CHAPTER 5
RESULTS AND DISCUSSION

5.1 INTRODUCTION

The significant results obtained from the analytical and experimental investigations of dual fuel combustion are presented and discussed in this section. The analytical part of the work has been presented first and discussed. In order to validate the proposed combustion model, the computed pressure-crank angle diagrams are compared with the experimentally obtained diagrams and discussed. The ignition delay correlation developed as a result of analytical and experimental investigations is also presented and discussed. The results of the experimental investigations which have been carried out for fixing the parameters of the correlation study are also presented and discussed. It is observed that a very satisfactory correlation has been obtained between the analytical and experimental results.

The combustion model proposed in this work considers mainly the influence of aspirating different proofs of ethanol on the peak pressure, rate of pressure rise, mean gas temperature of the cylinder contents, ignition delay of injected diesel and the burning rates of individual fuels at a full load bmep of 5.32 bar. The quantities of different proofs of ethanol required for replacing 20, 40 and 60 percent of diesel are determined from the known calorific
values so that at any instant of dual fuel operation, the net input energy remains constant, i.e., the energy corresponding to straight diesel operation.

5.2 HOT MOTORING

As mentioned earlier, the model developed relates to the closed period of dual fuel operations, which consist of compression, combustion and expansion processes. In order to study the effect of inducting ethanol along with intake air and the contained water in ethanol on the in-cylinder engine parameters, the model is tested initially for the motoring process alone.

5.2.1 MOTORING PRESSURE-TIME PREDICTIONS

The computed motoring pressure-crank angle diagrams for various flow rates of 190 proof ethanol corresponding to 20, 40 and 60 percent of diesel substitutions are compared with air only compression in Figs. 5.1 and 5.2 for 1500 and 1300 RPM respectively. It could be seen that compared with air only compression, there is a gradual decrease in the compression pressure with increasing flow of ethanol. Ethanol flow rates corresponding to 20 and 60 percent of energy shares cause a reduction of 6.7 percent and 20 percent respectively in the values of peak compression pressures at 1500 RPM. For the same quantum of ethanol flows, depression in peak pressures amounting to 4.8 percent and 17 percent have been indicated at 1300 RPM.
FIG. 5-1. PREDICTED CYLINDER PRESSURE VERSUS CRANK ANGLE FOR DIFFERENT ENERGY SHARES.
FIG. 5-2. PREDICTED CYLINDER PRESSURE VERSUS CRANK ANGLE FOR DIFFERENT ENERGY SHARES
5.2.2 MOTORING TEMPERATURE-TIME PREDICTIONS

Figs. 5.3 and 5.4 present the variations in the computed mean temperature of the charge with respect to crank angle for the same conditions of speed and percentage of energy shares. Compared with air only compression, a significant drop in the temperature of the charge of varying magnitudes with increasing percentages of energy share has been observed throughout the compression process. While the admission of 190 proof ethanol corresponding to 20 percent and 60 percent of energy shares exhibit reductions of 7.2 percent and 20.8 percent in the values of peak compression temperature at 1500 RPM, it could be seen that at 1300 RPM percentage drops of 5.8 percent and 14.5 percent have been indicated.

The observed reductions in the values of peak compression pressures and temperatures may be attributed to the reduction in the temperature of the charge at IVC, caused by the partial evaporation of ethanol in the processes of carburetion and induction as well as the subsequent vaporization of remaining ethanol droplets during the compression process. Besides, the enhanced values of the mean specific heats of the charge due to the presence of ethanol and water vapour also contribute in lowering the compression temperatures and pressures.
FIG. 5-3. MEAN GAS TEMPERATURE VERSUS CRANK ANGLE FOR DIFFERENT ENERGY SHARES - THEORY

SPEED: 1500 RPM
C.R.: 16.5
ETHANOL: 190 PROOF

- AIR ONLY
- 20% ENERGY
- 40% ENERGY
- 60% ENERGY

CRANK ANGLE (deg)
FIG. 5-4. MEAN GAS TEMPERATURE VERSUS CRANK ANGLE FOR DIFFERENT ENERGY SHARES - THEORY
5.2.3 INFLUENCE OF HIGHER COMPRESSION RATIO

The development of compression pressure and temperature of dual fuel system corresponding to 40 percent diesel substitution are compared with air only compression in Figs. 5.5 and 5.6 respectively at 1500 RPM and at a higher compression ratio of 18.5. It could be seen that increasing the compression ratio results in a lower percentage drop in the values of peak compression pressure and temperature.

Lowering the proof of ethanol causes a further reduction in the compression pressure and temperature of the charge. At 1500 RPM, compared with 190 proof ethanol, admission of 150 proof ethanol further lowers the peak compression pressure and temperature by 2.5 percent and 3.2 percent respectively. These reductions are probably caused by larger water content (29.6 percent by weight).

5.2.4 VARIATIONS OF INTERNAL ENERGY AND COMPRESSION WORK

Fig. 5.7 illustrates the influence of various flow rates of proof ethanol at 1500 RPM on the values of computed peak internal energy of the charge and compression work. It is clearly seen that these parameters show a decreasing trend with increasing percentage of ethanol energy share. Similar trend is observed by lowering the proof of ethanol as well. As the vaporization of ethanol during the compression process takes some of the energy supplied in the form of work, there
FIG. 5-5. PREDICTED CYLINDER PRESSURE VERSUS CRANK ANGLE AT HIGHER C.R.

SPEED: 1500 RPM
C.R.: 18.5
ETHANOL: 190 PROOF

- AIR ONLY
- 40% ENERGY
FIG. 5-6. MEAN GAS TEMPERATURE VERSUS CRANK ANGLE AT HIGHER C.R.
FIG. 5.7. VARIATIONS OF PEAK INTERNAL ENERGY AND COMPRESSION WORK WITH RESPECT TO PERCENTAGE OF ENERGY SHARE AND PROOF-THEORY
is a net reduction in the compression work. This reduction increases with increase in the flow rate of ethanol. Owing to the reduction in compression temperature due to ethanol induction, the incremental change in internal energy is decreased. It is also observed that the reductions in the compression temperature, pressure and work done are larger at lower proofs of ethanol. However, lowering the speed of the engine causes a smaller reduction.

From the foregoing, it is seen that the temperature and pressure prevailing at the moment of diesel injection differ significantly from the corresponding values of air only compression. These reductions influence the evaporation and mixture formation characteristic of injected diesel and preflame reactions leading to the formation of ignition nuclei and hence provide an unfavourable environment for the initiation of combustion. The predicted compression temperature and pressure for various flow rates of ethanol of different proofs and the experimentally measured delay periods enabled the development of dual fuel ignition delay correlation which will be presented in a later section.

5.3 PREDICTED PRESSURE - CRANK ANGLE DIAGRAMS AND MASS BURNING RATES OF FUELS DURING DUAL FUEL COMBUSTION

5.3.1 PREDICTION OF PRESSURE - CRANK ANGLE DIAGRAMS

The range of operating parameters included in the present study for predicting the apparent mass burning rates
of individual fuels are: varying percentages of ethanol energy share (0 to 60 percent), variations in the proof of ethanol (190, 170 and 150), variations in the injection timing (8°, 13° and 18° BTDC) of the igniting fuel and varying the speed of the engine (1300, 1500 rpm).

5.3.1.1 STRAIGHT DIESEL OPERATION

As mentioned earlier, the values of the coefficients of the shape factors $C_{p1}$, $C_{p2}$, $C_{d1}$ and $C_{d2}$ and the phase proportionality factor $\beta$ and their correlations with engine operating conditions, such as ignition delay, trapped equivalence ratio of diesel and the engine speed are adjusted from an extensive series of tests so that the computed pressure-crank angle diagrams of straight diesel operation closely agree with the experimental diagrams over a wide range of speeds and loads. The theoretical pressure - crank angle diagram so obtained for a straight diesel operation is given in Fig.5.8 for a speed of 1500 RPM and a full load bmep value of 5.32 bar. For a dynamic injection timing of 13° BTDC, the model predicts an ignition delay of 9.4°CA, peak pressure of 64.5 bar and a peak rate of pressure rise of 7.326 bar/°CA. The occurrence of peak pressure is at 10° ATDC.

5.3.1.2 DUAL FUEL OPERATIONS WITH 190 PROOF ETHANOL

The predicted pressure crank angle diagrams are presented in Figs.5.9, 5.10 and 5.11 for the dual fuel operations with 20, 40 and 60 percent diesel substitutions.
FIG. 5.8. COMPUTED CYLINDER PRESSURE DIAGRAM FOR STRAIGHT DIESEL OPERATION

SPEED: 1500 RPM
FUEL: DIESEL
INJ.: 13° BTDC
DLY.: 9.4°
PEAK PR.: 64.5 bar
PEAK δP/δθ: 7.326 bar/°CA
FIG. 5.9. COMPUTED CYLINDER PRESSURE DIAGRAM FOR DUAL FUEL OPERATION SHARE 20%
FIG. 5.10. COMPUTED CYLINDER PRESSURE DIAGRAM FOR DUAL FUEL OPERATION—SHARE 40%
FIG. 5.11. COMPUTED CYLINDER PRESSURE DIAGRAM FOR DUAL FUEL OPERATION - SHARE 60%
by 190 proof ethanol. It could be seen that for 20 percent ethanol energy share an increase of 6.5 percent in the delay period has been obtained. Also the peak rate of pressure rise and peak pressure are increased by 17.5 percent and 3.7 percent respectively, compared with straight diesel operation. Also seen that the occurrence of the peak pressure has been advanced by 1° CA.

At 40 percent energy share the ignition starts at 1° BTDC giving rise to a 25 percent increase in the delay period compared to straight diesel operation. Due to the delayed initiation of combustion very close to the TDC position a 12 percent increase in the value of maximum rate of pressure rise has been observed. The occurrence of peak pressure is further advanced and occurs at 8° ATDC. Despite this increase in the maximum rate of pressure rise, the peak pressure value is lowered to 61.5 bar as a result of the bulk of combustion taking place in the expansion process.

The induction of higher percentages of ethanol altogether causes a different effect. At 60 percent energy share the combustion starts at 2.5° ATDC. A lower peak pressure of 54.0 bar and a maximum rate of pressure rise of 7.12 bar/°CA have been predicted. An increase of 65 percent in the value of delay period is expected to result in the burning of a higher proportion of ethanol and diesel in the premixed mode of combustion. This should have resulted in a higher peak pressure and a higher rate of pressure rise.
However, owing to the delayed initiation of combustion well past TDC, the values of peak pressure and maximum rate of pressure rise are lowered. Compared with straight diesel operation, the occurrence of peak pressure in this case was shifted by 3° CA further away from TDC.

From the foregoing it is seen that the increase or decrease of peak pressure or maximum rate of pressure rise depend on the ignition delay and the point of ignition with respect to the TDC position during dual fuel operation.

5.3.1.3 DUAL FUEL OPERATIONS WITH 170 PROOF ETHANOL

Figs. 5.12, 5.13 and 5.14 illustrate the predicted pressure-crank angle diagrams for 170 proof ethanol dual fuel operations. At an energy share of 20 percent, admission of 170 proof ethanol enhances the delay period by 8 percent compared to 190 proof ethanol. However, the resulting changes in the peak pressure and the rate of pressure rise are only marginal.

It is also seen that increasing the percentages of energy share to 40 and 60 percent, further increases the delay period by 10.9 percent and 18.5 percent respectively. It is known that the mass flow rate of 170 proof ethanol has to be increased in accordance with the decrease in the heating value and this increased mass flow associated with the higher latent heat depresses the charge temperature
FIG. 5.12. COMPUTED CYLINDER PRESSURE DIAGRAM
FIG. 5.13. COMPUTED CYLINDER PRESSURE DIAGRAM

SPEED: 1500 RPM
FUEL: D+E (40%)
PROOF: 170
INJ.: 13° BTDC
DLY.: 13.25°
PEAK PR.: 56.2 bar
PEAK dp/dθ: 7.5 bar/°CA
FIG. 5.14. COMPUTED CYLINDER PRESSURE DIAGRAM

SPEED : 1500 RPM
FUEL : D+E (60 %)
PROOF : 170
INJ. : 13° BTDC
DLY. : 17.4°
PEAK PR. : 48.1 bar
PEAK dp/deg : 5.82 bar/°CA
considerably, which has a greater influence on the vaporization and diffusion of fuels. Consequently the peak pressure and rate of pressure rise are lowered and the ignition delay is increased.

5.3.2 MASS BURNING RATES AND CUMULATIVE HEAT RELEASE

5.3.2.1 STRAIGHT DIESEL OPERATION

The computed mass burning rate of diesel during straight diesel operation is shown in Fig.5.15. It is seen that the maximum rate of heat release, characterised by the peak rate of fuel burning of 2.35 mg/°CA, occurs at 1° ATDC. It is also predicted that nearly 64 percent and 87 percent of total heat supplied have been released at the termination of premixed and diffusion combustion periods which are identified as the points of occurrence of peak values of pressure and temperature respectively.

5.3.2.2 DUAL FUEL OPERATION WITH 190 PROOF ETHANOL

The predicted mass burning rates of individual fuels and the resulting cumulative heat release for the dual fuel operations of 20, 40 and 60 percent of diesel substitutions for 190 proof ethanol are presented in Figs.5.16, 5.17 and 5.18. It could be seen that the shapes of individual mass burning curves of the inducted and injected fuels are nearly identical though differing in magnitude. This may be attributed to the assumption made in the model pertaining to
FIG. 5.15. COMPUTED MASS BURNING RATE AND CUMULATIVE HEAT RELEASE
FIG. 5.16. MASS BURNING RATES OF FUELS AND GROSS CUMULATIVE HEAT RELEASE—THEORY
FIG. 5.17. MASS BURNING RATES OF FUELS AND GROSS CUMULATIVE HEAT RELEASE - THEORY
FIG. 5.18. MASS BURNING RATES OF FUELS AND GROSS CUMULATIVE HEAT RELEASE - THEORY
the normalized burning rates of ethanol and diesel. Compared with straight diesel operation, the peak burning rates of diesel are lowered by 10.6, 38.0 and 49.0 percent respectively during 20, 40 and 60 percent of ethanol energy share operations. The shifting of the occurrence of peak burning rates of fuels away from the TDC can also be noticed [65].

At 60 percent ethanol energy share, the mass burning rate of ethanol is obviously higher compared to 40 percent energy share. Owing to the lower heating value of ethanol and combustion occurring at relatively lower temperature and pressure levels, 60 percent of diesel substitution results in a lower rate of pressure rise and peak pressure.

5.3.2.3 DUAL FUEL OPERATION WITH 170 PROOF ETHANOL

Figs 5.19, 5.20 and 5.21 depict the mass burning rates of diesel and 170 proof ethanol and the cumulative heat release for 20, 40 and 60 percent of diesel substitutions. Dual fuel operation with 170 proof ethanol at 20 percent diesel substitution shifts the occurrence of the peak rate of fuel burning by 2° CA away from TDC compared with 190 proof operation. It could be observed that increasing the percentage of energy share further shifts the occurrence of the peak rate of fuel burning from TDC position.
FIG. 5.19. MASS BURNING RATES OF FUELS AND GROSS CUMULATIVE HEAT RELEASE - THEORY
FIG. 5.20. MASS BURNING RATES OF FUELS AND GROSS CUMULATIVE HEAT
RELEASE - THEORY
FIG. 5.21. MASS BURNING RATES OF FUELS AND GROSS CUMULATIVE HEAT RELEASE - THEORY

SPEED: 1500
FUEL: D+ E (60%)
INJ.: 13° BTDC
PROOF: 170

- - - DIESEL
- - - ETHANOL

FUEL BURNING RATE (mg/°CA)

CRANK ANGLE (deg)

CUMULATIVE HEAT RELEASE (J)
5.4 PREDICTION OF MASS AVERAGED GAS TEMPERATURE

The effects of inducting various percentages of 190 proof ethanol on the combustion temperature of dual fuel operation are shown in comparison with straight diesel operation in Fig. 5.22 on a crank angle basis. It could be seen that dual fuel operations result in a lower combustion temperature, the degree of which depends on the quantum of ethanol admission. Compared with straight diesel operation, the peak combustion temperatures of dual fuel operations at 20, 40 and 60 percent of diesel substitutions are lowered by 2.1, 11.4 and 14.5 percent respectively. Besides, the occurrence of the peak temperature during 20 percent and 40 percent operations are advanced by $2.5^\circ$ CA and $1.5^\circ$ CA respectively from $383^\circ$ CA corresponding to straight diesel operation. However, at 60 percent substitution the occurrence of peak temperature is retarded to $385^\circ$ CA. It is particularly interesting to note that the temperature during the expansion at 60 percent energy share operation is slightly higher than that observed during 40 percent energy share. This increase may be attributed to the slow burning during expansion.

Fig. 5.23 shows the variation of the computed mean gas temperature with crank angle during the dual fuel operation with 170 proof ethanol at 1500 RPM. Compared with the values for 190 proof, a further drop in the combustion temperatures has been indicated. There is also a shift of the point of
FIG. 5.22. VARIATIONS OF COMPUTED MEAN GAS TEMPERATURE AT 1500 RPM. ETHANOL: 190 PROOF
FIG. 5.23. VARIATIONS OF COMPUTED MEAN GAS TEMPERATURE AT 1500 RPM. ETHANOL: 170 PROOF
occurrence of peak temperature away from the TDC. From the foregoing it can be concluded that the temperature of the gas in the cylinder is greatly influenced by the changes in ignition delay and proof of ethanol.

5.5 PREDICTION OF IGNITION DELAY PERIOD

Computed ignition delay periods as a function of percentage of energy share with various proofs of ethanol tried in this investigation are illustrated in Fig.5.24. It could be seen that for all proofs of ethanol the ignition delay increases gradually with increasing percentages of ethanol induction upto about 35 percent, beyond which the increase becomes rapid. It can also be noted that the ignition delay increased with decreasing the ethanol quality. This trend is in close agreement with the results of Lestz and Heisey [37].

5.6 RATE OF PRESSURE RISE

Fig.5.25 illustrates the trend in the variation of computed maximum rate of pressure rise at 1500 RPM for various percentages of diesel substitution with different proofs of ethanol. It can be observed that the rate of pressure rise which is a function of the intensity of initial premixed burning of fuels and controlled by the delay period and the charge temperature, is significantly influenced by the proof of ethanol. As long as the combustion starts before
FIG. 5.24. COMPUTED DELAY PERIODS AS A FUNCTION OF ETHANOL ENERGY SHARE FOR VARIOUS PROOFS
FIG. 5.25. PREDICTED PEAK RATE OF PRESSURE RISE AS A FUNCTION OF ETHANOL ENERGY SHARE AT 1500 RPM
TDC, the rate of pressure rise increases with increase in the delay period, caused either by increasing the ethanol energy share or lowering the proof of ethanol. The rate of pressure rise increases with increasing the ethanol energy share to a maximum of 27 percent for 190 proof, 20 percent for 170 proof and 17 percent for 150 proof. Any further increase of ethanol flow reduces the rate of pressure rise gradually and reaches a value below that of straight diesel operation. The increase in the peak rate of pressure rise can be attributed to the large amount of fuels taking part in the uncontrolled initial phase of combustion.

5.7 CYLINDER PRESSURE AND MASS BURNING RATES AT 1300 RPM

Typical computed pressure-crank angle diagram at 1300 RPM and at full load bmep of 5.32 bar are shown in Figs.5.26, 5.27 and 5.28 for straight diesel and dual fuel operations during 20 and 60 percent diesel substitutions. It is noticed that the peak pressure and rate of pressure rise exhibit a decreasing trend from the corresponding values obtained at 1500 RPM excepting at 20 percent energy share.

Figs.5.29, 5.30 and 5.31 present the mass burning rates of fuels and the resulting cumulative heat release during the above operating conditions. For similar percentages of energy share, compared with 1500 RPM, the peak rates of fuel burning show a decreasing trend at 1300 RPM.
FIG. 5.26. COMPUTED CYLINDER PRESSURE DIAGRAM - AT FULL LOAD

SPEED: 1300 RPM
FUEL: DIESEL
INJ: 13° BTDC
DLY: 6.3°
PEAK PR: 62.0 bar
PEAK dp/dθ: 7.2 bar/°CA
FIG. 5.27. COMPUTED CYLINDER PRESSURE DIAGRAM - AT FULL LOAD

SPEED: 1300 RPM
FUEL: D+E (20%)
PROOF: 190
INJ.: 13°BTDC
DLY.: 8.7°
PEAK PR.: 67.5 bar
PEAK dp/dθ: 8.7 bar/°CA
FIG. 5.28. COMPUTED CYLINDER PRESSURE DIAGRAM
- AT FULL LOAD
FIG. 5.29. MASS BURNING RATE OF DIESEL AND GROSS CUMULATIVE HEAT RELEASE - THEORY
FIG. 5.30. MASS BURNING RATES OF FUELS AND GROSS CUMULATIVE HEAT RELEASE - THEORY

SPEED: 1200 RPM
FUEL: D+E (20%)
PROOF: 190
INJ.: 13° BTDC

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DIESEL
ETHANOL
FIG. 5.31. MASS BURNING RATES OF FUELS AND GROSS CUMULATIVE HEAT RELEASE

SPEED: 1300 RPM
FUEL: D+E (60%)
PROOF: 190
INJ.: 13° BTDC

- DIESEL
- ETHANOL
5.7.1 MASS AVERAGED GAS TEMPERATURE

The variations of the computed mean gas temperatures with respect to crank angle are shown in Fig.5.32 for 1300 RPM. Similar trends in the variation of temperature as obtained at 1500 RPM have been observed at this speed also. The shifting of the point of occurrence of the peak temperature as compared to straight diesel operation has also been observed in this case. The reasons for these variations may be attributed to the same effects mentioned earlier for 1500 RPM.

5.7.2 RATE OF PRESSURE RISE

The effect of proof of ethanol on the peak rates of pressure rise at 1300 RPM are presented in Fig.5.33. It could be seen that the peak rate of pressure rise of dual fuel operations increases gradually up to 20 percent of energy share and then attains values below that of straight diesel operation.

5.8 INFLUENCE OF RETARDED TIMING OF DIESEL INJECTION

In order to evaluate the influence of the injection timing of diesel on the mass burning rates of fuels and ignition delay, computations were also made for a dynamic injection timing of 8°BTDC. The resulting pressure – crank angle diagrams are presented in Figs.5.34 and 5.35 for
FIG. 5.32. VARIATIONS OF COMPUTED MEAN GAS TEMPERATURE AT 1300 RPM - ETHANOL: 190 PROOF
FIG. 5.33. PREDICTED PEAK RATE OF PRESSURE RISE AS A FUNCTION OF ETHANOL ENERGY SHARE AT 1300 RPM
FIG. 5.34. COMPUTED CYLINDER PRESSURE DIAGRAM FOR RETARD INJ. TIMING

SPEED: 1500 RPM
FUEL: DIESEL
INJ.: 8° BTDC
DLY.: 6.6°
PEAK PR.: 65.9 bar
PEAK dp/60: 7.82 bar/°CA
FIG. 5.35. COMPUTED CYLINDER PRESSURE DIAGRAM FOR RETARD INJ. TIMING

SPEED: 1500 RPM
FUEL: D+E (20%)
PROOF: 190
INJ.: 8° BTDC
DLY.: 8.6°
PEAK PR.: 57.6 bar
PEAK dp/deg: 6.86 bar/°CA
straight diesel and 20 percent ethanol (190 proof) energy share operations respectively. Compared with base line injection timing, it could be seen that the retarded injection timing of 8°BTDC results in a decrease of ignition delay of 28 percent and 17 percent as compared to the delay period observed during straight diesel and dual fuel operations respectively. The values of the peak pressure and rate of pressure rise of straight diesel operation show a marginal increase due to the combustion starting very close to TDC (1.4°BTDC). However, during the dual fuel operation at 20 percent diesel substitution the peak pressure and rate of pressure rise are decreased marginally. This can be attributed to the delayed start of combustion (0.3°ATDC).

The mass burning rates of individual fuels and the cumulative heat release during the operations of retarded injection timing are shown in Figs. 5.36 and 5.37. In the case of straight diesel operation, compared with the base line injection timing of 13°BTDC, retarding the injection timing to 8°BTDC results in an increase of 23 percent in the peak rate of diesel burning, owing to the occurrence of combustion starting at 1.4°BTDC. However, retarded dual fuel operation during 20 percent energy share shows only a marginal increase in the values of fuel burning rates.

5.9 PREDICTION OF PRODUCTS OF COMPLETE COMBUSTION

The percentage increase in the predicted concentrations of the products of complete combustion, namely $H_2O$ and $CO_2$ as
FIG. 5.36. MASS BURNING RATE OF DIESEL AND GROSS CUMULATIVE HEAT RELEASE - THEORY
FIG. 5.37. MASS BURNING RATES OF FUELS AND GROSS CUMULATIVE HEAT RELEASE - THEORY

SPEED: 1500 RPM
FUEL: D+E (20%)
PROOF: 190
INJ: 8° BTDC
- - DIESEL
- - - ETHANOL
a function of percentage of diesel substitution and proof of ethanol is shown in Fig. 5.38. It is observed that the increase in the concentration of $H_2O$ with ethanol energy share is rapid compared with the increase in $CO_2$. Similar trend has been observed while decreasing the proof of ethanol. It is believed that the higher concentrations of these tri-atomic gases associated with increasing values of specific heat with temperature might be the main cause for the drop in the mean cylinder gas temperature.

5.10 IGNITION DELAY CORRELATION FOR ETHANOL-DIESEL DUAL FUEL COMBUSTION SYSTEM

Among the many factors which control the combustion phenomena in a dual fuel combustion system, the ignition delay can be considered as the most significant and important parameter. Considerable data were collected during the investigations to analyse the influence of ethanol admission on ignition delay. The rapidity of heating of the injected diesel and the associated ignition delay are primarily influenced by the physical properties of the surrounding medium. The rate of heat transfer to the diesel particles during the ignition delay period is almost independent of fuel volatility and chemical structure. Thus, owing to the lower compression temperature and pressure of the charge and the presence of ethanol vapour surrounding the evaporating diesel particles, dual fuel operation always result in extended delay periods.
FIG. 5.38. PERCENTAGE INCREASE IN THE PREDICTED CONCENTRATIONS OF PRODUCTS OF COMPLETE COMBUSTION AS A FUNCTION OF ETHANOL ENERGY FOR VARIOUS PROOFS
The factors which govern the ignition delay are numerous and the manner in which they influence its magnitude are not clearly known. The key factors which influence the delay are the variations of temperature and pressure during the delay period, engine speed and combustion chamber design. As stated earlier, the temperature and pressure of the charge depend on the quantum of ethanol admission as well as the proof of ethanol. From the values of measured ignition delay, recorded over a large number of experimental runs at varying operating conditions and the corresponding computed temperature and pressure of the charge at the point of diesel injection, the following delay correlation has been developed for dual fuel combustion system.

$$\text{Delay (ms)} = 2.5 \ e^{1800/T} / p^{1.02}$$  \hspace{1cm} (5.1)

where $T$ and $p$ are temperature and pressure at the point of injection. A computer package called "Regression Technique" available in IBM 370 system has been used for developing this correlation. Fig.5.39 compares the delay periods, as computed from the above expression with the values obtained from the experimental runs at 1500 RPM and 1300 RPM and at full load for various percentages of energy share by 190 and 150 proof ethanol. It is quite evident that the measured values exhibit a close agreement with the corresponding computed values. For both the speeds the difference between the delay periods of 190 and 150 proof ethanol increases with increase in the percentage of energy share, which suggest the dependence of
FIG. 5.39. COMPARISON OF MEASURED AND PREDICTED IGNITION DELAYS

- **SPEED**: 1500 RPM
- **LOAD**: FULL
- **INJ.**: 13° BTDC
- **PREDICTED - 190 PROOF**
- **PREDICTED - 150 PROOF**
- **MEASURED - 190 PROOF**
- **MEASURED - 150 PROOF**

**DELAY (°CA)**

- **SPEED**: 1300 RPM
- **LOAD**: FULL

**PERCENTAGE OF ENERGY SHARE**
ignition delay on the water content of ethanol. However, the effect of proof appears to be less at the lower speed. These findings indicate that the charge temperature affects the delay period more than any other factor.

The validity of the above correlation has also been checked for the dual fuel operations with advanced and retarded injection timings of 5°CA from the recommended baseline timing of 13°BTDC. During these runs, the maximum observed deviations were found to be within ±10 percent.

5.11 COMPARISON OF PREDICTED AND MEASURED CYLINDER PRESSURE DIAGRAMS

5.11.1 INTRODUCTION

The main objective of this work is to develop a dual fuel combustion model capable of predicting the apparent mass burning rates of individual fuels and the resulting cylinder pressure. The validity of the model is checked by recording the cylinder pressure during the experimental runs. Typical predicted pressure–crank angle diagrams are compared with the measured ones, recorded under different operating conditions such as speed, quantum of ethanol admission and proof of ethanol, in the following sections. The correlation has been very satisfactory for the range covered in the experimental program.
5.11.2 HOT MOTORING RUN

Figs. 5.40 and 5.41 show typical motoring cylinder pressure diagrams with the predicted values shown in full lines and the experimental readings indicated for every 5° crank angle. These diagrams reveal that the correlation between the measured and simulated values is good and the theoretical predictions are close to the realistic behaviour of the dual fuel compression processes. As pointed out earlier, compared with air only compression, carburetion of ethanol markedly lowers the compression temperature and pressure of all dual fuel motoring runs. Among other factors, these reductions mainly depend on the quantum of ethanol, proof of ethanol and speed of the engine. It can be noticed that the predicted pressures are quite close to the experimental values in the expansion portion of the motoring run and differ by a margin of 5 to 8 percent only during the compression process.

The effect of ethanol induction on the measured and predicted peak compression pressures at various speeds are shown in Fig. 5.42. Lowering the speed of the engine decreases the peak pressure as a result of increased heat loss due to lengthening of compression time as well as the induction work.

It is estimated that at 1500 RPM, for 190 proof ethanol, the compression pressure of the charge at TDC, is varying according to the expression
FIG. 5.40. COMPARISON OF MEASURED AND PREDICTED CYLINDER PRESSURE - HOT MOTIONING
FIG. 5.41. COMPARISON OF MEASURED AND PREDICTED CYLINDER PRESSURE - HOT MOTORING

- **SPEED**: 1500 RPM
- **PROOF**: 190

- **SIMULATED - AIR ONLY**
- **MEASURED**
- **SIMULATED (59.2% ETH.)**
- **MEASURED**
FIG. 5.42. EFFECT OF ETHANOL INDUCTION ON THE PEAK COMPRESSION PRESSURE AT VARIOUS SPEEDS
\[ P = P^* - 0.085 \cdot \text{EES} \quad \ldots (5.2) \]

where \( P^* \) is the peak compression pressure for air only compression and \( \text{EES} \), the percentage ethanol energy share. For other proofs of ethanol, the above equation has to be multiplied by a proof correction factor, as given below:

\[
\text{Correction factor} = \left( \frac{0.25}{190} \right) \quad \ldots (5.3)
\]

The combined effect of the reduction in pressure as well as temperature causes a reduction in the compression work which is reflected by a marginal decrease in the dynamometer scale reading.

The results of computation have also shown that an error of \( +20^\circ \text{C} \) in the estimated temperature of the charge at IVC causes a variation in the magnitude of the pressure at IVC and peak compression pressure of 2.5 percent and 3 percent respectively. Further the peak compression temperature, internal energy and work done are increased by 3.5 (30 K), 4 (14.5 J) and 4 percent (9.4 J) respectively.

Few runs of hot motoring were also carried out with the heating of the air-ethanol mixture so that the temperature of the charge was raised to a value equal to the normal intake temperature of 35°C. The peak compression pressure in this case is marginally higher compared to motoring without
preheating the mixture. The effect of increasing the mixture temperature obviously reduces the quantum of air inducted. This reduction is due to more complete vaporisation of ethanol. It is observed that at 45 percent diesel substitution, preheating the mixture to 35°C lowered the air flow rate by 6.8 percent. Typical cylinder pressure diagrams recorded during the hot motoring under different operating conditions are given in plate 5.1.

5.11.3 COMPARISON OF FIRED CYCLES

5.11.3.1 CYLINDER PRESSURE DIAGRAMS AT 1500 RPM

Typical experimental pressure-crank angle data which are the average values of 3 cycles, are compared with the predicted pressure values obtained from the combustion model. Fig.5.43 shows such a comparison for straight diesel operation at 1500 RPM and at full load. While the measured gas pressure agrees well with the predicted values during the compression period, the measured pressures are lower by 3 to 5 percent during the initial period of expansion process. The magnitudes of peak pressure and their occurrence show a good agreement between predicted and experimental values. While the model predicts a delay of 9.4°C, actual combustion occurs after 9°C from the point of injection of 13°C BTDC. In the same diagram the measured duration of injection is shown as 16°C, giving a rate of injection of 1.65 mm/°CA as against the rated value of 1.8 mm/°CA.
PLATE 5.1. TYPICAL MOTORING CYLINDER
PRESSURE TRACES - 190 PROOF
FIG. 5.43. COMPARISON OF PREDICTED AND MEASURED CYLINDER PRESSURE - STRAIGHT DIESEL
Figs. 5.44 and 5.45 present the comparison of measured cylinder pressures with computed diagrams during the dual fuel operations at the maximum percentage of diesel substitutions with 190 (59.2%) and 170 (57.9%) proofs ethanol respectively. Compared with measured values the model predicts a higher peak pressure of 2.0 bar during the dual fuel operation with 190 proof ethanol. However, a closer agreement has been observed between the measured and predicted values with 170 proof ethanol.

5.11.3.2 ADVANCED AND RETARDED TIMINGS

The response of the model to changes in the injection timings of 5° advance and 5° retard from the base line timing of 13° BTDC was also investigated. Fig. 5.46 compares the measured values of pressure with the predicted diagram during the advanced injection timing at the maximum diesel substitution of 51.6 percent (150 proof). Though the delay is higher, advancing the injection timing causes the combustion to occur before TDC, thus resulting in a rapid pressure rise. It is also seen that the magnitude of measured peak pressure is higher than the predicted value by about 4 bar.

Fig. 5.47 shows the comparison of measured pressure with the predicted diagram during the dual fuel operation with retarded injection timing of 8° BTDC. It is seen that the occurrence of peak pressure is shifted away from TDC by 20° CA. As the bulk of the combustion take place during the
FIG. 5.44. COMPARISON OF PREDICTED AND MEASURED CYLINDER PRESSURE - ETHANOL 59.2%
**FIG. 5.45.** COMPARISON OF PREDICTED AND MEASURED CYLINDER PRESSURE - ETHANOL 57.9%
FIG. 5.46. COMPARISON OF PREDICTED AND MEASURED CYLINDER PRESSURE - ETHANOL 51.6%
FIG. 5.47. COMPARISON OF PREDICTED AND MEASURED CYLINDER PRESSURE - ETHANOL 50.7%
expansion process, compared with the base line operation, a reduction of 2.5 percent in the value of thermal efficiency has been observed. While the measured peak pressure is lower by 2 bar, the measured pressures are marginally higher than the predicted values during the main part of the expansion process.

5.11.3.3 CYLINDER PRESSURE DIAGRAMS AT 1300 RPM

Figs. 5.48 5.49 and 5.50 indicate the comparison of the predicted and measured values of cylinder pressure during the dual fuel operations at 1300 RPM for various operating conditions. It could be seen that at 70 percent (190 proof) as well as at 52.9 percent (150 proof) energy shares, the bulk of combustion occurs during the expansion process. However, it is observed that the thermal efficiencies are still comparable with straight diesel operation. Typical pressure-crank angle diagrams, superimposed with crank degree marks together with either needle lift trace or \( \frac{dp}{d\theta} \) trace are seen in plates 5.2 and 5.3. Table 5.1 shows the summary of the experimental results and the computed in-cylinder parameters at 1500 RPM and at full load for various proofs of ethanol. It is seen that a fair agreement exists between the two. Similar trends have been observed for the other speed.
FIG. 5.48. COMPARISON OF PREDICTED AND MEASURED CYLINDER PRESSURE - ETHANOL 21.5%
FIG. 5.49. COMPARISON OF PREDICTED AND MEASURED CYLINDER PRESSURE - ETHANOL 70%
FIG. 5.50. COMPARISON OF PREDICTED AND MEASURED CYLINDER PRESSURE - ETHANOL 52.9%
PLATE 5.2. TYPICAL RECORDED PRESSURE - CRANK ANGLE TRACES - INJ 13° BTDC
PLATE 5.3. TYPICAL RECORDED PRESSURE-CRANK ANGLE TRACES AT FULL LOAD AND AT ADVANCED AND RETARDED INJ. TIMINGS (FOR FURTHER DETAILS REFER TABLE 5.2)
TABLE 5.1

Predicted and experimental results of peak pressure and rate of pressure rise with quantum of ethanol at different proofs.

<table>
<thead>
<tr>
<th>S. No.</th>
<th>Proof/percent</th>
<th>Peak pressure (bar)</th>
<th>Rate of pr. rise (bar/°CA)</th>
<th>Predicted</th>
<th>Measured</th>
<th>Predicted</th>
<th>Measured</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Straight Diesel</td>
<td>64.0</td>
<td>64.5</td>
<td>3.52</td>
<td>3.40</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>190/22.2</td>
<td>67.45</td>
<td>65.5</td>
<td>3.92</td>
<td>4.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>190/47.9</td>
<td>63.27</td>
<td>61.0</td>
<td>2.86</td>
<td>3.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>190/59.2</td>
<td>53.52</td>
<td>51.5</td>
<td>1.61</td>
<td>1.75</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>170/21.1</td>
<td>66.50</td>
<td>68.0</td>
<td>4.17</td>
<td>4.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>170/44.6</td>
<td>57.34</td>
<td>55.5</td>
<td>3.45</td>
<td>3.27</td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>170/57.9</td>
<td>49.50</td>
<td>50.5</td>
<td>1.63</td>
<td>1.80</td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>150/21.8</td>
<td>66.80</td>
<td>68.5</td>
<td>4.22</td>
<td>4.11</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>150/42.8</td>
<td>52.57</td>
<td>50.0</td>
<td>3.28</td>
<td>3.44</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>150/48.1</td>
<td>41.72</td>
<td>43.5</td>
<td>1.31</td>
<td>1.24</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
5.12 RESULTS OF EXPERIMENTAL INVESTIGATIONS

5.12.1 INTRODUCTION

Manifold admission of ethanol has been experimented in compression ignition engines of various types used in practice. The quantum of ethanol that could be effectively burnt along with diesel varies greatly with the type of engine and operating conditions and is generally limited either by the occurrence of knocking or combustion quenching.

As reported by earlier investigators and confirmed in the present investigation, dual fuel operations always result in an increased delay period and this extended delay causes the occurrence of knocking or misfiring depending upon numerous factors, like air/ethanol ratio, proportion of diesel injected, type of engine and other operating conditions. A drop in thermal efficiency is always observed at lower loads, which can be attributed to the high overall air/ethanol ratio and the shorter penetration of diesel flame. However, at higher loads, induction of ethanol results in higher thermal efficiency values compared to straight diesel operation.

The results of experimental investigations carried out for understanding the dual fuel combustion mechanism in different types of engines are presented and discussed in the following sections.
5.12.2 EFFECT OF ETHANOL INDUCTION ON THE MASS. RATE OF AIR FLOW

It was believed that the induction of ethanol along with the intake air, due to its greater evaporative cooling would reduce the temperature of the charge and hence improve the mass flow rate of inducted air. However, experimental results do not indicate any such improvement. The mass rate of flow of air to the engine plotted against the percentage of diesel substitution with 190 proof ethanol for the speeds of 1300 and 1500 RPM is shown in Fig.5.51. It could be seen that the mass flow rates practically remain constant during the entire dual fuel operations. The lower molecular weight of ethanol vapour, on partial evaporation, might displace proportional volume of air and the beneficial effect of the reduction in the temperature of the charge is compensated by the displacement of air. Thus the rate of flow of air remains constant. This is also true for other speeds and proofs of ethanol and in conformity with the observations made by Starkman et.al [66].

5.12.3 EFFECTS OF PROOF ETHANOL ON THE PERFORMANCE OF DUAL FUEL OPERATION

Figs.5.52 and 5.53 represent the summary of typical results of the comparative performance of straight diesel versus dual fuel operations at 1300 RPM and 1500 RPM and at full load, as a function of the percentage of diesel
FIG. 5.51. VARIATION OF AIR FLOW WITH ETHANOL ADMISSION
FIG. 5.52. VARIATIONS OF MEASURED THERMAL EFFICIENCY, PEAK PRESSURE AND DELAY PERIOD WITH PERCENTAGES OF DIESEL SUBSTITUTION AT 1300 RPM
FIG. 5.53. VARIATIONS OF MEASURED THERMAL EFFICIENCY, PEAK PRESSURE AND DELAY PERIOD WITH PERCENTAGES OF DIESEL SUBSTITUTION AT 1500 RPM
substitution for different proofs of ethanol. It could be seen that there is a marginal improvement in brake thermal efficiency for both the speeds with increasing percentages of diesel substitution. This improvement in thermal efficiency may be attributed to many favourable factors associated with dual fuel operation. At higher loads the evaporation of ethanol appears to be more complete due to the higher temperature levels of the engine components. This might result in quicker mixture formation and preparation of both the fuels. Also, the increased ignition delay, associated with ethanol carburetion, provide a rapid and increased constant volume combustion close to TDC. These factors cause a part of fuel to burn earlier which otherwise will burn late in the cycle and contribute in increasing the expansion ratio. The reduced pumping losses, associated with the compression of an initially cooler charge, also contribute to the increase in thermal efficiency.

However, increasing the percentage of ethanol induction over and above 55 percent and 45 percent for 1300 RPM and 1500 RPM respectively indicate a decreasing trend in the values of thermal efficiency. Upto this point, the delayed ignition and combustion are compensated by the increased constant volume combustion and higher flame speed of ethanol. Any further increase in the percentage of ethanol induction results in the bulk of combustion taking place well after the commencement of expansion stroke. This increases the after
burning and reduces the effective expansion ratio as evidenced by the increase in the exhaust temperature and pressure. It is observed that at the points of maximum percentages of diesel substitution the exhaust back pressure increased by about 3 to 5 percent compared to the straight diesel operation. The exhaust temperatures at these points also increase by about 15°C to 25°C compared to the lower percentages of diesel substitution. It could also be observed from Figs.5.52 and 5.53 that the proof of ethanol had no apparent effect on the thermal efficiency of the engine. However, the percentage of ethanol that could be admitted before the onset of knock or rough running decreases with lowering the proof of ethanol. While it is possible to substitute 70 percent of diesel at 1300 RPM by 190 proof, the percentage decreases to 62 percent and 52.9 percent for 170 proof and 150 proof respectively. Similar trend has also been noticed for the other speed of 1500 RPM.

It has also been observed that the fuel economy suffers during dual fuel operations at light loads. This phenomena could be explained by many contributing factors, like long ignition delay, reduced quantum of diesel injection and increased levels of after burning of the fuels. As the quantity of diesel injection is reduced, large droplets with shorter penetration are likely to be introduced and some of the inducted ethanol particles may not be enveloped by the flame. In view of the above considerations and due to poor
mixture formation and incomplete combustion, substantive energy losses are likely to occur resulting in poor fuel economy.

The variations of peak pressure at full load with the quantum of the various proofs ethanol induction are also shown in Figs. 5.52 and 5.53 for 1300 RPM and 1500 RPM respectively. The peak pressure and rate of pressure rise depend on the proportion of ethanol vapour taking part in the uncontrolled combustion and are mainly governed by the delay period as well as the spray envelop of the injected diesel. It could be seen that the peak pressure decreases markedly at higher levels of ethanol admission and significantly increases during lower flow rates of ethanol.

It is quite evident from these figures that for all proofs of ethanol the maximum peak combustion pressures occur within a narrow range of 15 to 20 percent of diesel substitutions. Compared with straight diesel operation, the peak pressure increases to a maximum of 5.8 percent and 6.4 percent at 1300 RPM and 1500 RPM respectively. It is also seen that at the points of maximum percentages of diesel substitution the peak pressures are lowered far below that of straight diesel operation. Probably, factors such as reduced penetration of diesel spray, long ignition delay and commencement of combustion during the expansion stroke may be attributed for the reduction in peak pressure values.
5.12.4 DURATION OF INJECTION AND PERIOD OF INITIAL RAPID COMBUSTION

Typical results of injection duration of diesel, delay period, peak pressure, duration between the occurrences of peak pressure and ignition, mean rate of pressure rise and exhaust temperature are presented in Fig. 5.54 as a function of 190 proof ethanol admission at 1500 RPM and at full load. Diesel substitution amounting to about 35 percent by energy, gradually reduces the duration of $Q_{\text{Pmax}} - Q_{\text{ign}}$ from 11.5 to 9.5 and any further increase in the quantum of ethanol increases the duration. During the period where the duration $Q_{\text{Pmax}} - Q_{\text{ign}}$ decreases, the peak pressure and rate of pressure rise show an increasing trend.

It is also seen that diesel substitutions in excess of about 35 percent cause the injection of diesel to be completed before the start of ignition. Though this condition might allow substantial evaporation and mixing of diesel particles with ethanol vapour before the start of ignition, the peak pressure and rate of pressure rise are lowered due to the factors discussed already. With increasing proportion of ethanol admission, the exhaust temperature dropped initially and then gradually increased. This indicates that for small quantities of ethanol admission the combustion improves and at higher proportions the attendant evaporative cooling of the charge increases the ignition delay and causes late burning of the fuels.
FIG. 5.54. VARIATIONS OF MEASURED DURATION OF INJECTION, DURATION OF $\theta_{p_{\text{max}}} - \theta_{\text{ign}}$, EXHAUST TEMPERATURE AND RATE OF PRESSURE RISE WITH VARIOUS PERCENTAGES OF ENERGY SHARE
5.12.5 EFFECT OF ADVANCING OR RETARDING THE INJECTION TIMING OF DIESEL

Investigations were also carried out to evaluate the performance of the dual fuel engine with advancing and retarding the injection of diesel by 5° from the base line timing of 13° BTDC. The temperature and pressure of the charge prevailing at the points of advanced or retarded injection differ from those of standard timing and these influence the ignition delay and the maximum percentage of ethanol admission. The maximum percentage of ethanol that could be admitted before the occurrence of either misfiring or rough running increases marginally with advancing the injection timing. The maximum percentage of various proofs of ethanol, the ignition delay and the corresponding peak pressure are given in Table 5.2 for the injection timings tried in this investigation. Though the delay period is higher with increased advance of injection, it is seen that substantial amount of heat is released nearer to TDC position which is conducive for taking in larger quantum of ethanol. However, the maximum percentages of diesel substitution are reduced at retarded timing. It could also be observed that for any injection timing, the maximum percentage of diesel substitution decreases with lowering the proof of ethanol.

5.12.6 EXHAUST TEMPERATURE

The effect of dual fuel operation on the exhaust temperature are presented in Fig.5.55 as a function of
### TABLE 5.2

Variations of maximum percentage of Diesel substitutions, Delay period and Peak pressure with advanced and retarded injection timings.

<table>
<thead>
<tr>
<th>PARTICULARS</th>
<th>Proof of Ethanol</th>
<th>Speed</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>190</td>
<td>170</td>
</tr>
<tr>
<td></td>
<td></td>
<td>RPM</td>
</tr>
<tr>
<td>Inj.Timing BTDC</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>13</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>13</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>13</td>
</tr>
</tbody>
</table>

| Maximum percentage of Diesel Substitution | 64.4 (P4) | 70.0 (P1) | 72.6 (P5) | 59.3 (P2) | 62.0 (P6) | 64.8 (P3) | 50.3 (P5) | 52.9 (P1) | 55.6 (P9) | 1300 |
| Delay period (°CA)      | 14.0 (P10) | 16.0 (P7) | 16.5 (P11) | 14.0 (P8) | 14.5 (P12) | 14.0 (P9) | 14.0 (P10) | 14.5 (P7) | 16.5 (P11) | 1500 |
| Peak pressure (bar)     | 45.0 (P10) | 50.0 (P7) | 57.0 (P11) | 43.5 (P8) | 50.5 (P12) | 61.5 (P9) | 61.0 (P10) | 51.0 (P7) | 51.0 (P11) | 1300 |
|                        | 44.0 (P10) | 52.0 (P7) | 52.0 (P11) | 38.5 (P8) | 50.5 (P12) | 38.0 (P9) | 38.0 (P10) | 52.0 (P7) | 52.0 (P11) | 1500 |
FIG. 5.55. VARIATIONS OF EXHAUST TEMPERATURE AS A FUNCTION OF PERCENTAGE OF ETHANOL ENERGY - 1500 RPM
percentage of diesel substitution at 1500 RPM and at full load. These results are typical of those obtained by carburett ing various proofs of ethanol during the dual fuel operations of advanced and retarded injection timings. This figure reveals that for various injection timings tried in this investigation, the exhaust temperatures drop initially to a maximum of about 10 to 15 percent and then increases marginally as the percentage of diesel substitution approaches the maximum value.

The reduction in the exhaust temperature are indicative of the advancement of heat release or an increase in the expansion ratio as well as the combined effects of charge cooling and lower flame temperature of ethanol combustion. Though the effect of charge cooling and flame temperature are more significant, the small increase in the exhaust temperature may be attributed to the increased after burning at higher values of ethanol induction. Similar trends have been observed at the other speed.

5.12.7 PRE-HEATING OF AIR-ETHANOL MIXTURE

As stated earlier, the compression temperature and pressure of the charge are significantly lowered to varying degrees, depending upon the quantity and quality of ethanol inducted. This may be attributed to the high latent heat of ethanol as well as the increased heat capacity of the mixture. However, preheating the air-ethanol mixture gives
rise to a different effect on the operation of dual fuel system. On complete evaporation of ethanol the lower molecular weight of ethanol causes a reduction in the mass rate of air drawn into the engine. This phenomenon causes a reduction in the density of the charge significantly. At the point of the maximum energy substitution amounting to about 60 percent and at full load, pre-heating the mixture to 35°C results in a reduction of 6 percent in the mass rate of flow of air into the engine. A 11 percent reduction in the mass rate of air flow to the engine was observed when the mixture is heated to 50°C. At 50 percent diesel substitution and at a mixture temperature of 110°C the mass rate of air is reduced by 14.5 percent. Variations in ignition delay, peak pressure and rate of pressure rise as a function of percentage of diesel substitution are illustrated in Fig.5.56 for mixture temperatures of 35°C and 50°C at 1500 RPM. It could be seen that while there is a marginal difference in the ignition delay at lower percentages of energy share, at higher percentages the variation in delay is significant with regard to the temperature of the mixture. Increasing the temperature of the charge results in a smaller increase in the delay period with the percentage of energy share. At 60 percent energy share raising the mixture temperature to 50°C results in an ignition delay increase of 27 percent and 20 percent at 3/4th load and full load respectively. From the above it follows that at higher loads the increase in delay is smaller...
FIG. 5.56. VARIATIONS OF MEASURED IGNITION DELAY, PEAK PRESSURE AND RATE OF PRESSURE RISE WITH PERCENTAGE OF ENERGY SHARE AT VARIOUS MIXTURE TEMPERATURES
and the same can be attributed to the higher gas and component temperatures.

It may be also realised that the influence of reduction in the concentration of oxygen as a result of reduced air flow has less impact on the ignition delay than the increase in the mixture temperature. Despite the marginal increase in the delay at lower energy shares, increasing the mixture temperature results in a higher peak pressure and rate of pressure rise. However, at higher percentages of energy share, it is seen that both the peak pressure and rate of pressure rise tend to fall below the value of straight diesel operation as a result of delayed ignition taking place in the expansion stroke. As increasing the mixture temperature beyond 50°C resulted in pressure fluctuations and knocking combustion at higher values of diesel replacement, higher temperature dual fuel operations were not tried.

The variations of measured delay, peak pressure, mean rate of pressure rise and the duration between the occurrence of peak pressure and the start of ignition which reasonably indicate the burning period are presented in Fig.5.57 as a function of mixture temperature at a constant energy share of 50 percent and at different power outputs. A significant decrease in the delay period with increasing temperature of the mixture is evident from the diagram. Besides, it can be seen that the peak pressure and the rate of pressure rise show a steady increase with the temperature of the mixture.
FIG. 5.57. VARIATIONS OF MEASURED INCYLINDER PARAMETERS AS A FUNCTION OF TEMPERATURE
This is in good agreement with the results of Eston et.al [67]. The improvements in the evaporation and mixture preparation of the fuels prior to the start of ignition at high temperature, greatly contribute for the heat release to be weighted more towards the premixed mode of combustion than the diffusion mode. Besides, it is believed that ethanol molecules might become chemically very active and this might cause changes in the combustion characteristics of the mixture by promoting a higher speed of flame propagation. These factors contribute for the initial rapid burning of the mixture and hence the smaller duration of burning, identified by $Q_{p_{\text{max}}} - Q_{\text{ign}}$ as shown in the figure.

The reduction in the delay period and the increase in the peak pressure and rate of pressure rise appear to be linear with the temperature of the air-ethanol mixture. Further, it is also seen that the temperature of the mixture up to which the engine could be operated smoothly decreases with load, as the higher temperature is expected to result in Otto type detonation rendering the rough running of the engine.

5.12.8 EXHAUST EMISSIONS

It is known that the formation of oxides of nitrogen are the result of the nitrogen oxidation by oxygen of the ambient air and predominantly promoted by the operating conditions
that provide oxygen rich combustion at elevated temperatures and long residence time. Experimental results obtained for oxides of nitrogen as well as CO emissions are plotted in Figs. 5.58 and 5.59 as a function of percentage of energy share for speeds of 1300 and 1500 RPM at full load.

While the peak concentration of oxides of nitrogen occur during the straight diesel operation, the emission of oxides of nitrogen gradually decreases with the addition of ethanol, as a result of thermal quenching of NO/NOx producing reactions. Compared with straight diesel operation, dual fuel operation results in the reduction of both NO and NOx emissions to the extent of 50 percent. Lower values of reduced NOx emissions in the case of dual fuel operations may be attributed to the following factors:

(a) The significantly higher heat of vaporization and increased mass flow of ethanol primarily contribute to the reduced cycle temperature.

(b) The higher flame speed of ethanol combustion reduces the time available for the high temperature NOx formation reactions to complete.

(c) Lower adiabatic flame temperature of air-ethanol mixture.

The first two factors are supported by the values of computed temperatures, presented in the earlier sections as well as the measured exhaust temperature as indicated in Figs. 5.58.
and 5.59. The second one is also supported by the measured drop in the duration between the start of ignition and the occurrence of the peak pressure.

It is also observed that lowering the proof of ethanol further causes a reduction in the oxides of nitrogen. The reason for such a result may be due to the contained water in ethanol fuel, which act as a thermal sink and contribute to a further drop in the combustion temperature.

Increasing the percentage of energy share, keeping the mixture temperature constant also exhibit a reduction in the measured values of NO/NOx. At full load and at maximum substitution level during the constant mixture temperature of $35^\circ$C and $50^\circ$C, the oxides of nitrogen emission are about 30 to 35 percent less than the straight diesel operation, thus justifying the effect of lower combustion temperature of ethanol.

Compared with standard injection timing of $13^\circ$ BTDC (dynamic), advancing or retarding the injection of diesel show a different trend in the emission of oxides of nitrogen as shown in Fig.5.60. While the injection advance show an initially increasing trend upto 45 percent of energy share and a rapid decrease thereafter, the retarded timing results in a gradual decrease of the NO emission. Increasing trend of NO emission in the case of advanced injection timing could be attributed to the commencement of combustion well before TDC, which might increase the temperature of the cylinder gas.
FIG. 5.58. EFFECT OF VARIOUS PERCENTAGES OF PROOF ETHANOL ON NO/NOx FORMATION AND EXHAUST TEMPERATURE
FIG. 5.59. EFFECT OF VARIOUS PERCENTAGES OF PROOF ETHANOL ON NO/NOx FORMATION AND EXHAUST TEMPERATURE
FIG. 5.60. EFFECT OF INJECTION TIMING ON THE FORMATION OF NO₃ AS A FUNCTION OF ETHANOL ENERGY SHARE
From the measured CO emissions, shown in Figs. 5.58 and 5.59, it is seen that the percentages of CO emission increase with increase in the percentage of energy share for different proofs of ethanol tried in this investigation. The existence of local heterogeneity of the mixture could be the one of the most dominant factors for the formation of CO. Increasing the percentage of energy share can be expected to introduce higher local fuel rich zones due to the possibilities of charging the cylinder with more unevaporated ethanol droplets and continuing evaporation of ethanol in the process of compression.

While the CO concentration at 1300 RPM remained at about 50 percent of the values obtained at 1500 RPM, the percentage increase with the energy share for both speeds seem to be nearly equal. The smaller values of CO at 1300 RPM could be due to the lesser heterogeneity of the mixture resulting from the longer time of induction and compression processes. It was also observed that the admission of air-ethanol mixture at elevated temperatures showed a reduced CO emission of about 30 to 40 percent due to the fact that heterogeneity of the charge might be reduced as a result of the absence of ethanol droplets entering into the combustion chamber.

5.12.9 EFFECT OF NOZZLE OPENING PRESSURE

The effects of increasing the injection pressure of diesel fuel by about 10 percent over the recommended value of
185 atm for straight diesel operation, on the performance of
dual fuel engine in respect of ignition delay, thermal
efficiency, peak combustion pressure and exhaust temperature
are presented in Fig. 5.61 for 1500 RPM and 75 percent full
load.

It could be seen that a 10 percent increase of the
injection pressure reduces the ignition delay periods
marginally at higher percentages of diesel substitution. This
decrease in the delay period might be due to better
atomization resulting from higher injection pressure, which
is believed to decrease the mean diameter of the diesel
particles as well as the variations in the droplet size.
These factors enable a quicker heating and evaporation of
diesel particles as a result of the higher surface to volume
ratio and heat transfer coefficient.

It could also be observed that higher injection pressure
causes a marginal increase in thermal efficiency and peak
combustion pressure. Increasing the injection pressure is
likely to result in the formation of larger quantity of
diesel vapour at the start of ignition coupled with higher
penetration and dispersion of diesel spray. The combined
effects of all these factors might possibly increase the
constant volume combustion which is reflected by the
increased values of thermal efficiency and peak pressure.

A small reduction in the exhaust temperature is also
observed from the fig. 5.61. This drop may be due to the
FIG. 5.61. EFFECT OF INJECTION PRESSURE ON DUAL FUEL OPERATION AT VARIOUS PERCENTAGES OF ENERGY SHARE

RPM: 1500 RPM
LOAD: 75% FULL LOAD
ETHANOL: 190 PROOF

- 205 Atm
- 185 Atm

PEAK. PR. (bar)
DELY. (CA)
EXH. TEMP. (°C)
THER. EFF. (%)

PERCENTAGE OF ENERGY SHARE
increased constant volume combustion heat release as well as a reduction in the delay period both of which give rise to a higher expansion ratio and results in a lower exhaust temperature. No significant improvements in the performance of the engine has been noticed with further increase in injection pressure.

5.13 EXPERIMENTAL RESULTS FROM DUAL FUELED AUTOMOTIVE C.I. ENGINE

The experimental investigations carried out on a six cylinder automotive diesel engine revealed that the percentage of diesel substitution primarily depend on the load and speed of the engine and varies between 26 to 48 percent over a wide range of engine speeds [68]. Thermal efficiency values of dual fuel operations were found to be lower at light loads and comparable at higher loads. The drop in efficiency at lower loads could be mainly attributed to long ignition delay resulting in the combustion to take place in the expansion process and incomplete combustion due to charge cooling and lower operating temperatures. At higher loads, there was an improvement owing to larger heat release around the TDC position.

Preheating of the intake air enabled higher percentages of ethanol induction which varied from 32 to 60 percent. A marginal improvement in the specific fuel consumption compared to straight diesel and ambient air operations was also observed.
The influence of ethanol admission on the thermal efficiency at full load for the various speeds are shown in Fig.5.62. It could be seen that the thermal efficiency decreases at lower speeds and there is a gradual improvement in the values of thermal efficiency at higher speeds. An increase of 8 and 17 percent in the efficiency values could be obtained during preheated operation over pure diesel and ambient air operations respectively.

Few runs were also made with ethanol mixed with one percent of cyclo hexonal, an ignition improver. From Fig.5.63 it is observed that the use of such an additive improves the performance of the engine especially at lower loads and the efficiency values are higher than the straight diesel and dual fuel (ambient air) operations throughout the load range. It is believed that the presence of the additive contributes greatly to the presence of number of active particles and enables the early ignition of the charge at a lower temperature thereby resulting in earlier heat release.

The cylinder liner temperatures of the engine at 24 locations corresponding to the top horizontal plane, 10 cm from the cylinder gasket are shown in Fig.5.64 for straight diesel and dual fuel with ambient and preheated air operations at full load and at 1600 RPM. It could be seen that there is a drop in the liner temperatures during dual fuel operation as compared to straight diesel operation. This may be attributed to the lower flame temperature of ethanol.
FIG. 5.62. INFLUENCE OF SPEED UPON THERMAL EFFICIENCY AT FULL LOAD

FIG. 5.63. VARIATION OF THERMAL EFFICIENCY WITH ADDITIVE AT VARIOUS TORQUE LEVELS
a) neat diesel operation

b) bifuel operation with ambient air

c) bifuel operation with preheated air

FIG. 5.64. CYLINDER LINER TEMPERATURES AT 1600 RPM AND AT FULL LOAD
combustion, higher latent heat of evaporation and increased specific heat values of the constituents. These reductions are likely to reduce the heat losses to the cooling water. Preheated air operation does not cause much variation in the liner temperatures as compared to ambient air dual fuel operation.

5.14 PERFORMANCE CHARACTERISTICS OF DUAL FUELED TURBOCHARGED C.I. ENGINE

General trends observed during this phase of the investigation, among others, include an overall improvement in thermal efficiency at higher loads and some penalty at lower loads. It could be seen from Fig.5.65 that an improvement in thermal efficiency to the tune of 12 percent at higher loads for the speeds of 1200 and 1500 RPM has been achieved during dual fuel operations. This may be attributed to better vaporization and a more homogeneous air-ethanol charge resulting from the compression of the charge in the turbocharger as well as better heat exchange from components which are relatively at a higher temperature. The condition of the charge is more conducive for rapid heat release through nearly constant volume combustion. Besides, the faster burning of air-ethanol mixture at the early stages of combustion results in a higher expansion ratio and reduced heat loss. The combined effect of all these factors thus helped in the substantial improvement in the value of thermal efficiency at higher loads. However, dual fuel operations at
FIG. 5.65. VARIATION OF THERMAL EFFICIENCY AND AIR/ETHANOL RATIO WITH OUTPUT
lower loads and speeds, due to increased heat loss attributed to enhanced ignition delay, caused a reduction in thermal efficiency [69].

The effects of inducting ethanol, to the limit just before the cylinder pressure fluctuations become apparent, on the peak combustion pressure and ignition delay period are shown in Fig.5.66 at 1200 and 1500 RPM and at various loads. It could be seen that the peak pressures were always higher during dual fuel operations, except at very light loads and the increase in peak pressures were found to vary between 10 to 15 percent. This is in contrast to the trend noticed in naturally aspirated engines where the peak combustion pressures exhibit an initial increase followed by a decrease with increasing percentages of ethanol admission at higher loads.

Similar to the observations made on naturally aspirated engines, the delay period during dual fuel operation of turbocharged engine was always higher than that obtained during straight diesel operation. However, the increase in the delay period with increase in the load was found to be smaller in the case of turbocharged engine. Also the increase in delay was found to be higher at lower speeds than at high speeds. It may be noted that the increase in the delay period gradually decreases with increase in the load. This trend may be attributed to the increased quantity of available oxygen, as a result of higher input energy to the turbine, which
FIG. 5.66. VARIATIONS OF DELAY AND PEAK PRESSURE WITH LOAD
possibly compensate the reduction in the compression temperature and pressure of the charge obtained during the dual fuel operations. An increase of 32 percent and 26 percent in the mass flow rates of air could be achieved at 1500 RPM and 1200 RPM respectively at full load, as compared to mass flow rates at no load.

The variation of thermal efficiency, mean rate of pressure rise, exhaust temperature and boost pressure as a function of energy ratio, defined as ethanol energy/diesel energy, are presented in Fig.5.67 for a speed of 1500 RPM and at full load bmep of 9.2 bar. It could be seen that increasing the energy ratio resulted in an increase in the rate of pressure rise and thermal efficiency and a reduction in the exhaust temperature and boost pressure. While the rate of pressure rise increased by 52 percent at the knock limited energy ratio of 0.65, an energy ratio of 1.25 with a tolerable knock, causes a 100 percent increase, compared with straight diesel operation. The corresponding increase in the delay periods are only 3°CA and 4°CA respectively. Rate of pressure rise and thermal efficiency are possibly increased by the rapid heat release due to a large portion of the mixture burning under constant volume conditions. The fast burning characteristics of ethanol coupled with lower flame temperature results in the reduction of the exhaust temperature to the extent of 90°C which in turn reduces the available energy at the turbine inlet causing a reduction in
FIG. 5.67. VARIATIONS OF THERMAL EFFICIENCY, RATE OF PRESSURE RISE, EXHAUST TEMPERATURE AND BOOST PRESSURE WITH ENERGY RATIO
the boost pressure from 0.65 bar (gauge) to 0.52 bar (gauge) as shown in the figure.

The observed variations of the mean rate of pressure rise, thermal efficiency, air/ethanol ratio and the knock limited energy ratio are shown in Fig.5.68 over a speed range of 1200 to 1700 RPM and at full loads. It could be seen that compared with straight diesel operation, the dual fuel operation shows a higher thermal efficiency of 6 to 12 percent over the entire speed range. Similar to the trend observed in a naturally aspirated multicylinder dual fuel compression ignition engine, it is noticed that the knock limited energy ratio decreases with increase in speed. This reduction might be attributed to the higher turbulence level prevailing inside the cylinder and increased delay in terms of the crank angle. The combined effects of higher turbulence and ignition delay result in the combustion of a large quantity of air-ethanol mixture to take place at constant volume and limit the substitution of diesel. It is also seen that the air/ethanol ratio varies only marginally between 26 and 29 percent during the entire range of speeds tried in these investigations.

5.15 LONG DURATION ENDURANCE TESTS

At the end of 500 hours endurance running programme, the two identical single cylinder engines were dismantled to examine the effects of diesel and hydrated ethanol of 190
Figure 5.68: Variations of thermal efficiency, rate of pressure rise, energy ratio and air/ethanol ratio with speed.
proof dual fuel operations on the wear of engine components and engine deposits. On the inspection of the cylinder components it is observed that the components of dual fuel engine, like piston, valve seats and cylinder-head were neat and clean with lesser lacquering and deposits, compared with the components of neat diesel engine. Factors like lower C/H ratio of ethanol as well as the reduced cracking of fuel as a result of lower flame temperature of ethanol combustion can be attributed to this trend.

Throughout the duration of testing, both engines did not show any performance deterioration of lubricating oil degradation. Analysis of used oil samples, taken at regular intervals of 100 hours, showed satisfactory results in respect of variations in flash point and viscosity. At an ambient temperature of 32°C, while the viscosity of used oil samples removed from the dual fuel engine showed a shift of -5 percent from the rating of the fresh oil, possibly due to the escape of ethanol particles to the crankcase, the samples taken from the straight diesel engine showed a 4 percent increase as a result of contamination by the products of combustion. However, at higher temperatures the viscosity of the used oils remained fairly constant. It is possible that the high temperature would evaporate ethanol which might have entered into the crankcase. Lubricating oil consumption in both engines were within the permissible value.
Though it was believed that ethanol and the contained water and corrosive combustion products, like aldehydes and formic acid, could attack the engine components and contribute to increased corrosion and wear of engine components, the observed results shown in Table 5.3 indicate that the total wear of the liner and piston rings are comparable with the values obtained from the straight diesel operated engine.

Examination of the above results reveal that there is no evidence of corrosive attack on the piston, piston rings and liner by the contained water or the products of ethanol combustion. Compared with the straight diesel engine the exhaust port of the dual fuel engine was found to have reduced carbon deposits and lacquer and the valve seats exhibited the absence of guttering. Dimensional changes in the gudgeon pin, big and small end bearings of connecting rod as well as the main bearings of the crank shaft were observed to be minimal in both the engines.

5.16 INFLUENCE OF COMBUSTION CHAMBER CONFIGURATIONS

The variations in thermal efficiency as a function of the percentage of diesel substitution for different combustion chamber configurations are presented in Fig.5.69 at 1500 RPM and at full load. Compared with straight diesel operation for the various configurations tried, increasing the percentage energy share of ethanol results in a
<table>
<thead>
<tr>
<th>Particulars</th>
<th>I</th>
<th>Compression Rings</th>
<th>II</th>
<th>III</th>
<th>Oil Control Ring</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>DF</td>
<td>SD</td>
<td>DF</td>
<td>SD</td>
<td>DF</td>
</tr>
<tr>
<td>Piston Ring gap, mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Before Test</td>
<td>.28</td>
<td>.27</td>
<td>.30</td>
<td>.30</td>
<td>.30</td>
</tr>
<tr>
<td>After Test</td>
<td>.36</td>
<td>.34</td>
<td>.335</td>
<td>.35</td>
<td>.36</td>
</tr>
<tr>
<td>Percentage change</td>
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<td>25.9</td>
<td>11.6</td>
<td>16.6</td>
<td>20.0</td>
</tr>
<tr>
<td>Ring Weight, gms</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Before Test</td>
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<td>12.791</td>
<td>12.901</td>
<td>13.182</td>
<td>12.996</td>
</tr>
<tr>
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<td>12.64</td>
<td>12.81</td>
<td>13.062</td>
<td>12.855</td>
</tr>
<tr>
<td>Percentage Change</td>
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<td>1.20</td>
<td>0.70</td>
<td>0.90</td>
<td>1.08</td>
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<td>Cylinder Bore, mm</td>
<td></td>
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<td></td>
</tr>
<tr>
<td>side</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Non Thrust Before</td>
<td>79.99</td>
<td>79.99</td>
<td>79.99</td>
<td>79.99</td>
<td>79.99</td>
</tr>
<tr>
<td>side</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>DF: DUAL FUEL ; SD: STRAIGHT DIESEL.</td>
<td></td>
<td></td>
<td></td>
<td></td>
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</tbody>
</table>
FIG. 5.69. VARIATIONS OF THERMAL EFFICIENCY WITH VARIOUS PERCENTAGES OF ETHANOL ENERGY
substantial improvement in the values of thermal efficiency. It could also be seen that only marginal differences in thermal efficiency exist between the standard and toroidal pistons during the dual fuel operations.

Though the use of standard piston gave a maximum gain of thermal efficiency of only 11.5 percent, the improvements in other chambers vary from 25 to 39 percent, the maximum being for the smaller hemispherical chamber [70]. It is also evident that the maximum thermal efficiency of all the chambers occurs within a narrow range of 40 to 50 percent of diesel substitution. Higher percentages of ethanol induction extend the ignition delay period such that the injection of diesel terminates well before the start of combustion in which case the organised movement of air may not have any decisive influence in the process of combustion as the problem of distribution and mixing are now restricted to a smaller quantity of injected fuel.

It may also be noted that the toroidal chamber accepts a lower percentage of ethanol share before the engine starts to run roughly. The reduction in the percentage substitution of ethanol in the case of toroidal piston might be due to the intense mixing of diesel and air-ethanol mixture, which may account for the rapid heat release, consequent to the ignition of diesel fuel. It is also noticed that there is a significant increase in thermal efficiency values with increase in the percentage of diesel substitution for the
other two pistons with oblong and smaller hemispherical shapes. At higher levels of ethanol substitution thermal efficiency values are comparable to that of standard and toroidal pistons.

The influence of spray pattern of diesel on the performance as well as on the limit of maximum diesel substitution was also studied. From the data obtained it is observed that the combination of a single hole injector and toroidal piston allowed a maximum of 79 percent of diesel substitution with a marginal improvement in thermal efficiency. The higher percentage of diesel substitution might be possible due to the reduced volume of air-ethanol mixture being enveloped by the single spray thus permitting only a smaller volume of air-ethanol mixture to burn along with diesel, the remaining mixture being burnt in a manner similar to the combustion process in a S.I. Engine.

From the foregoing, it is believed that at higher levels of diesel substitution the piston configuration becomes less critical as the thermal efficiencies of all the chambers are comparable with the standard piston.

5.17 SUMMARY OF THE EXPERIENCES OF EXPERIMENTAL INVESTIGATIONS

From the extensive experimental program conducted it became evident that it is indeed technically feasible to run
compression ignited engines of different configurations in
dual fuel mode with ethanol as a supplementary fuel. The
analytical predictions very satisfactorily correlate with the
experimental findings. The salient findings of the present
investigations are presented in a nut shell in the next
chapter.