CHAPTER 2

REVIEW OF PREVIOUS WORK ON DUAL FUEL COMBUSTION SYSTEMS FOR C.I. ENGINES

2.1 INTRODUCTION

The use of alcohol fuels in internal combustion engines is not new. The characteristic features of lower alcohols, especially that of ethanol and methanol, like cetane rating, flammability limits and exhaust emissions are more akin to gasoline and in some respects better than gasoline. Hence these fuels have been accepted as promising alternate fuels providing possibilities for adoption for spark ignition engines [1, 2]. The fact that many automobile manufacturers have come out with production model of neat methanol and ethanol fueled S.I. engine powered automobiles, has proved beyond doubt that technical feasibilities have been well established. However, a review of literature indicates that not much attention has been paid earlier in establishing the effective utilisation of alcohol fuels in compression ignition engines, relieving the pressure of the increasing demand on middle distillates. Though in the past, Mclaughlin et.al [3] and Alperstein et.al [4] have carried out investigations in the adoption of alcohols in compression ignition engines, as auxiliary fuels, for effecting some improvements or beneficial consequences such as reduction in smoke, better air utilisation, improved thermal efficiency and capability of burning heavy distillate fuels, the quantum
of alcohol admission was limited to about 15 percent by volume of the total fuel consumption. It could be seen that the investigators were aiming to improve the combustion of diesel rather than trying to use ethanol or methanol as an alternative fuel.

However, in the recent past, considerable interest has been shown in the use of alcohols in compression ignition engines by some investigators. Alcohols' poor self-ignition and lubricity characteristics coupled with the high latent heat of evaporation preclude their use in compression ignition engines without introducing some suitable modifications of the engine or the fuel to overcome the above mentioned drawbacks. Efforts have been made in the last few years, especially after the oil embargo imposed by the OPEC countries in 1973, to identify appropriate techniques to allow compression ignition engine to operate on alcohol fuels either as a diesel extender or as a sole fuel. These include dual or pilot injection (PID), emulsification or blending, additivated alcohol fuel, spark or glow plug ignition (forced ignition), surface ignition and carburetion or fumigation. Each of these techniques require some specific modifications of the engine or the fuel handling system to allow the combustion of alcohols in compression ignition engines. The advantages, the characteristic features and the results of the previous investigations are presented and discussed in the following sections.
2.2 PILOT INJECTION OF DIESEL (PID)

This technique employs the injection of a small quantity of diesel fuel shortly before the injection of alcohol through a second injector. The high compression temperature of air resulting from the compression of air only ignites the diesel spray first and the resulting multiple ignition centres ignite the air-alcohol mixture formed subsequently. This approach is generally applicable to engines with different open chamber combustion systems as well as to swirl chamber engines.

Berg et al [5] have experimentally investigated the possibilities of using different concentrations of ethanol in water in a direct injection multicylinder diesel engine by adopting the dual injection system and has reported that the engine developed the rated power as long as the injection and ignition of the pilot fuel take place prior to the injection of ethanol-water mixture. They also observed that it would be possible to obtain consistent ignition of ethanol with a water content of 50 percent by weight resulting in a performance, similar to that of straight diesel operation. This might be possible, probably, due to the high ignition energy of diesel spray compared to spark energy. It has been also reported that consistent ignition of aqueous ethanol with 80 percent water was not possible and ethanol with higher water content resulted in a decrease of power and an increase in HC emissions. A fall in the exhaust temperature
with an increase in the quantity of water in ethanol has also been observed due to the cooling effect produced by the evaporation of water as well as the increase in the specific heat values of the products of combustion. Benefits like reduced emission of hydrocarbon, improved ignition and normal wear of engine components have also been reported.

From the investigations carried out on an air-cooled single cylinder engine Pischinger [6] has reported that as much as 90 percent of fuel energy could be replaced by alcohol through the technique of dual injection, while maintaining a high efficiency comparable to that of a standard diesel engine. Similar to the results of Berg et.al, lower emissions of gaseous pollutants and reduced combustion noise level were also achieved. Comparing the performances of ethanol and methanol operations, it was seen that the specific energy consumption of ethanol operation was little higher than that of methanol operation during the entire operating range. This increased fuel consumption may be attributed to the slightly prolonged combustion and the greater energy loss through heat transfer due to the higher mean temperature during combustion. Further, it was shown that the presence of water upto 40 percent by volume in both ethanol and methanol fuels resulted in acceptable levels of energy consumption and exhaust emissions.

In an attempt to optimize the dual injection combustion system, Hiromi Sugiyama [7] using high speed Schlieren
photography, has studied the influence of injection geometries on the combustion process. He showed that the partial mixing of the diesel and ethanol sprays exert little influence on the first stage of combustion process. The studies also confirmed that the ignition of diesel occurs first in the portion of the diesel spray envelop that has not yet mixed with the ethanol spray, suggesting the delayed formation of ignition nuclei of diesel, in the presence of ethanol vapour. Heat release studies reported by him suggest that the combustion processes of ethanol as well as diesel fuel are ruled by turbulent diffusion of fuel and air. It was also possible for him to substitute as high as 70 percent of diesel energy through ethanol, in both prechamber and swirl chamber engines without any loss in thermal efficiency and still keep NOx, CO and HC emissions on par with the levels of straight diesel operation.

Dietrich et.al [8] experimentally analysed the importance of the relative injection timings of pilot and main fuels as well as the injection pressure of the pilot fuel on the performance of a dual injection compression ignition engine. Their results confirmed that at rated load upto 92 percent of diesel could be substituted and for a wide range of speed and load levels the timing of ethanol injection did not have much influence on the fuel consumption, thus simplifying the ethanol injection system. Increasing the compression ratio from 16.5 to 20, caused an increase in peak pressure from 115
variety of triblend fuels, namely ethanol, castor oil and diesel. He was able to blend as much as 80 percent of ethanol in diesel in the presence of 10 percent castor oil which acted as a phase activator. Tests conducted on a precombustion chamber C.I. engine with different triblend fuels resulted in a comparable thermal efficiency to that of straight diesel operation. Increasing ethanol concentration in the triblend fuel caused a reduction in NOx emission levels and an increased HC and CO emissions similar to the findings of Khan and Gollahalli. Even so, HC and CO emissions were still comparable to those for straight diesel even at 80 percent ethanol content.

Hiromi Sugiyama [7] from his experimental results on different types of combustion chamber engines reported that the swirl chamber causes the maximum delay whereas the open chamber results in the minimum delay. However, only in the open chamber engine does the E/F (Ethanol/Ethanol + Diesel) ratio have a noticeable influence on the maximum rate of pressure rise.

Recently it has been reported that a mixture of ethanol and 35 percent of di-ethyl ether \((\text{C}_2\text{H}_5)_2\text{O}\), having a cetane number of around 100, which can be produced from ethanol and generally used as a cold start fluid for diesel engines at low ambient temperatures, is capable of mixing with commercial diesel at any proportion [19]. Though this blend is technically feasible, it is not a commercially acceptable
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The admixture of an additive such as cetanox or kerobrisol as an ignition or cetane improver to ethanol enables neat ethanol operation of compression ignition engines without any modification excepting the increase of the fuel flow through adjustments in fuel metering device so as to account for the lower heating value of the fuel. This approach enables the operation of a diesel engine either on neat diesel or on a mixture of ethanol and ignition improver according to the availability of fuels, thus alleviating the infrastructural problems and enabling the border crossing traffic for easy reversibility.

The minimum required admixture of ignition improver in ethanol to obtain the same ignition delay as that of diesel fuel depends upon the molecular structure of the ignition improver, the selection of which also depends on the manufacturing facilities, economy and safe handling. Most of them are only organic nitrates which means esters of aliphatic alcohols and nitric acid. As cetane number is the only true indicator of ignition delay for conventional fuels where the
physical delay component shows very little variation from one fuel to another, it follows that the usual methods for defining the cetane number of hydrocarbon fuels cannot be used for alternate fuels, like alcohols and mixtures of alcohols and ignition improved fuels, all of which have much different chemical composition and physical properties. The results obtained through the standard cetane number determination fail to achieve their original purpose in that they need by no means be representative of the ignition characteristics of a fuel in the practical diesel engine. For example, ethanol fuel with cetane improver having a cetane number of 30 which is an unacceptable value for diesel fuel, works quite well in a compression ignition engine. The mechanism by which the ignition improver initiates the combustion is not understood clearly. It is believed that the nitrate molecules are thermally decomposed at a lower compression temperature and result in the formation of radicals which on reaction with ethanol molecules initiate and accelerate the combustion through chain reaction [20]. The length of the reaction and the chain velocity determine the required quantity of ignition improver.

Pischinger et.al [21] have reported from their experimental investigations that the operation of a ethanol fueled diesel engine with 16 percent ignition improver of D-II-2 (mixture of primary alkyl nitrates) manufactured by ETHYL CORPORATION, U.S.A. and 5 percent castor oil, added to
improve the lubricity, resulted in a 3 percent increase in thermal efficiency compared with straight diesel operation.

Experimental investigations conducted by Stefan and Walde [22] on a supercharged diesel engine using ignition improved ethanol fuel, showed that a 2 percent increase, of hexyle nitrate, in ethanol from 13 to 15 percent drastically reduced the HC emissions. This indicates that at low concentrations of ignition improver the combustion becomes unstable and hampers the pre-ignition reactions.

Though it has been reported in the literature that combustion of ethanol fuel with ignition improver containing nitrogen will increase the emissions of oxides of nitrogen, the experimental results reported by Stefan and Walde does not support this view. Hardenberg and Schaeter [20] had also reported a similar trend.

It can be reasonably assumed that while all the nitrate-contained nitrogen is emitted in the form of nitrogen, all the nitrate-contained oxygen is consumed in the combustion. The above phenomena coupled with the lower flame temperature of ethanol could probably result in the reduction of the formation of oxides of nitrogen.

Tests carried out by the addition of small quantities of a new and more effective additive called triethylene-glycol di-nitrate (TEGDN) or Di-ethylene glycol di-nitrate, prepared from ethanol, via ethane, have shown satisfactory results. A
conventional direct injection engine requires less than 5 percent by volume of either of the above mentioned additives for smooth running on ethanol fuel over a wide spectrum of speed and load [23]. In addition, this admixture provides a very good cold starting characteristics which can be considered a big step forward, since other additives require high concentrations thus raising the fuel cost. Though this approach generally provides a satisfactory performance, the high cost of the additive and its explosive nature prohibits its wide use.

2.5 SURFACE IGNITION

Though the knock resistance of alcohols is very high, they tend to pre-ignite easily from hot surfaces due to their lower surface ignition temperatures which vary from 300°C to 400°C and this property is used for the successful development of surface ignition engines. Investigation on this property by several researchers revealed that the surface ignition temperature of lower alcohols like methanol and ethanol are substantially lower than most of the hydrocarbon fuels and this temperature decreases with an increase of the cylinder pressure and the surface area of the hot surface [24].

Based on the above mentioned characteristics of alcohol fuels, Nagalingam et.al [25] have developed a surface ignition engine with a compression ratio equal to that of a
conventional diesel engine and reported that the thermal efficiency of an alcohol fueled surface ignition engine was essentially the same as that of the conventional diesel engine. It was also shown that methanol fuel has the lowest delay period followed by ethanol and diesel, exactly the reverse to that of the trend expected from a conventional diesel engine. It has also been reported that the surface ignition engine operated smoothly at lower speeds with very low aldehyde emissions of only about 50 ppm. In view of the fact that combustion is initiated by the hot surface, the compression ratio becomes less critical. However, this technique needs major modification in the design of combustion chamber in accommodating a suitable hot surface and also suffers from retrofitability.

2.6 SPARK AND GLOW PLUG IGNITION

In an attempt to overcome some of the disadvantages encountered in dual fuel injection, addition of ignition improver and the preparation of emulsified fuels, Adelman and Pefley [26] have successfully used ethanol in a conventional diesel engine by adding a spark ignition system and a suitable injector to inject ethanol fuel. Thermal efficiency, power and exhaust emissions of a spark assisted swirl chamber engine are comparable to straight diesel operation. Due to poor mixing of fuel and air and the change in flame propagation characteristics of ethanol, unburned fuel and CO emissions increased, while NOx and particulate
emissions showed a decreasing trend. It has been also suggested that the location of the spark plug relative to the injector and the timing of injection are to be optimized for better results.

Miyamoto et.al from Japan [27] also investigated the engine performance on straight ethanol with aided ignition by spark and glow plugs and reported that the use of glow plug caused a significant reduction in the maximum attainable power, compared with the operation with spark plug which resulted in a higher output comparable with the straight diesel operation. This improvement in maximum power obtained with spark plug is considered to be due to shortening of the combustion duration which was possible to be achieved by the use of a high energy spark ignition. The reports also revealed that the peak combustion pressure, NOx emissions and ignition delay were less affected by the spark timing.

2.7 CARBURETION OR FUMIGATION

Historically alcohols have been used in compression ignition engines by introducing and mixing with intake air, generally referred as carburetion, fumigation or inlet air aspiration. The resulting homogeneous mixture of air and alcohol, compressed below the self ignition temperature of alcohol is forced to ignite by a diesel spray which acts as a high energy ignition source. The dual fuel combustion thus taking place in a compression ignition engine can be
visualised as a composite of diesel premixed and diffusive burning and premixed alcohol burning. The associated problems with this type of combustion system are:

(a) sluggish combustion with a drop in thermal efficiency at combinations of low flow rate of diesel and lean air-alcohol mixtures, and

(b) explosive and uncontrolled or knocking combustion at high diesel flow rates and rich air-alcohol mixtures.

A study of the properties of alcohols and diesel fuels reveals considerable differences in physical and chemical characteristics between these fuels. Among these, the high latent heat of vaporization which lowers the temperature and pressure of the charge and the poor self ignition quality of alcohols are the dominant factors resulting in such a complex combination of combustion processes.

As early as in 1952, Havemann et.al [28, 29] made extensive studies, on precombustion and open combustion chamber engines with compression ratios of 21.22 and 16.5 respectively, in order to investigate the effects of inducting ethanol along with intake air on smoke, power output and ignition delay. They reported that while it was possible to substitute diesel by ethanol up to 35 percent of total energy requirement in an open chamber engine before the onset of knocking, the precombustion chamber accepted only about 18 percent of energy. It was also established that the
open combustion chamber engine could be overloaded to about 40 percent, 2.5 times higher than the value possible in the precombustion engine, probably due to the higher excess air factor employed in the case of open chamber engines. It was also shown that while in open combustion chamber, diesel fuel having a cetane number of 45 could burn as much as 70 percent by volume of ethanol before the onset of knocking, it was only 60 percent by a diesel fuel with a cetane number of 40.

From the investigations carried out on a ASTM-CFR engine, Murthy and Pless [30] have indicated that in spite of the long ignition delay which initiates the combustion well after TDC, there was no penalty on thermal efficiency. This phenomenon could be attributed to the fast burning characteristics of alcohol. Contrary to many reports, it has been reported that concurrent combustion of alcohol and diesel resulted in an increase in the smoke density as compared with straight diesel operation. They have also reported that the engine developed an increased of bmep of 11 percent with a maximum reduction of a 4 percent in energy consumption at 20 percent of ethanol induction.

Tests conducted on a single cylinder Ricardo prechamber engine by Gupta [31] showed that ethyl alcohol corresponding to 25 percent of total energy input could be gainfully carburetted without causing any knock. At 2500 RPM, 25 percent of ethanol induction resulted in a reduction of 7 percent in specific fuel consumption, 50 percent in smoke density and an
increase in the peak pressure by 10.5 kg/cm² which compares well with the results of Murthy and Pless.

Panchapakesan et al. [32] carried out experimental investigations to examine the effects of ethanol carburetion on ignition delay, peak firing pressure and mean rate of pressure rise on a single cylinder direct injection compression ignition engine. They reported that dual fuel operations lead to very high air utilization and resulted in the development of 60 percent higher power output. They further reported that a flow rate of ethanol corresponding to 30 to 40 percent of diesel energy seems to be an optimum proportion for achieving satisfactory performance, in respect of peak pressure and mean rate of pressure rise. It was also shown that an increase of 4° CA in injection advance from the recommended value of 27° BTDC (static injection timing) resulted in a smooth running of the engine over its entire load range with only 30 percent of full load diesel flow rate. This could be attributed to the advancement of heat release close to TDC. It was further reported that an increase in the compression ratio from 16.5:1 to 20:1 allowed a higher substitution of diesel.

Felt [33] had reported that at higher levels of diesel substitution the dual combustion process becomes similar to that of spark ignition engine and all the factors such as high compression ratio and inlet temperature would affect the
dual fuel combustion through the development of knocking or rough running.

Daimler Benz [34] showed that compression ignition of ethanol was possible only with an abnormal compression ratio of 25:1. However, during the operations of part load and cold starting, even this was found to be unsatisfactory and the preheating of intake air was necessary. Soot free combustion of ethanol and the absence of engine deposits permitted the exhaust gas recycling which improved the part load performance. The Daimler-Benz data on the relationship between fuel rate to indicated load show that at part load the exhaust recycling caused fuel economy. As the peak pressure, rate of pressure rise and overheating of cylinder parts become excessive at full load due to exhaust recycling, a load dependent control of exhaust recycling has been recommended for good results.

Klaus Bro and Pederson [35] from their combustion photography studies performed on a research diesel engine, suggested that the combustion of air-ethanol mixture by a diesel spray could be classified either as single combustion, consequent combustion or simultaneous combustion. The type of combustion found in a specific case depends on the ethanol-air ratio, quantity of pilot fuel and the inlet mixture temperature. The quantity of pilot fuel required for achieving an optimum engine performance, employing ethanol as the inducted fuel, seem to be 35 percent by energy. They
also reported that the peak combustion pressures of dual fuel operations were generally higher than that of straight diesel operation. However, the increase was modest and not thought to prohibit the use of ethanol as an alternate fuel using the carburetion technique. The pronounced decrease in thermal efficiency and output at some proportion of air-ethanol ratio and the pilot quantity of diesel are in conformity with several researchers.

Marcio Cruz et al [36] from their investigation on a direct injection diesel engine with a compression ratio of 20:1 reported that ethanol substitution in a diesel engine through carburetion technique increases the maximum torque capabilities of the engine by making more complete use of oxygen and increased mass of oxygen resulting from the reduced temperature of the charge. Their investigation also revealed that only 50 percent of the maximum torque could be produced by employing the richest ethanol-air ratios and the maximum proportion of fuel energy that could be provided by ethanol at this condition amounts to about 60 percent. Increasing the pilot fuel quantity did not allow the richness of ethanol-air ratio which could be tolerated prior to the onset of knock, but resulted in the production of higher torque.

Lestz and Heisey [37] aspirated various proofs of ethanol by air atomization in a direct injection single cylinder diesel engine and reported that the percentage of water in
ethanol had no apparent effect on the performance of the engine. However, combustion quenching or misfiring occurred earlier for lower proof ethanol. Besides, the test data also indicated the dependence of ignition delay on the proof of ethanol. At full load for a 40 percent energy substitution, changing the proof of ethanol from 200 to 160 resulted in a 20 percent increase of ignition delay. At higher load levels increasing the energy share of ethanol resulted in the increase of peak pressure, rate of pressure rise and specific NOx emission. Though a 70 percent reduction in particulate emission is reported at full load, an enhanced biological activity of both raw particulate and soluble organic fraction (SOF) have been observed.

Broukhiyan and Lestz [38] reported that it would be possible to substitute upto 50 percent diesel energy through aqueous ethanol in a prechamber diesel engine at lower loads. However, a significant fall in thermal efficiency on increasing the percentage of ethanol was observed. This could be attributed to the incomplete combustion of ethanol-air mixture and the delayed start of combustion past TDC resulting in a lower expansion ratio. Their results also showed that the thermal efficiency increased with an increase in the speed of the engine, which is in close agreement with the results of Gupta [31].

Murayama et.al [39] investigated the effect of one of the important design parameters, namely the ratio of the volume
of prechamber to clearance, on the limit of admission of ethanol in a prechamber compression ignition engine and reported that a reduction in the above ratio allowed a higher percentage of diesel substitution. It is believed that the diesel knock might be caused by the large quantity of ethanol-air mixture in the vicinity of the pilot spray and a reduction in the prechamber to clearance volume ratio could result in the reduction of the volume of ethanol-air mixture enveloped by the pilot spray, thus reducing the rapid premixed combustion. A 50 percent reduction in the volume ratio from 0.222 resulted in the significant reduction of the peak pressure and ignition delay which suggest a smooth and reliable combustion at higher diesel concentration levels. As a result of this modification, the introduction of ethanol amounting to 80 percent of the total energy supply into the intake manifold was possible without the problems of knocking or misfiring. Further they reported that lowering the compression ratio reduced the knocking to a limited extent and the performance of the engine was not affected by the water content in ethanol, at least up to 40 percent by volume.

Foster et.al [40] investigated the effect of 200 proof ethanol fumigation into two types of multicylinder diesel engines. One was a direct injection turbocharged engine and the other an indirect injection naturally aspirated type. Their results showed that the emission and performance trends during ethanol fumigation were nearly the same for both the
engines. Increasing the admission of ethanol resulted in an increase in the delay period, rate of pressure rise and peak pressure which are consistent with the trends that have been well established by many investigators. Induction of ethanol in the indirect injection engine caused a larger percentage reduction of NOx and smoke than in the direct injection engine. Though the naturally aspirated engine showed a shorter delay period than the turbocharged engine, the percentage increase of delay was more with alcohol addition.

Foster et al. also attempted to evaluate the effects of direct liquid injection into the manifold and the introduction of prevaporized alcohol along with the intake air on the performance of a direct injection engine. It has been reported that the pre-vaporization of ethanol significantly eliminated the maldistribution of ethanol between the cylinders and also decreased the ignition delay, rate of pressure rise, peak pressure and the maximum limit of diesel substitution relative to the liquid injection. Though there was a reduction in the particulate emission, an increased emission of soluble organic fraction and enhanced bioactivity in both raw particulate and soluble organic fraction were noticed. This enhanced biological activity also conforms with the findings of Lestz and Heisey [37].

2.8 DUAL FUEL OPERATION ON TURBOCHARGED DIESEL ENGINES

Experimental investigations on the use of ethanol as a secondary fuel in turbocharged diesel engines have also been
carried out to some extent. In turbocharged engines, the dual fuel systems normally adapted include the admission of ethanol ahead of the compressor or into the intake manifold after the compressor. While the introduction of ethanol after the compressor generally results in an uneven cylinder to cylinder distribution, the admission ahead of compressor results in compressor blade damage.

Barnes et al. [41], admitting ethanol into the intake air just ahead of the compressor, substituted as much as 40 percent of fuel energy. Increasing the mass flow ratio of diesel to ethanol higher than 1:1 resulted in the dripping of liquid ethanol from the compressor housing as a result of inadequate heating of the mixture in the process of compression taking place in the compressor. Compared with neat diesel operation, a 15 percent increase in the peak pressure and about 35 percent increase in the rate of pressure rise were reported. Alcohol addition also increased the smoke limited power by 36 percent. Marginal improvements in brake thermal efficiency and exhaust emissions have also been reported.

In order to overcome the problems of uneven cylinder-to-cylinder distribution of alcohol or compressor blade erosion due to the admission of alcohol downstream or upstream of the turbocharger respectively, Foster et al. [40] suggested an elaborate arrangement to ensure the complete evaporation of ethanol so that only superheated ethanol vapour...
enters the intake air stream. This prevaporization of ethanol eliminated the problem of cylinder-to-cylinder maldistribution of ethanol and also decreased the ignition delay, rate of pressure rise and peak cylinder pressure compared with the values obtained by the injection of liquid ethanol into the inlet manifold. While it was possible to substitute 80 percent of energy by ethanol at lower bmep levels, the engine could not be run smoothly with more than 40 percent energy substitution at higher load levels.

Though the above mentioned technique allowed higher percentages of energy replacement over a wide spectrum of speed and load, the system necessarily requires elaborate heating and complicated controlling. An alternative method of fumigation has been investigated and reported by Duggal and Lyford Rike [42] in which ethanol was introduced just after the Vaneless diffuser of the turbocompressor. The high flow velocity field coupled with high static temperature existing at the compressor volute provided the necessary energy for the complete evaporation and improved mixing of ethanol with air, thus eliminating the problems of maldistribution and the blade damage. It was observed that the performance trend of the engine in respect of the maximum percentage ethanol admission, thermal efficiency, smoke and increase in delay period, was very much similar to the results of many investigators. This method is considered to be practical and as an optimum arrangement for the better utilization of ethanol in turbocharged compression ignition engines.
From the foregoing it could be seen that while considerable experimental data have been generated, a systematic modelling of the combustion process has not been attempted for dual fuel combustion system involving oxygenated fuels and diesel. The delay period which determines the occurrence of knock, power developed and smooth operation has also not been studied deeply in dual fuel combustion system. It is believed that there is enough scope to make a further study in this direction and for developing a combustion model to predict the various in-cylinder parameters during a dual fuel combustion cycle and to develop correlations for the delay period. These have been the main objectives of the present work. The various aspects of the development of a mathematical combustion model and the details of the experimental investigations carried out are presented and discussed in the following chapters.