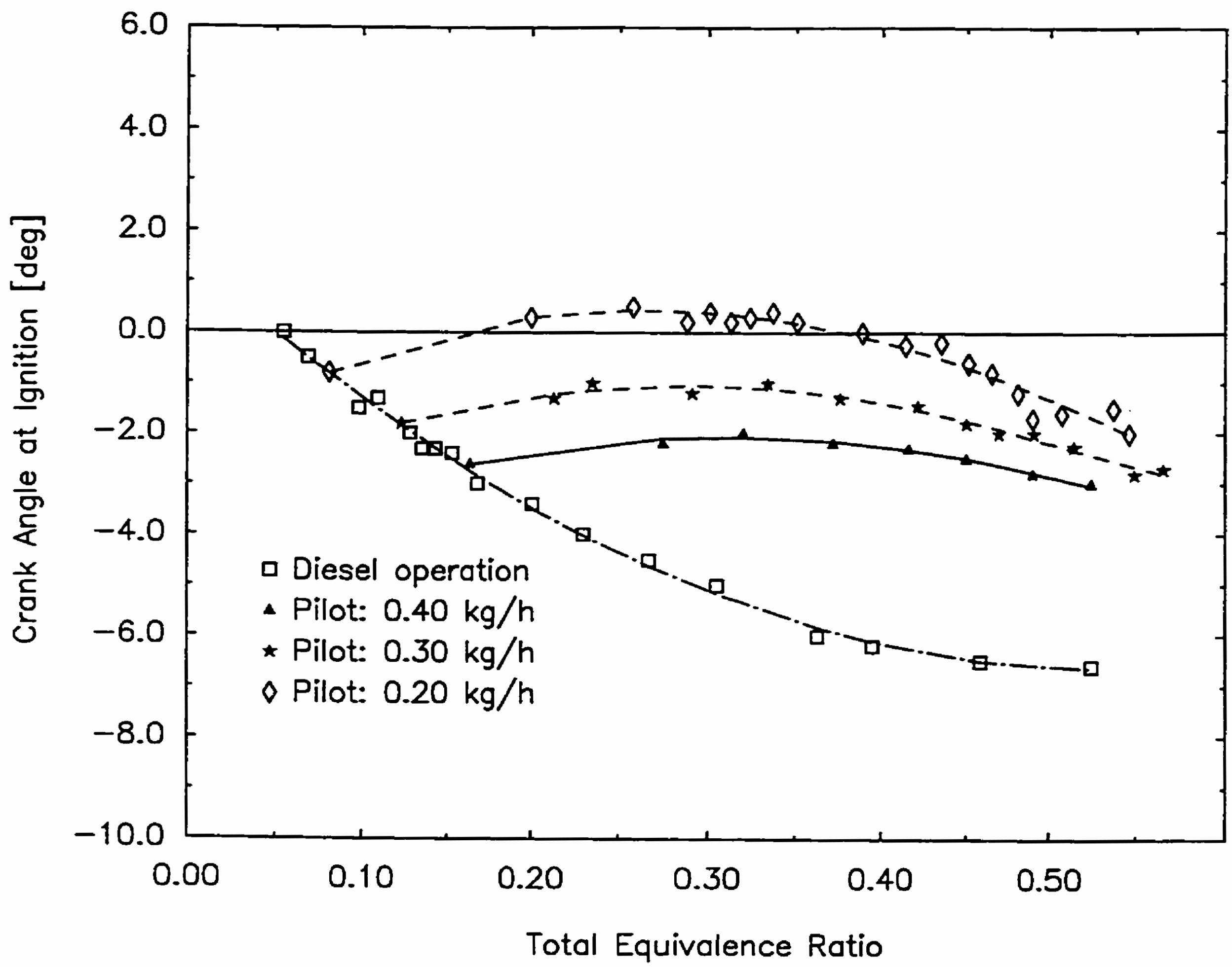


Figure 4.2.9 Variations of the Point of Ignition with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using a 58.0 Cetane Number Fuel with Different Pilot Quantities

4.4 Exhaust Emissions During Dual Fuel Operation with Different Cetane Number Diesel Fuel Pilots

Both the combustion of the diesel pilot fuel and the main premixed charge contribute to the formation of pollutants in dual fuel engine operation. Accordingly, an examination of the relative effects of the quality (especially cetane number) and quantity of the pilot fuel injection, as well as the gaseous fuel concentration on dual fuel engine exhaust emissions, with special reference to light load operation is presented in this chapter.

Fig.4.4.1 shows the variation of the CO concentration in dry exhaust with total equivalence ratio when the dual fuel engine was operated on commercial methane using different pilot amounts of commercial 41.5 CN diesel fuel. Carbon monoxide is the primary intermediate product in the methane oxidation and, for lean methane-air mixtures, its presence in the exhaust is an indication that some of the CO produced through the oxidation reactions of the methane could not be oxidized further to carbon dioxide. The reason for this is that at light load, with very small amount of methane introduced in the cylinder, even after the energy release and raising temperature following the ignition of the pilot, the concentration of methane in the fuel-air mixture is so small, that the oxidation reactions starting from the pilot ignition centres will be quenched. With very lean engine operation and small pilots within the partial motoring region, the CO concentrations recorded imply that they originate partially also from the incomplete combustion then of the small pilots. At light load, CO emissions increase as the amount of gaseous fuel is increased. When more methane is admitted, a maximum value is reached, indicating a change in the extent of completion of the oxidation reactions as more extension of the flame propagation to the mixture regions takes place, while approaching the corresponding local effective lean flammability limit. It appears that in the region of total equivalence ratio of above ϕ=0.5, when the CO concentrations begin to decline, propagated combustion reactions occur, even if the flame does not travel throughout the whole combustion chamber. For methane-air concentrations less than the observed effective lean flammability limit, the energy release process during dual fuel operation is similar to that observed with full diesel operation while the oxidation of the gaseous fuel is limited mainly to the regions surrounding the active ignition centres of the pilot spray [4,26].

The variations of unburnt CH_4 concentration in the exhaust gases have a very similar trend to that observed for the CO concentrations, with the maxima occurring at slightly leaner mixtures than the CO maxima (at ϕ~0.40). Fig. 4.4.2 shows that a considerable fraction of the methane representing significant quantities of fuel can pass through the engine cylinder unreacted. When the engine runs at very light loads, the poor atomization of the pilot with increased ignition delay, large cyclic variations and low charge temperature, result in a very low gaseous fuel utilization. The unburnt CH_4 concentration increases initially with the increase in the methane concentration in air of the cylinder charge. However, the utilization of the supplied methane improves continuously with increased gaseous fuel admission, as shown in Fig. 4.4.3. The percentage of unreacted methane in the intake charge was calculated from the knowledge of the dry exhaust concentration of methane and the fuelling rates. For mixture strengths above Φ=0.40, the rapid decline of the unreacted methane percentage expectedly occurs as the effective lean flammability limit of the mixture is approached.

The size of the pilot has a very significant influence on the level of the unreacted CH_4 in the exhaust gases and hence unburnt hydrocarbon emissions. At light load operation, the utilization of the gaseous fuel is improved significantly with the increase in the quantity of the pilot fuel due to the increased rates of combustion. For example, when a 0.20 kg/h pilot is used, about 50% of the intake CH_4 amount does not react up to a total equivalence ratio of 0.5, while the utilization fraction of methane becomes closer to 90% when the size of the pilot is doubled to 0.40 kg/h for the same equivalence ratio.

The shifting of the peak values of methane concentration in the exhaust towards weaker mixtures indicates that the effective lean flammability limit of the charge is lowered when using higher pilot quantities due to the increase in temperature, which is the main factor affecting the limit.

At higher loads, when the methane concentration in the cylinder charge is high enough for the flame to propagate throughout the whole charge, the pilot fuel quantity has little influence.

The use of larger pilots tends to produce higher levels of CO emissions. Carbon monoxide originates mainly from the methane surrounding the combustion zone of the pilot. With an increased quantity of the pilot fuel, for the same equivalence ratio, the volume of the pilot is be enlarged and hence the fraction of the gaseous fuel-air charge that undergoes partial oxidation will increase proportionally.

As shown in Figs. 4.4.4 and 4.4.5, when operating the engine on methane with increased CN of the pilot fuels, similar trends with respect to the exhaust emissions are observed. However, conflicting results are found when comparing the effects of different cetane number fuels when the pilot size is considered. Thus, for a 0.20 kg/h pilot, as shown in Figs. 4.4.6 and 4.4.7, the CN 58 fuel improved the utilization of the methane up to 20% and reduced the CH_4 concentration in the exhaust, as compared to the 41.5 CN fuel. The CN 46.5 fuel had a slightly adverse effect. Little differences were found at very light loads, as well as at full load.

The improvement is even larger when increasing the CN of the fuel from 41.5 to 58 and operating the engine above $\phi \sim 0.35$ on a 0.30 kg/h pilot, as represented in Fig. 4.4.8. When using relatively large pilots of 0.40 kg/h, the effect of increasing the CN of the fuel from 41.5 to 58 was reversed. Fig. 4.4.9 indicates that the level of CH_4 concentration in the exhaust is higher when increasing the fuel cetane number. The difference becomes larger around $\phi=0.38$ when the CH_4 concentration peaks, then diminishes and eventually becomes insignificant at very light load. The decrease in the percentage of methane converted was more significant at light and medium loads as shown in Fig. 4.4.10. The reason for this is that with a higher CN pilot displaying a shorter ignition delay, the combustion time is longer but with rapid pilot burning, without a good penetration and diffusion into the gaseous mixture.

With very small pilots, the atomization of the liquid spray is poor and the quenching of the reaction is likely to take place. Figs. 4.4.11-4.4.13 show the CO emissions when operating the engine with increased CN fuels for different pilot quantities. The level of CO concentration in the exhaust is higher with increased cetane number of the smaller pilots. The largest increase was found to be when operating the engine on a 0.30 kg/h pilot within the region of ϕ=0.30÷0.50. For a 0.40 kg/h pilot, the CO concentration recorded when operating with increased CN fuel was lower at light loads (below ϕ=0.40), but became larger at higher loads.

Some of the corresponding variations of CO_2 and O_2 concentrations in dry exhaust are found in Appendix F.

When operating the engine with a nitrogen-methane gas mixture and a 0.20 kg/h pilot, (Figs.4.4.14-4.4.16), the use of a higher CN pilot improved the methane utilization and lowered the unburnt CH_4 levels in the exhaust throughout the whole engine load. However, the CO emissions were increased for light and medium loads which is indicative of the slower burning rates associated with the presence of nitrogen.

The CO_2 addition to the intake methane-air mixture when using a higher pilot at high load did not influence the CH_4 emissions or the percent of CH_4 converted. A decrease in the CH_4 concentration in the exhaust gases was observed for lower loads and a decrease in the CO emissions level for all methane-air mixture concentrations considered when a higher CN pilot is used (Figs.4.4.17-4.4.19).

As expected, the levels of CO emissions were relatively higher when operating the engine on propane as compared to methane operation. It was found that an increase in the CN of the pilot fuel has no influence on the CO concentration in the exhaust when operating the engine on propane with a 0.30 kg/h pilot (Fig. 4.4.20).

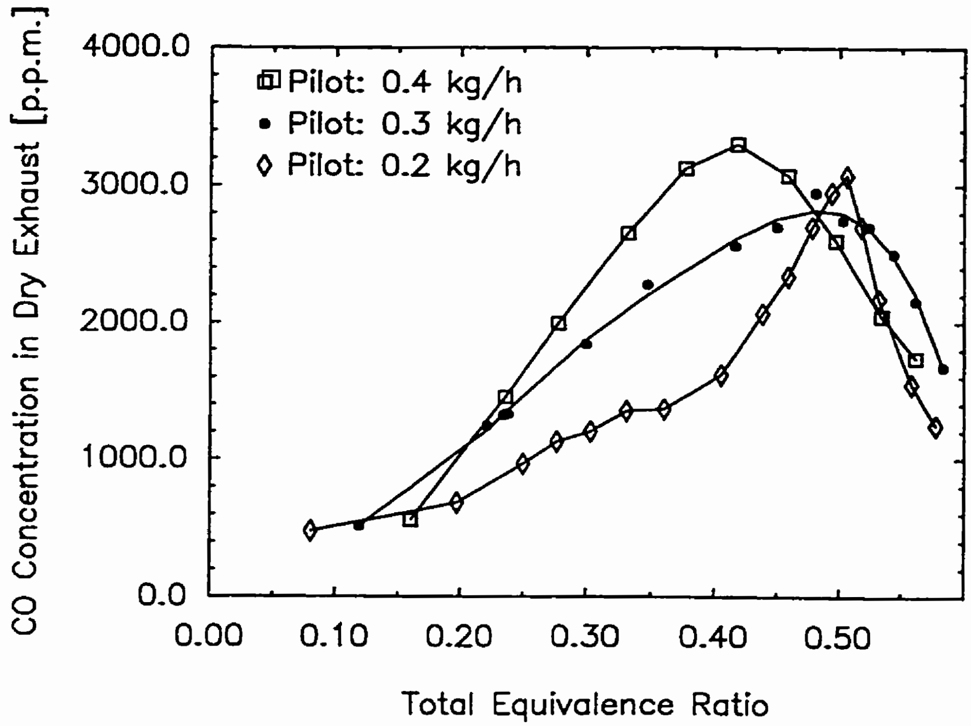


Figure 4.4.1 Variation of Dry Exhaust CO Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using a 41.5 Cetane Number Fuel with Different Pilot Quantities

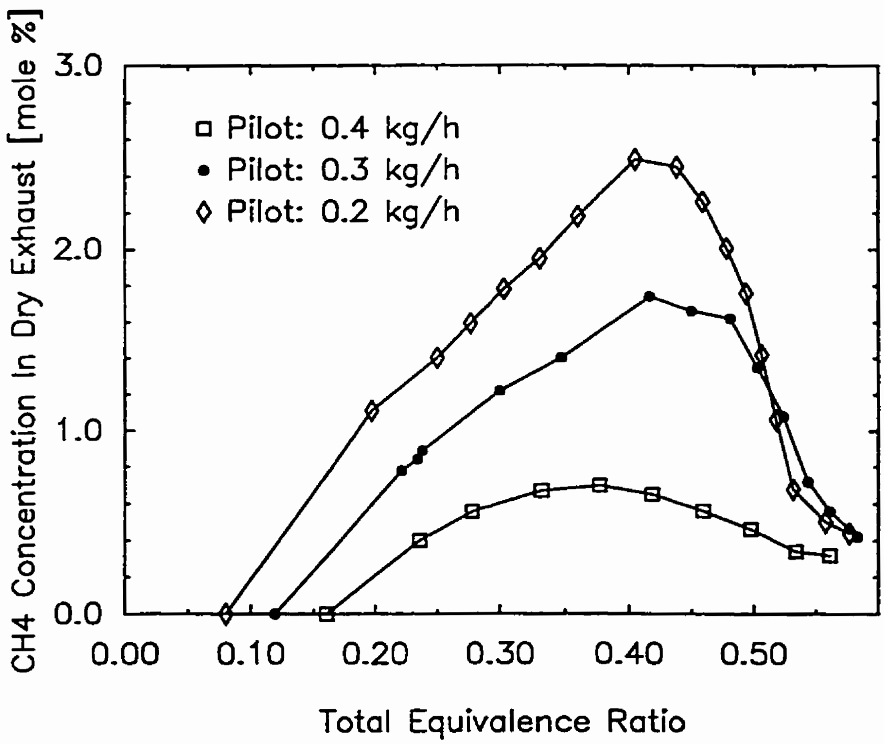


Figure 4.4.2 Variation of Dry Exhaust CH4 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using a 41.5 Cetane Number Fuel with Different Pilot Quantities

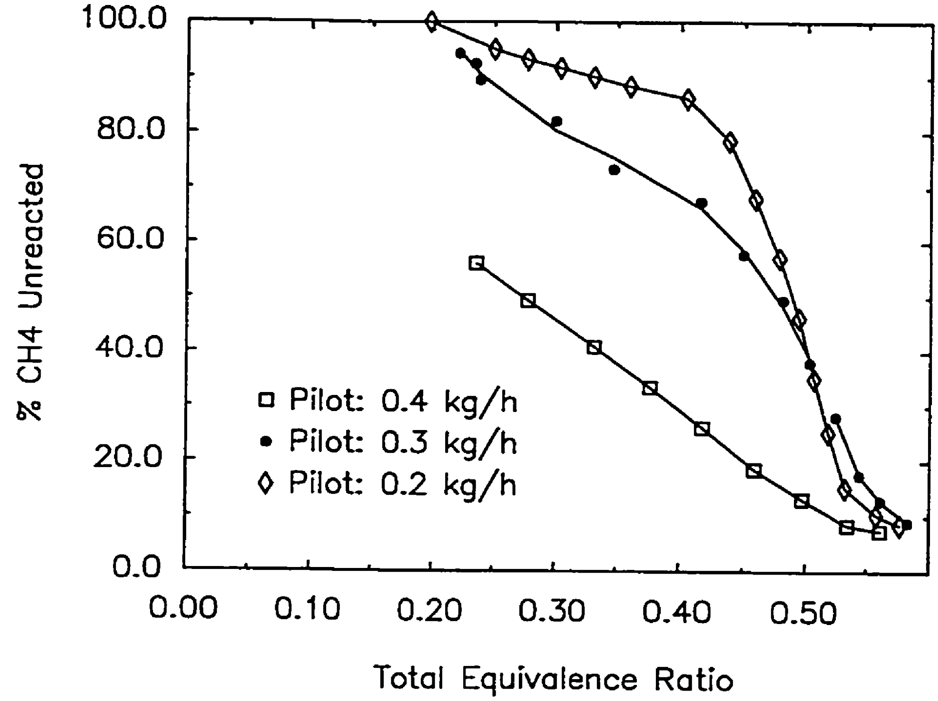


Figure 4.4.3 Variation of the Percent of CH4 Unreacted with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using a 41.5 Cetane Number Fuel with Different Pilot Quantities

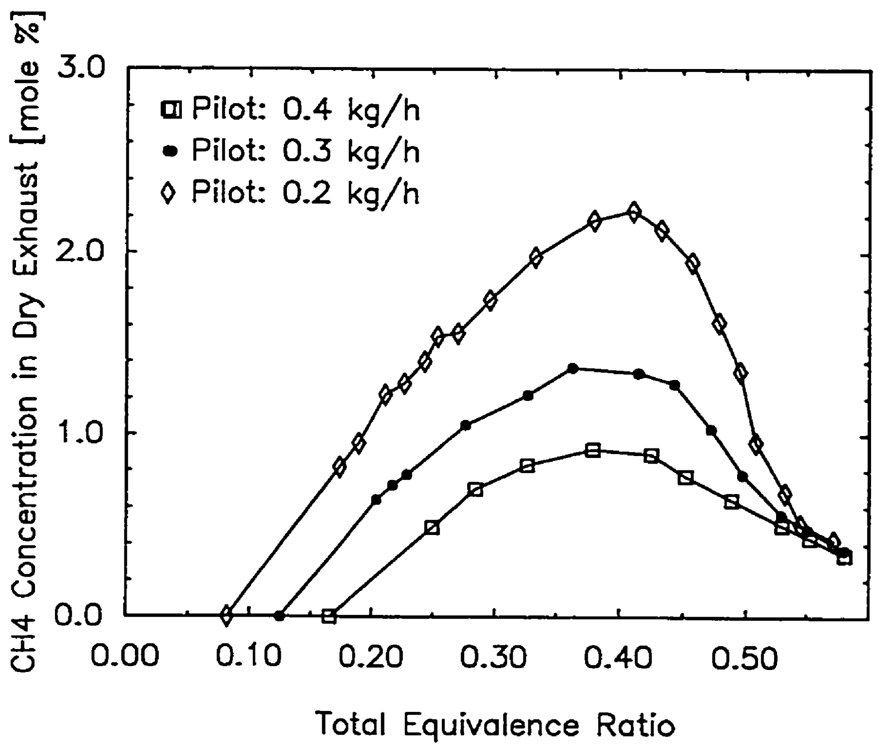


Figure 4.4.4 Variation of Dry Exhaust CH4 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using a 58.0 Cetane Number Fuel with Different Pilot Quantities

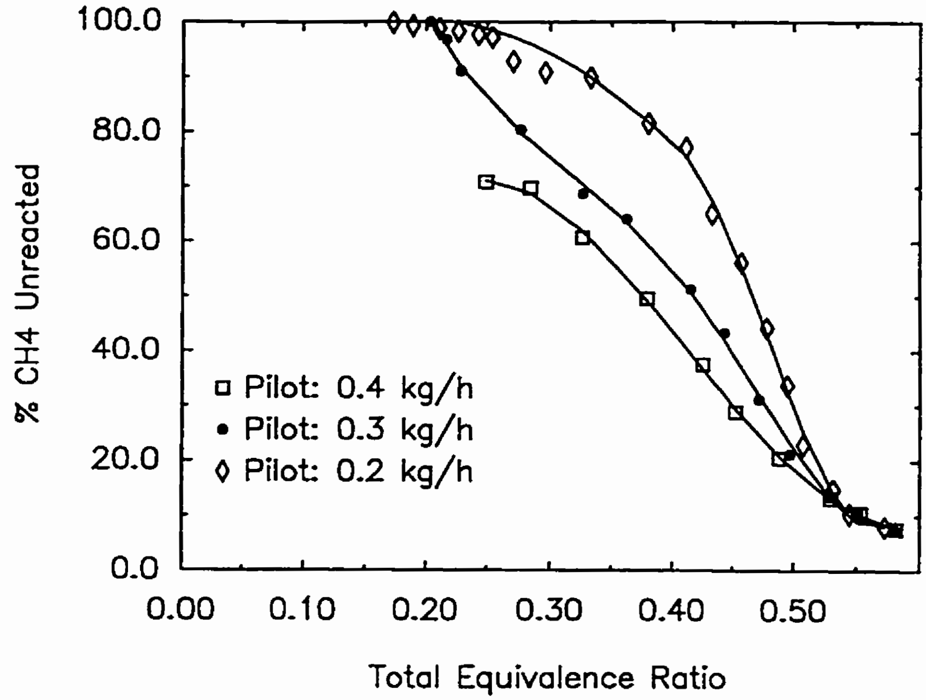


Figure 4.4.5 Variation of the Percent of CH4 Unreacted with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using a 58.0 Cetane Number Fuel with Different Pilot Quantities

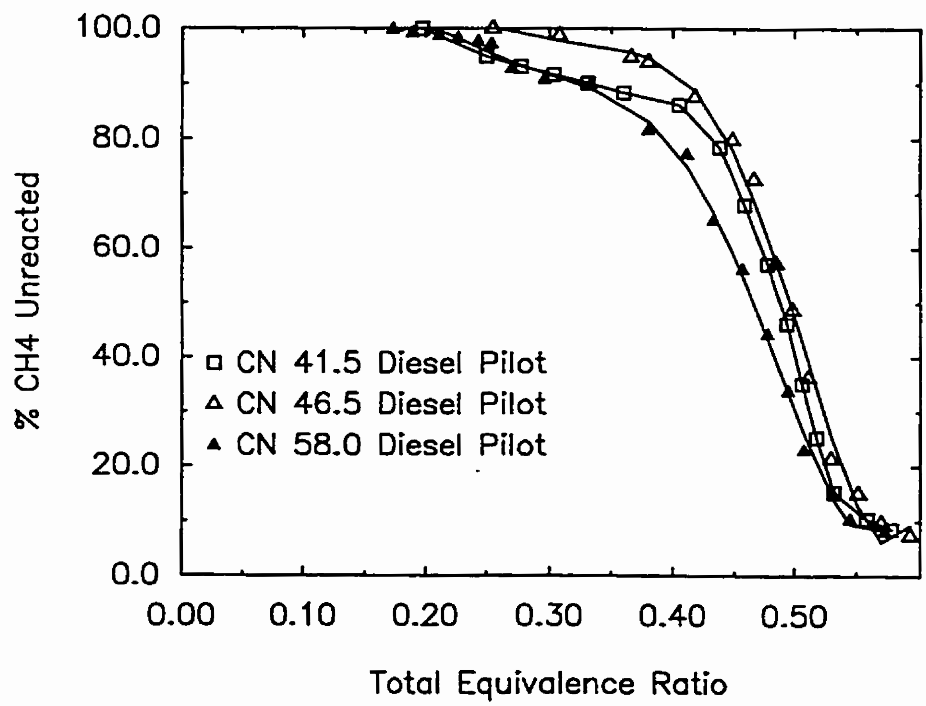


Figure 4.4.6 Variation of the Percent of CH4 Unreacted with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

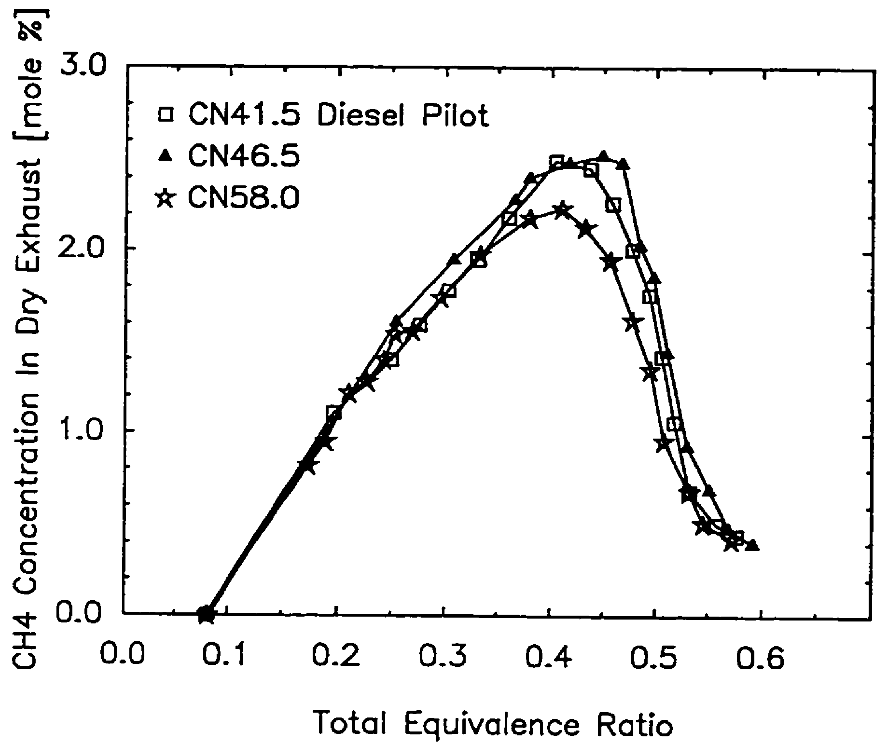


Figure 4.4.7 Variation of Dry Exhaust CH4 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

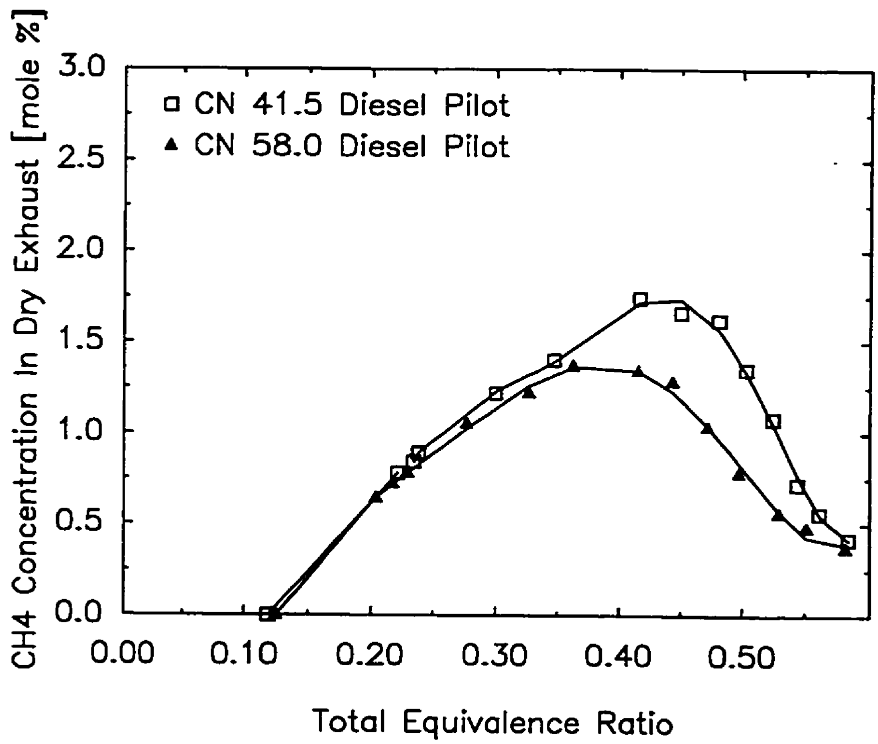


Figure 4.4.8 Variation of Dry Exhaust CH4 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.30 kg/h Pilot

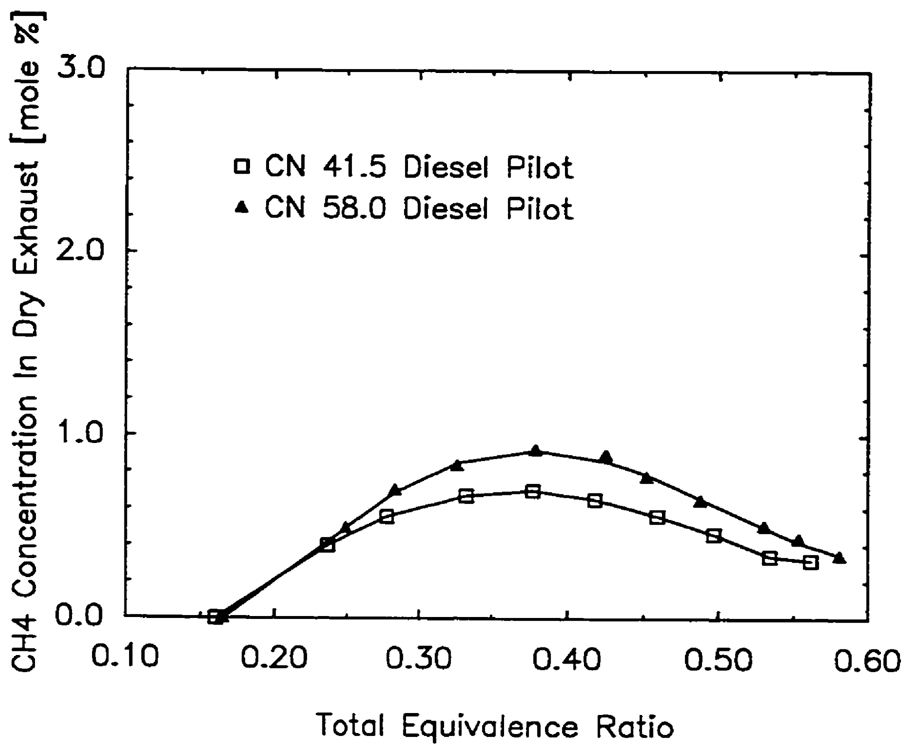


Figure 4.4.9 Variation of Dry Exhaust CH4 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.40 kg/h Pilot

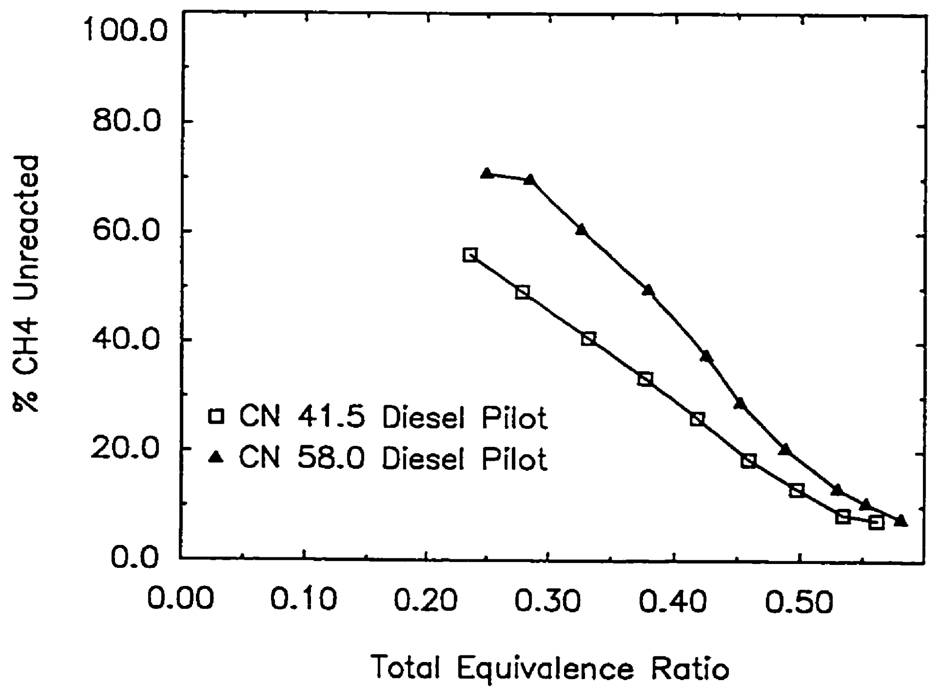


Figure 4.4.10 Variation of the Percent of CH4 Unreacted with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.40 kg/h Pilot

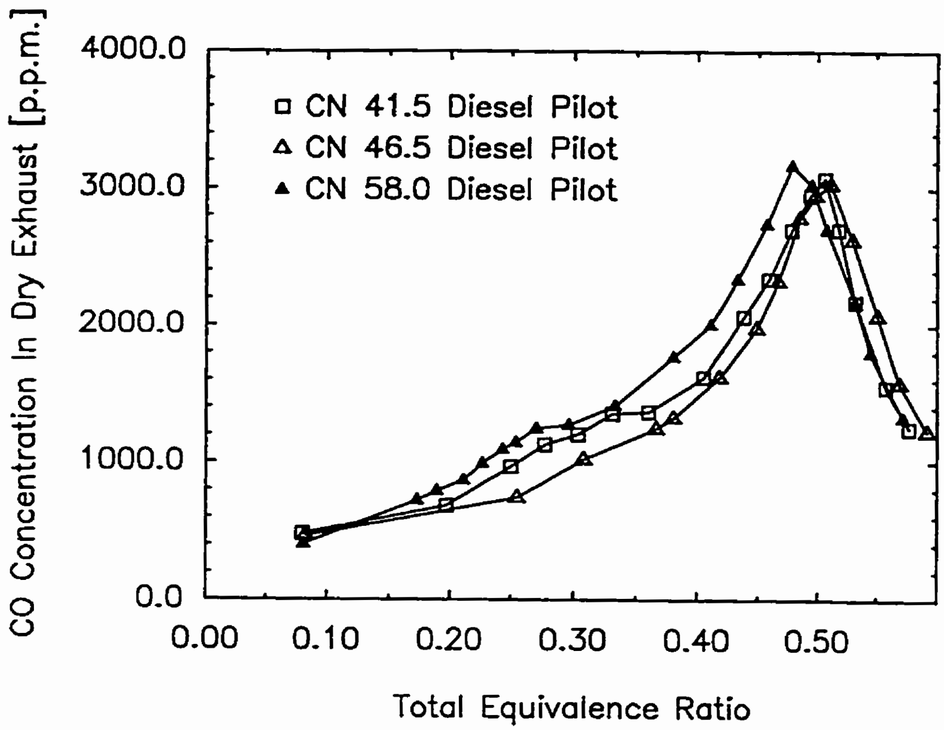


Figure 4.4.11 Variation of Dry Exhaust CO Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

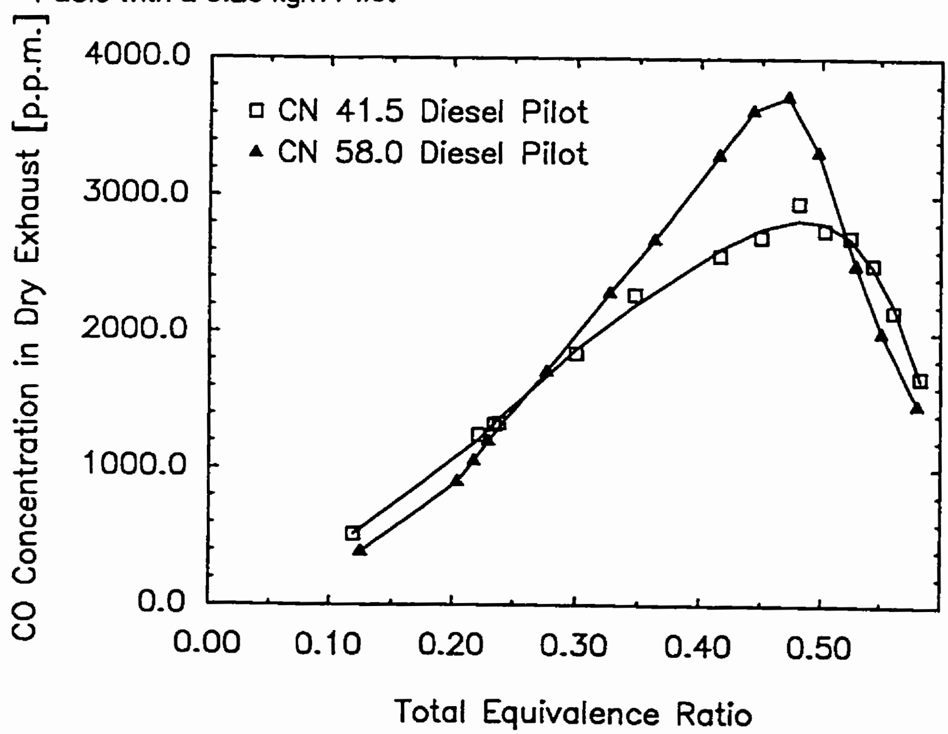


Figure 4.4.12 Variation of Dry Exhaust CO Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.30 kg/h Pilot

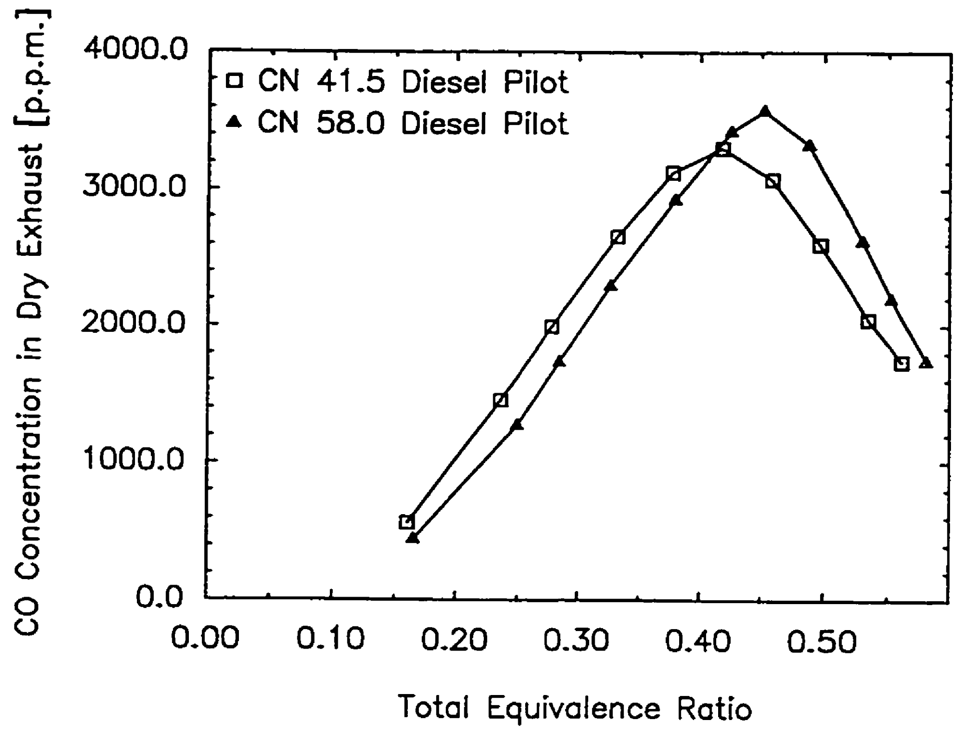


Figure 4.4.13 Variation of Dry Exhaust CO Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.40 kg/h Pilot

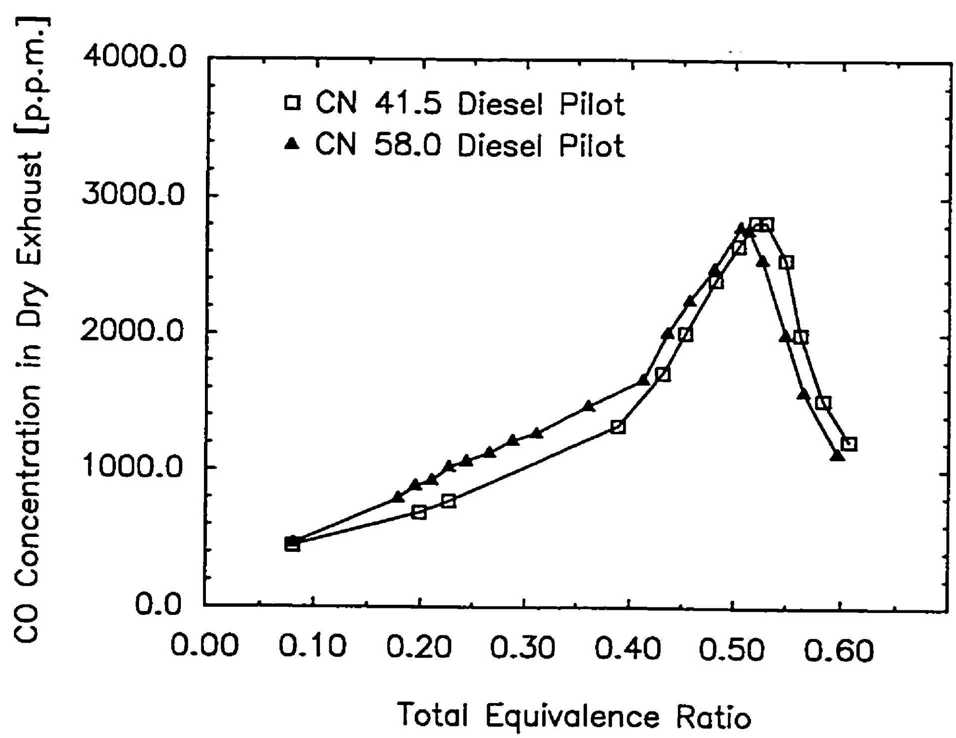


Figure 4.4.14 Variation of Dry Exhaust CO Concentration with Total Equivalence Ratio for Dual Fuel Operation with a CH4-N2 Mixture (50% by Mass CH4) when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

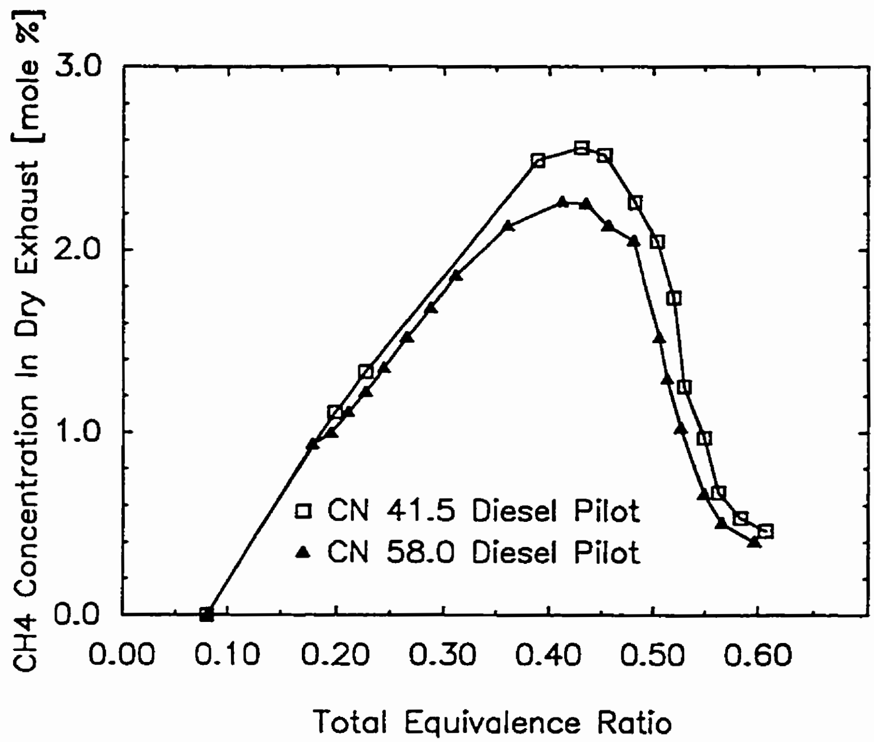


Figure 4.4.15 Variation of Dry Exhaust CH4 Concentration with Total Equivalence Ratio for Dual Fuel Operation with a CH4-N2 Mixture (50% by Mass CH4) when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

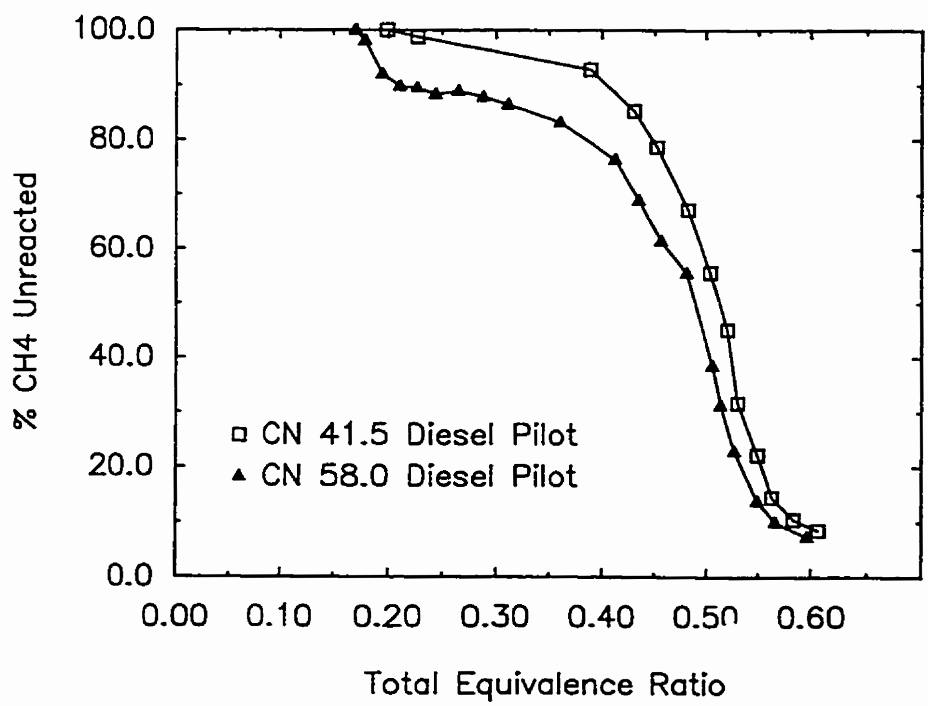


Figure 4.4.16 Variation of the Percent of CH4 Unreacted with Total Equivalence Ratio for Dual Fuel Operation with a CH4-N2 Mixture (50% by Mass CH4) when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

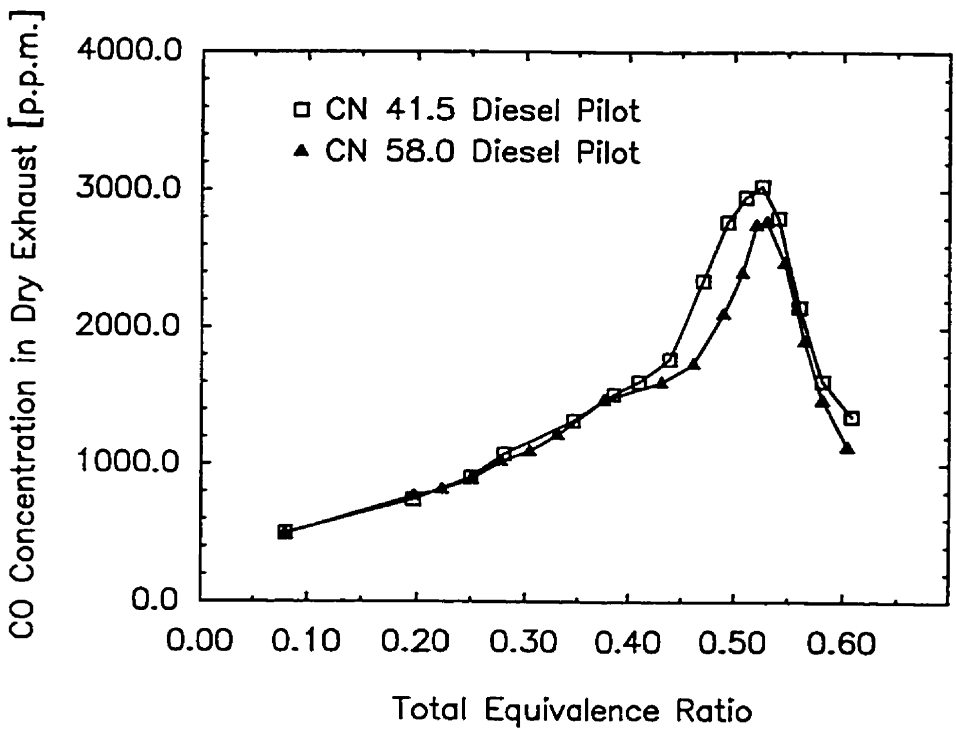


Figure 4.4.17 Variation of Dry Exhaust CO Concentration with Total Equivalence Ratio for Dual Fuel Operation with a CH4-CO2 Mixture (50% by Mass CH4) when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

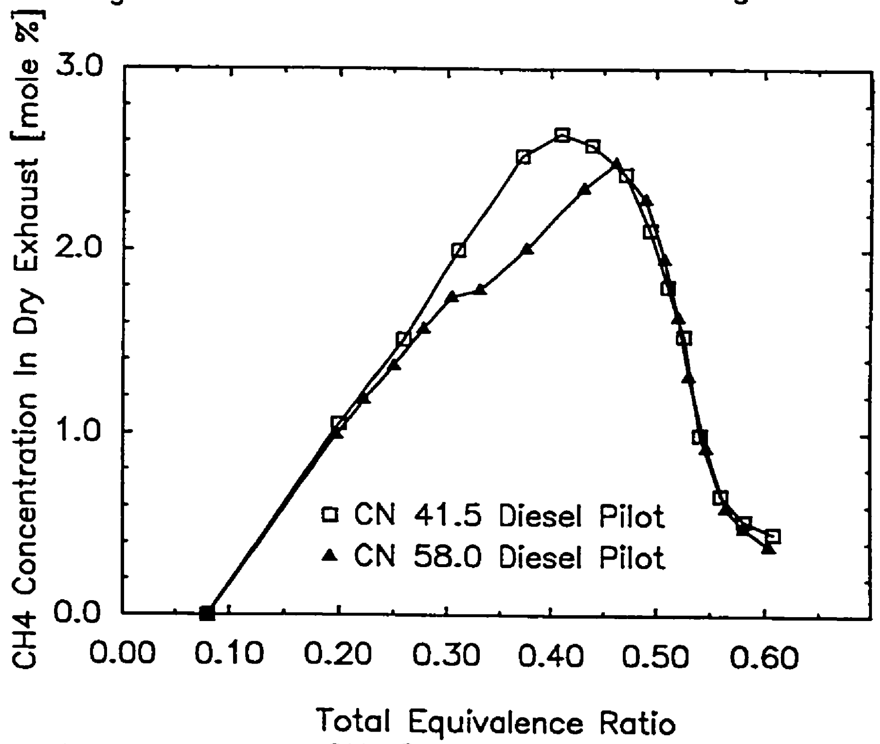


Figure 4.4.18 Variation of Dry Exhaust CH4 Concentration with Total Equivalence Ratio for Dual Fuel Operation with a CH4-CO2 Mixture (50% by Mass CH4) when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

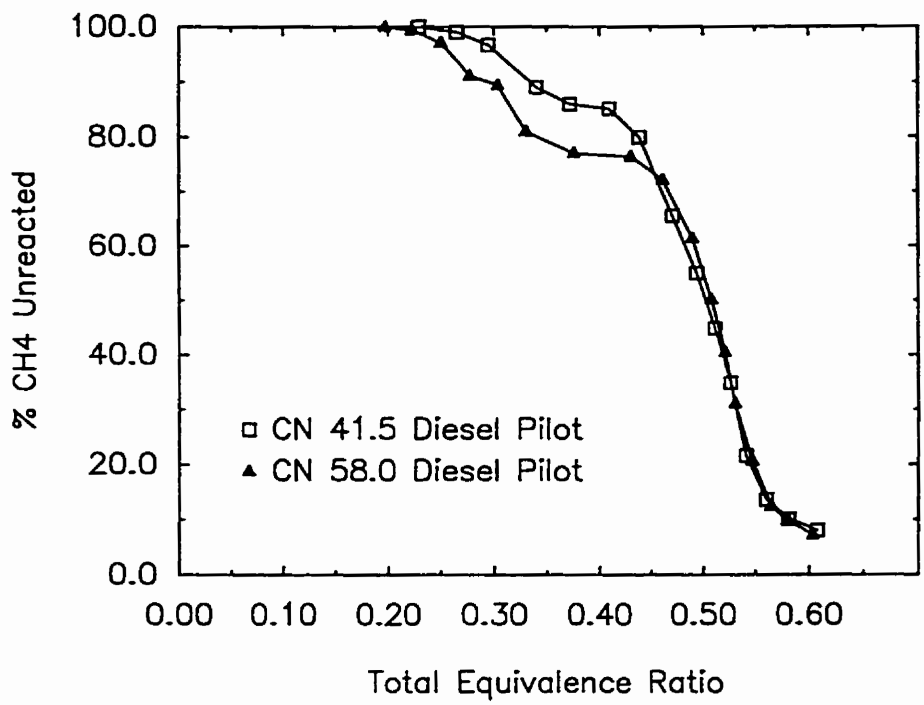


Figure 4.4.19 Variation of the Percent of CH4 Unreacted with Total Equivalence Ratio for Dual Fuel Operation with a CH4-CO2 Mixture (50% by Mass CH4) when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

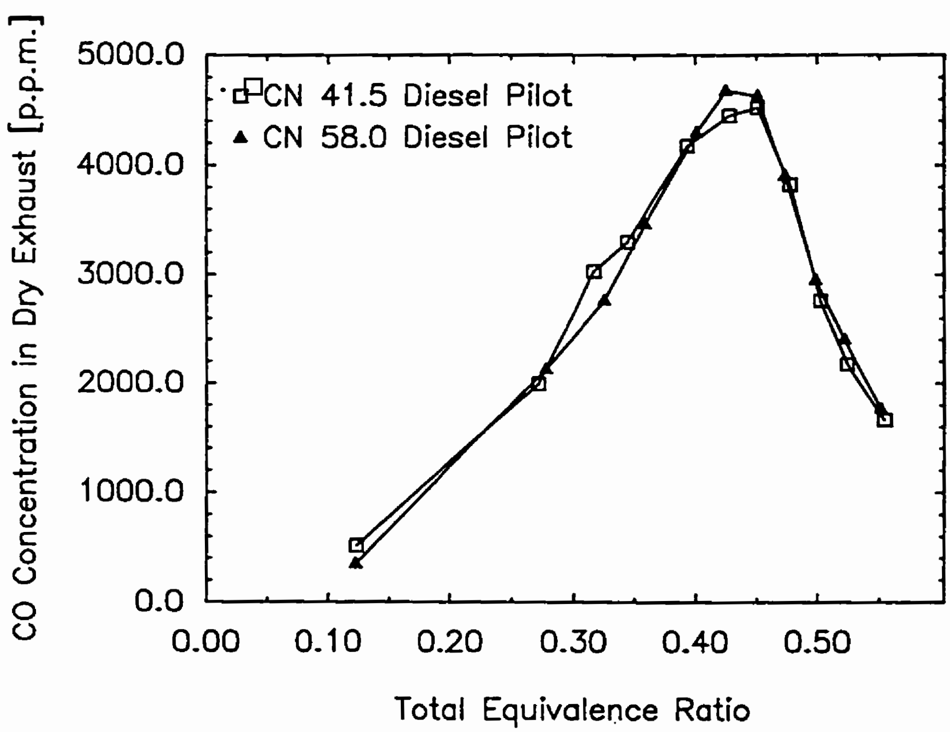


Figure 4.4.20 Variation of Dry Exhaust CO Concentration with Total Equivalence Ratio for Dual Fuel Operation with Propane when Using Different Cetane Number Fuels with a 0.30 kg/h Pilot

CHAPTER FIVE

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

The results of this study showed that the quality and quantity of the diesel pilot fuel can have significant effects on the performance of a gas fuelled diesel engine. Some of these are listed below:

- The use of relatively small pilot quantities with higher cetane number fuels can lead to an improved combustion at light loads, lower rates of pressure rise and smoother operation.
- The ignition delay in gas fueled diesel engines of the dual fuel type depends strongly on both the quantity and quality of the pilot fuel used.
- The superior autoignition behavior of higher cetane number diesel fuels is not inhibited by the presence of a gaseous fuel in the surrounding atmosphere within dual fuel engines. The reduction in the ignition delay period during dual fuel operation was more evident at light loads.
- The use of high cetane number pilots permits the employment of smaller pilot quantities and can improve engine operations on propane and low heating value gaseous fuel mixtures compared to operation with common diesel fuels having a relatively low cetane number. Hence a greater economy of the diesel fuel can be obtained.

- Exhaust gas analysis showed that the increase in the cetane number of the diesel fuel when operating the engine on methane or low heating value gaseous fuel mixtures with smaller pilots can improve the gaseous fuel utilization. However, the corresponding levels of carbon monoxide concentration in the exhaust were found to be higher when using higher cetane number pilots.

Recommendations

- Further experimental and analytical work is needed for establishing the influence of other liquid fuel characteristics besides cetane number (e.g. aromatic content) on dual fuel engine performance.
- A complete exhaust emission analysis would require the measurement of oxides of nitrogen even when they are expected to be lower than under diesel operation.
- Improvements to the engine electric loading system are needed to accommodate higher loads and to enable engine more conveniently and reliably partial motoring conditions.
- A computerized data acquisition system of sufficient capacity and accuracy is strongly recommenced to be employed in the testing to ensure a much more reliable and easier way of obtaining the necessary information about engine cylinder pressure-time events.

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APPENDIX A

1. Calibration of the Choked Nozzles for Metering Gases

Choke conditions are established in a nozzle when the velocity of the gas flowing through it is sonic at the nozzle throat.

From the theory of isentropic steady flow of ideal gases in nozzles it can be shown that for chocked conditions the mass flow rate of a gas through a well rounded nozzle is given by:

$$m = C_D \frac{AP_0}{\sqrt{RT_0}} \sqrt{\gamma \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma - 1}{\gamma - 1}}}$$

where:

m - mass flow rate of the gas

C_D - coefficient of discharge (to account for a real flow)

A - cross sectional area at the throat

P_0 - absolute upstream pressure

R - universal gas constant

T_0 - absolute upstream temperature

γ - specific heats ratio (C_P/C_V) of the gas

Hence,

For a specific nozzle and gas (A, C_D, λ, R are constant), under choked conditions, the gas mass flow rate is a function of the upstream conditions only:

$$m = K \frac{P_0}{\sqrt{T_0}}$$

where K is a constant for a particular nozzle and gas.

$$K = C_D \frac{A}{\sqrt{R}} \sqrt{\gamma \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}}$$

The nozzles used for metering gases in the engine experiments were calibrated for air using the orifice meter. The calibration curves are represented in Fig. A.1.1. There is a linear relationship between the mass flow rate and the upstream ratio $P_0/\sqrt{T_0}$.

When the nozzles are to be used for metering other gases, the value of coefficient K will change.

For a particular meter, assuming that the coefficient of discharge over a limited working range is dependent only on the specific meter being used and is not considerably influenced by the properties of the gas metered, for the same operating conditions of temperature and pressure, it can be shown that:

$$\frac{K_g}{K_a} = \sqrt{\frac{M_g \gamma_g \left(\frac{2}{\gamma_g + 1}\right)^{\frac{\gamma_g - 1}{\gamma_g - 1}}}{M_a \gamma_a \left(\frac{2}{\gamma_a + 1}\right)^{\frac{\gamma_a \; 1}{\gamma_a - 1}}}}$$

where:

- subscript "g" refers to any particular gas and "a" refers to air

- M_g, M_a - molecular weights of the gas and air

This enables the use of the nozzles for metering various gases without recalibration. The only requirement is the calculation of coefficient K for each particular gas.

Listed below are the values of γ, M and coefficient K for metering various gases:

	Air	CH_4	C_3H_8	CO_2	N_2
γ	1.399	1.304	1.128	1.288	1.400
M	28.967	16.043	44.097	44.011	28.016
K #9	0.552	0.401	0.631	0.661	0.543
K #10	0.257	0.187	0.294	0.308	0.253
K #13	0.238	0.173	0.272	0.285	0.234

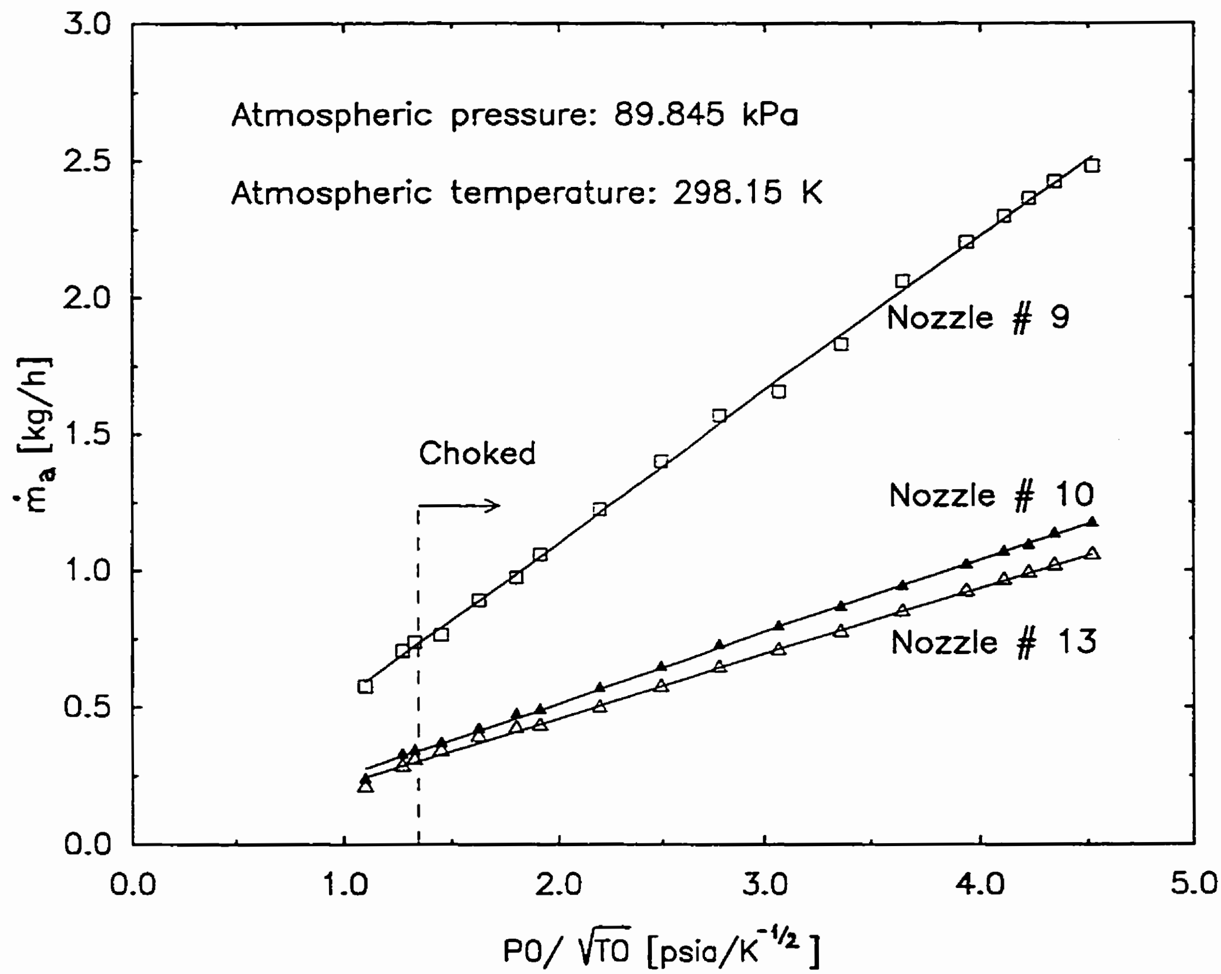

Figure A.1.1 Calibration of Choked Nozzles for Metering Gases

APPENDIX B

1. Calibration of the viscous flow air meter

The air flow mater consists of a tube containing a matrix of narrow triangular passages thus rendering the air flow viscous.

From the theory of laminar flow in pipes (Hagen - Poiseuille) it can be shown that the pressure drop due to the viscous flow in a pipe of fixed geometry is proportional to the mass flow rate of the fluid of invariant viscosity and density.

$$\Delta p = m\frac{8\mu L}{\pi\rho r^4} \tag{B.1}$$

where:

Δp - pressure drop across length L

m - mass flow rate of the fluid

μ - viscosity of the fluid

ρ – density of the fluid

L - length of the tube

r - radius of the tube

Therefore, the air mass flow rate can be determined from the knowledge of the pressure drop across the tube and the relation is linear.

$$m = \Delta p \cdot K$$

where:

$$K = \frac{\pi \rho r^4}{8 \mu L}$$

and can be determined experimentally.

The meter was calibrated using a standard orifice and a sufficiently large tank. The calibration curve is shown in Fig. B.1.1. The straight line function obtained experimentally confirms the above mentioned theoretical equation.

2. Calculation of the Correction Factor for Viscous Flow Air Meter Calibration Curve

The calibration of the viscous flow meter is affected by the daily changes in viscosity and density of the atmospheric air due to the changes in ambient temperature (T) and pressure (p). To quantify the effect of such changes, a correction factor has to be calculated.

Let

$$(\frac{m}{\Delta p})_c$$

be the slope of the calibration curve on the day of the calibration

and

$$(\frac{m}{\Delta p})_t$$

be the slope of the calibration curve for the day of the test.

From the equation (B.1) it can be shown that:

$$(\frac{m}{\Delta p})_t = (\frac{m}{\Delta p})_c \cdot (\frac{\rho_t}{\rho_c})(\frac{\mu_c}{\mu_t}) = (\frac{m}{\Delta p})_c \cdot (\frac{p_t}{p_c})(\frac{T_c}{T_t})(\frac{\mu_c}{\mu_t})$$

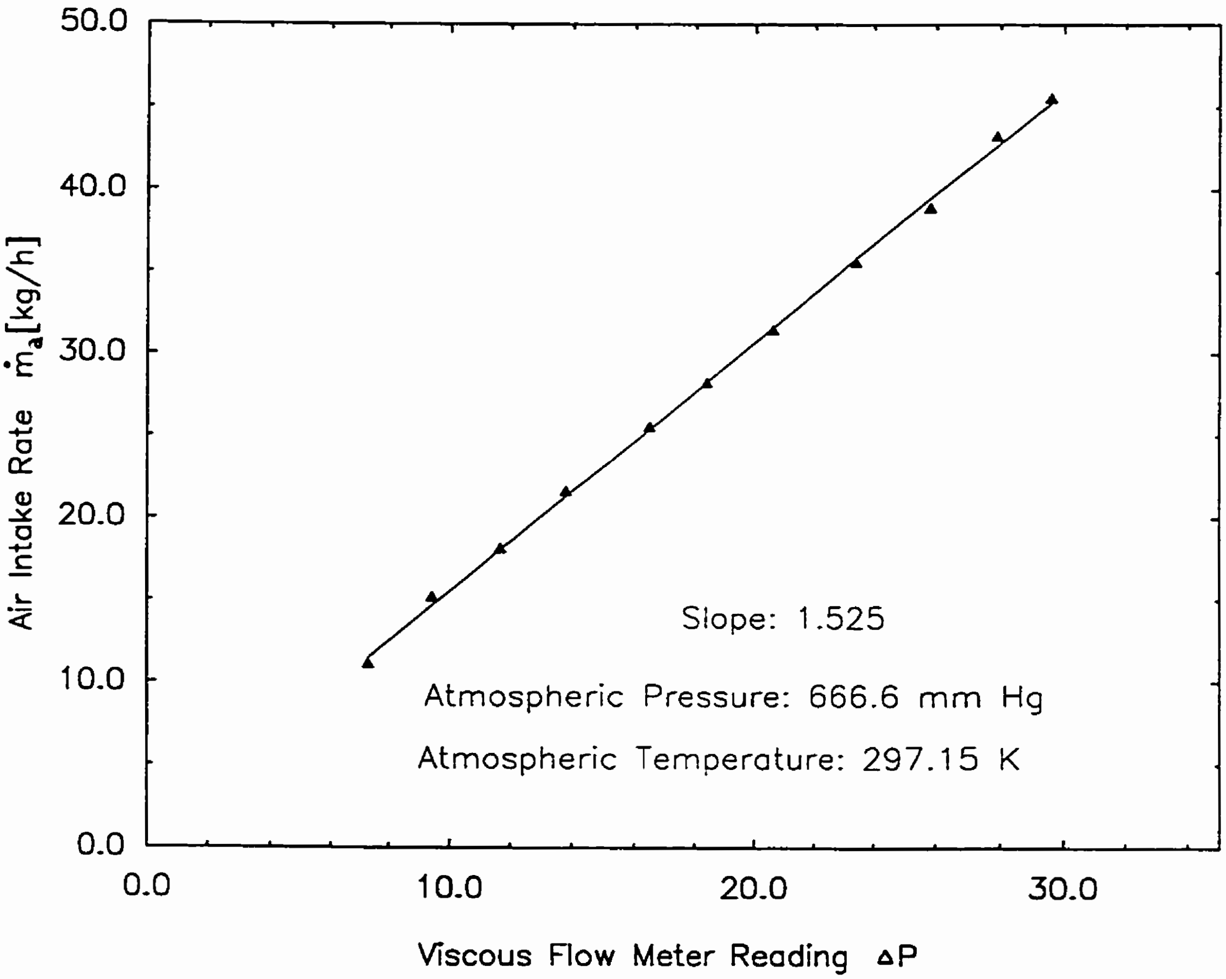

Fig. B.1.1. Calibration of the Viscous Flow Air Meter

APPENDIX C

Calculation of Total Equivalence Ratio

Equivalence ratio (ϕ) is defined as the ratio of actual fuel-oxygen ratio to the stoichiometric fuel-oxygen ratio (mass basis):

$$\phi=\frac{(\text{Fuel/Oxygen})_{\text{Actual}}}{(\text{Fuel/Oxygen})_{\text{Stoichiometric}}}=\frac{\Sigma(\text{Oxygen})_{\text{Stoichiometric}}}{\Sigma(\text{Oxygen})_{\text{Actual}}}$$

The stoichiometric reaction of combustion for any hydrocarbon fuel having the formula C_nH_m is:

$$C_nH_m+(n+\frac{m}{4})O_2 \rightarrow nCO_2+\frac{m}{2}H_2O$$

Therefore, the stoichiometric oxygen required for m_{HC} (kg/h) of hydrocarbon would be:

$$\frac{(n+\frac{m}{4})32}{12n+m}\times m_{HC} \qquad \text{(kg/h)}$$

The total stoichiometric oxygen required for the combustion of any number of such hydrocarbons will be:

$$\Sigma(\frac{32n+8m}{12n+m})\times m_{HC} \qquad \text{(kg/h)}$$

Hence,

$$\phi = \frac{\Sigma(\frac{32n+8m}{12n+m})\times m_{HC}}{0.233\times m_a}$$

where m_a (kg/h) is the air mass flow rate.

Listed below are the formulae employed for calculating the equivalence ratio when using different gaseous fuels with diesel pilot. For all practical purposes, diesel fuel was treated as cetane which has the chemical formula $C_{16}H_{34}$ and natural gas as methane, CH_4. The errors introduced by such an approximation are insignificant.

a) Methane as main gaseous fuel with Diesel pilot

$$\phi = \frac{17.167\times m_g + 14.888\times m_D}{m_a}$$

b) Propane as main gaseous fuel with Diesel pilot

$$\phi = \frac{15.607\times m_g + 14.888\times m_D}{m_a}$$

APPENDIX D

Calculation of the Unreacted Methane

The combustion reaction of methane and diesel pilot in air can be written as follows:

$$(\frac{m_m}{16})CH_4+(\frac{m_D}{226})C_{16}H_{34}+\frac{m_a}{28.96}(0.21O_2+0.79N_2)\rightarrow$$
$$\rightarrow aCO_2+bCO+cN_2+dCH_4+eNO_x+fH_2O+gO_2+hC$$

where m_m, m_D, m_a are the mass flow rates of methane, diesel and air respectively, and a,b,c,d, etc are the fractions of the respective components in the final products.

The amount of C and NO_X in the exhaust were considered negligible and for the dry exhaust gas analysis f = 0.

Let $[CH_4]$, $[CO]$, $[CO_2]$ be the mole fractions of CH_4, CO, CO_2 respectively, determined experimentally in the dry exhaust.

It can be written that:

$$[CH4]=\frac{d}{a+b+c+d+g}$$

$$[CO]=\frac{b}{a+b+c+d+g}$$

$$[CO_2]=\frac{a}{a+b+c+d+g}$$

And:

$$d = (a + b + c + d + g) [CH_4]$$

$$a + b + d = (a + b + c + d + g) ([CH_4] + [CO] + [CO_2])$$

Therefore:

$$\frac{d}{a+b+d} = \frac{[CH_4]}{[CH_4]+[CO]+[CO_2]}$$

Writing the carbon mass balance:

$$a+b+c = \frac{m_m}{16} + \frac{16}{226} m_D$$

Hence:

$$d = \left(\frac{m_m}{16} + \frac{16}{226} m_D\right) \frac{[CH_4]}{[CH_4]+[CO]+[CO_2]}$$

Therefore, the percent of methane unreacted will be:

$$\%CH_4 = 100 \times \frac{d}{\frac{m_m}{16}} = \left(1 + \frac{256}{226}\frac{m_D}{m_m}\right) \frac{[CH_4] \times 100}{[CH_4]+[CO]+[CO_2]}$$

Natural gas was supplied by the Canadian Western Natural Gas Co. Ltd. A variation of the natural gas composition from one bottle to another is to be expected. The analysis of a typical sample of natural gas as provided by the C.W.N.G. laboratory was given as:

Constituent	By Volume
CH_4	97.48 %
C_2H_6	1.24 %
C_3H_8	0.30 %
$i\text{-}C_4H_{10}$	0.02 %
N_2	0.71 %
CO_2	0.25 %

In all calculations natural gas was treated as methane.

APPENDIX F

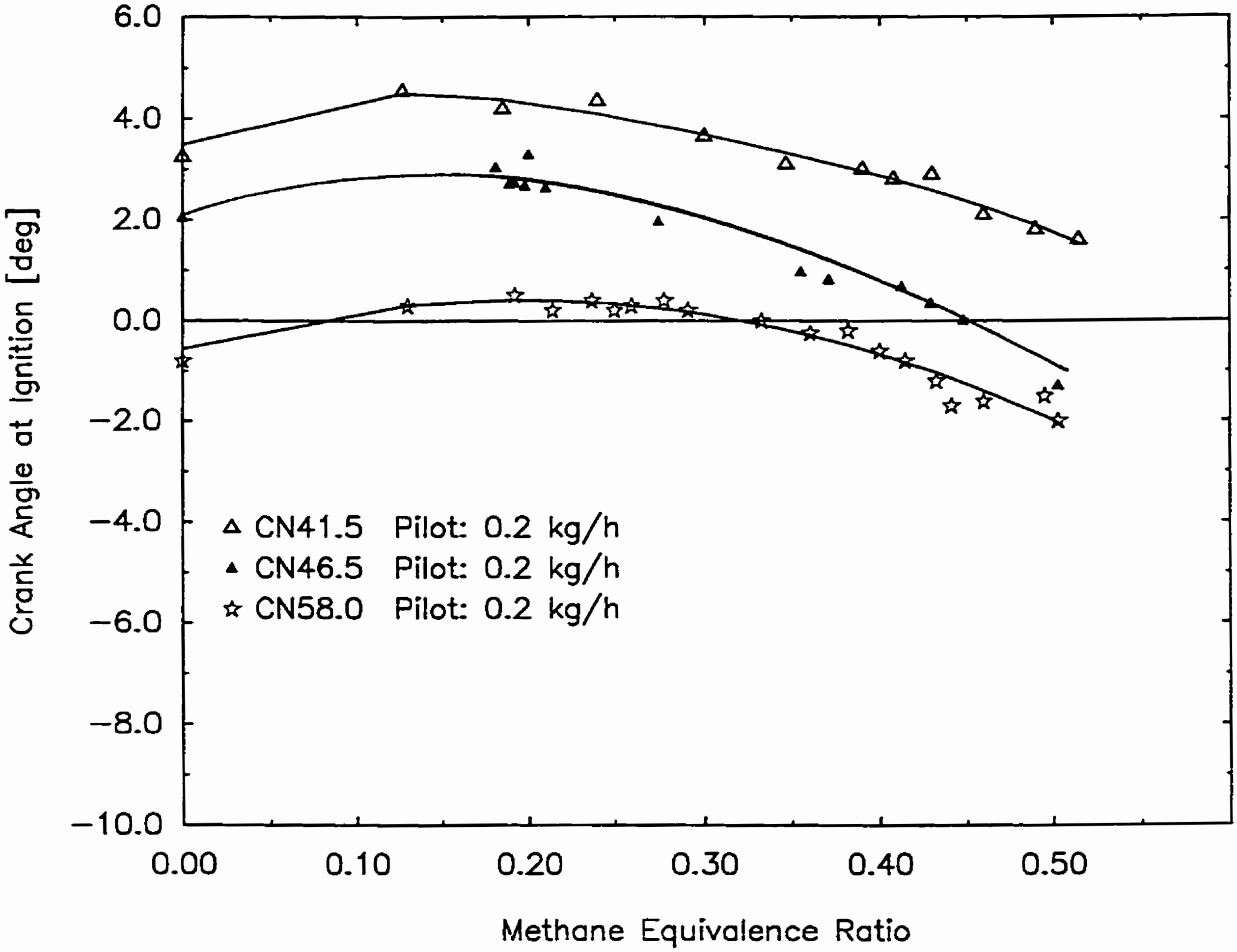

Figure F.1.1 Variations of the Point of Ignition with Gas Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

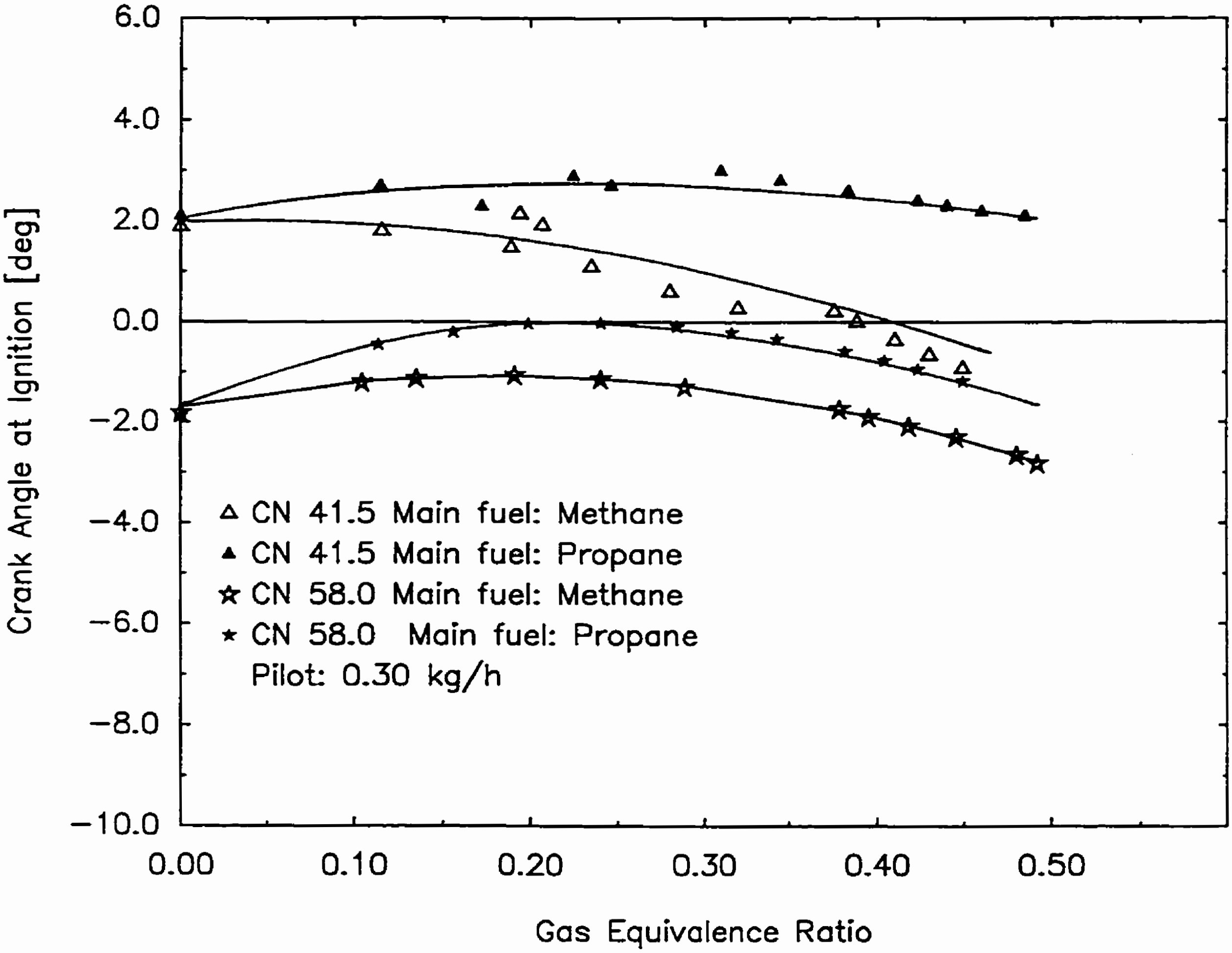

Figure F.1.2 Variations of the Point of Ignition with Gas Equivalence Ratio for Dual Fuel Operation with Propane when Using Different Cetane Number Fuels as Compared to the Corresponding Values Obtained when Operating with Methane.

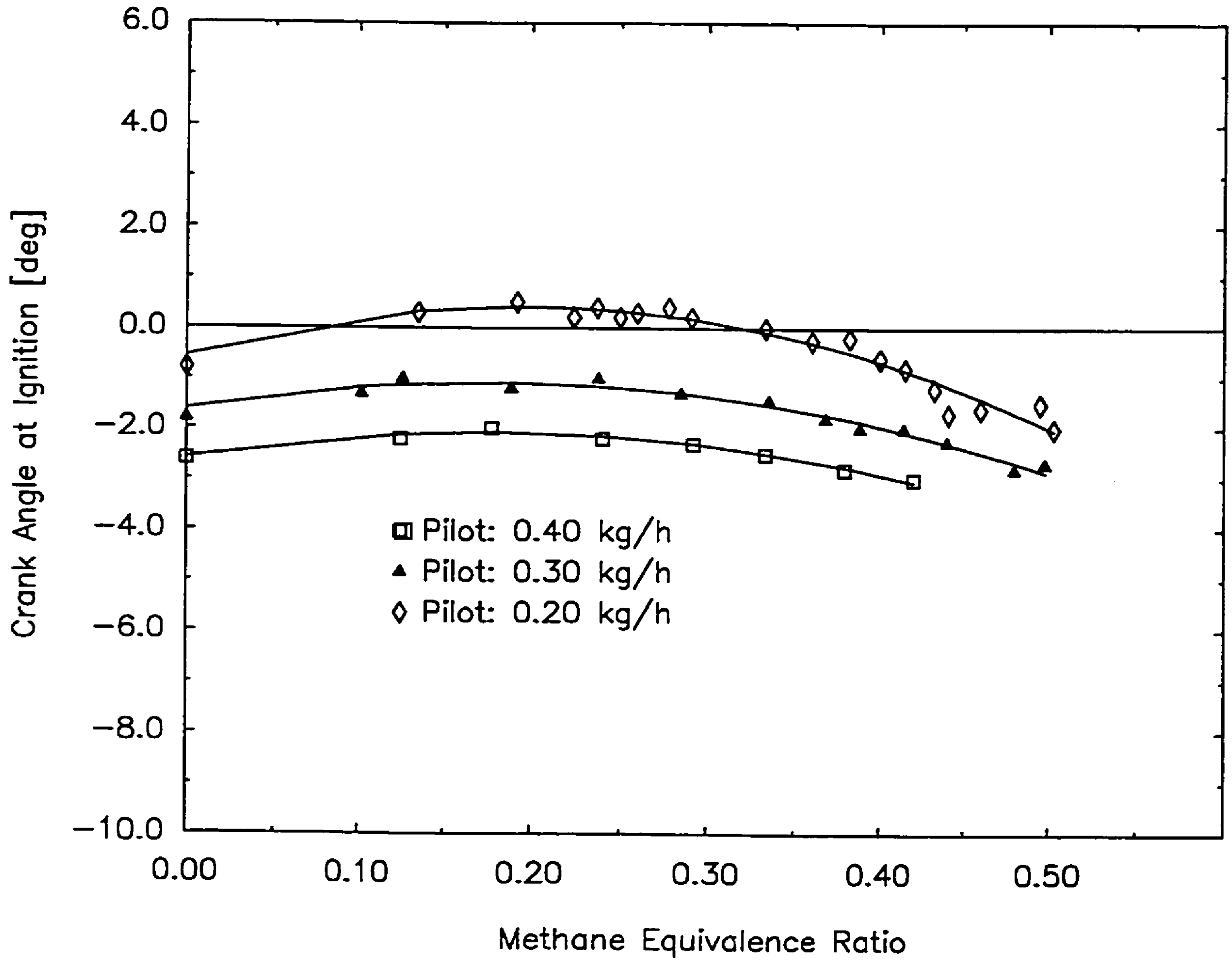

Figure F.1.3 Variations of the Point of Ignition with Gas Equivalence Ratio for Dual Fuel Operation with Methane when Using a 58.0 Cetane Number Fuel with Different Pilot Quantities.

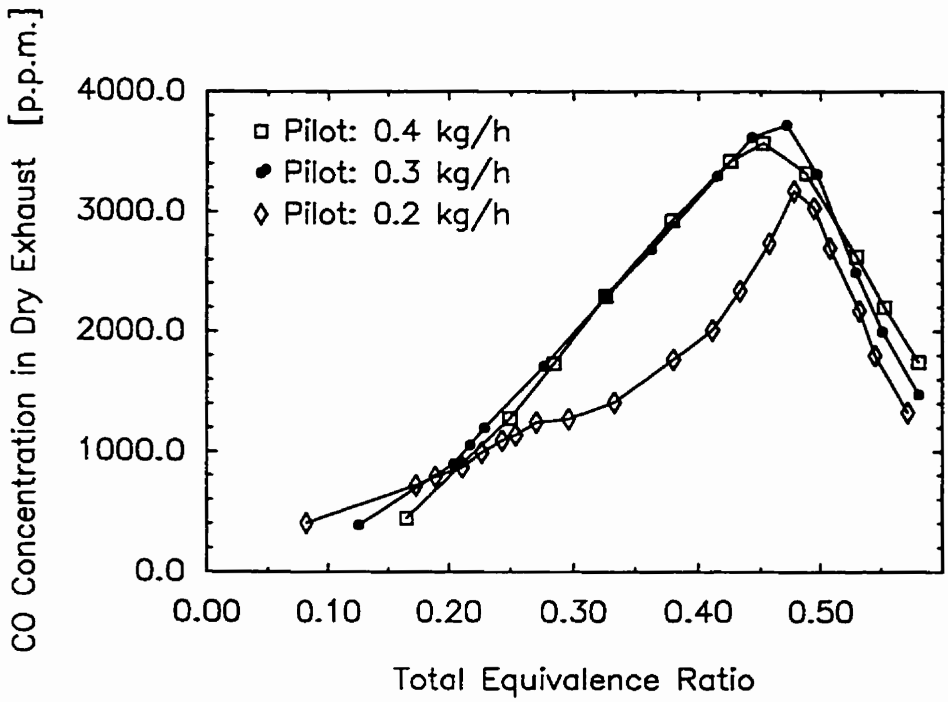

Figure F.1.4 Variations of Dry Exhaust CO Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using a 58.0 Cetane Number Fuel with Different Pilot Quantities.

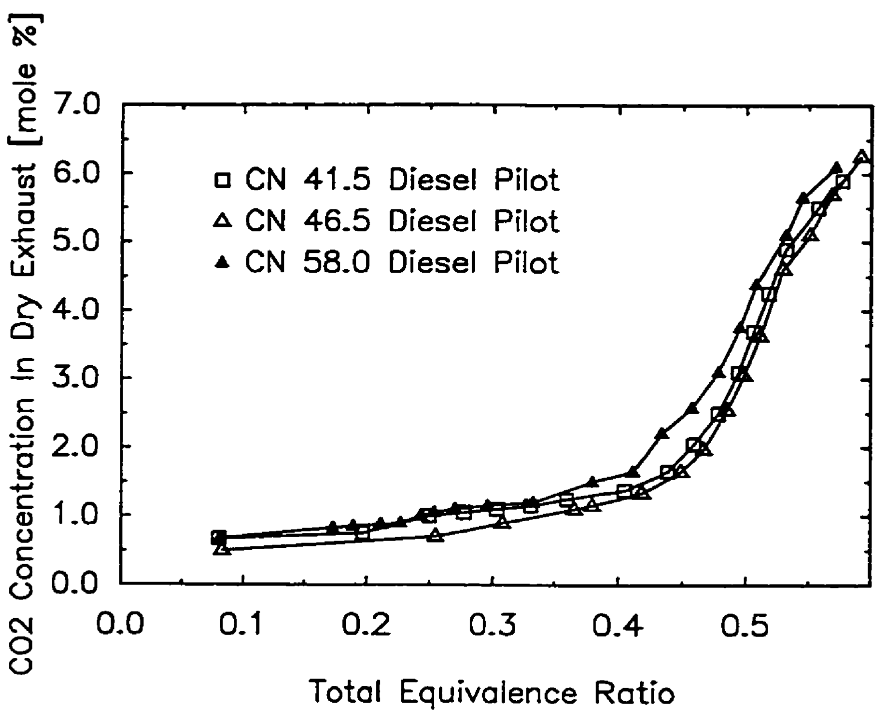

Figure F.1.5 Variation of Dry Exhaust CO2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

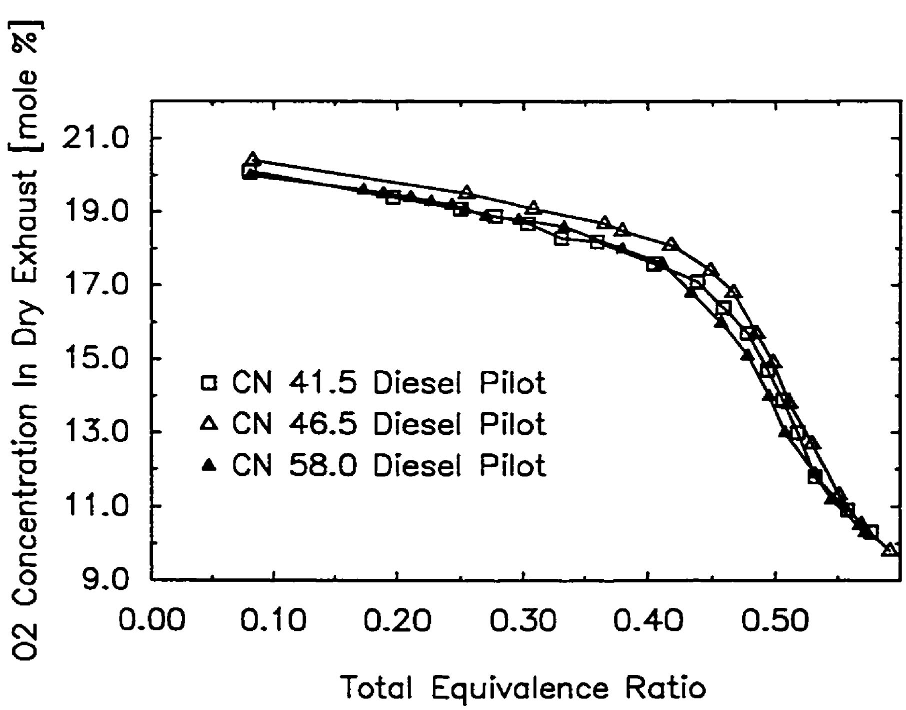

Figure F.1.6 Variation of Dry Exhaust O2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

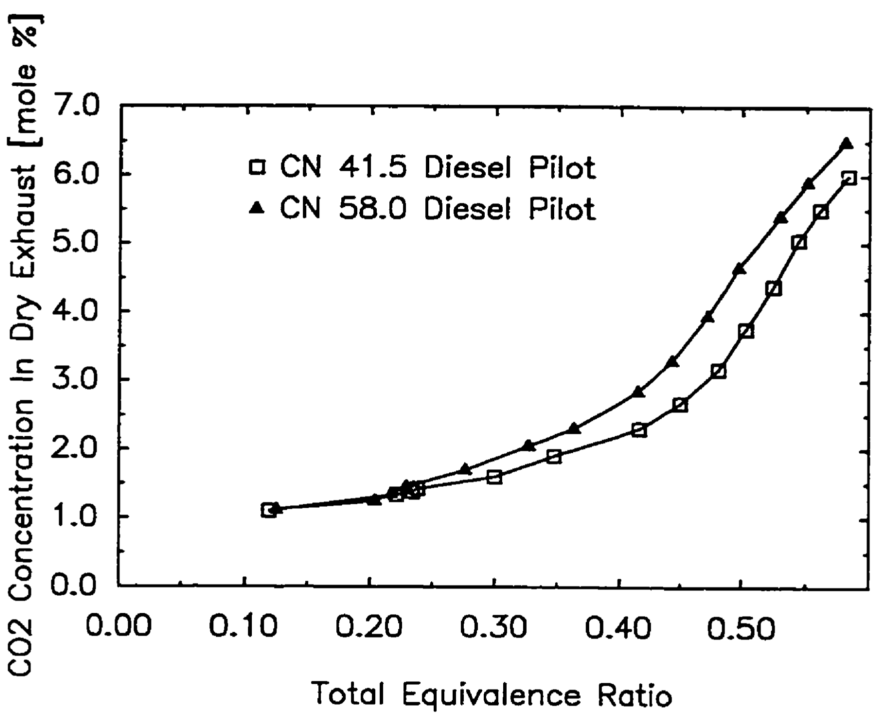

Figure F.1.7 Variation of Dry Exhaust CO2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.30 kg/h Pilot

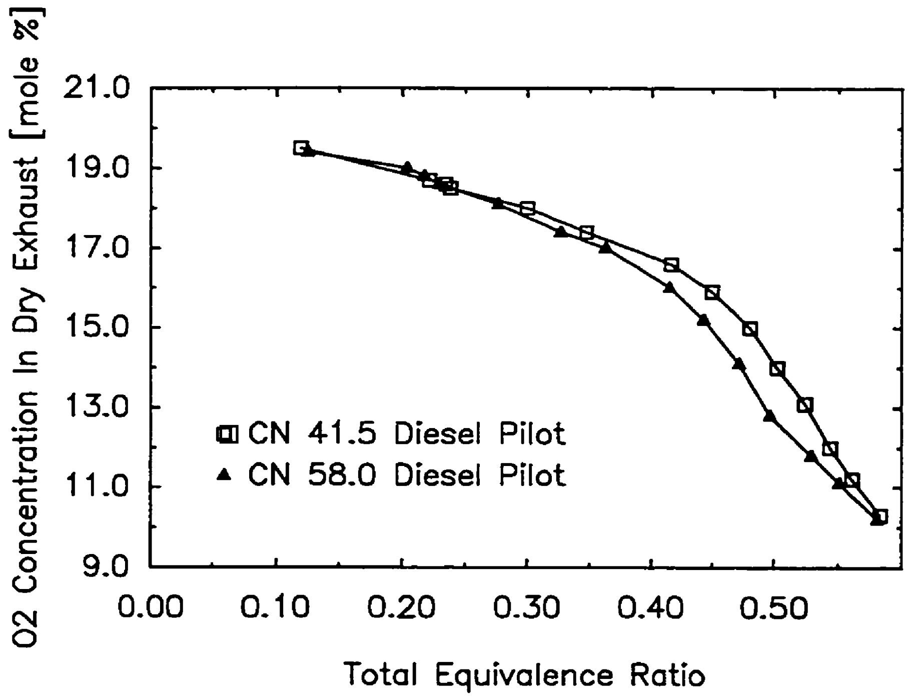

Figure F.1.8 Variation of Dry Exhaust O2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.30 kg/h Pilot

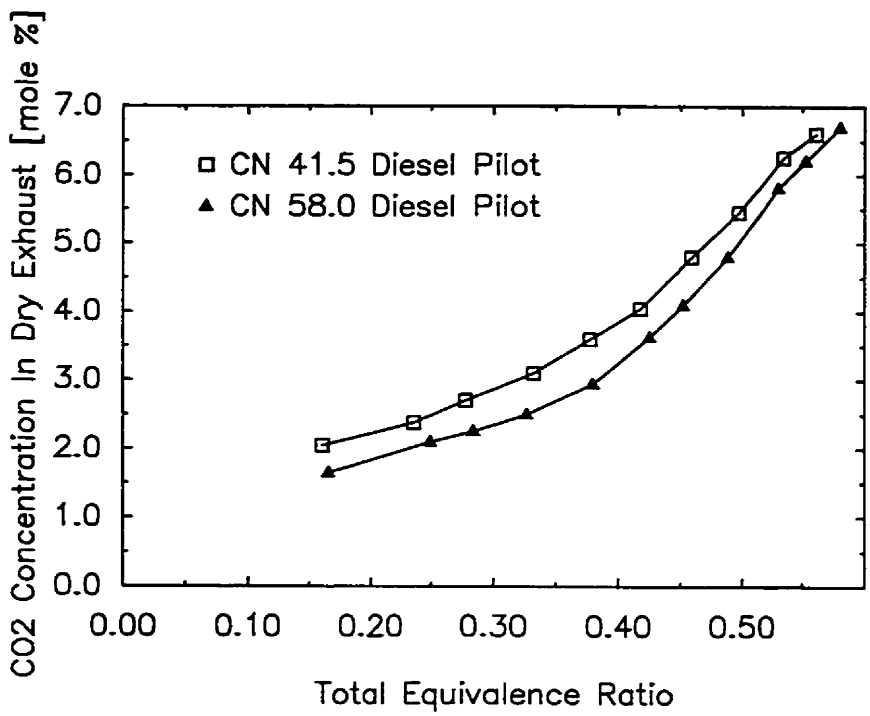

Figure F.1.9 Variation of Dry Exhaust CO2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.40 kg/h Pilot

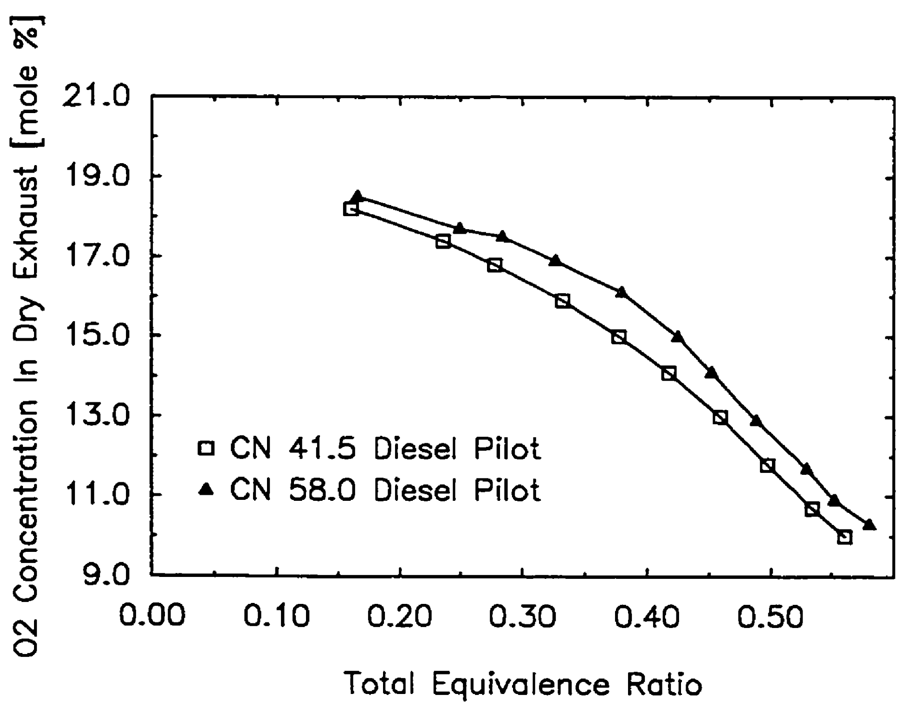

Figure F.1.10 Variation of Dry Exhaust O2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Cetane Number Fuels with a 0.40 kg/h Pilot

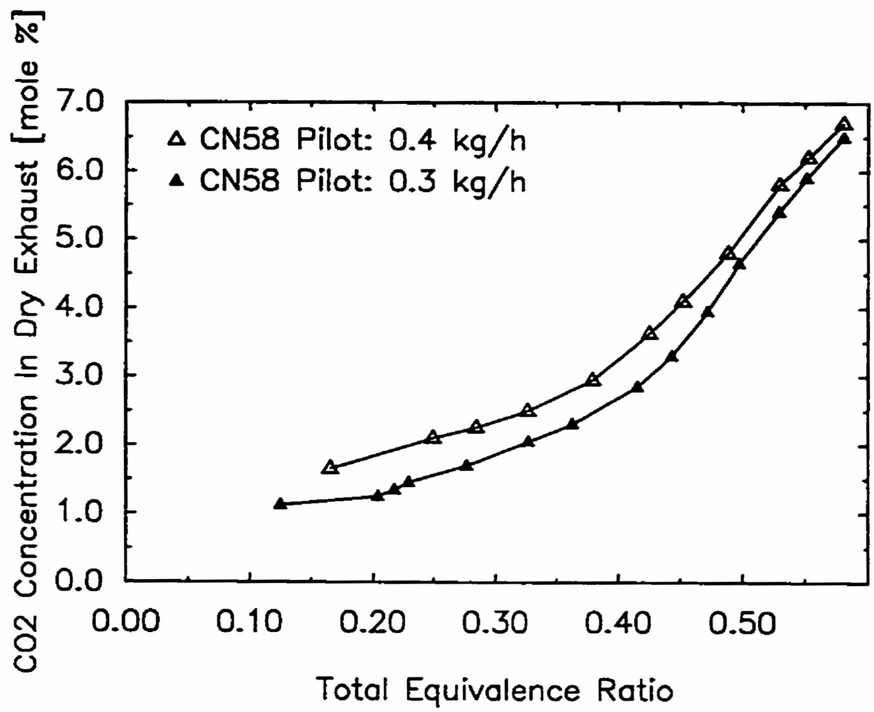

Figure F.1.11 Variation of Dry Exhaust CO2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Pilot Quantities with a 58.0 Cetane Number Pilot

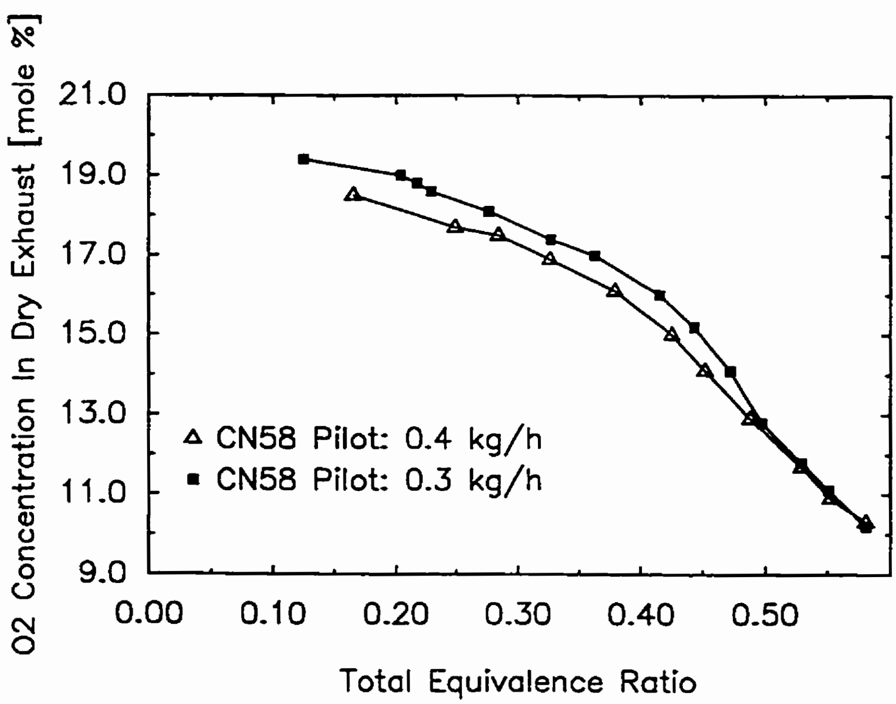

Figure F.1.12 Variation of Dry Exhaust O2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Methane when Using Different Using Different Pilot Quantities with a 58.0 Cetane Number Pilot

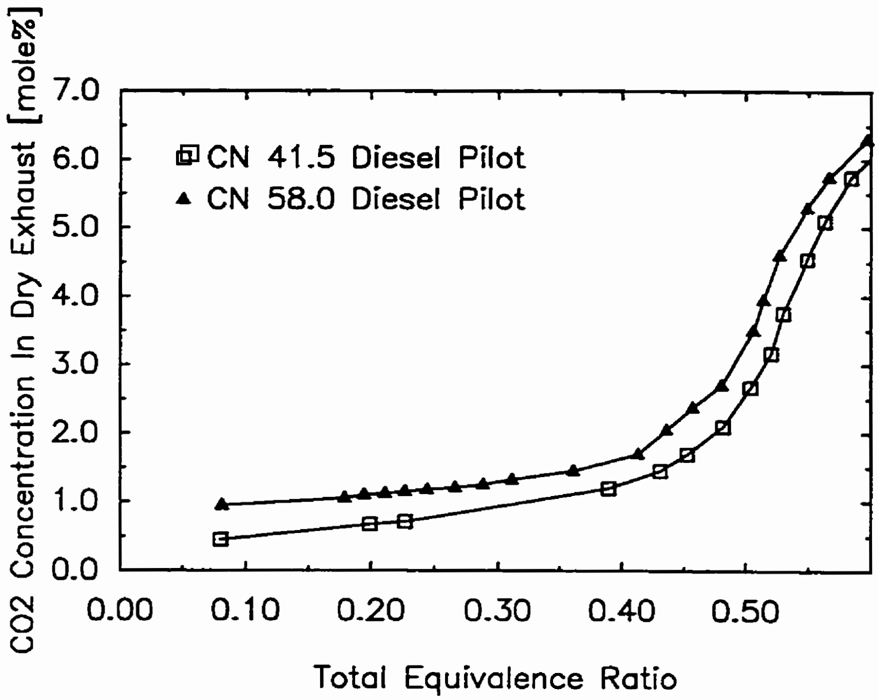

Figure F.1.13 Variation of Dry Exhaust CO2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with CH4-N2 Mixture (50% by Mass CH4) when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

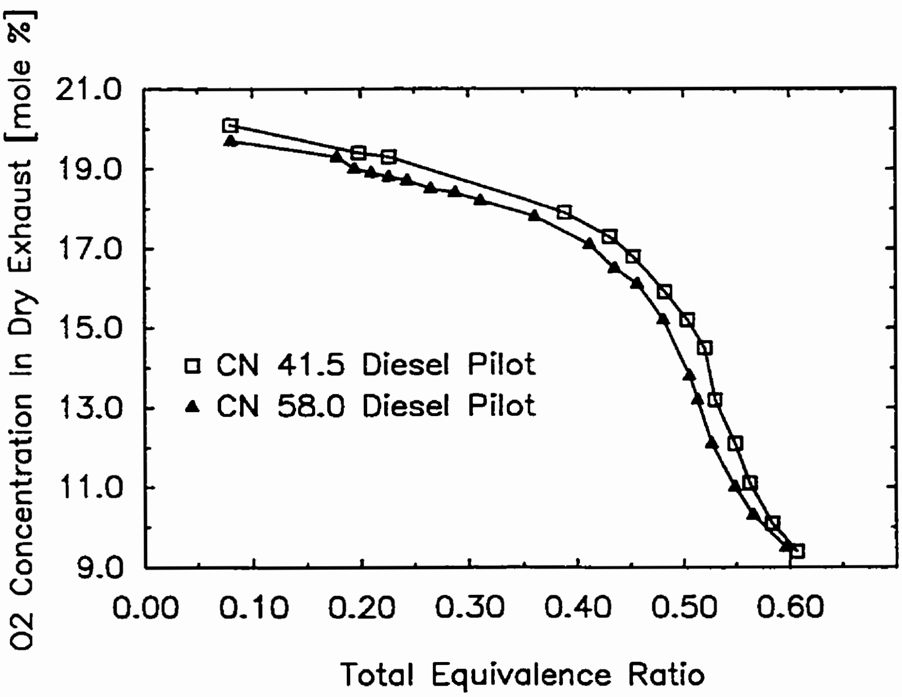

Figure F.1.14 Variation of Dry Exhaust O2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with CH4-N2 Mixture (50% by Mass CH4) when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

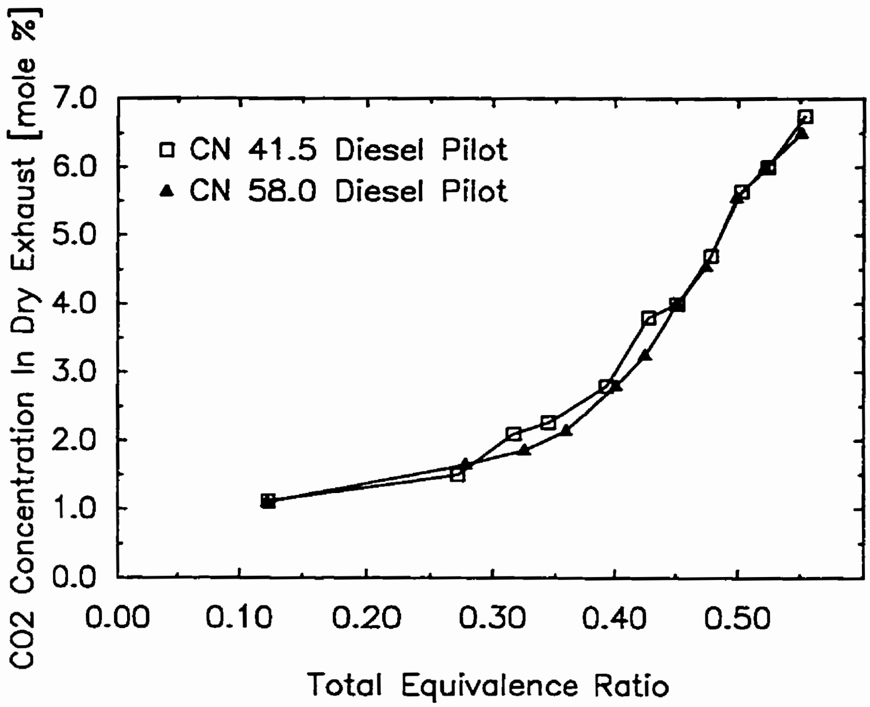

Figure F.1.15 Variation of Dry Exhaust CO2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Propane when Using Different Cetane Number Fuels with a 0.30 kg/h Pilot

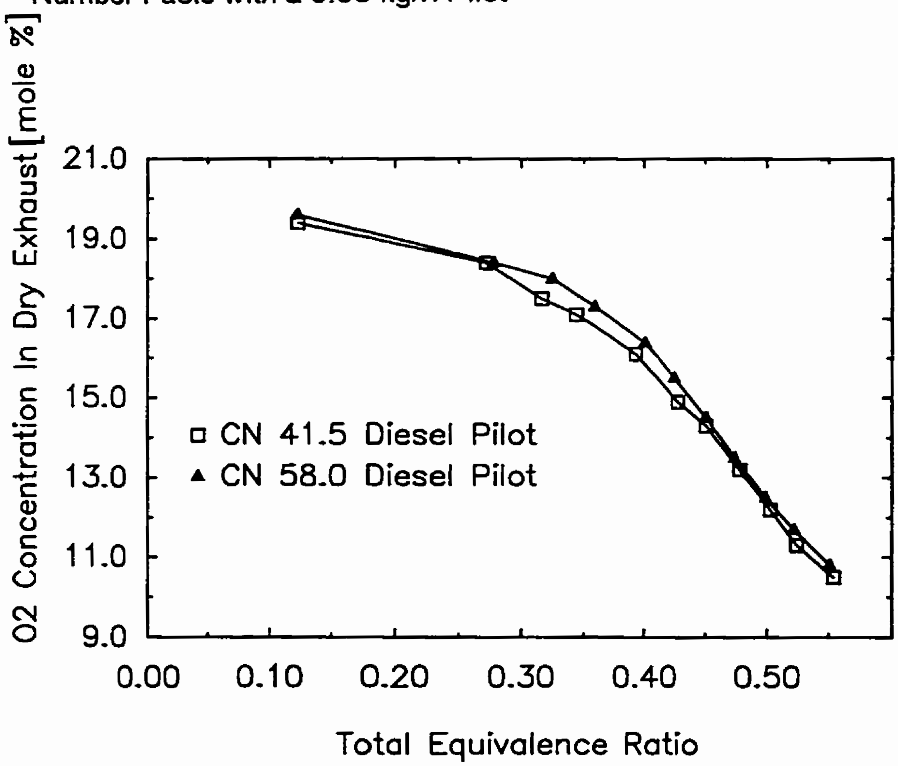

Figure F.1.16 Variation of Dry Exhaust O2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with Propane when Using Different Cetane Number Fuels with a 0.30 kg/h Pilot

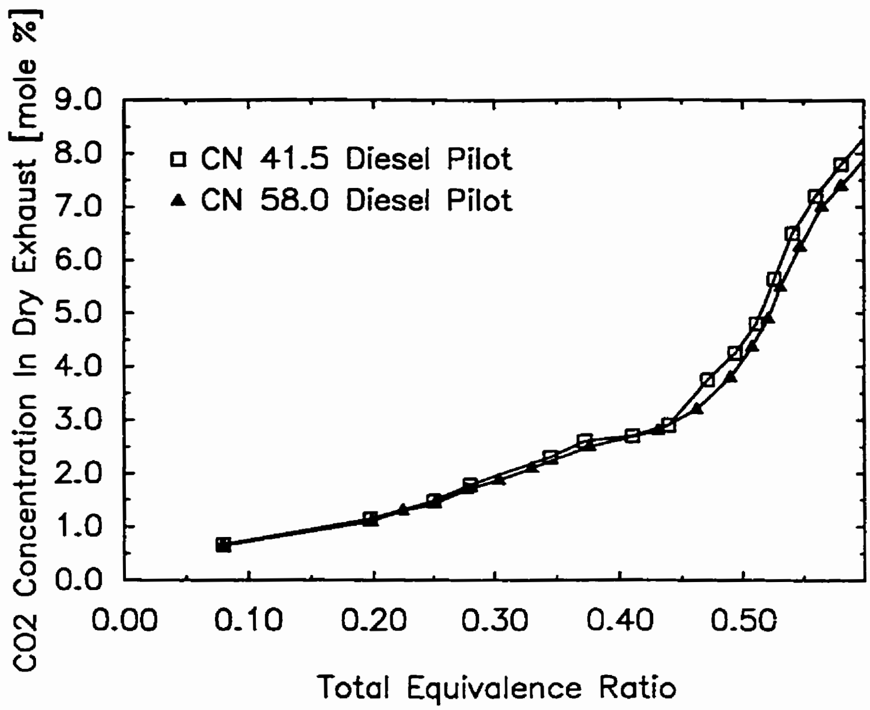

Figure F.1.17 Variation of Dry Exhaust CO2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with CH4-CO2 Mixture (50% by Mass CH4) when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

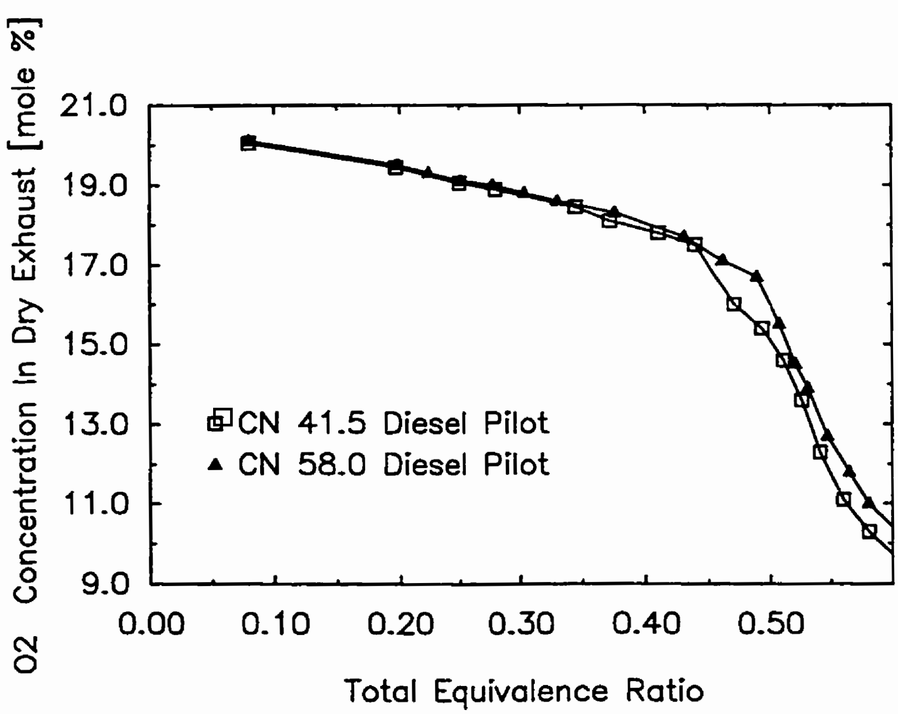

Figure F.1.18 Variation of Dry Exhaust O2 Concentration with Total Equivalence Ratio for Dual Fuel Operation with CH4-CO2 Mixture (50% by Mass CH4) when Using Different Cetane Number Fuels with a 0.20 kg/h Pilot

IMAGE EVALUATION
TEST TARGET (QA-3)

150mm

6"

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Rochester, NY 14609 USA
Phone: 716/482-0300
Fax: 716/288-5989