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**A Zero-Dimensional Model of  
Turbulent Combustion in a  
Homogeneous Charge  
Spark Ignition Engine**

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For Erika, Leonardo and Olavo



# Abstract

In the present work, a detailed study is made of the combustion process in a diesel engine. The results are presented in the form of a state diagram.

The state diagram is a graphical representation of the combustion process. It is a plot of the pressure and temperature of the gas mixture as a function of the crank angle. The diagram is divided into several regions, each representing a different stage of the combustion process.

The diagram is divided into several regions, each representing a different stage of the combustion process. The regions are labeled as follows: 1. Ignition, 2. Propagation, 3. Development, 4. Termination, and 5. Exhaust. The diagram is a plot of the pressure and temperature of the gas mixture as a function of the crank angle.

Two methods are used to determine the state diagram. The first method is based on the use of a pressure transducer and a temperature transducer. The second method is based on the use of a pressure transducer and a pressure transducer. The diagram is a plot of the pressure and temperature of the gas mixture as a function of the crank angle.

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Conclusions are drawn from the results of the study. The diagram is a plot of the pressure and temperature of the gas mixture as a function of the crank angle. The diagram is divided into several regions, each representing a different stage of the combustion process. The regions are labeled as follows: 1. Ignition, 2. Propagation, 3. Development, 4. Termination, and 5. Exhaust. The diagram is a plot of the pressure and temperature of the gas mixture as a function of the crank angle.

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In the present work a zero-dimensional model of turbulent combustion has been developed for the spark ignition engine. In addition to the prediction of thermodynamic state properties, the model includes the following elements: heat transfer from the combustion chamber, a geometrical model of turbulent spherical flame propagation and detailed post flame chemical equilibria.

The model is responsive to such engine operating parameters as spark timing, equivalence ratio, engine speed and also to such design parameters as compression ratio, spark plug position and combustion chamber geometry.

Two different geometries were studied extensively with the program, namely the disc chamber and the bowl-in-piston chamber. Ethanol and isooctane were investigated as fuels during the numerical simulations at variable conditions of spark advance, equivalence ratio, engine speed and compression ratio

The model was employed in a partly diagnostic role to predict the engine performance using different correlations for a critical model parameter, the flame speed ratio. These were taken from the work of different researchers and included both empirical correlations, derived from engine experiments, and correlations derived from more fundamentally-based turbulent combustion studies.

Comparison between prediction and experiment was made possible by a diagnostic investigation of the expanding flame front into the combustion chamber of a homogeneous charge spark ignition engine, performed in a Ricardo E6 research engine.

The turbulent flame speed was determined experimentally by a dual-beam laser deflection technique. The measurement of

flame transit times was performed using a specially designed and built system incorporating equipment for data acquisition based on a microcomputer.

The results obtained experimentally and those obtained by the numerical simulation in general show good agreement. In the absence of detailed in-cylinder turbulence information the more fundamentally-based turbulent flame propagation models often do not perform as satisfactorily as those developed specifically for zero-dimensional ( or thermodynamic ) engine simulation.

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# Nomenclature

|      |              |
|------|--------------|
| Ac   | Exhaust      |
| At   | Flame        |
| Cc   | Cylinder     |
| Al   | Inlet        |
| At   | Total area   |
| Aw   | Cylinder     |
| Cp   | Constant     |
| Cr   | Coefficient  |
| Cv   | Constant     |
| D, d | Diameter     |
| Dg   | Gasket       |
| Dt   | Time         |
| Dih  | Angle        |
| H    | Enthalpy     |
| h    | Height       |
| h    | Heat         |
| L    | Conductivity |
| L    | Integral     |
| N    | Engine       |
| Nu   | Nusselt      |
| p    | Pressure     |
| Pa   | Atmosphere   |
| Pl   | Pressure     |
| Po   | Stagnation   |
| Pr   | Prandtl      |
| R    | Creep        |
| R    | Unit         |
| Re   | Reynolds     |
| Rf   | Flame        |
| S    | Piston       |
| T    | Temperature  |
| Tad  | Adiabatic    |
| Tg   | Mean         |
| Tw   | Mean         |
| U    | Interfacial  |
| u    | Turbulent    |
| u    | Turbulent    |
| u    | Laminar      |

|      |                                 |                       |
|------|---------------------------------|-----------------------|
| Ae   | Exhaust valve area              | $m^2$                 |
| Af   | Flame front area                | $m^2$                 |
| Ah   | Cylinder head area              | $m^2$                 |
| Ai   | Inlet valve area                | $m^2$                 |
| At   | Total area                      | $m^2$                 |
| Aw   | Cylinder wall area              | $m^2$                 |
| Cp   | Constant pressure specific heat | $kJ\ kg^{-1}\ K^{-1}$ |
| Cr   | Coefficient of residual gases   | *                     |
| Cv   | Constant volume specific heat   | $kJ\ kg^{-1}\ K^{-1}$ |
| D, d | Diameter                        | m                     |
| Dg   | Gasket diameter                 | m                     |
| Dt   | Time increment                  | s                     |
| Dth  | Angle increment                 | deg                   |
| H    | Enthalpy                        | kJ                    |
| h    | Height                          | m                     |
| h    | Heat transfer coefficient       | $W\ m^{-2}\ K^{-1}$   |
| L    | Con-rod length                  | m                     |
| L    | Integral length scale           | m                     |
| N    | Engine speed                    | rpm                   |
| Nu   | Nusselt number                  | *                     |
| p    | Pressure                        | $N\ m^{-2}$           |
| Pa   | Atmospheric pressure            | $N\ m^{-2}$           |
| Pi   | Pressure at inlet manifold      | $N\ m^{-2}$           |
| Po   | Standard atmospheric pressure   | $N\ m^{-2}$           |
| Pr   | Prandtl number                  | *                     |
| R    | Crank throw                     | m                     |
| R    | Universal gas constant          | $kJ\ kg^{-1}\ K^{-1}$ |
| Re   | Reynolds number                 | *                     |
| Rf   | Flame radius                    | m                     |
| S    | Piston stroke                   | m                     |
| T    | Temperature                     | K                     |
| Tad  | Adiabatic flame temperature     | K                     |
| Tg   | Mean gas temperature            | K                     |
| Tw   | Mean wall temperature           | K                     |
| U    | Internal energy                 | kJ                    |
| u    | Turbulence intensity            | $m\ s^{-1}$           |
| u    | Turbulent burning velocity      | $m\ s^{-1}$           |
| u    | Laminar burning velocity        | $m\ s^{-1}$           |



|       |                        |             |
|-------|------------------------|-------------|
| v     | Volume                 | $m^3$       |
| $v_i$ | Velocity at inlet duct | $m\ s^{-1}$ |
| $v_p$ | Mean piston velocity   | $m\ s^{-1}$ |
| $v_r$ | Recess volume          | $m^3$       |
| $v_s$ | Swept volume           | $m^3$       |
| w     | Work                   | $kJ$        |

#### Subscripts

|   |                 |
|---|-----------------|
| a | Atmospheric     |
| b | Burned          |
| i | Inlet           |
| o | Reference state |
| u | Unburned        |

#### Greek symbols

|           |                                    |                     |
|-----------|------------------------------------|---------------------|
| $\gamma$  | Ratio of specific heats            |                     |
| $\delta$  | Ignition delay                     | s                   |
| $\eta$    | Kolmogorov scale                   | m                   |
| $\theta$  | Crank angle                        | deg                 |
| $\lambda$ | Thermal conductivity               | $W\ m^{-1}\ K^{-1}$ |
| $\lambda$ | Taylor microscale                  | m                   |
| $\mu$     | Absolute viscosity                 | $N\ s\ m^{-2}$      |
| $\nu$     | Kinematic viscosity                | $m^2\ s^{-1}$       |
| $\rho$    | Density                            | $kg\ m^{-3}$        |
| $\sigma$  | Molecular diameter (see Fig. 2.15) | $\text{\AA}$        |
| $\phi$    | Equivalence ratio                  |                     |
| $\omega$  | Angular velocity                   | rad                 |

## Abbreviations

|      |                            |
|------|----------------------------|
| ATDC | After Top Dead Center      |
| BTDC | Before Top Dead Center     |
| BIP  | Bowl-In-Piston             |
| CR   | Compression Ratio          |
| DCA  | Degree Crank Angle         |
| FGF  | Flame Growth Factor        |
| FPV  | Flame Propagation Velocity |
| FTT  | Flame Travel Time          |
| MFB  | Mass Fraction Burned       |
| SA   | Spark Advance              |
| SQR  | Squish Ratio               |
| TDC  | Top Dead Center            |
| WOT  | Wide Open Throttle         |



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## 1.1 Introduction

In this chapter an attempt is made to survey the literature on the cause most concern in the field of spark discharge.

The combustion process is the most important part of the operation, despite the fact that less than ten per cent of the effort has been spent on it in fifteen years in the most important process, engine consumption, power, etc.

For the sake of convenience, the extensive and complex subject is subdivided into various specific aspects of the problem.

The first topic is the modelling in spark discharge, the consideration of the discharge into two subsections, laminar and turbulent.

A review of previous work is presented next, followed by a discussion of the phenomena. The next section is about the problem of the engines.

A review of the experimental propagation of the discharge is then presented.

Thereafter, attention is

# Chapter 1

## Literature Survey

### 1.1 Introduction

In this chapter an attempt has been made to cover in detail the aspects of engine research that are believed to cause most concern to engineers, designers and researchers in the field of spark ignition engines.

The combustion process in spark ignition engines is by far the most important event during the entire cycle of operation, despite the fact that its duration accounts for less than ten percent of the whole cycle. Therefore great effort has been spent by researchers, during the past fifteen years in particular, aiming to improve this all important process, especially in relation to fuel consumption, power output and emissions.

For the sake of clarity in the presentation of so extensive and complex a subject, this chapter has been subdivided into various sections, each one dealing with specific aspects of engine combustion research.

The first topic is the important one of combustion modelling in spark ignition engines, followed by detailed consideration of flame propagation. This has been split into two subsections identifying the problems of both laminar and turbulent flame propagation.

A review of previous studies of engine heat transfer is presented next, followed by a brief discussion of knock phenomena. The next section addresses some considerations about the problem of cyclic variation in spark ignition engines.

A review of the experimental methods employed to study the propagation of the flame front in the combustion chamber is then presented.

Thereafter, attention is focused on the types of



combustion chamber most widely employed in the spark ignition engines and how the different geometries can affect the combustion rate.

The influence that the type of fuel has on the combustion process is examined next and emphasis is given to investigations of ethanol as an alternative fuel for spark ignition engines.

Finally, a discussion of the principal exhaust emissions unburned hydrocarbons, carbon monoxide and oxides of nitrogen, is presented.

## 1.2 Combustion modelling in spark ignition engines

In the past fifteen years the spark ignition internal combustion engine has been subjected to increasing restrictions regarding such characteristics as fuel consumption and pollutant emission levels. The more stringent regulations regarding the air quality level in the USA and more recently in the EEC, coupled with the intense competition between automotive manufacturers have led to the improvement of quality and performance of such powerplants.

Research into the internal combustion engine has consequently increased dramatically, especially in the field of combustion. From all the techniques used as tools in the development of the internal combustion engine, the mathematical model has proved to be very cost effective and reliable, witnessing dramatic improvements in the recent years.

This is a result of the continuous growth in both the recognition of the power of the mathematical model and also in the availability of increasing computer capability.

The use of mathematical models offers some significant advantages over the traditional method of design and extensive testing, not least being the development of a more complete understanding of the physical aspects that arise when formulating the model.

Another advantage is that with the model it is possible to identify key controlling variables which in turn provides guidelines for a less costly experimental test program.

Finally, the model has the ability to predict behaviour over a wide range of design and operating variables which can then be used to determine trends and trade-offs. Depending on the accuracy of the model it can also be used to optimise engine and control.

Today, models may be classified into two groups:

- .Phenomenological models and
- .Multidimensional models.

The most successful and useful models to date are in the group of the phenomenological models. These models, sometimes called zero-dimensional or thermodynamic models are structured around a thermodynamic analysis of the contents of the engine cylinder during the engine operating cycle.

They are constructed from the formalism of the first law of thermodynamics in which the only real independent variable is the time.

These models attempt to describe complex engine behaviour in terms of separate, physically based submodels of the identifiable phenomena. The submodels are based upon sound physical formulations representing our present understanding of the phenomena.

The modelling of flow into the engine during the induction process in zero-dimensional models is handled using



quasi-steady one-dimensional flow equations. The flow of gases through valves and other restrictions is modelled under conditions of isentropic adiabatic flow through a nozzle, using discharge coefficients to relate the effective area for a particular constriction to the ideal area for isentropic flow. The pressure ratio used in these computations is formed by the system pressures upstream and downstream of the passage.

Normally, plenum assumptions are made for intake and exhaust manifolds unless pipe dynamics are of interest. When dealing with pipe dynamics, unsteady gas-flow equations are to be used including friction, heat transfer and geometrical changes in pipe geometries.

Boundary conditions at pipe junctions, valves and open ends are determined by means of empirical constants. The working fluid is assumed to be a homogeneous mixture of fuel vapour and air with a certain amount of humidity and residual gases from the previous cycle at a frozen composition.

Due to the pulsating nature of the flow in both intake and exhaust manifolds, under certain conditions of throttled-engine operation, there may be a back-flow of unburned gases into the intake manifold during the period the valves overlap. It is assumed then that the burned gases do not mix with the fresh ones and that they are the first to reenter the cylinder during the intake process. This is an important condition to be fulfilled when calculating the amount of residual gases in the combustion chamber.

Specific heats of the individual components of the gaseous mixture are calculated through polynomial functions of the temperature.

Heat transfer is modelled using correlations between

Nusselt, Reynolds and Prandtl numbers developed for heat transfer in steady flows over flat plates and in pipes. They take the form

$$Nu = a Re^b Pr^c$$

where the constants  $a$ ,  $b$ ,  $c$  are obtained by curve fitting experimental data from a particular engine system. In evaluating Reynolds number, the cylinder diameter and mean piston velocity are used as the characteristic length and velocity, respectively.

The rate of mass burning may be specified by a functional relationship obtained by matching with previous experimental data. Burned gas properties are obtained by curve fitting thermodynamic-equilibrium values of burned gas properties or by approximate models of thermodynamic equilibrium. In both unburned and burned zones, individual gases are treated as ideal gases.

The validation of the phenomenological description can easily be made by comparisons between theory and experiments. The modeler has complete control over which phenomena are to be included and the level of detail to be achieved in the light of both our physical understanding and the available data.

The computational codes resulting from these formulations are relatively undemanding in terms of computer requirements and lend themselves nicely to extensive parametric optimisation or design evaluation.

However some deficiencies are found in phenomenological models, relating especially to the fact that they do not provide spatial resolution of variables such as fluid velocity and state variables such as temperature and chemical species. These are sometimes necessary for a model to be an effective tool, for example, in cases where

it is required to predict in detail the fluid flow within the engine cylinder, as in the study of the air-flow pattern at the time of fuel injection in diesel engines.

Furthermore, given that engine phenomena are very complex, the formulation of adequate submodels that represent them may be extremely difficult. Lack of physical insight and/or empirical data may then impose major limitations of this type.

In general the models are independent of each other but linked to the overall simulation. What then proves difficult to assess is the degree of interactiveness of the various phenomena via this approach.

Despite these problems the phenomenological models have been applied to internal combustion engines with considerable success.

The first useful models of the conventional spark ignition engine were concerned with the basic thermodynamics, focusing on the performance of the engine as a function of such operating conditions as fuel/air ratio, engine speed, spark advance and engine design variables. One of the earliest such models was that developed by Patterson and Van Willen (1962).

Further improvements to a similar model were introduced by Lavoie et al (1970), in which the basic thermodynamic relationships were supplemented by the Zeldovich chain reaction mechanism for nitric oxide formation. They used this model to verify, by comparison with engine measurements, that the extended Zeldovich mechanism was indeed valid in the engine environment.

Komiyama and Heywood (1973) also employed a similar model to predict nitric oxide concentration in the exhaust of a spark ignition engine over a wide range of fuel/air



ratios, proportions of exhaust gas recirculation and engine speed. Comparison of measured and calculated data showed good agreement over all operating conditions. Heat transfer was computed using the correlation developed by Eichelberg (1939).

Phillips and Orman (1966) developed a thermodynamic model based on the first law which took into account heat transfer between the working fluid and its environment, dissociation of the working fluid and finite rates of heat release. They utilised the thermal theory of Mallard and Le Chatelier (1883) for the laminar flame propagation, introducing modifications to include the effects of turbulence which permitted the calculation of the flame speed at any point in the engine cycle.

The study reported that the calculated burning velocity relative to the unburned mixture, increased rapidly towards the end of combustion but the variation of the rate of expansion of the burned mixture is such that the flame speed relative to the engine structure changes only slightly during combustion over a range of operating conditions. The model was able to simulate the effects of several parameters such as the ignition timing, initial pressure, initial temperature, engine speed, and fuel/air ratio on the performance of a single cylinder Ricardo E6 research engine.

In the phenomenological model the mass burn rate is in general specified a priori. This constitutes one of the major criticisms made of this model. Several workers have specified the burn rate through the expression

$$dM/dt = \rho_u A_f S_t$$

where  $dM/dt$  is the mass burn rate,  $\rho_u$  is the unburned gas density,  $A_f$  is the flame front area and  $S_t$  is the turbulent burning velocity. The specification of the

turbulent burning velocity is typically the point at which the major assumptions are made.

Until comparatively recently turbulent burning velocity has been related to the laminar burning velocity through the expression

$$S_t = S_l \cdot K$$

where the term  $K$  is assumed to be dependent on the engine speed and determined from a relationship of the form

$$K = 1.0 + C \cdot N$$

The value of the term  $C$  has been determined by several workers; Lucas and James (1973), working with a Renault IFP research engine found  $C$  to have a value of 0.00197, Phillips and Orman (1966) measuring flame propagation speed in a Ricardo E6 research engine gave to  $C$  the value 0.002 and Hodgetts (1966) suggested the value 0.0017 after experiments with a Ford six cylinder production engine.

Another important contribution that made use of this relationship was that of Benson et al (1975). The above equations are valid only if it is assumed that the turbulence intensity is both homogeneous and constant during compression of the unburned charge. This means that the turbulent burning velocity can then vary only through the laminar burning velocity for a given engine speed.

An alternative approach is to model the turbulent flame propagation process as that of the entrainment of discrete, coherent, turbulent eddies which are then burned at the laminar burning velocity. Blizard and Keck (1974) used a model based on this principle to calculate the rate of burning in a spark ignition engine. They considered that the entrainment front moves relative to the unburned



gas at a constant speed, proportional to the turbulence intensity. After entrainment, the eddies burn in a characteristic time  $\tau$  that is a function of its size and the laminar burning velocity, ie,

$$\tau = 1/u_l$$

Zelevnik and McBride (1977) presented a well formulated model for the Otto cycle including intake and exhaust portions of the cycle, instead of considering only the compression/expansion and combustion. The model also included heat release and a full kinetic treatment of the chemistry rather than using a truncated reaction mechanism, such as the Zeldovich mechanism.

Tabaczynski et al (1977) extended the model of Blizard and Keck (1974) postulating a detailed structure for individual eddies together with a technique for calculating the time required to burn them. Although this model is based on indirect experimental evidence, physical intuition and mathematical convenience it predicts the proper trends in ignition delay period and combustion duration.

The entrainment model was further extended by Hires et al (1978), who reformulated it introducing new parameters to take into account the geometry of a given engine. Although the model has shown excellent agreement with some experimental data, there are some questions to be resolved, for example the turbulence field for complex geometries using squish and swirl, the use of spherical flame front geometry, turbulent levels in the burned gases and heat transfer in a time and space dependent turbulent field.

Heywood et al (1979) developed a simulation model of the four stroke spark ignition engine, to study the effects of variations in engine design and operating parameters on engine performance, efficiency and NOx emissions. The model assumes that the mass fraction burned in the cylinder, at

each crank angle is specified by a Wiebe function of the type

$$x = 1.0 - \exp(-C)^m$$

$$C = a \{ (\theta - \theta_i) / \Delta\theta \}$$

where  $x$  is the mass fraction burned,  $a$  is the efficiency parameter,  $m$  a form factor,  $\theta$  the crank angle,  $\theta_i$  the crank angle at start of combustion and  $\Delta\theta$  is the burn angle duration.

Girgis and Tidmarsh (1979) presented a model based on that formulated by Blizard and Keck (1974) to study the effect of combustion chamber design on performance. The model was divided into two sections, one that considered the thermodynamics of the internal combustion engine, while the other solved a simplified version of the chemical species equations in order to predict engine performance. The burned mass fraction has been obtained from purely thermodynamic considerations.

Computer simulation models are becoming increasingly accepted in the field of fluid flow in internal combustion engines, where at the expense of considerable complexity, they offer the prospect of a more generally applicable method. Despite the fact that the subject is outside the scope of the present work, some important contributions have been made through this approach recently by Singh and Surakomol (1979), Sorenson and Pan (1979), Gosman and Johns (1978), Ramos and Sirignano (1980)a, Ramos and Sirignano (1980)b, Gosman and Johns (1980), Morel and Mansour (1982) and Davis and Borgnakke (1982).

It is appropriate to comment on engine knock modelling although the subject is covered in detail in section 1.4. Despite the fact that considerable effort has been devoted to the elaboration of mathematical models to study the

knock phenomenon in spark ignition engines, to date no model has proved capable of predicting in full all the variables involved in such an important phenomenon. The difficulties are related mainly to the formulation of a detailed chemical kinetic model.

The model formulated by Douad (1981) was a phenomenological model based on an overall kinetic reaction rate which describes the auto ignition delay of a fuel as a function of the pressure and temperature, which are supplied by a thermodynamic simulation of the engine cycle.

The model is capable of predicting the auto ignition delay that is calculated from the expression below:

$$\delta = A p^n \exp (B/T)$$

where A, n and B are parameters of pure fuels determined experimentally and p and T are the pressure and temperature respectively.

Schaepertons et al (1981) found that the best way to fully characterize the knock phenomenon by means of computer simulation, could only be achieved when dealing with a method that considered the spatial and temporal details of the combustion processes in the engine. This led to the formulation of a multidimensional model for that purpose. A two dimensional model formulation was used to study the effects of high compression ratio combustion chambers on the onset of knock. The model also showed that knocking characteristics are caused by the auto ignition of the gas ahead of the flame front.

Smith et al (1984) have described recently a model that solves the coupled conservation equations for mass, momentum, energy and chemical species to assess the influence of different operation parameters on hydrocarbon



ignition characteristics. The variable engine parameters introduced were the pressure, temperature and equivalence ratio. The results indicated that the parameter with greatest influence on the ignition delay time was the initial temperature.

To complete this section, some general comments about multidimensional models should be made. Such models, also called detailed models in the literature, utilize detailed numerical techniques such as finite difference and finite element approaches to obtain the solution of the full conservation equations of mass, chemical species, momentum and energy.

They have the potential to describe more exactly and in more detail the fluid dynamics and chemical processes that occur at any location within the engine cylinder and cycle. Due to limitations of computer capabilities at present, these models have been largely used as two-dimensional formulations and as such have had only limited applications to the analysis, design and optimization of engines. Their validation also relies on detailed experimental measurement which has only recently started to appear.

### 1.3 Heat transfer

Heat transfer in internal combustion engines has been always a subject of great interest by engine researchers and in an attempt to understand the processes of heat transfer to the cooling system and thermal loading of engine components, the subject has stimulated a great deal of research.

The study of heat transfer in engines presents serious difficulties since the processes inside the combustion chamber of an internal combustion engine are extremely

complex, involving rapidly changing gas pressure, temperature and perhaps also local fluid velocities.

Cylinder heat transfer varies between individual engines due to several factors such as the degree of quench, turbulence and specific chamber geometry. Here the shape of the gas and coolant flow systems is so complex that theory cannot predict in detail the flow patterns or temperature patterns involved. At present time, an adequate theoretical analysis of the heat transfer has not been achieved although a number of studies have been made with the aim of improving this situation.

Eichelberg (1939), presented a method to evaluate the heat transfer through the walls of an engine cylinder, making use of the following equation:

$$Q = C A V^{0.3} (p T)^{0.5} (T_g - T_w)$$

where C is a constant, A is the instantaneous area involved in the heat transfer process, V is the mean piston velocity, p is the pressure in the cylinder, T the temperature in the cylinder,  $T_g$  the mean gas temperature and  $T_w$  the chamber wall temperature.

The equation is very simple and explicit and accordingly very easy to apply. It is not however dimensionally homogeneous and its use requires some appreciation of the conditions for which its reliability is established.

Pflaum (1962) in an extension of the work of Eichelberg, proposed overall heat transfer coefficients for piston crowns and cylinder liners. Later on, in the course of a critical review of previous work on heat transfer, Annand (1963) pointed out the inconsistency of the previous methods, including the lack of dimensional consistency and proposed a more satisfactory alternative.



He suggested calculating the heat transfer by making use of the following equation:

$$Q = A a \lambda / D (Re)^{0.8} (T_g - T_w) + (T_g^4 - T_w^4)$$

where A is the instantaneous area involved in the heat transfer,  $\lambda$  is the thermal conductivity coefficient, D the cylinder diameter,  $T_g$  and  $T_w$  the mean temperatures of the gas and wall respectively, and a and c are numerical constants. The Reynolds number Re is determined from the expression:

$$Re = \rho V_p D / \mu$$

where  $\mu$  and  $\rho$  are the absolute viscosity and density of the gases respectively, both functions of the temperature of the gas.  $V_p$  is the mean piston speed.

Krieger and Borman (1966) used the Eichelberg formula to calculate approximate heat release for a compression ignition engine. The methods proposed do not require a detailed knowledge of the combustion phenomena but are particularly sensitive to the pressure data, thus requiring excellent quality pressure-time diagrams. Despite the fact that the method is delimited by the assumption of thermodynamic equilibrium, they concluded that it proved a reliable tool for predicting performance of the internal combustion engine.

Using laws of similarity governing convective heat transfer, Woschni (1967) derived an equation containing two convective terms one of which takes into account the piston velocity and the other the convection due to combustion. The heat transfer coefficient is calculated by the expression :

$$h = 110.0 \quad d \quad p \quad T \quad K$$

$$K = \{ C_1 V_p + C_2 V_s T_1 (p - p_0) / p_1 V_1 \}$$

where  $d$  represents the cylinder diameter,  $p$  the pressure and  $T$  the temperature in the cylinder at a given instant of the cycle,  $C_1$  and  $C_2$  are constants,  $V_p$  is the mean piston velocity,  $p_1$ ,  $V_1$ ,  $T_1$  are the pressure, volume and temperature respectively at some reference state, for example the inlet valve closing, and  $V_s$  is the instantaneous cylinder volume. The total heat transfer can then be evaluated by inserting the heat transfer coefficient in the equation:

$$Q = A h (T_g - T_w)$$

The Annand and Woschni expressions are both largely based on turbulent convection considerations and both have constants which may be adjusted in order that reasonable overall heat transfer coefficients are obtained.

In a later work, Annand (1980) carried out some experimental observations of the local heat flux for a range of compression ratios, speeds and inlet conditions in one cylinder of a six cylinder compression ignition engine in the absence of combustion.

He pointed out the great difficulty in obtaining a simple representation of the heat transfer in reciprocating engine cylinders, even in the absence of combustion. He also reported the strong influence of boundary layer behaviour when pressure and temperature vary together in a gas from which heat is being transferred to a solid surface. An important observation was that when combustion occurred, convective heat flux did not lead the driving temperature difference on expansion, but lagged behind. He concluded that this probably reflected the intensification of turbulence by charge movement induced by gas expansion during combustion.

Benson and Whitehouse(1983) report that experiments conducted by Summers (1970) in a single cylinder engine, confirmed that the local velocity changes can be caused by the expansion of the burned gases.

Borgnakke et al (1980) also developed a model to evaluate the instantaneous heat transfer in a spark ignition engine. The model is based on the growth of the thermal boundary layer after the flame passage. It is therefore heavily dependent on the description of the turbulent flow field in the engine cylinder. The turbulence levels are characterized in this simulation model by a  $K-\epsilon$  model which include the effects of compressibility and anisotropic behaviour of the flow field. The flame propagation process is modelled by an entrainment model.

The model seems to present good sensitivity to the changes in geometry and operating conditions of the engine. The turbulence model appears to predict the general behaviour of the turbulent field but the lack of experimental data does not allow a satisfactory evaluation of the model.

#### 1.4 Knocking

Since the early days of engine development, knock has been an intriguing phenomenon for engine design men and over the last fifty years much effort has been devoted to the task of qualifying and quantifying it.

Knock is one of the most serious limitations on the performance of the present day internal combustion engine and has been plaguing the designers and users of spark ignition engines since 1880 when Clerk (1921) suppressed the extremely violent knock by the use of water as reported by the author in a paper published in that year.

Later on, during the war years, important research has

been done by a team of researchers at the NACA in the USA, as mentioned in a paper by Miller (1947), the great majority of the workers favoring the theory of auto ignition to explain knock.

Downs and Wheeler (1952) published a paper whereby they suggested that the knock phenomenon could be explained by two theories still valid today : the detonation theory and the auto ignition theory.

The detonation theory considers knock to be a phenomenon in which shock waves originated at some point in the combustion chamber during the propagation of the normal spark ignited flame front, induced a reaction on the end gas. The detonation wave is thus a shock wave which is sustained by the energy of the chemical reaction in the highly compressed medium.

Under this theory some relevant facts about detonation can be identified :

- .the flame produced under detonation is very luminous.
- .the velocity of detonation is known to be several times the velocity of the sound in the unburned part of the charge.
- .it has been found to be extremely difficult to set up detonation waves in containers as small as the combustion chamber of spark ignition engines, what has caused workers to study the phenomenon in tubes. When the detonation wave occur, the gas already burned moves in the same direction as the flame, in the case of a tube, as opposed to the normal flame propagation where the flame and burned gases move in reverse directions.
- .the relative velocity between the detonation wave and the burned gas behind the flame is equal to the velocity of sound in that state.

Miller (1947) was convinced that the detonation wave originated behind the normal propagation flame front and



swept out ahead of it to ignite the sensitized end gas.

Curry (1963) however showed evidence for an acceleration of the primary flame front of the order of ten to twenty times its normal velocity but still below the sonic velocity.

The detonation theory did not gain many proponents among the researchers and at present the only accepted theory for knock in internal combustion engines is the auto ignition theory.

This theory considers knock as the spontaneous ignition of the end gas. The flame front, originating at the spark plug and then propagating at normal velocity through the unburned gas, compresses the end gas together with the piston motion, raising its pressure and temperature above the normal values in such a way that "pockets" of self ignited gas appear in the body of the unburned gas ahead of the flame causing it to react suddenly.

The mass of gas that burns in this way expands quickly, causing an abrupt increase in the pressure inside the combustion chamber. There are several factors that influence auto ignition such as temperature, pressure, and air/fuel ratio. The chemistry of auto ignition is not yet fully understood.

The most widely accepted theory is that due to Lewis and Von Elbe (1951) that considers the auto ignition as a chain reaction process. A chain reaction is established when chemical combinations involving active particles generate other active particles together with the products of combustion.

The minimum energy possessed by a molecule capable of producing more molecules by collision is called activation energy. The velocity of the reaction can be influenced by

increasing the concentration of the molecules, ie by increasing the pressure. In this way an increase in the frequency of the collisions that led to the formation of new active molecules is achieved. Augmenting the temperature also improves the reaction rate causing an increase in the level of energy of the individual molecules.

Caris et al (1956) in a study of the influences of the combustion chamber geometry on the combustion rate, found that compact chambers presented higher combustion rates thus requiring low octane fuels. The reason is that the combustion takes place in a much shorter time interval and the end gases are thus less exposed to higher pressures and temperatures. In this way the physical conditions for occurrence of knock are much reduced or even eliminated. Air/fuel ratio was also pointed out as a factor that affects octane requirements because any variation in air/fuel ratio implies variations of the flame propagation velocity, flame wall temperatures and that causes the reaction time of the end gas to be a minimum near the stoichiometric ratio, thus maximizing the tendency to knock.

Hoffman (1961) has presented a method of determining the reactive knock resistance of spark ignition engine fuels. The method which he proposed consisted of filtering the rate of pressure change signal in order to isolate the gas vibration frequencies associated with knock, thereby obtaining a fixed threshold value of the filtered output for a given engine and reference fuel. A knocking cycle was then defined as the cycle for which the filtered voltage exceeded the threshold voltage. The octane rating of the fuel under test could then be determined by comparing the data from the tests with the data from reference fuels with known octane rating. The method was successful in rating fuels but unsuitable for research when only one fuel is being used.

A more suitable method for knock detection was that due to Barton et al (1979) They measured the knock intensity by analysing the frequency spectra of the root mean square (RMS) values of the pressure and rate of pressure change. The method proved to be successful, permitting objective comparisons of knock for different fuels and engines. It also allowed comparisons to be made relative to engine performance parameters such as maximum power.

Further work was carried out by Johnson et al (1965) using an infrared radiation method to measure end gas temperatures up to the time of knocking using as variables the intake pressure, the intake temperature and engine speed. They also utilised the chain reaction theory in their work and came to two interesting conclusions from their experiments. The first was that the extent of the reaction was quite large for the end gas, where computations from measured pressures and temperatures showed that ten percent of the energy released by the fuel occurred before the passage of the flame front. The second conclusion was that the rates of reaction became appreciable in magnitude at lower temperatures for lower octane number fuels, and increased more rapidly with temperature for the higher octane number fuels.

Haskel and Bame (1965), analysing records of simultaneous light emission, pressure and ionisation signals, considered knock in the gasoline engine to result from a "volume explosion" of the unburned gases ahead of the flame front. Prior to the knock pulse itself the pressures were essentially the same as those on knock-free cycles. They argued that the pressure pulse originated in the previously unburned gases, propagates back through the burned gases at essentially sonic velocity compressing the already burned gases, what can explain the high luminosity waves evident in the photographs.

Duke et al (1970) examined the effect of knock intensity on exhaust emissions in a single cylinder spark ignition engine using the method described by Barton and co-workers. The exhaust components that they were interested were CO, CO<sub>2</sub>, NO<sub>x</sub> and HC.

Another interesting method to determine the knock resistance of a fuel was that proposed by Douad and Eyzat (1978) and called by them the "Four-octane-number method". The aim of the method is to determine the knock resistance of the fuel based on the auto ignition delay concept which is obtained through simple processing of the standard "Research" and "Motor" indices, combined with two other specially defined indices. The four octane method can forecast, without experiment the behaviour of a fuel. Evidence for this was obtained by predicting with an average accuracy of better than 0.5 octane point the behaviour of several samples of fuels-vehicles-rotation speeds for which the experimental results were already known.

Other methods can be used to detect knock in addition to the cylinder pressure method. The air-borne noise method is still in use today and consists basically of a microphone, amplifier with filter and ear-muffs. They have a wide field of application with a realistic noise reproduction. However this method implies a subjective knock recognition, but such recognition is only that above a knock threshold and there is no cylinder identification.

As the rate of pressure rise is elevated during the knock process, causing the individual components of an engine to vibrate at their natural frequency, a method that utilises this vibration can be devised. The structure-borne noise method uses an accelerometer mounted on the engine block instead of a microphone or a pressure transducer. The method has a short set up time, entire engine speed range and cylinder allocation with the addition of the ignition



signals but necessitates empirical location of the sensor points.

photographic techniques can also be used with advantage when studying the knock phenomena in spark ignition engines, provided an optically accessed engine is available.

A great step towards the wider use of high speed photography in combustion research was given by Withrow and Rassweiler (1936). They used an engine with quartz windows in the cylinder head through which high speed motion pictures could be taken of the entire space of the combustion chamber, at a rate of five thousand frames per second.

These pictures confirmed most of the previous understanding of the engine combustion process, such as distinct flame fronts, high luminosity behind the flame front, ie in the vicinity of the spark plug and also the formation of different points of autoignition ahead of the flame front when operating the engine under knock conditions. The technique of studying combustion through high-speed motion pictures has been improved since then, with engines offering better access to the combustion chamber but unfortunately the method has been restricted to the optically accessed research engines only.

Recently, Lightfoot and Negus (1984) using a high speed photography technique, confirmed that the burned gas near the spark plug became brighter after the flame front had reached the far side of the combustion chamber, this intensity being maximum when the pressure reached its maximum value.

They suggested that the re-illumination was due to the delayed combustion of the carbon monoxide (CO) and/or hydrogen (H<sub>2</sub>). In addition, the fact that the burned gas

expands locally from the flame front causes the previously burned gas to be compressed. This leads in turn to a gradient of temperatures with the hottest point at the spark plug. The results of the photographic technique were confirmed by studying the light emission from the combustion chamber using a photodetector.

Lee and Lightfoot (1985), working in an optically accessed single cylinder spark ignition engine and also using a high speed photography technique, undertook several experiments aimed at establishing the correlation between knock and flame propagation. Using probability histograms of volume burnt history, they concluded that knocking cycles had overall slower flame propagation, in opposition to other investigators that defend the point of view that during knock the burning velocity increases.

### 1.5 Cyclic variation

In recent times the pressures towards increased fuel economy, lower emissions and higher efficiency of the spark ignition engine has led to the use of higher compression ratios and lean burn processes. This approach has proved to be very effective but leads to an increase in the cyclic variation of the combustion process.

It is well known that cyclic variations in engine operation are fundamentally caused by variation in combustion from one cycle to the next which manifests itself as variations in the pressure development and engine output.

The fact whether the combustion process is of the fast burn type or slow burn type plays an important role in the severity of the cyclic variations. If the engine follows a fast burn process, a large fraction of the energy release occurs near the top dead center when the combustion

chamber volume is changing very slowly and in this case the pressure variation is due mainly to the combustion variations.

However, when the engine operates according to a slow burn process, a large portion of the charge is burned later in the cycle where the movement of the piston is greater, then the variation in pressure due to combustion variations is augmented due to rapidly changing in cylinder volume.

Fuel economy and emissions are also adversely affected by these combustion variations together with an increase in the roughness of engine operation. Therefore, if cyclic pressure variations are to be minimized in an engine the problem of combustion variations should also be minimized and combustion rates should be maximized.

It has been found that variations from cycle to cycle are originated during the initial burning period and continue into the fully turbulent period.

Starkman et al (1959), making use of the ionization probes technique, monitored the flame arrival times at various locations of the combustion chamber over many cycles and noticed that most of the variations occurred near the spark plug.

The fact that the combustion is dependent on the early burn period was also reconized by Soltau (1961). He studied the effects of the early stages of the flame development on the overall combustion using high speed movies and noted that the fully developed flame travel phase was the most repeatable combustion stage, therefore suggesting that if it were possible to reduce the variations of the early combustion period, then the variations in the later stage would also be reduced.

These conclusions were confirmed in a later study by Peters and Borman (1970) and by Arrigoni et al (1973) who observed that the effects of combustion variation on pressure development in cycles having lower burning rates during the early stages of flame development, continued to have lower rates even after combustion was well underway, thus resulting in lower peak pressures.

However, there is a very early stage of flame development in which the flame growth is very repeatable, perhaps corresponding to a laminar flame growth.

Curry (1963) like Starkman et al (1959) used ionization probes technique to measure the flame travel times at various positions in the combustion chamber. He found that the arrival times at the ionization probe closest to the ignition source were the most repeatable, while large variations occurred thereafter. This fact was later confirmed by Ohigashi and Hamamoto (1970) working with modified spark plugs to act as ionization probes.

Hirao and Kim (1970) also found that the very early kernel development remained essentially constant for all combinations of ignition location and timing investigated, even though variations in flame arrival times at other distant ionization probes varied.

The effects of engine operating and design variables on cyclic variation were considered by many investigators. They can be treated under two different points of view : those variables depending upon chemical factors and those depending upon physical factors.

By chemical factors it is understood to refer to variables that determine the composition of the trapped charge in the cylinder and they are :  
 .equivalence ratio



.dilution of the charge due to exhaust gas recirculation,  
and also due to the residual gas left in the cylinder from  
previous cycle

.fuel type

All these factors were thought to influence cyclic  
variations by affecting the laminar burning velocity.

However, Mattavi et al (1979) suggested that at early  
stages of the combustion process the expansion velocity  
was more important than the laminar burning velocity. This  
expansion velocity is strongly affected by energy density,  
ie chemical composition of the charge trapped inside the  
combustion chamber.

From the three factors mentioned above, the equivalence  
ratio seems to be the most important factor influencing  
cyclic variations of combustion and the subsequent  
pressure development in the engine.

Generally, minimum cyclic variations occur at a slightly  
rich equivalence ratio corresponding to the mixture  
strength for maximum power. This has been confirmed by  
Phillips and Orman (1966) using ionization probes  
technique in the combustion chamber of a Ricardo E6  
engine. They observed minimum variations in flame arrival  
times at 1.15 to 1.25 equivalence ratio. The variations in  
arrival times were characterized by the time interval  
between the arrival of ten percent and ninety percent of  
the cycles monitored.

Karim (1967) also observed minimum percent variations in  
peak cycle pressures for equivalence ratios in the range  
1.25 to 1.37.

The second parameter, charge dilution, is an important  
factor especially for engines using exhaust gas  
recirculation (EGR) systems where increases in the  
dilution were generally found to result in increased cyclic

variations in pressure development.

Increased residuals resulted in higher cyclic variability as pointed out by Karim (1967). The increase in residuals were obtained by closing the throttle, which caused the coefficient of variation to change from eight percent at wide open throttle (WOT) to twenty five percent at one eighth throttle.

Recently, Kuroda et al (1978) showed that progressive addition of exhaust gas recirculation (EGR) resulted in four identifiable combustion patterns namely : normal burn, slow burn, partial burn and misfire, corresponding to increasing levels of variations in engine output. An acceptable stability limit was found to correspond to levels of EGR at which five percent of the cycles were slow burners.

Finally, the characteristics of the fuel were found to be an important factor affecting cyclic variations in engines. The equivalence ratio at which the maximum burning velocity occurs and the magnitude of this maximum are two fuel-related characteristics that may affect cyclic variations.

Fuels having greater burning velocity can release a larger part of their energies near top dead center and because of this better combustion-volume phasing, pressure variations are smaller compared to slower burning fuels with substantial energy release later in the cycle.

Contrasting with the chemical factors above described there are the physical factors now considered. By physical factors it is understood those which alter the physical conditions under which combustion proceeds, consequently affecting cyclic combustion variations.

The first to consider is the ignition system. It was found that some ignition systems do not affect cyclic variations but others do. Soltau (1961) and Karim (1967) found that ignition characteristics such as spark energy, spark duration, spark gap and electrode shape had little effect on cyclic variations of pressure development. The ignition characteristics that had noticeable effects on cyclic variation were spark gap location, number of ignition points and spark timing.

Kuroda et al (1978), experimenting with a dual spark plug system were able to obtain shorter overall combustion times with consequent reduction in cyclic variation which allowed the engine to operate at higher levels of EGR, necessary for controlling the level of NOx emissions.

The effects of combustion chamber geometry on cyclic variations of combustion and engine output were considered in a study by Young (1980). He found that the chamber shape in conjunction with the spark plug location were responsible for alterations in the length of the main combustion period and in its variation from cycle to cycle. For a given position of the spark plug the chamber which allowed the greatest amount of cylinder volume to be engulfed for an assumed spherical flame propagation produced the best results from the point of view of cyclic variations.

Another cause for cyclic variations is the engine speed and it is well known that cyclic variations are proportional to engine speed. Harrow and Orman (1965) utilising ionization probes to detect flame arrivals, noted that for a given mixture strength and spark timing, if the engine speed were increased, thus producing shorter flame travel times, this also produced an increase in the cyclic variations.

The effect of compression ratio on cyclic variations appears to be weak. It was found that variations in

average flame velocity increase with increasing compression ratio. Increasing the compression ratio reduces the amount of residual gases trapped in the combustion chamber and the smaller the charge dilution the greater the combustion rate and the lesser the cyclic variation.

Another factor is that the higher the compression ratio the higher the temperature of the charge and hence the higher the combustion rate, implying also the reduction of the cyclic variation, other factors being equal.

Another physical factor that influences the cyclic variations is the mixture preparation. Based on the findings of the effect of equivalence ratio on cyclic combustion and pressure development variations, it can be expected that a more homogeneous mixture would cause smaller cyclic variations. However, several investigators attempting to determine the influences of mixture formation on cyclic variation, using various types of mixers, gaseous fuels, etc, found that the results of these experiments do not provide sufficient evidence to support or refute the conclusion that a more homogeneous mixture will produce a lower cyclic variability. The main deficiencies of these experiments were the confounding effects of fuels, geometric factors, and the lack of qualitative and quantitative data to characterize the mixture homogeneity. Among the investigators that defend the concept above described are Patterson (1966) and Soltau (1961).

However, more recent studies like those of Matthes and McGill (1976) and Peters and Quader (1978), have shown that non-homogeneous mixtures can result in lower cyclic pressure variations, which is in opposition to the findings of the previous mentioned investigators.

To finalize it must be mentioned that the flow of gases



inside the cylinder has been shown to be the most influential factor affecting cyclic variations in combustion and pressure development. Many investigators have addressed studies to this subject, focusing most on the possibilities of extending the lean combustion limit aiming to improve fuel economy. In a recent paper, Petrovic (1982) presented results from a theoretical and experimental studies of cyclic variations of flame propagation periods in a spark ignition engine. He showed that cyclic variations were most influenced by mixture turbulence and homogeneity characteristics. )

The main conclusion was that a high turbulence reduces variations of the developed flame propagation period. For lean mixture running it is preferable to have homogeneous mixture of moderate turbulence during the phase of flame initiation and intense turbulence during the developed flame.

#### 1.6 Flame propagation monitoring

The expression flame propagation is understood as the process by which a flame front propagates through the mixture in the combustion space from the ignition source. The rapidity with which this flame front is displaced is called the flame propagation velocity.

This flame propagation velocity, measured in cylinder wall coordinates, is the summation of two velocity components: the turbulent burning velocity which is the actual rate of flame propagation through a stationary reactant mixture and is therefore measured relative to the unburned mixture and the expansion velocity which is the flame velocity component resulting from the expansion of the hot gas behind the flame front as a result of differences in density.

The first attempt to investigate the propagation of a flame front in a spark ignition engine was carried out by Glyde (1930) who mounted six circular small windows forming a line in the cylinder head of a spark ignition engine. Despite the fact that the visibility was poor, he was able to perform several experiments directed towards the knock phenomenon.

Withrow and Boyd (1931) replaced the circular windows by installing a slit window and using a revolving-drum camera with the rotational axis parallel to the window in order to obtain a record of flame-travel distance versus time. Again, they were mainly concerned with the knocking problem.

Marvin and Best (1931) made some advances by installing thirty one circular windows over the area of the cylinder head obtaining a good view of the combustion chamber with sufficient spatial resolution to plot flame position as a function of the crank angle.

Finally, Withrow and Rassveiler (1936) succeeded in covering the entire cylinder head of a side valve engine with a quartz window in order to obtain an unobstructed view of the combustion space. Since then the method of access to the combustion chamber through a transparent window in order to be able to photograph the combustion event is common place.

However, it should be noted that this method is not suitable for use with production engines, but is restricted to research engines specially built for that purpose.

The use of optically accessed engines is in itself no guarantee that a non-intrusive technique is being used, at least nowadays. Engines, such as those used during the introduction of the optical techniques, were of the "L"

head type. In those engines the valves were placed laterally in the combustion chamber, thus allowing space for the mounting of a transparent window in the cylinder head giving optical access to the combustion space.

But with modern engines, all with overhead valves, this method of access is not possible. This leads to different methods which are arguably not truly non-intrusive techniques. The literature contains many such examples and some will be considered briefly here.

It is known that optical engines are sometimes fired only in a few cycles each time. The main reason for this is probably that due to the difficulty in keeping the quartz window free from the soot deposits which make them totally unusable for the purpose they were built.

The skip-fired method of operation tends to scavenge completely the cylinder of exhaust residuals, which cause a totally atypical mixture to be used on the next firing cycles.

The side valve engine of the type described previously is totally obsolete nowadays and therefore there is no reason to expect that the different geometry gives results which approach those that would be obtainable if a modern geometry was to be used.

To overcome this difficulty, Bowditch (1961) used an engine in which an elongated piston with a quartz window on the top and a slot on the side reciprocates about an inclined mirror fixed to the block and protruding inside the piston slot. The image received through the window could then be reflected horizontally to a camera. The system provides access to, but not through, the combustion chamber.

A better arrangement was used by General Motors, building

an engine with a piston crown quartz window at the cylinder head, thus giving view through the combustion chamber which makes it very suitable for schlieren photography of the combustion process. The drawback is that due to the peculiar arrangement of the quartz window in the cylinder head, only one poppet valve is used for both intake and exhaust processes. The streams are separated outside the engine by a rotary valve. Because there is always a certain amount of exhaust gas after the exhaust stroke trapped between both valves, the engine can be used only in the skip-fired method. Furthermore it gives an abnormally high level of hydrocarbon emissions due to the abnormally high crevice above the piston ring, necessary to accommodate the quartz window.

Sandia Laboratories, in the USA have constructed an engine that provides a good view of the combustion space, substituting the whole cylinder head by a quartz window. To do so, the valves were mounted in a position ninety degrees relative to the cylinder axis. Optical access can also be obtained via the cylinder wall. The main drawbacks to this construction are: firstly that in order to provide sufficient space for the valves to be opened radially, the clearance height is large which results in low compression ratios and secondly, the flow pattern obtained during intake with such a lay-out is quite different from that obtained when the valve-in-head configuration is utilized. Lightfoot and Negus (1984) utilised such an engine for their investigation on knock, as mentioned previously in section 1.5.

At MIT, the engine utilized by combustion research provided optical access to the cylinder from the side. A square piston was employed to avoid optical distortion and reciprocated between flat walls. The flow pattern in such an engine tends to be totally atypical. Namazian et al (1980), utilised such an engine in a schlieren visualization of the flow in the cylinder of a spark

ignition engine.

It is worth to mention that all optically accessed engines operate with relatively low compression ratios, ranging from 4.5 to 7.0, which is also atypical for modern spark ignition engines operating in the range of 9.0 - 12.0 : 1 compression ratios.

Despite such concerns, the utilization of optically accessed engines for combustion research has increased dramatically in the last years and is always associated with the ever increasing utilization of laser-based techniques.

Optical fibres are beginning to be used nowadays to allow three-dimensional measurements of in-cylinder flow velocities and flame propagation. Recently, Spicher and Velji (1984), described the development of a new optical fibre technique for the experimental determination of the spatial flame propagation, monitored through a large number of optical fibres. In such investigation the optical fibres were distributed in ten planes parallel to the plane of the cylinder head, each of them containing twenty-four optical fibres radially disposed, providing two hundred and forty measuring points. The data acquisition system was provided by on-line computer. They concluded that the technique can be successfully applied to detecting spatial flame propagation in engines operating under real conditions.

Recently, a technique based on the schlieren principle that a light beam can be refracted in the vicinity of the strong refractive index gradients that occur near a flame front, has been used to measure the flame propagation velocity in the combustion chamber of a spark ignition engine.

The method consists of measuring the time delay between



the deflection signals from the laser beams running parallel, one to the other, at a known distance apart. The equipment employed in the measurement of the flame propagation velocity by this technique consist of a low power 1mw HeNe laser tube, a beam splitter that provides two parallel beams with variable spacing, auxiliary optics and position sensitive photodetectors. This subject will be described with more detail later in this work.

Swords et al (1982) describe the utilization of a similar equipment used to perform experiments in a Ricardo E6 research engine with the object of determining the flame propagation velocity and cyclic variation during the early stages of flame development. They found that for lean mixture operation, cyclic variation depends strongly on the mean flame propagation velocity, while for rich mixture operation the cyclic variation tends to be smaller for a given mean flame speed.

Alessandretti and Violino (1983) utilized the same technique to measure transit times of the flame front at specific locations inside the combustion chamber of a spark ignition engine. They constructed a specially designed combustion chamber of cylindrical shape over the normal combustion chamber, similar to a pre-combustion chamber with 35mm in diameter and it held two quartz windows. The compression ratio was 9 : 1 and the fuel selected was gasoline. The average time delay and cycle fluctuations have both been investigated as a function of engine speed and load.

The effects of changing the compression ratio, and different fuels on early flame development in a spark ignition engine has been investigated by Kalghatgi (1985). The results showed that decreasing the compression ratio reduces flame velocity sharply, mainly due to the increase of the residual mass fraction. Cyclic variation was found to be considerable and varying inversely with the average

burning velocity, which is in agreement with previous studies.

### 1.7 Combustion chambers for spark ignition engines

The performance of an internal combustion engine will be mainly dependent on the combustion process, therefore the design of the combustion chamber, the core of any engine, is of great importance.

Variables such as fuel/air ratio, ignition timing, exhaust gas aftertreatment, valve timing, to name the most significant, also play a very important role in engine performance.

It is clear that with the increasing effort to make engines more efficient in terms of fuel consumption and reduce the exhaust emissions, particularly nitric oxides, systems for mixture control, ignition timing and exhaust gas aftertreatment have reached a high degree of development. It is naturally becoming extremely difficult to make significant progress in these areas of engine control. Furthermore, the tendency towards the use of unleaded gasoline, to minimize ambient pollution by internal combustion engines, has led to the concentration of efforts towards the research and development of more efficient combustion chamber geometries.

The combustion chamber geometry has been the object of research since the early days of engine development. Ricardo (1929) developed the so called "Turbulent Head", aiming to create additional turbulence during the compression stroke in order to increase the combustion rate, reduce the surface to volume ratio of the combustion chamber with a consequent reduction in the losses during combustion.

Ricardo continued his experiments until 1929 when more than sixty different types of combustion chamber had been examined. The investigations were concerned particularly with the determination of the effects that parameters such as combustion chamber geometry, compression ratio, turbulence, spark plug location and quench distance had on normal and knocking combustion. In that time, these and related studies led to an increase in compression ratio from 4.8 to 7.0 and an increase in output power of the order of fifteen per cent.

A remarkable and exhaustive research programme was carried out during the fifties by Heron and Felt (1950), Caris and Mitchel (1956) and Caris and Nelson (1959). They studied the requisites for the reduction of the octane number by an appropriate design of the combustion chamber. The combustion chambers considered were those of the bowl-in-piston type, derived from diesel engines with direct injection. In those days the bowl-in-piston chambers proved to be relatively inefficient, returning a low specific power output and presenting thermal problems which prevented the general breakthrough of this type of geometry.

From the seventies onwards, the main trends in combustion chamber design and development have been towards a more efficient compromise between sometimes conflicting factors such as specific power output, low production costs, low fuel consumption and reduced exhaust emissions. The reduction in exhaust emissions in particular, brings some conflicting factors into the scene.

In order to have effective operation of the catalytic converter, the engine combustion chamber must operate at high temperatures. The need to build engines with a very low fuel consumption has led to the introduction of the lean burn process, which produces low exhaust gas temperatures, therefore making the exhaust gas

aftertreatment very inefficient. This causes the combustion process to be deliberately deteriorated in order to create conditions for the catalytic converter to operate properly. Combustion chambers with low compression ratios, without quench areas, relatively large diameters and favourable surface to volume ratio (SVR) were introduced.

An interesting study was performed by Scheffler (1966) who determined the effects of combustion chamber surface to volume ratio characteristics on the exhaust hydrocarbon emissions. He noticed that HC emissions increase in proportion to the surface to volume ratio and that for a given engine configuration, compression ratio is the most important factor influencing the surface to volume ratio.

The most promising solution to overcome this problem is the fast burning combustion chamber concept. One of its greater advantages is an improved tradeoff between efficiency and NOx emissions combined with a great tolerance to dilution of the charge.

Mattavi (1980) pointed out some of the advantages of a fast burning combustion process, among them :

- .fast burn chamber requires a lower octane fuel when compared to low burn chamber operating at MBT
- .cyclic variations are reduced when operating with fast burn chambers.
- .fast burn chambers presented a greater resistance to knock with consequent improvement in fuel economy through the use of higher compression ratios.

Overington and Thring (1981) carried out investigations aiming at the determination of the effects of operating gasoline engines with different combustion chambers. The tests were performed in a single cylinder research engine Ricardo Hydra, with high compression ratio compact combustion chambers.

Those chambers, compared with conventional ones, allowed an increase in octane numbers for a given fuel from 1.0 to 2.5 points. The fuel economy figures were ten percent better when these chambers were operated with lean mixtures. The bowl-in-piston compact chamber presented the best results, with lower nitric oxide and hydrocarbon emissions when compared with open chambers.

Similar results were obtained by Gruden and Hahn (1980). These investigators came to the conclusion that the combustion chamber located in the piston crown could be better suited to comply with today's requirements of fuel consumption, emissions and power output than the chamber in the cylinder head. In fact, the bowl-in-piston combustion chamber facilitates the choice of spark plug location, the section of the quench area, it requires a low octane number to operate knock-free at a given compression ratio, the conditions for leaning the mixture are more favorable and it is less sensitive to exhaust gas recirculation (EGR). The optimum value of the compression ratio seems to be in the vicinity of thirteen to one which has also been found by Thring and Overington (1982).

Sutton (1983) described the effects that a high compression combustion chamber located in the cylinder head under the inlet valve and operating in the lean burn concept has on knock, fuel economy and emissions. Again a 13 : 1 compression ratio utilising 98 RON gasoline has been found to give very good results, with improvements of the order of ten percent over conventional designs. Some advantages claimed for the combustion chamber under the inlet valve are :

- .faster burning of the charge improved significantly the knock-free operation
- .scavenging of the spark plug was better when it faced the incoming fresh charge
- .the chamber under the inlet valve has a relatively cooler environment and it is possible that the end gas remote



from the spark plug is also at lower temperature.

Poulos and Heywood (1983) investigating the effects of chamber geometry on the combustion rate of spark ignition engines presented some interesting conclusions regarding the location of the spark plug in the combustion space. They noted that model predictions underscored the importance of combustion chamber geometry in determining burn duration. The spark plug location was found to be much more important to burn duration than chamber geometry. Relocation of the spark plug from the side to the center reduced drastically burn duration, the main combustion stage being more affected than the early combustion stage.

Engels and Pirault (1984) found that the relative quiescent area in the vicinity of the spark plug of a high compression ratio compact chamber may help to reduce: ignition delay and that a shrouded position does not favor a fast kernel development. The more prominent position of the spark plug in the open chamber favours a fast kernel growth.

The modern tendency in combustion chamber geometry includes the adoption of three or four valves configuration in order to improve volumetric efficiency. Four valve engines were found to return up to sixty percent higher power output values than the conventional two-valve configuration. The position of the spark plug in the four-valve lay-out allows a combustion chamber configuration which closely approaches the ideal spherical shape.

Gruden et al (1984) found that the modern spark ignition engine is more efficient, presenting a lower fuel consumption, high power output, more favourable exhaust emissions when configured to the four-valve system.

Colins and Stokes (1984) used a Ricardo Hydra research

engine to investigate the performance of four-valve combustion chambers, high compression combustion chambers and open chambers. The performance at full load was very similar for the three chambers operating at low speeds but for high speed operation, the four-valve with its improved volumetric efficiency was superior. For partial load operation, the high compression compact chamber showed the best performance allowing the most extended lean limit.

#### 1.8 Ethanol- alternative fuel for spark ignition engines

From the earliest development of the internal combustion engine, alcohol has been investigated as an engine fuel. The most widely used alcohol, until some years ago, was methanol as a high performance racing fuel. Here, its high octane quality and evaporative cooling properties, made it ideal for the application.

Ethanol has been used intermittently, but not on a large scale, since the early 1900 when it was burned in farm tractors during periods of economic crisis.

Brazil has become the first country to utilize ethanol on a large scale basis as an alternative fuel for spark ignition engines, with nearly two million vehicles planned to be running on pure ethanol by the end of 1986. Anhydrous ethanol is also being blended with gasoline in a proportion of twenty percent ethanol and eighty percent gasoline. The excellent anti-knock property of ethanol makes it a very attractive substitute for tetra-ethyl lead as a knock suppressor.

Sugar cane is the most important resource for ethanol production in Brazil. The technology is comparatively simple and has not changed considerably for many years. Most ethanol from sugar cane is produced by fermentation of molasses, a by product of sugar processing containing

about fifty to fifty five percent fermentable sugars.

However, in any large scale plant the whole juice of cane may be fermented as is the practice in some of the newer distilleries.

The most important problem of a large scale ethanol industry seems to be the effluent disposal. For every litre of ethanol, about ten litres of highly polluting stillage, or slops, are produced. To minimize the problem of its disposal, this stillage disposal can be applied as a fertilizer in sugar cane fields either in irrigation water or direct spraying from road tankers. Another practice is to treat by evaporation to produce a concentrate which may be used for stock feed or as fertilizer. Despite the efforts in this area, it seems that a satisfactory position has not yet been achieved in solving problems of stillage disposal from ethanol in sugar cane areas throughout the world.

Firstly, ethanol is separated from the residue of fermentation which forms the so called vinasse, a high polluting constituent which represents a major problem. Ethanol is subsequently concentrated to near the azeotropic limit of about ninety six percent ethanol by volume. Further azeotropic distillation using benzene, produces the anhydrous ethanol generally 99.6 percent ethanol or better.

Some interesting characteristics of ethanol as a motor fuel will be identified here. While gasoline and diesel fuels are essentially a mixture of hundreds of different hydrocarbon compounds each having its own specific boiling point, ethanol is a pure substance having a unique boiling point. Therefore, gasoline and diesel fuel, depending on blending exhibit a range of properties whereas ethanol has specific properties.

They also have important differences in their chemical composition. In particular ethanol contains in its molecule oxygen together with carbon and hydrogen, which implies a lower heating value considerably below the values of gasoline and diesel fuel. The high octane rating of ethanol makes it ideal for fueling spark ignition engine but its low cetane rating makes it difficult to use in diesel engines. The most important properties that characterize the fuels discussed are represented in Table (11).

The higher heat of vaporization of ethanol has a great effect on its evaporative cooling capacity, which means that at least 3.7 times more heat is necessary to vaporize ethanol when compared with that necessary to vaporize gasoline for an equivalent engine power output. Furthermore, the low vapour pressure of ethanol at relatively low temperatures causes problems of cold starting, requiring the use of auxiliary devices when the temperature drops below ten degrees Celsius.

Ethanol also acts as a solvent and thus attacks some of the materials normally found in vehicle fuel systems designed to handle gasoline.

One further important characteristic of ethanol is its much wider flammability limits when compared with gasoline, which means that a storage tank may have saturated vapour within the explosive range at normal ambient temperatures. With gasoline this concern does not exist because the saturated vapour is too rich to ignite.

If ethanol is to be used as a fuel for spark ignition engines, the engine has to be properly designed to accommodate these features

The main modifications to be introduced are :

- .carburettor recalibration
- .ignition timing

Table 1.1 (From Heywood, J.B. - see Bibliography)

## Data on fuel properties

| Fuel              | Formula<br>(phase)                                     | Molecular<br>weight | Specific<br>gravity:<br>(density,†<br>kg/m³) | Heat of<br>vaporization,<br>kJ/kg‡ | Specific heat        |                                     | Higher<br>heating<br>value,<br>MJ/kg | Lower<br>heating<br>value,<br>MJ/kg | LHV of<br>stoich.<br>mixture,<br>MJ/kg | (A/F) <sub>s</sub> | (F/A) <sub>s</sub> | Fuel octane rating |       |
|-------------------|--------------------------------------------------------|---------------------|----------------------------------------------|------------------------------------|----------------------|-------------------------------------|--------------------------------------|-------------------------------------|----------------------------------------|--------------------|--------------------|--------------------|-------|
|                   |                                                        |                     |                                              |                                    | Liquid,<br>kJ/kg · K | Vapor c <sub>p</sub> ,<br>kJ/kg · K |                                      |                                     |                                        |                    |                    | RON                | MON   |
|                   |                                                        |                     |                                              |                                    |                      |                                     |                                      |                                     |                                        |                    |                    |                    |       |
| Practical fuels   |                                                        |                     |                                              |                                    |                      |                                     |                                      |                                     |                                        |                    |                    |                    |       |
| Gasoline          | C <sub>8</sub> H <sub>1.878</sub> (l)                  | ~110                | 0.72–0.78                                    | 305                                | 2.4                  | ~1.7                                | 47.3                                 | 44.0                                | 2.83                                   | 14.6               | 0.0685             | 92–98              | 80–90 |
| Light diesel      | C <sub>12</sub> H <sub>1.88</sub> (l)                  | ~170                | 0.84–0.88                                    | 270                                | 2.2                  | ~1.7                                | 44.8                                 | 42.5                                | 2.74                                   | 14.5               | 0.0690             | —                  | —     |
| Heavy diesel      | C <sub>16</sub> H <sub>1.78</sub> (l)                  | ~200                | 0.82–0.95                                    | 230                                | 1.9                  | ~1.7                                | 43.8                                 | 41.4                                | 2.76                                   | 14.4               | 0.0697             | —                  | —     |
| Natural gas§      | C <sub>3.88</sub> H <sub>8</sub> N <sub>0.18</sub> (g) | ~18                 | (~0.79†)                                     | —                                  | —                    | ~2                                  | 50                                   | 45                                  | 2.9                                    | 14.5               | 0.069              | —                  | —     |
| Pure hydrocarbons |                                                        |                     |                                              |                                    |                      |                                     |                                      |                                     |                                        |                    |                    |                    |       |
| Methane           | CH <sub>4</sub> (g)                                    | 16.04               | (0.72†)                                      | 509                                | 0.63                 | 2.2                                 | 55.5                                 | 50.0                                | 2.72                                   | 17.23              | 0.0580             | 120                | 120   |
| Propane           | C <sub>3</sub> H <sub>8</sub> (g)                      | 44.10               | 0.51 (2.0†)                                  | 426                                | 2.5                  | 1.6                                 | 50.4                                 | 46.4                                | 2.75                                   | 15.67              | 0.0638             | 112                | 97    |
| Isooctane         | C <sub>8</sub> H <sub>18</sub> (l)                     | 114.23              | 0.692                                        | 308                                | 2.1                  | 1.63                                | 47.8                                 | 44.3                                | 2.75                                   | 15.13              | 0.0661             | 100                | 100   |
| Cetane            | C <sub>16</sub> H <sub>34</sub> (l)                    | 226.44              | 0.773                                        | 358                                | —                    | 1.6                                 | 47.3                                 | 44.0                                | 2.78                                   | 14.82              | 0.0675             | —                  | —     |
| Benzene           | C <sub>6</sub> H <sub>6</sub> (l)                      | 78.11               | 0.879                                        | 433                                | 1.72                 | 1.1                                 | 41.9                                 | 40.2                                | 2.82                                   | 13.27              | 0.0753             | —                  | 115   |
| Toluene           | C <sub>7</sub> H <sub>8</sub> (l)                      | 92.14               | 0.867                                        | 412                                | 1.68                 | 1.1                                 | 42.5                                 | 40.6                                | 2.79                                   | 13.50              | 0.0741             | 120                | 109   |
| Alcohols          |                                                        |                     |                                              |                                    |                      |                                     |                                      |                                     |                                        |                    |                    |                    |       |
| Methanol          | CH <sub>4</sub> O(l)                                   | 32.04               | 0.792                                        | 1103                               | 2.6                  | 1.72                                | 22.7                                 | 20.0                                | 2.68                                   | 6.47               | 0.155              | 106                | 92    |
| Ethanol           | C <sub>2</sub> H <sub>6</sub> O(l)                     | 46.07               | 0.785                                        | 840                                | 2.5                  | 1.93                                | 29.7                                 | 26.9                                | 2.69                                   | 9.00               | 0.111              | 107                | 89    |
| Other fuels       |                                                        |                     |                                              |                                    |                      |                                     |                                      |                                     |                                        |                    |                    |                    |       |
| Carbon            | C(s)                                                   | 12.01               | ~2§                                          | —                                  | —                    | —                                   | 33.8                                 | 33.8                                | 2.70                                   | 11.51              | 0.0869             | —                  | —     |
| Carbon monoxide   | CO(g)                                                  | 28.01               | (1.25†)                                      | —                                  | —                    | 1.05                                | 10.1                                 | 10.1                                | 2.91                                   | 2.467              | 0.405              | —                  | —     |
| Hydrogen          | H <sub>2</sub> (g)                                     | 2.015               | (0.090†)                                     | —                                  | —                    | 1.44                                | 142.0                                | 120.0                               | 3.40                                   | 34.3               | 0.0292             | —                  | —     |

(l) liquid phase; (g) gaseous phase; (s) solid phase.

† Density in kg/m³ at 0°C and 1 atm.

‡ At 1 atm and 25°C for liquid fuels; at 1 atm and boiling temperature for gaseous fuels.

§ Typical values.

RON, research octane number; MON, motor octane number.

## Sources:

E. M. Goodger, *Hydrocarbon Fuels: Production, Properties and Performance of Liquids and Gases*, Macmillan, London, 1975.E. F. Obert, *Internal Combustion Engines and Air Pollution*, Intext Educational Publishers, 1973 edition.C. F. Taylor, *The Internal Combustion Engine in Theory and Practice*, vol. I, MIT Press, 1966.J. W. Rose and J. R. Cooper (eds.), *Technical Data on Fuel*, 7th ed., British National Committee, World Energy Conference, London, 1977.



- .valve timing
- .heat to inlet manifold should be increased
- .compression ratio should also be increased to compensate for the low stoichiometric air/fuel ratio

Due to the lower stoichiometric air/fuel ratio of ethanol and its high heat of vaporization, carburettor recalibration becomes essential and heating of the incoming mixture to increase driveability highly desirable. As the fuel has low heat of combustion it is highly desirable to operate the engine at high compression ratios in order to reduce the specific fuel consumption.

The water content in the fuel is also a critical issue and must be kept to very tight limits, unless the engine is equipped with a self-adjusting metering system which controls the fuel/air ratio at a constant value, using for example an exhaust oxygen sensor with feed back to the carburettor.

Emissions by spark ignition engines operating with ethanol has received greater attention by researchers in the past twenty years or so.

Starkman et al (1964), for example, did not include an investigation of exhaust emissions in their comparative study of the performance of a single cylinder spark ignition engine operating on ethanol, methanol, benzene and iso-octane. They confirmed that the output of the engine operating with alcohols is increased in relation to the operation with the other two hydrocarbon fuels at the same compression ratio of 9:1. A heavy increase in fuel consumption following this increase in output was also reported.

Lowry and Devoto (1976) have however conducted investigations on exhaust emissions in a single cylinder engine fueled with ethanol. They measured exhaust

emissions as a function of equivalence ratio but using the same compression ratio as that used when operating the engine with gasoline.

They came to the conclusion that the emissions of unburned hydrocarbons, on a mass basis, were significantly lower for methanol than for either ethanol or gasoline. However, gasoline engines presented a higher percentage of NOx emissions, on a mass basis, than either ethanol or methanol in the range of equivalence ratios from 0.8-1.0.

Pischinger and Pinto (1979) have presented data for vehicles operating on ethanol but neither studied the effects that changes in compression ratio and equivalence fuel/air ratio would have on the performance of the engine.

Knock in spark ignition engines operating with ethanol has been investigated by Loeck and Menrad (1981) who found that the maximum ignition advance limited by knock is approximately equivalent to the maximum advance for the best torque. Knock was also found to be highly dependent on the temperature, not only on the ambient temperature but also on the general level of engine temperature. Consequently, the knock limits of alcohol fuels will be much more sensitive to temperature changes of the fuel/air mixture and/or coolant temperature than normal hydrocarbon fuels.

The high sensitivity of the alcohol fuels, represented by the difference between the Research Octane Number RON, and the Motor Octane Number MON, points out in the same direction, according to the authors.

In fact, knock is today the dominating factor in the design and adjustment of modern engines. Menrad et al (1982) showed that in the adjustment of high compression alcohol-fueled engines, the pre ignition problem also

seems to be an important factor and therefore requires a more careful investigation, in addition to the knock problem.

It is well established that knock is due to the auto-ignition of the unburned mixture near the end of the combustion process, whereas pre-ignition is due to auto-ignition of the unburned mixture prior to spark occurrence. This can be attributed to several factors such as carbon deposits that becomes incandescent as the load in the engine increases, to some engine parameters and the way in which sensitive fuels react to them, and finally to overheated parts in the combustion chamber after the engine has been subjected to severe knocking.

Because some fuels, such as methanol, have a strong tendency to decompose under relatively low pressure and temperature, which makes them very susceptible to auto-ignition, great care must be taken to ensure that the combustion chamber will not have components susceptible to become overheated during operation so producing "hot-spots". The most common sources of "hot-spots" are the spark plugs and the valves. These must have the right thermal capacity for heat dissipation in order to avoid the possibility of overheating thus leading to pre-ignition. For ethanol fueled engines, the risks of pre-ignition are minimized since ethanol is less susceptible to decomposition and consequently to auto-ignition.

The authors concluded that although fuel composition has some importance on the pre-ignition phenomenon, the strongest influence results from the choice of spark plug. The Heat Number must be as high as possible, without producing a deterioration in the cold-starting process. Other sources of pre-ignition, such as overheated valves, must also be avoided if the spark plug selected is to give the best performance. The highest pre-ignition sensitivity

was found for the equivalence ratio in the range 1.05-1.10.

### 1.9 Emissions in spark ignition engines

The beginning of a new era in automotive engine combustion research appears in the 1950s when it became evident that hydrocarbon emissions were responsible for the photochemical smog in the atmosphere. New techniques had to be developed to measure emitted hydrocarbons in ppm concentrations of exhaust gas and design strategies developed to respond to the challenges posed.

Since that time a great deal of resource and effort has been put into combustion research in internal combustion engines with the aim of reducing their exhaust emissions. Today, the main types of pollutant emissions from a spark ignition engine have been exhaustively investigated: unburned hydrocarbons (HC), carbon monoxide (CO) and oxides of nitrogen (NO<sub>x</sub>).

#### 1.9.1 Unburned hydrocarbons

Three sources are known to be responsible for most of the hydrocarbon exhaust emissions, namely wall quenching, incomplete combustion of the charge and scavenging of the gases in the cylinder.

Wall quenching is likely to be present everytime a flame tries to propagate in the vicinity of a cold wall. The effect is then a significant slowing down of the chemical reaction due to both a break in the sequence of chain reactions and to heat loss in the layer of the charge adjacent to the cold wall.

Daniel (1957) was the first to study wall quenching in an engine, using a photographic technique to study the combustion process through a quartz window in a CFR

engine. The pictures suggested that as the flame passed through the chamber a dark region of unreacted mixture was left adjacent to the wall. He argued that this unreacted mixture subsequently reacts with the burned charge and escapes out of the exhaust. Further experiments showed however that a significant amount of the non-burned hydrocarbons produced by wall quenching of the charge, were burned inside the cylinder when in contact with the hot combustion products containing oxygen later in the cycle during the expansion and exhaust strokes.

Property gradients are high in wall regions and therefore diffusive fluxes are also large, promoting mixing and combustion in the product gases.

Higher exhaust gas temperature, obtained by either insulating the exhaust pipe or using leaner mixtures in conjunction with a retarded spark timing, also enhanced the hydrocarbon oxidation. It was found that the residence time of the gases also influenced the oxidation of hydrocarbons.

The dimensions of the combustion chamber, expressed as a ratio of its surface area to its volume, plays a very important role in the formation of the hydrocarbon layer. The thickness of that layer is very small, of the order of a few hundredths of a millimeter, the volume then being proportional to the combustion chamber surface area as demonstrated by Daniel (1957).

The volume and area measurements are complicated by the movement of the piston but for purposes of comparison between different combustion chambers the areas and volumes are normally taken with the piston at TDC. This simplification turns out to be sufficiently accurate for most applications.

Surface to volume ratio is not the only method used to



compare exhaust hydrocarbon emissions between different combustion chambers, other factors such as air/fuel ratio, the variability with time of the amount of HC passing through the exhaust valve and the oxidation of HC during and after the exhaust stroke are also important.

Incomplete combustion in an engine may arise as a consequence of mixtures being extremely lean or rich or where exhaust gas dilution is excessive. Under these conditions, incomplete flame propagation may occur in some cycles and in some cases a complete misfire may take place

The factors which promote incomplete flame propagation and misfire include:

- .poor condition of the ignition system, including spark plugs
- .low charge temperature
- .poor charge homogeneity
- .mixture too rich or too lean
- .large exhaust residual gases
- .lack of fuel in the residuals from previous cycle
- .poor distribution of the residuals within the cylinder
- .scavenging of the cylinder in a two-stroke engine, produces large amount of hydrocarbons once the fuel is blown through the cylinder directly into the exhaust and escapes the combustion process.

In four-stroke engines this problem may be considered practically non-existent except in supercharged engines which may have some HC emissions from this source.

Huls et al (1968) investigated the mechanisms of combustion that affect exhaust emissions and the relationship of those mechanisms to engine design and operating variables. They came to some interesting conclusions from their experiments which are presented in sequence.

Low HC exhaust emissions from an engine could be obtained by leaning the mixture to 16:1 air/fuel ratio. Reductions in HC emissions can also be obtained when the spark timing is set to retard the ignition which causes an increase in the exhaust temperature with a consequently increase in burned-out HC. Increasing the intake manifold pressure and/or decreasing compression ratio also was found to reduce the amount of HC emissions.

The coolant temperature was found to have little influence on cooled-exhaust HC emissions when the coolant temperature was increased from the boiling point of water to the boiling point of ethylene glycol, but HC emissions increased dramatically when the coolant temperature was decreased from the boiling point to the ambient temperature. Retarding the ignition was found to cause an increase in the HC leaving the cylinder but not the exhaust system, the reason being that the HC reacts in the exhaust port and exhaust system. Compression ratio was found to play an important role in the HC emissions and a reduction in CR from 8:1 to 6:1 resulted in a substantial reduction in the levels of cooled-exhaust HC emissions, of the order of forty three percent.

Knock was found to have some influence on the HC exhaust emissions of a spark ignition engine as demonstrated by Duke et al (1970). They examined the effect of knock intensity on exhaust emissions in a single cylinder engine and concluded that the overall effect of knock intensity on unburned HC emissions is to reduce their exhaust concentration. The HC emissions increase first to a maximum near the condition of trace knock and then decrease with increasing knock intensity. The authors do not present an explanation for that phenomenon but one possibility is that as the knock intensity increases, greater maximum cycle temperatures are attained in the combustion chamber bringing as a consequence, a reduction on the HC emissions. They also found that maximum

reduction in HC occurs when operating with stoichiometric mixtures unlike the findings of some previous workers mentioned earlier.

Lee (1971) investigated the effect of compression ratio and other parameters upon the exhaust emissions of a single cylinder engine operating at constant speed and found that spark advance and air/fuel ratio largely determined the concentration of exhaust HC, followed by coolant temperature. Compression ratio was not found to have a pronounced effect on the HC emissions.

Spark plug position was also found by Quader (1973) to influence the amount of HC emitted from a spark ignition engine, using a single cylinder engine having a modified combustion chamber. The effects of spark plug location and combustion duration on HC emissions were such as to give the lowest HC emissions when the spark plug was placed near the exhaust valve and highest with the spark plug near the intake valve.

Heywood (1976) pointed out that the crevices in the combustion chamber can also have importance in the production of unburned HC, once they are too narrow for the flame to enter. It was found that the most important crevice is the volume between the piston and cylinder wall above the first compression ring where the flame is quenched by heat loss at the crevice entrance.

#### 1.9.2 Carbon monoxide

Carbon monoxide is an intermediate product formed during the combustion process. In the spark ignition engine the CO is primarily controlled by the fuel/air ratio. For fuel rich mixtures, the carbon monoxide concentrations in the exhaust increase proportionally with the equivalence ratio whilst for lean mixtures concentrations of CO in the exhaust vary little with equivalence ratio.

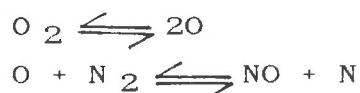
Huls (1968) pointed out that low carbon monoxide emissions are obtained by lean operation and the only engine operating variable that affects CO is the fuel/air ratio. In a rich mixture, the mechanism of formation of CO is clearly insufficiency of oxygen but in a lean mixture the controlling mechanism is not clear. It is possible that inhomogeneities due to exhaust residuals or the formation of CO in the quench zone or at the interface between quench zone and bulk gas may cause CO to appear. As the hydrocarbon oxidation route is via the CO, the occurrence of quenching will restrict the CO burn-out.

Lee (1971) performed tests to observe the effects of varying compression ratio, stoichiometry and coolant temperature on the exhaust emissions. He found that concentration of CO was affected mainly by stoichiometry. To a certain extent he confirmed the point of view of Huls (1968) when he stressed that increasing the coolant temperature caused a slight increasing in the CO concentration and that compression ratio and spark advance produced still smaller effects.

### 1.9.3 Oxides of nitrogen

Nitric oxides are formed at the peak combustion temperature within the combustion chamber and persist during expansion and exhaust in non-equilibrium concentrations. Oxides of nitrogen can react in sunlight to produce photochemical smog, a peculiar pollution problem common to areas with great concentration of vehicles and peculiarly stable atmospheric conditions.

The pioneering work on nitric oxides was carried out by Newhall and Starkman (1967) who identified the important contribution of the so called Zeldovich chain reaction:

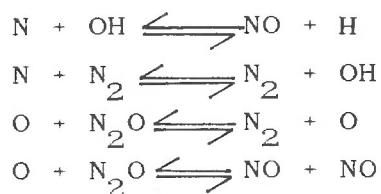




as a mechanism for NO formation.

In this mechanism the chain reactions are initiated from the second equation by the atomic oxygen formed in the first equation describing the dissociation of oxygen molecules at the high temperatures reached during the combustion process. In the third equation, molecular oxygen reacts with atomic nitrogen producing further nitric oxide and atomic oxygen.

Lavoie, Heywood and Keck (1970) extended the Zeldovich mechanism with the addition of the reaction between the nitrogen and hydroxyl radicals and other reactions involving nitric oxide.



It was found that the removal process of NO under the engine conditions was very slow and thus the concentration of nitric oxide remains nearly constant through the expansion process. The extended Zeldovich mechanism is by far the most commonly used and satisfactory model of NO formation in spark ignition engines. The majority of the investigation carried out since the pioneering work of Newhall and Starkman (1967) on the mechanism of nitric oxide formation have sought to associate the formation of oxides of nitrogen with the important parameters of engine operation.

Huls et al (1968) observed that low NOx emission may be obtained by operating the engine with lean mixtures or by lowering the peak temperature achieved during the combustion cycle. As expected, increasing the inlet air



temperature caused the NO<sub>x</sub> formation to increase but it decreased again if very high inlet air temperature is used. The reason is that excessively high inlet air temperature causes the volumetric efficiency to decrease. Thus the exhaust gas dilution of the charge is increased and also the heat losses per unit charge. At some value of the inlet air temperature the effects above mentioned cause the maximum temperature reached during the combustion process to decrease, thus reducing the formation of nitric oxide. The influence of the coolant temperature on NO<sub>x</sub> formation was found to be small.

Duke et al (1970) investigating how knock could affect exhaust emissions found that the nitric oxide emissions increased with the knock intensity for lean mixtures whilst for rich mixtures the NO in the exhaust was usually almost constant.

Felt and Krause (1971) found that the compression ratio had no effect on NO<sub>x</sub> emissions provided equal power is developed by the engines under test at steady state conditions as measured by a dynamometer.

Komiyama and Heywood (1973) developed a theoretical model for predicting the nitric oxide concentrations in the exhaust of a spark ignition engine. The theoretical model was based on the previous work of Lavoie et al (1970) and makes use of the extended Zeldovich mechanism for NO<sub>x</sub> formation. The model was validated in a CFR single cylinder engine and the results were reported to give good agreement with the predicted values over all operating conditions.

From that work it could be inferred that the process of NO<sub>x</sub> formation in rich and lean mixtures have slightly different characteristics. In lean mixtures the NO<sub>x</sub> chemistry freezes early in the expansion stroke, thus little NO decomposition occurs while in rich mixtures,

the freezing of the NOx chemistry occurs later in the expansion stroke after the charge is totally burned, which causes a substantial amount of NOx to be decomposed.

Spark plug location was found to have some influence on NOx emissions. Quader (1973) pointed out that the combustion chamber operating with a dual spark plug system or having the spark plug placed in the center of the chamber presented the highest NOx emissions levels, while no significant variation on these levels were detected for the spark plug at the side of either exhaust valve or the inlet valve. It can be concluded then that any attempt to reduce the combustion duration time by relocating the spark plug or by using dual spark plugs increases the NOx emissions, as reported by the authors.

Sakai et al (1973) investigated the effect of combustion chamber shape on oxides of nitrogen in a single cylinder spark ignition engine. They came to the conclusion that combustion chamber geometry has only a slight influence on the NOx emission. If the position of the spark plug is such as to permit a lowering of the peak combustion temperature, the NOx level can be greatly reduced. Engine output, however, is not strongly affected by combustion chamber geometry and spark plug location.

Heywood et al (1979) developed a computer simulation of the four stroke spark ignition engine to study the effects of load, engine speed, combustion duration, spark timing, equivalence ratio, exhaust gas recirculation and compression ratio on the efficiency and NOx emissions. After validation of the model using a 5.7 litres displacement engine, the conclusions regarding NOx emissions may be summarised as follows: increasing load also increases the brake specific nitrogen oxide (bsNO) due to an increase in charge density; increasing speed cause bsNO to be increased since the NO chemistry freezes earlier in the expansion stroke, while retarded timing

decreases bsNO; also, bsNO decreases with increasing compression ratio.

A simulation model of the direct injection stratified charge engine from Texaco has been developed by Hiraki and Rife (1980) to predict the performance and NOx emission of that engine. The model makes use of the extended Zeldovich mechanism on NO formation, with a steady state assumption for the N concentration and equilibrium values used for H, O, O<sub>2</sub>, and OH concentrations. They found that NOx values were predicted with good accuracy. NOx concentration is very sensitive to the mixing process and it can be drastically reduced by rapid mixing after the spark plug. Because a higher swirl ratio can produce rapid cooling of the burned gas, a beneficial effect on NOx concentration at high equivalence ratio was found. At low equivalence ratio, on the other hand, higher swirl ratio has an unfavourable effect on NOx due to the coincidence of higher gas temperatures and peak cylinder pressures.

# Chapter 2

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## Theoretical Investigation

## 2.1 Introduction

The computer programme which is to be presented is based on a phenomenological model. Recent efforts in combustion modelling have in general been aimed at detailed multidimensional models which are intended to characterize engine processes.

While these models are of great help in understanding the detailed thermodynamic processes of engine combustion and in principle are capable of calculating the evolution of the fuel/air mixture in both time and space they suffer many limitations. For example, the theoretical formulations are inevitably incomplete and there is lack of experimental data that can be used for their validation. In particular there remains a significant lack of understanding of turbulent processes and uncertainty concerning the kinetics of hydrocarbon fuel oxidation. These factors continue to limit the usefulness of such models as engine design tools.

On the other hand, phenomenological models, though less general, often appear to be more realistic in their specific application to spark ignition engines, judging from the successful application of these models to so many cases.

They can reduce drastically the time and cost during the development phase of a new engine, predicting with good accuracy, the trends of specified parameters when others are varied in a certain operating range. This shortens the the design time since it eliminates the need to first build and test each engine configuration in order to obtain relevant data of its performance.

The submodels, representing the physical phenomena can be very useful to understand the complex processes involved. All the parameters considered to have influence on the



process can be varied, therefore it is easy to obtain data for the whole range under study.

Phenomenological models are essentially characterized by the fact that they are constructed from the formalism of the first law of the thermodynamics. The only real independent variable considered is the time. Structurally speaking the model comprises several separate submodels, each one representing a particular physical process. The degree of accuracy of the model is evidently dependent of the rigour with which the submodels are formulated.

The present model is intended to be used as a predictive tool for the study of different combustion chamber geometries for spark ignition engines operating on different fuels, including in particular hydrated ethanol.

The theoretical work presented here consists of the development of a computer simulation model that allows alternative combustion chamber designs to be evaluated simply and rapidly. The model can be divided in two sections, namely:

- .a thermodynamic model which is essentially based on the first law of thermodynamics and includes features such as heat transfer between gases and cylinder walls, turbulent flame propagation, detailed combustion chemistry, to mention the most important of them.
- .A geometric model that assumes spherical flame propagation into a cylindrical combustion chamber extended to simulate bowl-in-piston combustion chamber types. The flexibility of the models allows other geometries to be added.

## 2.2 The thermodynamic model

The present work considers the period during which both the inlet and exhaust valves remain closed ie, during

the power cycle of the engine. The open period, corresponding to the gas exchange process has not been considered. The power cycle includes the compression, combustion and expansion processes.

In order to develop the model some key assumptions were made and these are summarised as follows:

.The pressure and temperature of the gases are uniform throughout the cylinder, the temperatures of the burned and unburned gases being calculated by means of ideal gas laws.

.At any instant, the unburned mixture consists of homogeneous air-fuel mixture and residual gases and moisture without any chemical reaction and further details are given later.

.The heat transfer area was assumed to be constituted of five distinct areas, namely, the cylinder head without valves, the piston crown, the instantaneous cylinder wall area, the admission and exhaust valve areas. The metal temperatures of each of these surfaces were considered to be constant throughout the cycle, however, the mean wall temperature used for heat transfer calculations varies with time.

.The heat transfer from the gas to the walls of the combustion chamber has been calculated by making use of the instantaneous heat transfer coefficient, the respective area and the difference between the mean gas temperature and the mean wall temperature.

The model uses a straightforward iterative technique in which the condition of the gas at a specific crank angle is used as the initial data for the calculation of the properties at an increased crank angle and so on until the end of the calculation stage. Details will be given when describing each process separately.

The model can simulate the effects of engine speed, combustion chamber shape, spark plug position, equivalence

ratio, ignition timing, compression ratio, differences in fuel and different engine geometries.

Important features such as dissociation of the combustion products, residual gas dilution, variable specific heats and heat transfer are also included in the computer simulation model. Within the framework of this zero-dimensional model, cylinder average properties such as charge temperatures, pressures, turbulent burning velocity, flame radius, heat transfer, work done, indicated mean effective pressure, indicated specific fuel consumption, and some other parameters are calculated.

In order to avoid the simple assumption of data to start the computations, such as the conditions of the inlet mixture at the end of the admission process, these parameters were calculated using relations derived from gas laws, as presented in the next section. This approach does not consider, as pointed out previously, the dynamics of the admission process.

#### 2.2.1 Conditions during admission

During the admission process a certain amount of mixture formed by atomized fuel and air with certain amount of humidity, enters the cylinder and mixes with a certain amount of residual gases from the previous combustion. For limited overlapping of the valves, for example, less than 30 degrees crank angle (DCA) there is no possibility that the total scavenging of the residual gases occurs which will limit the amount of fresh charge admitted in the cylinder. For this reason, the process of admission should be analysed taking into account the parameters that describe also the exhaust process, considering the process of gas exchange as a whole.

The present model does not consider the gas exchange

processes, only the processes occurring during the closed valves period being modeled. However, in order to obtain values for the initiation of the cycle calculations, such as temperature, pressure, volume, residual gas fraction, etc., at the conditions when the inlet valve closes, some theory is presented to explain the method adopted to calculate the above mentioned parameters.

The amount of fresh mixture admitted during the admission process depends on the following factors:

.The flow resistances characteristic of the intake system, which causes a pressure drop in the intake manifold.

.The presence of certain amount of residual gases in the combustion chamber, reducing the total volume available for the fresh mixture

.Heating of the charge by the surface of the intake manifold and cylinder walls, reducing the density of the charge.

Due to the pressure drop caused by the resistance to the flowing of the mass of mixture through the inlet system, the total amount of charge is reduced because the density of the charge decreases. The pressure in the cylinder at the end of the admission process can be estimated with good accuracy, considering the admission to be a steady-state process. In this case, using the Bernoulli equation results

$$P_i / \rho_i + v_i^2 / 2 + g H_i = P_a / \rho_a + \beta v_a^2 / 2 + \xi v_a^2 / 2 + g H_a \quad (1)$$

where  $\beta$  is a coefficient that takes into account the drop in the charge velocity at the cross section of the cylinder and  $\xi$  is a coefficient for the intake system referred to its narrowest section. Both coefficients will be given numerical values later in this section.

Assuming that  $V_i = 0$  and  $H_i = H_a$ , and neglecting the change in density of the fresh charge when it passes through the inlet manifold  $\rho_i = \rho_o$ , the Bernoulli equation can be written as

$$P_i / \rho_i = P_a / \rho_i + (\beta + \xi) V_a^2 / 2 \quad (2)$$

$$\Delta P_a = P_i - P_a = (\beta + \xi) V_a^2 \rho_i / 2 \quad (3)$$

in a four stroke engine without supercharging,

$$P_i = P_o \quad \text{and} \quad \rho_i = \rho_o$$

Therefore, the decrease in pressure when the charge moves through the intake system is proportional to the square of the velocity in the minimum cross section and depends upon the resistance coefficients and the velocity drop.

The continuity equation for the minimum section of the intake system and the cylinder section is:

$$V_a A_v = V_p A_p \quad (4)$$

where  $A_p$  is the area of passage through the intake valve, considered to be the narrowest section and given by

$$A_v = \prod D_i h \quad (5)$$

The maximum velocity of the piston is estimated from

$$V_p = R \omega (1 + R^2 / L^2)^{0.5} \quad (6)$$

The use of the maximum piston velocity instead of the mean velocity, permits to obtain velocities closer to the actual velocities expected in a four stroke engine, because the area of the maximum cross section through the valve is obtained approximately in the instant when changes in the velocity of the piston are small and the velocity is close



to its maximum value.

Hence, the temperature of the charge at the instant the inlet valve closes can be estimated substituting the values given by equations (4), (5) and (6) in equation (3). The density of the charge at inlet conditions is found by the equation of state

$$\rho_o = P_o / R_u T_o \quad (7)$$

where subscript o indicates atmospheric conditions and  $R_u$  is the gas constant of the mixture.

The numerical value of the term  $(\beta^2 + \xi)$ , the coefficient that takes into account the resistances to the fluid flow in the intake system and the deceleration of the flow when it enters the cylinder is found experimentally. The values suggested by Arkhangelsky et al (1976) are in the range 2.5 - 4. A value of 3 was found to give a good approximation for the pressure at the end of admission in the present work.

The amount of fresh charge that can be admitted to the cylinder will be slightly reduced due to the presence of a certain volume of residual gases left in the cylinder after the exhaust process. This residual gas is at temperature  $T_r$  and pressure  $P_r$ . In order to estimate the amount of residual gas, a coefficient of residual gases is defined as the ratio of the number of moles of residuals to the number of moles of fresh charge admitted

$$C_r = N_r / N_m$$

The volume occupied by the residual gas may be assumed to be the volume of the combustion chamber at the end of the exhaust stroke, which is

$$V_{cc} = V_s / (C_r - 1) \quad (8)$$

From the equation of state the number of moles of residual gas can be estimated

$$N_r = P_r V_{cc} / R_o T_r \quad (9)$$

The value of the pressure exerted by the residual gas will be dependent upon the pressure of the medium into which discharge takes place. If no supercharger or muffler is installed after the exhaust manifold, the pressure is taken to be the atmospheric  $P_o$ .

The temperature of the residual gas depends upon the mixture composition, the expansion ratio and the process of heat exchange during expansion and exhaust. The pressure of the residual gases in the present work was assumed to be  $1.15 P_o$  due to the configuration of the exhaust system of the Ricardo E6. The temperature of the residual gas was taken to be 770 K.

During the admission process, the amount of heat transferred to the charge depends upon the charge velocity, the duration of the admission process and the difference in the temperatures of the charge and walls. In fact  $\Delta t$  is formed by two terms, one that takes into account the elevation in temperature of the charge due to heat transfer and another that considers a decrease in the temperature of the charge due to vaporization of the fuel, ie,

$$\Delta t = \Delta t_{th} + \Delta t_{tv} \quad (10)$$

The value of  $\Delta t$  was determined experimentally in the Ricardo E6 engine by means of the thermocouple placed in the vicinity of the intake port and with the temperature of the mixture being set by an electric heater. The heater was adjusted to give a mixture temperature at the measuring point equal to  $40 \pm 2^\circ \text{C}$ . In the cycle synthesis

program a value of  $40^{\circ}\text{C}$  was taken for  $\Delta t$ .

When the charge enters the cylinder it mixes with the residual gas and as a consequence its temperature is further increased. The temperature of the fresh mixture can be estimated from a heat balance between the fresh mixture and the residual gas before and after the mixing process, which is assumed to occur at constant pressure.

$$N_m C_{pm} (T_o + \Delta T) + N_r C_{pr} T_r = C_{pm} (N_m + N_r) T_a \quad (11)$$

$$N_m + N_r = N_m (1 + Cr) \quad (12)$$

Hence

$$T_a = (T_o + \Delta t + K Cr T_r) / (1 + Cr) \quad (13)$$

The term  $K = C_{pr} / C_{pm}$  for the purpose of estimate the temperature of the mixture in order to start the cycle can be assumed to be unity without introducing a significant error since the difference between  $C_{pm}$  and  $C_{pr}$  is small.

The efficiency with which the cylinder volume can be occupied by the fresh mixture during the admission process can be estimated by the coefficient of admission. It is defined as the ratio of the volume of fresh mixture present in the cylinder when the inlet valve closes to the maximum theoretical value that could be admitted to the cylinder if there were no losses in the process ie,

$$\eta_v = v_i / v_s \quad (14)$$

From the equation of state, the total number of moles of fresh mixture plus residual gas in the cylinder after the intake valve is closed is

$$P_a V_a = R_o T_o (N_m + N_r) \quad (15)$$

But at the beginning of the process

$$P_o V_i = R_o N_m T_o \quad (16)$$

From equations (14), (15) and (16) and considering that

$$V_a / V_s = CR / (CR - 1)$$

$$N_m + N_r = N_m (1 + CR)$$

an expression for the coefficient of admission can be written as

$$\eta_v = CR T_a T_o / [P_o T_a (CR - 1) (1 + CR)] \quad (17)$$

or in a percent basis also referred to as the volumetric efficiency

$$E_v = \eta_v 100 \%$$

If in equation (9) the volume of the combustion chamber  $V_{cc}$  is substituted for  $V_s / (CR - 1)$ , this results in

$$N_r = P_r V_s / [R_o T_r (CR - 1)] \quad (18)$$

Since  $N_m = P_o \eta_v V_s / R_o T_o$  and  $CR = N_r / N_m$  then

$$CR = P_r T_o / [\eta_v P_o T_r (CR - 1)] \quad (19)$$

Solving equation (13) for  $T_a (1 + CR)$  and substituting in equation (17) gives

$$\eta_v = CR P_a T_o / [P_o (CR - 1) (T_o + \Delta t + CR T_r)] \quad (20)$$

If the value of  $CR$  given by equation (19) is inserted in equation (20) and rearranging gives

$$\eta_v = T_o (CR P_a / P_o - P_r / P_o) / [(T_o + \Delta t) (CR - 1)]$$

$$C_r = P_r (T_o + \Delta t) / (T_r C_r P_a - P_r) \quad (21)$$

The volume occupied by the fuel, air and residual mixture after the intake valve closes is given by

$$V_a = \int D S + V_{cl}$$

where S is the position of the piston relative to the top dead center,

From what was above exposed, the set of equations represented by equations (3), (13), (17) and (21) can be used to calculate the conditions of the charge in the cylinder immediately after intake valve closing.

### 2.2.2 The compression process

The compression process was assumed to begin when the inlet valve closes after a certain volume of the combustible mixture has been trapped in the cylinder. During the compression process the dissociation of the combustion products or residual gases is considered to be insignificant and the composition can then be assumed to be constituted by fuel vapour, air and residuals from the previous cycle. Details of the calculation of the charge composition are given in appendix 1.

The amount of fresh charge induced will be dependent upon the volumetric efficiency and the amount of residual gases left in the cylinder. Pressure and temperature of the gases at I.V.C. are estimated making use of the equations derived in the previous section. The value found for the temperature at I.V.C. is then used to calculate the specific heat at constant pressure and constant volume and the ratio of the specific heats is then determined.



The charge is then compressed isentropically from initial state at temperature  $T_1$ , volume  $V_1$  and pressure  $P_1$  to a state at temperature  $T_2'$ , volume  $V_2'$  and pressure  $P_2'$ . At the end of the step, the mixture releases heat to the cooling medium through the combustion walls. As a consequence, there is a decrease in temperature from  $T_2'$  to the final value  $T_2$  and consequently a decrease in pressure from  $P_2'$  to  $P_2$  while the volume remains unchanged. Therefore, after equilibrium is reached, the mixture condition is defined by state  $P_2$ ,  $V_2$ ,  $T_2$ . The heat losses to the coolant are computed using a correlation for an average heat transfer coefficient as detailed in section 2.4.

The crank angle interval that defines the compression step was taken as  $D\theta = 3$  DCA, which gives a corresponding time increment equal to

$$Dt = D\theta / 6 N \quad (22)$$

The process is repeated using the final values of pressure, volume and temperature as the initial values for the next step, the compression process being terminated when the ignition angle is reached. Mathematically the process above can be represented by the set of equations:

$$P_2' = P_1 (V_1 / V_2')^{\gamma} \quad (23)$$

$$T_2' = P_2' V_2' / M R \quad (24)$$

$$T_2 = T_2' - Q_{12} / M C_v \quad (25)$$

$$P_2 = M R T_2 / V_2 \quad (26)$$

$$\gamma = C_p / C_v \quad (27)$$

$$Q_{12} = A h (T_g - T_w) \quad (28)$$

$$W_{12} = 0.5 (P_1 + P_2) (V_2 - V_1) \quad (29)$$

### 2.2.3 The expansion process

The expansion process has been modeled bearing in mind that the gases undergo a dissociation process and therefore the composition cannot be regarded as constant. During the expansion process, also called the power or working stroke, the heat energy released by the combustion of the fuel is converted into mechanical work.

During the initial stage of the expansion process, the fuel is not yet fully burned and the temperature reaches its maximum value some degrees of crank angle after the maximum pressure is reached. The process of expansion was based on the same principles used for the compression process, in particular, the gases are expanded isentropically from state 1 at pressure  $P_1$ , volume  $V_1$  and temperature  $T_1$ , to state 2 at pressure  $P'_2$ , volume  $V'_2$  and temperature  $T'_2$ . Then heat is released to the coolant via the cylinder walls.

Furthermore, there is a decrease in the internal energy of combustion products due to dissociation and both factors considered cause the gas to reach the equilibrium at a new point 2 defined by pressure  $P_2$ , volume  $V_2$  and temperature  $T_2$ . The heat is lost at constant volume and is computed using the the mean heat transfer coefficient as defined in section 2.4.

As in the compression process the crank angle interval that define the expansion step was taken as  $Dth=3$  DCA leading to a time increment equal to

$$Dt = Dth / 6 N \quad (30)$$

The final values of the pressure, volume and temperature

are considered as the initial values for the next step, the expansion phase being terminated when the exhaust valve opens, E.V.O. The set of expressions that characterize what has been exposed above is given below:

$$P_2' = P_1 \left( V_1 / V_2 \right)^\gamma \quad (31)$$

$$T_2' = P_2' V_2 / M R \quad (32)$$

$$T_2 = T_2' - (Q_{12} / M C_v) - U_p / M \quad (33)$$

$$p_2 = M R T_2 / V_2 \quad (33)$$

$$\gamma = C_p / C_v \quad (34)$$

$$Q_{12} = A h (T_g - T) \quad (35)$$

$$W_{12} = 0.5 (P_1 + P_2) (V_2 - V_1) \quad (36)$$

#### 2.2.4 The combustion process

The success of the cycle simulation model will depend predominantly on the accuracy with which the combustion process can be predicted and also on the validity of the assumptions and approximations made for the events that precede and follow the combustion process.

The combustion of the mixture in the cylinder of a spark ignition engine is a very complicated physical and chemical process whose initiation, development and completion depend on the detailed features and speed of the chemical reactions, the conditions of heat and mass exchange in the flame zone and heat transfer to the walls.

The model developed here assumes that a two-zone configuration exists, with the burned and unburned gases being separated by an infinitesimally thin flame front

with the two zones at different temperatures but at the same pressure. Between the two zones there is no heat transfer through the boundary surface, the flame front. Heat is exchanged only between the hot gases and the combustion chamber walls and is estimated using a mean heat transfer coefficient calculated individually for each zone according to the method outlined in section 2.4.

Unburned gases are assumed to be a mixture of non-reactive vapour of fuel, air and exhaust gas residuals with constant composition. Its composition needs therefore to be calculated only once at the start of the cycle (I.V.C.) from the values of the fuel composition, fuel/air ratio and the percentage of residual gases in the fresh charge.

The burned zone contains the products of combustion in chemical equilibrium. The composition is not constant since dissociated products are recombined as the temperature falls.

The pressure, assumed to be spatially uniform throughout the combustion chamber, is a function of time only.

The combustion takes place in the flame front which propagates from the spark plug following a spherical geometry. Blowby and valve leakage losses are not considered in this work.

The method followed to calculate the thermodynamic state of the burned and unburned subsystems at the end of each incremental step is presented in sequence.

#### 2.2.4.1 Combustion

At the begining of each crank angle interval, the thermodynamic state of the two zones are represented by

$MB_1$ ,  $TB_1$ ,  $VB_1$ ,  $MU_1$ ,  $TU_1$ ,  $VU_1$ . The volume of the combustion chamber is  $V_1$  and the uniform pressure is  $P_1$ . In the interval allowed for combustion to occur, the flame front will move to a position 2 consuming a mass of unburned charge  $M_i$  at the turbulent burning velocity. The new position of the flame front will be determined by the relationship

$$Rf_2 = Rf_1 + V_f Dt \quad (37)$$

where  $V_f$  is the flame speed.

The flame radius calculated as above will define a new position for the flame front, corresponding to the end of the combustion step for the new time interval. This flame radius is used to calculate the volume occupied by the burned gases.

The elemental mass  $M_i$  can be calculated by the expression

$$M_i = \rho_u A_f u_t Dt \quad (38)$$

where  $u_t$  is the turbulent burning velocity as defined in section 2.3.3.

The energy released in the combustion of the mass  $M_i$  is

$$E_i = M_i C_p (T_{ad} - T_{u_1}) \quad (39)$$

This energy is used to compress the burned mass  $MB$  and the unburned mass  $MU$ , varying the pressure from  $P_1$  to  $P_2$  and raising the temperature of the unburned charge to  $T_{u_2}$  and that of the burned charge to  $T_{b_2}$ . At the end of the combustion step, the temperature of the elemental mass  $M_i$  will be the same as that of the burned charge, assuming a fully mixed model. Therefore, an energy balance yields the equation



$$\begin{aligned}
M_i C_{p_b} (T_{ad} - T_{u_1}) &= M_{b_1} C_{p_b} (T_{b_2} - T_{b_1}) \\
&+ (M_{u_1} - M_i) C_{p_u} (T_{u_2} - T_{u_1}) \\
&+ M_i C_{p_b} (T_{b_2} - T_{u_1}) \quad (40)
\end{aligned}$$

The LHS of equation (40) is the energy released during the combustion of the elemental mass  $M_i$  and is used to  
 .raise the temperature of the burned charge  $M_{b_1}$  from  $T_{b_1}$  to  $T_{b_2}$ ,  
 .raise the temperature of the unburned charge  $M_{u_2}$  from  $T_{u_1}$  to  $T_{u_2}$ ,  
 .raise the temperature of the burned increment mass  $M_i$  from  $T_{u_1}$  to  $T_{b_2}$ .

From isentropic compression of the products and reactants by the increment mass burned

$$T_{b_2} / T_{b_1} = (P_2 / P_1)^{\frac{\gamma_B - 1}{\gamma_B}} \quad (41)$$

$$T_{u_2} / T_{u_1} = (P_2 / P_1)^{\frac{\gamma_u - 1}{\gamma_u}} \quad (42)$$

Equation (40) can be rearranged as follows, calling

$$M_{u_1} - M_i = M_{u_2}$$

$$\begin{aligned}
M_i C_{p_b} (T_{ad} - T_{u_1}) &= M_{b_1} C_{p_b} T_{b_1} (T_{b_2} / T_{b_1} - 1) \\
&+ M_{u_2} C_{p_u} T_{u_1} (T_{u_2} / T_{u_1} - 1) \\
&+ M_i C_{p_b} (T_{b_2} - T_{u_1}) \quad (43)
\end{aligned}$$

Substituting  $T_{b_2} / T_{b_1}$  and  $T_{u_2} / T_{u_1}$  by their expressions given by (41) and (42) yields

$$\begin{aligned}
Mi \, Cp_b \, (T_{ad} - T_{u1}) &= Mb_1 \, Cp_b \, T_{b1} \left\{ (P_2 / P_1)^{\frac{\gamma_b-1}{\gamma_b}} - 1 \right\} \\
&+ Mu_2 \, Cp_u \, T_{u1} \left\{ (P_2 / P_1)^{\frac{\gamma_u-1}{\gamma_u}} - 1 \right\} \\
&+ Mi \, Cp_b \, (T_{b2} - T_{u1}) \quad (44)
\end{aligned}$$

From perfect gas relationships applied to both burned and unburned phases, results

$$P_2 \, V_{u2} = Mu_2 \, R_u \, T_{u2} \quad (45)$$

$$P_2 \, V_{b2} = Mb_2 \, R_b \, T_{b2} \quad (46)$$

From (46) and from the fact that  $V_{b2} = V - V_{u2}$ , results

$$T_{b2} = P_2 \, (V - V_{u2}) / R_b \, Mb_2 \quad (47)$$

where  $Mb_2 = Mb_1 + Mi$

Considering only the last term of equation (44) and substituting  $T_{b2}$  by the expression given by (47) and rearranging results

$$\begin{aligned}
Mi \, Cp_b \, T_{u1} / R_b \, Mb_2 \, (P_2 \, V / T_{u1} - R_u \, T_{u2} \, Mu_2 / T_{u1} \\
- R_b \, Mb_2)
\end{aligned}$$

After substituting  $T_{u2} / T_{u1}$  by the expression (42) and returning the resulting expression to equation (44), the latter becomes

$$\begin{aligned}
Mi \, Cp_b \, (T_{ad} - T_{u1}) &= Mb_1 \, Cp_b \, T_{b1} \left\{ (P_2 / P_1)^{\frac{\gamma_b-1}{\gamma_b}} - 1 \right\} \\
&+ Mu_2 \, Cp_u \, T_{u1} \left\{ (P_2 / P_1)^{\frac{\gamma_u-1}{\gamma_u}} - 1 \right\} \\
&+ Mi \, T_{u1} \, \gamma_b / C \, Mb_2 \, (P_2 \, V / T_{u1} - R_u \, (P_2 / P_1)^{\frac{\gamma_u-1}{\gamma_u}} \, Mu_2 \\
&- R_b \, Mb_2) \quad (48)
\end{aligned}$$

Equation (48) is solved iteratively to determine  $P_2$  which enables to calculate  $Tu_2$  and  $Tb_2$  by substituting  $P_2$  back in expressions (41), (42). Therefore, all the thermodynamic properties of the gas at the end of the combustion step are determined, viz:  $P_2$ ,  $Vb_2$ ,  $Vu_2$ ,  $Tb_2$ ,  $Tu_2$ ,  $Mu_2$ ,  $Mb_2$ .

#### 2.2.4.2 Piston movement

The change in the conditions of both gases due to piston movement can be determined as follows. The total amount of piston travel is calculated as function of the crank angle increment selected, and both gases are isentropically compressed during this process. The work produced during the compression/expansion of the burned and unburned gases is equal to the variation of the internal energy of the system. For a closed system

$$Q_{23} = W_{23} + \Delta U \quad (49)$$

Assuming adiabatic process  $Q_{23} = 0$ , hence  $W_{23} = -\Delta U$

$$W_{23} = \int_2^3 p \, dv = C \int_2^3 v^{-\gamma} \, dv$$

$$\text{or } W_{23} = C v^{1-\gamma} / (1-\gamma) = C (v v^{-\gamma}) / (1-\gamma)$$

$$\text{but } W_{23} = p v / (1-\gamma) \Big|_2^3 \text{ therefore}$$

$$W_{23} = (P_3 V_3 - P_2 V_2) / (1-\gamma) \quad (50)$$

The internal energy of the cylinder contents before piston movement is

$$U_2 = Mu_2 C_{v_u} Tu_2 + Mb_2 C_{v_b} Tb_2 \quad (51)$$

and after piston movement is

$$U_3 = M u_3 C_{v_u} T_{u_3} + M b_3 C_{v_b} T_{b_3} \quad (52)$$

but  $W_{23} = - (U_3 - U_2) = U_2 - U_3$

Substituting  $U_2$  and  $U_3$  by the expressions given by (51) and (52) results

$$W_{23} = M u_2 C_{v_u} T_{u_2} (1 - T_{u_3} / T_{u_2}) + M b_2 C_{v_b} T_{b_2} (1 - T_{b_3} / T_{b_2}) \quad (53)$$

but  $T_{u_3} / T_{u_2} = (P_3 / P_2)^{\frac{\gamma_u - 1}{\gamma_u}}$ , and also

$T_{b_3} / T_{b_2} = (P_3 / P_2)^{\frac{\gamma_b - 1}{\gamma_b}}$ , therefore

$$W_{23} = M u_2 C_{v_u} T_{u_2} \left\{ 1 - (P_3 / P_2)^{\frac{\gamma_u - 1}{\gamma_u}} \right\} + M b_2 C_{v_b} T_{b_2} \left\{ 1 - (P_3 / P_2)^{\frac{\gamma_b - 1}{\gamma_b}} \right\} \quad (54)$$

Substituting  $W_{23}$  by its expression (50) and assuming for  $\gamma$  a value given by  $\gamma = (\gamma_u + \gamma_b) / 2$ , results

$$P_3 = 1 / V_3 \left\{ M u_2 C_{v_u} T_{u_2} \left\{ 1 - (P_3 / P_2)^{\frac{\gamma_u - 1}{\gamma_u}} \right\} + M b_2 C_{v_b} T_{b_2} \left\{ 1 - (P_3 / P_2)^{\frac{\gamma_b - 1}{\gamma_b}} \right\} \right\} + P_2 V_2 / V_3 \quad (55)$$

Equation (55) when solved iteratively will give the value of  $P_3$  and consequently the values of  $T_{b_3}$  and  $T_{u_3}$ .

#### 2.2.4.3 Heat transfer

Heat transfer can now be calculated assuming the process to occur at constant volume.

$$Q_b = h_b A_b (T_{b_3} - T_w) Dt \quad (56)$$

$$Q_u = h_u A_u (T_{u_3} - T_w) Dt \quad (57)$$

where  $h_b$  and  $h_u$  are the mean heat transfer coefficients for the burned and unburned zones respectively calculated as indicated in section 2.4.

The final temperatures  $Tb_4$  and  $Tu_4$  are given by

$$Tb_4 = Tb_3 - Qb / Mb_2 Cv \quad (58)$$

$$Tu_4 = Tu_3 - Qu / Mu_2 Cv \quad (59)$$

Again, the equilibrium between the pressure on both sides of the boundary have been disrupted so a new value of the equilibrium pressure must be calculated. From the equation of state

$$P_4 = Ru Tu_4 Mu_2 / Vu_4 = Rb Tb_4 Mb_2 / Vb_4 \quad (60)$$

After rearranging and substituting  $Vu_4$  by

$$Vu_4 = V_3 - Vb_4, \text{ results}$$

$$Vb_4 = (Mb_2 Rb Tb_4 V_3 / Mu_2 Ru Tu_4) / (1 + Mb_2 Rb Tb_4 / Mu_2 Ru Tu_4)$$

Substituting  $Vb_4$  back in equation (60) the value of the final pressure in the combustion chamber is determined.

### 2.3 Turbulence in spark ignition engines

Turbulence has been considered one of the most important factors that affect the engine combustion process. Indeed, in order to determine, for example, heat transfer rates, ignition delay times and minimum ignition energy, it is necessary to know the details of the turbulence structure in the engine.



despite the great difficulties encountered in its quantification, there is a large body of literature that addresses gas flow within the cylinder of a reciprocating engine and attempts have been made to relate such measurements to the engine operating parameters and to the combustion process as illustrated by Semenov (1963), Dent and Salama(1975) and Lancaster(1976) to mention just a few.

The turbulent field in the cylinder of an internal combustion engine is produced by high shear flows during the intake process and/or near the TDC of the compression stroke for engines whose combustion chambers are provided with large squish areas.

In an internal combustion engine, the flow field inside the cylinder is periodic and this periodicity necessitates the use of ensemble averaging techniques to obtain data for analysis of the flow. This ensemble is composed of measurements performed at the same point in the engine cycle for a large number of cycles. When the flow can be considered quasi-steady, conventional time averaging techniques are still useful for determining turbulence quantities from which integral time scales can be determined.

The turbulent flow can be considered as mean flow upon which are superimposed vortices of different size randomly dispersed in the flow. These vortices have finite lifetimes and seem to appear at random times. To fully characterize a turbulent flow it is necessary to know its r.m.s turbulent velocity, also called turbulence intensity, one of the three different scales that specify the average dimension of the random vortices that make up the turbulence, and the mean velocity of the flow upon which the r.m.s turbulent velocity is superimposed.

The instantaneous velocity in the cylinder at any crank

angle theta can be defined as

$$U(\theta) = \bar{U}(\theta) + u(\theta) \quad (61)$$

$$\bar{U}(\theta) = 1 / N \sum_{i=0}^{N-1} U(\theta + i n \pi) \quad (i = 1, 2, \dots, N-1) \quad (62)$$

where  $n$  is the number of strokes in an engine cycle,  $\bar{U}$  is the mean velocity and  $u$  is the fluctuating velocity. The r.m.s turbulent velocity or turbulence intensity can be determined from

$$u'(\theta) = \left\{ 1 / N \sum_{i=0}^{N-1} (U^2(\theta + i n \pi) - \bar{U}^2(\theta)) \right\}^{1/2} \quad (63)$$

The integral length scale is defined as the integral of the autocorrelation coefficient of the fluctuating velocity at two points in the flow separated by a variable distance and is a measure of the large scale structure of the flow field. If the autocorrelation coefficient is defined by a relationship between  $u(x)$  and  $u(\theta)$

$$R_x = 1 / N \sum_{i=0}^{N-1} u(x_0) u(x) / u_0 u_x \quad (64)$$

Then the integral length scale is given by

$$L = \int_0^{\infty} R_x dx \quad (65)$$

Fig 2.1 illustrates the above definitions.

The Taylor micro scale  $\lambda$ , is defined by the second derivative of the correlation coefficient  $R_x$  evaluated at the origin  $x_0$  or  $t_0$ . It is represented by the intercept with the abscissa of the parabola coinciding in level and curvature with the function  $R_x$ , ie

$$\lambda^2 = - 2 / (\partial^2 R_x / \partial x^2) \quad (66)$$

The Taylor micro scale is an indication of the magnitude of the fluctuating strain rate of the turbulent field. It can also be interpreted as a measure of the spacing

between the smallest eddies whose size is defined by the Kolmogorov micro scale.

The Kolmogorov micro scale,  $\eta$  representing the size of the smallest eddies, is related to the rate of turbulence dissipation per unit mass

$$\eta = (\nu^3/\epsilon) \quad (67)$$

where  $\nu$  is the kinematic viscosity of the gas.

The technique used for measuring the integral length scale as defined above, requires the use of two probes and the correlation of a large number of data. Due to difficulty inherent to the application of such a technique, the procedure in which only one probe is used is adopted whenever possible. In such cases the integral time scale turbulence  $L_t$ , is defined as a correlation between two velocities at a fixed point in space but separated in time and indicated by

$$L_t = \int_0^{\infty} R_t \, dt \quad (68)$$

The autocorrelation coefficient  $R_t$  between fluctuations  $u_0$  at the reference time  $t_0$  and  $u_z$  at  $t_0 + z$  is given by

$$R_t = \overline{u_0 u_z} / u_0^2 \quad (69)$$

Similarly, a Taylor micro time scale is defined as

$$\lambda_t^2 = -2 / (\partial^2 R_t / \partial t^2)_{t=t_0} \quad (70)$$

and a Kolmogorov micro time scale, is defined by

$$\eta_t = (\nu/\epsilon)^{1/2} \quad (71)$$

The integral time scale represents the mean transit times of the eddies past a reference point or in cases where

there is no mean flow, it represents the mean life time or mixing time of the energy containing eddies. The micro time scale characterizes the inverse of the frequency of occurrence of the Kolmogorov eddies and  $\eta_t$  is a measure of their life time. A relationship between the two integral scales is given by

$$L = \bar{U} L_t \quad (72)$$

which is valid if the turbulence is homogeneous, the mean flow is constant and  $u' / \bar{U}$  is small compared to 1.

The intake process is responsible for the generation of the highly turbulent field that appears inside the cylinder of an internal combustion engine and which persists throughout the compression and expansion strokes. According to Hoult and Wong(1975) if the bore to stroke ratio is less than or equal to 1, the the vortex generated during intake lasts to BDC and maintains itself during compression and may even be amplified during the compression process. However, if the bore to stroke ratio is greater than 1, the vortex breaks up due to instabilities of the flow and the turbulence decays. They offered no explanation for this behaviour.

As the piston approaches TDC, the turbulence decays rapidly once no energy is being supplied to the flow, as illustrated in Fig 2.2 taken from Groff and Matekunas(1980). The intake stroke basically depend only on the jet flow through the valve to generate the high turbulent field, while compression and expansion strokes are highly dependent on the combustion chamber geometry. For disc chambers with no swirl or squish, the dissipation process generally continues as the piston moves to BDC, but the rate of dissipation is reduced near BDC.

Although the flow field in the cylinder is highly turbulent, the averaged data over a large number of cycles

is extremely repeatable, which implies that models of the turbulent combustion process representing the averaged process should present good agreement with engine performance data.

Near the TDC the flow field is highly dependent upon combustion chamber shape and inlet swirl. For combustion chambers having no squish areas eg, disc chambers, the flow field near TDC, when averaged over many cycles appears to be homogeneous and isotropic, whilst during the intake process the same flow is highly anisotropic and the mean velocity and the turbulence intensity is a function of the position inside the combustion space.

For combustion chambers having large squish areas ( $> 30\%$ ) or high swirl rates, the turbulence generated is greater than that produced by piston motion. Engine variables such as engine speed, compression ratio, load and intake pressure are known to affect the turbulence quantities in a different manner.

The most known result is that the mean velocity and the r.m.s turbulent velocity increases linearly with engine speed. As the rate of straining of the flow field increases with the turbulence intensity, more energy is dissipated, the Reynolds number based on the Taylor micro scale increases and therefore the dissipative eddies are also reduced.

Therefore, as engine speed increases, the length scales tend to be reduced, as indicated in Fig 2.3 taken from Smith(1982), which represent microshadowgraphs of flame propagation showing increased wrinkling and so reduced length scale, as the engine speed is increased.

If the flow field is in the relaxed regime, where the structure of turbulence is governed by the chamber dimensions, the integral length scale should be of the

order of chamber height and should not change with engine speed. In a relaxed flow field the integral length scale is expected to vary inversely with the compression ratio. Such a variation was demonstrated by Dent and Salama(1975) who reported a decrease in the Taylor micro scale with an increase in compression ratio.

Another engine variable of importance is the inlet pressure, directly connected with the volumetric efficiency. If the intake pressure is increased, for example, through the use of a supercharger, the density of the charge is expected to increase, provided the inlet temperature is controlled and therefore, the flow through the inlet valve increases, thereby increasing the volumetric efficiency. This essentially increases the amount of kinetic energy supplied to the flow and it can then be expected that the turbulence intensity be enhanced. In this manner the turbulence intensity can increase with load and length scales are expected to show a decrease with volumetric efficiency.

### 2.3.1 Turbulent combustion

The initial phase of the turbulent combustion process is highly dependent on the local turbulent field in the vicinity of the spark plug at the instant of ignition. This initial phase lasts until several eddies of the size of the combustion chamber height are fully consumed. It is during this initial phase of the combustion process that appears much of the cyclic variation, so peculiar to spark ignition engines. After this initial phase the flame front accelerates rapidly, entering the fully developed turbulent phase where most of the charge is burned.

Several experimental studies of turbulent flame propagation in engines have been carried out which have helped to improve the present understanding of the global



features of engine combustion such as the behaviour of turbulent burning velocities, the duration of combustion and ignition delay.

However, only recently, with the ever increasing use of laser instrumentation to measure the properties of the flow inside the combustion chamber of an internal combustion engine, have experimental studies addressed the question of flame structure in engines and their influence upon the flow field, as reported by Witze et al(1984). According with the authors, the accurate measurements of turbulence intensity during combustion are difficult to perform due to cyclic variation of the combustion process and the uncertainties in the distinction between large scale turbulence and cyclic variations in the fluid motion.

Their measurements suggest an anisotropic enhancement of the turbulence ahead of the flame front with the change in turbulence in the direction of flame propagation being twice that of the component parallel to the flame surface. In the unburned gas region the data from the conditional sampling suggested also an increase in the turbulence intensity. The findings of Witze et al(1984) are in agreement with the Rapid Distortion theory proposed by Wong and Hoult(1979), according to whom the turbulence intensity can increase in certain directions due to rapid compression. However, the rapid rise in temperature of the unburned gases increases the fluid viscosity which influences directly the rate of decay.

Further to the investigations specifically related to engines, there have been several studies recently of turbulent V-shaped and burner-type premixed flames such as that by Moss(1980), which have sought to improve our understanding of the structure of turbulent premixed flames in general.

This increase in information on turbulent flames, obtained experimentally has brought advances in theoretical understanding of flames and offers the prospect of improvements in capabilities to predict them. Most of the theoretical work related to engines has however assumed simplified structures for turbulent flames. Nevertheless, such results should prove useful in improving our knowledge of the interaction between turbulence and combustion.

Models of premixed charge engine combustion usually rely on the fact that an average burning velocity is the flame parameter that most influences engine pressure history. Once this burning velocity is known, then with suitable approximations the flame front position can be computed. In particular, with curve fitting for the parametric dependence for velocity and assuming spherical flame propagation, zero-dimensional models can often reproduce measured pressures with considerable accuracy provided the flame front has negligible thickness. This problem will be addressed later in this work.

Apart from the turbulence parameters already presented , some factors such as departures from isotropy, homogeneity and equilibrium may also be of some importance in engine applications but given the limited parametric information concerning their effects they will not be considered here.

The most direct way in which turbulence affects the combustion process is by its action upon the flame propagation velocity. A flame is a rapid self-sustained propagating chemical reaction which liberates heat and occurs in a discrete reaction zone. In a non turbulent reactant mixture, flame propagation is laminar, presenting a smooth surface and a thin reaction zone of less than one millimeter thickness.

When turbulence is present, the time-averaged flame front

reaction zone is significantly thicker and the flame speed may be several times the laminar value depending, inter alia, on the intensity of turbulence.

For a flame front expanding spherically from the central ignition point, the laminar flame could be visualized as the smooth curve on the left of Fig 2.4, whilst the turbulent flame would present a surface which is highly wrinkled.

Different models have been proposed to explain the increase in flame speed due to combustion. They are classified into two categories, one based on molecular transport and the other on turbulent transport processes.

The best known of the molecular transport models was formulated by Damkohler(1947) and is known as the wrinkled laminar flame model. He realized that the mechanism by which turbulence affects flame propagation depends upon whether the scale of turbulence is small or large as compared to the thickness of the reaction zone. The model considered the turbulent combustion as the propagation of a laminar flame possessing a constant normal velocity over the whole surface that separates burned from unburned gases ie, the reaction zone. Therefore, a turbulent flame could be thought of as a laminar flame having an increased surface area as a consequence of the action of small scale turbulence.

However, all authors who use the concept of laminar model consider the development of the combustion surface under the action of large scale turbulence as the basic factor responsible for the acceleration of the combustion process but assuming a constant normal flame propagation velocity. The role of the small scale turbulence in this case is to increase the transport processes within the reaction zone, thereby increasing the local rate of flame propagation.

Damkohler(1947) expressed the turbulent burning velocity by

$$u_t / u_l = 1 + u' / u_l \quad (73)$$

There is a wealth of experimental data to support the wrinkled flame model and the model is claimed to be applicable to more intensely turbulent flames. The wrinkled flame front model has been adopted by several researchers since then, among them Schelkin(1947), Karlovitz(1953), Scurlok(1953), Sokolik(1963), and Kosachenko(1965). For further details and mathematical expressions the interested reader should address the review by Andrews and Bradley(1975).

More recent models such as the eddy entrainment theory put forward by Blizard and Keck(1974) are in fact an extension of Damkohler(1947) work on the small scale turbulence theory. They modeled the turbulent flame propagation as the entrainment and subsequent laminar burn up of discrete coherent, turbulent eddies. The combustion was considered as the propagation of a flame front having finite thickness through the unburned mixture at a speed determined by the rate at which eddies are ignited due to diffuse transport of radicals such as H, OH, and O between adjacent eddies.

These eddies are then assumed to burn at the laminar burning velocity in a characteristic time

$$\tau = L / u_l \quad (74)$$

where L is the eddy radius, function of engine speed, geometry and size of the combustion chamber. The weaknesses of this model are the assumption of a constant entrainment velocity over the entire combustion period and the constant characteristic time to burn an eddy. Entrainment velocity is dependent on the turbulence intensity whereas laminar burning velocity involved in the

calculation of the characteristic time is not constant under cylinder conditions.

A further improvement was given by Tabaczynski et al(1977) who postulated the eddy burning time to be dependent of eddy size, laminar burning velocity and turbulence intensity, therefore eliminating most of the Blizzard and Keck(1974) model drawbacks. It is worth to notice that Tabaczynski(1977) assumed that turbulence intensity increased during the compression while experiments performed by Lancaster et al(1976) revealed that turbulence level decreases as the gases are compressed. Both models are not yet fully developed because of unanswered questions related to the uniformity of the turbulent field in complex combustion chamber shapes and turbulence levels in the burned gases. In addition some study about the aerodynamics of intake and compression processes need to be extended so as to prescribe as functions of engine geometry, the flow conditions at which combustion is initiated.

Daneshiar and Sahota(1985) examining the propagation of flames inside the combustion chamber through a quartz window observed such an intense turbulence suggesting that the propagation rate and the thickness of the combustion zone do not follow the pattern of a wrinkled laminar flame. The combustion zone is much more thick than would be expected from a wrinkled laminar flame.

The thick combustion zone was also described by Ballal and Lefebvre(1975) when high levels of turbulence are present. They concluded that the combustion process cannot be interpreted as being confined within well defined boundaries near a contiguous flame surface. The process is more likely to occur 'in depth', whereby eddies of the unburned gas are consumed during their passage through the reaction zone.

This point of view seems to be shared also by Gatowski et al(1984) after observation of flames propagating inside the cylinders of spark ignition engines. To illustrate their point of view, Ballal and Lefebvre(1975) produced the photographs shown in Fig 2.5. The first is characteristic of the low turbulence process, producing a thin wrinkled flame while the second shows a much thicker combustion zone due to the higher turbulence. The flames are produced by the combustion of premixed propane and air. It can be noticed that the characteristic length scales are dramatically changed from the first to the second case, according to the increase in turbulence

Experiments with turbulent flames gave evidence of the existence of spatially non-uniform distributions of viscous dissipation, which induced Tennekes(1968) to formulate a theory whereby he suggested that the small scale structure of turbulence may be modelled as constituted of small vortex tubes of diameter  $\eta$ , the Kolmogorov micro scale. These vortex tubes are stretched by eddies of size  $\lambda$ , the Taylor micro scale. The model revealed to be consistent with what is known about viscous dissipation in locally isotropic turbulence and is supported by experimental evidence as was demonstrated by Narayanan et al(1977).

### 2.3.2 Laminar burning velocity correlation

The laminar burning velocity is a well defined characteristic of a combustible fuel-air mixture. It represents the speed at which a one dimensional laminar flame propagates into the stationary unburned gas under adiabatic conditions. It depends explicitly on the pressure, temperature and composition of the unburned gas.

As can be seen from Fig 2.6, Fig 2.7 and Fig 2.8, the



laminar burning velocity shows a maximum for slightly richer than stoichiometric mixtures and is a strong function of unburned gas temperature and a weak function of pressure. It also decreases linearly with the residual gas fraction.

Laminar burning velocities may be determined experimentally for each fuel according to various methods.

#### 2.3.2.1 Conical stationary flames on cylindrical tubes and nozzles

The cylindrical tube burner can be used to determine the burning velocity by the Gouy's flame area method after Gouy(1879). The method is recommended for preliminary research on unknown mixtures and is claimed to give a burning velocity within 20 %. With this method, the flame is photographed and its area calculated dividing the cone into a number of sections using a planimeter. The main source of errors with this method is due to the fact that the tip of the flame is rounded instead of being pointed, but the error here is not very great.

The main source of error is due to the reduction of the velocity at the base of the flame due to heat loss and removal of active species at the burner wall. The base itself is difficult to locate because the luminosity dies out gradually as the dead space is approached. An alternative to ameliorate the difficulties pointed out was through the use of a nozzle, to give a uniform velocity profile. This improves the definition at the base of the flame so that it can be located more easily. Other advantage is that with nozzles higher Reynolds numbers can be attained and as a consequence, larger flames can be used without problems of turbulence. The main disadvantage of the use of nozzles is that the gas flows from a wide tube into a relatively narrow nozzle and explosions may occur if the flame strikes back into the

wider part, what is difficult to prevent in practice.

#### 2.3.2.2 Soap bubble method (spherically expanding flames)

Through this method, a combustible mixture is blown into a bubble and ignited centrally. As the surface of the bubble expands freely as the explosion proceeds, the combustion occurs at constant pressure. The velocity measured is a spatial velocity, the expansion of the burned gases causing a bodily movement of the gases through which the flame front is progressing.

The method gives directly the velocity of the propagating flame, corrected only for the expansion and should therefore give accurate results. The greatest experimental limitation is the determination of the initial and final size of the bubble which is very important to determine accurately because the expansion ratio appears in the calculation as the cube. This expansion ratio is defined as  $D^3 / d^3$  where  $D$  and  $d$  are the final and initial values for the diameter of the bubble respectively, which are determined by means of high speed cine photographs.

#### 2.3.2.3 Constant volume combustion in spherical bombs

The method is similar to the soap bubble method, but the bomb is filled with the combustible mixture and instead of determining the expansion ratio from the final size of the bubble, simultaneous records are made of the size of the spherical shell of the burned gases and of the pressure inside the bomb. The method is of great value for studying effects of pressure on burning velocity and for substances which are available only in small quantities.

Bradley and Hundy(1971) have improved the method by using a hot wire anemometer to measure the gas flow ahead of the

flame front. They also used a laser interferometer system with high speed cine recording to study the flame front itself. Closed vessel techniques, accompanied by density corrections also offer the prospect of accurate measurement of laminar burning velocity.

In the present work, laminar burning velocity has been calculated using empirical correlations which are functions of mixture strength, unburned mixture temperature, pressure and residual gas fraction developed by Gulder(1984). Two different fuels were considered in the present application namely iso-octane and ethanol.

The correlation has the form

$$Su(\phi, P, T) = Su_0(\phi) [T_u / T_0]^\alpha [P / P_0]^\beta \quad (75)$$

where  $Su(\phi)$  is the velocity measured at  $T_u = T_0$  and  $P = P_0$  for a given equivalence ratio  $\phi$  and  $\alpha$  and  $\beta$  are constants which may also be weakly mixture dependent.

Previous workers like Andrews and Bradley(1972), Metghalchi and Keck(1980), Metghalchi and Keck(1982) and Gulder(1983), have successfully adopted similar correlations.

The laminar burning velocity at room temperature  $Su_0$  for pure ethanol, hydrated ethanol and iso-octane as a function of mixture strength, unburned mixture temperature and pressure are given by the expression

$$Su_0 = z w \phi'^m \exp \{-d(\phi - 1.075)^2\} \quad (76)$$

Values of the constants that appear in the above equation are taken from Gulder(1984) and are as follows

| Fuel          | z   | w (m/s) | n      | d    |
|---------------|-----|---------|--------|------|
| iso-octane    | 1.0 | 0.4658  | -0.326 | 4.48 |
| pure ethanol  | 1.0 | 0.4650  | 0.250  | 6.34 |
| hydr. ethanol | *   | 0.4650  | 0.250  | 6.34 |

$$* z = 1.0 - 3.25 C_w$$

where  $C_w$  is the mole fraction of water in a stoichiometric fuel-air mixture. Fig 2.6 and Fig 2.7 illustrate the variation of laminar burning velocity with equivalence ratio for ethanol-air and iso-octane-air mixtures respectively. Fig 2.8 indicates the effect of water concentration on burning velocity of ethanol-air mixtures.

For the pressure dependence of the laminar burning velocity, a power law expression of the form

$$Su(\phi, P) = Su_0(\phi) (P / P_0)^\beta \quad (77)$$

has been used, where the values of the exponent are taken to be -0.22 for iso-octane-air mixtures. For ethanol, two cases must be considered, depending upon whether the equivalence ratio  $\phi$  is greater than or equal to one or less than one. In the first case ( $\phi \geq 1$ ),  $\beta = -0.17 \phi$  while in the second case ( $\phi < 1$ )  $\beta = -0.17 / \phi$ . For blends of ethanol and water, regardless of variations in equivalence ratio,  $\beta = 0.28 C_w - 0.17$ .

For the temperature dependence of the laminar burning velocity, a similar power law expression is adopted of the form

$$Su(\phi, T) = Su_0(\phi) (T_u / T_0)^\alpha \quad (78)$$

where the values of the temperature exponent  $\alpha$  are taken as 1.56 for iso-octane and 1.75 for pure hydrated ethanol, according to Gulder(1984). Fig 2.9 and Fig 2.10 indicate

the variation of the burning velocity of iso-octane-air and ethanol-water-air mixtures with pressure. Fig 2.11 indicates the temperature dependence of the burning velocities of ethanol-air and ethanol-water-air mixtures, while Fig 2.12 refers to the temperature dependence of the laminar burning velocity of the iso-octane-air mixtures.

### 2.3.3 Turbulent burning velocity correlation

Many turbulent flame propagation theories consider the turbulent burning velocity dependent upon the laminar burning velocity and turbulence intensity according to the general expression

$$u_t = A u_l + B u' \quad (79)$$

Mattavi, Groff and Matekunas(1979), Groff and Matekunas(1980) and Mathur et al (1983) also made use of such a functional dependence. Abdel-Gayed and Bradley(1976) performed experiments aiming to correlate out-of-engine steady flow measurements of flame speed to various turbulence parameters and the laminar burning velocity.

The results of such studies emphasize the importance of laminar burning velocity in the interpretation of turbulence burning velocity and turbulence intensity interactions. Other theoretical models for premixed turbulent combustion have been proposed and they are summarized in a paper by Andrews et al(1975)a. Due to the variety of forms presented by the equations proposed by distinct researchers, experimental data should be used to evaluate the adequacy of each equation. Andrews et al(1975)b, supplemented a body of existing data with some of their own and concluded that the turbulent Reynolds number based on the Taylor micro scale was a good parameter to correlate the ratio of the turbulent to laminar burning velocity, denominated flame speed ratio.

Further study by Bradley et al(1976) showed that a correlation between the ratio of the turbulent to laminar burning velocity and the ratio of the turbulence intensity to the laminar burning velocity gives better results.

Length scales are difficult to infer accurately from engine turbulence measurements. Generally they are estimated to fall in the range 3 to 12 mm, yielding turbulent Reynolds number based on the integral scale of the order of 1500 to 4000, according to data provided by Mattavi, Groff and Matekunas(1979).

In the present work, integral length scale was estimated by making use of the ratio of the combustion chamber volume to the combustion chamber surface area

$$L = V_{cc} / A_{cc}$$

which will give the following final expression

$$L = 0.5 D^2 h / (D^2 + 2 D h) \quad (80)$$

For a cylindrical combustion chamber with the piston near TDC, the above expression estimates for the integral length scale a value that is roughly equal to the clearance height  $h$ , whereas near BDC the value of the integral length scale will be roughly equal to the bore. The advantage of using an expression such as (80) is that integral length scale is made variable during the combustion process as the piston change its position.

Ballal and Lefebvre(1974) presented studies regarding integral scale effects using steady flow burner experiment. They concluded that flame speed ratio tends to increase with scale under conditions of low turbulence, but decreases with scale for conditions of high turbulence, the separation between the two regions presenting a value of  $u' / u_{\ell}$  equal to 2.



For  $u' / u_L$  less than 2, they reported integral scales of less than 1 mm which makes comparisons with engine data difficult specially since the results show an increase in FSR with integral scale in this region. Further complication arises when comparing engine data with out-of-engine data because of the interpretation of the turbulence intensity value used in correlating the data.

In engine experiments, the turbulence intensity is normally measured approximately at the time of ignition, while the turbulent burning velocity is calculated for a fully developed flame ie, with a radius greater than 30 mm. For a fully developed flame, using linear curve fitting to the experimental data forced through the theoretical intercept of unity  $u_L / u_L$  at  $u' / u_L$  equal to zero, Mattavi, Groff and Matekunas(1979) derived the following expression

$$u_L / u_L = 1 + 4.01 (u' / u_L) \quad (81)$$

During the period of flame development the flame speed ratio can be corrected by multiplying the last term of the above expression by

$$C = (R_f / R_{fm})^m \quad (82)$$

Where  $R_{fm}$  is the radius for a fully developed flame assumed to be 30 mm. The coefficient  $m$ , determined by Lancaster(1976) measuring turbulence in a CFR engine, falls in the range between  $m=5/3$  and  $m=2$ . Therefore, for a flame radius less than 30 mm the expresion used was

$$u_L / u_L = 1 + 4.01 (u' / u_L) C \quad (83)$$

with  $m = 2$  giving an exponent of 0.5.

#### 2.3.4 Turbulence intensity

The turbulent flow can be understood as mean flow upon which are superimposed vortices of different sizes randomly dispersed in the flow. These vortices appear to be born at random times and have a finite life time. They also can assume random orientations.

In internal combustion engines, the turbulence characteristics of flows have been measured using hot wire anemometry and nowadays the preferred technique of laser doppler velocimetry, LDV. A detailed comparison between the two techniques is presented by Witze (1980).

The direct measurement of turbulence intensity lies outside the scope of the present work. Therefore, the approach adopted was to use an interpolation process upon the available data produced by Groff and Matekunas (1980).

Two simplifying assumptions are made. Firstly, that the turbulence intensity continues its decay during the combustion process and secondly, the representative turbulence intensity varies linearly with engine speed. Fig 2.2 and Fig 2.13 illustrate the position adopted.

The data used was obtained by the authors from several experiments with a spark ignition engine operating in a motored mode. The range of speeds varied from 500 to 2000 rpm and spark timing varied from 0 to 30 DCA BTDC. The data was then nondimensionalized by the piston speed. It cannot be expected that such a correlation is valid for all types of engines. However, the curve can perhaps be adusted by means of a correction factor determined experimentally in order to obtain a more suitable curve for a specific engine geometry.

The relationship between turbulence intensity and piston velocity is presented by Liou et al (1984) using results

of measurements of flow near the TDC found by several investigators using motored engines. These results cover a wide range of engine configurations including engines with and without swirl. It broadly confirms that the turbulence intensity does depend significantly on the piston velocity.

However, there are significant differences from engine to engine at the same piston speed, which are caused partly by differences in the engine design and partly because it is not possible to characterize a particular flow regime by measuring just one velocity component at a single point. Even in the same engine, with and without swirl, turbulent velocity can be increased by swirling the gas.

For the range of speeds used in the present work, 600, 900, and 1200 rpm the mean piston speed  $V_p$  for the Ricardo E6 was 2.22, 3.33 and 4.45 m/s respectively. The engine used by Groff and Matkunas (1980) had mean piston speeds 46 % lower than these presented by the Ricardo E6. Consequently, other factors being equal, corresponding higher turbulence could be expected in this engine.

However, it has been reported by Lancaster (1976), measuring turbulence in a motored CFR engine that the piston-generated velocities are an insignificant contribution to the turbulence intensity in a disc chamber. The Ricardo E6 was operating with compression ratios of 10 and 12 to 1 while the engine used by the above researchers was operating with a compression ratio of 6.85 to 1. Despite such difference no significant variation in turbulence levels would be expected according to observations made by Lancaster (1976) on the variation of turbulence with compression ratio.

Valve lift appears to have the most important effect upon the generation of turbulence in the internal combustion engine, due to the effect of jet flow through the intake valve. Therefore, by reducing valve size and/or valve lift

and consequently valve flow area, induced turbulence can be increased. Normally, valve lift is reduced when valve area is increased. The Ricardo E6 engine has a lift nearly double that of the engine used in the experiments reported by Groff and Matekunas (1980) and a diameter approximately equal. Therefore, it could be expected to find lower levels of turbulence in the Ricardo E6.

The criterium adopted to match the turbulence correlation to the Ricardo E6 was based on the fact that the turbulence intensity near TDC can be estimated as

$$u' < 0.5 V_p \quad (84)$$

as suggested by Liou et al (1984). The curve was then adjusted by displacing it until the value corresponding to the TDC agreed with that determined by expression (84). In this way, varying the velocity of the engine the curve representing turbulence intensity versus time in DCA, displace up or down according to whether the velocity is increased or decreased, thus permitting to estimate the turbulence intensity from 200 DCA to 460 DCA.

It seems perfectly reasonable to adopt such an approximation given the uncertainties regarding the turbulence intensity. As pointed out by Liou et al (1984) measured values of turbulence intensity are scattered by a factor of 5 even under similar operating conditions as a consequence of adopting different techniques in measurements and also due to differences in the procedures adopted to process the data.

### 2.3.5 Adiabatic flame temperature

In the design of any type of combustion system and in particular the spark ignition engine, it is desirable to estimate the maximum flame temperature that products can

reach during the combustion process.

The maximum temperature that products could achieve will occur if the process is adiabatic, otherwise any heat produced during the combustion process tend to reduce the internal energy of the products thereby reducing their temperature. The maximum temperature for a given reaction is called the adiabatic flame temperature.

For a combustion process occurring at constant pressure application of the first law leads to

$$\Delta H = Q$$

that is, the enthalpy of the products is equal to the enthalpy of the reactants plus any heat transferred to the system. Therefore, given the heat transferred per unit mass the adiabatic flame temperature can be found by an iterative procedure.

The process adopted in the present work consists in evaluating the enthalpy of the reactants at a given condition of pressure, temperature, residual gas fraction and equivalence ratio. Next, the enthalpy of the products is evaluated started from a guessed temperature and the enthalpy compared with that of the reactants. If they do not agree a new estimate is made and the process repeated until agreement is reached within a specified tolerance, according to

$$T_{n+1} = T_n + (H_1 - H_2) / C_p \quad (85)$$

## 2.4 Heat transfer

A detailed knowledge of the science of heat transfer associate with the ability to analyze,

quantitatively, problems involving the transfer of heat have becoming increasingly important as modern technology has become more and more complex.

Mechanical and chemical engineers are particular concerned with problems of heat transfer. Modern power generation, in particular in internal combustion engines, involves the production of work from a combustible fuel through a combustion process taking place in the combustion chamber of the engine. A fraction of the heat generated by the combustion process is transferred from the hot products of combustion, through the walls of the combustion chamber, to the cooling medium and dissipated in the atmosphere.

There are three processes through which heat can be conveyed from the hot gases to the cooling medium through the combustion chamber walls, namely

- .conduction
- .convection
- .radiation

Heat conduction is the term applied to the mechanism whereby energy is transferred from one body to another, or from part of the body to another part, by the exchange of the kinetic energy of the molecules by direct contact or by the drift of free electrons as in the case of heat transfer in metals. The flow heat passes from the higher energised molecules to the lower energised ones.

The mechanism by which heat is transferred from a fluid to a solid wall or by a fluid to another fluid due to gross movement of the mass of fluid is called convection. If the fluid motion is caused by mechanical external means, for example a fan, the process is called forced convection whilst if the movement is caused by differences in density, which are due to temperature gradients in the bulk of the fluid, the process is termed natural convection.



In a fluid, it is virtually impossible to observe pure conduction because as soon as a temperature gradient is imposed on the fluid, natural convection currents will occur as a result of density differences. The basic laws of heat conduction must be coupled with those of fluid motion in order to describe, mathematically, the process of heat convection.

Thermal radiation is the term employed to describe the electromagnetic radiation which has been observed to be emitted at the surface of a body which has been thermally excited. This electromagnetic radiation, when it strikes another body may be partly absorbed, partly transmitted and partly reflected. This process of heat transfer is totally different from the other two previously described in the sense that it no longer necessitates a medium of transport such as air, for example.

For internal combustion engine application, the most important process of heat transfer is that performed under forced convection and therefore is the process that will be considered in detail here. Radiation has some importance, for example, when dealing with heat transfer from the hot products of combustion in a diesel engine to the walls of the combustion chamber but this aspect can be neglected in spark ignition engines. The reason is that in a diesel engine, the combustion process produces relatively large amounts of soot, which is responsible for the process of heat radiation whilst the combustion in the spark ignition engine can be regarded as soot-free unless operating with very rich mixtures which is totally undesirable and must be avoided.

When a fluid, in particular a gas, flows along the surface of a solid body, individual molecules of gas collide with the solid surface and thus heat is conveyed from the surface to the gas or vice-versa, depending upon their relative temperatures. The time scale of the heat exchange

process depends on the type of flow around the surface of the solid body.

Due to the viscosity of the gas the velocity of the flow is not equal throughout the whole cross section of the stream but rather decreases in the direction towards the solid surface becoming zero at the interface. Since the region in which the retarding effect of the gas viscosity plays a dominant role will often be a thin layer near the solid surface, the description of the convection process can be simplified by introducing the concept of velocity boundary layer.

The velocity boundary layer can be defined as the thin layer near the solid surface in which the viscous effect is important. Within this region the effect of the solid surface in the motion of the gas is significant and outside the boundary layer the effect of the wall may be neglected. Outside the boundary layer the gas is assumed to be inviscid whereas inside the boundary layer the viscous flow may be either laminar or turbulent. For engine heat transfer processes only the turbulent flow has interest.

In the event that the gas motion in boundary layer is turbulent, the mean flow is essentially parallel to the surface but it has superimposed upon it a fluctuating motion, in directions both normal and parallel to the solid surface. The effect of the transverse fluctuations is to cause some additional gas mixing, therefore increasing the rate at which heat is transported in the direction perpendicular to the surface.

Due to differences in temperature between the solid surface and the gas, a variation in the temperature of the gas is observed which is similar to the velocity variation described above and a thermal boundary layer may then be defined as the region between the surface and a point at

which the gas temperature has reached a certain percentage of the temperature of the bulk gas.

Generally, the thermal boundary layer does not coincide with the velocity boundary layer although it is certainly dependent on it ie, the velocity boundary layer thickness, the variation of the velocity whether the flow is laminar or turbulent, etc are all factors which determine the temperature variation in the thermal boundary layer.

From the above discussion it is clearly apparent that the prediction of the rates at which heat is convected from the hot gas to the surface involves a thorough understanding of the principles of heat conduction, fluid dynamics and boundary layer theory. All the complexities involved in such an analytical approach may be put together in terms of a single parameter by introduction of Newton's law of cooling

$$Q = A h (T_g - T_w) \quad (86)$$

The quantity  $h$  is known as the mean heat transfer coefficient, HTC or unity thermal conductance and is a complex function of the fluid composition, the geometric properties of the solid surface and the hydrodynamics of the gas flow past the surface. It is worth to mention that  $h$  is not a property of the material as is for example the thermal conductivity. In chapter 1 was presented a review of the different approaches given by some researchers as a means to estimate the mean heat transfer coefficient.

In the present work a correlation for the calculation of the HTC heat transfer based on the transport properties of gas mixtures is presented. The basic correlation is that due to Woschni (1967), which states that

$$Nu = h D / \lambda = 0.035 Re^{0.75} \quad (87)$$

where  $Nu$  is a Nusselt number and  $Re$  is a Reynolds number. Fig 2.14 taken from Taylor (1977) shows the dependence between Reynolds and Nusselt number for experiments performed by several researchers utilising production engines ranging from automobile, truck, aircraft and locomotive engines, diesel and otto cycles, two and four stroke cycle.

The characteristic length on which both numbers are based was arbitrarily chosen to be the cylinder bore. The reason to take the cylinder bore is that the characteristic length ought to increase with the size of the engine and as a matter of convenience. Valve lift could also have been chosen, for example, but they can be varied independently of the bore which is not satisfactory as far as the choice of a characteristic length is concerned. The thermal conductivity is represented by  $k$  and  $h$  is the mean heat transfer coefficient to be determined.

In order to evaluate the Reynolds number, the velocity of the gas inside the combustion chamber must be calculated. The correlation used to calculate the velocity of the gas was taken as

$$U_g = 2.28 V_p + 0.00324 V T_o / V_o P_o (P - P_m) \quad (88)$$

The expression assumes that the gas velocity is proportional to the mean piston velocity but introduces a factor to account for gas motion induced by combustion. The gas velocity induced by combustion is assumed to be proportional to the pressure rise due to combustion, with the subscript  $o$  denoting some reference condition here taken by convenience, as the inlet valve closing IVC.  $P_m$ , the motored pressure at the same point of the cycle, was estimated by use of the isentropic relation

$$P V^\gamma = P_o V_o^\gamma = \text{constant}$$

The constants in Woschni (1967) correlation were determined by matching experimental results obtained from a particular diesel engine, therefore when applied to any other engine the constants are estimates, at the best, and an adjustment may be necessary to better suit a particular engine.

The Reynolds number is estimated by

$$Re = \rho U_g D / \mu \quad (89)$$

The method employed to evaluate the coefficient of viscosity is supported by the kinetic theory of gases. This theory permits the accurate computation of the thermal conductivity and the viscosity of gases according to Hirschfelder, Curtiss and Bird (1954) and is the approach utilised in the present work.

The coefficient of viscosity of a pure component is given by

$$\mu = 26.963 \cdot 10^{-6} (M T / \sigma^2 \Omega) \quad (90)$$

The reduced collision integral  $\Omega$  is dependent upon the intermolecular forces of the gas molecules and therefore it is necessary to know the potential energy of interaction of the colliding molecules in order to evaluate this integral. For the present work, The Lennard-Jones model for the potential was assumed. According to this model, the potential is given by

$$\phi(r) = 4 \epsilon [ (\sigma/r)^{12} - (\sigma/r)^6 ] \quad (91)$$

The model assumes that molecules are rigid and impenetrable spheres surrounded by an inverse-power attractive force. In this way, molecules tend to attract one another when they are far apart and exert strong repulsive forces upon one another when they are close

together. In the expression above  $\phi(r)$  is the potential energy of interaction,  $r$  the intermolecular separation of the colliding molecules and  $\epsilon$  is the minimum value of  $\phi(r)$  as in Fig 2.15.

Collision integrals for this potential were obtained from Hirschfelder, Curtiss and Bird (1954) where they were tabulated as a function of the reduced temperature  $k T / \epsilon$  and are reproduced below for the species of interest.

| Species          | $\sigma(\text{\AA})$ | $\epsilon / k \text{ (K)}$ |
|------------------|----------------------|----------------------------|
| CO <sub>2</sub>  | 3.941                | 195.2                      |
| H <sub>2</sub> O | 2.641                | 809.1                      |
| N <sub>2</sub>   | 3.798                | 71.4                       |
| O <sub>2</sub>   | 3.467                | 106.7                      |
| CO               | 3.690                | 91.7                       |
| H <sub>2</sub>   | 2.827                | 59.7                       |
| H                | 2.708                | 37.0                       |
| O                | 3.050                | 106.7                      |
| OH               | 3.147                | 79.8                       |
| NO               | 3.492                | 116.7                      |
| Ar               | 3.542                | 93.3                       |

The reduced collision integral can finally be calculated from the expression due to Hattikudur and Thodos (1979)

$$\Omega = 1.155(T^*)^{0.1462} + 0.3945 \exp(-0.6672 T^*) + 2.05 \exp(-2.168 T^*) \quad (92)$$

where  $T^*$  is the reduced temperature given by expression (93) where  $K$  is the Boltzman constant.

$$T^* = K T / \epsilon \quad (93)$$

Thermal conductivity in the gas was calculated using the



following expressions taken from Svehla (1962)

$$(\lambda' + \lambda'') 10^6 = R / M^{1.32} [15 / 4 + (C_p / R - 5 / 2)] (\eta \cdot 10^6) \quad (94)$$

$$\lambda' 10^6 = 15 R / 4 M \eta 10^6 \quad (95)$$

$$\lambda'' 10^6 = 1.32 R / M (C_p / R - 5 / 2) (\eta 10^6) \quad (96)$$

where  $\lambda'$  is the translational thermal conductivity,  $\lambda''$  is the internal thermal conductivity and  $\lambda$  is the total thermal conductivity. All the above exposed is valid for a single component gas. If a mixture of gases is considered as in the case of the reactants or the products mixture the following expressions should be considered for the viscosity of a mixture of 11 species listed above

$$\mu = \sum_{i=1}^{11} \frac{\mu_i}{1 + \sum_{j=1}^{11} \beta_{ij} \frac{y_j}{y_i}} \quad j \neq i$$

$$\lambda' = \sum_{i=1}^{11} \frac{\lambda'_i}{1 + \sum_{j=1}^{11} \psi_{ij} \frac{y_j}{y_i}} \quad j \neq i$$

$$\lambda'' = \sum_{i=1}^{11} \frac{\lambda''_i}{1 + \sum_{j=1}^{11} \beta_{ij} \frac{y_j}{y_i}} \quad j \neq i$$

$$\beta_{ij} = \frac{[1 + (\mu_i / \mu_j)^{1/2} (M_j / M_i)^{1/4}]^{1/2}}{2 \sqrt{2} (1 + M_i / M_j)^{1/2}}$$

$$\psi_{ij} = \left[ 1 + 2.41 \frac{(M_i - M_j)(M_i - 0.142 M_j)}{(M_i + M_j)^2} \right] \beta_{ij}$$

$$\lambda = (\lambda' + \lambda'') C_p / C_{pF}$$

where  $C_p$  is the equilibrium specific heat and  $C_{pf}$  is the frozen specific heat. The symbols for the above expressions can be found in the nomenclature.

The mean heat transfer coefficient can now be evaluated by substituting the values for the thermal conductivity  $\lambda$  in the Nusselt number.

$$h = \lambda \text{ Nu} / D \quad (97)$$

To evaluate the amount of heat that is transferred from the hot gases to the cooling medium through the combustion chamber walls, the areas involved in the heat transfer process and the corresponding temperatures must be known. Also a slightly different approach must be considered when the gases are constituted either by the reactants or the products mixture or even when the flame is present or not.

In the first case, when there is no flame present as in the compression and expansion, the areas for the heat transfer were evaluated as follows

$$A_i = \pi D_i^2 / 4 \quad (98)$$

$$A_e = \pi D_e^2 / 4 \quad (99)$$

$$A_c = \pi D S \quad (100)$$

$$A_p = \pi D^2 / 4 \quad (101)$$

The heat transfer between the common area of the valve and valve seat is not considered once the heat exchange between hot gases and metal surfaces alone are of interest.

The area of the cylinder wall  $A_c$  in contact with the hot gases is the instantaneous cylinder area and is a function of the position of the piston. The cylinder head area was assumed to be that of a disc shape, the area of the valve

heads being subtracted.

$$A_h = \pi D^2 / 4 - (A_i + A_e) \quad (105)$$

The temperatures were assumed based on the values proposed by French and Atkins (1973) and are as follows

$$T_p = 593 \text{ K}$$

$$T_h = 623 \text{ K}$$

$$T_c = 398 \text{ K}$$

$$T_e = 1073 \text{ K}$$

$$T_i = 473 \text{ K}$$

The mean wall temperature was then estimated using the relationship

$$T_w = \sum T_j A_j / \sum A_j \quad (j = 1, 2, \dots, n) \quad (106)$$

The total area for heat transfer is

$$A = \sum A_j \quad (j = 1, 2, \dots, n) \quad (107)$$

Therefore, the heat losses from the gases to the cooling medium can be evaluated either for each element for example, the inlet valve, exhaust valve, etc or for the total area, having an average temperature given by (106). When there is the presence of a flame inside the combustion chamber, two subsystems coexist in the chamber separated by the reaction zone. In this case a slightly different approach was taken. The temperature of the wall was taken as that given by expression (107) but the areas for heat transfer were split in two: the area wetted by the burned gases and the area wetted by the unburned gases. The process adopted to calculate both areas is considered in section 2.5 where the geometrical model is described. Also, different heat transfer coefficients are considered whether they refer to the region occupied by the products of combustion or to the region occupied by the unburned

mixture.

## 2.5 Geometric model

It is necessary for the simulation of the combustion process in spark ignition engines using digital computer techniques, to know the enflamed volume, the flame front area, and the area for the combustion chamber surface in contact with the enflamed material as a function of the flame radius and these can then be used to calculate the flame speed, the heat transfer area and the burned volume.

The above mentioned parameters could be calculated using the interpolation method where the burned volume, heat transfer area, burned volume and flame front area at a number of piston and flame front locations are input to the program as array of data or by means of polynomial equations. These data may be obtained either experimentally or using theoretical expressions.

The burned volume and surface areas at a specific flame radius and crank angle can be calculated by interpolation between the input value. The advantage of using such a method is that the computations are simple and that various flame front profiles can be assumed. However the method presents the serious disadvantage that large amount of data is necessary for each chamber design and consequently introduces a time consuming process to generate the input data.

This method was therefore discarded in favour of a more straightforward procedure which consists of computing the burned volume and flame front areas as a function of the flame front radius by means of a numerical integration.

In this method expressions relating the geometrical parameters to burned volume, flame front area and flame

radius are determined and are then solved numerically. One serious drawback to this procedure is that it can only be applied to chamber shapes which can be readily geometrically defined, but it retains the advantages that it requires a minimum of input data and gives a high degree of accuracy.

In the present work the method utilized was that of numerical integration. A similar process was also used by Tabaczynski (1977), Blizzard and Keck (1974) and by Annand (1970) with good results. They applied the process to a disc chamber with the spark plug in the upper flat face of the combustion chamber and assumed a spherical propagation of the flame front.

Relationships may then be derived to express the burned volume, flame front surface and wall surface area enveloped by the enflamed material in terms of the flame front radius. If the procedure is to be applied to other forms of combustion chambers it has to be modified to a certain extent. The modifications introduced should allow for the calculation of the parameters required for the bowl-in-piston combustion chamber utilised in this work.

#### 2.5.1 Basis of calculation

In what follows, the theory is applied to a disc shape combustion chamber having the spark plug located at the upper flat surface of the chamber, which is the case of the Ricardo E6 engine and in the end of the section the bowl-in-piston chamber is considered. The geometry and the symbols used can be seen in Fig 2.16 for the disc chamber and in Fig 2.17 for the bowl-in-piston chamber.

The chamber is a right circular cylinder of diameter  $D$  and height  $H$ . A spherical flame of radius  $R_f$  is imagined to have originated at point  $P$  on the upper flat surface and

at a distance  $R_c - R_s$  from the edge.

The parameters to be determined are the enflamed volume  $V_f$ , the area of the flame front  $A_f$  and the area of the chamber surface in contact with the enflamed volume. This area is the summation of three areas ie, the area of the flat upper surface  $A_h$ , the area of the lower surface  $A_p$  and the area of the cylinder wall  $A_w$ , the total area being  $A_t$ .

Three cases should be considered whether the flame radius is

- .less than or equal to the distance from the spark plug to the cylinder wall and less than the chamber height
- .greater than the distance from the spark plug to the cylinder wall and smaller than the chamber height
- .greater than the distance from spark plug to cylinder wall and greater than the chamber height.

In the first two cases defined as above the geometrical parameters are easily computed. The perimeter of the flame front, measured over the reference plane, parallel to the flat surfaces and distant  $y$  from the upper surface, is found to be

$$p = 2 \int R_f \quad (108)$$

The area enclosed by the flame, measured over the reference plan is

$$b = \int R^2 \quad (109)$$

The perimeter of the flame front in contact with the cylinder wall, measured over the reference plane is  $q = 0$ . The enflamed volume is that of a spherical zone defined by the reference plane and the plane of the cylinder head separated by a distance  $y$ .

$$V_b = \int x (R_f^2 - x^2 / 3) \quad (110)$$



The area of the flame front surface is given by

$$A_f = 2 \int R_f \, x \quad (111)$$

and the areas wetted by the flame front can be calculated by

$$A_h = \int R_f^2 \quad (112)$$

$$A_w = 0 \quad (113)$$

$$A_p = \int (R_f^2 - x^2) \quad (114)$$

and the total area of the flame front in contact with the combustion chamber walls is

$$A_t = A_p + A_w + A_h \quad (115)$$

In the last case where the flame radius is greater than the distance from the spark plug to the cylinder wall, the process is not so simple and a numerical integration method must be used. The set of equations used to solve this stage of flame propagation is well established and is listed below.

.enflamed volume

$$V_b = \int_0^h (\alpha R^2 + \beta R_c^2 - R_s R_c \sin \beta) \, dy \quad (116)$$

.cylinder head area wetted by flame

$$A_h = \alpha(0) R^2(0) + \beta(0) R_c^2 - R_s R_c \sin \beta(0) \quad (117)$$

.Piston crown area wetted by flame

$$A_p = \alpha(h) R^2(h) + \beta(h) R_c^2 - R_s R_c \sin \beta(h) \quad (118)$$

cylinder wall area wetted by flame

$$A_w = \int_0^h 2 R(y) \beta(y) dy \quad (119)$$

flame front area

$$A_f = \int_0^h 2 R_f \alpha dy \quad (120)$$

with the parameters  $\alpha$ ,  $\beta$ , and  $R$  defined by

$$\alpha = \arccos \left[ \frac{(R_s^2 + R^2 - R_c^2)}{2 R_s R_c} \right] \quad (121)$$

$$\beta = \arccos \left[ \frac{(R_s^2 + R_c^2 - R^2)}{2 R_s R_c} \right] \quad (122)$$

$$R = (R_f^2 - y^2)^{1/2} \quad (123)$$

The total area of burned gas in contact with wall is

$$A_b = A_h + A_p + A_w \quad (124)$$

and the total area of unburned gas in contact with the wall

$$A_u = 2 \pi R_c (R_c + H) - A_b \quad (125)$$

The above set of equations are solved using Simpson's Rule. The process consists of dividing the region of interest into an even number of equal width strips and evaluating the value of the function at each ordinate.

When modelling the bowl-in-piston chamber some modifications should be introduced in the process above described in order to take into account the differences in the geometry of the combustion space. Two main cases should be considered, whether either the flame front intercepts the edge of the bowl but not the bottom or intercepts both the edge and the bottom as depicted in Fig 2.18.

When the flame front does not enter in contact with the piston, the geometric properties are evaluated considering the flame with the shape of a hemisphere. In both cases where the flame enters in contact with the piston, a process of integration of the functions showed above is necessary.

In that case, two regions are considered namely, one corresponding to the portion of the combustion chamber above the piston and the second corresponding to the region inside the bowl. The functions are then evaluated separately, for each region and the results added to give the final values of the areas wetted by the flame and the burned volume. The total volume of the burned gases will be the same as in the disc chamber assuming the compression ratio is not changed. However, the area of the cylinder wall wetted by the flame will be much reduced for a given crank angle, while the area of the piston wetted by the flame will be larger than in the disc chamber. The flame front area also will be larger than in the case of a disc chamber.

Fig 2.19 represents the areas wetted by the flame for the three cases considered in this section for the disc chamber illustrating the heat transfer regions considered by the model.

## 2.6 Prediction of the squish velocity

Despite the fact that some spark ignition engines use swirl as a means to augmenting the turbulence inside the combustion chamber, squish is a by-product of the use of a Heron combustion chamber.

This type of combustion chamber has been widely applied to diesel engines due to their ability to intensify the swirl of the gases, a rotational flow within the cylinder about

its axis. In diesel engines and stratified charge engines, swirl is a highly desirable feature aiming to ensure a rapid mixing between fuel and air and to promote a fast burn of the combustible mixture.

Squish, on the other hand is a radial flow that appears when the piston reaches the TDC when the gases are forced into the piston bowl due to the drastic reduction of the cylinder volume. Squish induces a high turbulence in the combustion chamber and also a rotational flow inside the cup, which resembles a rotating torus with an internal swirl about a circumferential axis as depicted in Fig 2.20.

Since the early days of engine development, it is understood that the flame propagation in the cylinder of a spark ignition engine is strongly dependent upon the motion possessed by the charge and squish itself is a highly effective mean to induce high turbulence near the TDC, accelerating the burning process.

However, the higher the squish the better for the combustion process can not be regarded as always true, because squish can also increase the heat losses, therefore reducing charge temperature and consequently peak pressure and temperature achieved during combustion.

Early references to squish prediction have been described by Dicksee (1949) who does not gave much detail of the method proposed for computing the radial velocity of the charge and by Loeffler (1954) who assuming that the air was incompressible, obtained the following expression for the squish velocity in a bowl-in-piston combustion chamber

$$U_s = N \quad L \quad (D^2 - d^2) / 1440 \quad d$$

where L is given by

$$L = (\sin \theta + 1 \sin (2 \theta)) /$$

$$\{(1 - \cos \theta) + 1 \sin^2 \theta + 2 Bc / S\} \quad (126)$$

Woods and Ghirlando (1975) developed a method that includes the effect of inertia of the gas, skin friction and piston ring leakage, all the phenomena being represented by differential equations. In a comparison of the effects of engine speed on the squish velocity, they found that a very small difference exists between the results produced by both methods.

Gosman and Johns (1978) have also developed a finite difference computer program which solves the differential conservation equations governing flow and heat transfer in an axi-symmetrical bowl-in-piston combustion chamber. In common with Woods and Ghirlando (1975) they found that the squish velocities predicted by Fitzgeorge and Allison (1963) method were almost identical to the values given by their far more complex and comprehensive models.

From all the above evidence it was decided to utilise a simple theory in the present work which aims to the prediction of the squish velocity in the spark ignition engine.

The method employed in the present work assumes that pressure is uniformly distributed inside the cylinder. This is not rigorously true in the case of a bowl-in-piston combustion chamber, for example, when the mixture moves rapidly into the bowl when the piston moves toward TDC during the compression stroke. This in turn implies the existence of a significant pressure difference between the annular space and the bowl space. Similar reasoning can also be applied to other types of combustion chamber that makes use of a squish area to produce the squish effect.

The movement induced by the portion of the charge being squished out from the annular space toward the bowl is of a toroidal nature as indicated in Fig 2.20. Although the squish velocity can be greatly enhanced by the reduction in the bumping clearance of squish height, the toroidal rolling speed is not increased by the same magnitude. Additionally, bowl diameter plays an important role in the development of the squish velocity. A reduction in bowl diameter, which is equivalent to an increase in the quench area, causes an increase in the squish velocity.

The theory below considers the gas inside the cylinder as having negligible inertia and viscosity and is not subjected to any surface friction effects. The instantaneous pressure inside the cylinder during the compression stroke may be assumed to be uniformly distributed throughout the combustion chamber volume. The flow is axy-symmetric which implies that the axis of the bowl coincides with the axis of the cylinder. For the definition of the symbols employed refer to Fig 2.20 and Fig 2.21.

Consider a small movement of the piston towards TDC reducing the clearance height from  $S$  to  $S - dS$ . The area of the annular face between the general radius  $R$  and the cylinder radius  $R_c$  can be represented by  $A_r$  where

$$A_r = \pi (R_c^2 - R^2) \quad (127)$$

The volume displaced by this annular area during the piston displacement  $dS$  is given by

$$V_r = A_r dS \quad (128)$$

while the total change in volume of the cylinder is represented by

$$V_c = A_c dS \quad (129)$$



provided the heat losses can be assumed to be small during the process, the temperature variation can be assumed to be negligible and therefore density can also be considered to be uniform. During combustion this assumption is no longer admissible since large variations in density is to be expected between the burned and unburned gases. As the pressure is assumed to be uniformly distributed, the mass of the portion of the charge in the annular space can be initially found by the expression

$$M_a = M_t A_r S / (A_c S + V_b) \quad (130)$$

where  $M_t$  is the total mass of the cylinder content and  $V_b$  is the volume of the bowl.

After the displacement  $dS$  the mass  $M_a$  can be estimated from

$$M_a = M_t A_r (S - dS) / \{A_c (S - dS) + V_b\} \quad (131)$$

The flow  $dM$  through the cylindrical surface represented by EF in Fig 2.20 is estimated by taking the difference between the two expressions above and assuming the instantaneous density as

$$\rho_i = M_t / (A_c S + V_b) \quad (132)$$

results for the expression of the volume in the annular space

$$dV = M_t A_r \{ (A_c S + V_b) / M_t \} \{ S / (A_c S + V_b) - (S - dS) / \{A_c (S - dS) + V_b\} \} \quad (133)$$

After rearranging

$$dV = A_r V_b dS / (A_c S - A_c dS + V_b) \quad (134)$$

The term  $A_c dS$  may be omitted to first order and the equation (134) becomes

$$dV / dS = A_r V_b / (A_c S + V_b) \quad (135)$$

The rate of volume flow with respect to time can then be obtained by multiplying  $dV / dS$  by the mean piston velocity  $dS / dt$

$$dV / dt = A_r V_b / (A_c S + V_b) dS / dt \quad (136)$$

The linear velocity of the air may be obtained by dividing the flow rate  $dV / dt$  by the area of flow ie, the cylindrical area  $2\pi R S$ . Substituting  $dV / dt$  by  $U_r$  and  $dS / dt$  by  $U_p$ , equation (136) becomes

$$U_r = A_r V_b U_p / 2\pi R S (A_c S + V_b) \quad (137)$$

or using equation (127)

$$U_r = (R_c^2 - R^2) V_b U_p / 2 R S V_t \quad (138)$$

The above expression give an estimate of the mean radial velocity at any radius  $R$  in the annular space. For the particular case of a bowl-in-piston combustion chamber, with bowl radius  $R_b$ , the squish velocity at the edge of the bowl is given by

$$U_s = U_p V_b (R_c^2 - R_b^2) / 2 R_b S V_t \quad (139)$$

where  $V_t = V_b + \pi R_c^2 S$ .

The same principle can be applied for the derivation of an expression that allows the estimate of the squish velocity of any chamber, provided consideration is given to differences in geometry. The geometry of the squish chamber and the symbols used are given in Fig 2.21.

Starting from equation (136), the expression that gives

the linear velocity of the air may be obtained by dividing the flow rate by the flow area, which in this case is  $L S$ , where  $L$  is the length of the sector chord. Denoting the linear velocity by  $U_s$  and the squish area by  $A_s$  results in

$$U_s = A_s V / \{L S (A_c S + V)\} dS / dt \quad (140)$$

where  $dS / dt = U_p$  is the mean piston velocity. In equation (140)  $L$  is the length of the chord that determines the segment whose area is the squish area. From considerations of geometry, the area of a segment of circle limited by the central angle  $\phi$  can be calculated by the expression

$$A_s = 0.5 R_c (\phi - \sin \phi) \quad (141)$$

where  $\phi$  is expressed in radians. The length of the chord can then be determined from

$$L = 2 R_c \sin (\phi / 2) \quad (142)$$

The above model was formulated for circumstances in which the engine operated in motored mode when equation (140) represents a good approximation for the velocity. This is clearly not the case for the firing engine where large variations in density are expected to occur between the burned and unburned gases which will cause the squish velocity under firing conditions to deviate from that determined in the motored mode.

The effect of combustion on the squish velocity can be qualitatively evaluated. However, considering that the spark plug is normally located in opposition to the squish area, the burned gases from the spark source compress the unburned gases ahead of the flame front, causing a reduction in the velocity of the gases moving from the squish area towards the spark plug. Spark timing is therefore expected to influence decisively the variation

of the squish velocity.

The experimental determination of the squish velocity under firing conditions is extremely difficult to perform. Recently, with the advent of the laser, it has become possible to perform measurements of local and instantaneous velocity within the reacting flow without the insertion of intrusive probes. However, no work on the particular subject of measurements of the effect of combustion on the squish velocities has been found by the author.

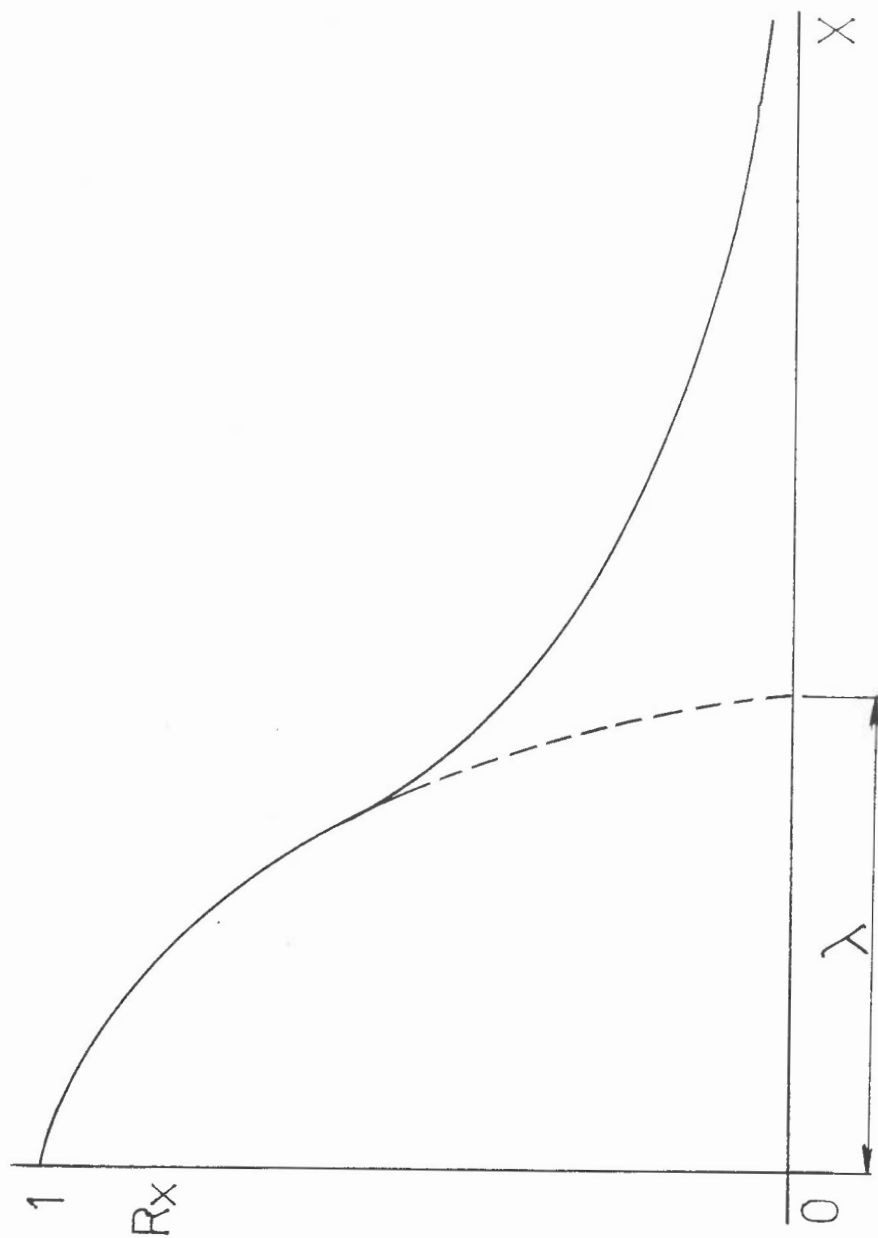


Fig. 2.1 Correlation coefficient  $R_x$  and estimate of the Taylor microscale

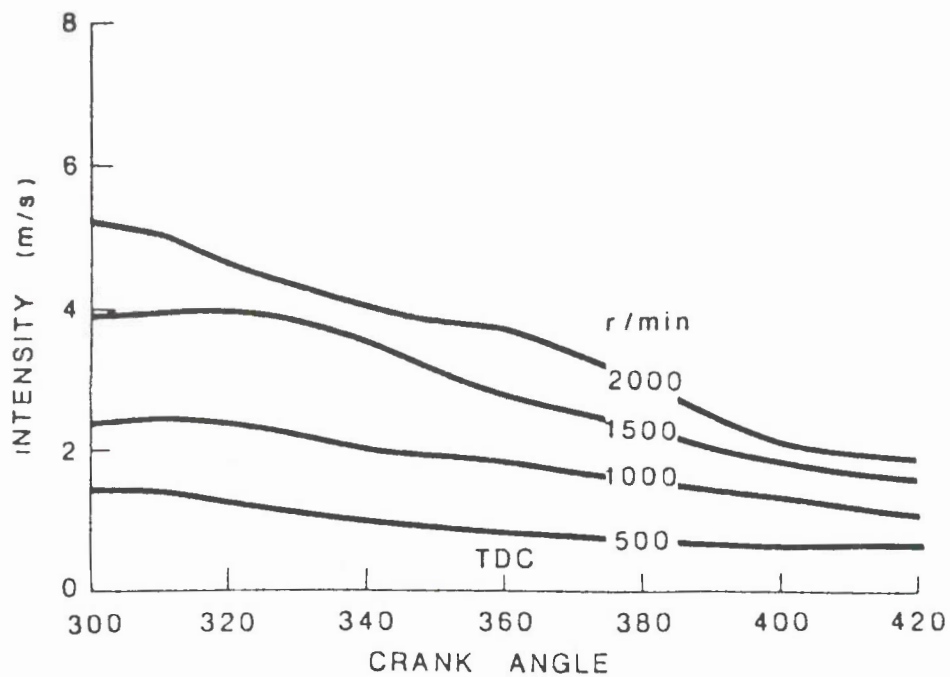


Fig. 2.2 Variation of the turbulence intensity with time. (from Groff, 1981)

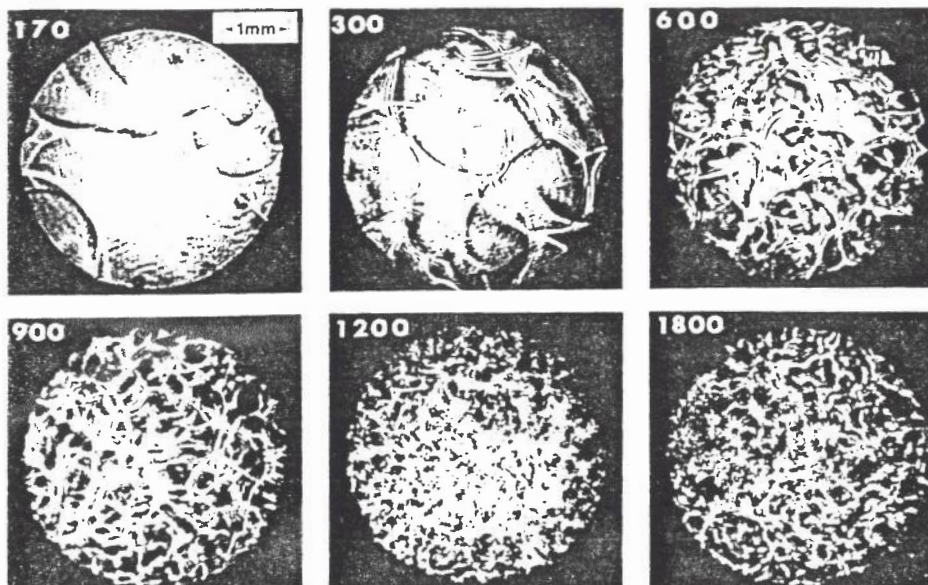


Fig. 2.3 Microshadowgraphs of flame propagation, showing increasing wrinkling with engine speed, on top left corner. (from Smith, 1982)



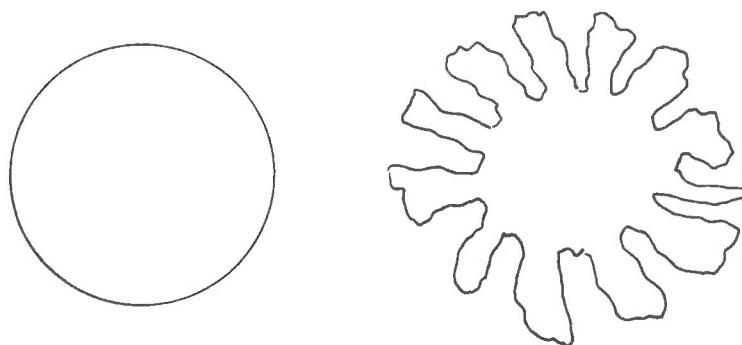


Fig. 2.4 laminar and turbulent propagation of a spherical flame.

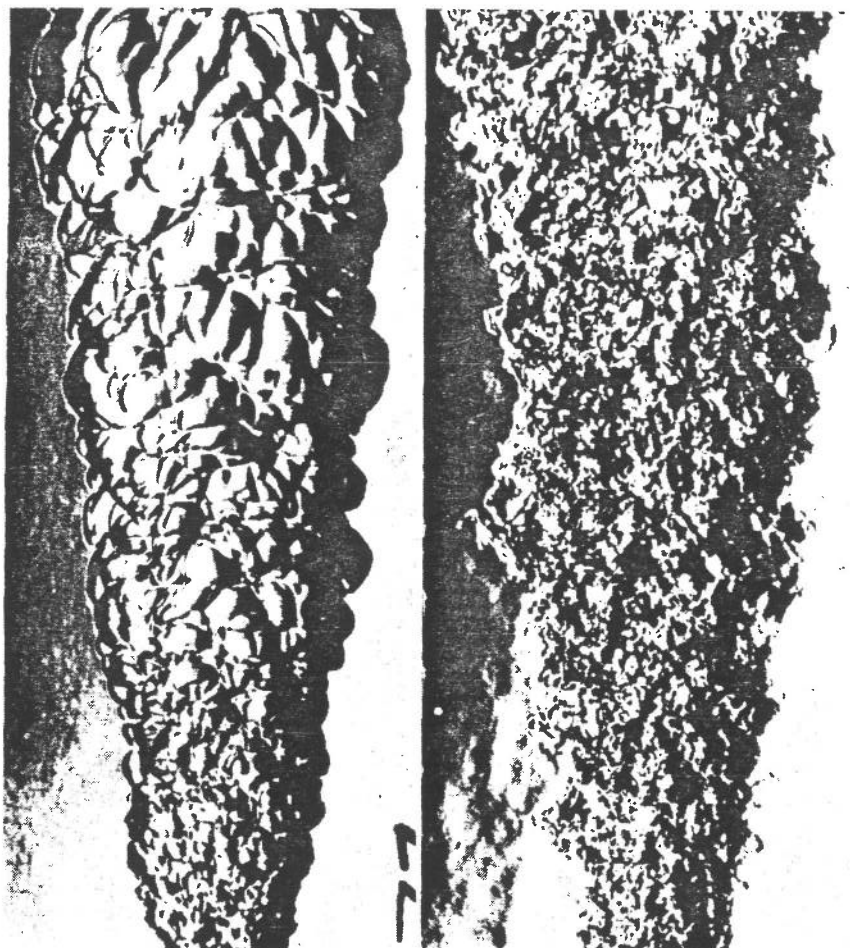


Fig. 2.5 Stoichiometric propane-air flames under conditions of low and high turbulence.  
(from Ballal and Lefebvre, 1975)

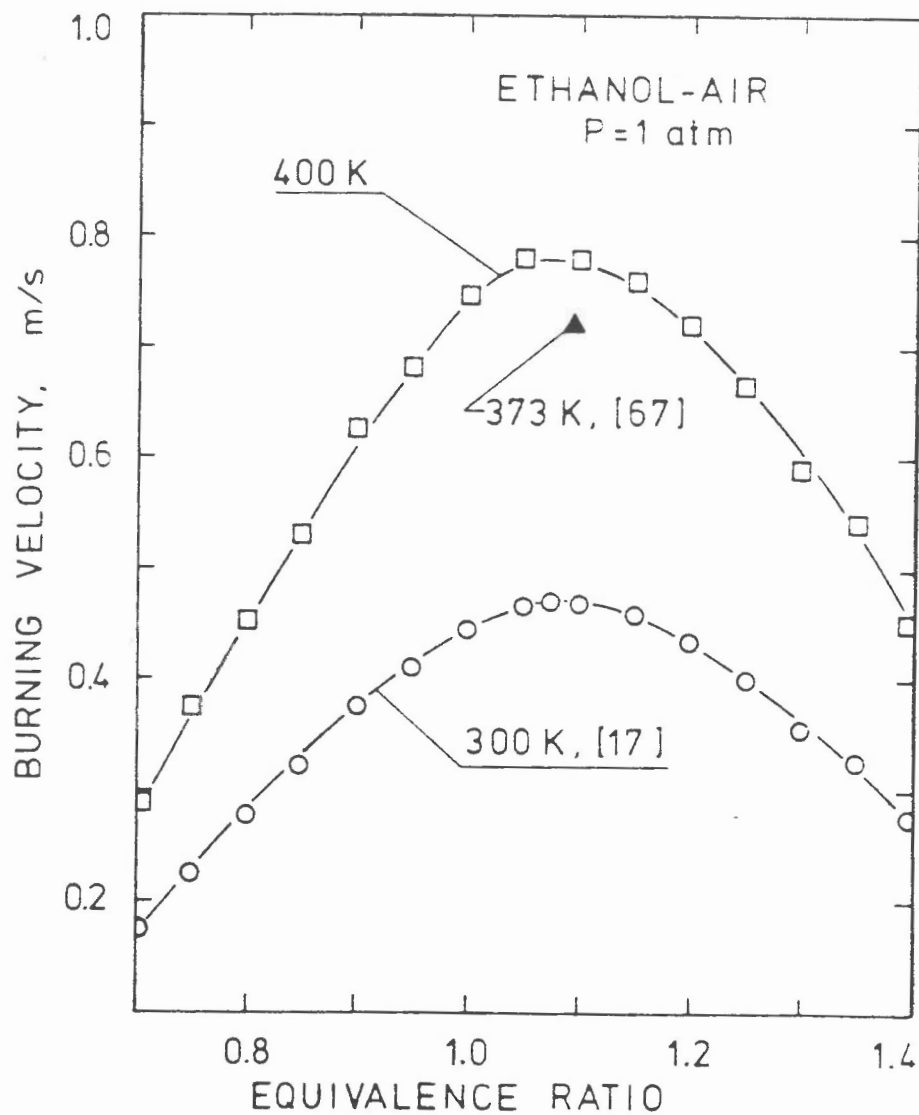


Fig. 2.6 Laminar burning velocity of ethanol-air mixtures. (from Gulder, 1984)

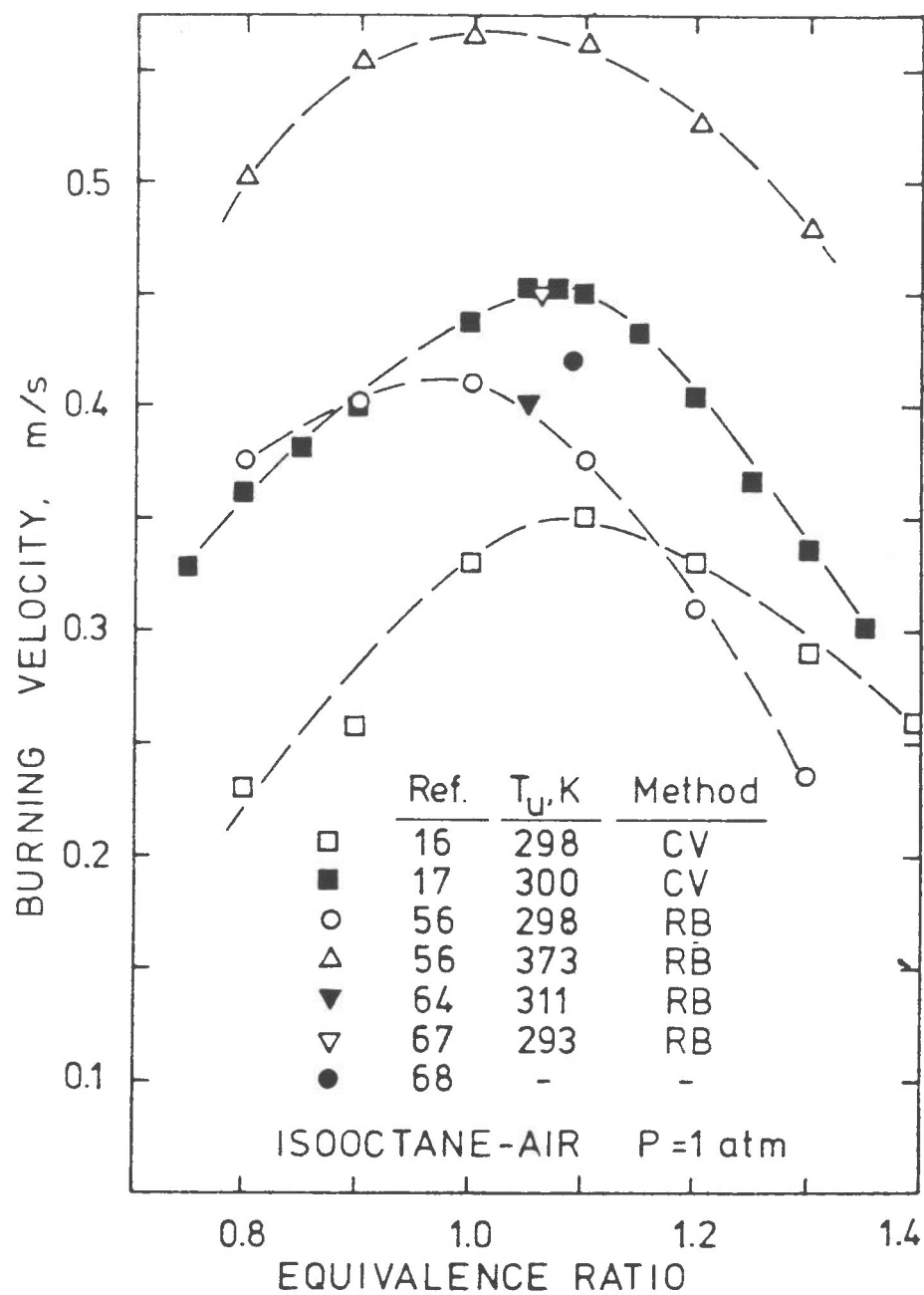


Fig. 2.7 Laminar burning velocity of isooctane-air mixtures. (from Gulder, 1984)

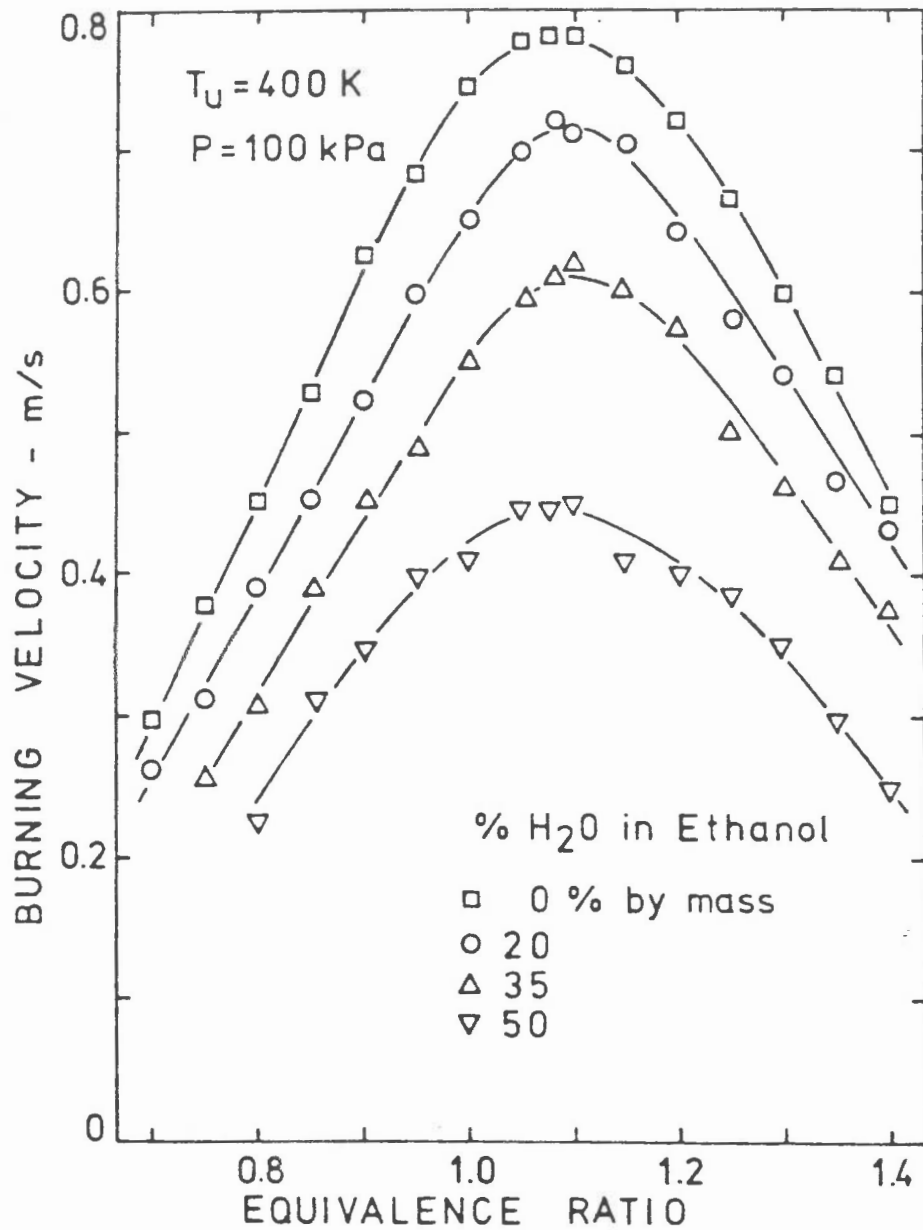


Fig. 2.8 Effect of water concentration on the burning velocity of ethanol-air mixtures. (from Gulder, 1984)

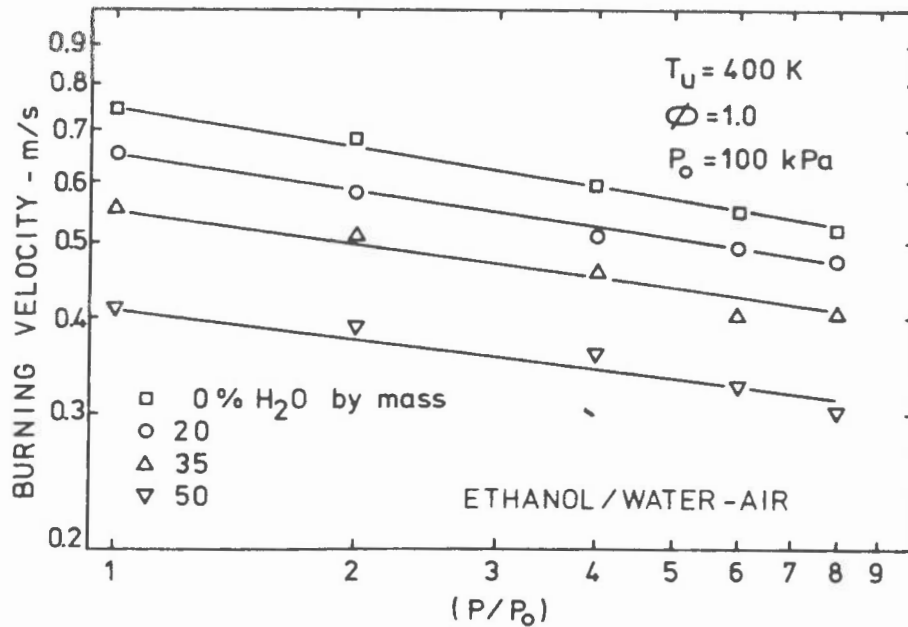


Fig. 2.9 Variation of the burning velocity of isooctane-air mixtures with pressure. (from Gulder, 1984)

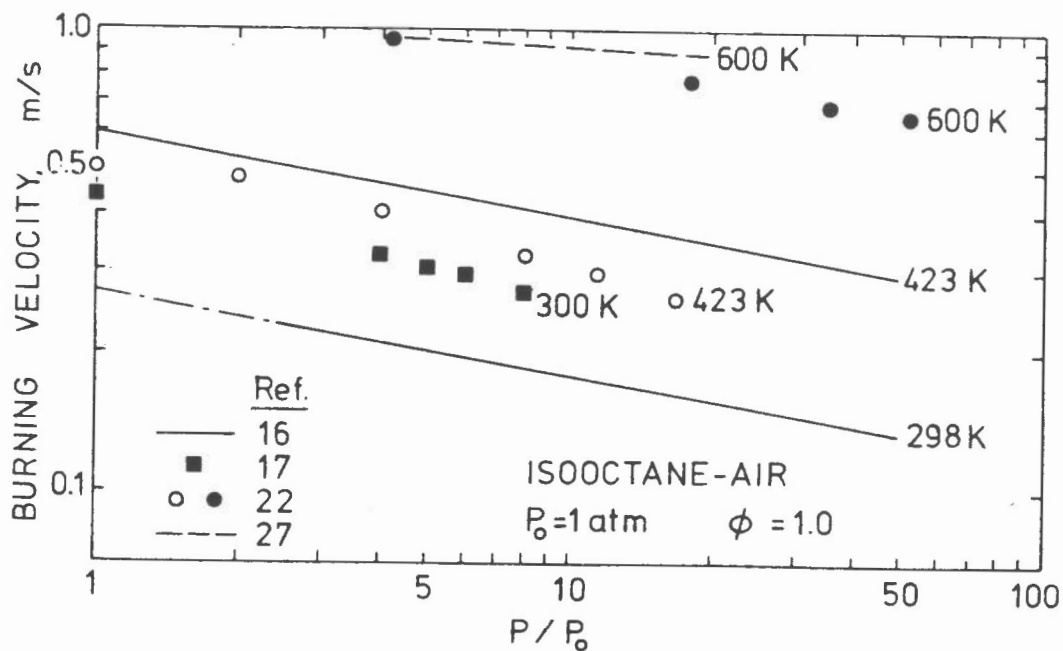


Fig. 2.10 Variation of the burning velocity of ethanol-water-air mixtures with pressure. (from Gulder, 1984)

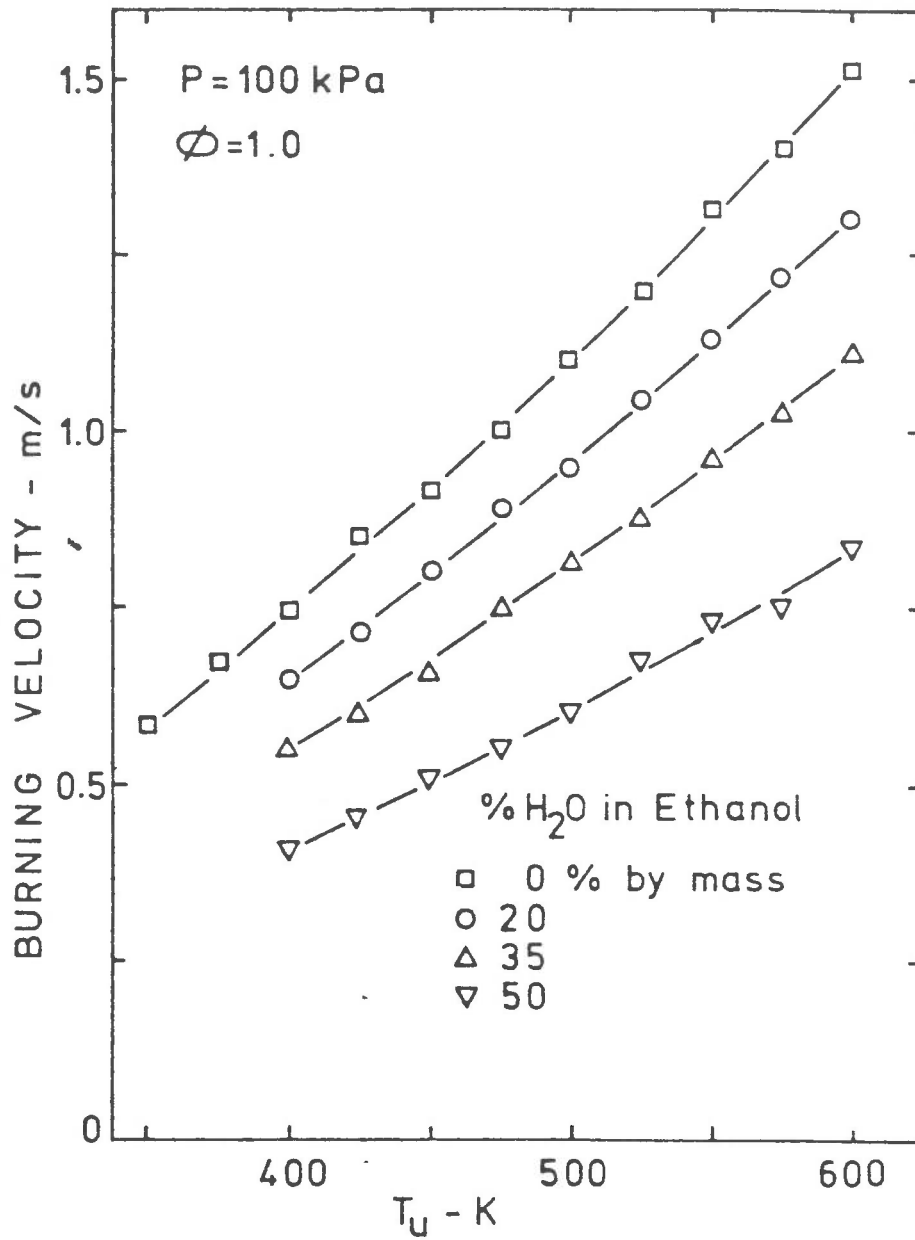


Fig. 2.11 Temperature dependence of the burning velocity of ethanol-air and ethanol-water-air mixtures. (from Gulder, 1984)



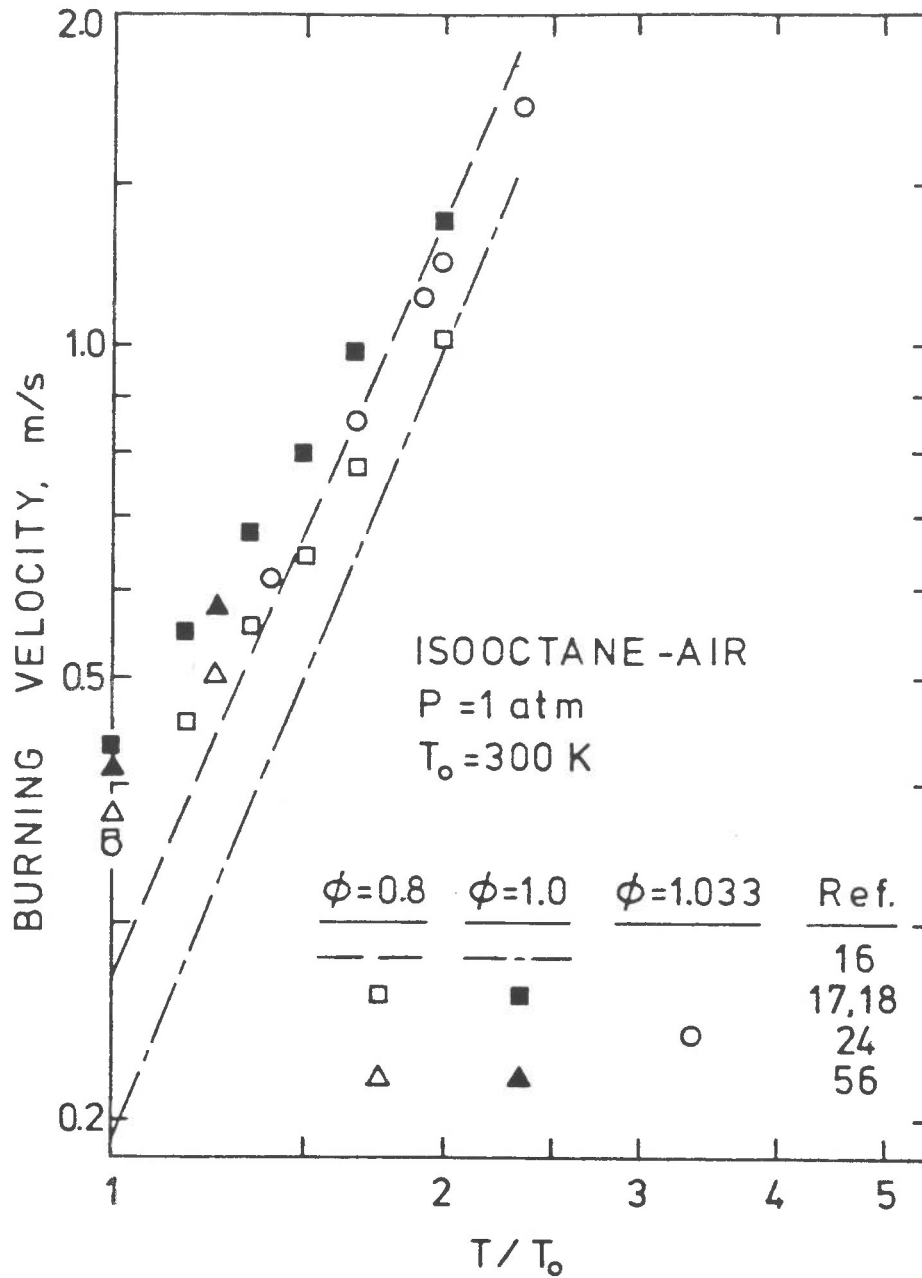


Fig. 2.12 Temperature dependence of the burning velocity of isooctane-air mixtures.  
(from Gulder, 1984)

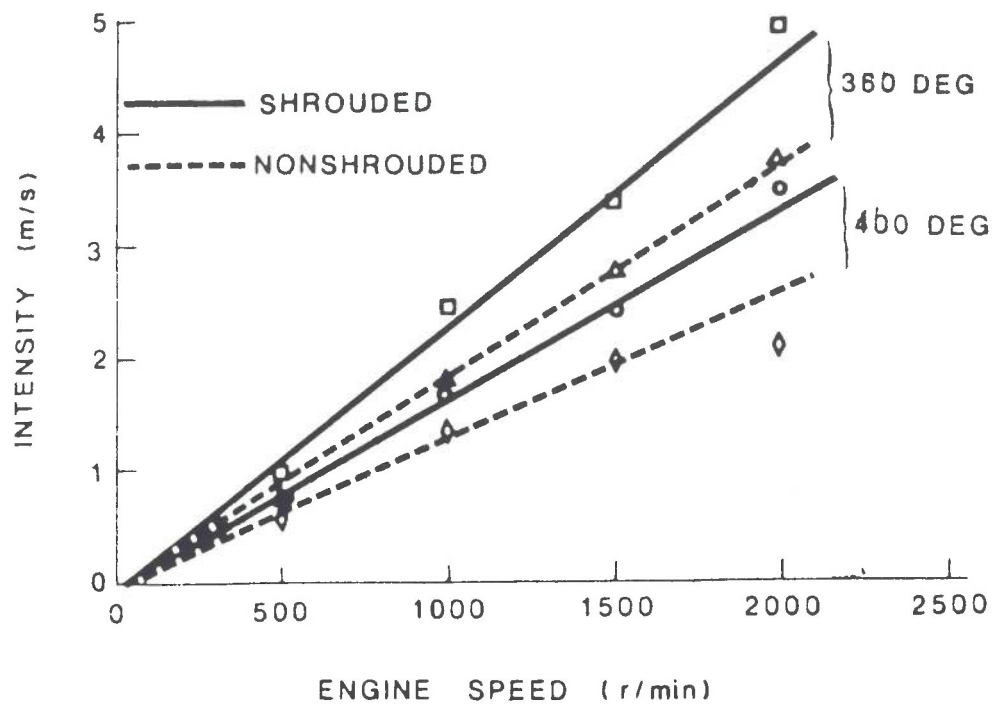
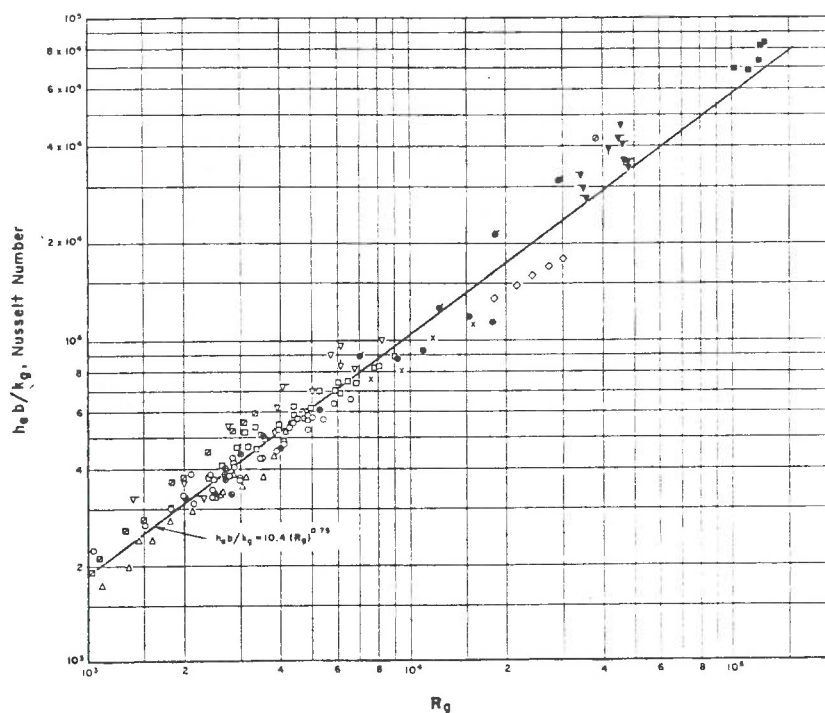


Fig. 2.13 Variation of turbulence intensity with engine speed. (from Groff, 1980)

Fig. 2.14 Variation of Nusselt number with the Reynolds number. (from Taylor, 1977)

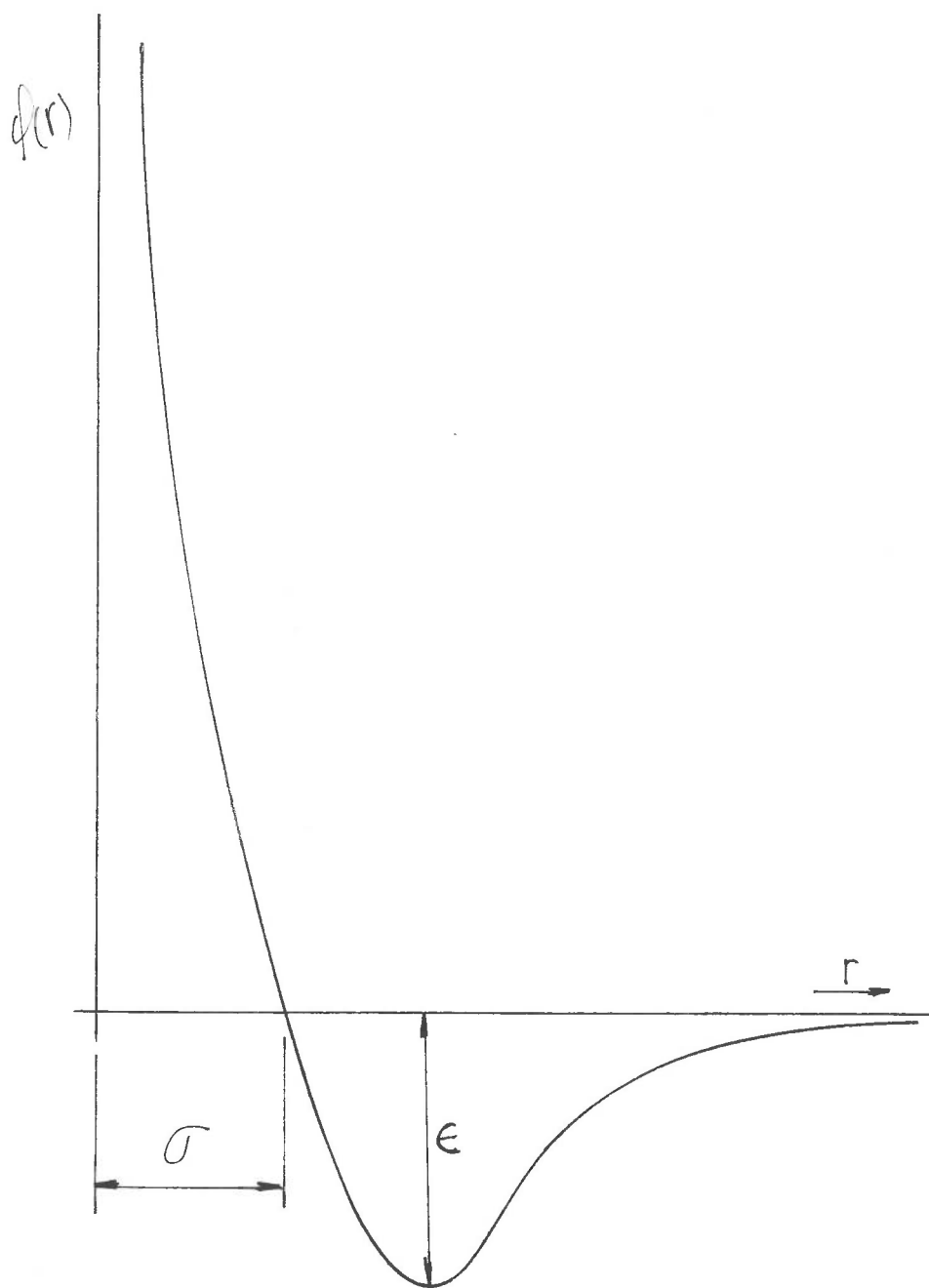


Fig. 2.15 Lennard-Jones potential curve.

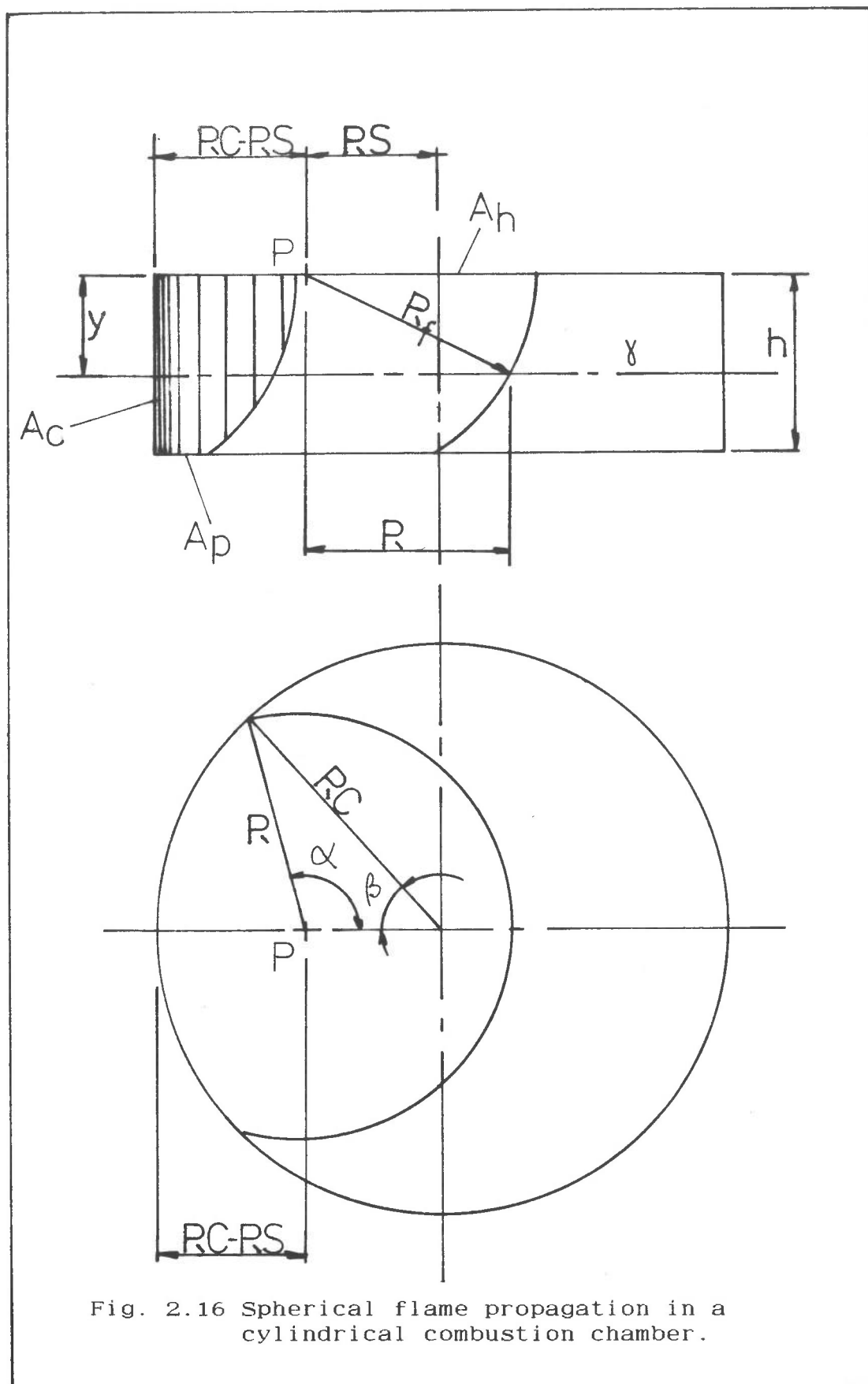


Fig. 2.16 Spherical flame propagation in a cylindrical combustion chamber.

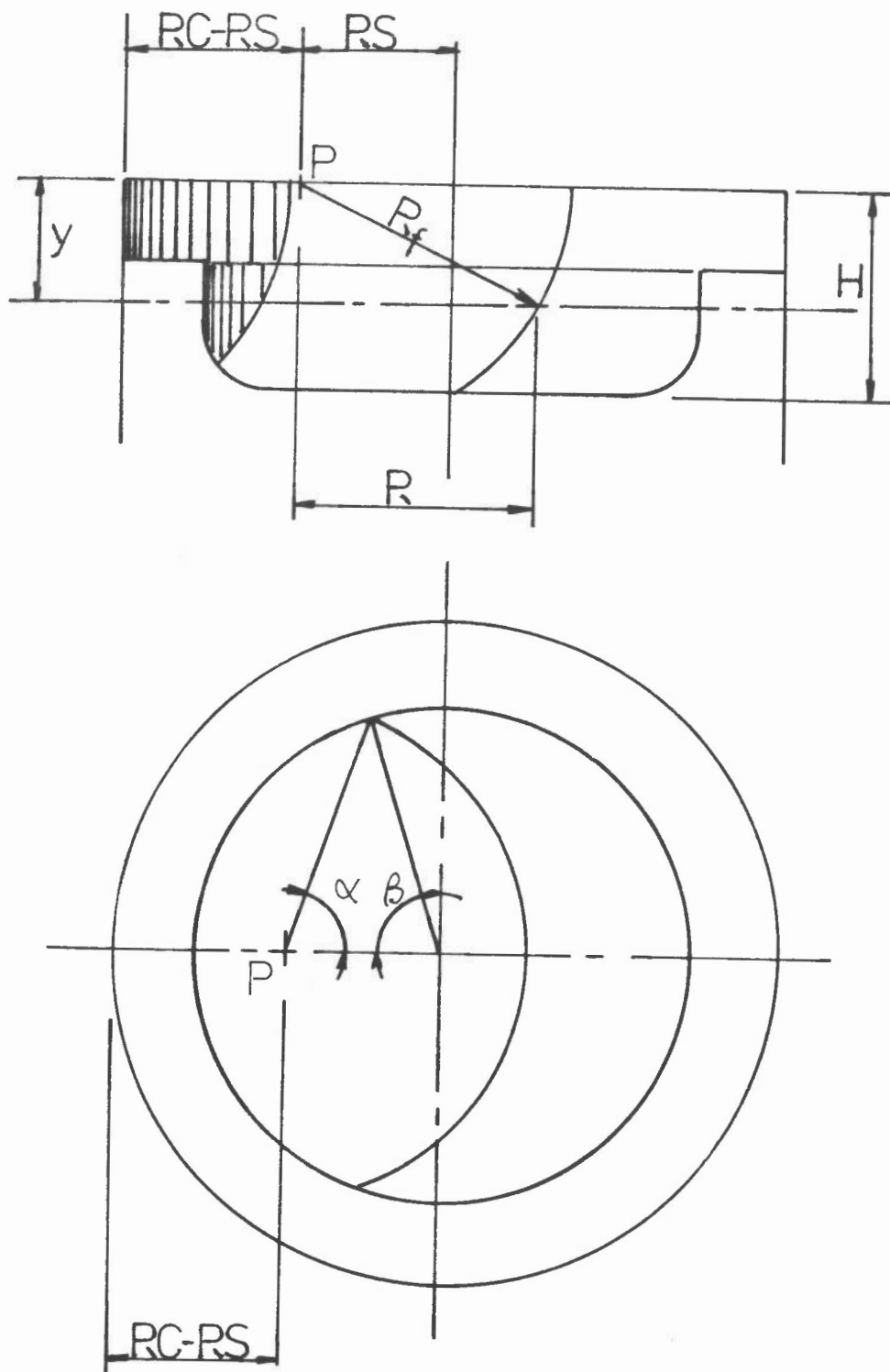
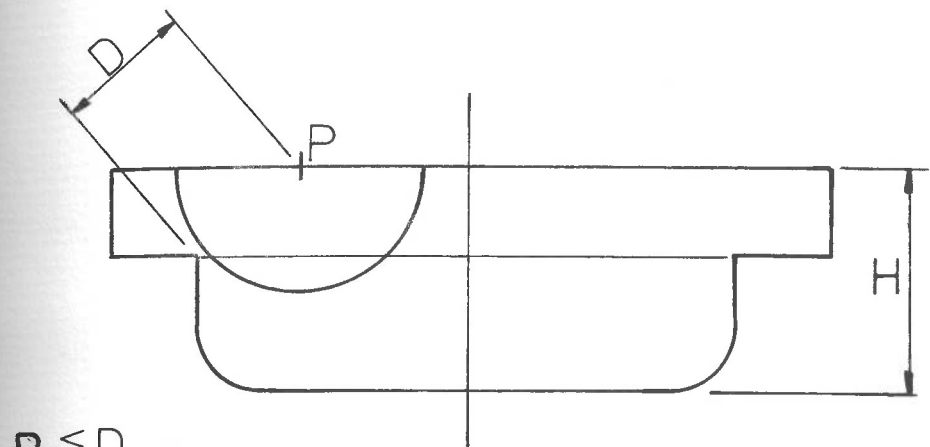
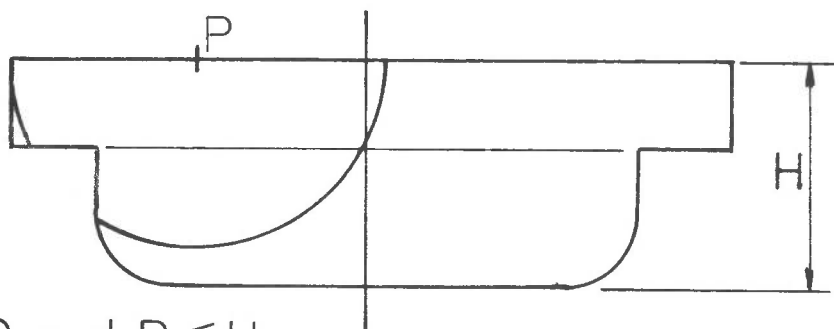


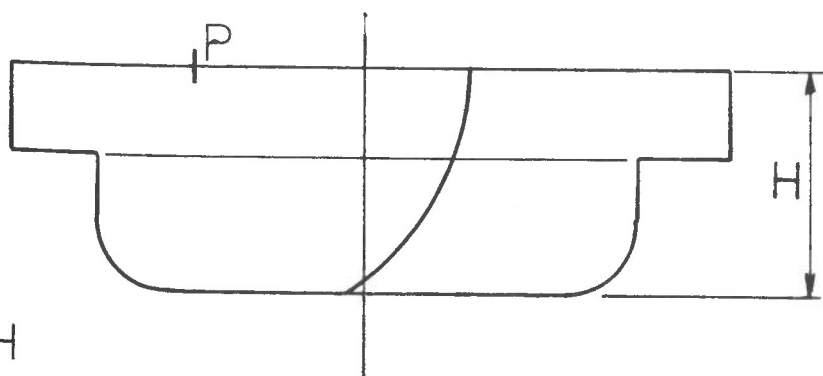
Fig. 2.17 Spherical flame propagation in a bowl-in-piston combustion chamber.



$$R_f \leq D$$



$$R_f > D \text{ and } R_f \leq H$$



$$R_f > H$$

Fig. 2.18 Interactions between a spherical flame front and a bowl-in-piston chamber.



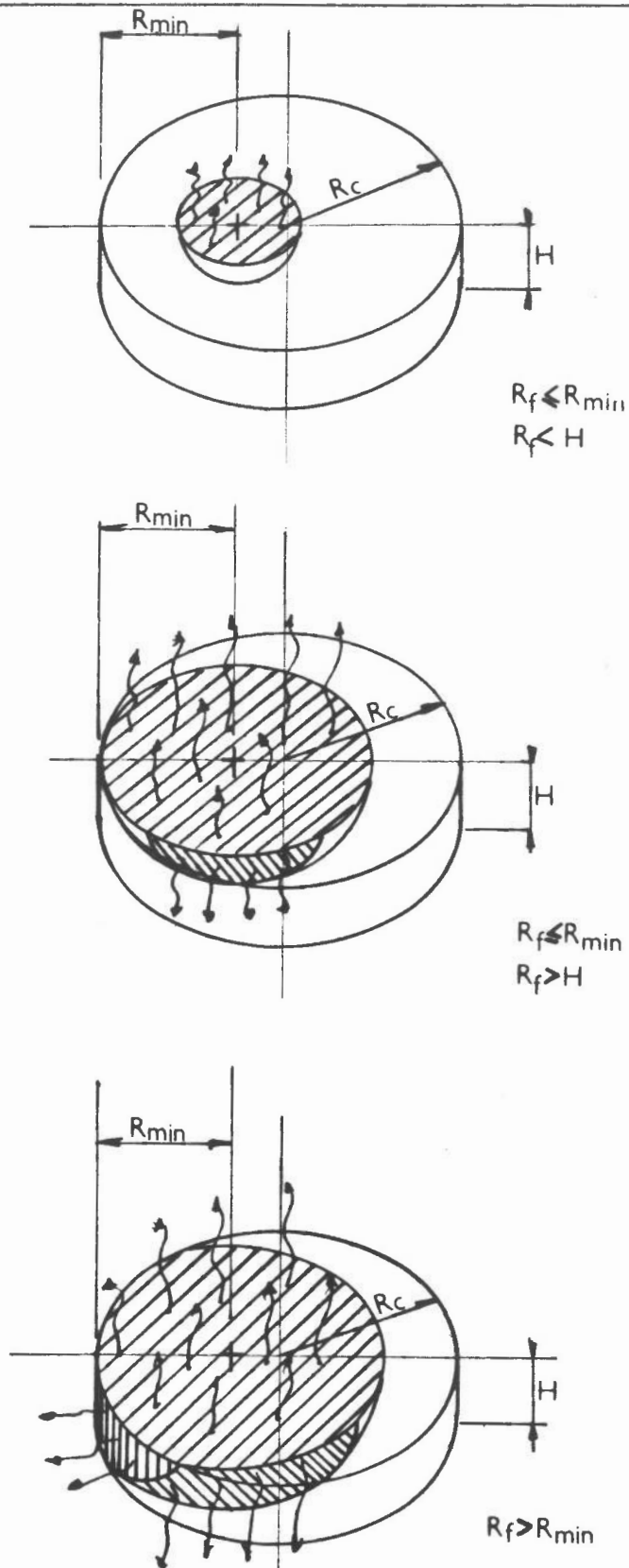


Fig. 2.19 Heat transfer surfaces in a cylindrical combustion chamber.

2.19

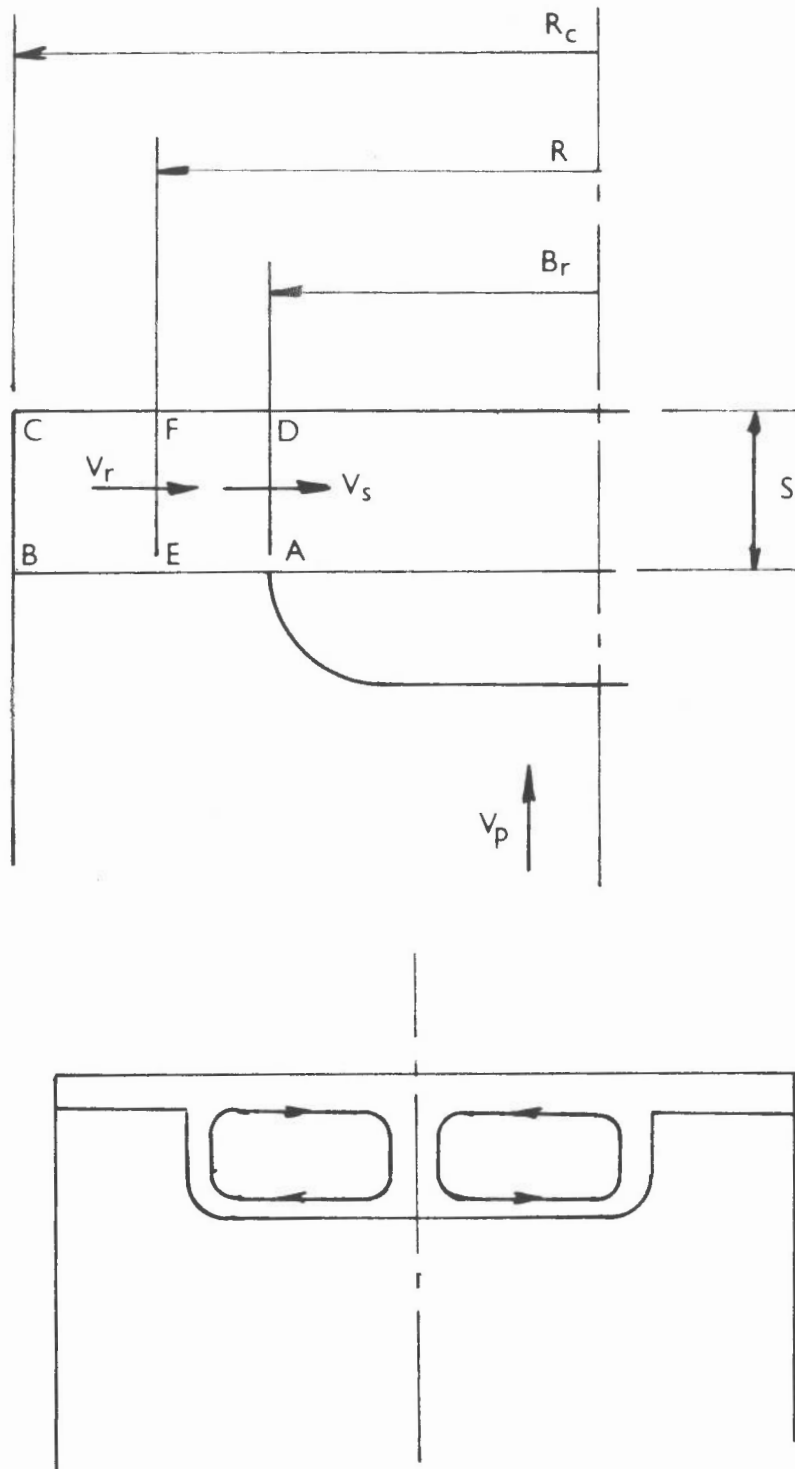


Fig. 2.20 Squish velocity in a bowl-in-piston combustion chamber.

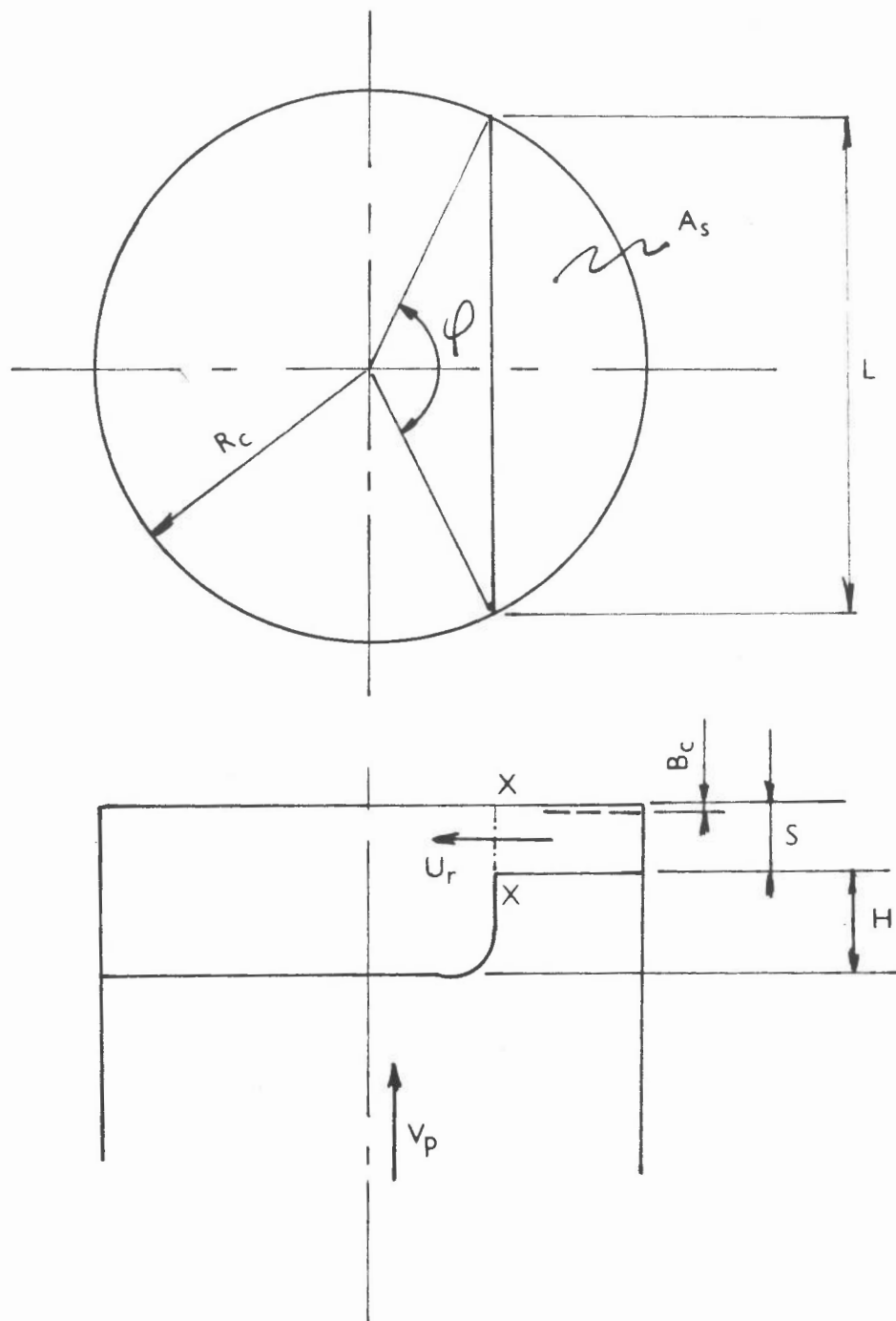


Fig. 2.21 Squish velocity in an open chamber with a concentrated squish area on the piston

# Chapter 3

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## Experimental Investigation

### 3.1 Introduction

This section deals with the description of the equipment utilised to perform the experimental part of this work. Firstly, the Ricardo E6 research engine is described followed by auxiliary equipment.

The modifications introduced and the description of the combustion chambers used are presented in sequence. A description of other equipment necessary to the tests is also introduced followed by a description of the laser equipment used to measure the flame propagation velocity or flame speed, together with the associated optical system.

The knock detection system specially developed for this work is then introduced. The chapter ends with a description of the data acquisition system followed by the presentation of the experimental procedures adopted in this work

### 3.2 The Ricardo E6 engine

The Ricardo E6 engine has proved to be a reliable and well-accepted instrument among researchers for investigations of the combustion process. It is an engine that could not be considered as representative of the modern spark ignition engine in relation to its operational conditions, notably crankshaft speed, combustion duration time, specific power output and geometry.

Nevertheless, its simplicity of operation, running stability, the possibility of easily and quickly mounting and dismounting among other features, make it a very useful tool for combustion research. Furthermore and more importantly, the availability of a great amount of data is

an encouragement to its use.

The Ricardo E6 engine is a single-cylinder, overhead poppet valve, four stroke cycle mounted on a common bed plate with an electrical dynamometer. The engine is adaptable to run as either a spark ignition engine or as a compression ignition engine. When set to spark ignition configuration, the compression ratio may be varied, while the engine is running, from 4.5 to 20 : 1. Alterations to ignition timing and fuel flow rates can also be made while the engine is running.

Lubrication is by means of a separately driven gear pump and engine cooling water is circulated by means of a separately-driven centrifugal pump. The engine can be seen in Fig 1 and its sectional view shown in Fig 3.2 while the main specifications are indicated in Table 3.1.

3.2.1 The cylinder head is made of cast iron with two valves, the larger being the inlet valve and the smaller the exhaust valve. Both valves are actuated by finger followers driven from an overhead camshaft.

3.2.2 The combustion chamber originally is approximately cylindrical in shape, the ends being formed by the flat cylinder head and the piston that is used in the diesel version, with the cavities in the crown so characteristic of the Ricardo Mark V compression swirl combustion chamber. The modifications introduced to the original chamber are described later.

3.2.3 The cooling system of the engine encompasses a high rate of circulation, the temperature being controlled by means of a heat exchanger. The water flow is measured by calibrated orifice and manometer system.



3.2.4 The ignition system is of the capacitor discharge type. A capacitor charged from a DC source is discharged via silicon controlled rectifier through the primary windings of an ignition coil. The discharge is triggered by a signal from a magnetic pick-up mounted on the crankshaft.

3.2.5 The carburettor is a Zenith, type WIP 36 of descending flux, fitted with a 36 mm diameter choke. This carburettor has a variable main jet consisting of a tapered needle valve which permits the mixture strength to be varied between wide limits during tests.

3.2.6 The air metering is performed through an Alcock viscous flow meter which also incorporates a flame trap on the engine side of the meter. The pressure drop across the meter, measured by a variable slope manometer, is used with the meter calibration constant to calculate the air flow. There is also an air heater incorporated within the flow meter, with a capacity of 1 kW.

3.2.7 The dynamometer is of the electric swinging field DC current type and is driven by the engine via a flexible coupling, the two units being mounted on a common cast iron bedplate. The dynamometer is of sufficient capacity to absorb the maximum power produced by the engine at all normal speeds and to motor the engine over the speed range. It is controlled by an AC solid state controller which also rectifies the main current so that the DC dynamometer can be used. It can easily be switched to the motor mode of operation.

3.2.8 All the relevant temperatures in the Ricardo E6 are measured, in its standard form, by means of mercury in

glass thermometers. This method of temperature measurement is a source for inaccuracies and loss of time. For that reason, a new system based on thermocouples was devised.

### 3.3 Modifications to the Ricardo E6

The modifications introduced in the Ricardo E6 engine consisted of the design and fabrication of a steel spacer into which the quartz windows were fitted and the design and fabrication of different types of piston crown that enable the different types of combustion chambers to be tested.

3.3.1 The spacer was made of steel and fitted between the cylinder head and the cylinder barrel, 12 mm thick with a central hole equal to the cylinder bore. The thickness of 12 mm resulted from the necessity to use high compression ratios and therefore, the windows must be small in order to achieve that purpose.

Holes were provided throughout the spacer to allow the cooling water to flow between the cylinder barrel and the cylinder head. The fact that the spacer is also water cooled helped in keeping the temperature of the quartz window holders at a tolerable level.

The spacer is positioned concentrically with the cylinder bore by means of elastic pins. The accurate positioning of the spacer in relation to the cylinder bore is essential in order that the piston can pass through the hole and reach a closer position from the cylinder head. The purpose of the spacer is to hold the two tubes in which the quartz windows are mounted.

These quartz windows provide optical access to the combustion chamber in order that the two parallel laser

beams can traverse the combustion chamber. There are two pairs of holes in the spacer positioned in such a way to permit the use of the quartz windows either in the vicinity of the spark plug, at a distance of approximately 15 mm, or more distant from the spark plug, at a distance of approximately 50 mm. The spacer can be seen in Fig 3.3. The dimensions are given in Fig 3.20.

3.3.2 The quartz window holder has a cylindrical shape and has been manufactured in stainless steel, having the tapered quartz window mounted in the fire side. The angle of the cone is such that the window may be kept in place by frictional forces only. The sealing between glass and metal is made with a special heat resistant type of Araldite. The temperature reached in the tubes was found to be around  $110^{\circ}\text{C}$  maximum. The quartz window tube is shown in Fig 3.4.

3.3.3 The combustion chambers investigated in this work were of the cylindrical and the Heron or bowl-in-piston types. The normal piston of the engine was substituted by a new one with a much thicker crown in order to allow machining of a centered recess and the six holes used to fix the heads to the piston. The heads were fabricated from an aluminum alloy containing silicon whose purpose was to reduce thermal expansion to a level comparable to that of the original piston material. The heads were then mounted on the piston and fixed by means of six bolts which were then locked one to another by means of a steel wire.

3.3.3.1 The disc-shaped combustion chamber is the normal chamber in the Ricardo E6. In this work, due to the introduction of the spacer between the cylinder head and the cylinder barrel, the cylindrical chamber has been

defined by the cylinder head, piston crown and spacer wall. The piston was fitted with a false head having no recess in the crown as opposed to that which is provided as standard with the engine as can be seen in Fig 3.5.

The compression ratio was chosen using the conventional mechanism of the engine. The clearance volume was accurately measured in order to obtain the compression ratios as close as possible to the theoretical values,  $s$ , according to a procedure presented in appendix 5.

3.3.3.2 The bowl-in-piston combustion chamber. Two types of bowl-in-piston combustion chambers were selected having compression ratios of 10 and 12 : 1 and both with squish ratios of 50 %. The different values of the compression ratio were obtained by varying the diameter and height of the bowl accordingly, once the bumping clearance had to be held at a specified value dictated by the intensity of squish desired. Valve recesses were provided in order to prevent them striking the head of the piston when this was at the TDC position.

A channel was also machined in order to provide clearance for the two laser beams when they traverse the combustion chamber. To determine accurately the clearance volumes for each of the compression ratios, the same procedure described in appendix 5 was executed. Fig 3.6 represents a view of the Heron chamber utilised in the experiments. The laser channel and the valve recesses are clearly visible.

### 3.4 Flame propagation monitoring

The combustion process in an internal combustion engine is arguably regarded as the most important phenomenon that occurs in the powerplant. Therefore, if the objectives in

designing an engine are to obtain good fuel economy, with adequate characteristics of torque, high power output without knock and low levels of exhaust emissions, the combustion process must be studied with great care.

The combustion process, however, cannot be considered separated from the place in which it takes place, that is, the combustion chamber. As the combustion process takes place in a completely enclosed space it naturally becomes difficult to conduct experiments leading to the investigation of flame propagation inside this space.

Several methods employed to study the development of flames inside the combustion chamber have been described in chapter 1, section 1.6. Possibly, the most widely used method until recently has been the flame ionisation probe technique, first used in the twenties and still proving to be very reliable, despite its intrusive character. It is a good method when dealing with fully developed flames.

The method is based upon the fact that, during combustion, ions, electrons and radicals such as O, OH, are produced whilst the reactants are transformed into products. The negatively charged electrons are attracted by the positive electrode of the probe, thereby reducing the effective probe resistance and causing an increase in current. The current after being amplified and processed produces a signal that can be displayed on an oscilloscope and even counted by specially built equipment.

In order that this technique can be applied, several specially built probes are introduced in the combustion chamber, obeying a specific geometry. Basically, the system consists of a DC power supply connected to the ionisation probe via a resistor. The resistance of the probe, if no combustion occurs, is very high and therefore the current flow and the voltage drop across the resistor can be negligible. However, if a flame enters in contact with the

probe, the current and hence the voltage drop across the resistor will increase.

The signal amplitude and pulse width will be functions of the reaction zone thickness, probe dimensions, circuit design and supply voltage. The system is very reliable, easy to operate, cheap to construct and possible to use in most engines, but presents poor spatial resolution.

Today, the increasing utilisation of laser-based techniques for research in fluid flows and combustion makes them attractive for engine combustion diagnostics. In the present work, a laser schlieren or differential laser refraction technique has been successfully employed to measure the flame propagation velocity in the combustion chamber of the Ricardo E6 research engine. The non-intrusive character of the method makes it preferable and more accurate than conventional ionisation probes.

The technique is based on the principle that a laser beam is deflected by the strong gradients of the index of refraction that exist inside the combustion chamber during flame propagation. At the flame front a reaction zone separates the burned gases at a temperature of approximately  $2600^{\circ}\text{C}$  from the unburned gases at a temperature of roughly  $400^{\circ}\text{C}$ , which causes a steep density gradient which refracts the laser beam.

The laser beam which traverses the combustion chamber is deflected towards the region of higher density as the flame front crosses the combustion chamber. A detailed review of schlieren phenomenon is clearly outside the scope of this work and the interested reader is referred to the work of Weinberg (1963).

The key requirement in applying this technique in an engine is that optical access must be provided to the combustion chamber. For that purpose a special spacer



providing access to the chamber through quartz windows was constructed as described in section 3.3.1. Two parallel laser beams that emerge from the beam splitter traversing the combustion chamber through the quartz windows at a distance of 15 mm from the spark plug electrodes gap, are deviated from the faces of photodetectors when traversed by the flame front.

The time elapsed between the spark and the deflection of each laser beam is detected and recorded as described in section 3.5.4. The fact that this technique depends for its application on an internal access to the combustion chamber makes it hardly applicable to conventional automobile engines. The reason is that it is nearly impossible to provide optical access to the combustion chamber of these engines without introducing major changes in the geometry of the existing chamber, therefore, restricting the technique to specially built engines or to single cylinder research engines, an example of which is the Ricardo E6.

The flame propagation velocity in the combustion chamber is measured relative to the walls of the combustion chamber and is also known as the flame speed. This flame speed is the sum of two components, namely, the turbulent burning velocity and the expansion velocity of the combustion products. The former represents the rate at which the flame front propagates into the unburned mixture and is measured using as reference the unburned gas.

It can be understood as the rate at which the unburned mixture is entrained and consumed by the reaction zone. The expansion velocity is the rate of expansion of the burned gases by virtue of the difference in densities between the burned and unburned gases resulting from combustion.

In planar flame propagation, the flame speed can be

expressed mathematically as

$$v_f = u_t (\rho_u / \rho_b - 1)$$

where  $v_f$  is the observed flame propagation velocity,  $u_t$  is the turbulent burning velocity and  $\rho$  the density.

Subscripts u and b denotes unburned and burned gases respectively. The expression above does not take into account the component of the normal flame propagation due to local flow velocity. If this is to be considered, the flame speed will be the vector sum of the actual flame speed  $v_f$  and the local flow velocity. In the Ricardo E6, according to measurements of local flow velocities performed by Cole and Swords (1979) using laser doppler anemometry, mean values of flow velocity near the spark plug and near the time of ignition are small for the range of speeds considered in the present work. Accordingly the effect of the local flow velocity component can be regarded as of no influence on the flame speed.

### 3.5 Data acquisition and analysis system

The events taking place in the cylinder of a spark ignition engine are in essence of a transient nature which cause difficulties for the development of a satisfactory measuring and data acquisition system.

The main difficulty is the consistency of the data once the probability of obtaining the same data is extremely low due to the high degree of cyclic variation in the engine operation. The fact that the engine can run at high rotational speed only aggravates this situation.

Despite the fact that an individual value for a certain variable may never be repeated, the events are essentially cyclic which makes them highly suitable to be accessed by

means of a computerised process.

The use of a microcomputer as a central element around which all the data acquisition system is built is very convenient. Nowadays, they are cheap, highly reliable, fast in sampling and easy to operate. The use of microcomputers to collect data from scientific instruments necessitates digitisation of the signal. Therefore the signal must be sampled at regular intervals and the values stored.

The microcomputer itself is not fast enough to be used directly as a digital storage device for high frequency signals, however, for comparatively low frequency signals of the order of a few kHz, a fast converter can be used to capture the data.

In view of the transient nature of the events within the cycle, it is necessary to ensure that the same point in a cycle is added to the ensemble for each cycle, which means that full cycles must be sampled and that the correct number of crank angle pulses are obtained during a certain elapsed time, for example, from IVC to EVO. Only if this condition can be assured can the data be averaged over many cycles.

The usual approach to obtain samples equally spaced in time is to initiate the sampling process by means of an external signal which acts as a trigger. The most simple and accurate way of doing this is by means of a slotted disc fitted to the crank shaft of the engine as described in section 3.5.7.

All the signals are then obtained as a function of the crank shaft position. Accurate timing is crucial since additional time-dependent effects, velocity fluctuations for example, may occur within certain cycles or between cycles, especially with single cylinder engines.

In this work, the data acquisition system was constructed around a BBC microcomputer, with 64K RAM memory. The machine itself is a very well known piece of equipment, extensively used in laboratories to control scientific experiments.

Two interfaces were utilised to collect data from cycle to cycle and feed to the computer. Pressure signals from the pressure transducer were gathered by means of a UNILAB microcomputer interface, manufactured by Unilab Ltd, as seen in Fig 3.7, directly connected to the user port of the BBC microcomputer.

Time signals from the photodetectors and knock time signals were gathered by a purpose-built interface, Fig 3.8, a description of which is found in section 3.5.4.

A description of the sampling process will be given in sequence. A single slot in the slotted disc, positioned at 213 DCA after TDC, triggers the system for data acquisition. The UNILAB interface is commanded to collect signals from the pressure transducer-charge amplifier and transfer them to the BBC microcomputer. The next trigger initiated is the spark signal. Here, the signal cannot be obtained directly from the magnetic transducer that triggers the spark. The reason is that there are two sparks per cycle in the Ricardo E6, one before TDC in the compression stroke and another before TDC in the exhaust stroke.

In order to eliminate the undesirable spark a photodetector was mounted to pick-up signals from the camshaft that correspond to an angle of the crank shaft of approximately 250 DCA. This signal commands the interface that the next signal from the ignition system corresponds to the useful spark.

The signal originating from the spark system is used to

trigger the clocks in the second interface. In this interface, three clocks were used. The first counts the time from the spark event until it is disabled by the signal emitted by the photodetector associated with the first laser beam, the one closer to the spark plug. The second counts the time which has elapsed from the time the spark occurred until the second signal arising from the photodetector which corresponds to the more distant laser beam. The third clock relates to the knock detector device described in section 3.4.6. All these signals are sent to the BBC via the appropriate interface and processed by specially-written software.

The passage of the single slot between the corresponding photodetector resets the system which is then ready to sample another cycle.

The data acquisition system comprises the following elements

- .laser tube and power supply
- .beam splitter
- .photodetectors
- cylinder pressure and flame travel time interfaces

3.5.1 The laser utilised in this work was a He-Ne gas laser manufactured by Rofin-Sinar from West Germany. The model was the RSK-20 with an output of 2 mW and is depicted in Fig 3.9. The complete operating laser system is mounted in an aluminum tube of 35 mm diameter with the exception of the power supply that comes as a separate unit.

3.5.2 The beam splitter, Fig 3.10 made by Malvern Instruments in UK, was an adjustable unit type RF 307 incorporating a polarization unit. The advantage of using an adjustable unit is that the distance between the beams can be continuously variable and the direction could be

changed from convergent through parallel to divergent.

These could be achieved by traversing or rotating the prism mounted on a precision mechanism which provides for the adjustment of beam separation and beam angles. This unit was mounted directly to the laser through an adjustable aluminum adaptor. The system was modified by machining the portion of the table in contact with the movement screw responsible for the translational movement of the saddle, in order to be able to obtain distances between the laser beams smaller than 3 mm.

3.5.3 The photodiodes employed in other similar studies in sensing the deflections of the laser beam were position-sensitive detectors, invariably of the type UDT PIN SC-10 manufactured by United Detectors Technology, USA. This type of detector gives three signals; one that is proportional to the intensity of the light that falls on its active surface and the other two are signals that are proportional to the displacement of the beam spot in the X and Y directions.

The signals then have to be amplified by means of a special amplifier to ensure that the deflection signals are independent of any changes in the beam intensity. Swords et al (1982) and later Kalghatgi (1985) used such equipment to measure the flame propagation velocity in a spark ignition engine.

In the present work, several different types of photodetector have been tested during an initial phase of the experimental programme. The final choice was a planar silicon light sensor, model LS 602, manufactured by Texas Instruments. As the external dimension of the photodetector was very much reduced, approximately 1.2 mm diameter, it was possible to mount two of them one for each laser beam, in a support placed directly in front of



the exit window of the spacer.

This permitted the abandonment of all the intermediate handling optics, always an expensive part of the equipment. The two photodetectors were connected by screened cables, 4 m long, to the event timer interface where two LM 311 N comparators condition the signals. To minimize the time delay caused by the capacitance of these cables two load resistors are also necessary.

The values indicated in the diagram of Fig 3.11, gave a satisfactory compromise between speed and sensitivity. Each detector was positioned as stated above, to intercept one of the undeflected laser beams. The signal line is held at zero volts by the current in the detector. When a beam is deflected by refractive index changes within the cylinder and no longer falls on a detector, the signal line rises to a maximum of 2 volts, the actual values depending on the residual light level at the detector.

As the comparator reference voltage is one volt, the first time the signal level is greater than that value, it will result in a pulse being fed to the logic circuits and an event to be registered. Once the system has been armed it will react when the input first exceeds the one volt comparator reference level.

Contamination of the inner surfaces of the combustion chamber windows had two effects on the laser beams, namely attenuation and scattering. It was found that sufficient attenuation of the beam intensity prevented the detector from working satisfactory, forcing the signal line to a value below that of the comparator signal reference. Conversely, when the beam was deflected, typically by two diameters, scattered light, when sufficiently intense, held the signal below the threshold. In both cases detection failed. The first case could only be prevented by constant

cleaning of the windows. To avoid the inconvenience of the second case, a polarizing filter with continuous variation of the degree of polarization was used.

#### 3.5.4 The event timer interface

The block identities of the event timer interface circuit are shown in Fig 3.12. The whole interface operates synchronously, using a one megahertz two-phase clock. The blocks labelled SC0, SC1, SC2, SC3, SC4 represent five synchronising circuits each of which produces a single pulse output locked to the internal clock when the relevant external event occurs.

Next comes the counter which is 16 bits long (65536) and can be reset to zero. The output of the counter is connected to the input of all three 16 bit registers. Each register is capable of being loaded with a binary number and can store it until it is overwritten or cleared. When a register is addressed to read its contents, its output assumes the stored value, otherwise its input is open. In this way, they can all be connected to a common bus going to the computer. Practical constraints insist that each register is read in 4 bit sections.

Because of the strong possibility of electrical interference causing false readings input, SC1-4 locks out after an event and requires re-arming to make it receptive to a further event. Input SC0 is always receptive. The events assigned are as follows

|       |     |       |         |         |       |
|-------|-----|-------|---------|---------|-------|
| Event | IVC | Spark | Laser 1 | Laser 2 | Knock |
| Input | SC0 | SC1   | SC2     | SC3     | SC4   |

When the inlet valve closes, the output of SC0 re-arms SC1 and sets the registers to zero.

When the spark occurs the output of SC1 sets the counter to zero and re-arms SC2, SC3, and SC4. If and when any of the other three events occurs, the consequent pulse output loads the assigned register with the time of its occurrence in microseconds, measured from the spark event. This arrangement of input re-arm ensures that the false spark produced at the end of the exhaust stroke is ignored; that the inputs for the time variable events are open only when the events are expected and that if an event does not occur, its register will contain zero rather than the previous reading. The computer reads the registers and stores the data during the exhaust and induction parts of the cycle. The electronic diagram of the timer event interface is given in Fig 3.13.

#### 3.5.5 The ignition detector

The Ricardo E6 engine is fitted with a high voltage discharge ignition system as outlined in section 3.2.4. In this arrangement a capacitor charged to 350 V is discharged into the ignition coil primary winding via an electronic switch to produce a spark.

In order to produce a marker pulse as close in time to the spark as possible, without carrying electrical interference to the interface, a specially designed electronic circuitry was utilised. A resistance was placed in series with the capacitor discharge path to provide a few volts to drive the input of the opto-isolator. Because the discharge current is oscillatory, a diode is used to protect the isolator from damaging reverse voltages. The isolator output switches on the ZN 3702 transistor to give a standard pulse for the interface. A schematic drawing of the circuit utilised can be seen on Fig 3.14.

#### 3.5.6 The knock sensing device

In chapter 1, several methods utilised to detect knock were presented and in this section the method employed in this work is described.

The system is based on the attenuation of the low frequency components of the pressure wave given by the pressure transducer output and amplification of the components existing in a frequency band extending from 5.5 kHz to 6.5 kHz. Further circuitry results in output pulses of suitable amplitude to feed the interface unit for each burst of signal within the pass band of the knock detector.

The cylinder pressure signal output of the charge amplifier provides the knock detector input. The input stage is a variable gain, frequency selective amplifier, where frequencies below 1 kHz are rejected by the system. The collector load of TR1, in Fig 3.15, is a resonant circuit, L and C2 are tuned to the 'ringing' frequency expected from the cylinder under knock conditions. The bandwidth is set by R3.

The envelope of the 'ring' is recovered by the emitter-follower-rectifier TR2, and the high frequency components are attenuated by the filter R6 and C3. The envelope signal drives TR3 which, with TR4 forms a Schmitt trigger. The associated resistor values are chosen to give a trigger threshold of about 1 volt, hysteresis of 0.3 volts and output swing of 0 volt to 4 volts.

### 3.5.7 The crank angle pulse generator

It is useful to associate the values of the pressure developed in the cylinder of the internal combustion engine, with the angular position of the engine crank shaft because each angle can be associated with a volume of the combustion chamber, characteristic of the engine

geometry. Furthermore, for a constant rotational speed there is a constant ratio between the time and the angular crank shaft position.

In the present work, the angular position of the crank shaft was measured by means of a slotted disc having 180 slots equally spaced every 2 DCA, as described earlier. The source of the signal was a light emitting diode (LED) placed at one side of the slotted disc while in the other side a photo-sensitive transistor creates an impulse from the light beam received from the LED each time a slot is placed between them.

The origin of the angles, the angle at which the system starts to record the pressuer signals, was taken as the IVC which is equal to 213 DCA in the Ricardo E6. The impulse for this origin angle is given by a second pair of light emitter / photodetector placed at the side of the 2 DCA emitter / photodetector and actuated in a single slot of the disc. A third individual slot identifies the TDC.

Adjustment is made by releasing six bolts on the flange that supports the disc which then becomes free to rotate. The TDC center can be adjusted by moving the emitter / photodetector support relative to the disc. TDC was verified with the engine disassembled using the dial gauge method. A schematic diagram of the electronic circuitry is given in Fig 3.16. The mechanical components are depicted in Fig 3.17 and the system installed in the crank shaft of the Ricardo E6 is represented in Fig 3.18.

### 3.5.8 Cylinder pressure

The measurement of the cylinder pressure by means of a Piezo-electric transducer is a very straightforward

process. The pressure can be obtained relatively easily using standard experimental methods with calibrated transducers.

In the present work, a Kistler miniature pressure transducer was utilised to gather information on peak cylinder pressure and to permit an evaluation of the cyclic variation of the cylinder pressure during the duration of the engine cycle.

Piezo-electric systems for cylinder pressure acquisition presents some advantages over other systems, namely resistive, capacitive or inductive systems such as

- .good linearity
- .high signal/noise ratio
- .compactness
- .high natural frequency
- .good signal resolution

These characteristics have made the piezo-electric system the only choice for pressure data acquisition in recent years.

The piezo-electric element, normally consisting of a quartz cristal, produces an electrical charge which is proportional to the pressure exerted. The charge produced is very small and therefore needs to be amplified to be of any usefulness. This electrical charge produced by the transducer is then fed to a charge amplifier which transforms it into a voltage proportional to the pressure. The pressure signal originated in this way, was fed to the UNILAB interface and, after digitisation, the signal can be read by the BBC computer and stored in floppy discs for subsequent retrieval.

The transducer was mounted flush in an adapter which was then mounted in place of the diesel injector nozzle in the cylinder head, diametrically opposed to the spark plug. The



face of the transducer was in contact with the flame in the cylinder once no flame trap was provided.

The miniature Kistler transducer requires no water cooling and according to the manufacturer is capable of working reliably under temperatures of up to 350° C and withstands intermittent flash temperatures of up to 2500° C. No calibration of the transducer was performed because the transducer was relatively new, and therefore it was decided to accept the calibration performed by the manufacturer. Fig 3.19 represents the pressure transducer and its adapter.

The pressure samples were taken and recorded each 2 DCA, from the start of the sampling, chosen at IVC as a matter of convenience, and terminating 250 DCA later, giving a total number of 125 samples of pressure.

### 3.6 Experimental procedure

The experimental work consisted primarily of the examination of the flame propagation times using the differential laser refraction technique.

The examination of the onset of knock as an indication of abnormal combustion has also been considered however. For both fuels under study, cyclic variation has also been examined during the operation of the engine.

#### 3.6.1 Parameters studied

Two types of fuel have been investigated  
.hydrated ethanol, 96 % ethanol by volume  
.iso-octane

The other parameters that have been varied during the

experiments are as follows

- .compression ratio, CR = 10 and 12 : 1
- .engine speed, N = 600, 900, 1200 rpm
- .equivalence ratio,  $\phi$  = 0.8, 0.9, 1.0
- .spark timing, SA = 10, 15, 20 DCA BTDC

### 3.6.2 Warming up the engine

To standardise the starting and warming up of the Ricardo E6, the following procedure has been employed after starting the engine

- .step 1 : running of the engine for a period of twenty minutes at 900 rpm, 25 % load
- .step 2 : the speed was increased to 1500 rpm and the engine run for a further twenty minutes
- .step 3 : the speed was adjusted to that required for the test condition, full load and WOT, for a further fifteen minutes
- .step 4 : with the sump oil temperature and cooling water temperature at nominal values, the mixture strength was set up
- .step 5 : the engine was then readjusted, before any measurement taken to the conditions of test.

This procedure has been applied whenever it was necessary start the engine.

### 3.6.3 Setting the mixture strength

The Ricardo E6 engine is fitted with a variable main jet carburettor. The mixture strength is varied by adjusting a tapered needle valve which allows the mixture to be varied between wide limits during the test. The ratio is determined by measuring the time for a given amount of fuel to be consumed. The air flow was measured with the Alcock viscous flow meter supplied with the engine. For

details see appendix 4.

#### 3.6.4 Setting the spark timing

The ignition system adopted in the Ricardo E6 is of the capacitor discharge type. The ignition sensor is mounted in a ring concentric with the crank shaft, whose movement permits the ignition timing to be varied over a range of 60 DCA. The ignition point is indicated by means of a graduated scale each 5 DCA. The spark advance has been manually adjusted by means of a lever actuating over the above mentioned ring.

#### 3.6.5 Setting the compression ratio

It is possible with the E6 engine to vary the compression ratio while the engine is running. The cylinder clamp is released to allow the cylinder nut to be rotated by means of the controlling handle: Clockwise to lower the ratio and anti clockwise to raise it. A micrometer head is used to measure the cylinder travel.

##### 3.6.5.1 Disc combustion chambers

The procedure described above has been used to vary the compression ratio when the disc type combustion chambers were tested. Due to the fact that a spacer has been assembled between the cylinder head and the cylinder barrel, it is not possible to take direct readings from the micrometer in order to determine the compression ratio. A graph has therefore been plotted which takes into account the thickness of the spacer. The method is presented in appendix 6.

### 3.6.5.2 Bowl-in-piston combustion chambers

The compression ratio of this combustion chamber could not be varied in the same way as the disc chambers, since squish distance or bumping clearance was assumed to be fixed at 1 mm. The compression ratio with this type of combustion chamber was made variable by varying the volume of the bowl during manufacturing, which implies that the engine had to be dismantled whenever a change in compression ratio was necessary. Small adjustments, using the variable compression ratio mechanism were revealed to be necessary in order to set the correct squish height.

### 3.6.6 Combustion chamber change

In order to test two different types of combustion chamber, the engine had to be dismantled each time a new type of combustion had to be fitted. The procedure adopted has been that recommended by the manufacturer, namely the piston and connecting assembly has been removed from the engine through the bottom of the crank case. With the normal piston this is a straightforward procedure having no requirement to disassemble all the head of the engine.

However, using the modified piston, the total height with false crown was above the limit that could pass between the crank shaft and the bottom end of the cylinder. Consequently, the whole head had to be dismantled, and the piston had to be removed from the top, after a laborious process to separate it from the connecting rod.

The combustion chamber could then be dismantled from the piston and the new one fitted in place, the bolts being wired one by one. The whole process could not be performed in less than four to five hours. The squish height was verified before reassembly utilising a micrometer to measure depths.

### 3.6.7 Other measurements

Temperature measurements, during each sample of 100 cycles were read from the display of the thermocouple monitor and recorded on cards for later utilisation.

Load was taken for each condition of test and recorded, allowing the calculation of the brake torque and power.

3.6.8 Knocking was observed with a CRT oscilloscope and the time of onset measured and recorded on floppy disc simultaneously with the times for the deflections of the laser beams due to the flame front passage.

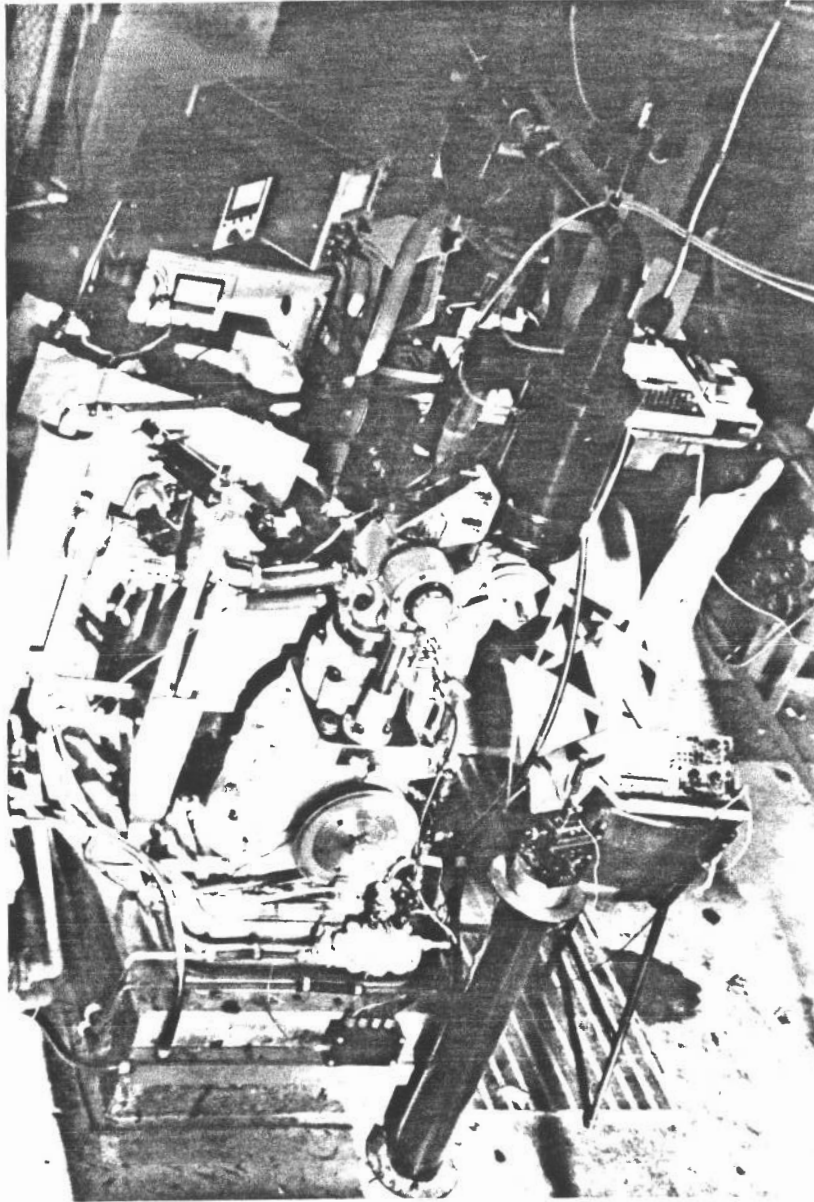


Fig. 3.1 General view of the laboratory, showing the engine and the instrumentation.



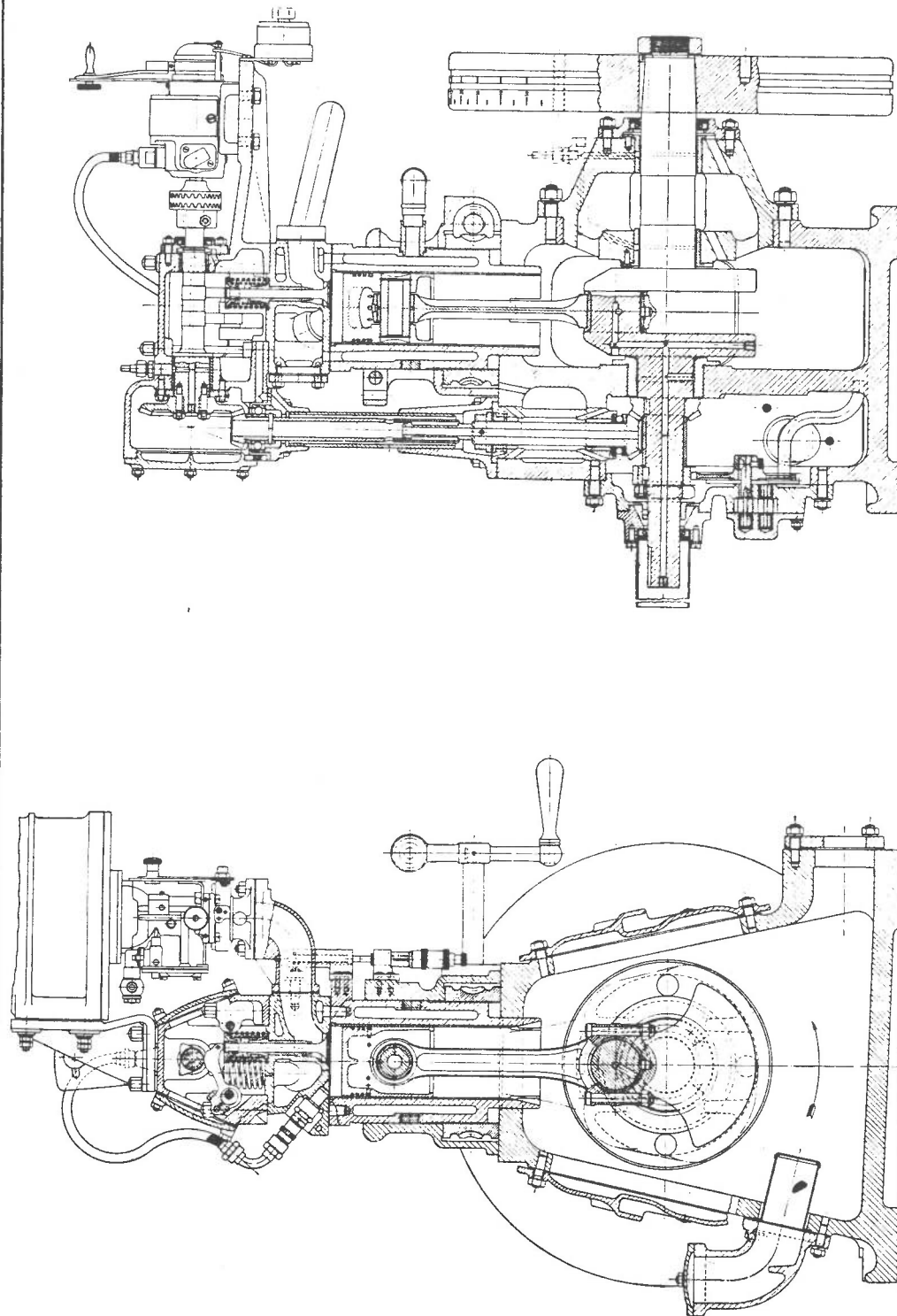


Fig. 3.2 Sectional views of the Ricardo E6 engine.

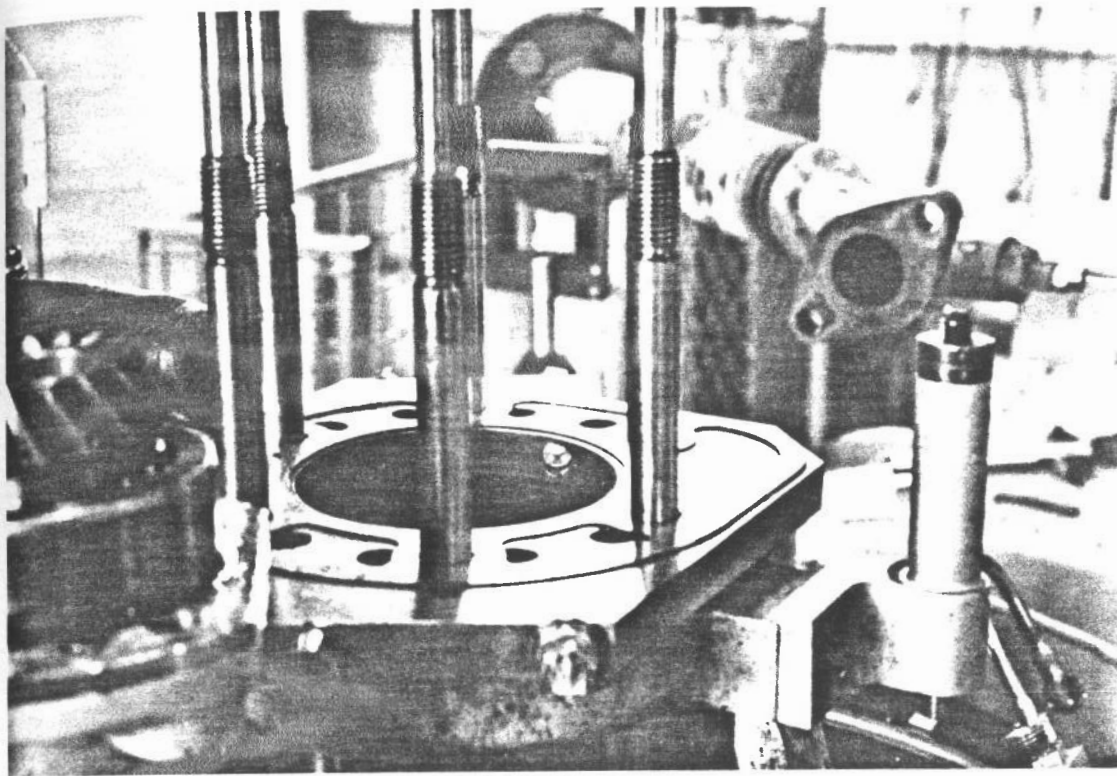
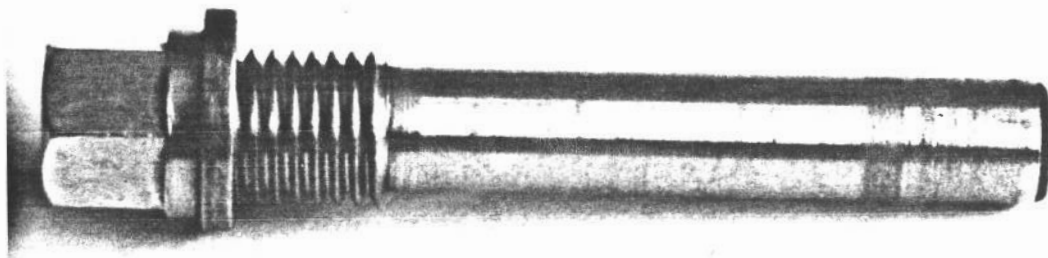


Fig. 3.3 The spacer with the quartz window holder.

Fig. 3.4 The quartz window holder. Window is seen in the right extremity of the tube.



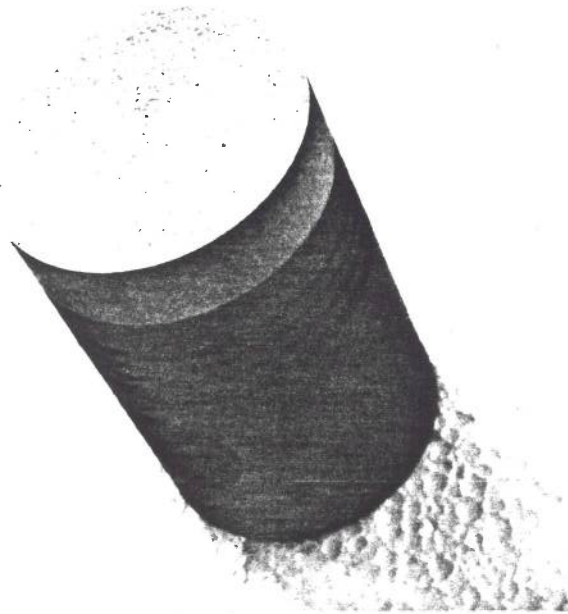
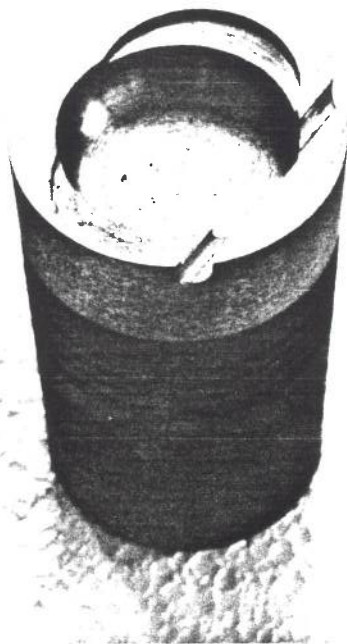


Fig. 3.5 The Ricardo E6 piston showing the false head for the cylindrical chamber.

Fig. 3.6 The Ricardo E6 piston showing the false head for the bowl-in-piston chamber.



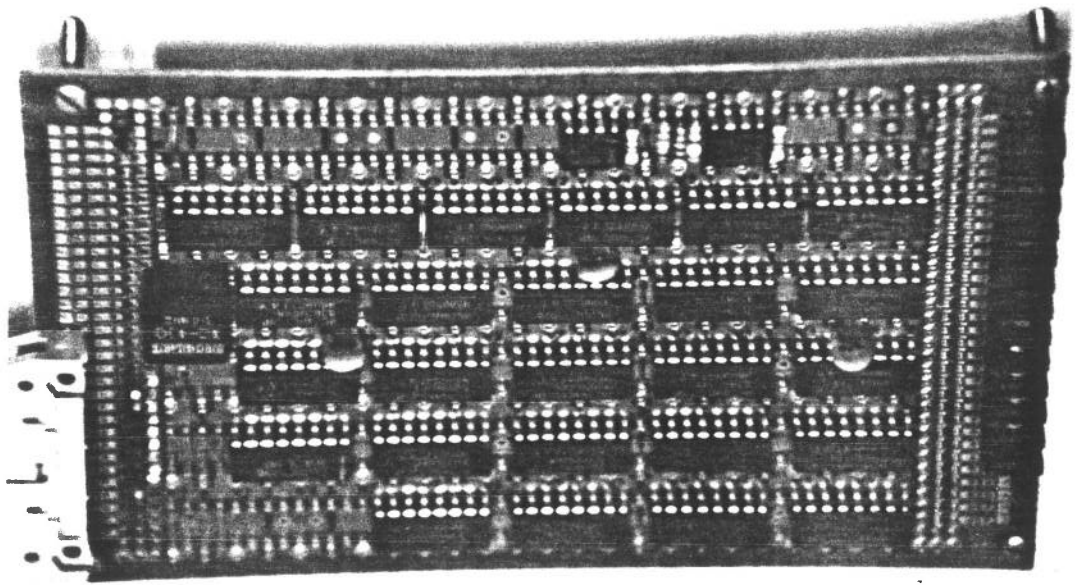
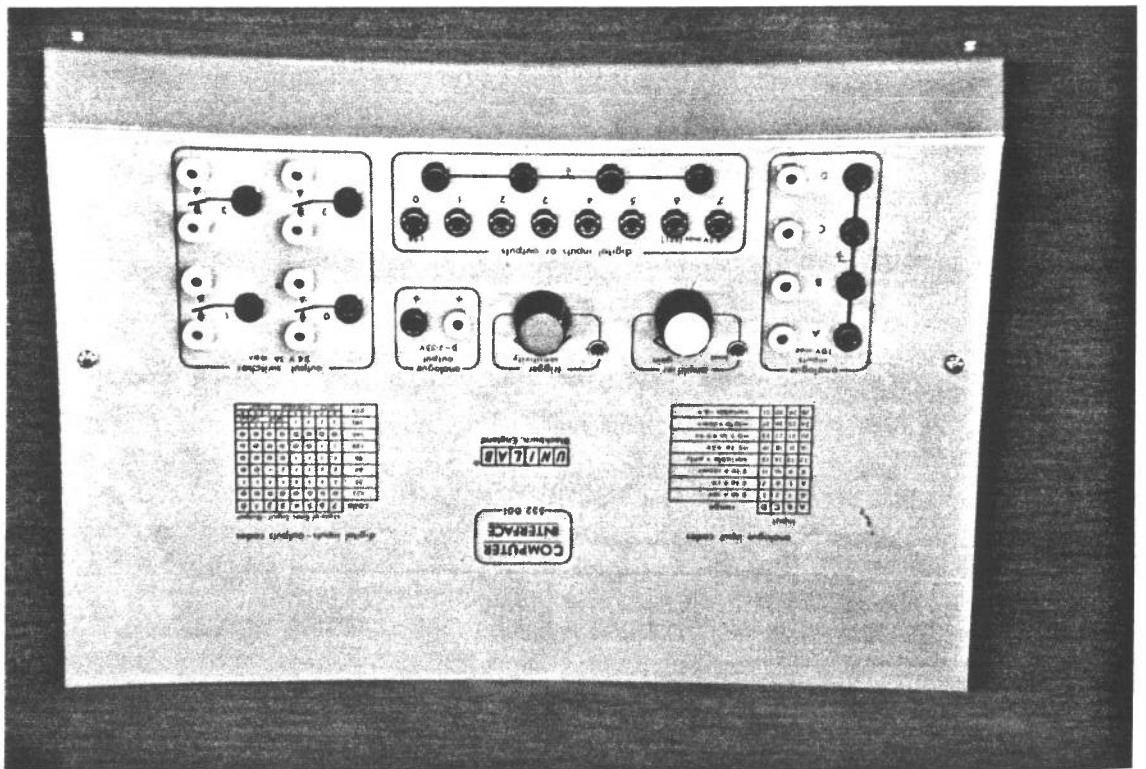


Fig. 3.8 The interface built to gather the laser pulses

Fig. 3.7 The UNILAB interface .



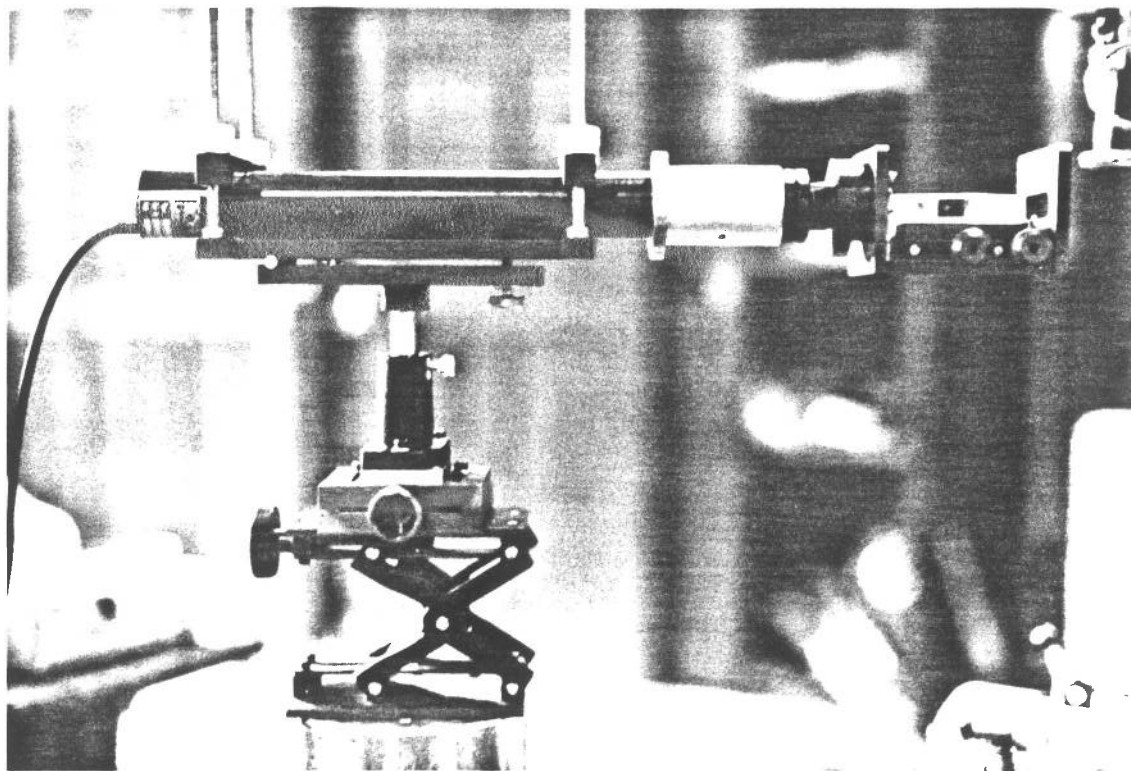
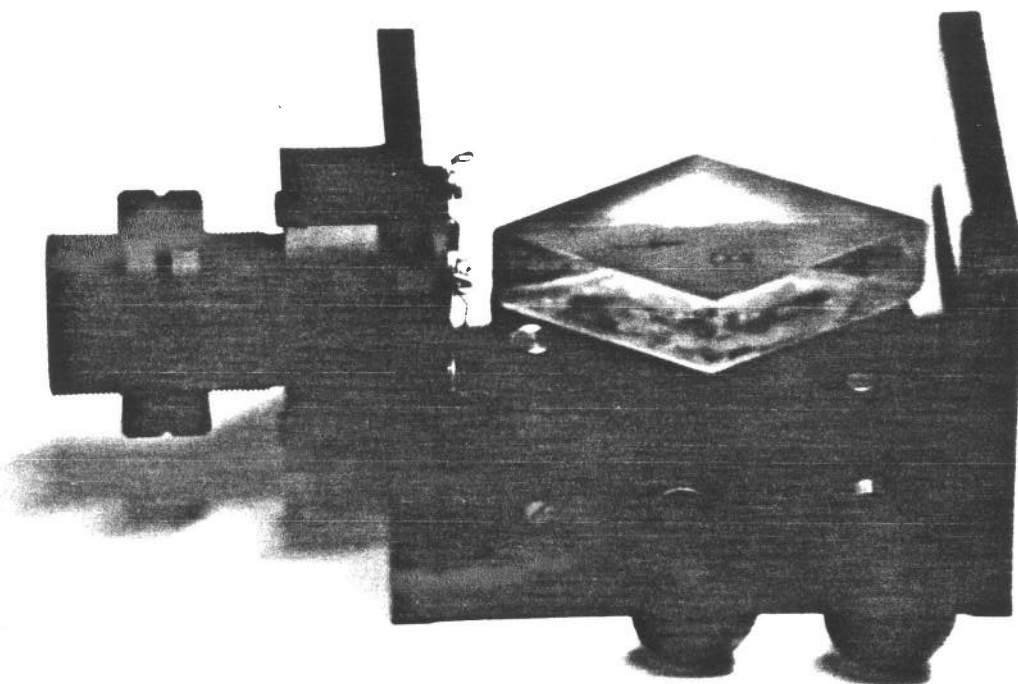


Fig. 3.9 The laser tube and beam splitter.

Fig. 3.10 Detail of the Malvern beam splitter.



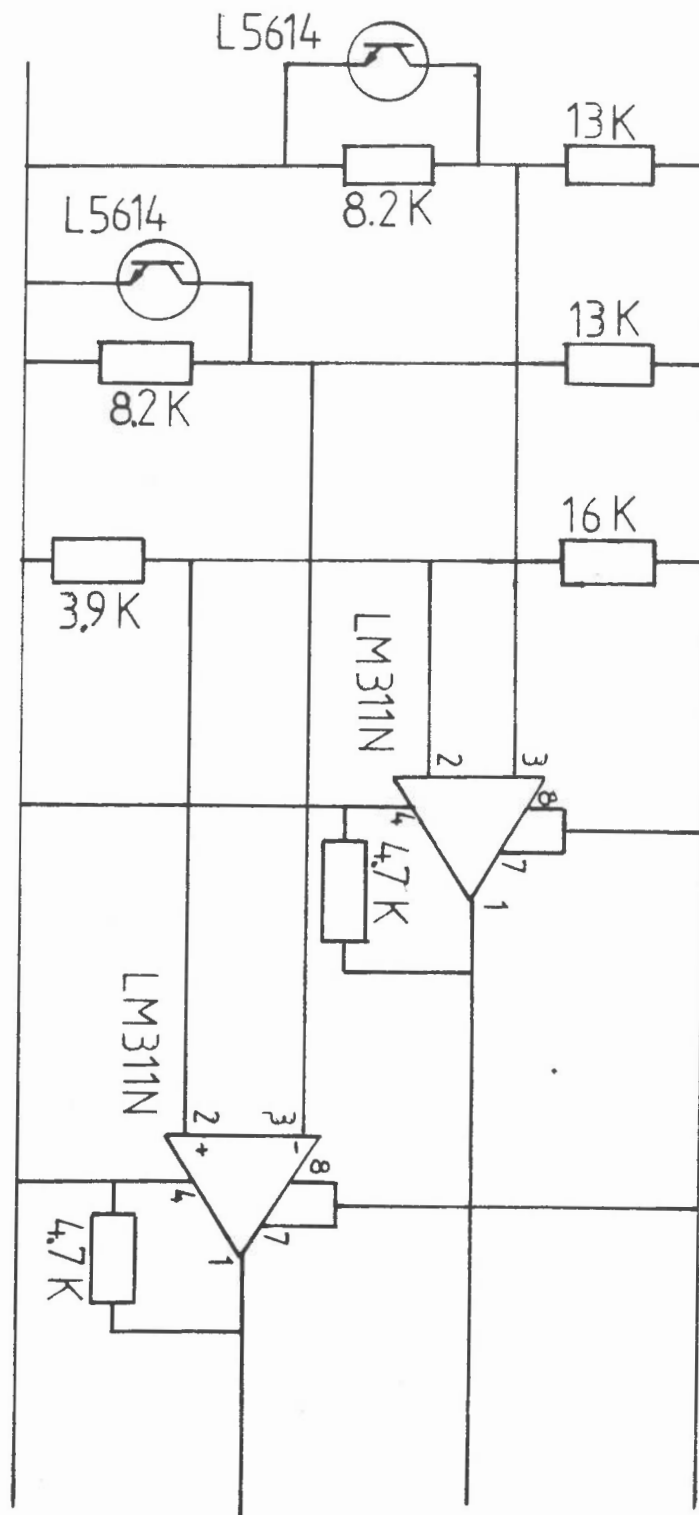


Fig. 3.11 Circuit diagram for the laser detector



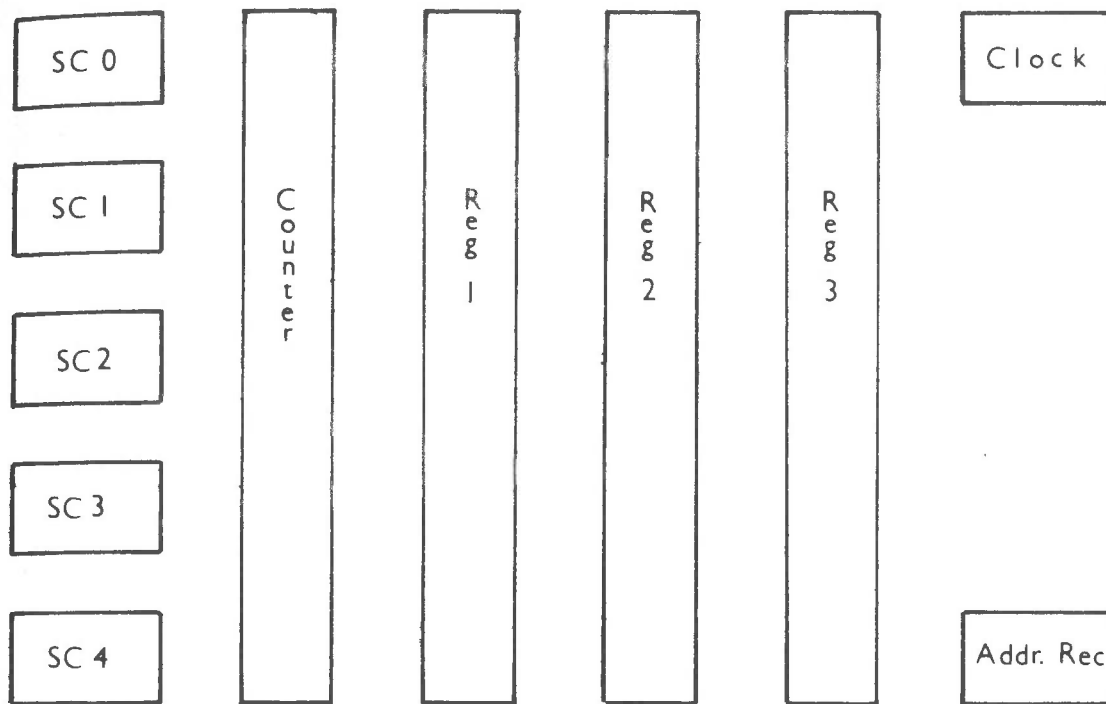


Fig. 3.12 Block diagram for the event timer interface.



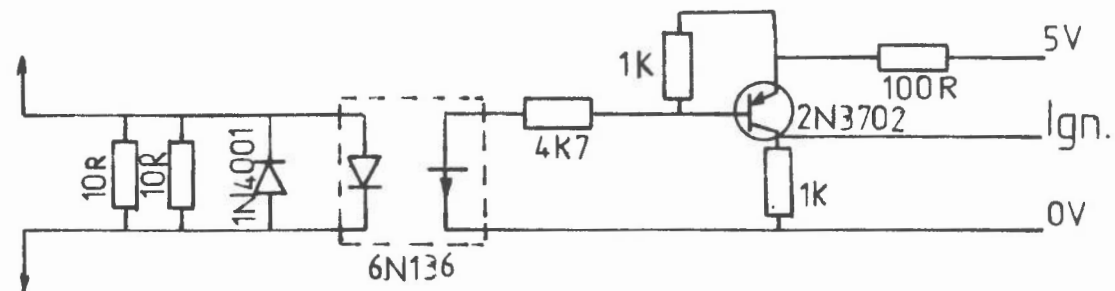


Fig. 3.14 Circuit diagram for the ignition detector

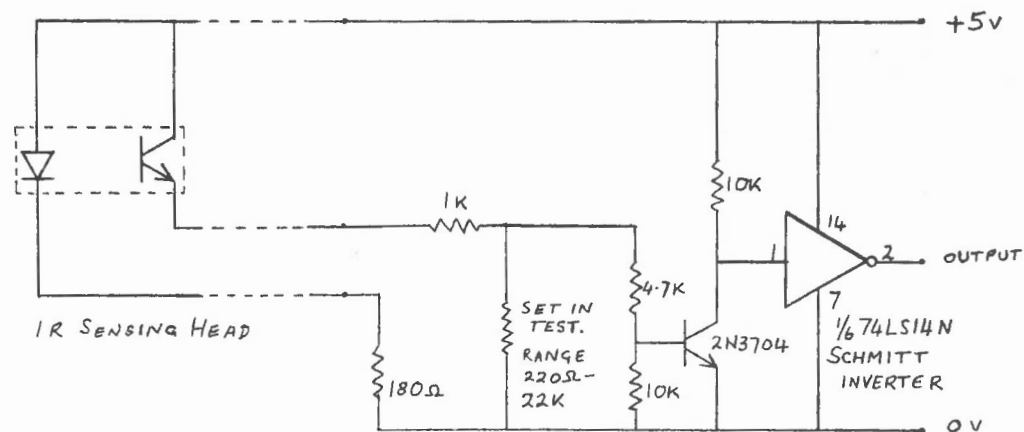


Fig. 3.16 Circuit diagram for the crank angle pulse generator.

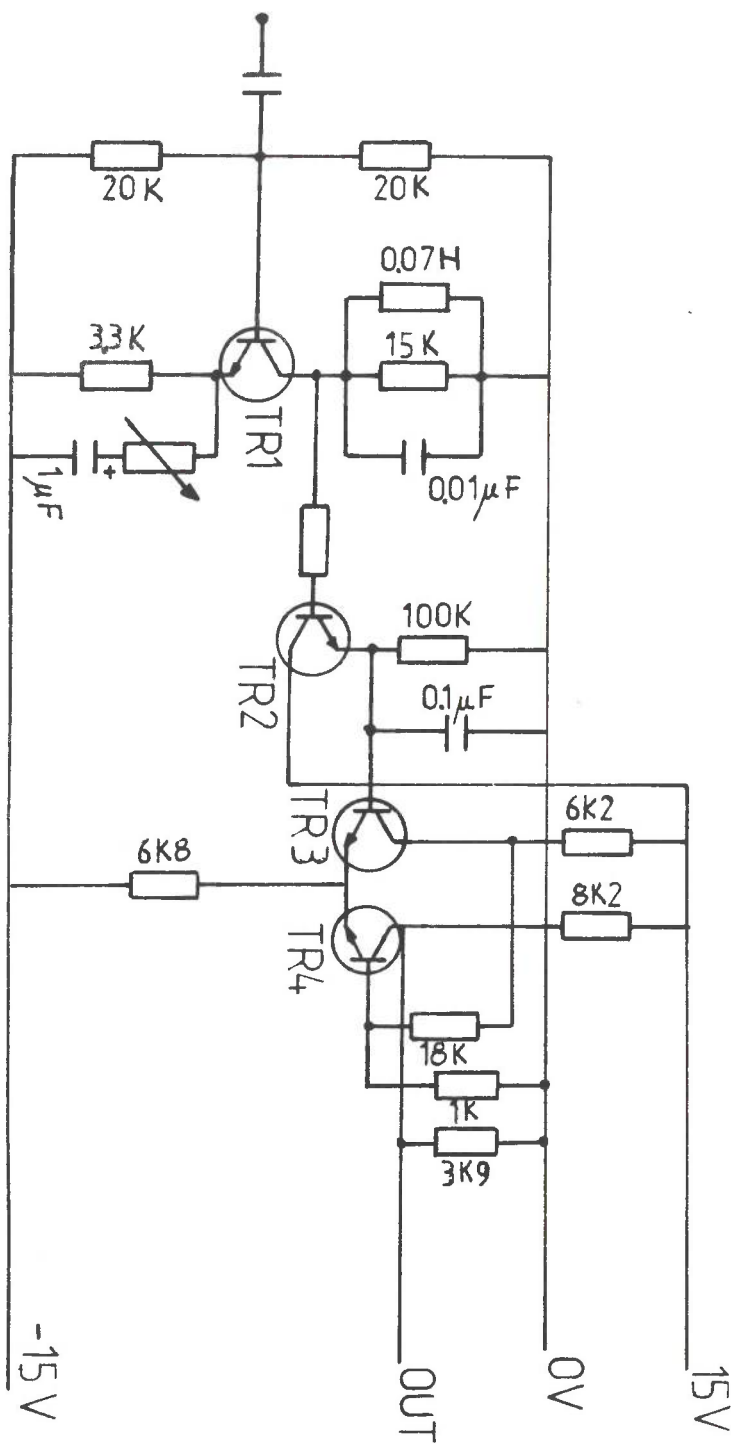


Fig. 3.15 Circuit diagram for the knock sensing device.

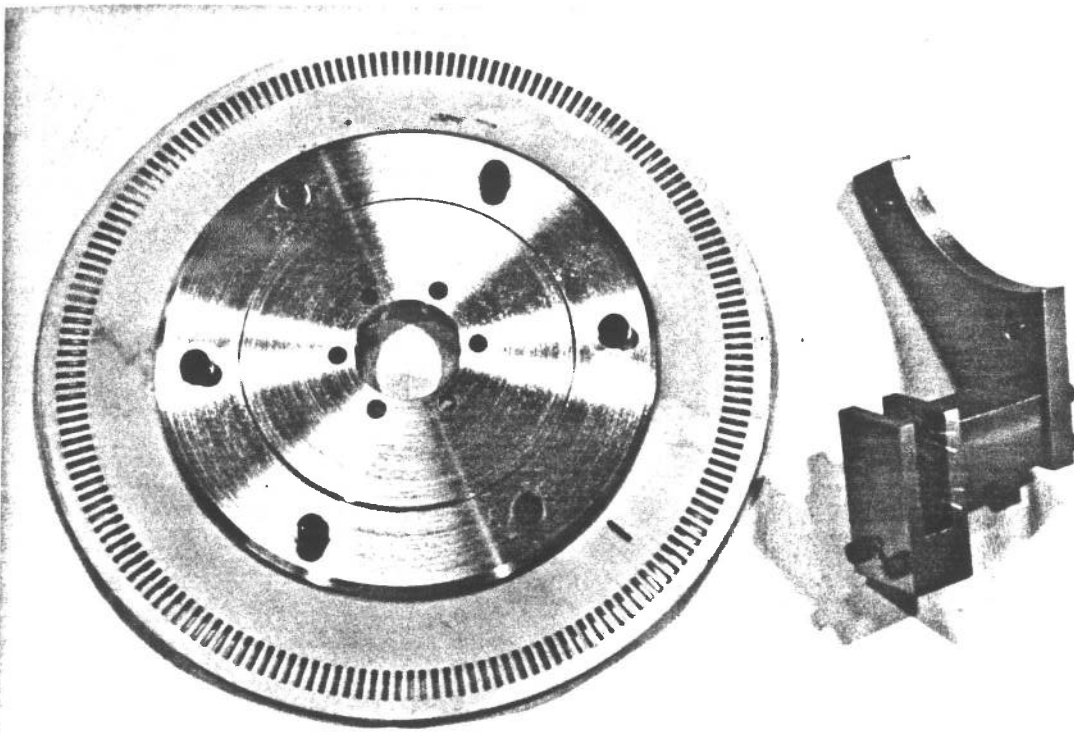
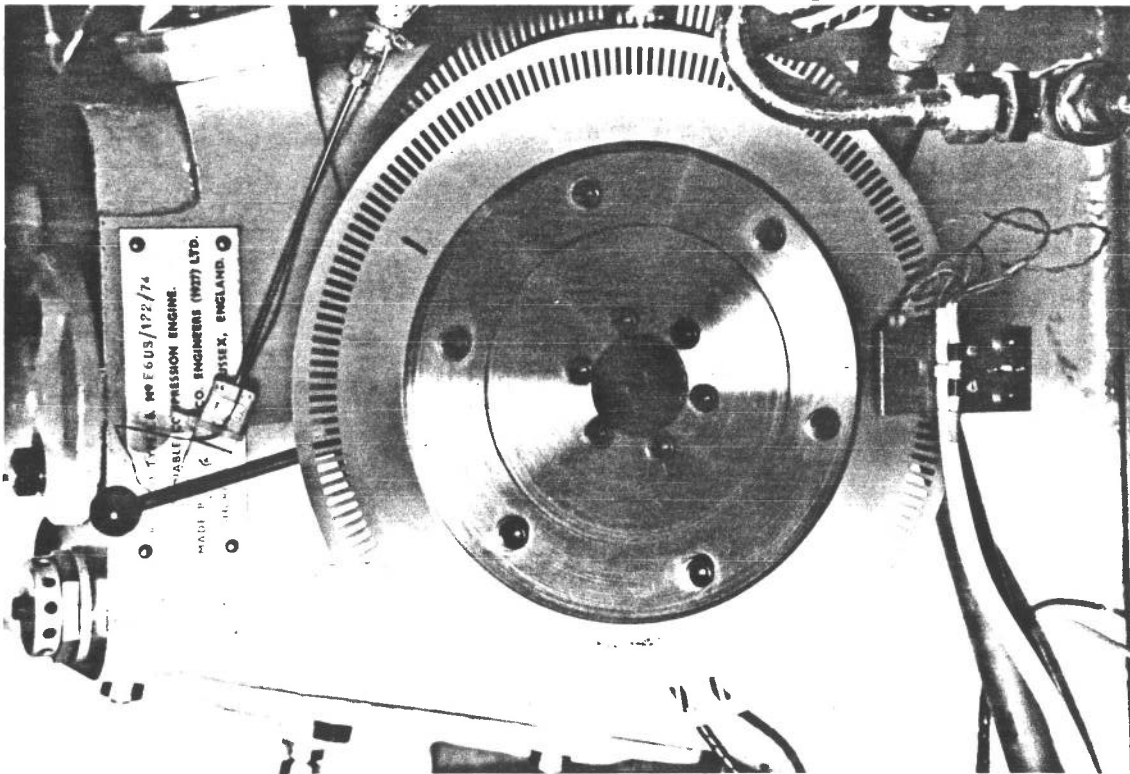


Fig. 3.17 The components of the crank angle generating pulses system : slotted disc and photodiodes support.

Fig. 3.18 The system to generate crank angle pulses as assembled on the engine.



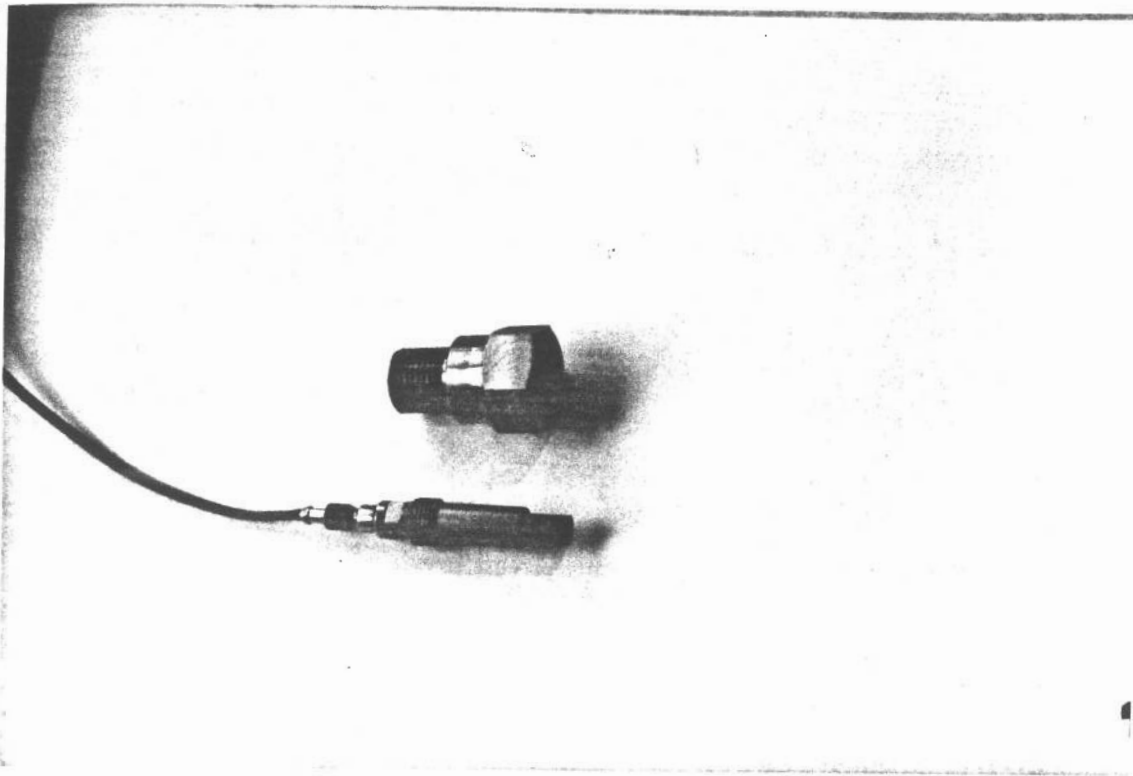
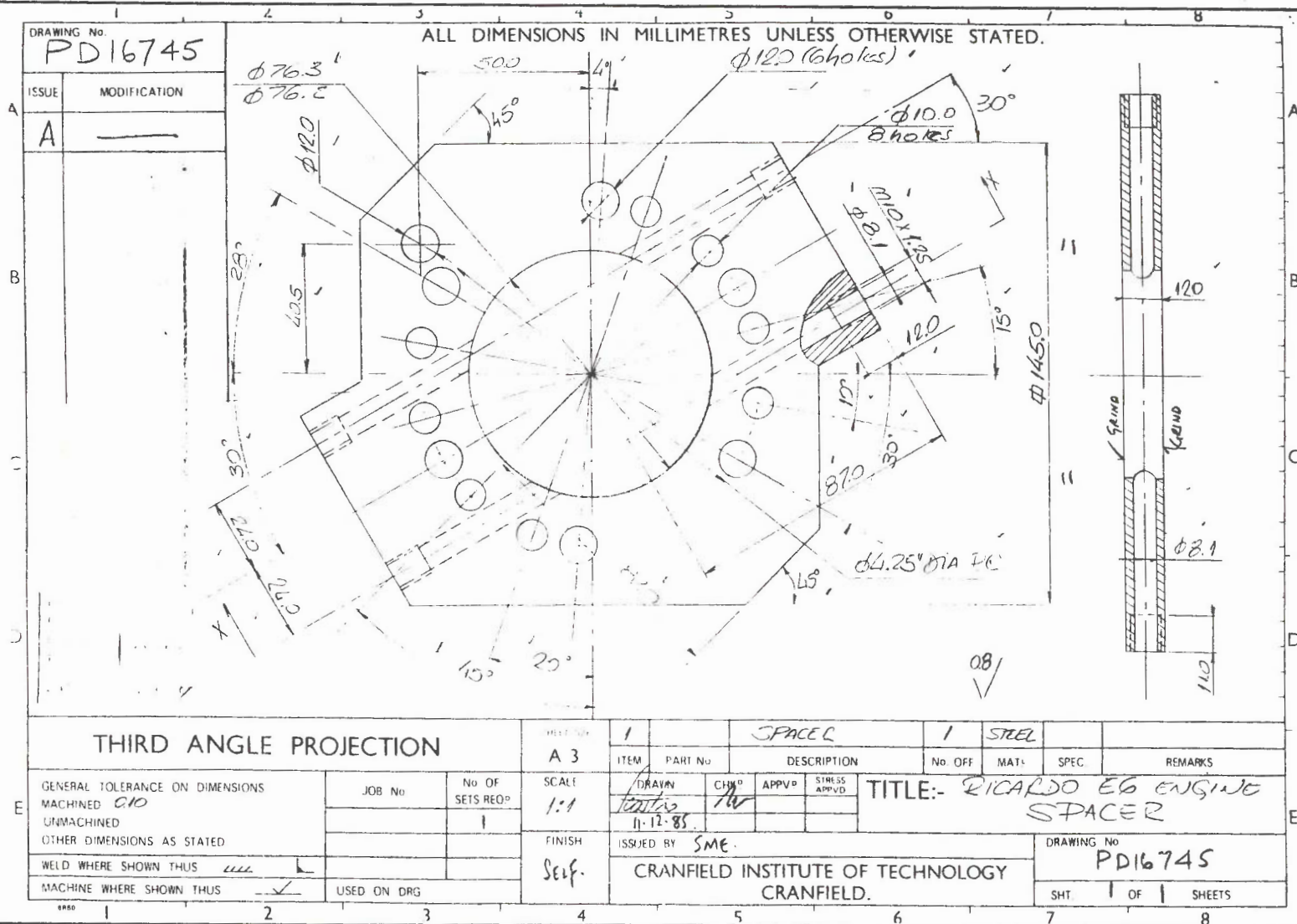


Fig. 3.19 The pressure transducer and adapter.



Fig. 3.20 Detail drawing of the spacer.



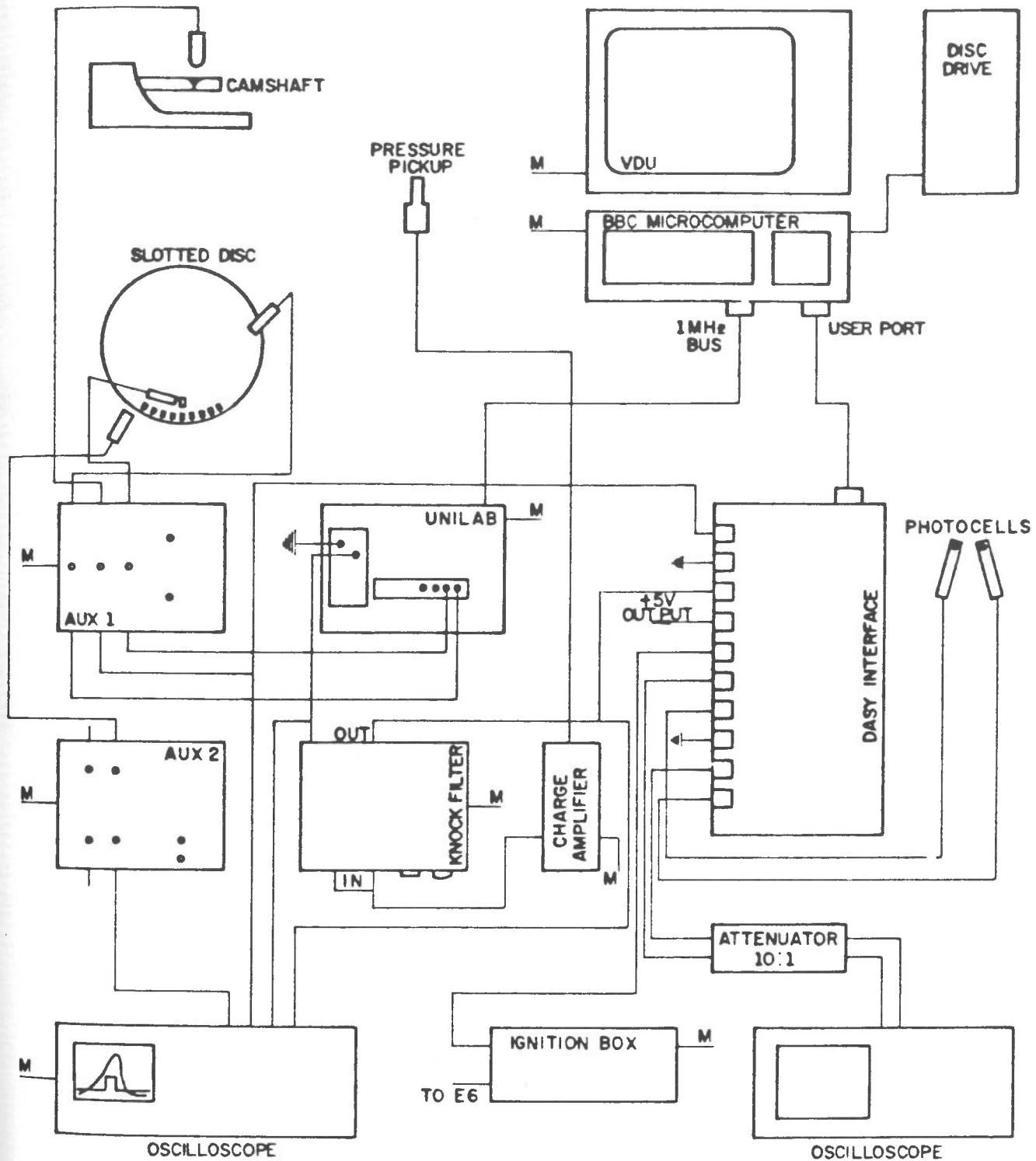


Fig. 3.21 Schematic of the equipment utilised in the present work for data acquisition.

# Chapter 4

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## Analysis of Experimental Results

#### 4.1 Introduction

In this chapter the results obtained during the experimental part of the work are presented and discussed. The aim of the experimental study was to measure the flame propagation velocity in the Ricardo E6 engine using the dual-beam laser deflection technique. The equipment employed to make the measurements of the flames time of passage inside the combustion chamber has been described in Chapter 3.

Two broad categories of data can be obtained during the experiments

- .steady state or time-independent measurements such as the various temperatures of interest during the experiments, namely, exhaust temperature, air intake temperature, oil temperature and water temperature, and the fuel consumption.

- .time-dependent measurements, such as cylinder pressure and flame travel.

The first category of data required only one measurement for each set of parameters selected, with the derived parameters such as brake thermal efficiency, brake specific fuel consumption being estimated from standard expressions existing in the literature, example of which are to be found in the book by Taylor (1980) presented in the Bibliography

The second category of data required the measurement of data through a number (typically 100) of cycles and the results submitted to a statistical analysis in order to determine the ensemble-averaged flame travel times and the standard deviation. The latter is taken as an indication of the degree of dispersion from cycle to cycle of the combustion process.

Two types of combustion chamber were considered namely,

the disc chamber having a compression ratio equal to 10:1 and the Heron or bowl-in-piston with compression ratios of 10:1 and 12:1.

The fuel utilized during the experiments was hydrated ethanol of the type commercially available for motor vehicles in Brasil, having a water content of 15 percent by volume but containing no denaturant. Iso-octane, without any additive was also utilized for both chambers at compression ratio of 10:1 only, to minimize knock problems.

The conditions set for the experiments and the parameters considered for study are presented in detail in Table 4.1.

#### 4.2 General trends

Despite the fact that the measurement of time-independent parameters was outside the scope of the present work, some comments about the general behaviour of the engine based on some of the parameters obtained will be presented in sequence. One important parameter, the fuel flow rate and its measurement will be presented later in Appendix 4.

One parameter, indicative of the efficiency with which the combustion process is occurring inside a given combustion chamber is the exhaust temperature. This parameter has been measured for nearly all conditions of test and some results are presented in Fig 4.1, Fig 4.2 and Fig 4.4. The thermocouple was located at a distance of approximately 50mm from the exit of the cylinder head, in the exhaust pipe. The results correspond to the BIP chamber with a compression ratio of 10:1 operating with ethanol at a spark timing of 15 DCA BTDC. Fig 4.3 depicts the exhaust temperature versus engine speed for the disc chamber.

The decrease in temperature with the diminishing

equivalence ratio may be the result of an increase in the thermal efficiency. When decreasing the equivalence ratio, ie, leaning the mixture, more air is available to burn the fuel supplied, reducing the difference between the heat supplied by the burning of the fuel and the heat effectively utilised during the cycle, therefore, increasing the thermal efficiency.

In Fig 4.2, the differences in the exhaust temperature for the same combustion chamber configuration, with the engine running at 900 RPM is shown. Iso-octane produced a higher exhaust temperature when compared at the same conditions with ethanol. The difference is roughly of the order of  $60^{\circ}\text{C}$ .

Increasing the compression ratio to 12:1, resulted in an exhaust gas temperature greater than that of iso-octane at 10:1, as depicted in the upper curve. At higher CR the temperature attained before combustion is higher, therefore, it can be inferred that the burning velocity will be higher and also the combustion temperature.

As the combustion process terminates before that required for lower CR, the walls will be exposed to burned gases for longer, which in turn implies that a higher rate of heat losses can be expected at higher CR.

Fig 4.3 depicts the exhaust temperature versus engine speed plot. Three conditions of spark advance are indicated for a disc combustion chamber with compression ratio 10:1, operating on ethanol at equivalence ratio of 0.9. The tendency is towards a higher exhaust temperature for higher engine speeds. The reason is that due to the increasing in engine speed, less time is available for the burned gas exchange heat with the cooler cylinder wall and therefore, more heat is released via exhaust gases.

The spark advance determines the course of combustion

process with respect to TDC and affects directly the indicated thermal efficiency. A change in the ignition timing alters the temperature, pressure, and may also change the turbulence of the charge at the beginning of the combustion. Reducing the spark advance causes most of the combustion process to occur after TDC has been reached, thus less heat is used to expand the gas producing useful work and is wasted in the exhaust process, producing higher discharge temperatures.

The phenomenon has been also depicted in Fig 4.4, here a plot of exhaust gas temperature as a function of spark timing for a bowl-in-piston combustion chamber operating with ethanol at equivalence ratio equal to 0.8 and CR equal to 10:1.

The trends are basically the same as described earlier. The temperatures attained at each condition of spark timing and speed are slightly greater for the bowl-in-piston chamber, probably due to the higher cylinder pressures developed by this configuration, as will be shown later.

Fig 4.5 and Fig 4.6 are plots of power versus engine speed. Also depicted in the upper portion of each figure is the brake mean effective pressure (bmep) versus crank angle. Fig 4.5 is for a disc chamber, CR equal to 10:1, spark timing of 20 DCA BTDC, and equivalence ratio equal to 0.9. For the same compression ratio, the power generated when using iso-octane was higher than that produced when operating with hydrated ethanol. The main reason for this resides in the lower heat of combustion of ethanol ( 27 MJ/kg) when compared to that of the iso-octane ( 43 MJ/kg).

The difference can be eliminated and biased towards ethanol if an increase in compression ratio is allowed. Fig 4.6 shows that the power developed when the



compression ratio is raised to 12:1 is greater when running on ethanol, though small. It should be borne in mind that the bowl-in-piston combustion chamber used was not optimized as explained in section 4.3. This is believed to explain the small amount gained in power output. A greater power output should be expected for this type of combustion chamber when its geometry is more favourable to the combustion process, eg if an offset bowl is used, leaving the spark plug electrodes directly inside the bowl.

#### 4.3 Combustion chamber analysis

The combustion process in a reciprocating engine is a decisive phase in the transformation of the chemical energy available in a given fuel into mechanical work. Combustion is influenced by numerous engine design and operating parameters and is also dependent on the chemical composition of the fuel.

The combustion process is mainly dependent upon the turbulent mixing produced by the flowing of mixture through the comparatively narrow sections of the intake system such as intake pipes and valve sections. As this turbulence tends to decay during the compression stroke the combustion process will be greatly enhanced if further turbulence could be produced during the flame kernel growth. This turbulence enhancement process can be achieved by careful design of the combustion chamber, as will be shown later in this section.

The geometry of the combustion chamber also greatly influences the heat exchange process in the internal combustion engine. The effectiveness of this exchange is characterized by the surface to volume ratio. The surface to volume ratio is also an indication of the expected reduction in combustion efficiency due to flame quenching

and the consequent increase in unburned hydrocarbon emissions. Combustion chambers with large values of the surface to volume ratios naturally tend to produce higher concentrations of unburned hydrocarbon.

The surface to volume ratio is increased if the chamber is provided with squish area as a means to enhance turbulence near TDC. Here, a compromise between high unburned hydrocarbon emissions and high degree of turbulence must be taken into consideration when designing the combustion chamber.

The position of the spark plug also plays a major role in the development of the combustion process. It should be placed, preferably, far from the squish area. The reason is that if the spark plug is placed too close to the squish area, the high intensity of turbulence generated in that region near the instant of ignition, will tend to quench some of the flame kernels, thus increasing the cycle to cycle variation of the combustion process. The nearer the spark plug to the center of the combustion chamber, the shorter the path described by the flame front and the greater the flame front area. As a result, the rate of pressure rise will be greater for the central spark plug than for the one placed laterally in the combustion chamber.

Therefore, the localisation of the squish areas far from the spark plug ensures that at the instant the spark occurs, the mixture is relatively quiescent in the region of the spark plug electrodes. As the piston reaches the TDC, the mixture is strongly ejected from the squish area, producing high flow velocities of the order of 10 times or higher than the mean piston velocity, and so ensuring a high degree of turbulence to assist the combustion process, when the flame has already attained the stage of full development.

A high degree of turbulence can also be obtained by tangentially orientating the inlet duct or by preswirling the incoming flow by the use of a helical port. Both systems are capable of generating intense turbulence inside the combustion chamber during intake and are far more common in the diesel engines.

The bowl-in-piston combustion chamber has been initially developed for diesel engines as a means to intensify the swirl effect, thereby ensuring a high degree of fuel-air mixing. Their use in spark ignition engines has increased recently and has led to their competing successfully with the traditional in-cylinder head combustion chambers. Two important features of the bowl-in-piston combustion chamber result. Firstly they reveal a lower octane requirement when compared to other combustion chambers; the second is dictated by economic factors. Cost effective production methods must be established before they can provide a satisfactory alternative to simpler geometries in spark ignition engines.

The Heron or bowl-in-piston combustion chamber appeared in the late fifties, derived from diesel engines, in an attempt to reduce octane requirement through combustion chamber design. This geometry produced particularly good results as reported by Heron and Felt (1950), Caris and Mitchel (1956) and Caris and Nelson (1959) in papers from that time. However there is still some divergence of view as to whether the combustion chamber should be placed in the cylinder head or in the piston or in between both of them.

The mode in which turbulence should be produced in order to ensure a highly efficient combustion process is still open to discussion: turbulence created in the intake manifold during the intake stroke or turbulence created at the end of the compression stroke by controlling the shape of the combustion chamber.

Several recent studies of the performance of the combustion chambers such as those of Overington and Thring (1981), Thring and Overington (1982), Gruden et al (1984) and Engels and Pirault (1984) do suggest that the bowl-in-piston chamber presents the best overall performance.

In the present work, the bowl-in-piston combustion chambers adopted both had a bowl diameter of 53.88 mm and a bowl height of 22.69 mm for the 10 : 1 CR and 18.54 mm for the 12 : 1 CR. The root radius at the bottom of the bowl was 8 mm, and the squish ratio, defined as the ratio of the annular area to the bore area was 50 %. In order to simplify the modelling process, an axi-symmetric bowl was chosen and this solution was extended to the fabricated chamber. This is not the general tendency with this type of combustion chamber. The majority of the designs nowadays prescribe a bowl displaced towards the spark plug, thereby leaving the majority of the squish area in direct opposition to the spark plug as depicted in Fig 4.7.

The channels opened for the passage of the laser beams are thought to have produced a peculiar flow pattern in the vicinity of the spark plug recess, suggesting the formation of recirculation of the flow and destroying the purely toroidal motion inside the bowl.

The position of the spark plug relative to the bowl was not thought to be particular favourable for the development of the flame front. In the Ricardo E6 engine, the electrodes of the spark plug are located in a recess in the cylinder head, 5 mm from the face of the cylinder head exposed to the gases of combustion. This implies that, when the piston is at TDC, the spark electrodes are 6 mm distant from the edge of the bowl. Assuming that the flame has some sphericity, the radius at TDC should range from

5 to 10 mm for a spark advance of 20 DCA. Therefore, the flame front makes contact with the cold walls of the piston at an early stage of development. If this is the case it could explain the higher incidence of cyclic variation presented by this configuration, due to partial quenching of the flame.

The bowl-in-piston combustion chamber presented, for all operating conditions, an increased resistance to knock when compared with the disc chamber at compression ratio 10 : 1. Even at higher compression ratio, 12 : 1, the bowl-in-piston presented negligible tendency to knock, relative to the disc chamber. Possibly, the greater surface to volume ratio of the bowl-in-piston configuration, implies that a larger surface area is available for heat transfer. This, associated with higher turbulence generated by the squish surface avoided overheating of the end gas, thereby reducing the risk of the onset of knock. Another possibility is that the greater flame surface area was responsible for an increased flame speed, thereby reducing the time that the end gases were subjected to higher pressure and temperature, reducing the tendency to knock.

The engine produced an intense knock when operated at low speed (600 rpm) and lean mixture ( $\phi = 0.8$ ) with iso-octane. At higher speed it became unstable, losing power and becoming incapable of sustaining the full load condition. For values of  $\phi = 1.0$  the reverse condition occurred namely, the engine developed a relatively high incidence of knock at higher rpm but could be operated at a stable condition for 600 rpm and 900 rpm.

The engine instability can be identified with a reduction in combustion efficiency. When the mixture is very lean ( $\phi = 0.8$ ), the amount of heat generated by the combustion process diminishes, hence the temperature of the process and the relative content of  $\text{CO}_2$  and  $\text{H}_2\text{O}$  in the products of

combustion are reduced. However, these factors reduce the specific heat of the working fluid and increase the mean adiabatic exponent of expansion, thereby increasing the thermal efficiency.

Misfires occurred relatively frequently when operating at  $\phi = 0.8$  and 1200 rpm. In cases where a stable condition could not be attained, the tests were abandoned. Some tests with light to moderate knock were used to evaluate the sensitivity of the knock detector constructed for the present work, a description of which can be found in section 3.5.6.

The knock detected during the tests ranged from heavy, when all cycles presented the phenomenon to light, when less than ten cycles out of 100 sampled, knocked. Unfortunately the equipment does not allow a classification of knock in terms of its intensity. A close examination of the traces of pressure on the oscilloscope revealed that some knock, of very low intensity and inaudible, was occurring in the combustion process, even when operating with ethanol.

One possible cause of the relatively high incidence of knock during the tests was associated with the relatively high inlet mixture temperature, which was adjusted by means of an electric heater in the Alcock flow meter. The temperature selected for all test conditions was  $40 \pm 2$  C, measured in the vicinity of the inlet port. That value of the temperature is representative of the mixture temperature encountered in engines fueled with hydrated ethanol in current production and it also gives a high probability that all the mixture entering the combustion chamber is completely vapourized.

Another good reason to heat the fresh mixture is that according to the findings of Hilden and Parks (1976), improving the vaporization in the intake system of a

single cylinder engine can reduce the amount of unburned fuel and aldehyde emissions.

A typical curve of pressure versus time produced by the knock detector device can be seen in Fig 4.8. The curve was taken for a bowl-in-piston combustion chamber, at compression ratio equal to 10 : 1 and operating with iso-octane. The leading edge of the square wave indicates the instant when knock starts, as visualised in the pressure-time curve. The origin of the time scale was the spark event.

#### 4.4 Operational details of the laser system

In the present work, the time dependent parameter of most interest was the flame speed as measured by the time elapsed between the deflections of the two laser beams, separated by a distance of 3 mm in the case of the bowl-in-piston chamber and 4 mm in the case of disc chambers, caused by the moving flame front. This flame travel time was the result of the difference between the times of two distinct events.

The first event was the deflection of the first laser beam positioned close to the spark plug. This deflection identified the time for the flame to travel from the spark plug electrodes to the beam. The origin of this time was the instant when the spark was triggered. Obviously, this time includes both the time for the flame to travel a specified distance and also any ignition delay. The second time distinguishes the total time elapsed before the flame front reaches the second laser beam. The time of flight of the flame front, used to compute the flame propagation velocities was represented by the difference between those two times. The equipment used to obtain the flame travel times was described in chapter 3. The position of the laser beams inside the combustion chamber can be



visualised from Fig 4.9 and Fig 4.10.

Fig 4.11 represent typical signals from the photodetectors, caused by the passage of the flame front, while Fig 4.12 represents the same signals after being processed by the interface.

Prior to the beginning of the experiments, some tests were performed to evaluate the effects of engine vibration on the laser system. A target was positioned at 10 m from the exit of the beam splitter in order to amplify and hence to evaluate the deflections of the laser spot caused by engine vibration.

Starting from the upper end of the rotation speed range ie, 3000 rpm, it was observed that the vibrations produced by the engine caused the appearance of a figure of ellipsoidal shape, very flattened in the sense of the transverse axis, with a longitudinal axis measuring roughly 150 mm. Reducing the speed of the engine, the shape and size of the figure also reduced, and finally at a speed of 1200 rpm, a small trace measuring approximately 8 mm long was produced. This value was accepted as a compromise given that the initial objective was to run the tests at the higher speed possible.

The deflections at a point distant 800 mm from the origin of the laser light was then estimated to be approximately 0.4 - 0.5 mm in the region where the photodetectors were placed. This was confirmed to be totally acceptable, compared with the deflections caused by the passage of the flame front, of the order of 1.0 - 1.5 mm.

The distance between the first laser beam and the spark plug electrodes was approximately 15 mm when measuring the flame travel times for both types of combustion chamber. However, the distance between laser beams had to be varied in order to accommodate the passage of the pair of beams in

the Heron combustion chamber. In the case of both combustion chambers the maximum spacing allowed by the windows was 4.5 mm but for the disc chamber, 4 mm was selected whereas for the bowl-in-piston, 3 mm was used, the choice being dictated by the channel width.

The plane defined by the two laser beams was inclined in relation to the horizontal plane (floor) at approximately 40 degrees in an attempt to make the plane of the beams pass as close as possible to the center of an imaginary spherical flame originating at the spark plug.

The system was initially conceived as operating with photodiodes having their active surfaces placed in the dark. They were illuminated when the beam reached their active surfaces after being deflected by the propagating flame front. This configuration worked fairly well in preliminary tests outside the engine environment, using the flame of a bunsen burner to deflect the beams.

The beams were directed onto a small black spot target placed over the active face of the photodetector, leaving part of this surface in the shadow. When the beams were deflected by the flame they struck the active surface, producing an output voltage. Between the exit of the beam splitter and the photodetectors, two 45 degrees prisms deviated the beams 90 degrees and directed them through 2 convergent lenses. The system worked fairly well but was difficult to set up. Fig 4.13 illustrates the configuration described.

This configuration was subsequently abandoned and the direct projection of the beams over the photodetectors was adopted. This system was much easier to adjust but still presented a significant drawback. Spreading of the light of the laser over the targets, produced some marginal illumination of the active surfaces, causing noise in the detection system. Tests with the system in the engine

environment proved to be worse than in the laboratory with the bunsen flame.

The final solution adopted a reversed illumination principle. In this configuration, the photodetectors were normally illuminated and produced a signal, indicating the passage of the flame front, when the spot deviated from the photodetectors. Different photodetectors, as described in section 3.5.3 were selected which permitted the system to operate in that manner.

Given the small deflection of the beams (1 - 1.5 mm), a very small active area was required for the photodetectors. The model selected had a diameter of approximately 2 mm and therefore the laser beams could easily be made to fall outside the active surface when deflected by the flame. This system when assembled in the engine environment, proved to be easy to operate and reliable. As the photodetectors were now extremely small, much of the beams handling optics could be eliminated.

However the photodetectors had to be mounted in different planes, a step of 20 mm between them proving to be perfectly adequate. This solution had to be adopted in order to avoid the mutual interference which originates from the scattered laser light from the glass protection of the photodetectors. As a further improvement, all the surfaces were covered with black anti-reflective paint. Fig 4.14 represents the solution adopted.

During operation of the system, one interesting phenomenon was observed when changing the spark timing. As the spark timing was advanced, the beam spot moved from the position occupied previously, requiring that the photodetectors be moved for each condition of spark advance.

It was observed that the beams tended to move further away

from the spark plug, the total deflection being of the order of 1 - 1.5 mm approximately. It is believed that the most plausible cause for this deflection is that an increase in the spark advance, causes a sharp rise in pressure, higher temperature gradients in the combustion chamber and consequently, higher refractive index gradients causing the beams to deviate still further. Once they reached their new position, they continue to present the same magnitude of deflections as previously observed. The sense of this deviation is in accordance with the fact that a light beam subjected to a gradient of refractive index, will deviate towards the higher density gas, in this case towards the unburned gas ahead of the flame front.

The quartz windows also presented problems regarding their operation, namely soot deposits. The tests were initially to be performed using commercial gasoline but this soon proved totally impossible. The windows, even after the engine had attained its nominal operating temperature became totally covered with soot deposits. The laser beams were then totally extinguished and the test had to be aborted after a few minutes in order to clean the windows.

To avoid such inconvenience a pure fuel, iso-octane, was selected as a substitute for the gasoline. The soot deposits were drastically reduced, and after the engine had reached its normal temperature the windows remained quite clean, sometimes for 2 to 3 hours, depending on the extent of time running with higher than 0.8 equivalence ratio. For this reason, all tests started with lean mixtures and finished at stoichiometric equivalence ratio.

One important observation should be made regarding the way the flame travel times were recorded. As the engine was running in the range of stoichiometric to lean mixtures, some misfiring was anticipated. In order to guarantee that all of the cycles sampled by the data acquisition system

were high pressure cycles, the lower pressure threshold was limited by the software to 65 % of the maximum pressure for each condition of operation. In this manner, misfiring cycles were disregarded by the data acquisition and not sampled, leaving only fully developed cycles to be utilised for the calculation of the flame speed. The number of cycles neglected were counted and displayed on the VDU as complementary information.

Normally, with  $\phi = 0.8$  some 5 to 10 cycles were neglected for each sample of 100 good cycles. For a spark advance of 20 DCA the limiting pressure was in the range of 35 - 38 bar at 900 rpm.

It was interesting to note that during the analysis of the flame travel times, it was observed that sporadically a negative flame travel time was recorded, indicating that the flame crossed the two beams in the reverse order. This suggests some significant departures from the sphericity of the flame at that stage of development, pointing towards a substantially greater degree of wrinkling.

#### 4.4.1 Flame travel times.

In this section are presented the results of the measurements of the flame travel times using the differential laser refraction technique. The results represent the average of a sample of one hundred consecutive firing cycles recorded by the data acquisition system for each condition of running of the engine. The manner in which the system senses the misfiring cycles has already been described in the previous section.

The data gathered was then processed statistically in situ, using software developed by the BBC microcomputer, for a first (and rapid) evaluation. They were then stored on floppy disc to receive further and more detailed



statistical analysis later. From such analysis a mean flame travel time  $T$  and a standard deviation  $\sigma$  were obtained such that either  $\sigma$  or  $\sigma/T$  could be used to represent the cyclic variation of the flame travel times.

The distribution of the flame travel times shows some departure from a normal distribution as can be seen from some typical histograms obtained from the processing of the data collected from the tests with the bowl-in-piston combustion chamber running on ethanol as shown in Fig 4.15, Fig 4.16, Fig 4.17 and Fig 4.18. From the examination of such histograms some skewness is evident, which means that the arithmetic mean coincides neither with the median nor with the mode.

Key parameters utilised during the analysis of the flame propagation process are defined as follows

- .mean flame velocity, taken as the ratio of the flame travel distance to the mean flame travel time
- .flame travel angle is defined as the crank angle required by the propagating flame to travel between two adjacent points in the combustion chamber. These points may be the spark plug and one of the laser beams or between each laser beam. Therefore,

$$\Delta\theta = 6 N \Delta t$$

where  $N$  is the engine speed in rpm and  $\Delta t$  is the flame travel time in seconds.

- .flame dispersion is a measure of the cyclic variation of the flame travel times and is represented by the standard deviation of the mean flame travel time .

The conditions for the tests and the parameters varied were presented in section 4.1. The flame travel times were recorded for each combination of parameters except for those tests which presented excessive knock or instability and which therefore had to be abandoned. A representative

sample of typical histograms of flame travel times corresponding to some of the conditions of testing is presented. Fig 4.15 is the histogram corresponding to the flame travel times in a bowl-in-piston chamber operating with ethanol, CR = 12 : 1, SA = 10 DCA, N = 900 rpm and equivalence ratio  $\phi = 1$ . Fig 4.16 and Fig 4.17 corresponds to histograms of flame travel times for the same combustion chamber but, the spark advance was changed to 15 DCA in the first case and 20 DCA in the latter case. Fig 4.18 represents the histogram for the same combustion chamber with the difference that the speed was now 600 rpm.

Table 4.1 shows some of the flame travel times recorded during the tests and in particular those utilised to illustrate this chapter. The engine was operated at wide open throttle (WOT) during all test conditions. In order to evaluate the influence that operating engine parameters had on the time of flight of the flame front, probability density curves versus time in milliseconds and probability density curves versus velocity in m / s were produced by processing data from 100 samples, as can be seen from Fig 4.19 to Fig 4.34. The advantages of using statistical analysis of data averaged over many engine cycles instead of single cycle or combustion bomb measurements is obvious, specially due to the ability of such techniques to identify small variations in data presenting large fluctuating components.

The distribution curves for the flame travel times revealed some asymmetry, with a skewness towards longer times for most of the curves. Nevertheless, the distribution curves for the flame speed presented a slight asymmetry with a skewness towards greater flame speeds ie, smaller times. The apparent discrepancy is caused by the fact that in the first case,  $p(\Delta t)$  is a distribution of  $\Delta t$ , whilst in the second case  $p(v)$  is a distribution of  $1 / \Delta t$ , for the same values of  $\Delta t$ , which causes a skewness on the side of small values of  $\Delta t$ .



|          | MEAN  | STD DEV | STD ERR | CFT VAR | SKEW  |
|----------|-------|---------|---------|---------|-------|
| ID120159 | 0.420 | 0.062   | 0.006   | 14.83   | 0.075 |
| ID120209 | 0.449 | 0.065   | 0.006   | 14.80   | 0.071 |
| ED115109 | 0.556 | 0.110   | 0.011   | 19.80   | 0.069 |
| ED120209 | 0.392 | 0.053   | 0.005   | 12.82   | 0.079 |
| EB110109 | 0.462 | 0.165   | 0.016   | 35.72   | 0.128 |
| EB115109 | 0.397 | 0.138   | 0.014   | 34.61   | 0.159 |
| EB115100 | 0.282 | 0.101   | 0.010   | 35.92   | 0.082 |
| EB120209 | 0.284 | 0.245   | 0.025   | 87.33   | 0.539 |
| EB210109 | 0.414 | 0.119   | 0.012   | 28.80   | 0.090 |
| EB215109 | 0.371 | 0.116   | 0.012   | 31.20   | 0.099 |
| EB215100 | 0.248 | 0.065   | 0.006   | 26.34   | 0.108 |
| EB220109 | 0.266 | 0.138   | 0.014   | 51.80   | 0.152 |
| EB210158 | 0.403 | 0.081   | 0.008   | 20.00   | 0.027 |
| EB210159 | 0.345 | 0.120   | 0.012   | 34.80   | 0.179 |
| EB210150 | 0.278 | 0.084   | 0.008   | 30.00   | 0.121 |
| EB215150 | 0.222 | 0.049   | 0.005   | 22.20   | 0.146 |
| EB215200 | 0.240 | 0.072   | 0.007   | 29.70   | 0.083 |

Obs. The code in the first column is interpreted as below

E B 2 15 20 0

.1st letter : E = ethanol or I = iso-octane

.2nd letter : B = bowl-in-piston or D = disc

.1st number : 1 = CR 10 : 1 or 2 = CR 12 : 1

.2nd number : 10 = 600 rpm, 15 = 900 rpm, 20 = 1200 rpm

.3rd number : spark advance = 10, 15 or 20 DCA

.4th number : 8  $\rightarrow \phi = 0.8$ , 9  $\rightarrow \phi = 0.9$ , 0  $\rightarrow \phi = 1.0$

Table 4.1 Statistical analysis of flame travel times

Fig 4.19 to Fig 4.22 show the influence of the type of fuel on both types of combustion chamber. The difference in mean flame travel time and as a consequence, in flame speed, was greater when operating with ethanol for the disc chamber than for the BIP chamber. One possible explanation is that because the disc chamber presents a much lower turbulence than the bowl-in-piston, flame speed will also be lower and consequently the dependence of flame speed on the laminar properties is greater for the disc chamber.

With both types of combustion chamber, the flame travel times corresponding to the operation with ethanol were smaller than those corresponding to the operation with iso-octane. This is most simply explained by the greater laminar burning velocity of the ethanol compared to that of the iso-octane. This trend was repeated in the cycle simulation program as will be shown in chapter 5.

The cyclic variation of the flame travel times and of the flame speed was generally greater for the bowl-in-piston combustion chamber independent of the fuel employed, as shown in Fig 4.23 to Fig 4.26. It is believed that this behaviour is due to problems caused by the choice of the geometry of these chambers. When the piston reaches TDC, during the combustion phase, the flame has approximately 6 to 10 mm radius and because the bowl was machined axi-symmetrically in the piston to facilitate the modelling process, some of the flame, still in its early stages of development may be quenched in the annular area surrounding the bowl. In this way the cyclic variation could be enhanced with this type of combustion chamber. The problem could, perhaps, be minimised by the utilization of a non-symmetrical bowl as described in section 4.3.

The increase in compression ratio caused a decrease in the

mean flame travel time as depicted in Fig 4.28, where the distribution  $p(\Delta t)$  for the BIP chamber with a compression ratio equal to 10 : 1 present a greater dispersion than for compression ratio 12 : 1. For example, the mean flame travel time and the standard deviation for a BIP chamber operating on ethanol, SA = 10 DCA, N = 900 rpm,  $\phi = 0.9$ , was  $T = 0.40$  ms and  $\mathcal{T} = 0.14$  ms when the compression ratio was 10 : 1 and  $T = 0.37$  ms and  $\mathcal{T} = 0.12$  ms when the compression ratio was 12 : 1. This is in agreement with the fact that increasing compression ratio, increases the temperature of the mixture and also reduces the amount of residuals, thus increasing the laminar burning velocity. One consequence of this is a reduction in the cyclic dispersion. This may reflect the early stages of flame development as suggested by Kalghatgi (1985).

The effect of engine operating parameters such as equivalence ratio, spark timing and engine speed on the mean flame travel time and flame speed are depicted in Fig 4.29 to Fig 4.34. The mean flame travel times decrease as the equivalence ratio increase, according to Fig 4.29. The cyclic dispersion also tends to increase as the equivalence ratio is decreased.

The variation of the mean flame travel time with the spark timing is depicted in Fig 4.31. The minimum flame travel time occurred for a spark timing of 35 DCA BTDC. Above and below that value, the tendency has been towards an increase in flame travel time for all the conditions of testing. The cyclic dispersion is small for the three cases, with SA = 15 DCA presenting the least amount of cyclic variation, as indicated in Table 4.1.

Finally, Fig 4.33 illustrates the variation of the mean flame travel time with engine speed. The tendency is towards an increase in flame travel time with a reduction in engine speed as is well established. The reason is that reducing engine speed causes turbulence intensity in the

chamber to be reduced with a consequent reduction in the turbulent burning velocity. Therefore, the flame propagates at a slower rate, increasing the time to cover the distance between the laser beams. With respect to cyclic variation, the tendency is towards a reduction in cyclic dispersion with reducing engine speed

Fig 4.35 to Fig 4.44 show plots of mean flame speed versus spark timing for different equivalence ratios, engine speed, fuel type and compression ratios. As a general rule the peak flame speed corresponded to the spark timing of 15 DCA BTDC.

The variation in flame speed as a function of equivalence ratio is indicated in Fig 4.43 for two compression ratios. The fuel was ethanol and the combustion chamber that of the bowl-in-piston. As is clearly seen, the tendency is towards an increase in the flame speed towards stoichiometrical mixtures as would be expected.

The dependence of flame speed on the engine speed can be seen on Fig 4.44 for the bowl-in-piston and disc combustion chambers, both running on ethanol, the difference being slightly greater for the bowl-in-piston.

The comparison between the experimental findings described in the present chapter, and the predictions from the computer simulation program presented and discussed in chapter 5, will be presented in chapter 6.

### Summary and conclusions

A single-cylinder engine was operated on hydrated ethanol at compression ratios  $CR = 10 : 1$  and  $CR = 12 : 1$ , with two types of combustion chambers, namely disc and bowl-in-piston, at various conditions of engine parameters



such as spark timing, equivalence ratio, and engine speed. Iso-octane was also used for both combustion chambers at  $CR = 10 : 1$ . The aim of the experiments was to measure the flame speed utilising an equipment designed and constructed for that purpose.

The laser-based system designed and built to measure the flame travel times in the combustion chamber of the Ricardo E6 research engine proved to be a reliable but not easy to operate system. However, the initial intention to design and build an equipment for engine combustion diagnosis that was simple, easy to operate and principally, cost effective, was nearly totally fulfilled, the only drawback being the fact that operating the equipment, presently, is a highly time-consuming operation.

The bowl-in-piston combustion chamber, despite the impaired performance caused by simplifications adopted during design and manufacturing, as explained during the discussions, produced a superior overall performance when compared to the disc chamber for the same operational conditions. It is believed that the higher resistance to knock presented by both the bowl-in-piston chamber and ethanol fuel, recommends both as a good option for spark ignition engine application.

Operation with hydrated ethanol presented an increase in engine consumption and a decrease in power output for both types of combustion chamber when compared with iso-octane for the same engine parameters. However, increasing the compression ratio brought an increase in power output and torque, an increase in thermal efficiency and a reduction in fuel consumption. However, the fuel consumption with ethanol was still greater than the fuel consumption obtained when operating the engine with iso-octane. Ethanol also presented a much higher resistance to knock

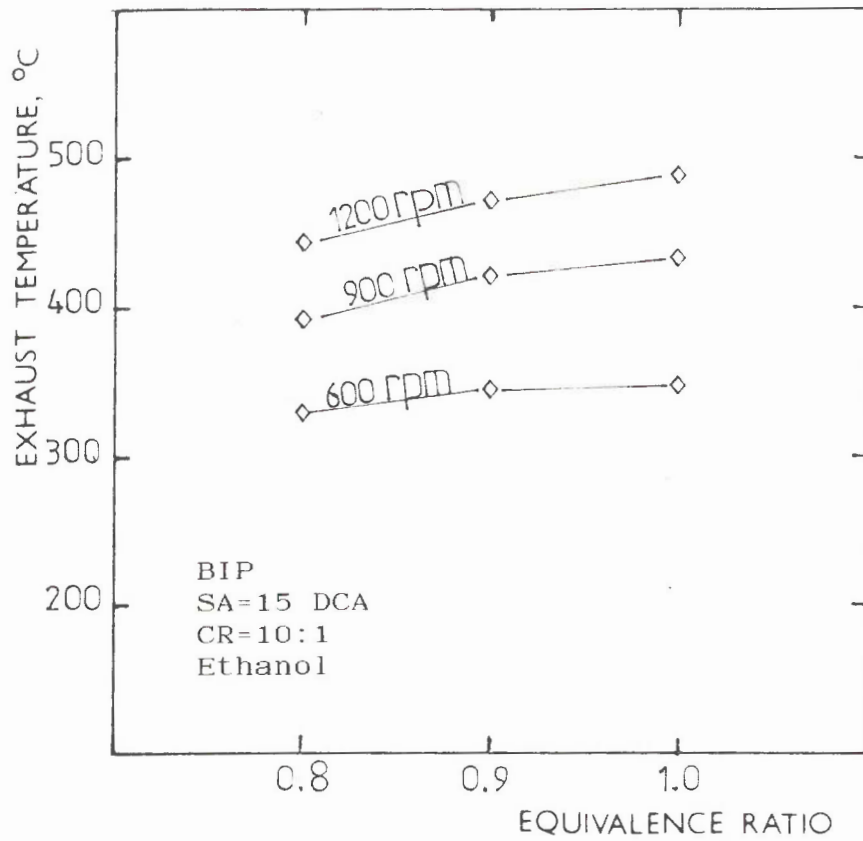
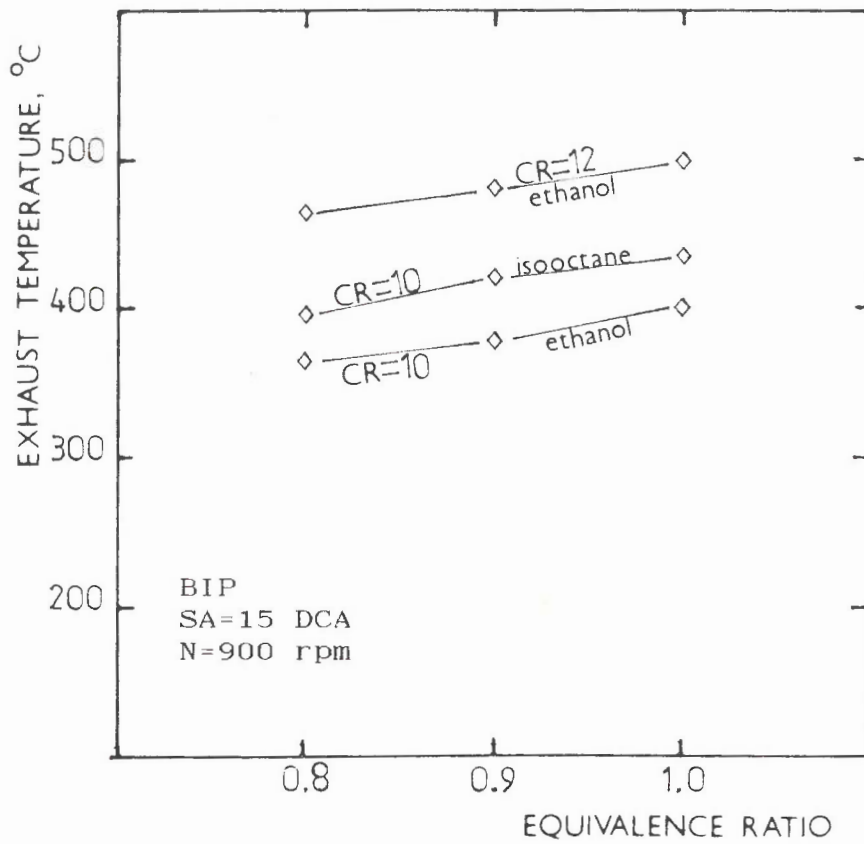


Fig. 4.1 Exhaust temperature versus equivalence ratio for BIP chamber at different speeds.

Fig. 4.2 Exhaust temperature versus equivalence ratio for BIP chamber for different fuels.

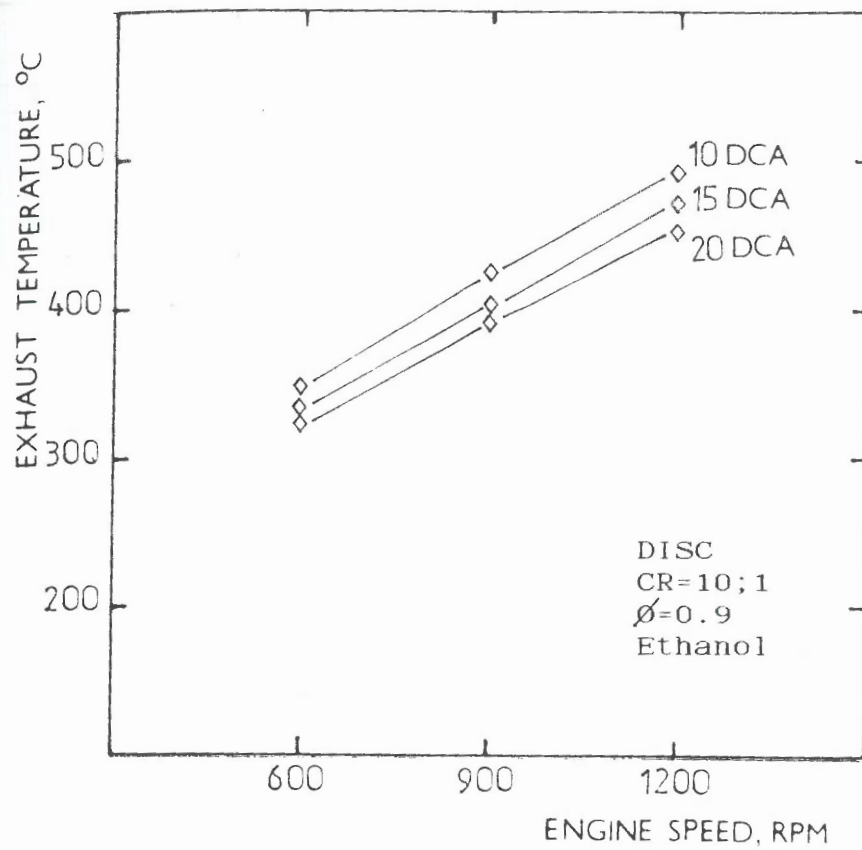
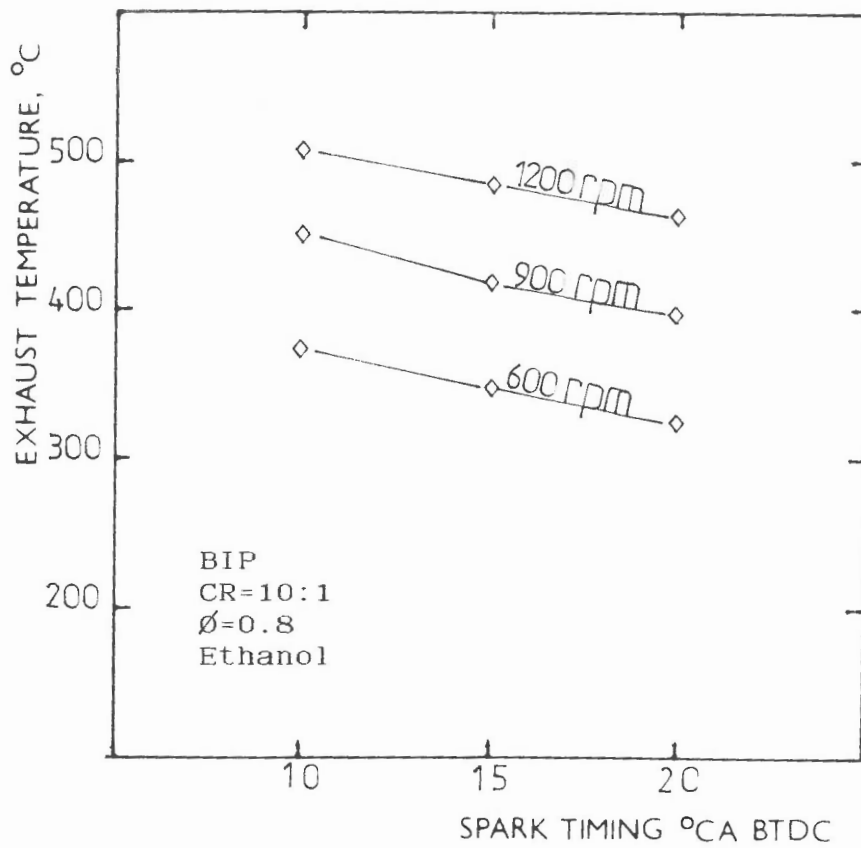


Fig. 4.3 Exhaust temperature versus engine speed for DISC chamber at various spark timing.

Fig. 4.4 Exhaust temperature versus spark timing at different engine speeds, for BIP chamber.



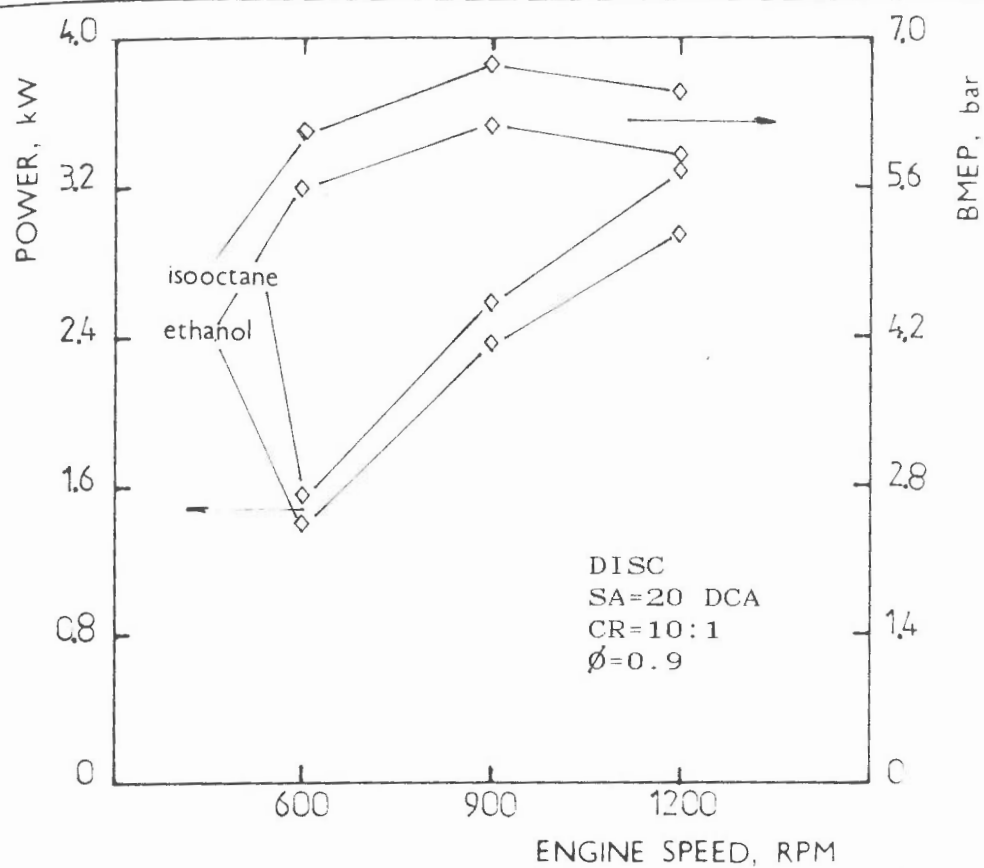
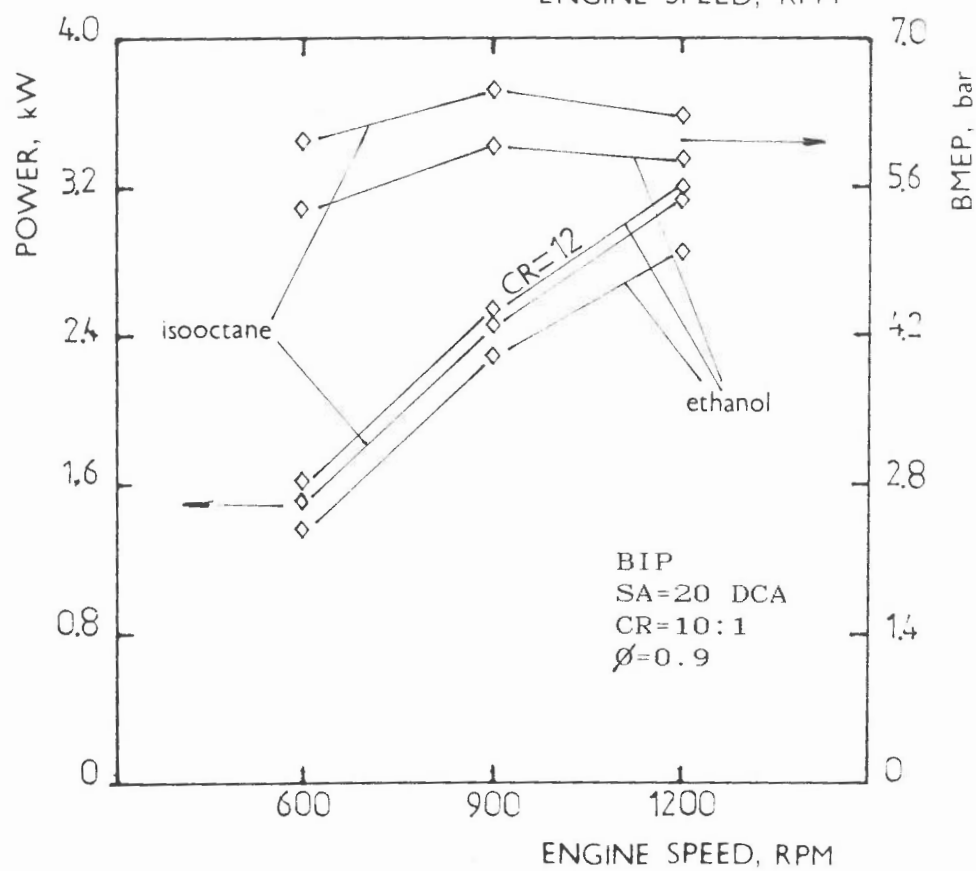


Fig. 4.5 Power and BMEP versus engine speed for DISC chamber as a function of fuel type.

Fig. 4.6 Power and BMEP versus engine speed for BIP chamber as a function of fuel type.

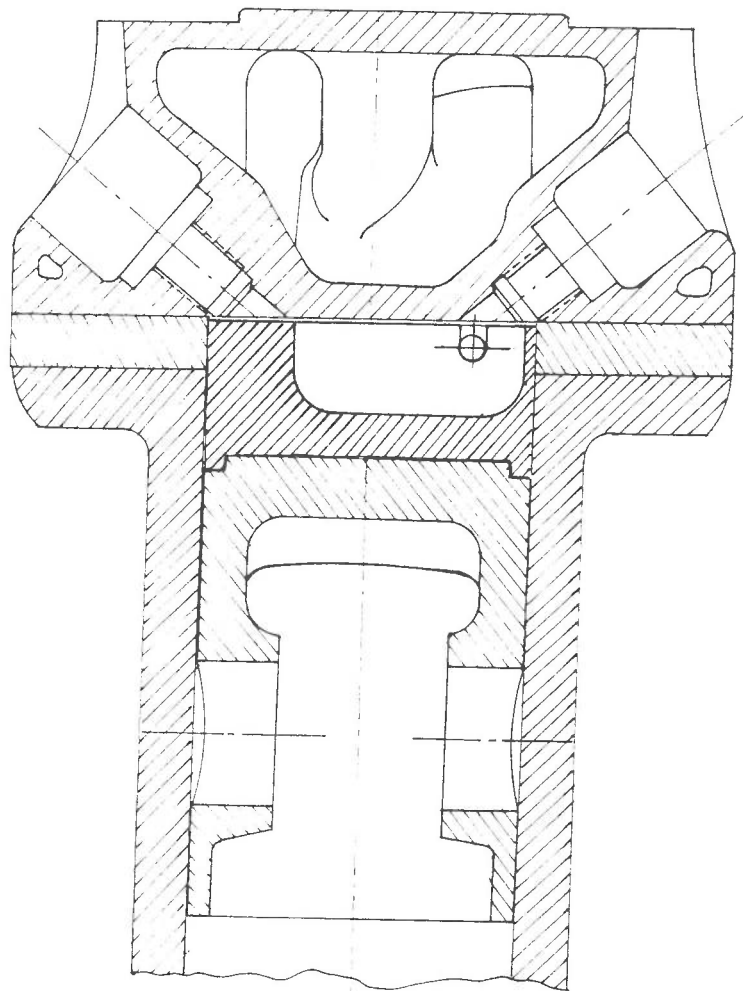


Fig. 4.7 Detail of the excentric bowl-in-piston combustion chamber.

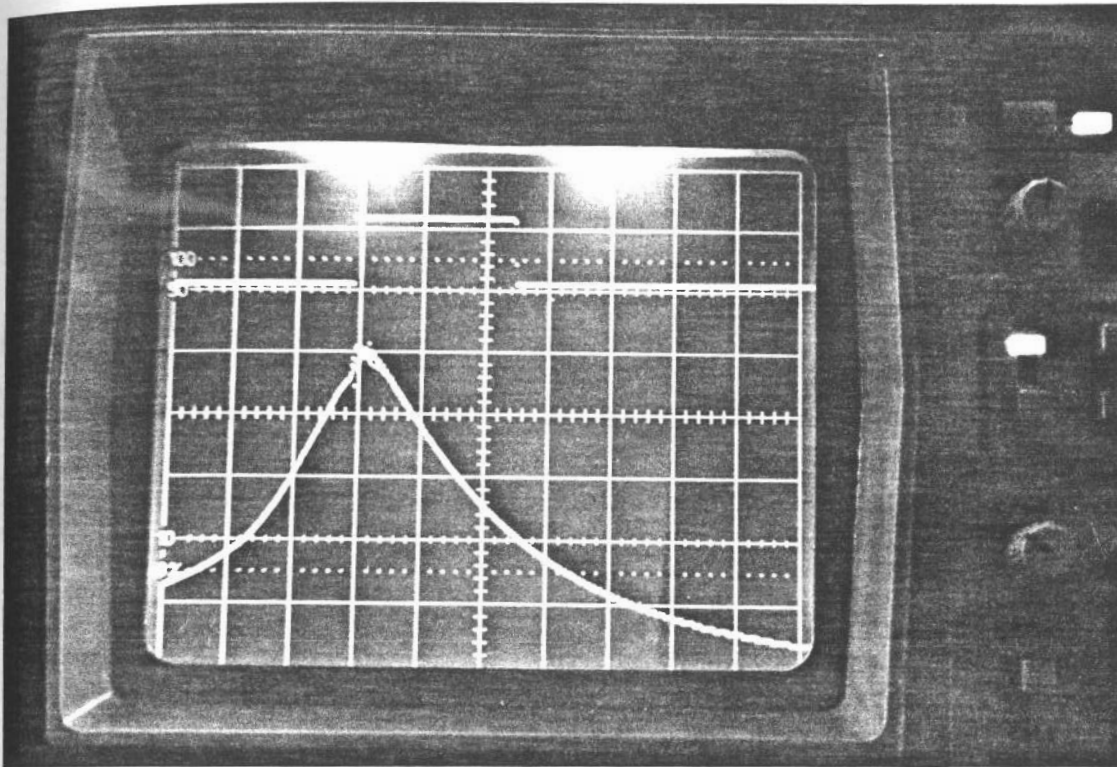
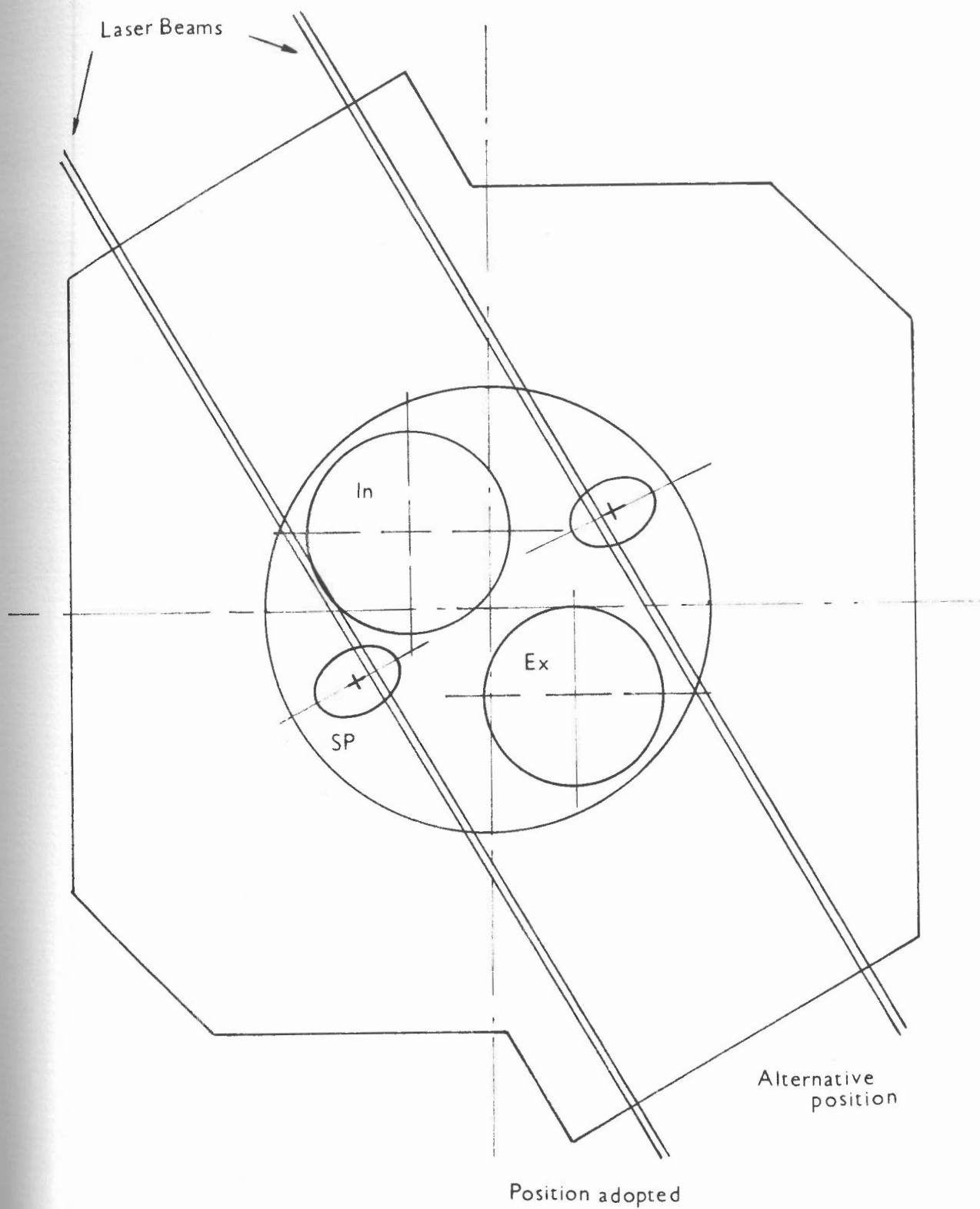


Fig. 4.8 Cylinder pressure versus crank angle of a knocking cycle, showing the trace of the knock sensing device. Origin of time is the spark event. Engine speed : 1200 rpm, time scale : 2 ms/div.

Fig. 4.9 Relative positions between laser beams and spark plug.



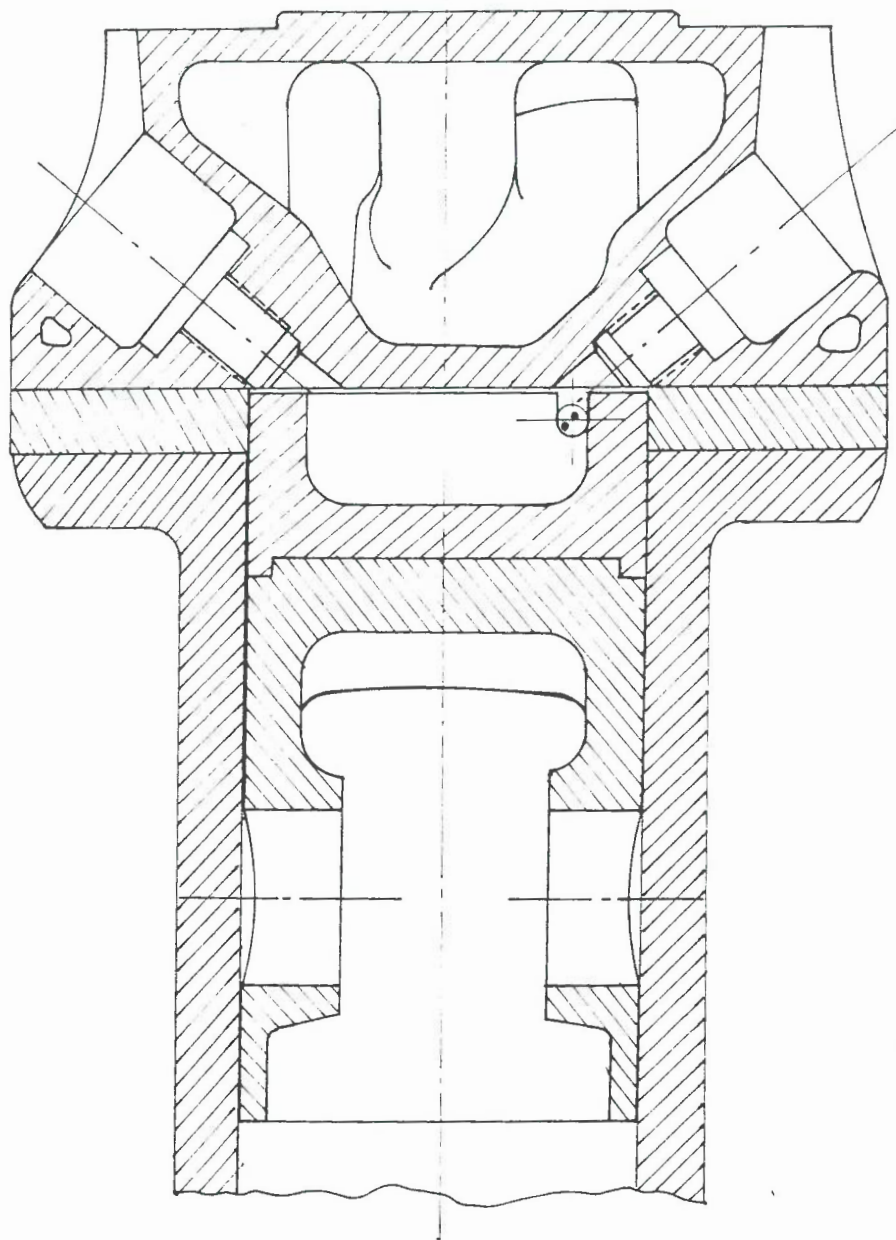


Fig. 4.10 Position of the laser beams inside the window, relative to the spark plug.



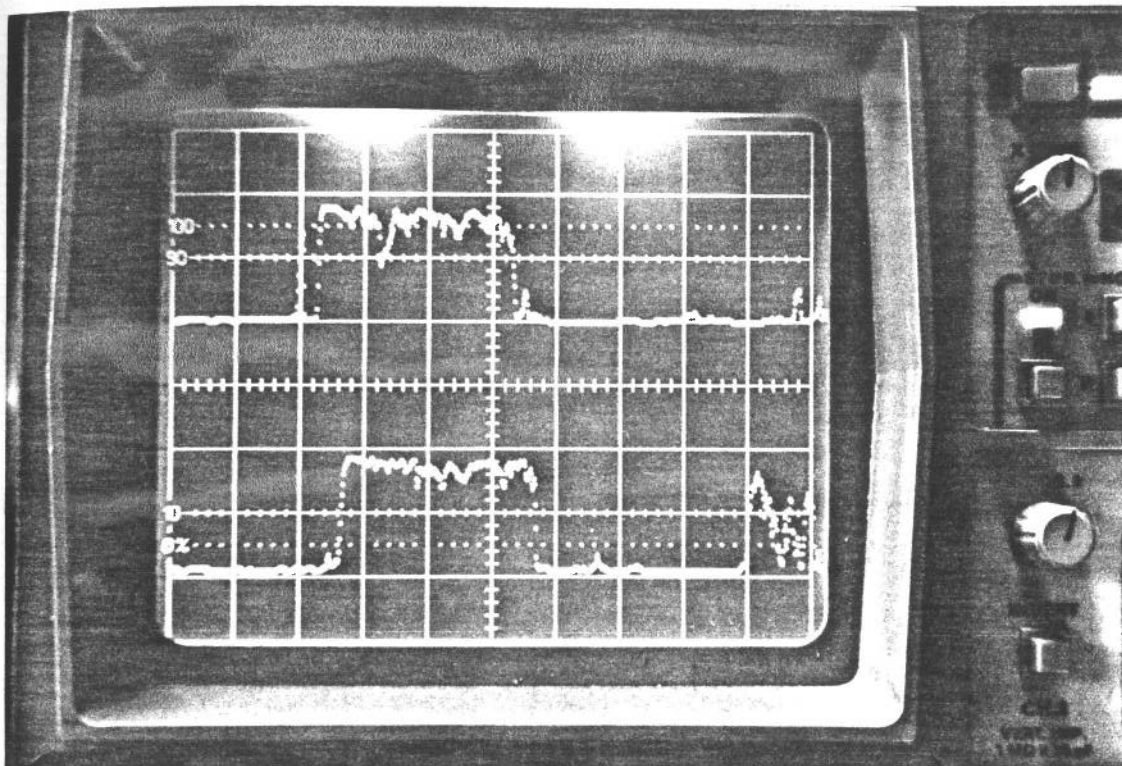
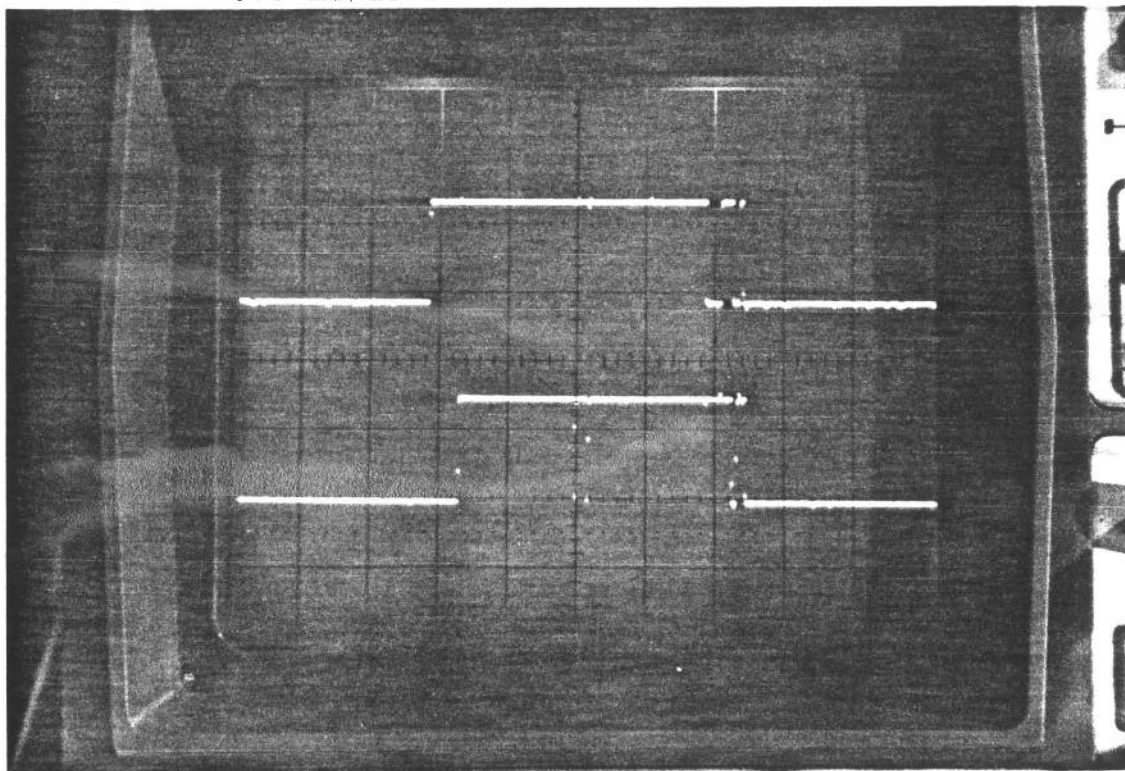
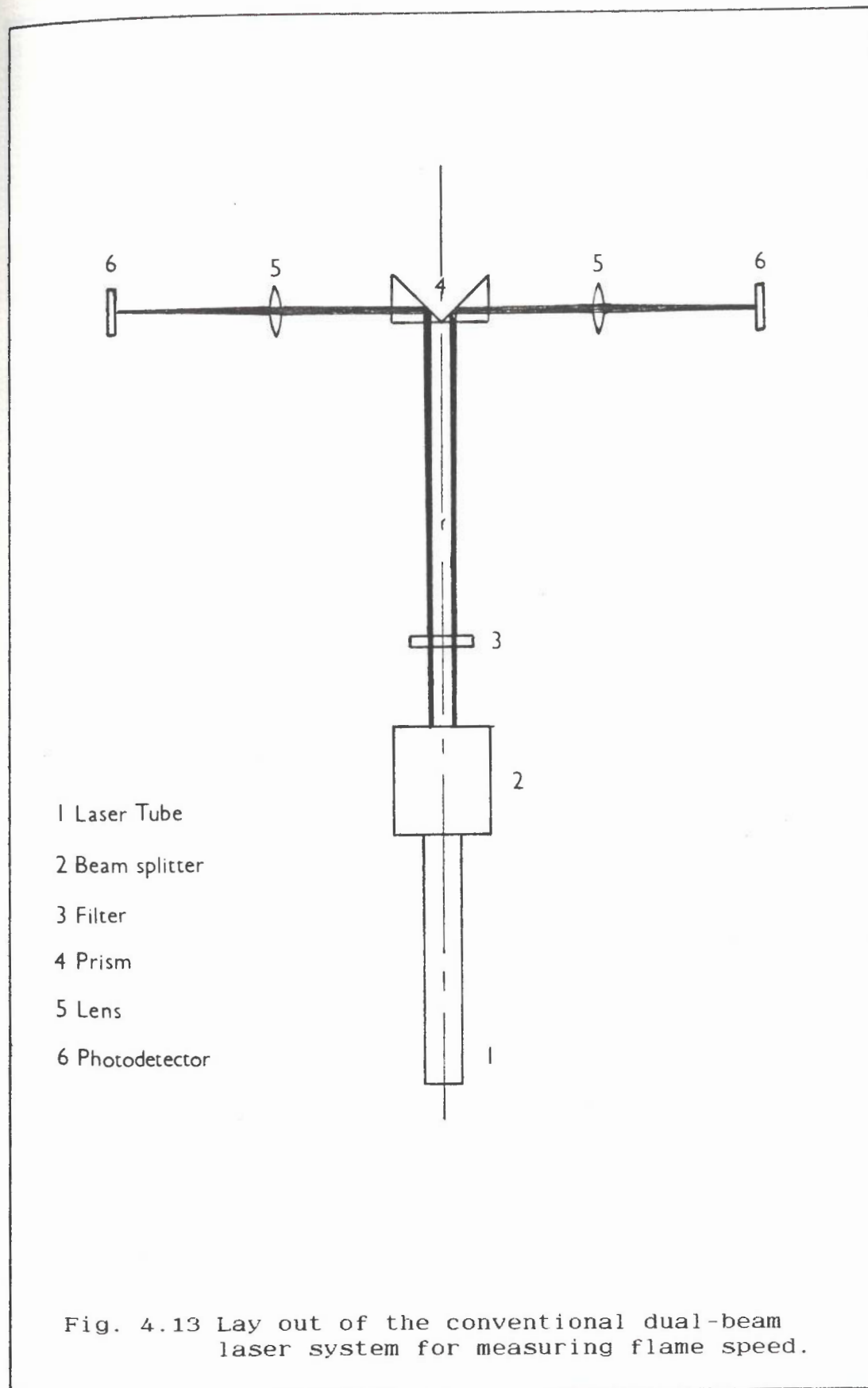


Fig. 4.11 Photodetector signals before processing.  
1 V/div versus 1 ms/div.

Fig. 4.12 Photodetector signals after processing by  
the time event interface. 0.2 V/div versus  
0.5 ms/div.







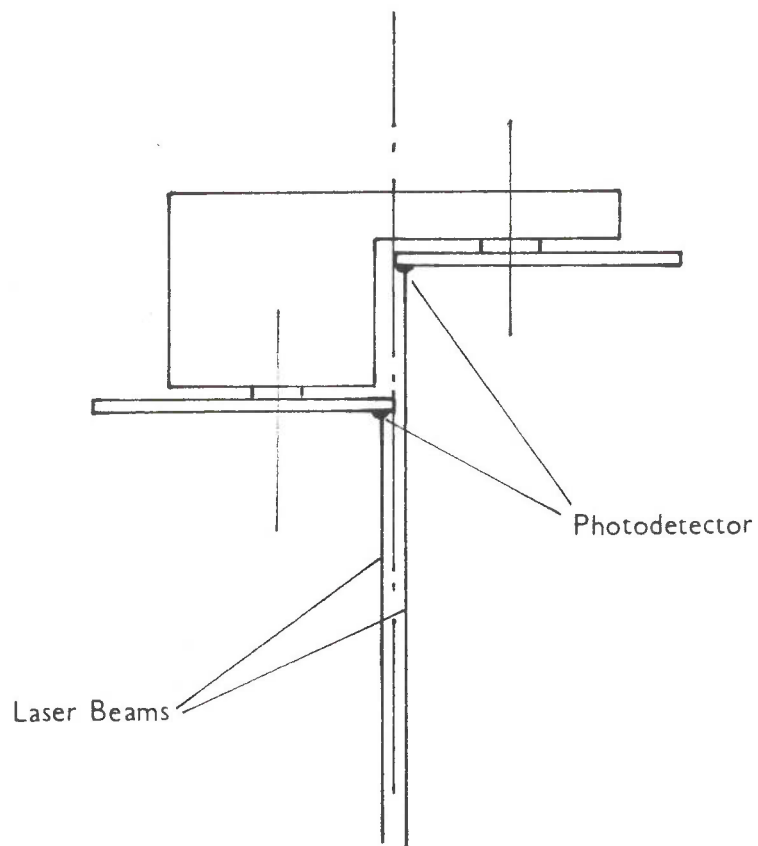


Fig. 4.14 Detail of the photodetectors support.

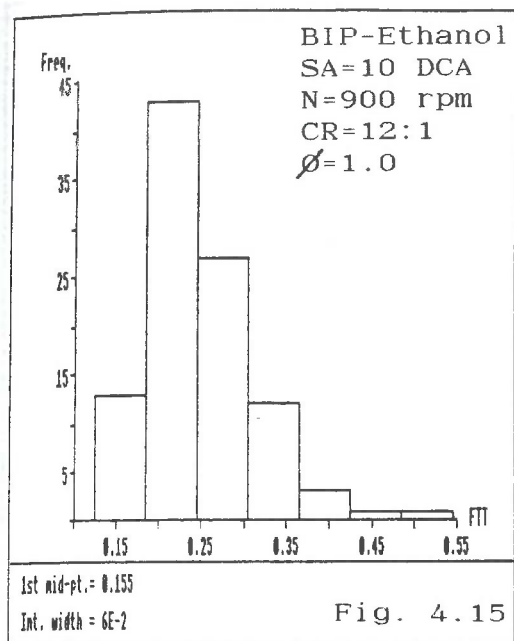
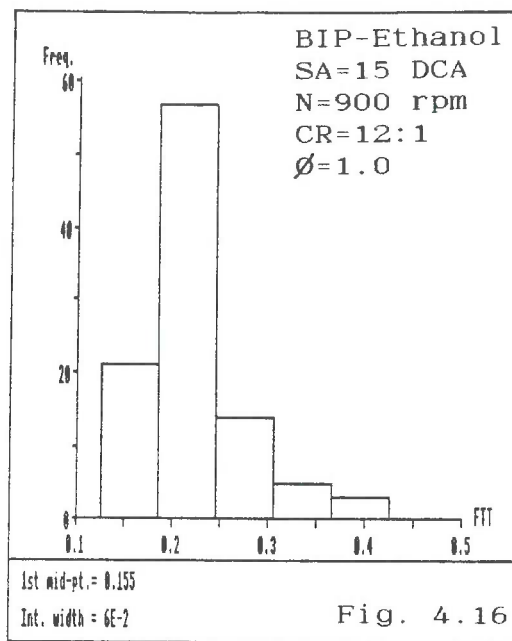
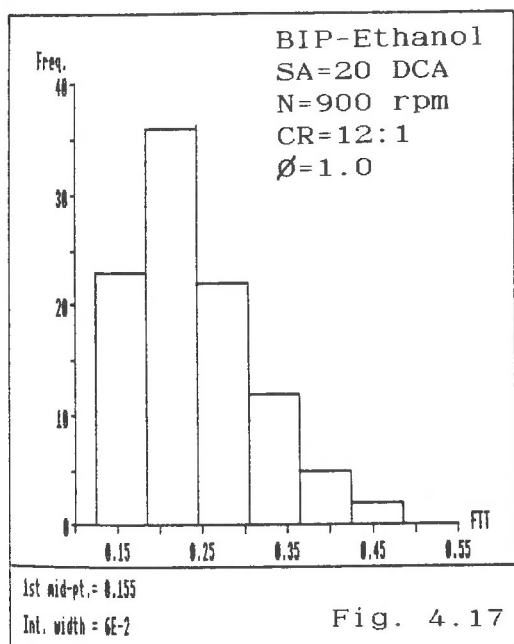
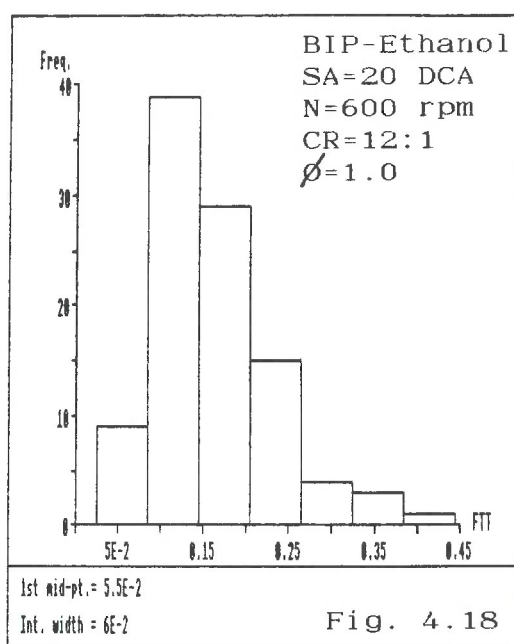


Fig. 4.15 to 4.18 Histograms of frequency versus flame travel times for BIP chambers.

Fig. 4.19 - 4.20 Influence of the type of fuel on the FTT and FPV for the DISC chamber.

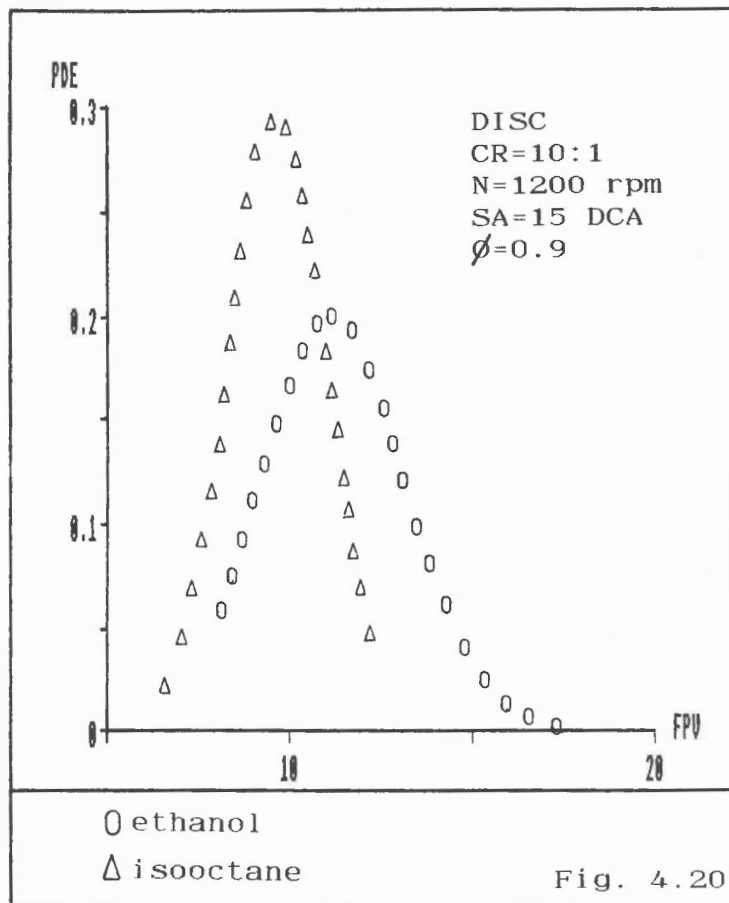
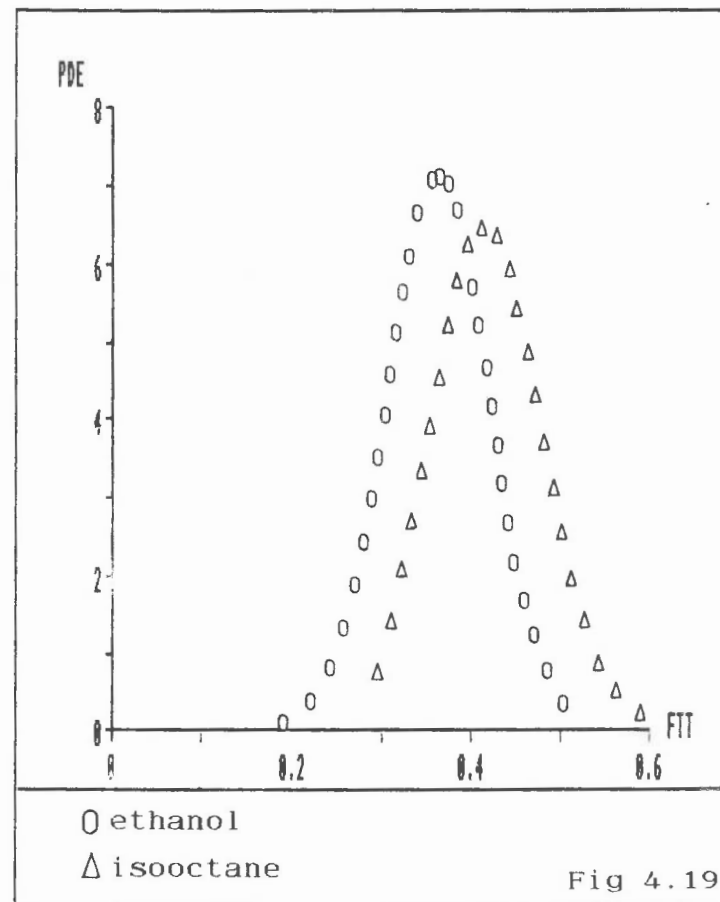


Fig. 4.21 - 4.22 Influence of the type of fuel on the FTT and FPV for the BIP chamber.

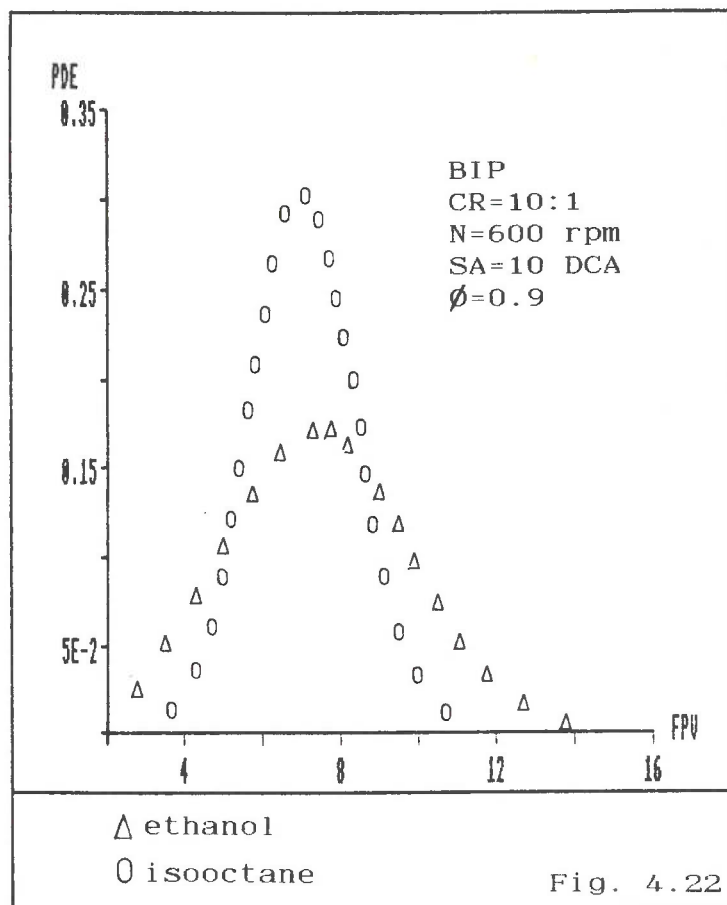
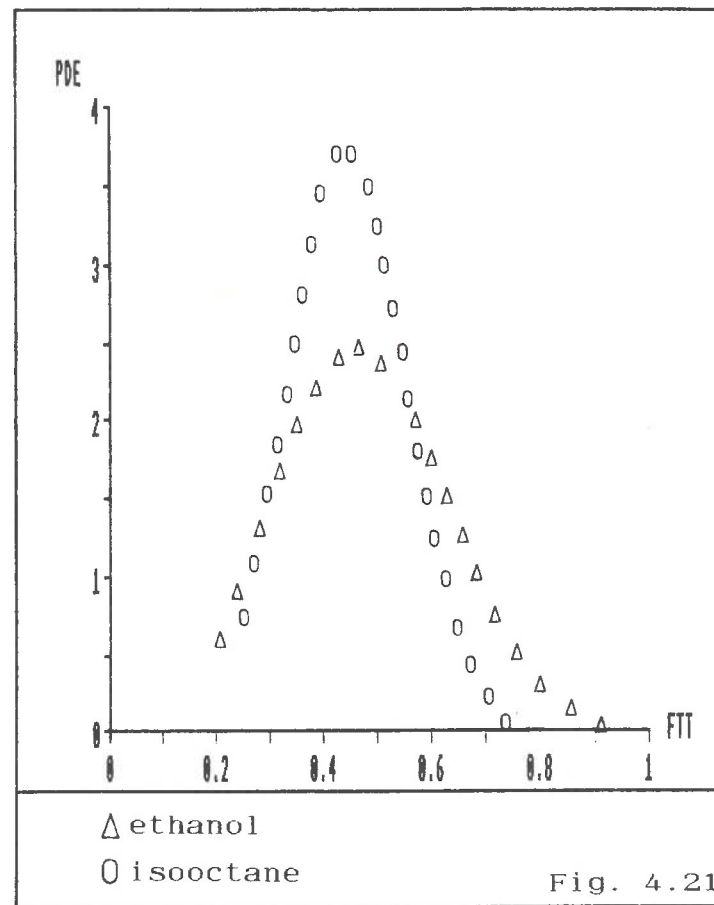


Fig. 4.23 - 4.24 Influence of combustion chamber geometry on the FTT and FPV.

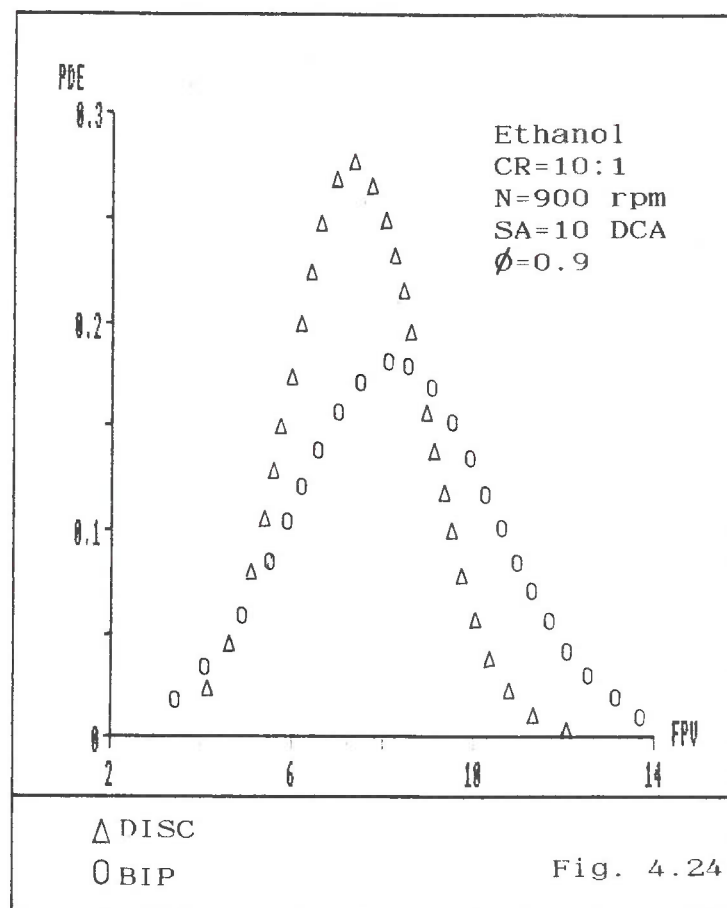
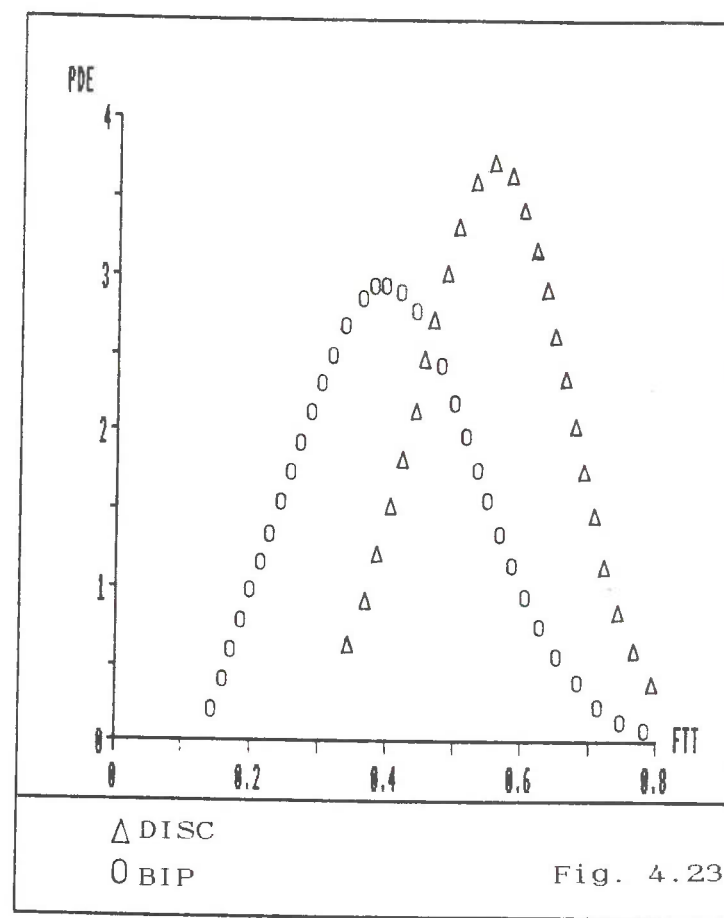




Fig. 4.25 - 4.26 Influence of combustion chamber geometry on the FTT and FPV.

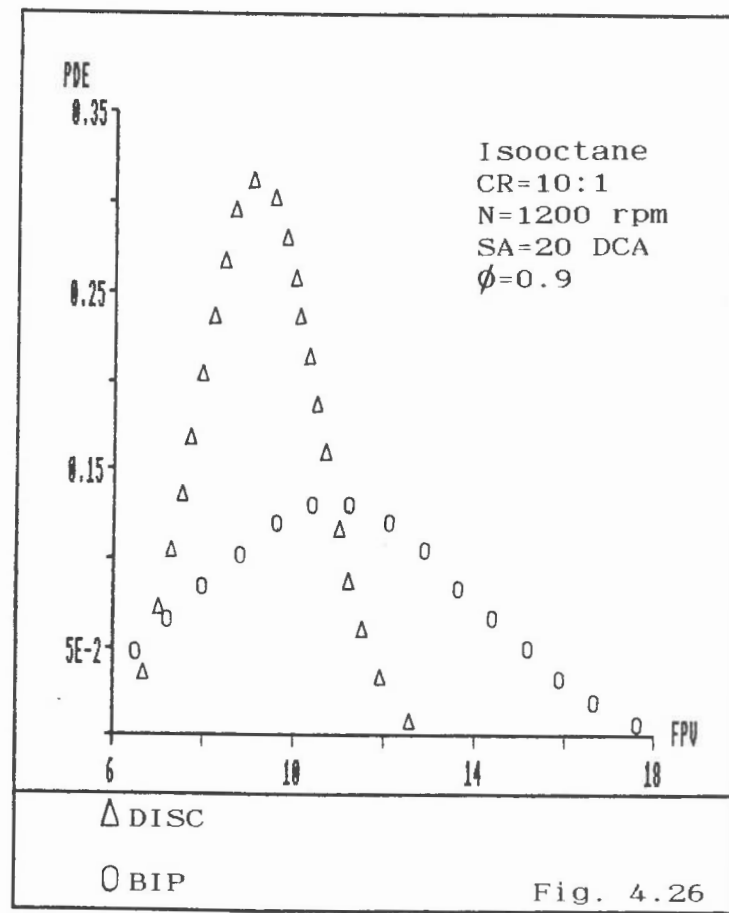
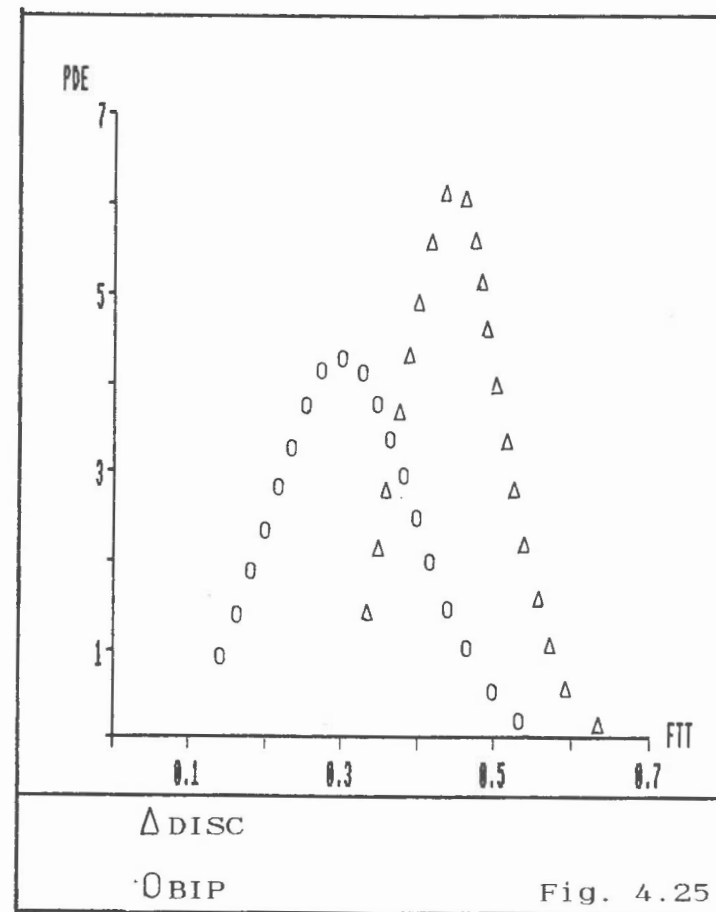


Fig. 4.27 - 4.28 Influence of the compression ratio on the FTT and FPV for a BIP chamber.

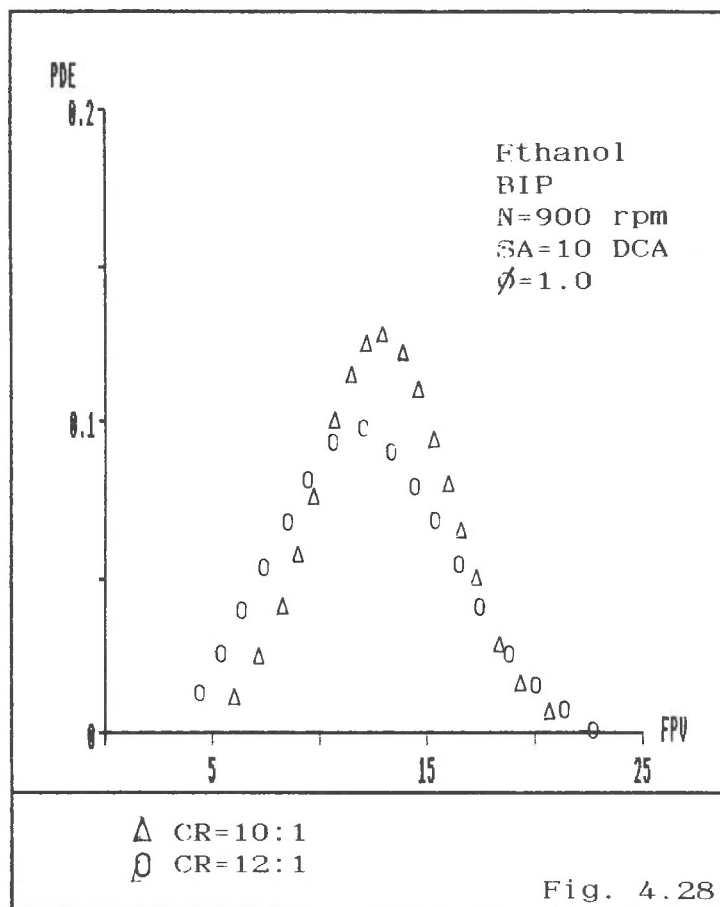
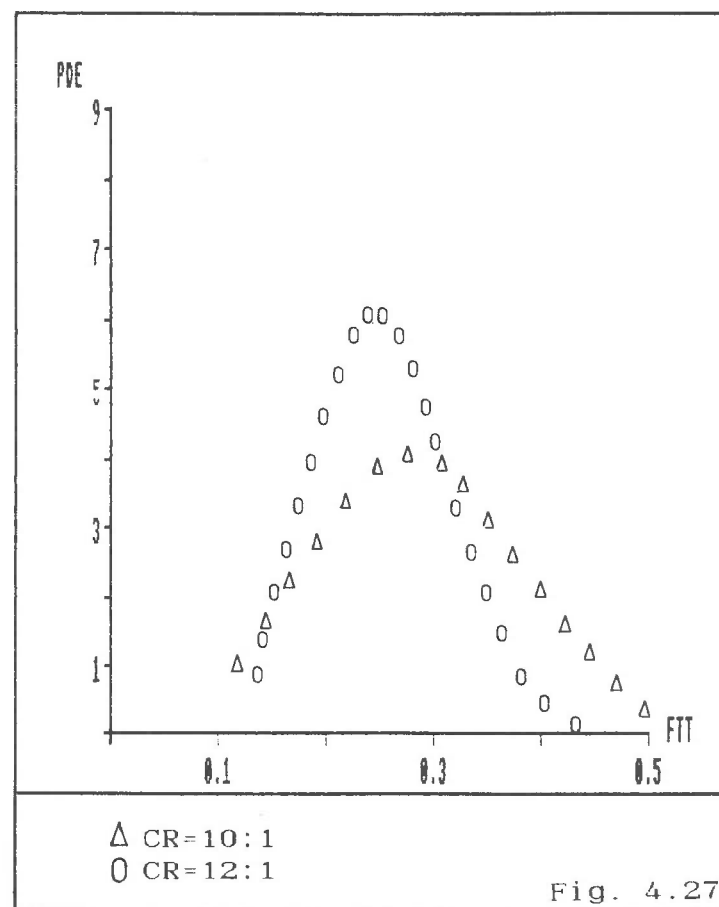


Fig. 4.29 - 4.30 Influence of equivalence ratio on the FTT and FPV for a BIP chamber.

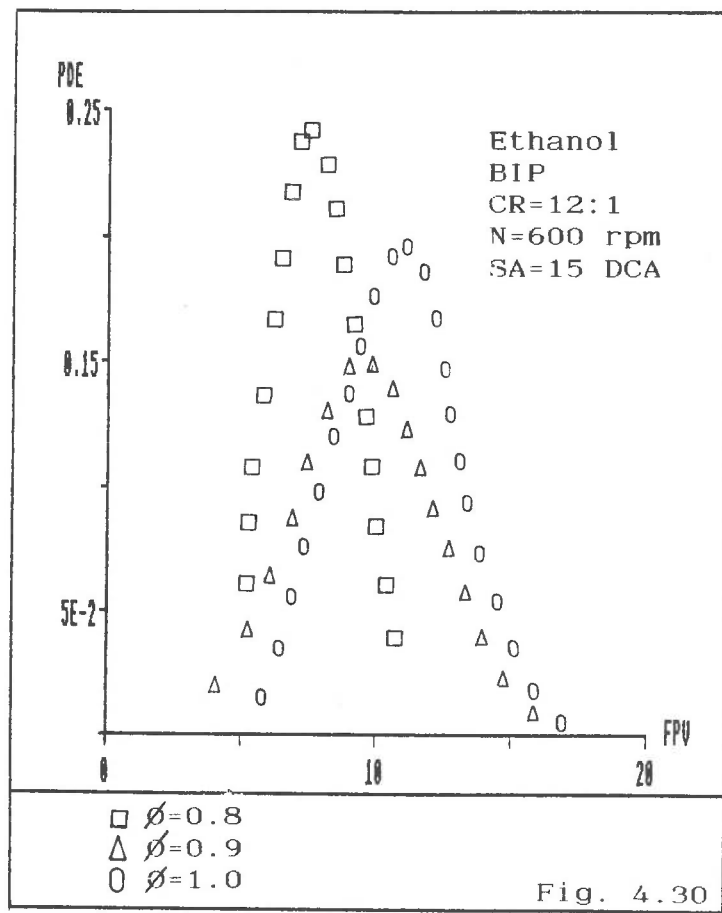
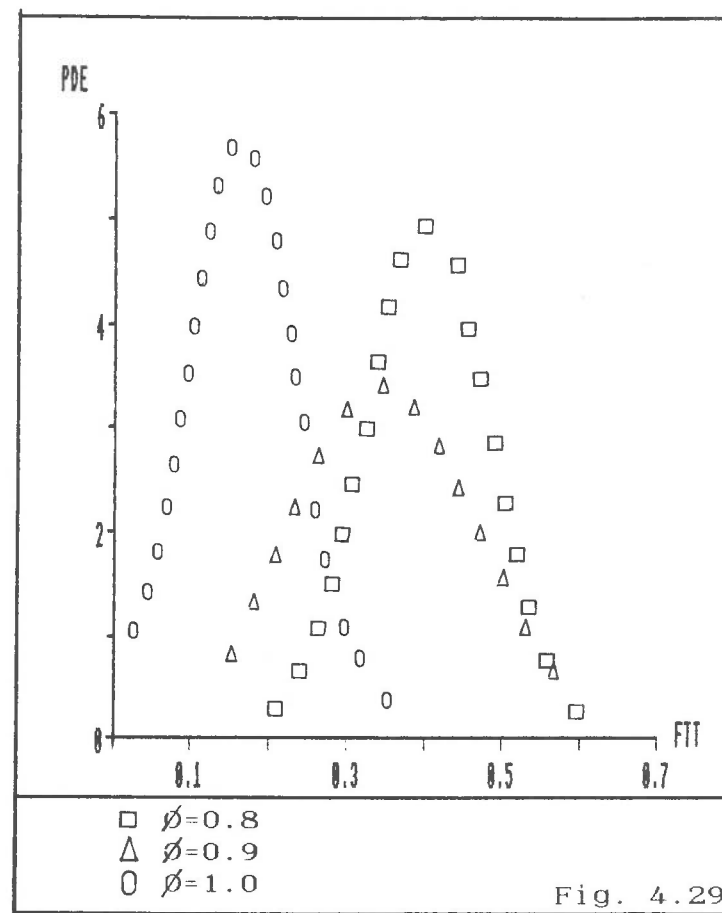


Fig. 4.31 - 4.32 Influence of the spark timing on the FTT and FPV for BIP chamber.

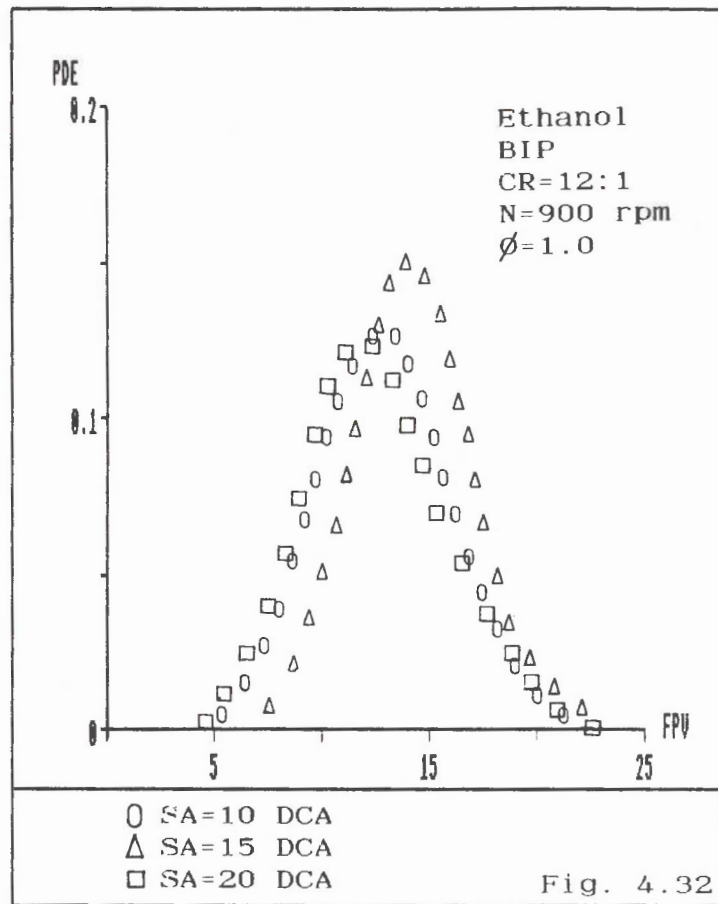
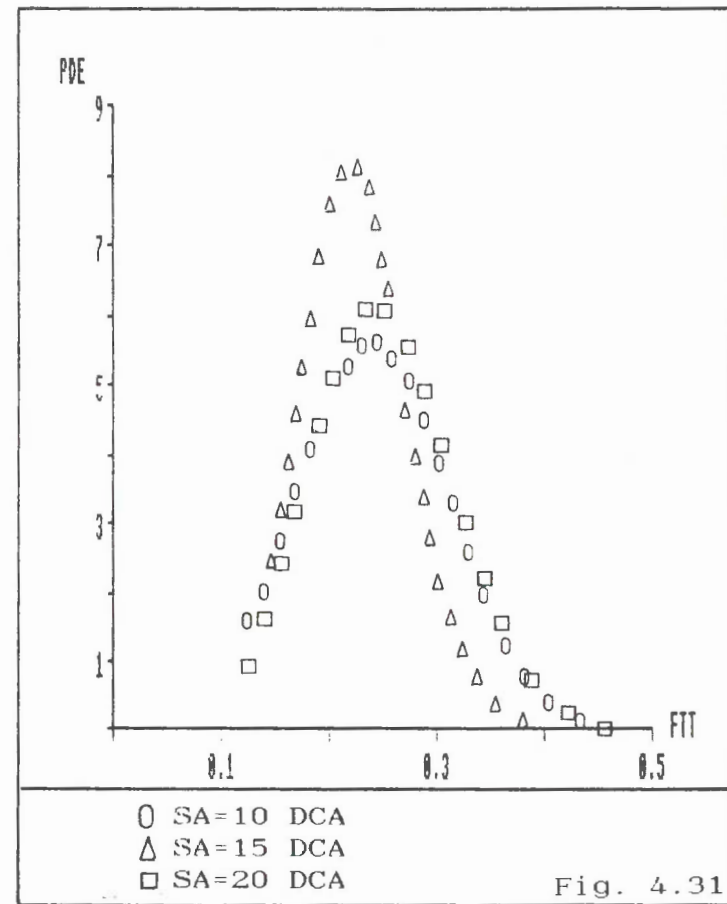
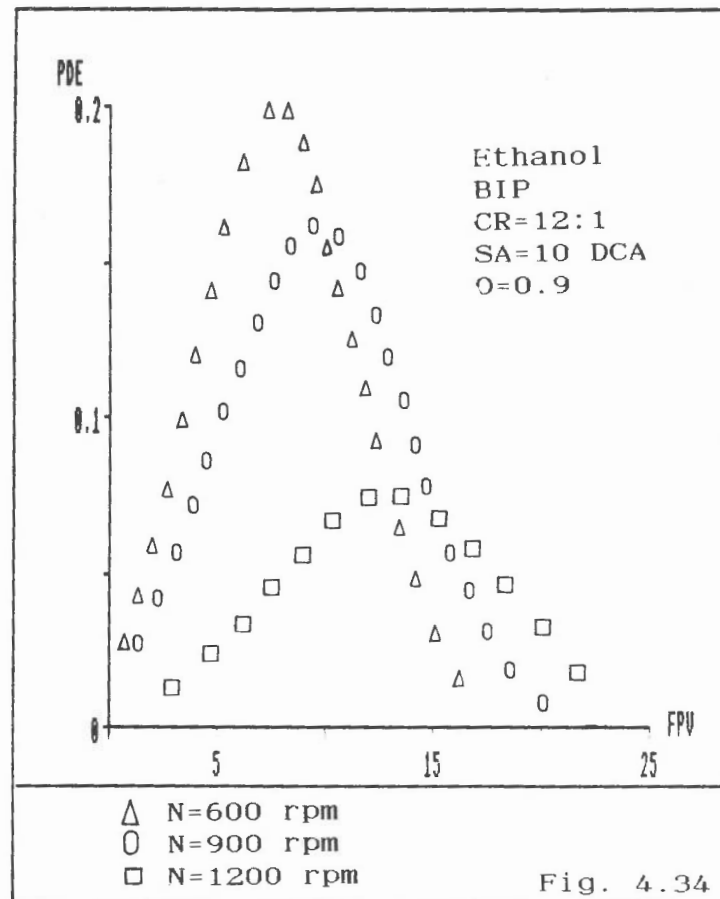
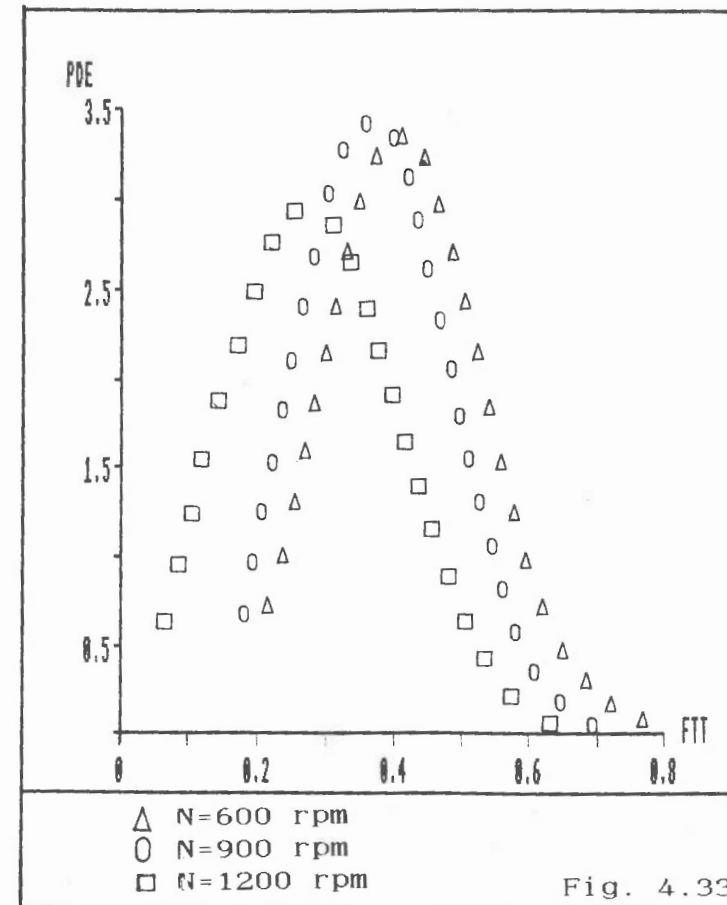


Fig. 4.33 - 4.34 Influence of the engine speed on the FTT and FPV for a BIP chamber.



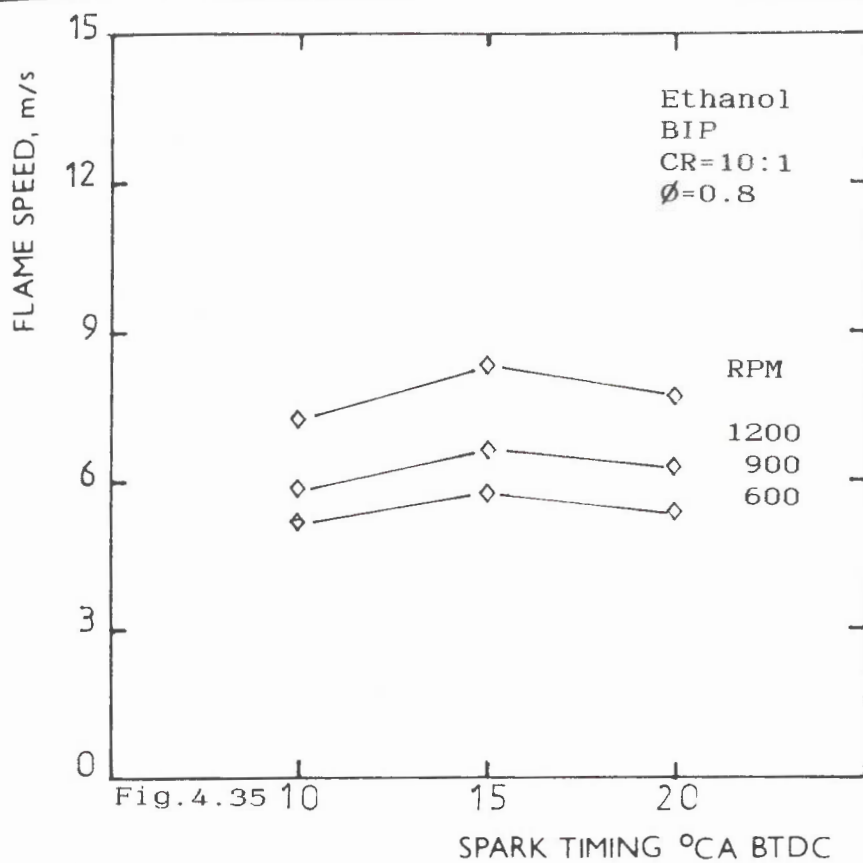
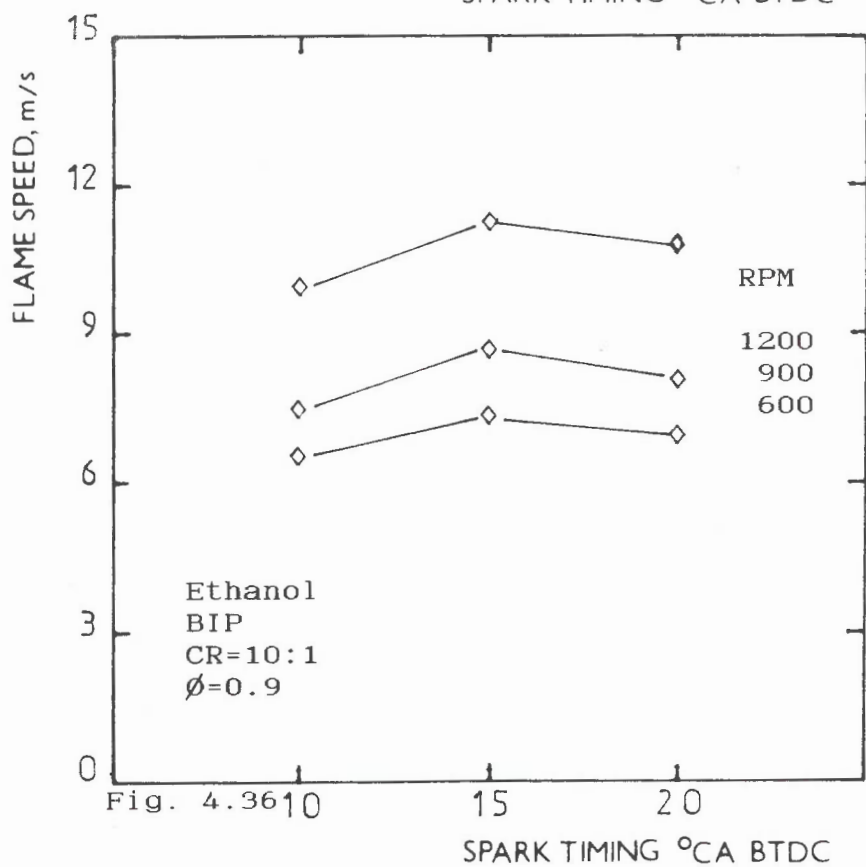
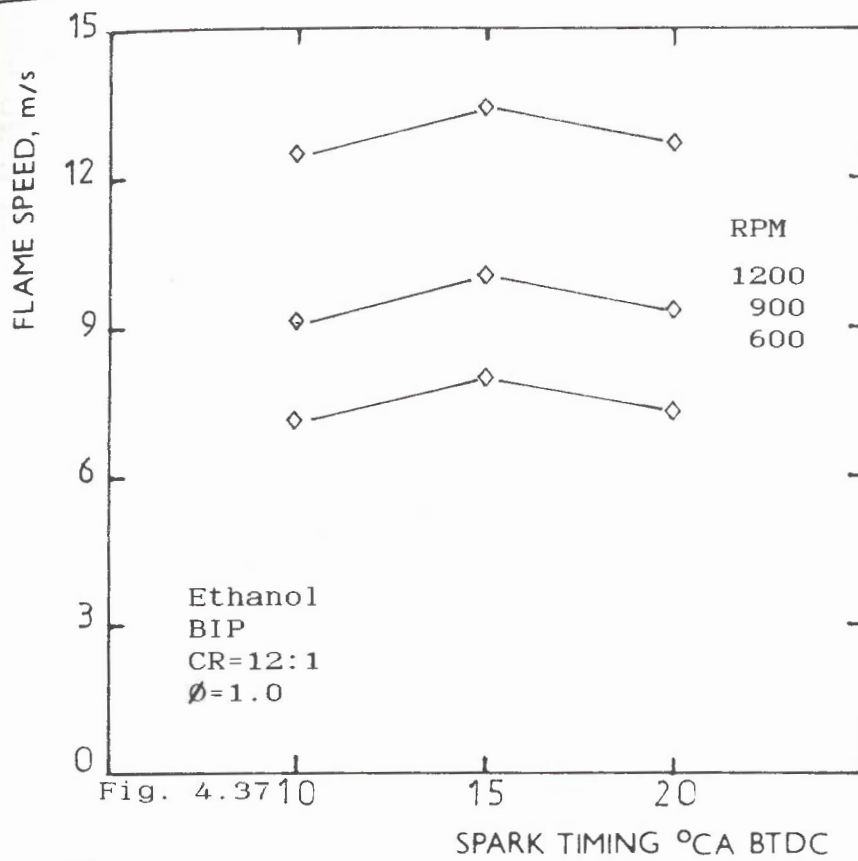
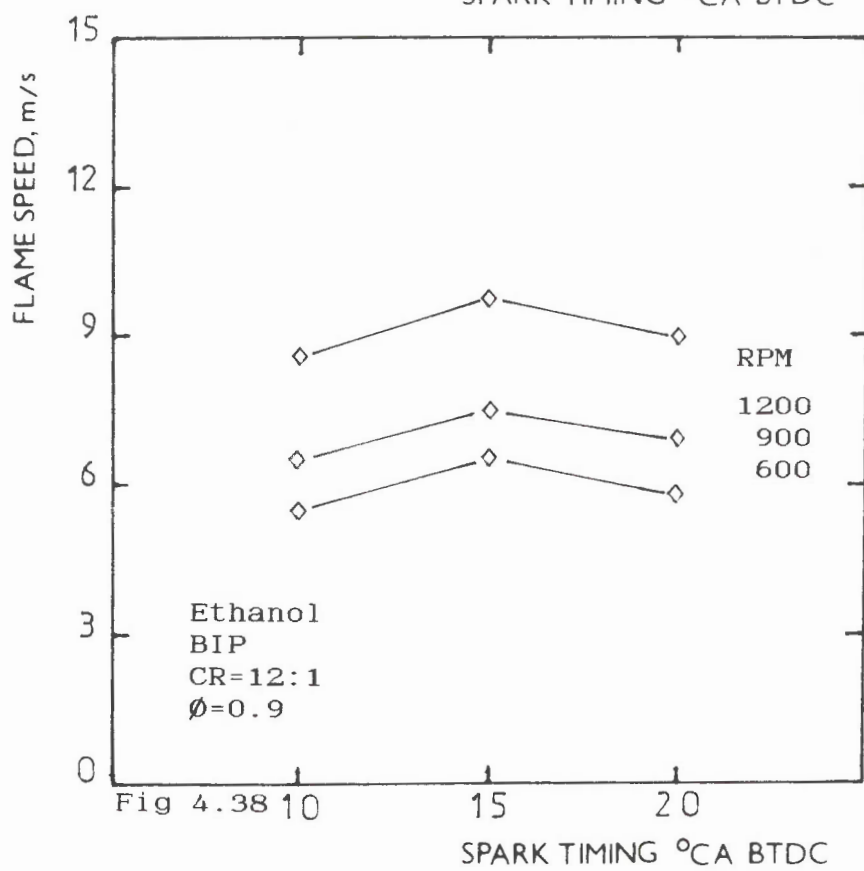
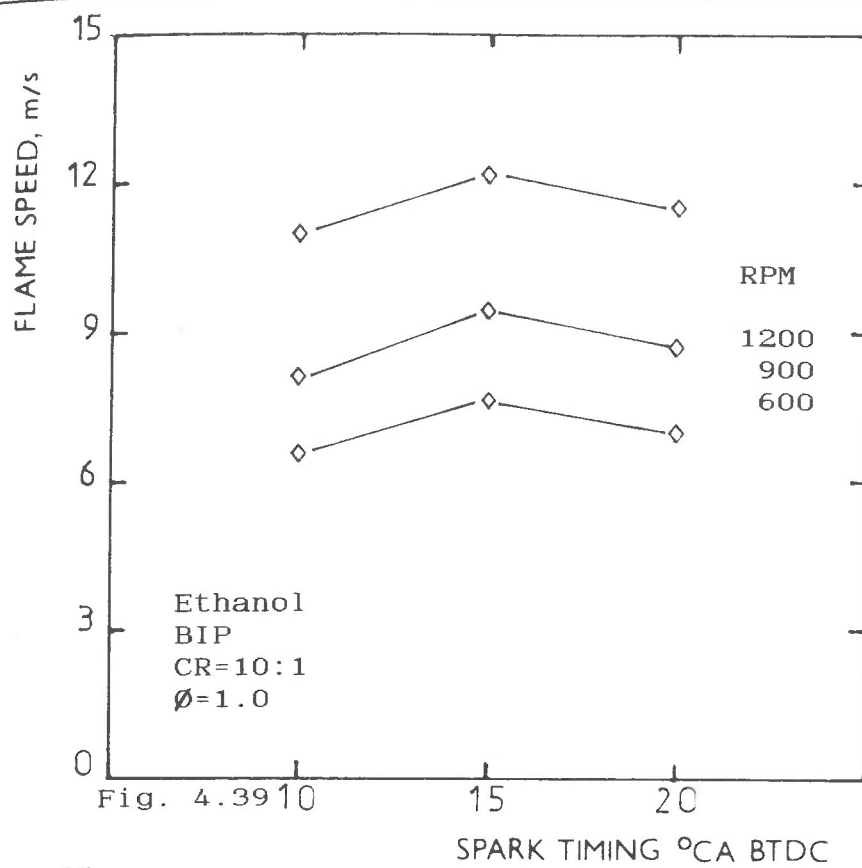
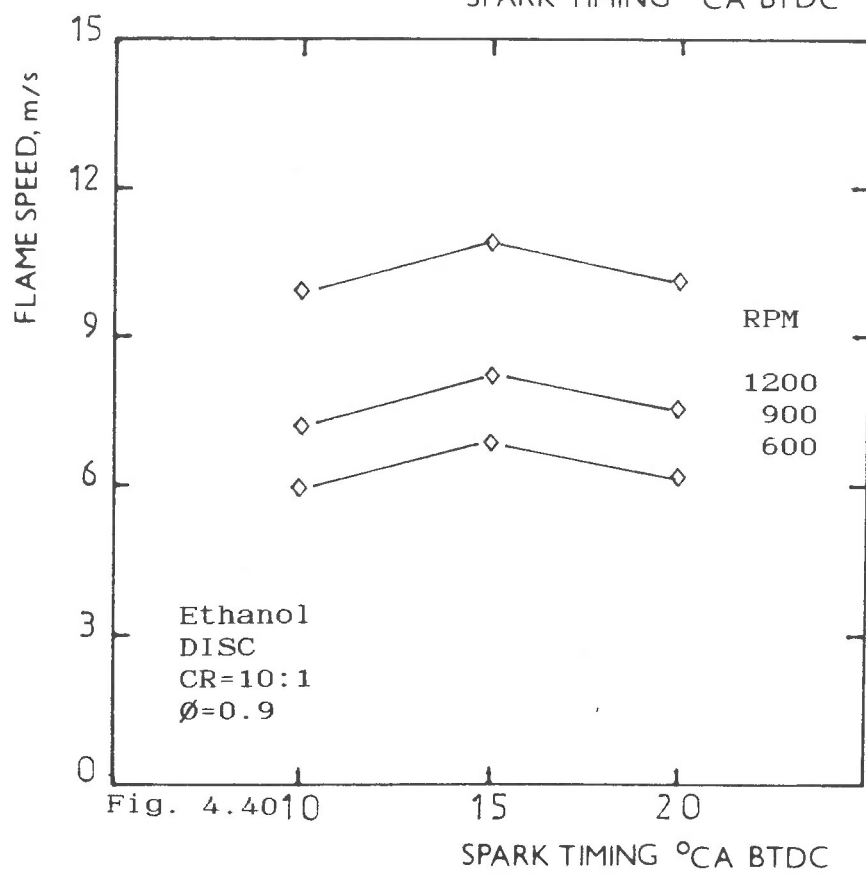
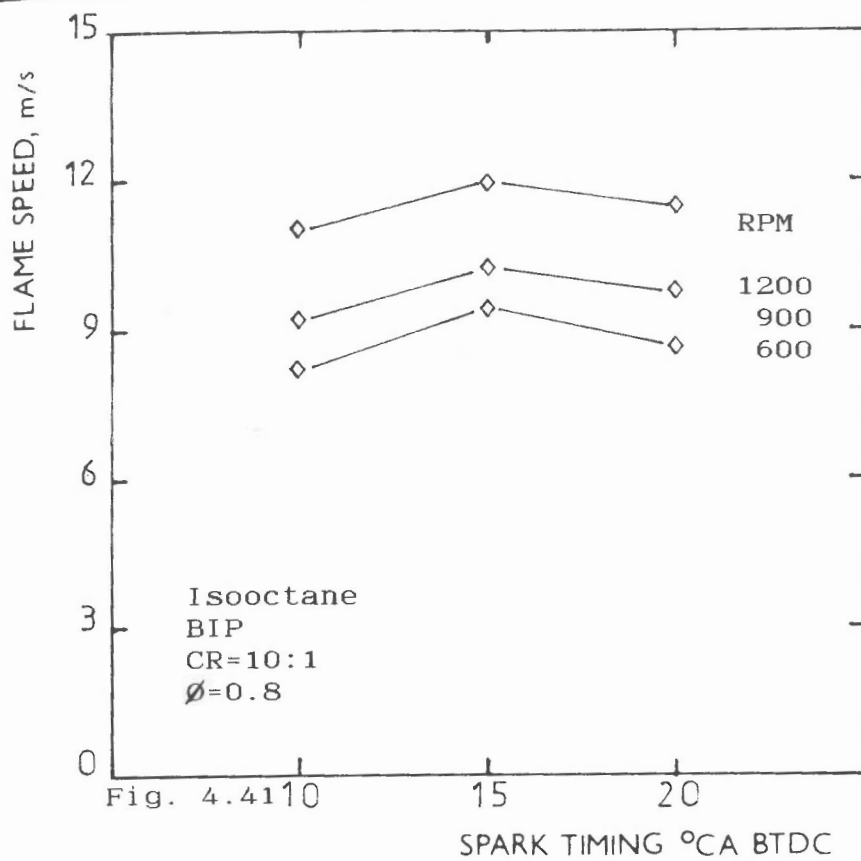
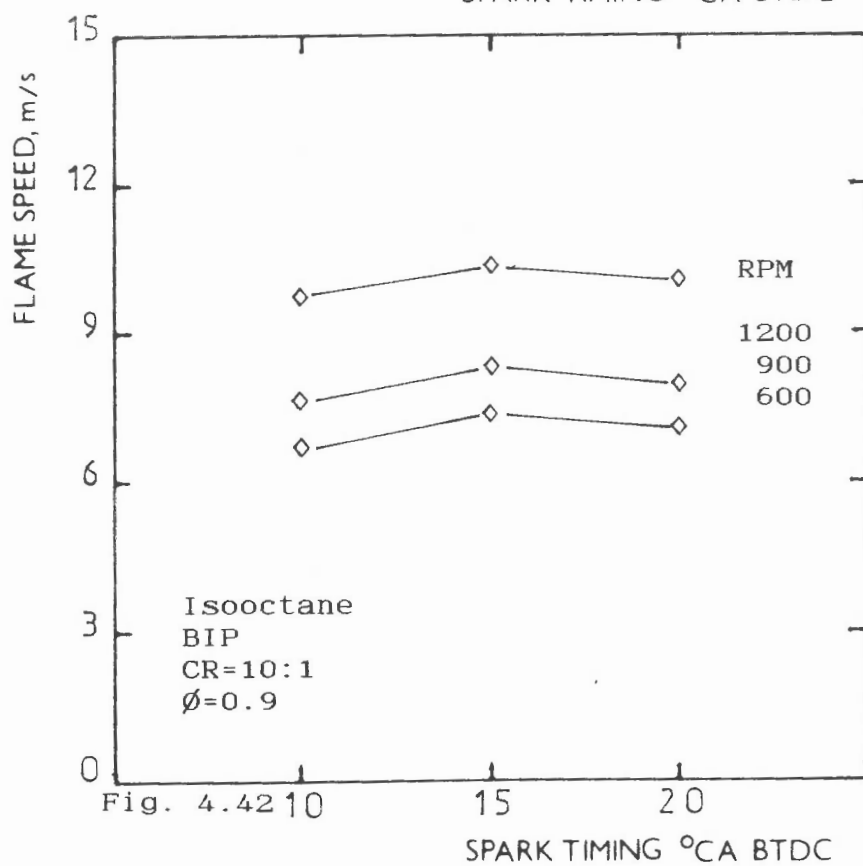


Fig. 4.35 - 4.42 Flame speed versus spark timing for DISC and BIP chambers









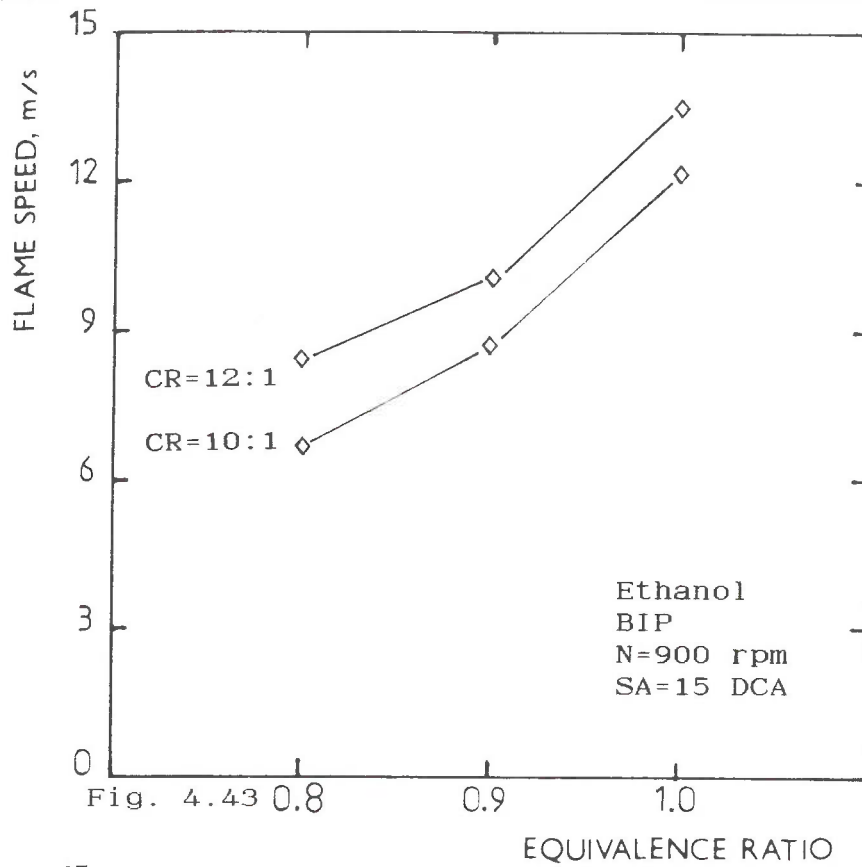
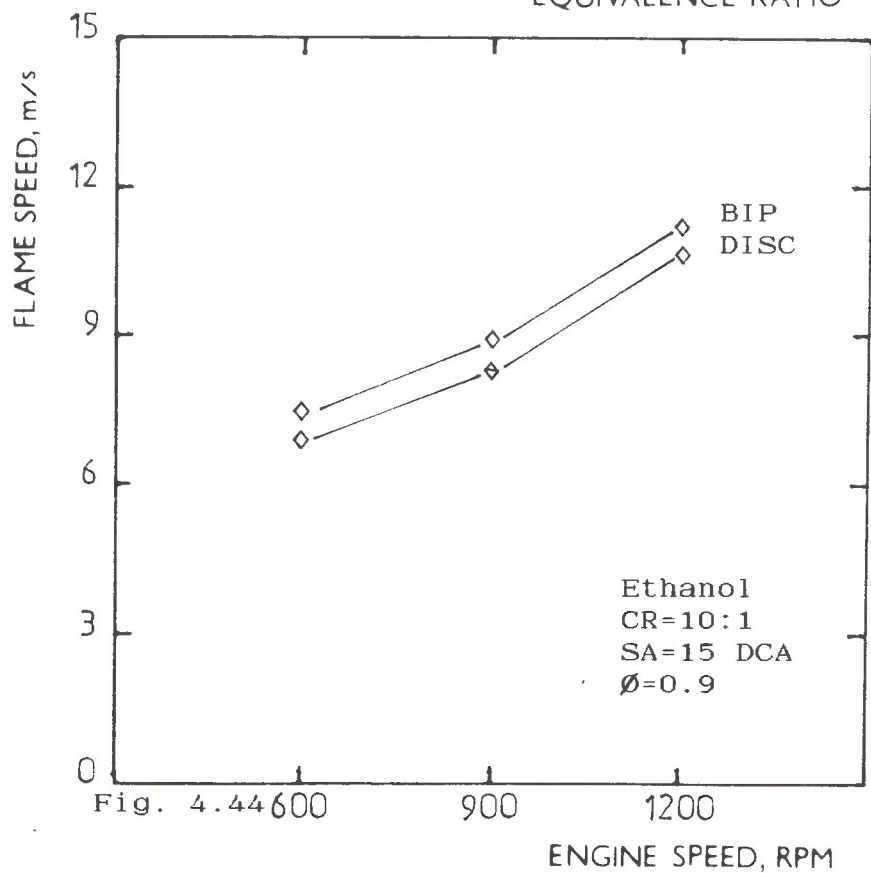


Fig. 4.43 Flame speed versus equivalence ratio for the BIP chamber at two compression ratios.  
 Fig. 4.44 Flame speed versus engine speed for DISC and BIP chambers.

# Chapter 5

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## Analysis of the Computer Model Predictions

## 5.1 Introduction

In the present chapter, the results obtained using the predictive computer programme are presented. The programme allows the effect of changes in engine operating parameters such as equivalence ratio, spark timing and engine speed to be readily assessed. The impact of design parameters such as spark plug position, compression ratio, combustion chamber geometry and type of fuel are also simulated.

Results for the disc and bowl-in-piston combustion chamber, both operated with ethanol and iso-octane are presented and discussed for a range of engine parameters. Results of the geometrical model simulation and the influence of engine parameters on flame front area and burned volume, for example, are also presented and discussed.

Next, the combustion and geometrical models are used to investigate the influence of parameters such as turbulence intensity, turbulent Reynolds number and laminar burning velocity on some of the widely reported correlations used to calculate turbulent burning velocity, a critical component of the modelling.

Finally, a comparison between the theoretical predictions and experimental measurements is presented and discussed.

## 5.2 Geometrical model results

The encouragement by design of faster burning processes in spark ignition engines brings the advantages of combusting very dilute fuel-air mixtures with attendant increases in engine efficiency. It also permits the use of higher compression ratios without the onset of knock, increasing still further opportunities for improved engine efficiency.



Two important processes can be employed to increase the burning processes in the spark ignition engine. One is by means of increasing the turbulence of the fresh charge via intake flow restrictions or combustion chambers with high squish areas.

When turbulence is increased, the transport rates within the flame front are increased and also flame front area is enhanced, both increasing the mass burning rate. However, this method of producing high turbulence must be utilised with care because the increased turbulence in the combustion chamber results in greater heat losses to the coolant medium via the cylinder walls, and at very high levels, extinction of flame propagation by local strain is also possible.

The second approach of producing a higher burning rate is via the increase in average frontal area of the flame. In order to achieve faster burning with this technique, it is necessary to reduce the contact between the wall and the flame to a minimum, which implies the utilisation of a small surface to volume ratio combustion chamber. It would be highly desirable to have also the spark plug moved to a more central position.

The great advantage of this technique for achieving high burning rates is that it minimizes the heat losses to the coolant. According to the definition given by Engels and Pirault (1984), compact chambers may be characterized by squish ratios of 30 - 50 %, with piston to head clearances varying from 1 - 1.5 mm. In these chambers, swirl ratios are below 1 : 1, with compression ratios in the range of 10.5 - 12.5.

According to this classification, the bowl-in-piston combustion chambers utilized in the present work may be regarded as a compact chamber, having squish ratio of 50

%, bumping clearance of 1 mm and compression ratio of 10 and 12 : 1. The Ricardo E6 engine, by its geometry, has a very low level of intake swirl and thus may be regarded as having a quiescent combustion chamber.

As stated earlier in this work, the geometrical model assumes spherical flame front propagation inside the combustion chamber. An evaluation of the interactions between the average spherical flame front and the contours of the combustion space can, therefore, be made from purely geometrical considerations. The results can be made largely independent of any modelling assumption made on the thermodynamic model, since the only input parameter to the geometrical model is the evolution of the flame radius with time.

Basically, three types of plots are necessary to realise an evaluation of the combustion chamber performance, namely

- .flame front area versus flame radius
- .enflamed volume versus flame radius
- .wetted areas versus flame radius

The flame front radius has been normalised by the cylinder bore, the areas by the cylinder bore squared and the volume by the cube of the cylinder bore. Another parameter of interest introduced in the simulation programme is the spark plug position, which was found to produce a dramatic influence in the evolution of the flame front area.

Typical plots of non-dimensionalized piston wetted area, wall wetted area, head wetted area and total area wetted by the flame are depicted in Fig 5.1. The flame travels along a curve of increasing flame radius. As the ignition occurs before TDC, the flame first travels on a path of decreasing clearance volume as it reaches the TDC and then travels on a path of increasing clearance volume during the expansion, a function of the crank angle

For small values of the flame radius, the crank angle has no influence upon the flame front area. The latter will be affected by the crank angle when the flame radius is equal to the distance from its origin to the nearest combustion chamber wall, which may be the piston crown or the cylinder wall. For central ignition chambers, this distance will be greater than for lateral ignition chambers. It can also be seen from Fig 5.2 that the flame radius required for the flame to propagate through all the combustion chamber is smaller for the central ignition chamber than for the lateral ignition.

The flame front areas vary greatly from the disc combustion chamber to the bowl-in-piston chamber in the normally useful range of 15 - 25 DCA from TDC. The bowl-in-piston chamber with lateral ignition showed a greater area, approximately 20 %, than the flame area presented by the disc chamber with lateral ignition as depicted in Fig 5.3 and Fig 5.4.

From the above, it can be inferred that flame front area is heavily dependent upon spark plug position for a given combustion chamber geometry. Again the largest gain in area is for the bowl-in-piston chamber. Moving the spark plug from a position 26 mm from the cylinder axis to a position 10 mm from this axis, flame front area increased approximately 67 % near TDC as can be seen in Fig 5.4.

When comparing the different curves produced by both types of chamber, a pattern appear to exist. For chambers with lateral ignition, the flame front area increases slowly with the flame front radius, remains approximately constant during most of the burning period and then decreases slowly till the end of the combustion process. For the cases where the spark plug is positioned centrally, there is a steep rise in flame front area up to a maximum value and then a sharp decrease as the flame front reaches

the combustion chamber walls. Fig 5.3 illustrates this point.

Squish ratios, defined as the ratio of the area of the bowl annulus to the cylinder bore area also affect the maximum flame area developed during the combustion as can be seen in Fig 5.5, a plot of flame area versus flame radius and in Fig 5.6, a plot of flame area versus enflamed volume. The chamber with squish ratio of 50 % produces a larger flame front area as compared to the 20 % squish ratio, both chambers having lateral ignition.

Fig 5.7 and Fig 5.8, are plots of flame front area versus enflamed volume, both normalized by the cylinder bore. The ratio of the flame front area in the case of the chamber with central ignition to that of the flame front area for the chamber with lateral ignition, increases in direct proportion with the enflamed volume, suggesting that the influence of the spark plug location is much greater for fully developed flames than for flames in the initial stages of formation.

Heat loss to the coolant is another important variable that is strongly dependent upon the geometry of the combustion chamber and of the areas of the later, wetted by the expanding flame front. The greater influence on heat transfer is produced by the cylinder head wall, where the largest and for some time the only one encountered area for heat transfer is located. Later in the cycle the flame front reaches the piston crown and the wetted piston area starts to contribute to the heat losses from the enflamed volume. Normally, the last wall to contribute to the heat losses is the cylinder wall.

The contribution of each area to the heat losses from the enflamed volume will be strongly dependent upon the

position of the spark plug. The greater loss will be produced by a combustion chamber with the spark plug positioned laterally in the combustion chamber since the flame wets the walls much earlier than if the spark plug was placed in the center of the chamber. A further delay, in the contact of flame with the cylinder head wall can be produced if the spark plug electrodes are located well inside the combustion space.

### 5.3 Combustion model results

The effects that parameters such as spark plug position have on the flame front area and burned volume have been considered in the previous section. In this section, predictions with the computer model showing the effects of the geometrical parameters on the combustion process are presented.

In order to allow comparisons to be made, the engine parameters and operating conditions have been maintained at fixed values for most of the predictions. These conditions are as follows

|                    |           |
|--------------------|-----------|
| cylinder diameter  | 76.20 mm  |
| stroke             | 111.13 mm |
| con-rod length     | 240.00 mm |
| spark plug off set | 26.00 mm  |

The operating parameters varied during the study were the spark advance, engine speed and equivalence ratio and their effects upon the cylinder pressure and mass fraction burned examined. Compression ratio and fuel type were also varied and their influence upon combustion assessed. The extent to which geometric parameters affect the combustion characteristics is also considered.

Fig 5.9 and Fig 5.10 represent typical plots of the



pressure versus time, expressed in crank angles, and the mass fraction burned versus crank angle, respectively, for a disc combustion chamber operating on ethanol.

The effect that compression ratio has on the combustion is indicated by the pressure-time and mass fraction burned plots depicted in Fig 5.11 and Fig 5.12, respectively. As is clearly shown, an increase in CR from 10 : 1 to 12 : 1, caused an increase in the peak cylinder pressure of the order of 26 %. The peak pressure also occurred approximately 2 DCA before. This fact can also be seen in the mass fraction burned curves, where for a given CA, the amount of mixture burned is greater for the higher compression ratio.

Besides a pressure increase with increased compression ratio, the temperature of the combustible mixture is also increased at the moment of the spark event and the amount of residual gases is reduced with a greater compression ratio. Both conditions, higher temperature and lower residuals tend to produce a favourable influence upon the laminar burning velocity, thus producing a faster combustion. However, increasing the compression ratio, leads to an increase in the surface area to volume ratio of the combustion chamber, thereby augmenting the heat losses to the coolant medium. The emission of unburned hydrocarbons is also expected to increase due to an increase in the surface area to volume ratio. The higher temperature attained in the combustion chamber tends to increase the production of NO<sub>x</sub> when the engine operates at higher compression ratios on a lean mixture.

Fig 5.13 and 5.14 depict the effect that the spark advance has on the pressure-time and mass fraction burned. Increasing the spark advance caused the peak pressure to increase and also to occur earlier in the cycle. The temperature of the combustion process is also affected by the spark advance. Higher spark advance leads to higher pressures and temperatures which can increase the NO<sub>x</sub>

emissions and the tendency to knock.

The pressure-time plots corresponding to the variation in the engine speed do not show much variation. As the engine speed increases the flame travel times, expressed in ms, decrease and the flame travel times expressed in crank angles increase. Consequently, for a given crank angle, the amount of mixture is reduced when the engine speed increases as depicted in the mass fraction burned curves of Fig 5.16. The curves of cylinder pressure also show this tendency, with the curve representing the higher engine speed placed below the one representing the lower engine speed. The higher speed diagram crosses that for the low speed because combustion is still taking place in the former case whilst in the second, combustion is completed. At the same time, at a higher speed the heat losses per cycle decrease because less time is available for the heat exchange between the gas and the walls.

The effect that equivalence ratio has upon the combustion process can be seen on Fig. 5.17 and Fig. 5.18. The mixture composition influences the rate of combustion, changing the pressure and temperature of the gases inside the cylinder. For a given engine speed and spark advance, decreasing equivalence ratio decreases the maximum peak cylinder pressure. For a given crank angle, the mass fraction burned is reduced. Also, the rate at which fresh mixture is consumed is smaller for the smaller equivalence ratio. This is in agreement with observations made on the flame propagation velocity experimentally, as was demonstrated in the previous chapter. The slower burn rate, with consequently greater flame travel times, implies reduced flame propagation velocities,

The duration of the combustion process is extended some crank angles, as can be seen from Fig 5.19, by the instant when the flame radius reaches its maximum value. In the present case, for  $\phi = 0.8$  and  $\phi = 1.0$ , the difference in



combustion duration is approximately 4 DCA. In terms of flame front area, there had been an increase, however small, for the higher equivalence ratio as depicted in Fig 5.20, which supports the increase in flame speed with the increase in equivalence ratio.

The effect of the spark plug position upon the combustion process is represented in Fig 5.21 and Fig 5.22 for a bowl-in-piston combustion chamber. It can be seen that the changes produced by the spark plug position are much more dramatic than the changes produced by equivalence ratio and engine speed, for example.

When the spark plug is moved to a central position in the combustion chamber, the area of the flame front is maximized in relation to the area corresponding to a laterally positioned spark plug. The benefits produced are evident from the pressure-time curve where an increase of approximately 10 bar in the peak pressure resulted from the prediction. The rate of pressure rise is also increased and the peak pressure occurs earlier in the cycle, roughly 2 DCA.

The mass fraction burned also presents a steeper curve as depicted in Fig 5.22. However, higher rates of mass burned, with the consequently higher rates of pressure rise, may not be interesting for a real engine, where smoothness of operation may dictate a less severe burn rate. This behaviour is typical for the bowl-in-piston chamber. Fig 5.23 and 5.24 are plots of burned volume fraction versus flame radius and burned volume fraction versus crank angle for the two cases of spark plug location ie, lateral and central.

The rate at which the enflamed volume grows is enhanced by positioning the spark plug centrally in the combustion chamber. It can also be seen that the maximum flame radius is smaller for the central spark plug when compared to the

lateral spark plug position. Combustion duration is reduced by placing the spark plug in the center of the combustion chamber as can be seen from the smaller crank angle at which the burned volume curve terminates in the case of the central positioned spark plug.

Still considering this type of combustion chamber, the other variable of interest was the squish ratio. This parameter has been defined as the area of the piston annulus divided by the area of the cylinder bore, and is expressed as a percentage. The two values examined were 20 % and 50 % squish ratios, for a chamber operating on ethanol at SA = 10 DCA and  $\phi = 1.0$ . The changes produced in the cylinder pressure were negligible, with a slightly smaller crank angle at which pressure is maximum, as depicted in Fig 5.25. The mass fraction burned curves, show a slightly different pattern, showing a higher mass burn rate for the chamber with 50 % squish ratio. This can be explained by the higher flame front area produced by the 50 % squish ratio combustion chamber as shown in Fig 5.26.

Squish velocity, as predicted by the programme according to the theory developed in chapter 2 is presented in sequence. Fig 5.27 depicts the effect of engine speed upon the squish velocity. Doubling the engine speed, from 600 to 1200 rpm caused the squish velocity to nearly double its value. The maximum value for each condition of speed always occurs at 10 DCA BTDC, a characteristic due to the peculiarity of the piston movement. The maximum squish velocity goes through a maximum later than the piston velocity, peaking 10 DCA BTDC.

Fig 5.28 indicates the effect of the squish ratio on the squish velocity. The increase in squish velocity when the squish ratio is increased from 20 % to 40 %, for example, is of the order of 130 % ie, doubling the squish ratio more than doubles the squish velocity.

The effect of the distance between cylinder head and piston crown or bumping clearance, when the piston reaches TDC, is shown in Fig 5.29. By halving the bumping clearance, from 1.0 to 0.5 mm, the maximum value of the squish velocity is increased by approximately 55 %. However, a basic difference appears from the Fig 5.29, When the squish velocity is increased by one of the two methods described previously, the squish velocity is substantially increased during relatively early stages of the compression stroke. However, when the bumping clearance is reduced, the remaining conditions being the same, a substantial increase in the squish velocity occurs only later in the cycle when the piston is approaching TDC, as depicted in Fig 5.29.

In practice, the process of increasing the squish velocity by increasing the squish ratio is preferred because the process that uses small bumping clearances tends to increase manufacturing costs due to the adoption of smaller dimensional tolerances dictated by the very reduced distance between the piston crown and the cylinder head.

Fig 5.30 and Fig 5.31 are plots of burned gas temperature versus crank angle and unburned gas temperature versus crank angle for a disc combustion chamber operating on iso-octane at  $SA = 15$  DCA and  $O = 1.0$ . Both curves show the expected tendencies, and reference to them will be made as and when necessary.

Fig 5.32 shows the variation of the flame radius with crank angle for the same combustion chamber.

Fig 5.33 present a typical plot of turbulence intensity versus crank angle according to the model described in chapter 2. The magnitude of the turbulence intensity is a function of engine speed and the curve shown is for a disc chamber and engine speed of 900 rpm.

The integral length scale is an additional parameter used to calculate turbulent burning velocity and other related parameters such as the Reynolds number. As the integral length scale represents in the same sense the maximum possible size of coherent eddies, it is interesting to make this parameter variable once the boundaries of the combustion chamber are also varying during the combustion process.

Therefore, the maximum size of the turbulent eddies is varied according to the piston position. For a three dimensional enclosure, an integral length scale can be estimated from the surface area to volume ratio of the combustion chamber

$$L = V / A$$

This procedure has been described in chapter 2. A typical plot of the variation of the integral length scale with crank angle is given in Fig 5.34. This result is interesting because it is sensitive to the geometry of the combustion chamber.

The surface area varies between different chamber types while the volume is constant for a given compression ratio, therefore, making the integral length scale dependent on the chamber area is a realistic approach towards varying  $L$  with geometry.

#### 5.4 Sensitivity of the model to various turbulent burning velocity correlations.

The zero-dimensional turbulent combustion model developed

in the present work has been used to study the effect that different correlations for turbulent burning velocity proposed by various researchers have on engine simulation parameters.

The present model, as described in chapter 2, was developed making use of the correlation for turbulent to laminar burning velocity ratio or flame speed ratio, proposed by Lancaster (1976). The results of the simulation using such a correlation were presented earlier in section 5.3. It was found to have a good agreement with the results found during the experiments with the Ricardo E6 engine as will be shown later in section 5.5.

However, it was decided to attempt to study the differences produced by the simulation when alternative correlations were used in place of that due to Lancaster (1976). The alternative correlations used in this study are presented in Table 5.1.

From the seven correlations presented, those due to Lancaster (1976), Tabaczynski (1977) and Groff and Matekunas (1980) were derived by correlating data from several engine experiments. Lancaster (1976) derived his correlation by considering the results of experiments with a CFR engine, while Groff and Matekunas (1980) derived a correlation by curve fitting to the data of 96 experiments performed on a research engine where four different positions for the spark plug were investigated.

During the experiments, different degrees of turbulence were generated by the use of shrouded and non shrouded valves. Burning velocities were determined from the pressure-time data by means of a diagnostic spherical flame model. In their correlation, they introduced a pressure dependent term in order to account for the thickness of the flame front. In a similar manner to the correlation proposed by Lancaster (1976), their



correlation is valid only for the period where the flame is considered fully developed which they claim to be 30 mm. Below that value a correction factor, from now on called flame growth factor FGF, should be used.

The FGF suggested by Lancaster (1976), as presented earlier, is

$$FGF = (R_f / 30)^{0.5} \quad (2)$$

which should multiply the fully developed FSR in order to evaluate the FSR during the early flame development.

The expression proposed by Groff and Matekunas (1980) to be used as a multiplier in the fully developed flame correlation is given by

$$FGF = 1.07 (R_f / 30)^{1.04} \quad (3)$$

which suggests a nearly linear dependence of the flame radius. In the expressions presented in Table 5.1 the factor  $S_f$  varies from 1.0 to 1.2 according to the expression

$$S_f = 1.0 + 0.05 \theta^{0.4} \quad (4)$$

when  $\theta$ , the spark advance varies from 0 to 30 DCA and is introduced as an optimization parameter according to the authors.

However, the magnitude of the flame radius considered to represent a fully developed flame seems to be somewhat arbitrary. Although the above investigators considered the flame to be fully developed after it reaches a radius of at least 30 mm, Patterson (1973) has claimed that, in his experiments, the flame was fully developed after the flame front had traveled 10.7 mm. Recently, Kalghatgi (1986) using a schlieren technique to measure flame propagation



velocity in a Ricardo E6 engine, found that the flame could be considered totally developed after its radius reached the 11 mm. The criterion adopted was that of the constancy of the flame propagation velocity during a certain distance in the combustion chamber, from the spark plug.

Experiments conducted in the present work, with a technique similar to that used by Kalghatgi (1986) also in a Ricardo E6 engine, suggest that flame propagation velocities measured 15 mm from the spark plug are in agreement to those found by the previous investigator.

However, in the present work, the distance of the point of measurement from the spark plug could not be varied and thus the verification of the flame propagation velocities according to the criterion mentioned earlier could not be achieved.

Therefore, the criterion of whether a flame is fully developed at 30 mm radius or less than that is open to debate. Some attempts to use a flame radius less than 30 mm in the above correlations were made and the results will be discussed later.

The third and fourth correlations considered were those due to Tabaczynski (1977) and Daneshiar and Hill (1987). Tabaczynski (1977) derived his correlation by considering more fundamental properties of turbulent flow, for example, the integral length scale, micro-scale and turbulence intensity. His model is based on the concept that the flame spreads via a turbulent entrainment process and that burning occurs on a scale characteristic of the non-uniformities within the entrainment front.

In the expressions presented on Table 5.1, the turbulent burning velocity has a strong dependence of  $u'$ ,  $(u'^{2/3})$  and  $u_t$  ( $u_t^{2/3}$ ) and a weaker dependence on the scale ( $L^{1/3}$ ). The

| AUTHOR                                       | REFERENCE                              | CORRELATION                                                                                                                                                                                                           |
|----------------------------------------------|----------------------------------------|-----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| LANCASTER                                    | LANCASTER et al<br>(1976)              | $u_t/u_\ell = 1.0 + 4.01 (u'/u_\ell)$                                                                                                                                                                                 |
| GROFF + METEKUNAS                            | GROFF AND MATEKUNAS<br>(1980)          | $u_t/u_\ell = 2.0 + 1.21 (u'/u_\ell) (P/p_m)^{0.82} S_f$                                                                                                                                                              |
| DANESHIAR AND HILL                           | DANESHIAR AND HILL<br>(1987)           | $u_t/u_\ell = 1.0 + \sqrt{\frac{2}{3}} \sqrt{\rho_{u/\ell_b}} \left(\frac{u'}{u_\ell}\right)$                                                                                                                         |
| KLIMOV                                       |                                        | $u_t/u_\ell = C (u'/u_\ell)^{0.7}$                                                                                                                                                                                    |
| CLAVIN + WILLIAMS                            | CLAVIN AND WILLIAMS<br>(1979)          | $u_t/u_\ell = 1.0 + (u'/u_\ell)^2$                                                                                                                                                                                    |
| TABACZYNSKI, FERGU-<br>SON AND RADHAKRISHNAN | TABACZYNSKI et<br>al (1977)            | $u_t/u_\ell = C (u'/u_\ell)^{1/3} (\rho_u/\rho_b)^{1/9} (L_{u'}/\gamma)^{1/3}$                                                                                                                                        |
| ABDEL-GAYED + BRADLEY                        | ABDEL - GAYED<br>AND<br>BRADLEY (1981) | $u_t/u_\ell = \left[ 1.34 RL \frac{u'}{u_\ell} \left[ 1 - \exp(-3.178 RL \frac{u'}{u_\ell}) \right] \right. \\ \left. + 10.9 RL \frac{u'}{u_\ell} \left[ 1 - \exp(-0.391 RL \frac{u'}{u_\ell}) \right] \right]^{0.5}$ |

TABLE 5.1

term  $L u' / \gamma$  is a turbulent Reynolds number based on the integral scale, while the constant  $C$  is to be determined experimentally for a specific engine geometry. An estimate of the turbulence intensity and the integral length scale is given by the expressions below, derived by considering the conservation of angular momentum by the turbulent eddies during compression and expansion as postulated by the Rapid Distortion theory.

$$u' = u'_i (p / p_m)^{1/3 \gamma} \quad (5)$$

$$L = L_i (p / p_m)^{1/3 \gamma} \quad (6)$$

According to expression (6), the integral length scale grows continuously during the combustion process, from its value at ignition, for example, and is independent of the geometry of the combustion chamber, an approach different from that adopted originally in the present work which considers an integral length scale as a function of the geometry of the combustion chamber, thus varying during the combustion process according to Fig 5.34. However, to simulate the cycle, the expressions for turbulence intensity and integral length scale were substituted for the relations utilized in the present work. The constant  $C$  was found to be 0.33 in order to give pressure-time curves similar to the experimental findings with the Ricardo E6. The results of the simulation utilizing such a correlation will be presented later.

Daneshiar (1987) presented a correlation that prescribes a linear dependence of the turbulent burning velocity on the turbulence intensity to laminar burning velocity ratio. He derived his correlation based on the Tennekes (1968) model of small scale turbulence structure. According to this model, the small scale structure of turbulence may be modeled as vortex tubes of diameter of the Kolmogorov scale which are stretched by eddies of the size of the Taylor microscale. The main combustion implications of the

Tennekes (1968) model are the slow burning at the laminar burning velocity in the relatively quiescent regions of the size of the Taylor micro scale and the fast burning along the vortex filaments of the size of the Kolmogorov scale.

This means that when a large eddy of the size of the integral scale is entrained within a thick turbulent flame it will contain several regions of the size of the Taylor micro scale where the burning is slow, but the boundaries of these regions may be rapidly enflamed because of the rapid flame propagation along the dissipation regions of the size of the Kolmogorov scale.

The expression given in Table 5.1 was simplified by its author assuming a density ratio of 6, corresponding to the burning of hydrogen fuel stoichiometrically in air at atmospheric pressure, which reduced the expression in Table 5.1 to

$$u_t / u_l = 1 + 2 u' / u_l \quad (7)$$

The expression was found to correlate well with data on turbulent burning velocities during burning at atmospheric pressure presented by Abdel-Gayed and Bradley (1981) for turbulent Reynolds number  $R_L > 1000$  and rms turbulent to laminar burning velocity ratio,  $u' / u_l > 10$ . Expression (7) is similar to the expression derived by Lancaster (1976) and the results obtained with the simulation programme using these expressions are presented in section 5.5.

The remaining 3 correlations incorporated in the present work were not developed from internal combustion engines experiments. The correlations are those due to Clavin and Williams (1979), Abdel-Gayed and Bradley (1981) and Klimov (1983).

The expression suggested by Clavin and Williams (1979) was

derived from a statistical theory for the structure and propagation velocity of premixed flames in turbulent flows having scales larger than laminar flame thickness. For isotropic turbulence and for small values of rms turbulent to laminar burning velocity ratio, the expression proposed by the authors is

$$u_t / u_l = 1 + (u' / u_l)^2 \quad (8)$$

Because the limit of validity of the expression is for low values of  $u' / u_l$  the relationship is not well-suited for engine combustion applications where larger values of  $u' / u_l$ , in excess of 2 are normally expected to be found. This was confirmed by the results of the simulation presented in section 5.5.

Abdel-Gayed and Bradley (1981) suggested that, on the basis of a wide range of experimental evidence, the ratio of the turbulent to laminar burning velocity,  $u_t / u_l$  could be correlated with the ratio of the RMS turbulent velocity fluctuations of the reactant mixture to its laminar burning velocity  $u' / u_l$ , and also with the turbulent Reynolds number based upon the integral length scale  $L$ , for the combustion of turbulent premixed gases.

The theory developed assumes uniform, isotropic turbulence and that the large and small eddy structures coexist in dynamic equilibrium. The smaller scales of turbulence are produced by the vortex stretching of larger eddies by the increasing turbulence in the flow field due to velocity gradients.

Through the agency of viscous action, the energy is dissipated in randomized molecular motion. As the level of turbulence increases the combustion process is increasingly concentrated within smaller length scales. The theory developed, called by the authors the Two-Eddy Theory, considers that the spectrum of turbulence is



simplified into the two eddy sizes, namely  
 .large eddies of the size given by the integral length scale, having a life time  $t_L = u' / L$  and  
 .small dissipative eddies of the size given by the Kolmogorov micro scale  $\eta$ , having a life time  $t_\eta = u' / \lambda$ .

Based on fundamental quantities for turbulent flow, the theory yields the following approximate expression for the turbulent to laminar burning velocity ratio

$$u_t / u_l = \{ 1.34 R_L^{0.47} u' / u_l \} \\ \{ 1 - \exp (-3.178 R_L^{-0.5} u' / u_l) \} \\ + 10.9 R_L^{0.12} u' / u_l \\ \{ 1 - \exp (-0.391 R_L^{-0.25} u' / u_l) \} \}^{0.5} \quad (9)$$

The correlation presents a slight departure from the linearity as can be seen by the evidence presented by the authors in their paper. The above expression was applied in the present simulation using the same procedures adopted by the other correlations regarding their limit of validity ie, from flame radius greater than or equal to 30 mm and the results are presented and discussed in section 5.5.

Klimov (1983) based on a large scale turbulent model ie, a model in which the integral length scale of the turbulent flow is larger than the flame front thickness, presented the following expression for the estimate of the turbulent to laminar burning velocity ratio

$$u_t / u_l = (1 / \alpha)^{0.3} (u' / u_l)^{0.7} \quad (10)$$

where  $\alpha$  is a constant considerably smaller than unity. Based on experimental evidence, for  $u' \gg u_l$

$$u_t / u_l \approx 3.5 (u' / u_l)^{0.7}$$



and it is this expression which is utilized in the present comparative study and from which results are presented in the next section.

### 5.5 Results predicted using different turbulent burning velocity correlations

In this section, the results of the simulation of the model using the 7 correlations described in section 5.4 are presented and discussed.

To assess the sensitivity of the model to the various correlations, the following operational parameters were selected as indicated below, and kept constant for all correlations

|                   |                |
|-------------------|----------------|
| chamber type      | bowl-in-piston |
| spark off set     | 26 mm          |
| compression ratio | 12 : 1         |
| rotational speed  | 900 rpm        |
| spark advance     | 25 DCA         |
| fuel              | ethanol        |
| equivalence ratio | $\phi = 1.0$   |

The reason underlying the choice of such parameters was to diversify the applicability of the model, and to test a range of different parameters from those analysed previously in section 5.3. In particular, the choice of a higher value of the spark advance reflects the observation that 15 DCA proved insufficient to produce a satisfactory rate of pressure rise in the cylinder.

Fig 5.35 presents the variation of cylinder pressure with crank angle for the 7 correlations studied. The correlations due to Lancaster (1976), followed by Tabaczynski (1977), presented the highest rate of pressure

rise for SA = 25 DCA. The correlation due to Groff and Matekunas (1980) might plausibly be taken as the lower limit acceptable.

The other 4 correlations, those due to Clavin and Williams (1979), Abdel-Gayed and Bradley (1981), Klimov (1983) and Daneshiar (1987) presented cylinder pressure significantly below this rather arbitrarily conceived limit. One key explanation for the observed behaviour evidently resides in the values of the turbulent burning velocity returned by the simulation programme using the respective correlations. The model is non linear and therefore no simple correspondence can be established.

However, since the mass fraction burned at each crank angle is estimated by the expression

$$\dot{m}_b = \rho_u A_f u_t \quad (10)$$

its value at each crank angle step depends strongly upon the turbulent burning velocity of the combustible mixture. Evidently, if the turbulent burning velocity is small, the amount of mixture burned during the crank angle step will also be small causing the mixture overall to burn more slowly. This in turn will cause the cylinder pressure rise to be delayed with respect to TDC. In some cases, the peak cylinder pressure was so retarded in relation to TDC, at SA = 15 DCA, that almost all the combustion occurred after the piston had passed TDC. In these cases there was no appreciable cylinder pressure rise, the engine behaving as if it had an extremely retarded spark timing. Fig 5.36 illustrates this point

The turbulent burning velocity is dependent on the ratio of the rms turbulent to the laminar burning velocity and here may reside one of the primary causes for the behaviour described. The turbulence intensity, as demonstrated by several workers among them Semenov (1963),

Groff and Matekunas (1980) and Liou et al (1984) decays with time in the crank angle interval during which combustion is said to occur.

The decay of turbulence intensity with time incorporated in the model was based on measurements performed in an engine in the motored mode, therefore, any increasing effects that combustion has upon the turbulence intensity as demonstrated for example by Witze (1984), are not included in the simulation. This caused the model to simulate turbulence intensity as in a non-firing condition.

Meanwhile, the laminar burning velocity  $u_L$ , is increasing during the combustion process since the charge temperature is steadily rising. This causes the ratio  $u'/u_L$  to be further reduced for each crank angle step. Accordingly the predicted ratio  $u_L/u_L$ , the flame speed ratio, may be significantly impaired by these effects, causing the mass fraction burned as calculated by expression (10) to evolve too slowly.

To minimize such shortcomings larger values of the spark advance were selected and the model again tested. The results presented in Fig 5.36 for the Daneshiar (1987) correlation confirm that advancing the spark leads to the mixture burning at a more favourable position of the piston relative to TDC, with a consequent increase in the rate of pressure rise.

Curves of mass fraction burned versus crank angle are presented in Fig 5.37 again for the seven correlations studied. For the same spark advance of 25 DCA, the total angle of combustion duration produced by each correlation tested is readily apparent. There is, for example, a remarkable difference between the two extreme cases, the fastest burn rate produced by Lancaster (1976) correlation and the slowest burn rate given by the Clavin and Williams

(1979) correlation, a difference of almost 50 DCA.

A complementary view of the combustion process is presented to that inferred from the pressure diagram. In the first case, the mixture is totally consumed after 33 DCA (6 ms at 900 rpm), while in the second case is totally consumed only after approximately 89 DCA (16.5 ms at 900 rpm). The second lowest rate of combustion was produced by the Abdel-Gayed and Bradley (1981) correlation, taking approximately 77 DCA (14.2 ms at 900 rpm) to complete.

The incorporation of alternative correlations has thus far been fairly simplistic and some considerations must be given to the limits of applicability. With the exception of the correlations due to Lancaster (1976) and Groff and Matekunas (1980) no other correlation prescribes a correction factor to take into account the early stages of flame kernel development.

In order to introduce some elements of consistency all the correlations were subjected to the same modification to account for this period of flame development, ie, they were all multiplied by the factor

$$FGF = (Rf / 30)^{0.5}$$

with exception of the expression suggested by Groff and Matekunas (1980) who indicate for their correlation the alternative factor

$$FGF = 1.07 (Rf / 30)^{1.04}$$

It was found in a subsequent simulation that the flame growth factor plays a very important role in determining the rate at which the flame speed ratio varies and as a consequence, the rate at which the mass is burned.

In order to evaluate the extent to which the coefficient FGF affects the correlations, two of them were used for simulations in which this factor was omitted. This implies that the correlation is valid from the point at which all reactant mixture is consumed.

The correlations considered were those due to Abdel-Gayed and Bradley (1981) and Clavin and Williams (1979). The reason behind the choice of these two particular correlations was because they presented the longest delay period during the earlier simulations. Both correlations were tested for the same engine parameters as in the first simulation.

Fig 5.38 represent a plot of pressure versus crank angle for the three spark advances selected and corresponds to the correlation due to Abdel-Gayed and Bradley (1981). As can be clearly seen, the changes in peak cylinder pressure are quite dramatic when compared with the earlier curve shown in Fig 5.35 at SA = 25 DCA. The maximum pressure attained in the first simulation was 35 bar while in the , second was 60 bar, an enhancement of 70 %. This improvement can also be seen in Fig 5.39 which presents the variation of mass fraction burned with crank angle.

The angle of combustion duration was reduced from approximately 75 DCA to 50 DCA which displaces the point of 50 % mass fraction burned MFB, from approximately 25 DCA ATDC in the first simulation to approximately 5 DCA ATDC, leaving most of the piston travel during the expansion stroke subjected to the high pressure gases.

The simulation with the correlation due to Clavin and Williams (1979) produced the same type of evidence with the omission of the FGF, leading to a dramatic increase in pressure and reduction of the time for combustion to be completed, as can be seen in Fig 5.40 and Fig 5.41.



A comparison between the cylinder pressure and mass fraction burned obtained from the simulation using both correlations is given in Fig 5.42 and Fig 5.43. There is an approximately 14 % gain in cylinder pressure when the Abdel-Gayed and Bradley (1981) correlation was used and a reduction of 6 DCA in the time spent to consume all the mixture. The results of this simulation confirm that the FGF factor does have a strong delaying effect over the evolution of the turbulent burning velocity.

The effect that the RMS turbulent velocity to the laminar burning velocity ratio has upon the combustion process in the engine was also verified in a further simulation. The correlation utilised was again that due to Abdel-Gayed and Bradley (1981). The values of the turbulence intensity calculated originally by the simulation model were multiplied by a numerical factor, this producing enhanced values of the turbulence intensity at a given crank angle.

The value of the factor was based upon the findings of Cole and Swords (1979) who reported on experiments undertaken in a Ricardo E6 engine, aiming to measure the turbulence intensity levels in the vicinity of the spark plug at TDC. They used laser doppler anemometry to measure turbulence intensity at various speeds of the engine in non fired mode.

For the Ricardo E6 engine running at 800 rpm, at points some 5 mm around the spark plug and piston at TDC, the turbulence intensity could be taken as approximately 2.5 m/s, which is close to the piston velocity, in this case 2.9 m/s.

Witze (1984) measuring turbulence intensity in a research engine in a firing mode found, at TDC, values of the turbulence intensity in the region of 0.8 m/s, with a rotational speed of 300 rpm. The piston velocity of his engine at 300 rpm was approximately 0.8 m/s, which again,



gives a turbulence intensity nearly equal to the mean piston velocity.

On the other hand the turbulence intensity calculated by the program is approximately equal to half the mean piston velocity, in conformity with the suggestions of Liou et al (1984). They also measured turbulence intensity in a motored engine using laser doppler velocimetry and came to the conclusion that at TDC, the turbulence intensity could be taken approximately as

$$u' \leq 0.5 V_p$$

If the relationship implied by the measurements of Witze (1984) is applied to an engine speed of 900 rpm, the corresponding mean piston velocity of 2.5 m/s would be representative also of the turbulence intensity. As both engines have the same combustion chamber geometry and are claimed to be fairly quiescent, the comparison seems to be realistic.

A further simulation using this increased value of the turbulence intensity was performed, employing the correlation of Abdel-Gayed and Bradley (1981) and making use of the FGF factor as before and the results can be seen in Fig 5.44.

When the turbulence intensity was thus enhanced, even retarding the combustion by the use of the FGF factor, the peak cylinder pressure was comparable to the cylinder pressure when the FGF factor was not present but the turbulence intensity was smaller. Clearly the effect of one factor tends to annul that of the other.

## 5.6 Comparison between the computer predictions and the experiments

The purpose of the present section is to present a comparison between the data obtained during the experimental work with the Ricardo E6 engine and the computer predictions using the same operational parameters.

The immediate aim of the experimental work was to obtain data on flame propagation velocity in the engine employing the laser technique described in chapter 4 and using the various equipment developed for that purpose and described in the same chapter.

In addition to the flame propagation velocity, cylinder pressure, fuel flow rate, air mass flow rate and engine load were measured and some results were presented in chapter 4. In this chapter additional information concerning these data will be given. The data presented are then utilised to verify the extent to which the model is able to simulate the behaviour of the engine.

### 5.6.1 Cylinder pressure

The objective of monitoring the cylinder pressure was to obtain information on peak cylinder pressure and to observe the cyclic variation in order to evaluate the behaviour of the engine during operation.

Highly precise measurements of engine cylinder pressure are required when the data is to be used to compute the combustion rates, for example. In this case the cylinder pressure data may be used in the diagnosis for engine simulation models but a small error in the cylinder pressure measurement may induce large errors in the derived parameters. A detailed analysis of the subject on cylinder pressure measurements can be found in the work of

Although practically the whole cycle can be displayed on an oscilloscope screen, only restricted degrees of interest were digitised, ie, the interval for digitisation was selected by software and limited to 50 DCA BTDC to 75 DCA ATDC. The data digitised correspond to the cylinder pressure averaged over 100 cycles selected as described in chapter 4.

In this chapter, cylinder pressure obtained experimentally in the Ricardo E6 engine is plotted with cylinder pressure predicted by the computer simulation model. The purpose is simply to verify the accuracy of the simulation programme, in relation to the real engine for the same operating parameters.

Fig 5.45 to Fig 5.48 provide plots of cylinder pressure versus crank angle for various conditions of operation for the bowl-in-piston combustion chamber and disc combustion chamber operating on both ethanol and isooctane. The operational parameters are indicated in the figures.

The experimental values, represented by the diamond symbols are plotted together with the predicted values and in general indicated a good agreement. The model has the tendency to predict a slightly lower value for the peak cylinder pressure. However, this is perfectly acceptable since the simulation is not fine tuned to the Ricardo E6 engine.

Furthermore, as explained in chapter 5, the estimation of the turbulence intensity levels during the combustion period gave reduced values compared to those which should be expected. As discussed earlier, this affects directly the turbulent burning velocity and consequently the rate at which mass is burned in the cylinder.

#### 5.6.2 Flame propagation velocity

The elapsed time between the deflections of the two laser beams, when disturbed by the moving flame front is called the flame travel time, FTT, and the processing of FTT data has been described in chapter 4.

The data averaged over 100 cycles define an ensemble mean flame travel time. The flame propagation velocity is then defined as the ratio of the distance between laser beams, to this mean flame travel time  $T$ ,

$$v = \Delta / T$$

The laser beams were positioned at a distance of 15 mm from the electrodes of the spark plug, therefore the velocity measured at that point is assumed to be that of a spherical flame front propagating from the spark plug and having a 15 mm radius at the moment the measurement is made.

Flame propagation velocity obtained from the computer simulation program should then be compared only at this point. The flame propagation velocity (or flame speed) obtained from the simulation programme must be understood as an apparent or effective flame speed because  $v$  is compared using a model which is dependent on a range of simplifying assumptions, for example, spherical flame propagation, turbulence decaying with time and turbulence intensity proportional to engine speed.

Some results obtained for the flame speed using the simulation programme in its original configuration, ie, using the Lancaster (1976) correlation are presented in Table 5.2. In that table, the experimental values of the flame speed are also indicated, corresponding to the conditions presented in chapter 4.

As can be seen, the data obtained from the simulation

|          | RF (mm) | D (mm) | $\bar{T}$ (m/s) | VF (m/s) | VF (m/s)    | RF (mm)     |
|----------|---------|--------|-----------------|----------|-------------|-------------|
| ID120159 | 15.0    | 4.0    | 0.420           | 9.52     | 12,2 / 16,0 | 10.6 / 18.8 |
| ID120209 | 15.0    | 4.0    | 0.449           | 9.09     | 13.9 / 18.4 | 11.1 / 20.5 |
| ED115109 | 15.0    | 4.0    | 0.556           | 7.19     | 8.4 / 11.3  | 10.4 / 18.6 |
| ED120209 | 15.0    | 4.0    | 0.392           | 10.20    | 13.9 / 18.4 | 11.1 / 20.5 |
| EB110109 | 15.0    | 3.0    | 0.462           | 6.49     | 6.2 / 8.1   | 9.9 / 17.3  |
| EB115109 | 15.0    | 3.0    | 0.397           | 7.56     | 8.4 / 10.8  | 9.4 / 15.6  |
| EB115100 | 15.0    | 3.0    | 0.282           | 10.64    | 8.9 / 11.8  | 9.6 / 16.9  |
| EB120209 | 15.0    | 3.0    | 0.284           | 10.56    | 13.9 / 18.4 | 11.1 / 17.6 |
| EB210109 | 15.0    | 3.0    | 0.414           | 7.25     | 5.9 / 7.7   | 9.9 / 16.4  |
| EB215109 | 15.0    | 3.0    | 0.371           | 8.09     | 7.9 / 10.3  | 9.4 15.6    |
| EB215100 | 15.0    | 3.0    | 0.248           | 12.10    | 8.5 / 11,1  | 9.7 / 15.9  |
| EB210109 | 15.0    | 3.0    | 0.266           | 11.28    | 9,2 / 12,1  | 9.5 / 15.8  |
| EB210158 | 15.0    | 3.0    | 0.403           | 7.35     | 6.1 / 7.8   | 9.9 / 16.3  |
| EB210159 | 15.0    | 3.0    | 0.345           | 8.70     | 6.8 / 8.8   | 11.4 / 18.9 |
| EB210150 | 15.0    | 3.0    | 0.278           | 10.79    | 7.4 / 9.6   | 11.7 / 19.5 |
| EB215150 | 15.0    | 3.0    | 0.222           | 13.51    | 9.8 12.5    | 10.1 / 17.6 |
| EB215200 | 15.0    | 3.0    | 0.240           | 12.50    | 11.2 / 14.8 | 11.6 / 19.2 |

TABLE 5.2



programme are in general in good agreement with the data obtained experimentally in the Ricardo E6. It is important to bear in mind that the programme is whole predictive on the basis of the model elements and that these have not been tuned to the Ricardo E6. Fig 5.49 is a plot of the flame propagation velocity determined experimentally versus the flame propagation velocity predicted by the programme.

In Table 5.2 the column identified by Rf indicates the minimum/maximum values of the flame radius as returned by the simulation programme. In column identified by Vf are indicated the minimum/maximum values of the flame propagation velocity, also calculated by the programme, corresponding to the flame front position given by Rf.

The values of  $Vf_{sim}$  in Fig 5.49 were obtained by interpolating on data from Table 5.2.



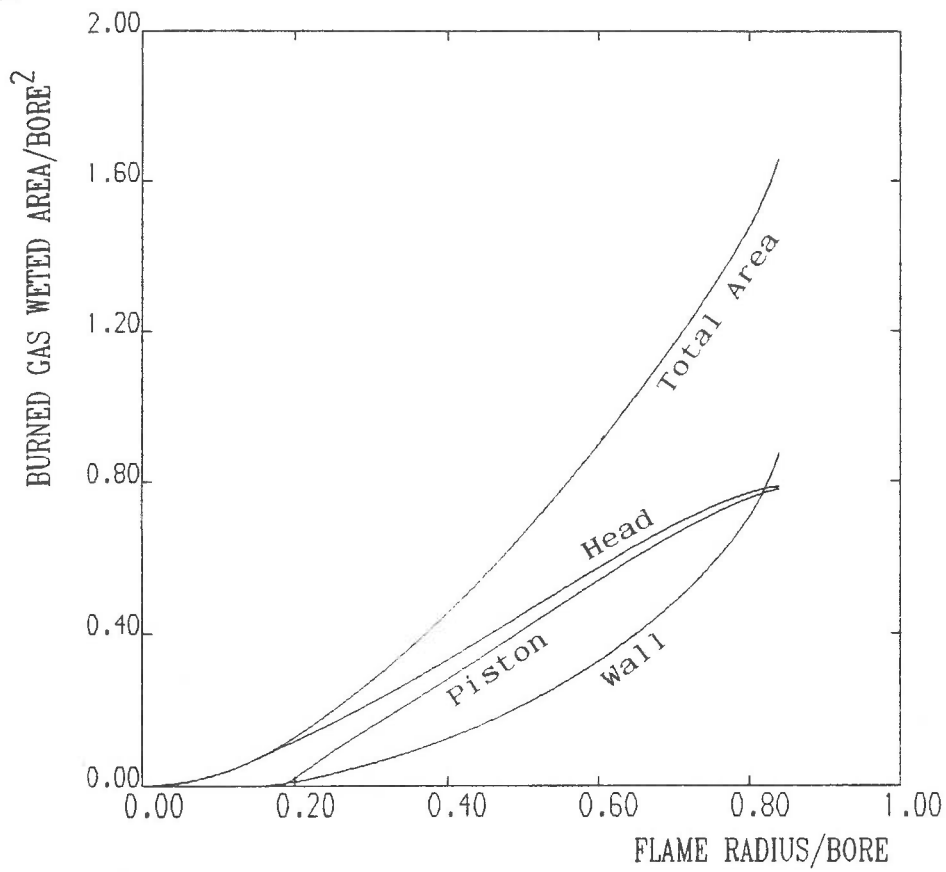
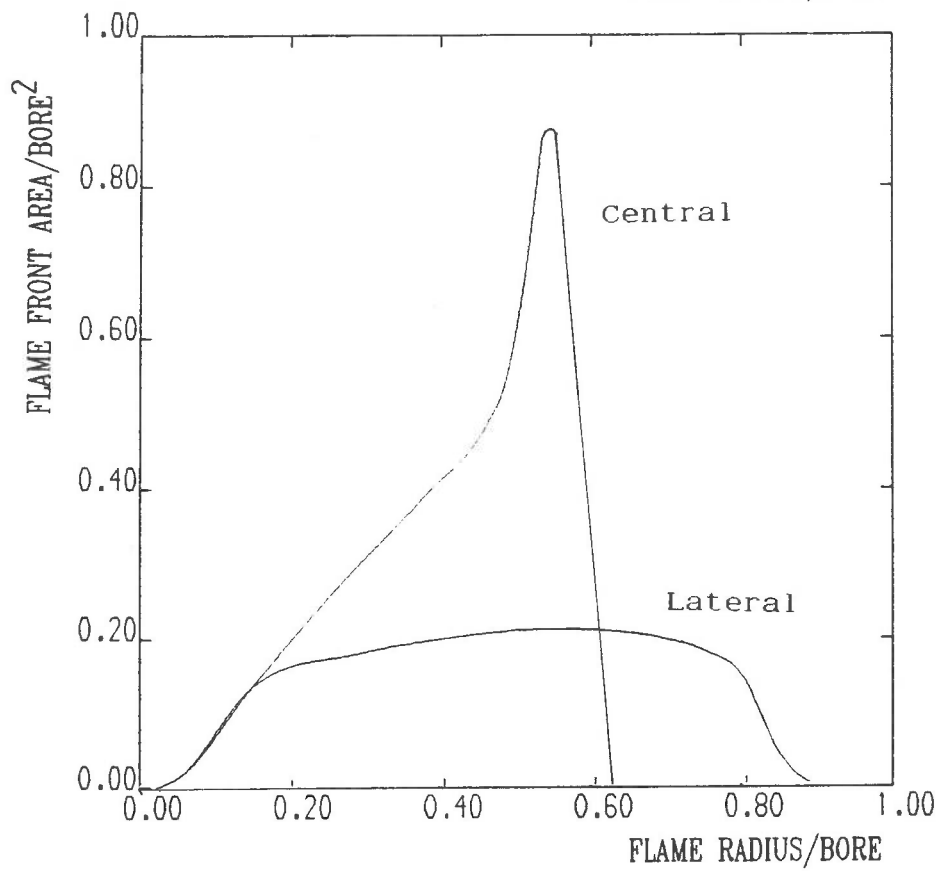


Fig. 5.1 Flame front wetted areas

Fig. 5.2 Flame front areas

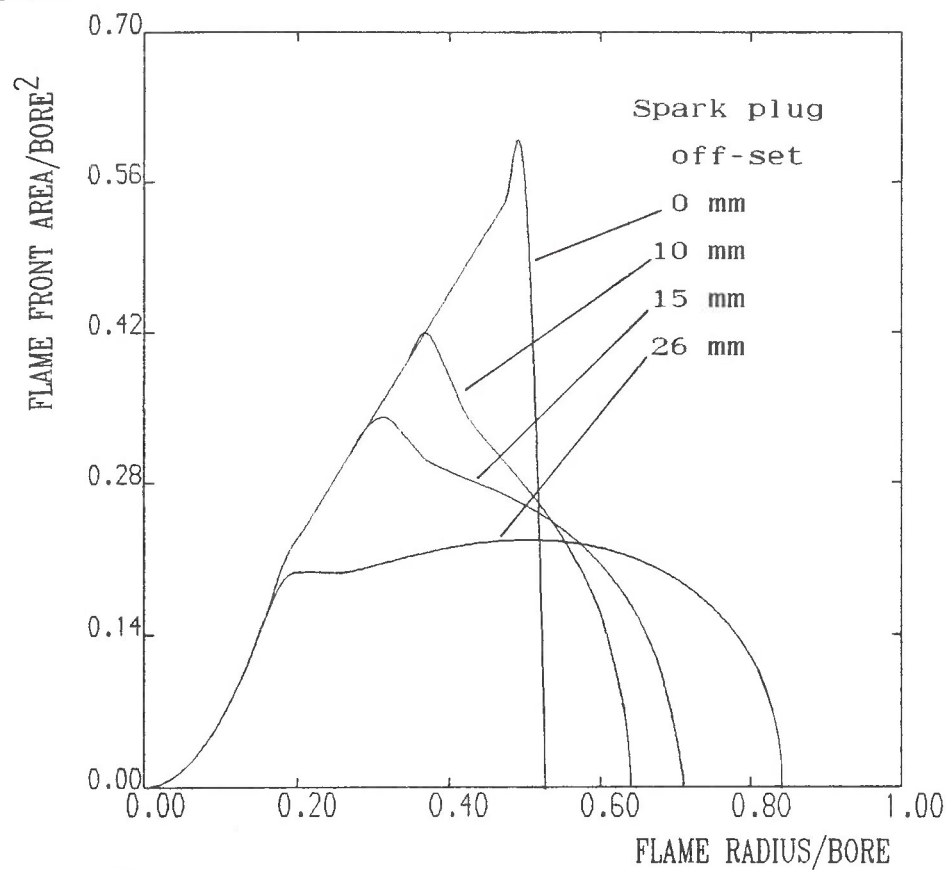
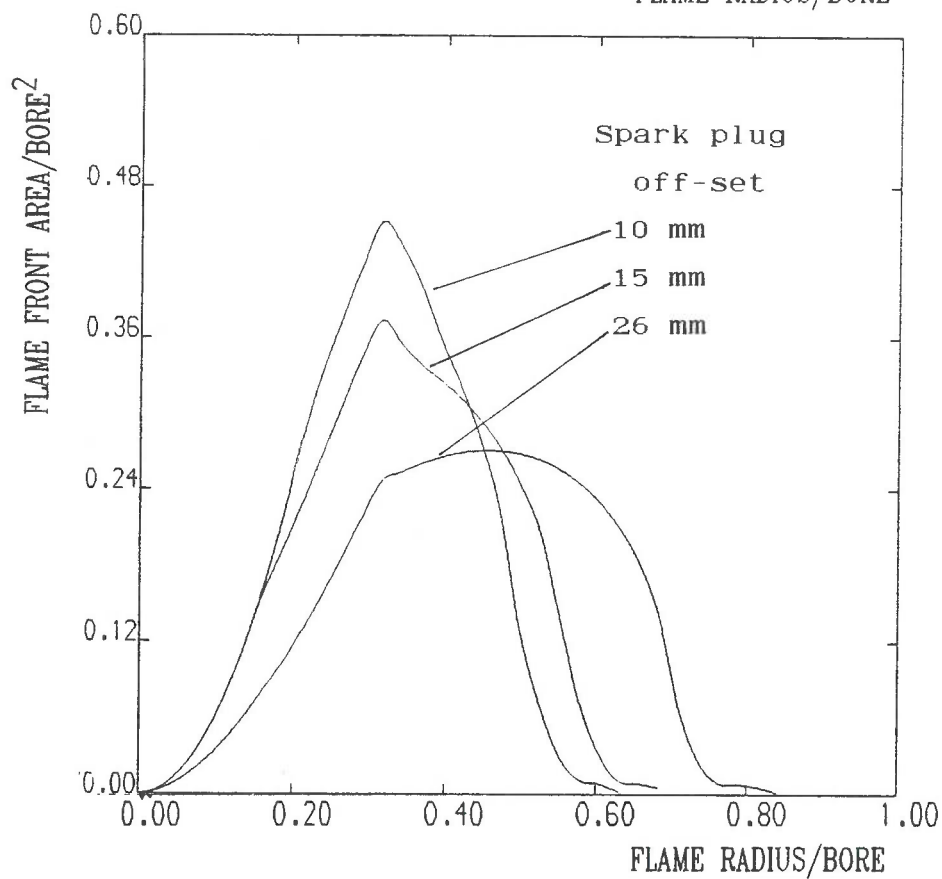


Fig. 5.3 Influence of spark plug position in a disc chamber

Fig. 5.4 Influence of spark plug position in a BIP chamber

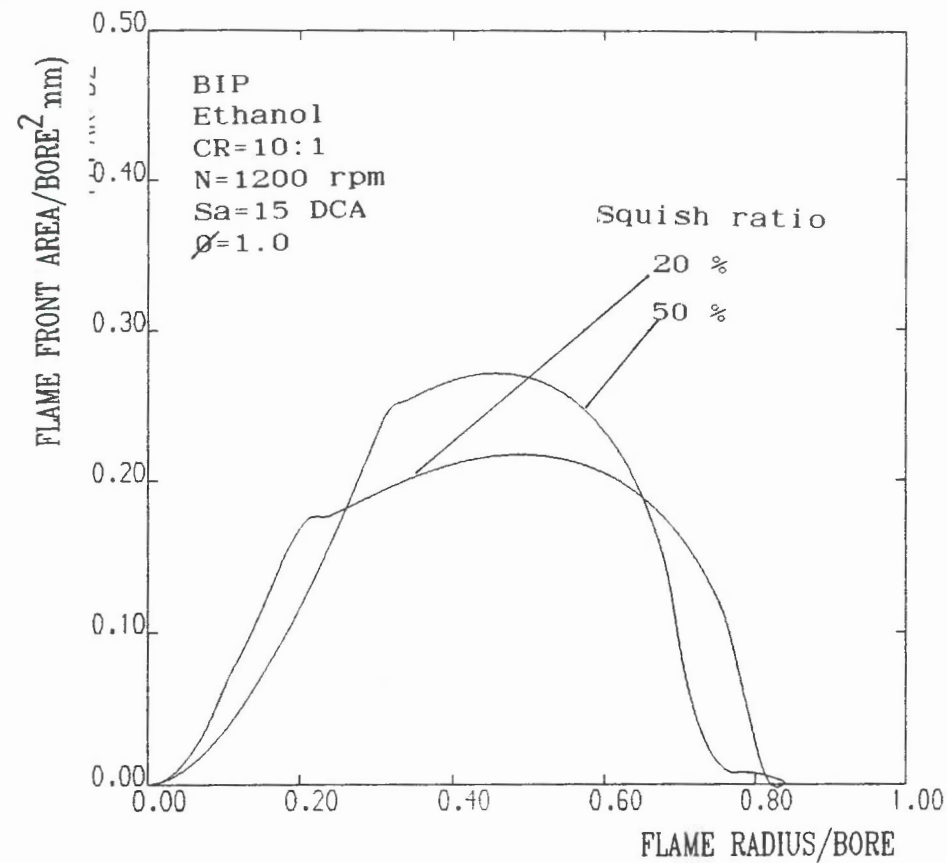
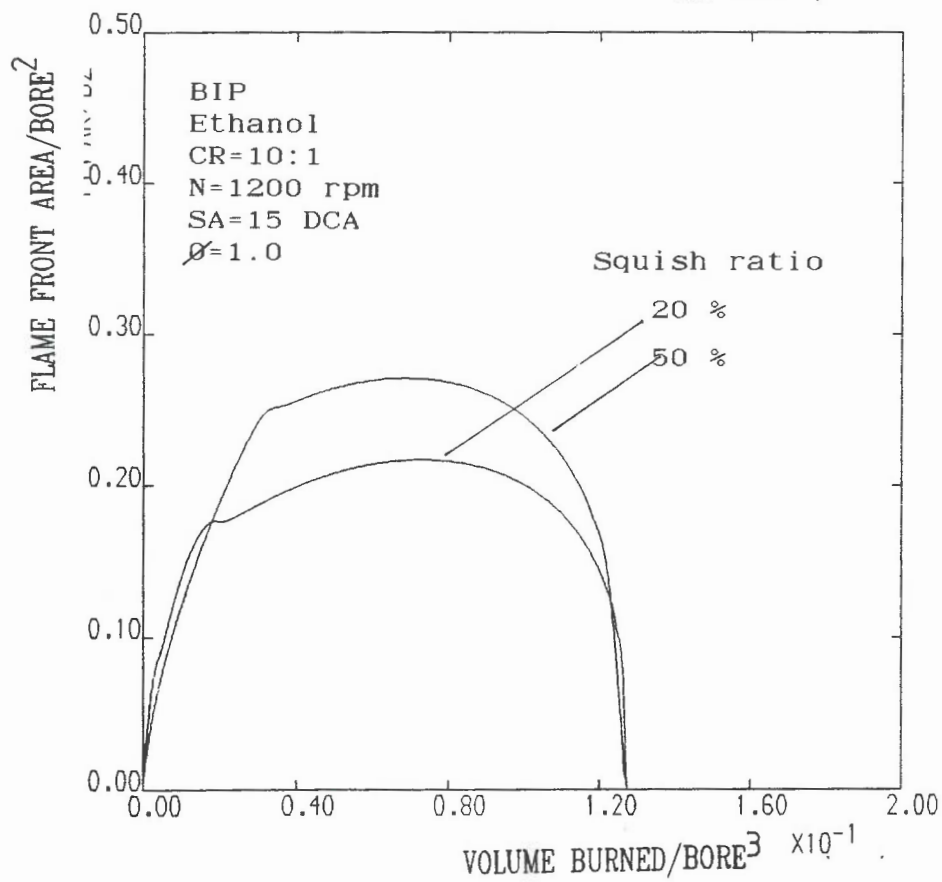


Fig. 5.5 and Fig. 5.6 Influence of squish ratio on flame front area versus flame radius and burned volume

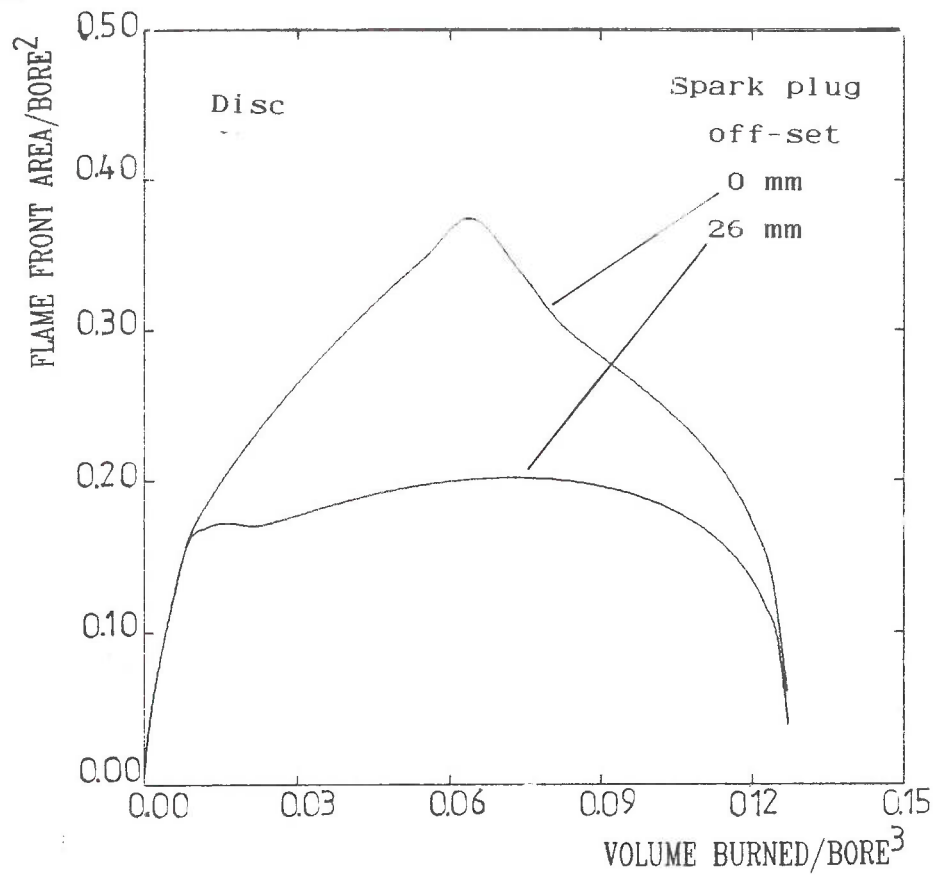
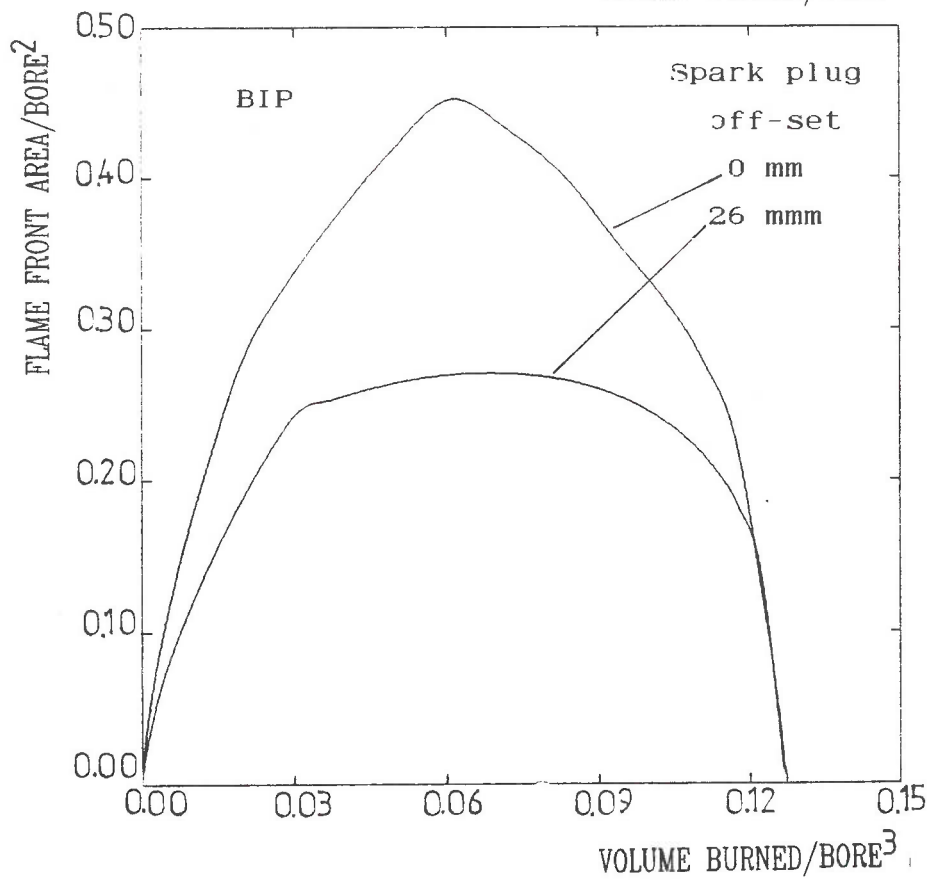


Fig. 5.7 and Fig. 5.8 Influence of the spark plug off-set on the flame front area versus enflamed volume.

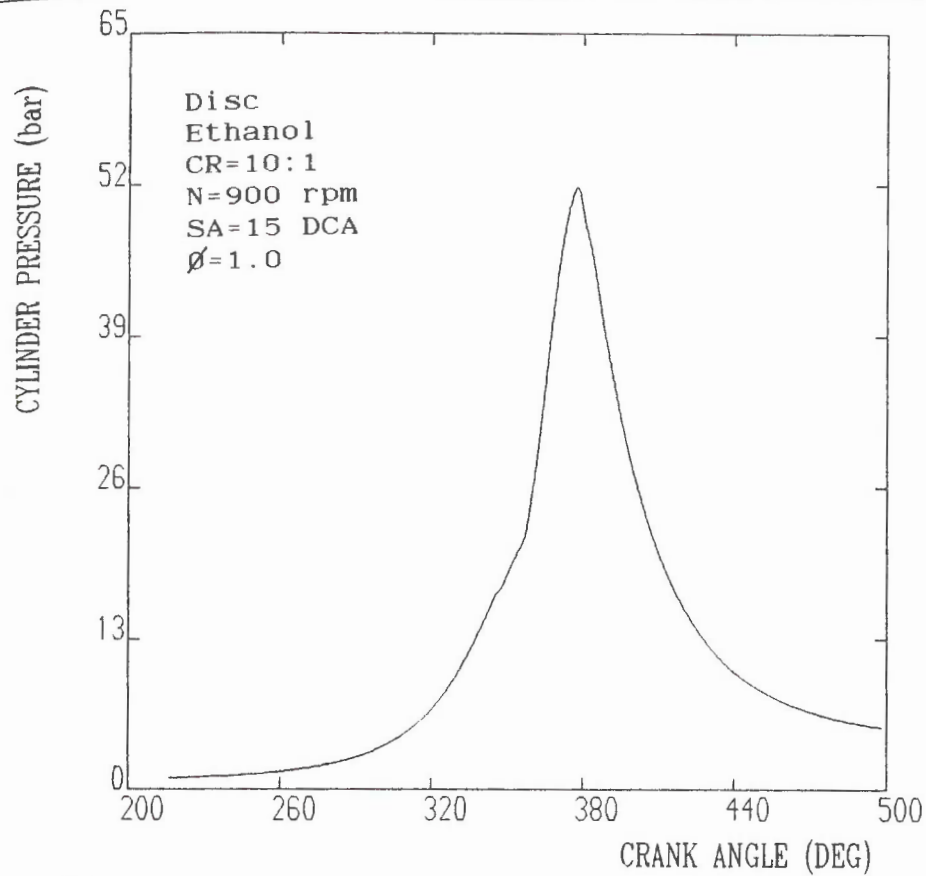
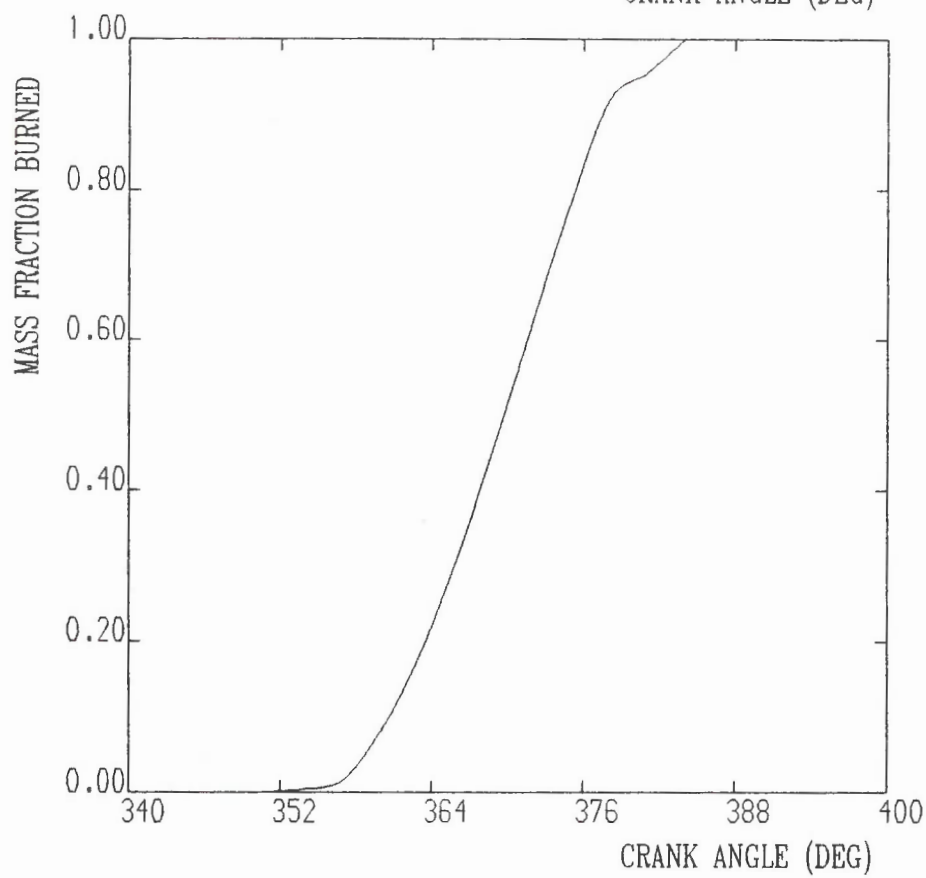


Fig. 5.9 Typical pressure versus time curve

Fig. 5.10 Typical mass fraction burned curve

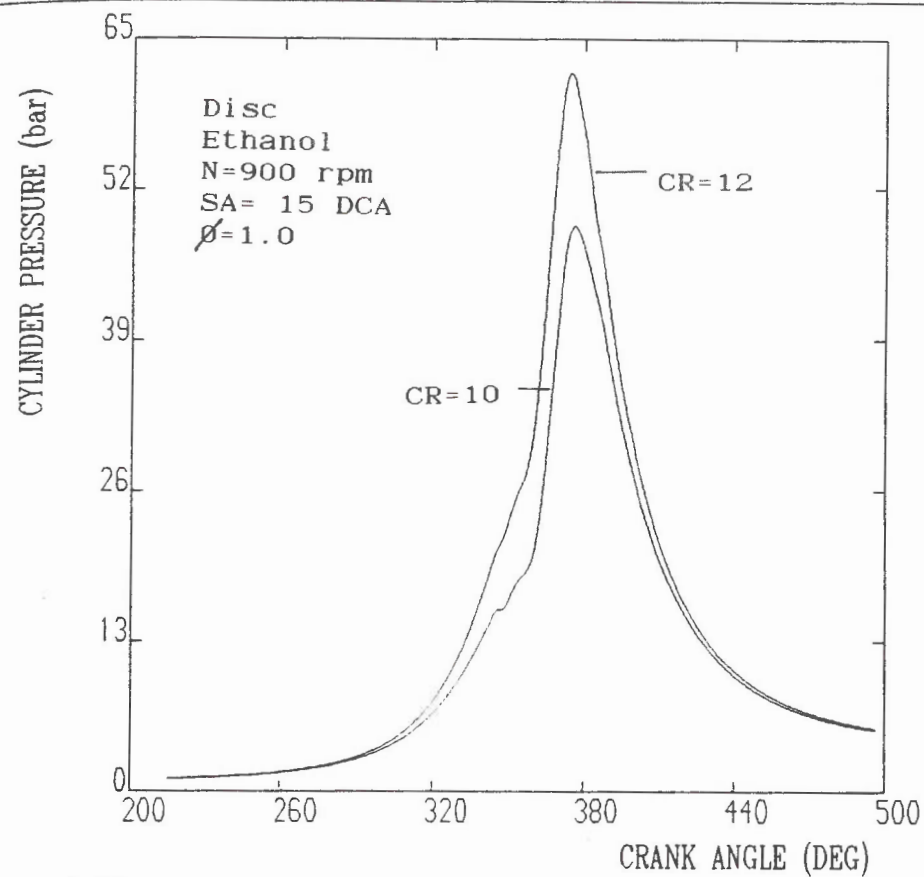
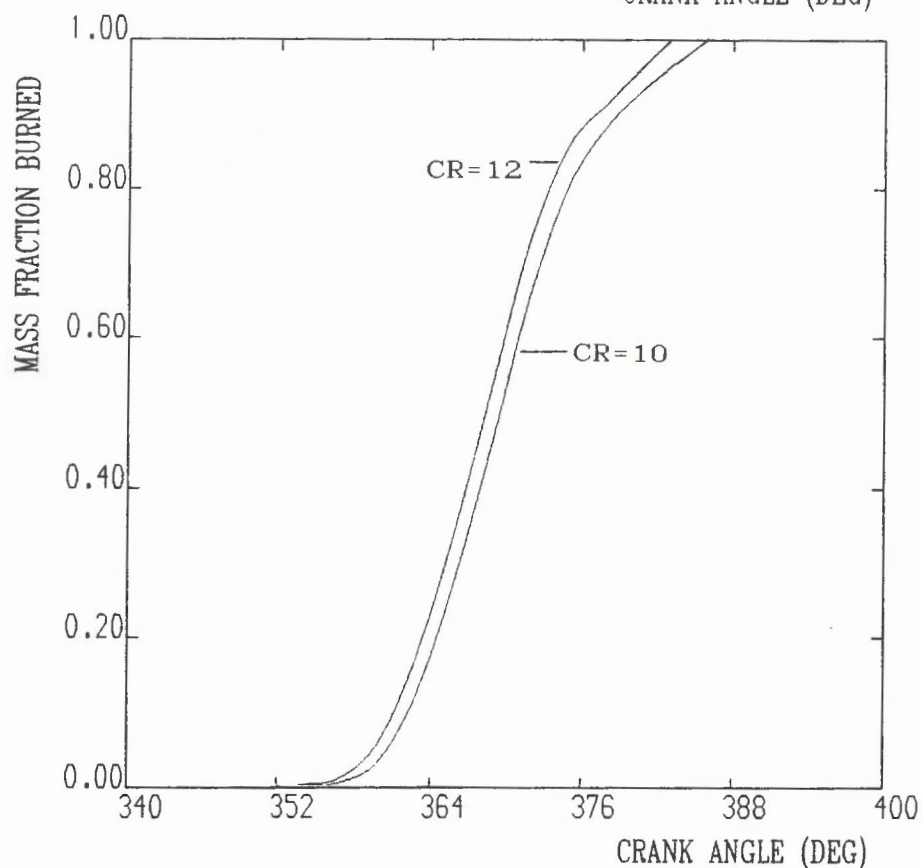


Fig. 5.11 Influence of CR on pressure versus time curve

Fig. 5.12 Influence of CR on mass fraction burned curve



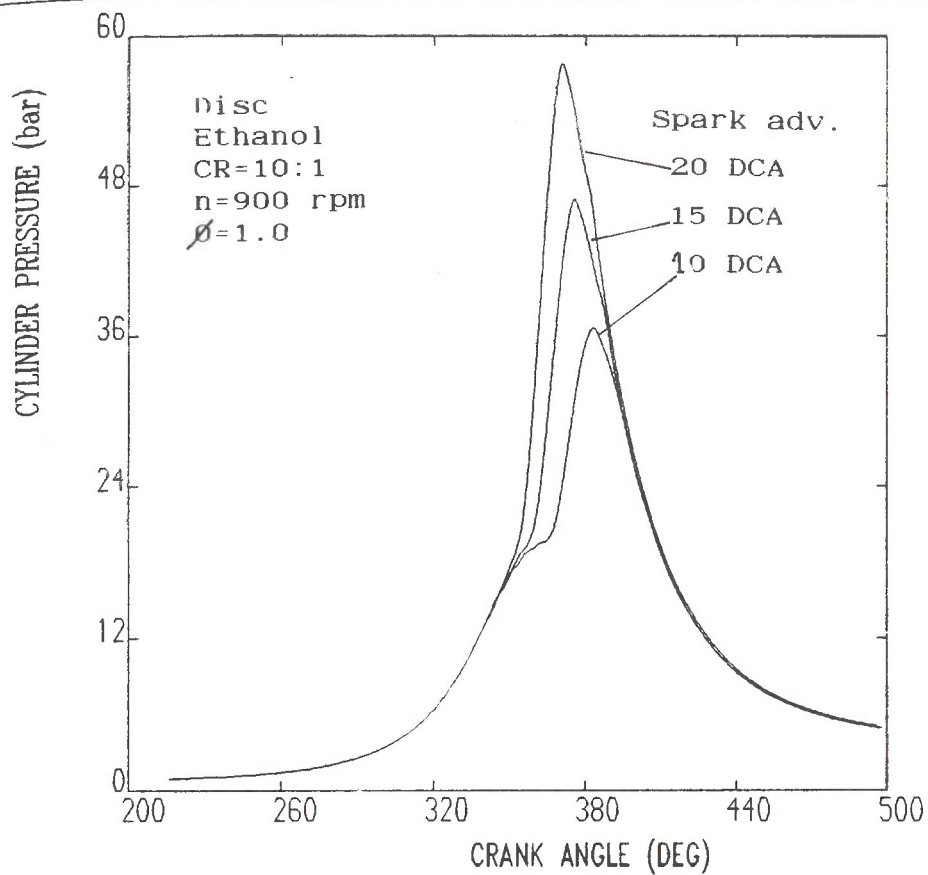
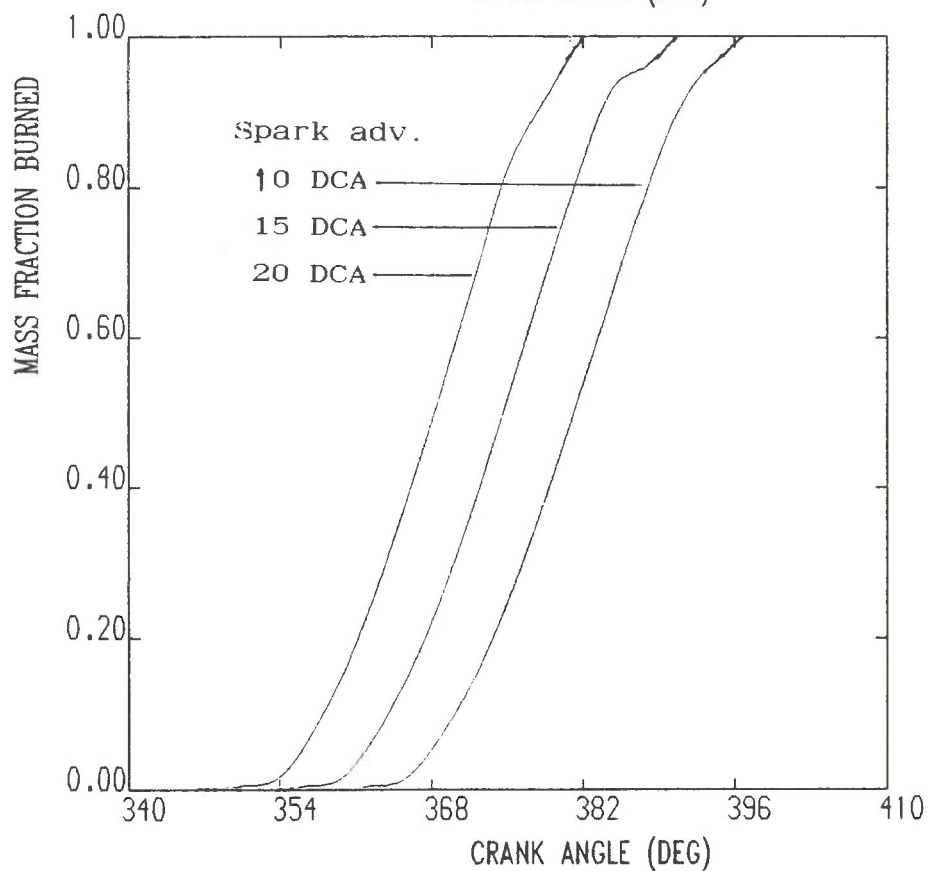


Fig. 5.13 Effect of spark advance on pressure curves

Fig. 5.14 Effect of spark advance on mass fraction burned

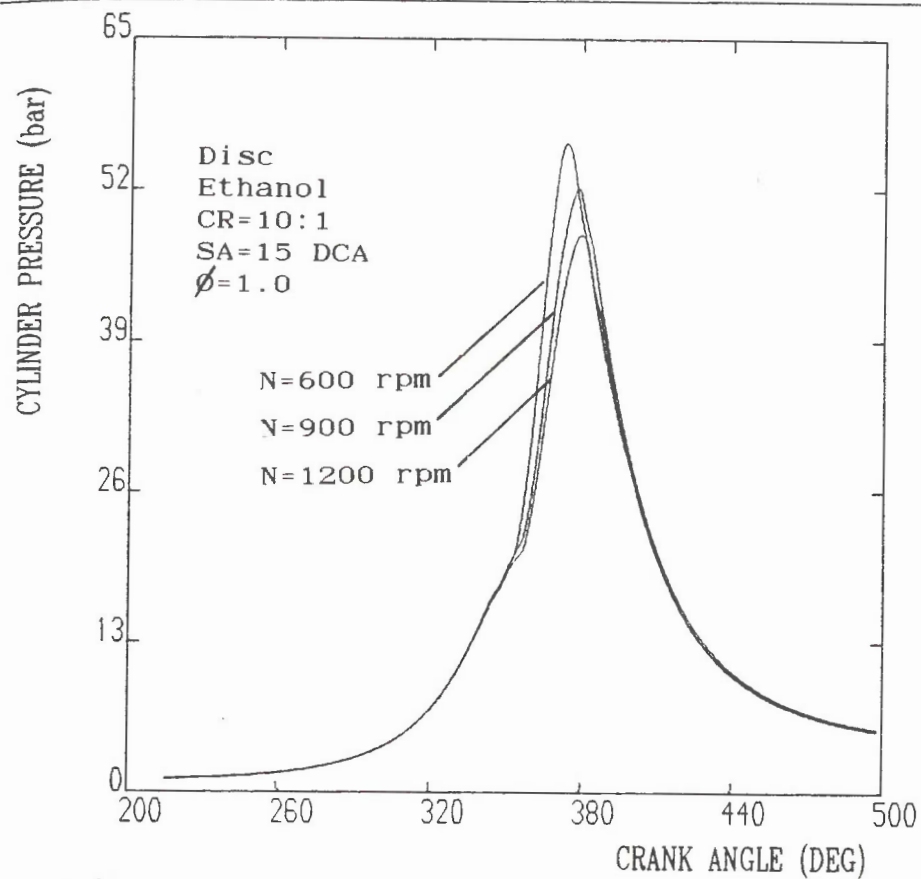
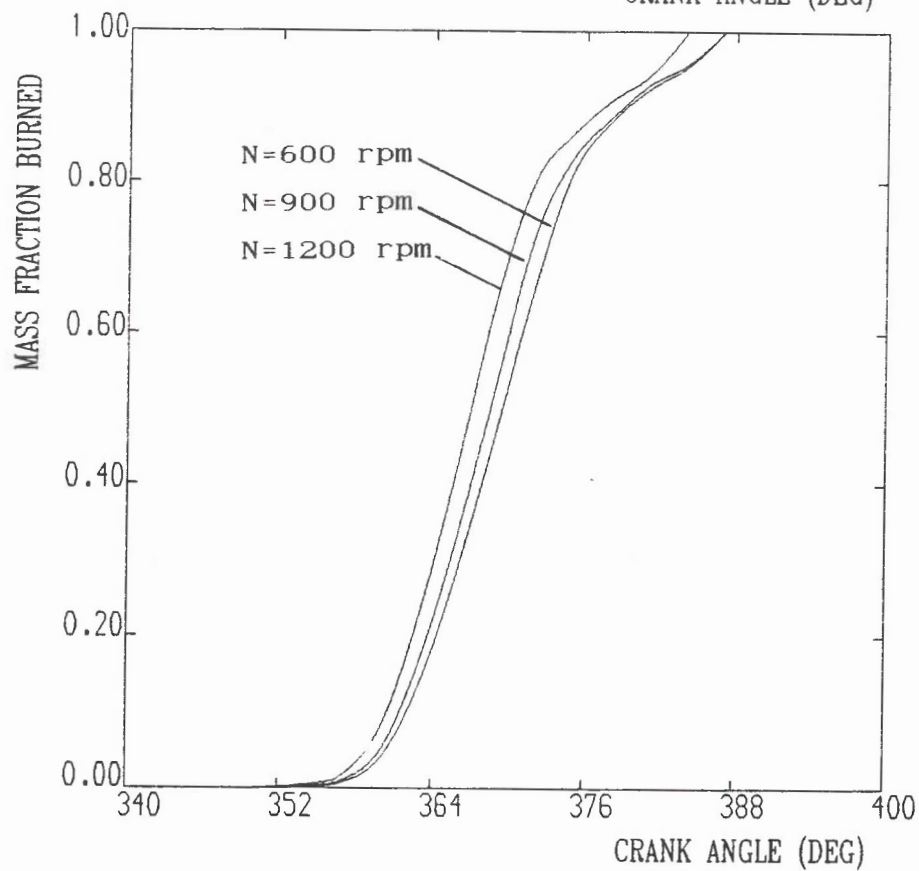


Fig. 5.15 Effect of engine speed on Pressure curves

Fig. 5.16 Effect of engine speed on mass fraction burned

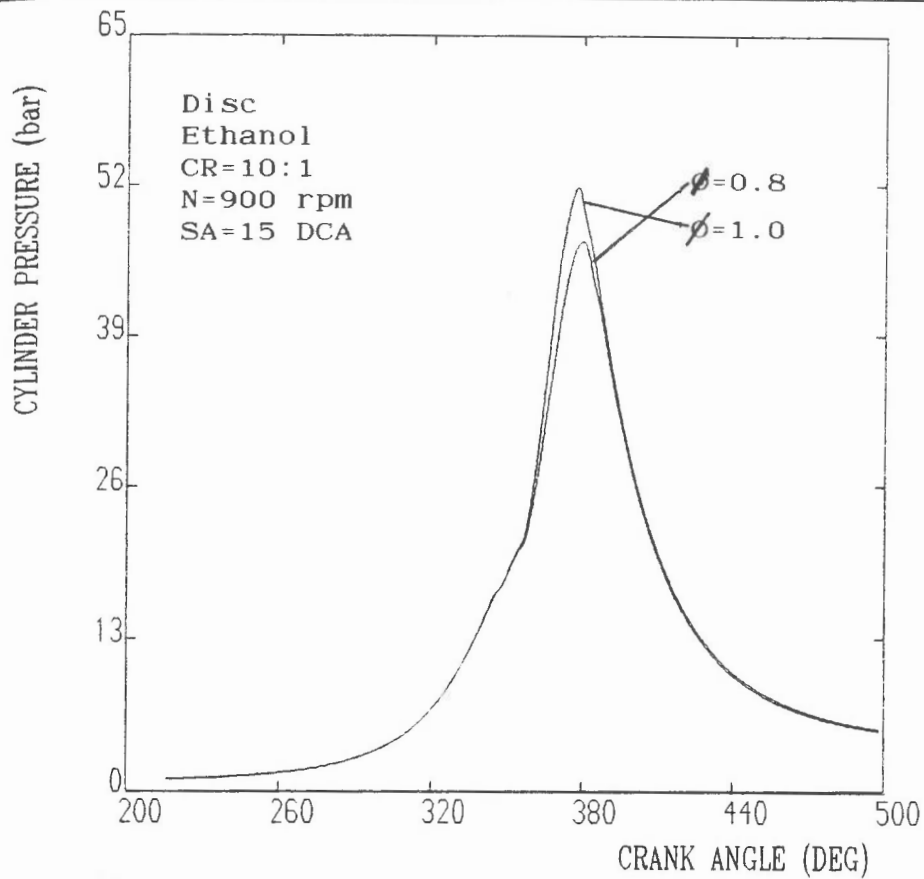
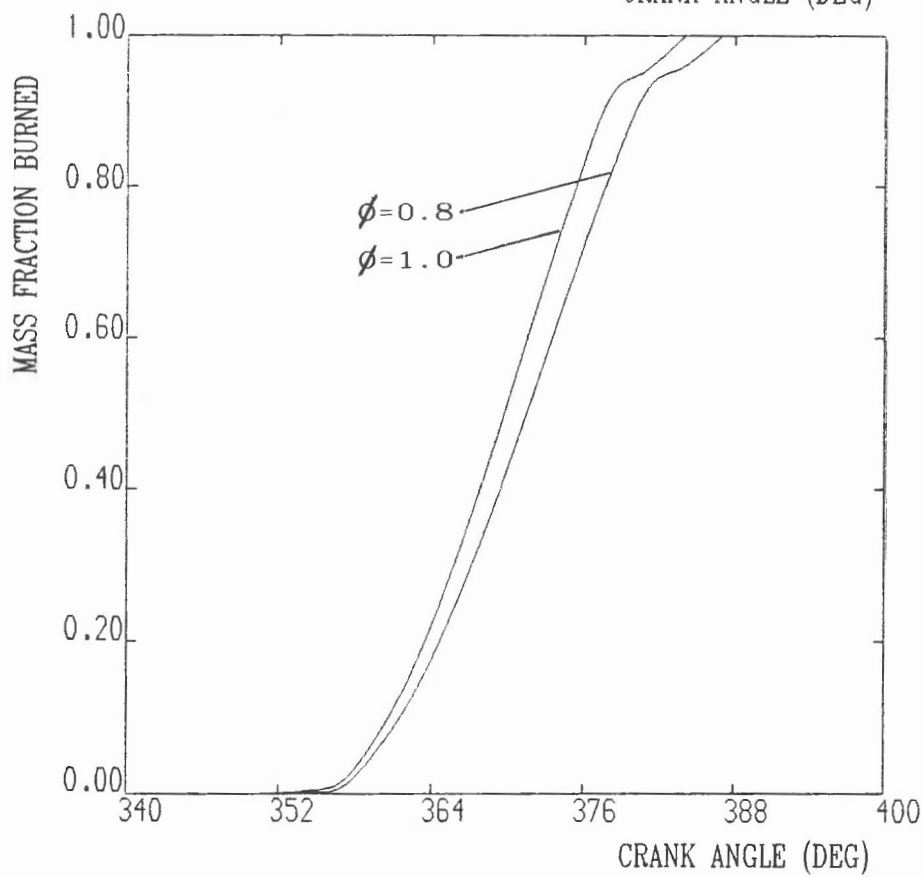


Fig. 5.17 Effect of equivalence ratio on pressure curves

Fig. 5.18 Effect of equivalence ratio on mass fraction burned

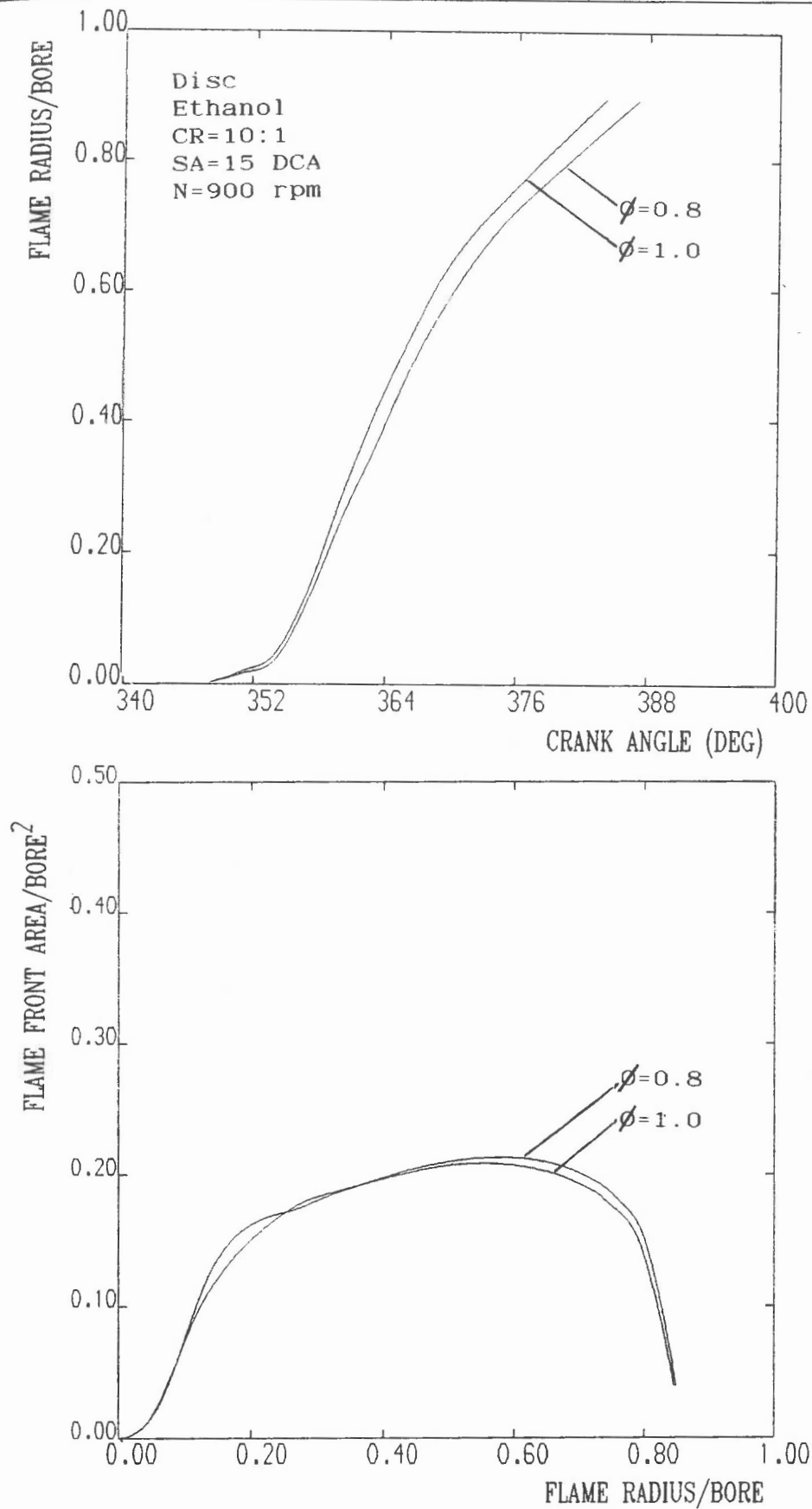


Fig. 5.19 Effect of equivalence ratio on duration of combustion

Fig. 5.20 Effect of equivalence ratio on flame front area

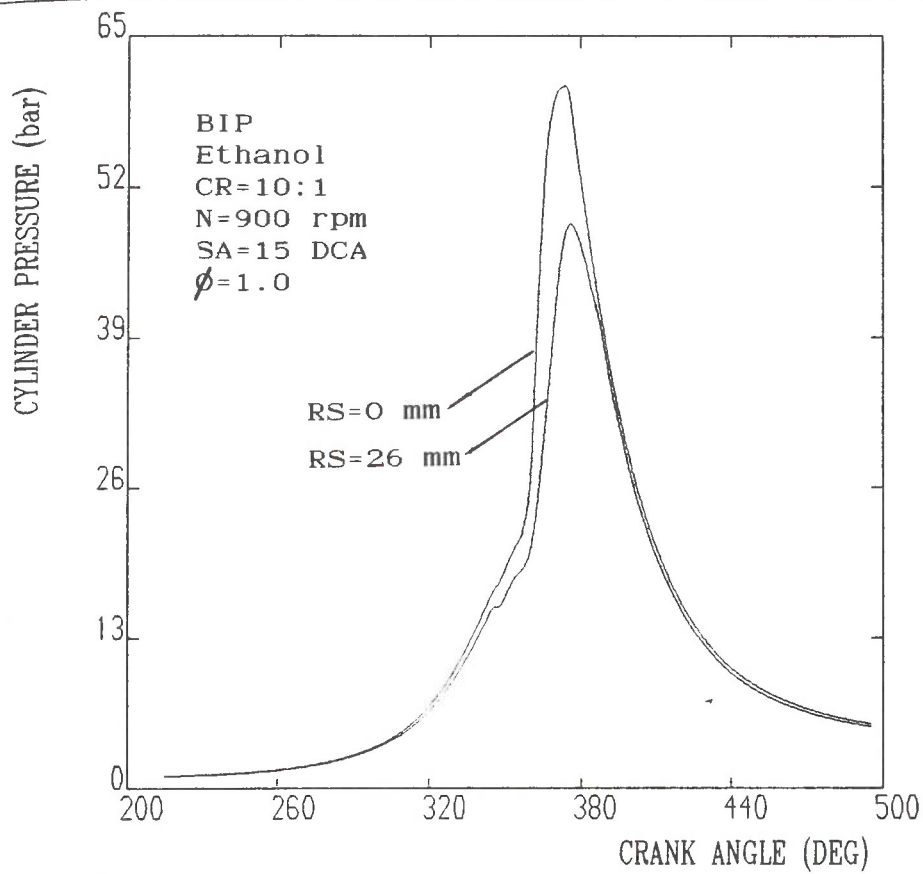
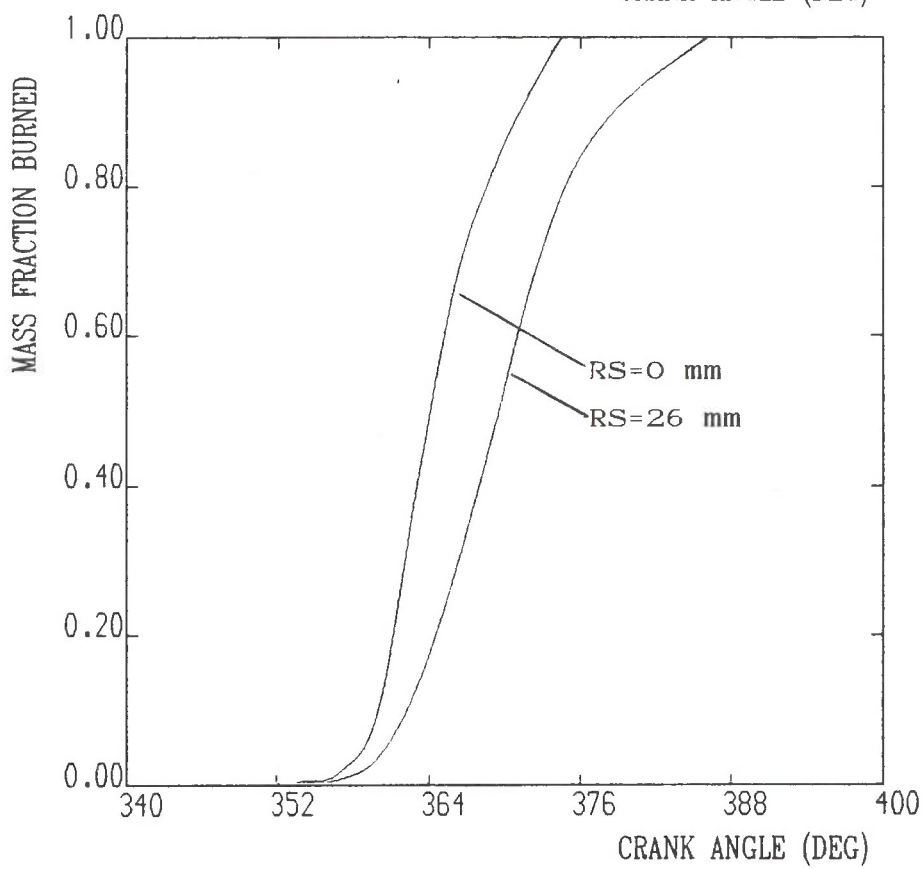


Fig. 5.21 Effect of spark plug position on pressure curves

Fig. 5.22 Effect of spark plug position on mass fraction burned

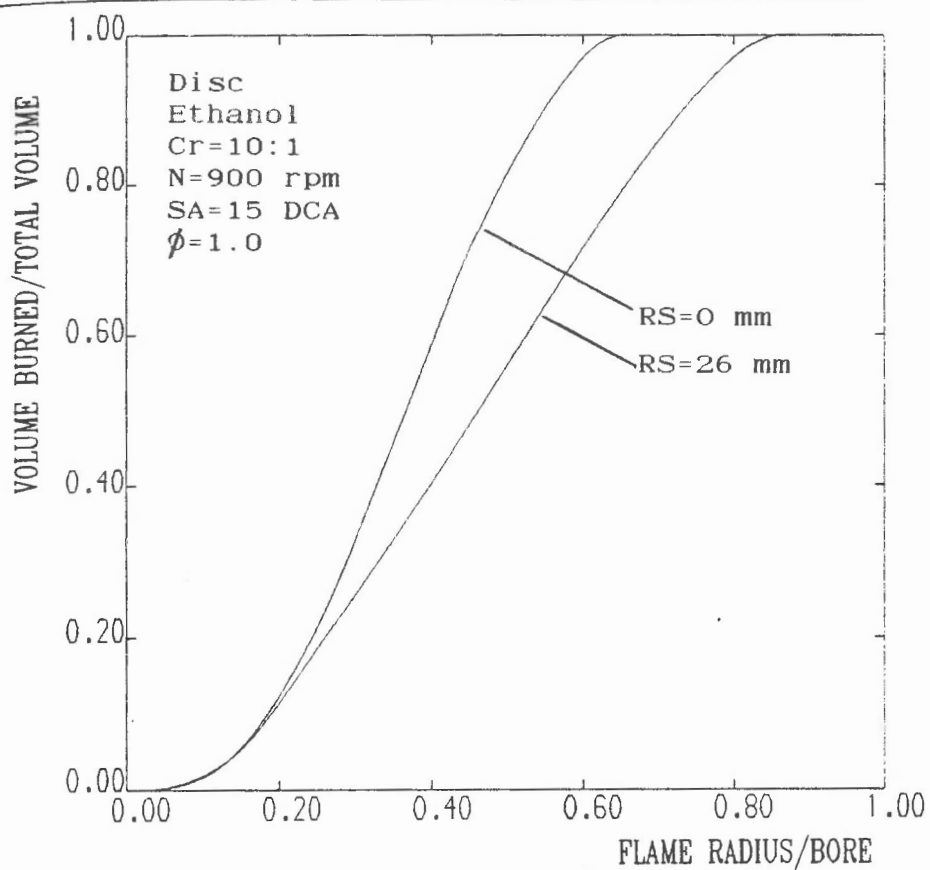
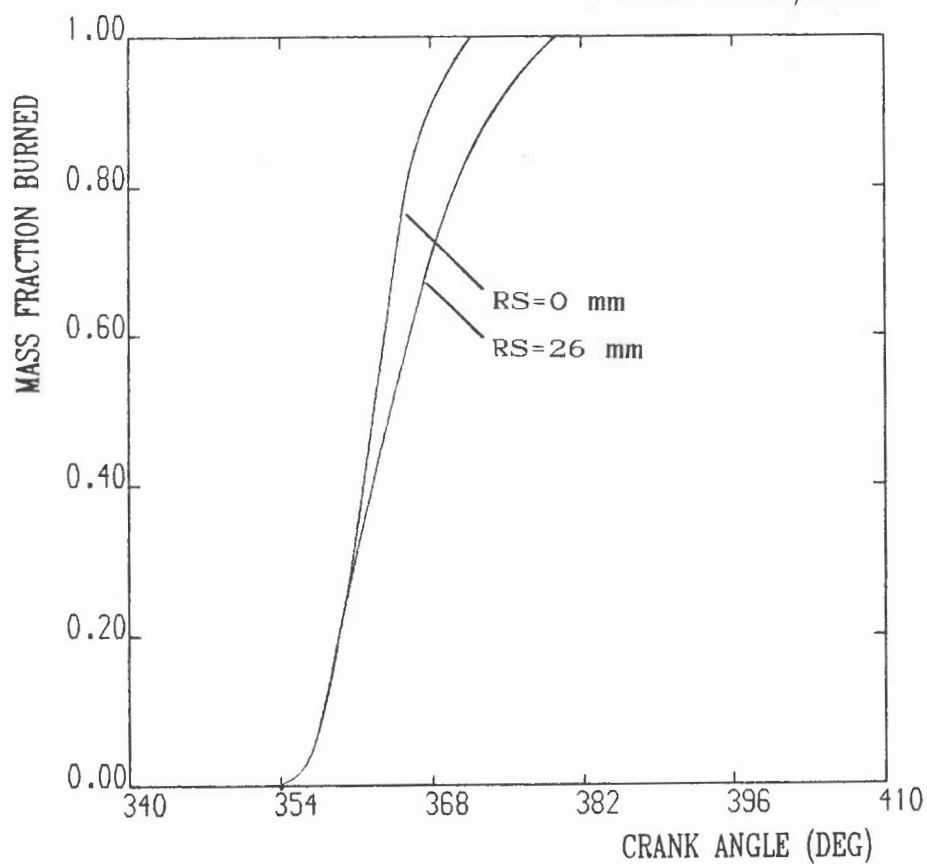


Fig. 5.23 Spark plug position and volume fraction burned

Fig. 5.24 Spark plug position and mass fraction burned



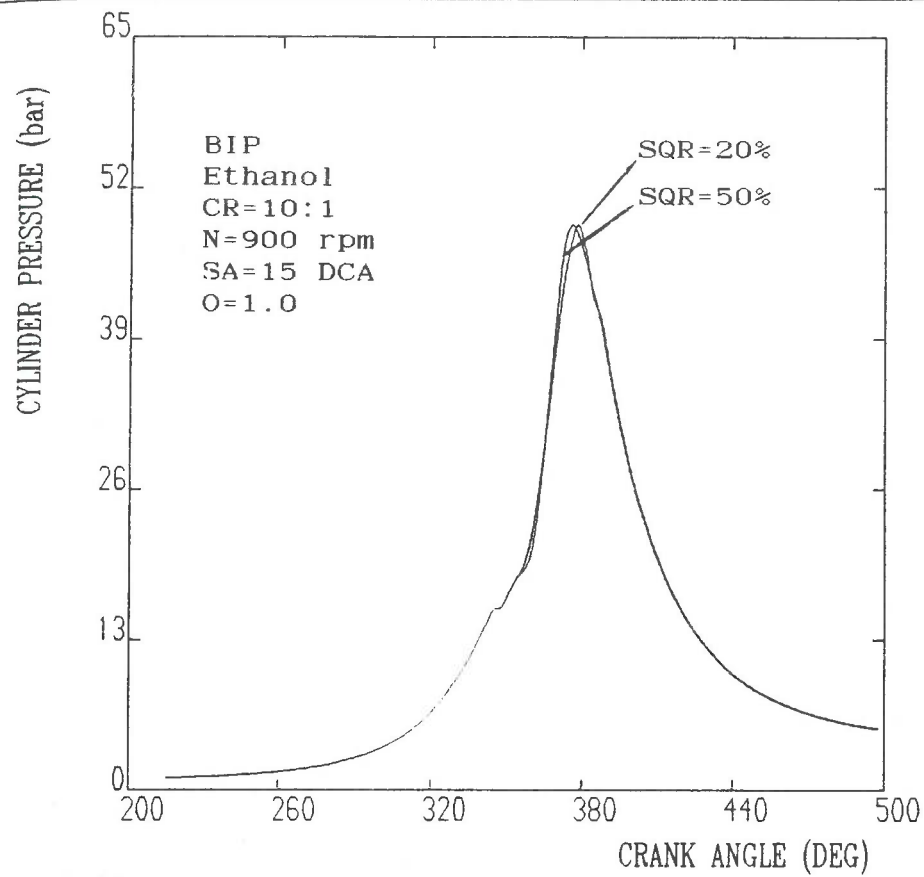
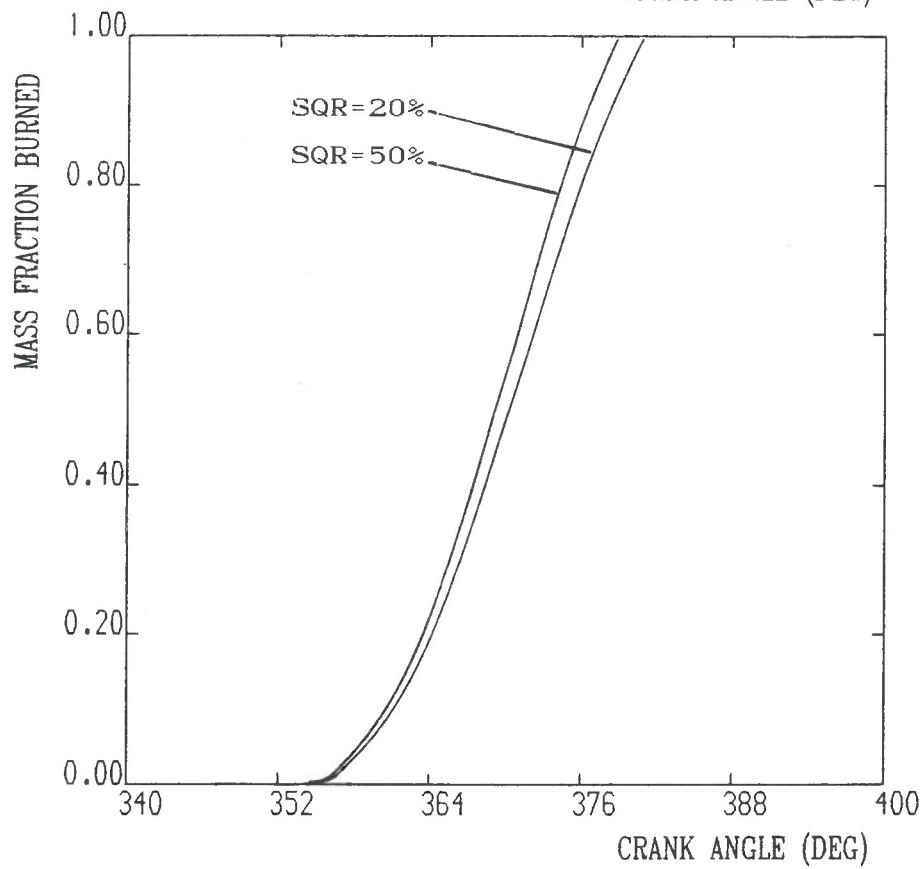


Fig. 5.25 Effect of squish ratio on pressure curves

Fig. 5.26 Effect of squish ratio on mass fraction burned

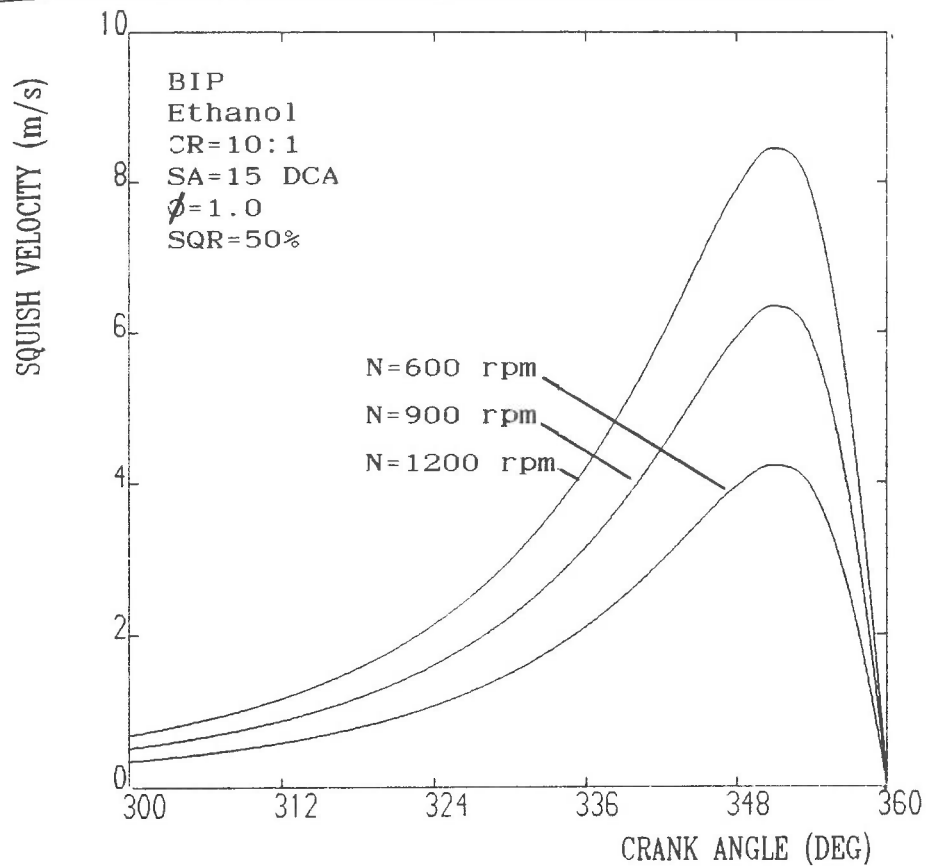
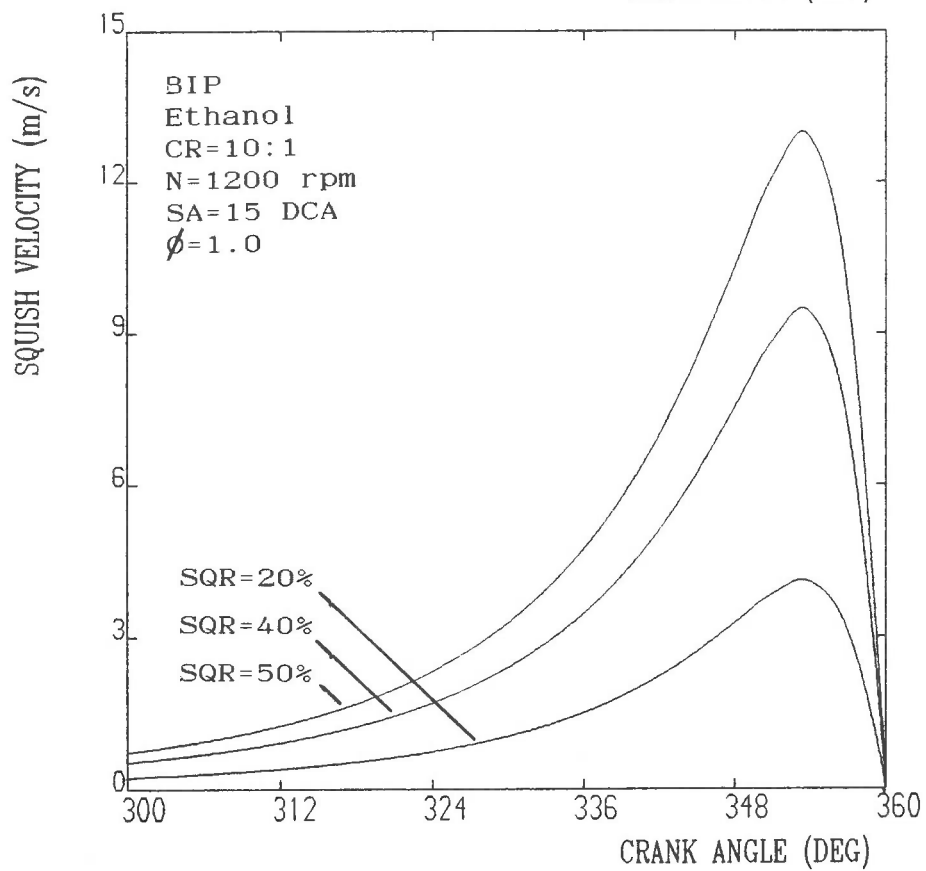


Fig. 5.27 Effect of engine speed on squish velocity

Fig. 5.28 Effect of squish ratio on squish velocity

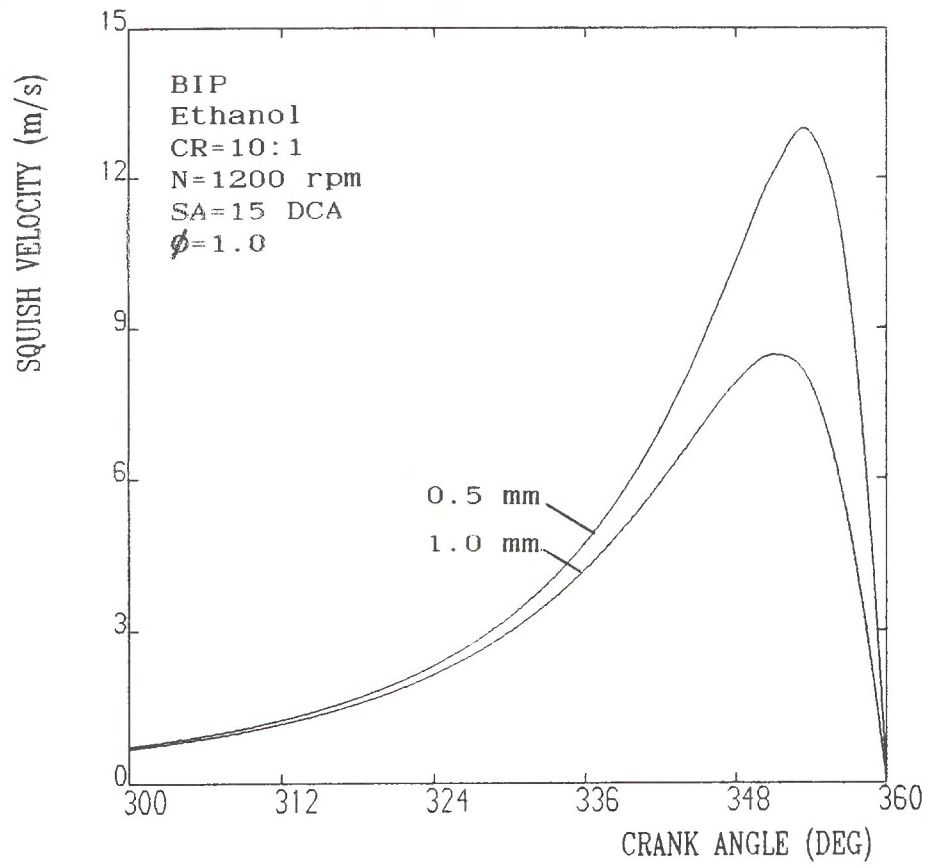


Fig. 5.29 Effect of bumping clearance on squish velocity

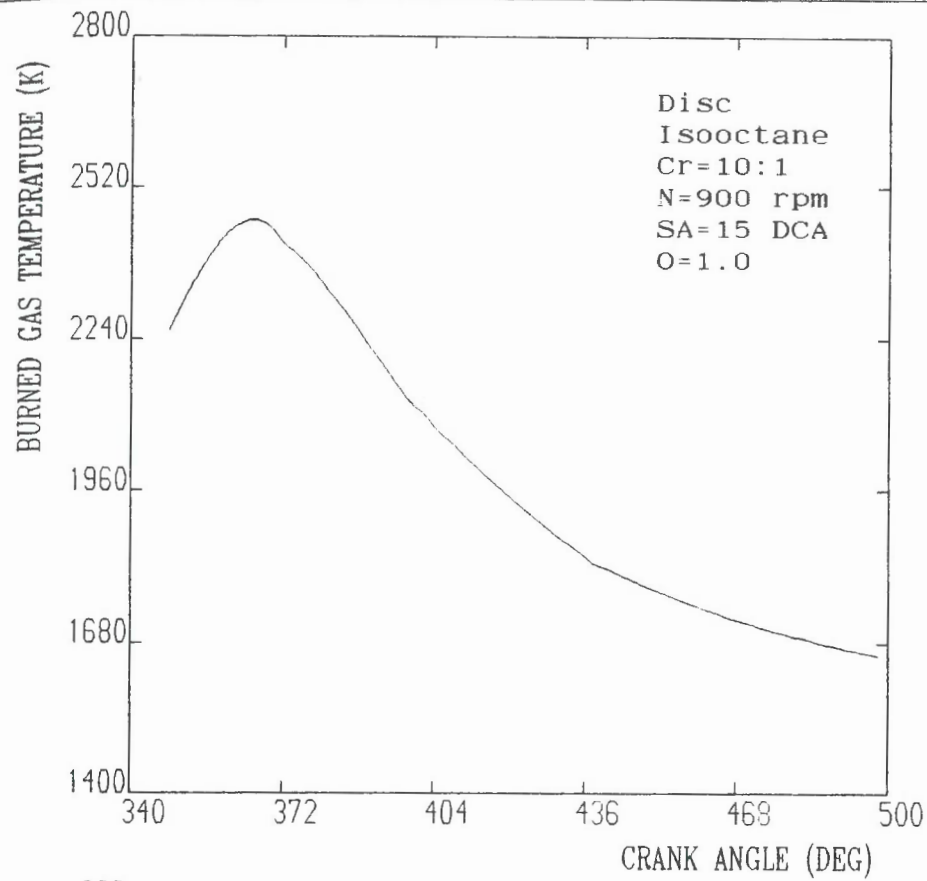
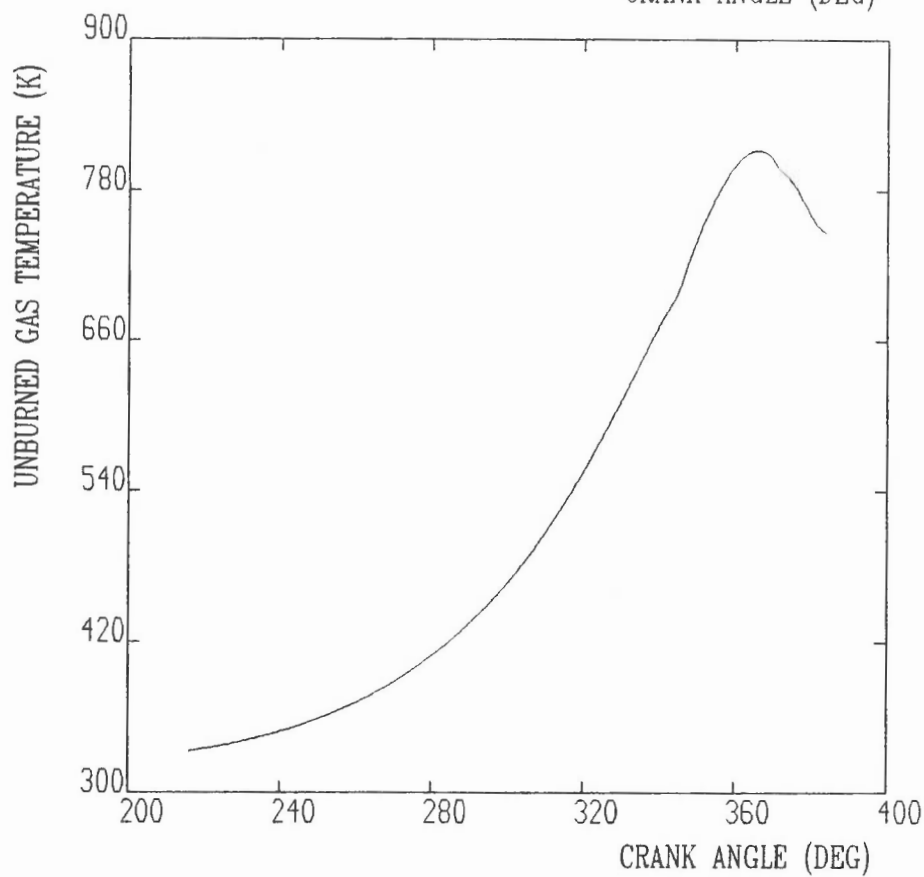


Fig. 5.30 Burned gas temperature versus crank angle

Fig. 5.31 Unburned gas temperature versus crank angle

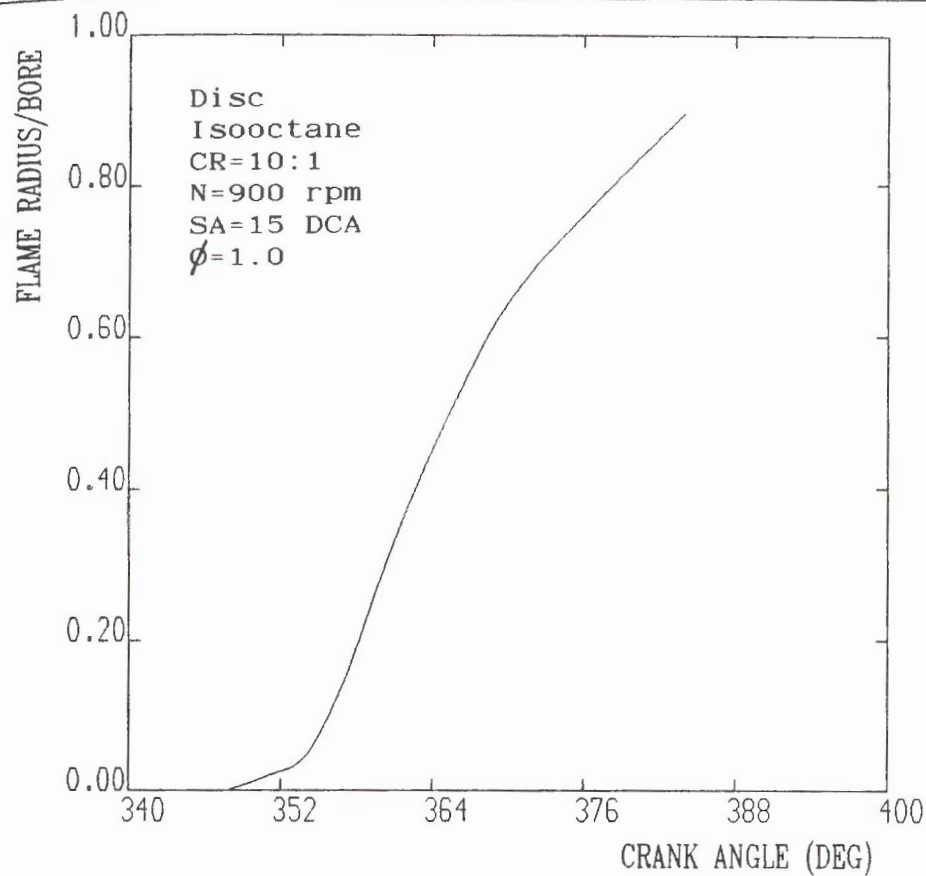
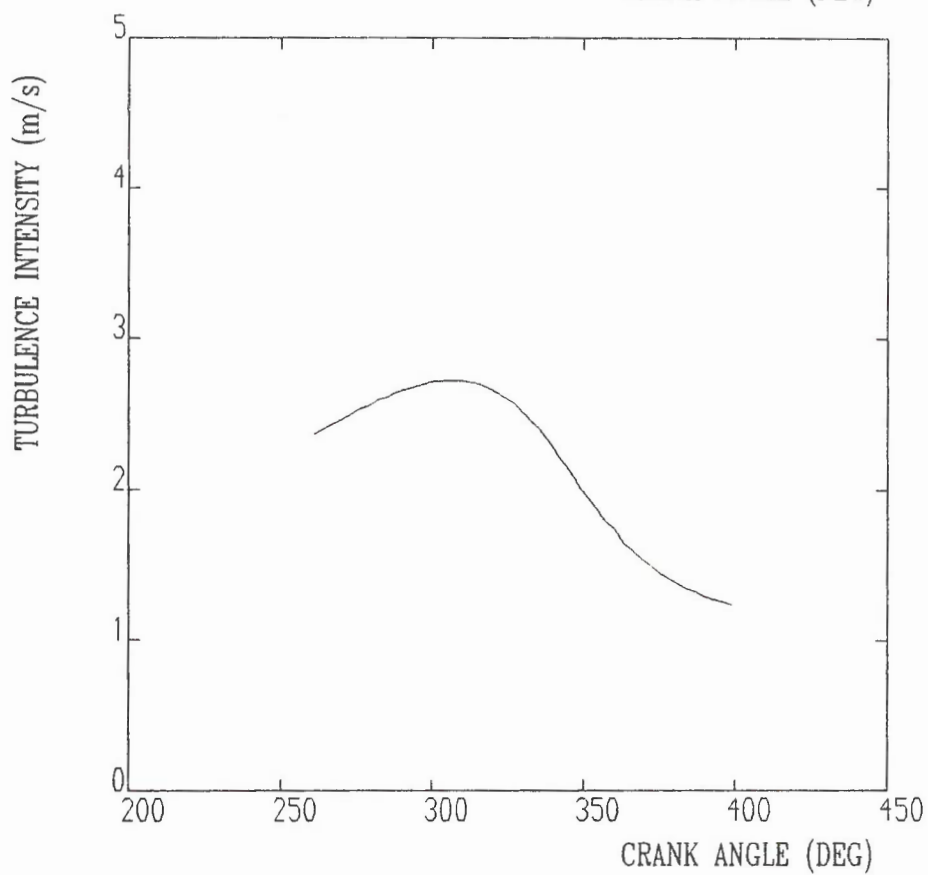


Fig. 5.32 Variation of flame radius with crank angle

Fig. 5.33 Turbulence intensity versus crank angle

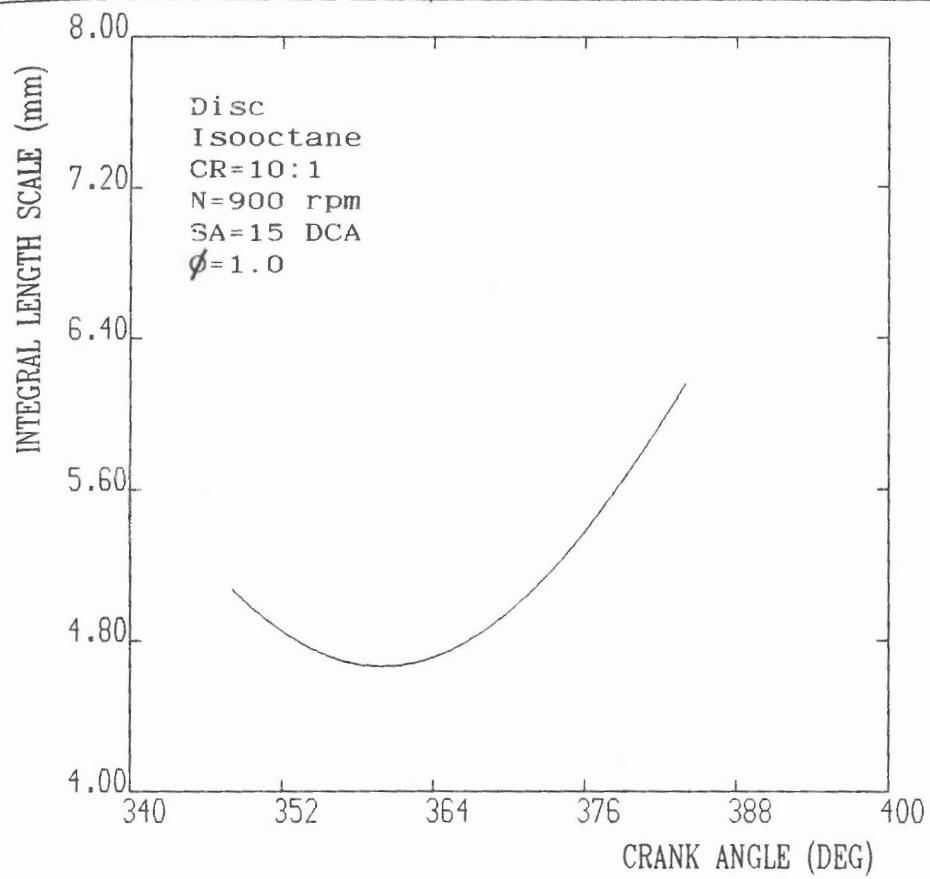


Fig. 5.34 Variation of integral length scale with crank angle



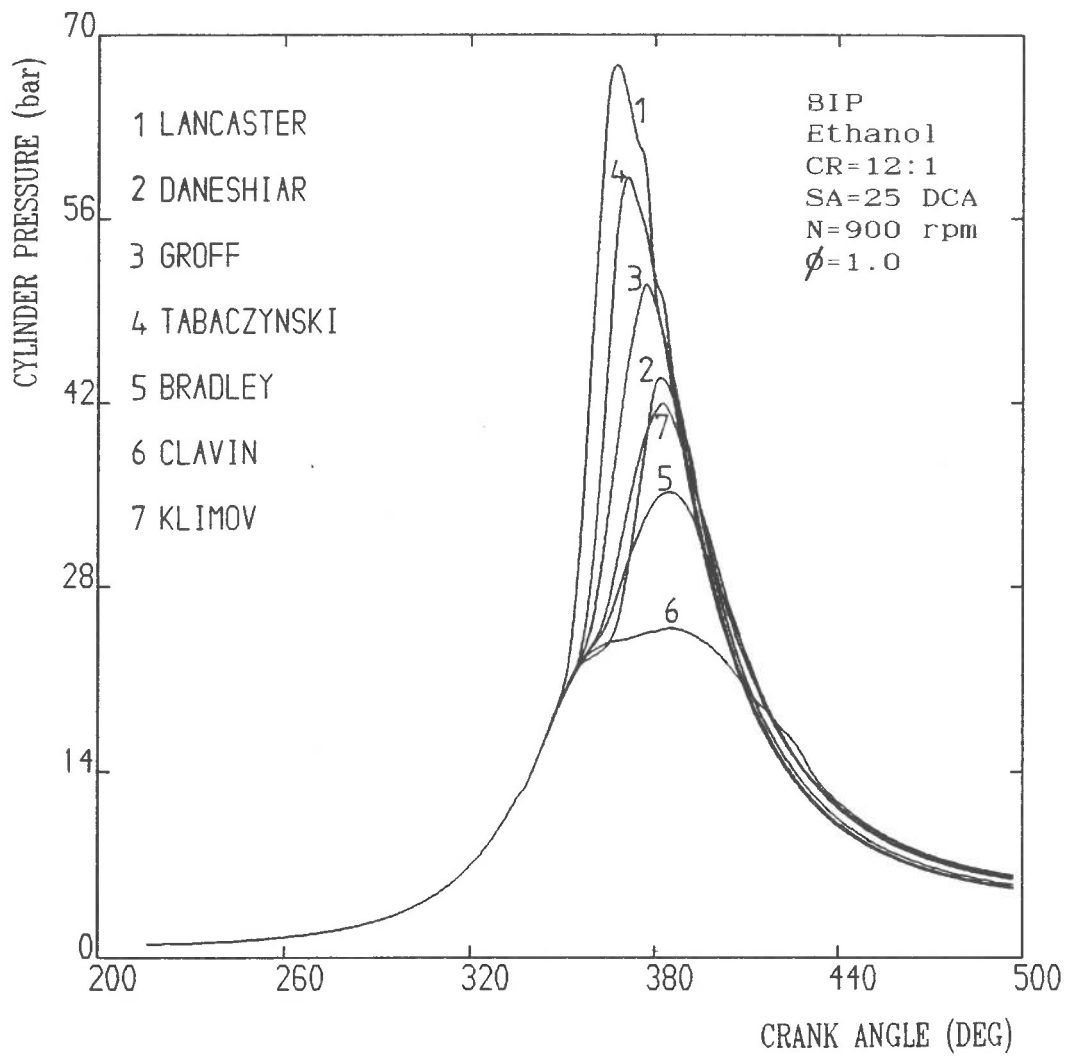


Fig. 5.35 Effect of different turbulent burning velocity correlations on cylinder pressure.

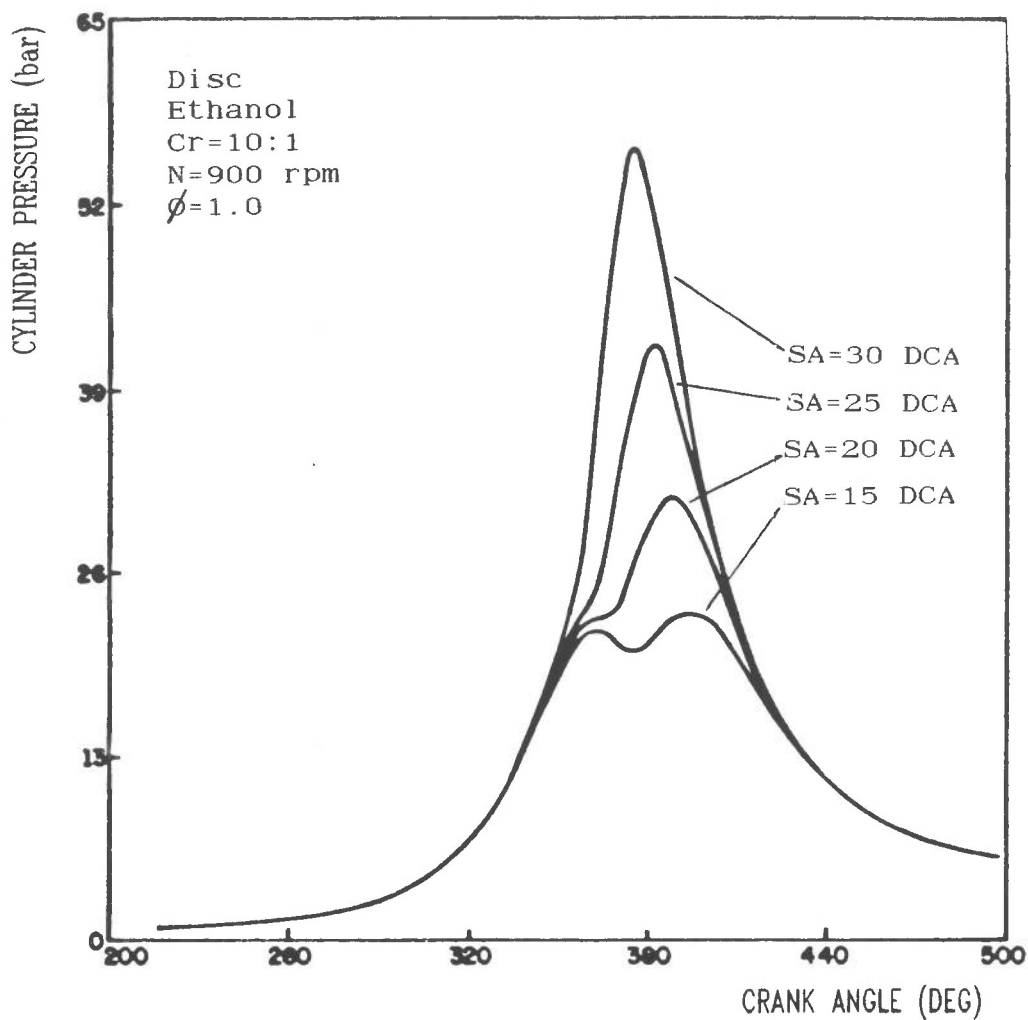


Fig. 5.36 Effect of spark timing on cylinder pressure using the correlation due to Daneshiar.

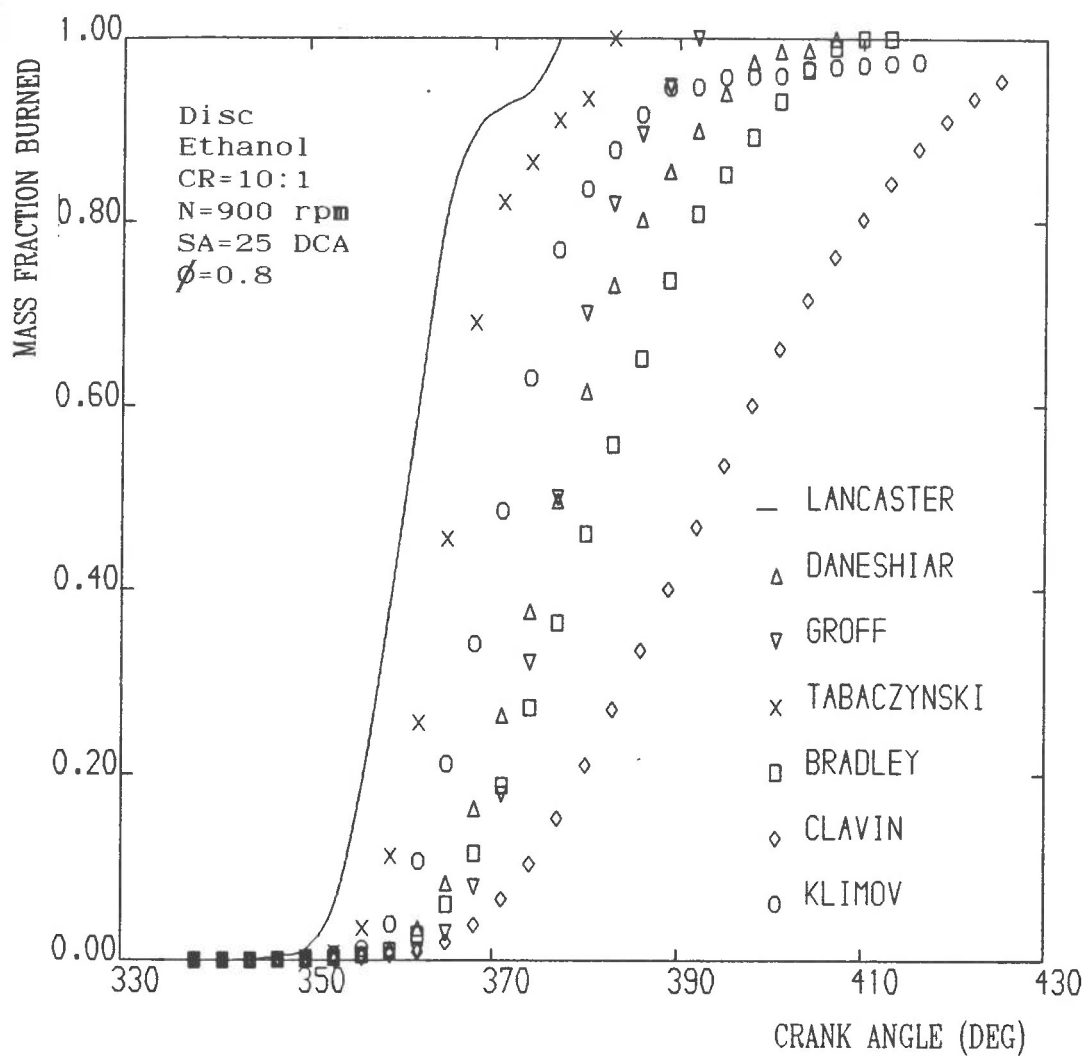


Fig. 5.37 Effect of different turbulent burning velocity correlations on mass fraction burned.

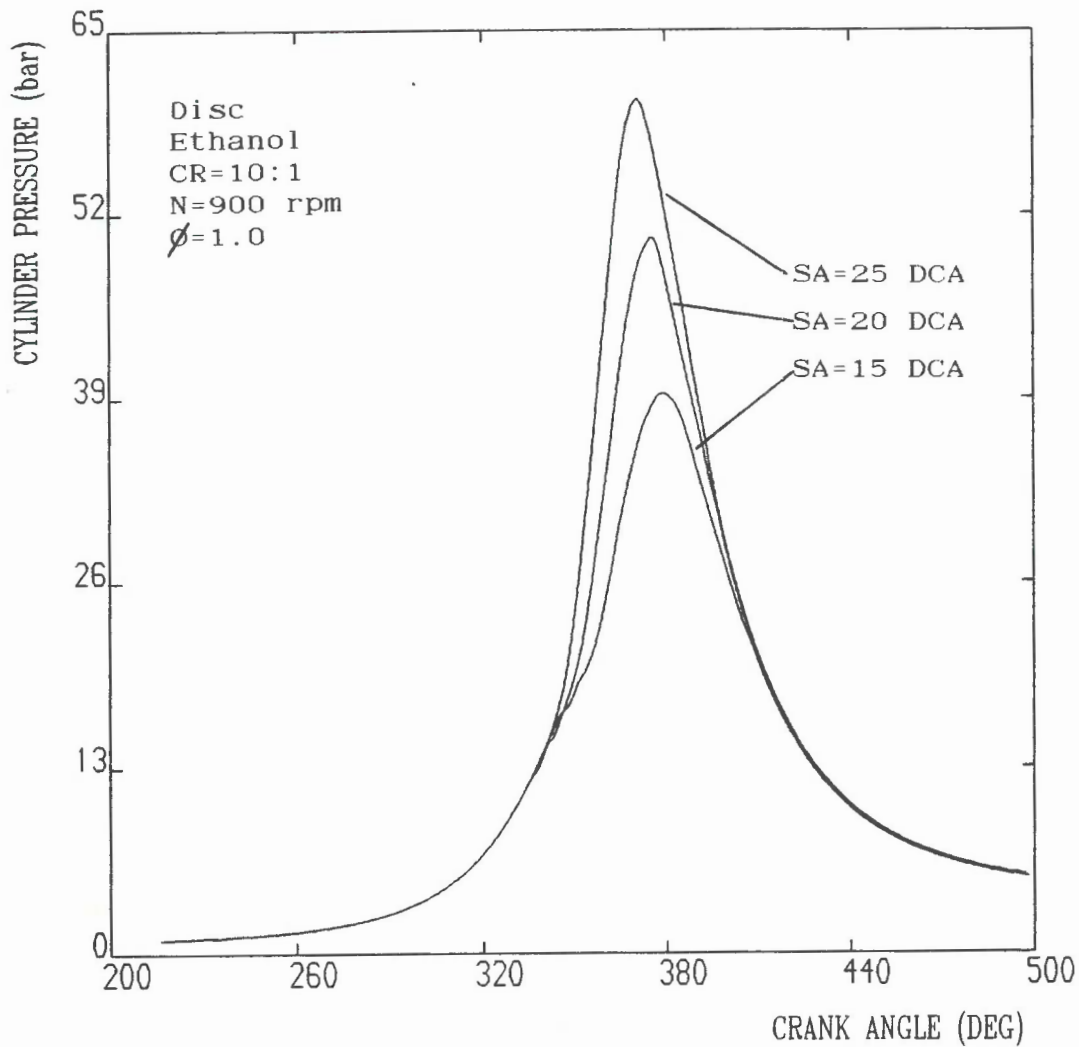


Fig 5.38 Effect of FGF on the turbulent burning velocity correlation due to Abdel-Gayed and Bradley. Pressure versus crank angle curves for various conditions of spark advance.

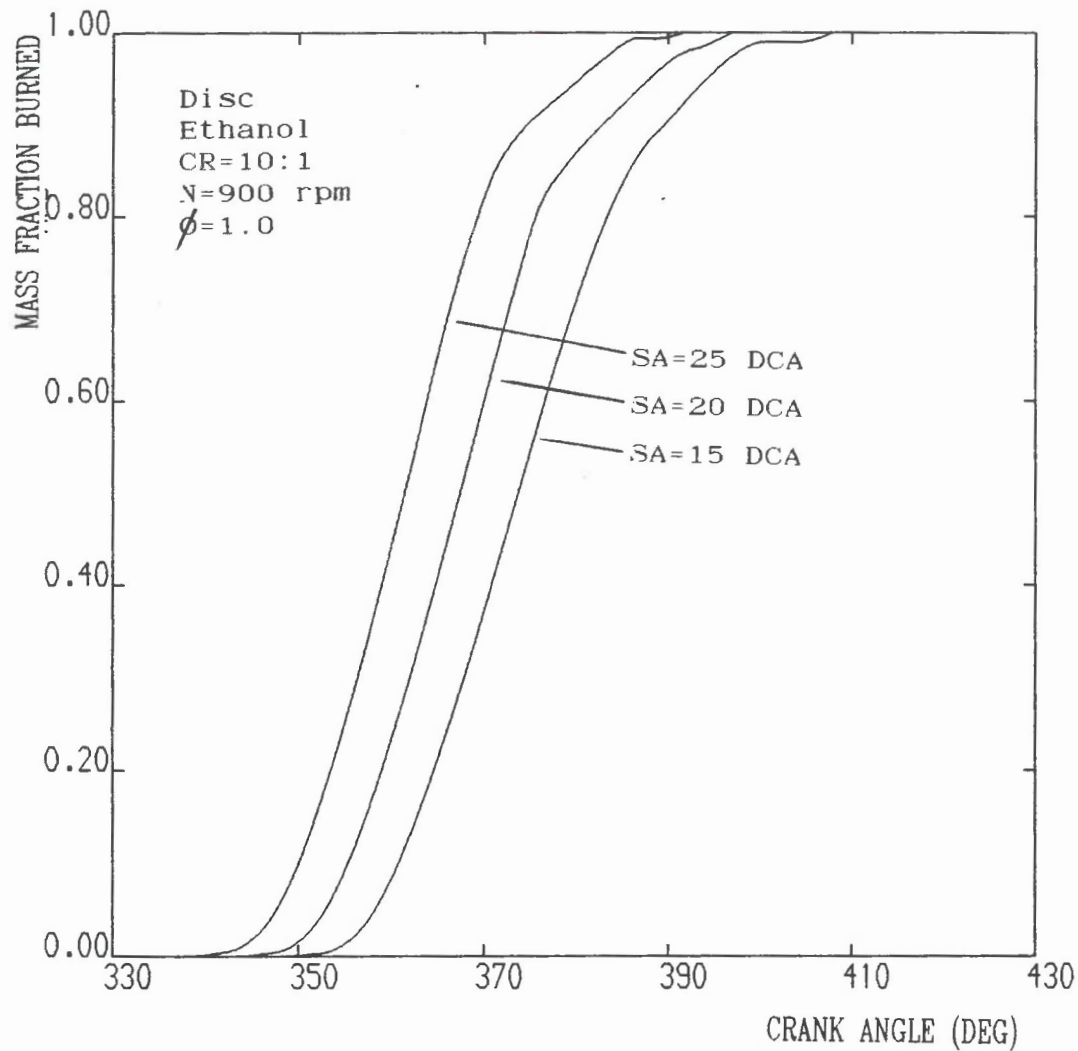


Fig 5.39 Effect of the FGF on turbulent burning velocity correlation due to Abdel-Gayed and Bradley. Mass fraction burned versus crank angle for various conditions of spark advance.

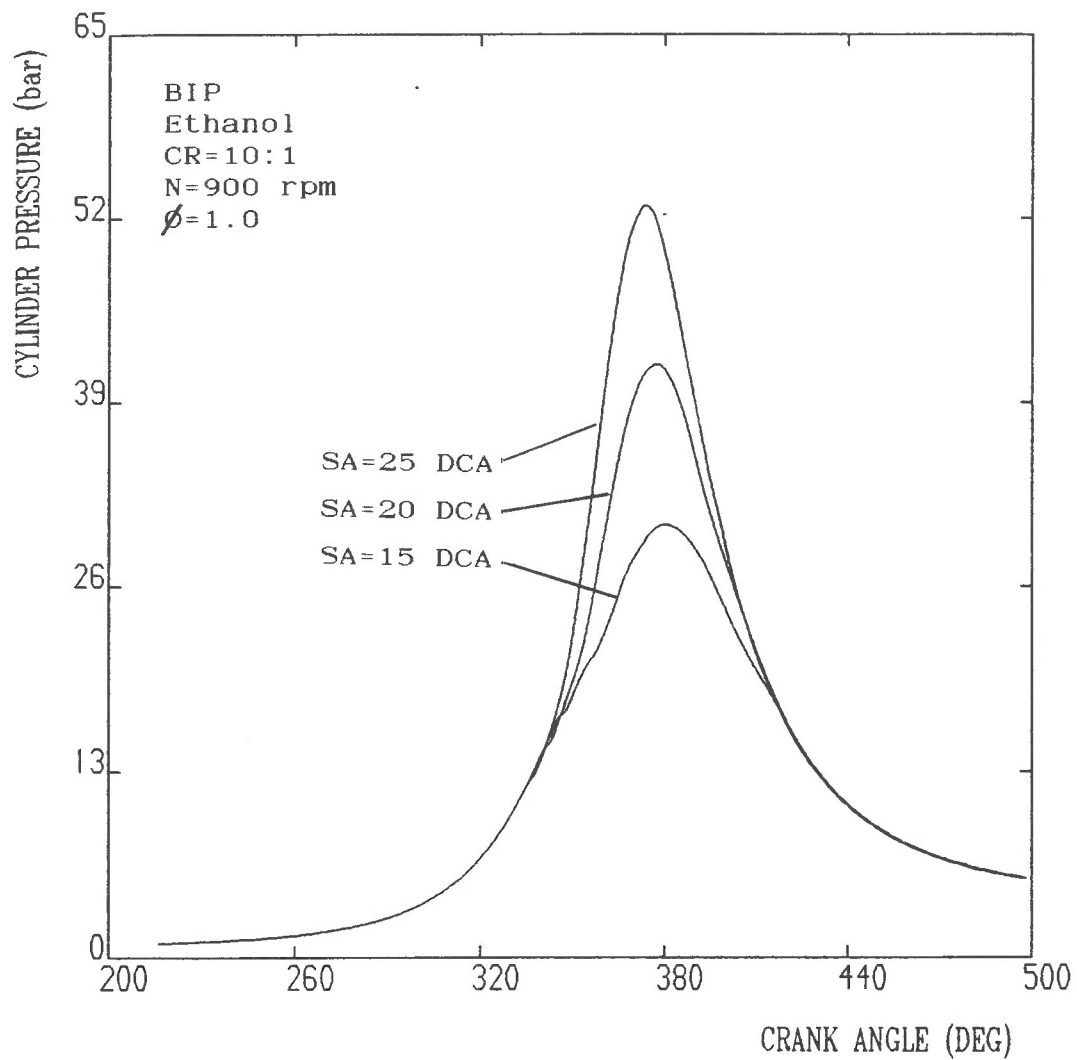


Fig 5.40 Effect of FGF on turbulent burning velocity correlation due to Clavin and Williams. Pressure versus crank angle curves for various conditions of spark advance.



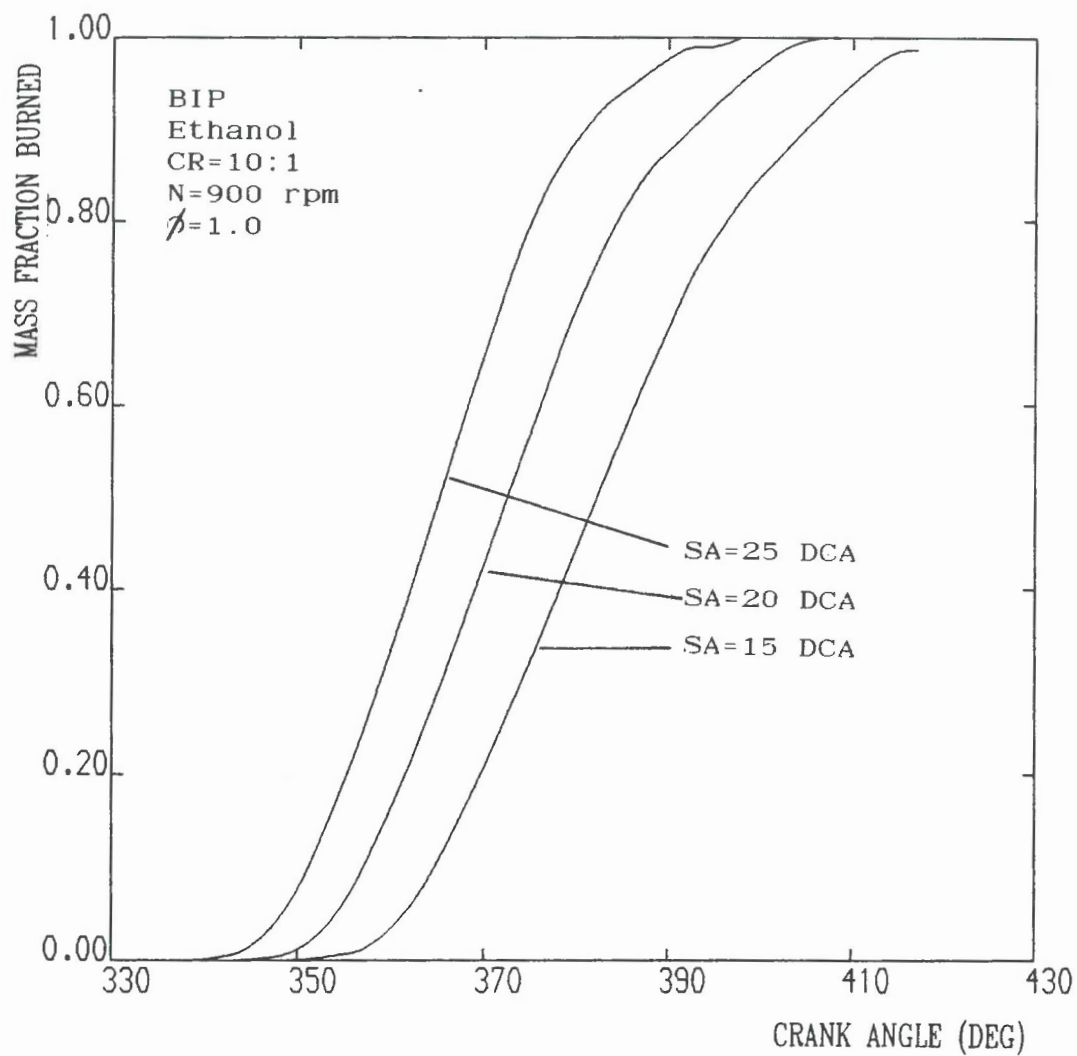


Fig 5.41 Effect of FGF on turbulent burning velocity correlation due to Clavin and Williams. Mass fraction burned versus crank angle for various conditions of spark advance.

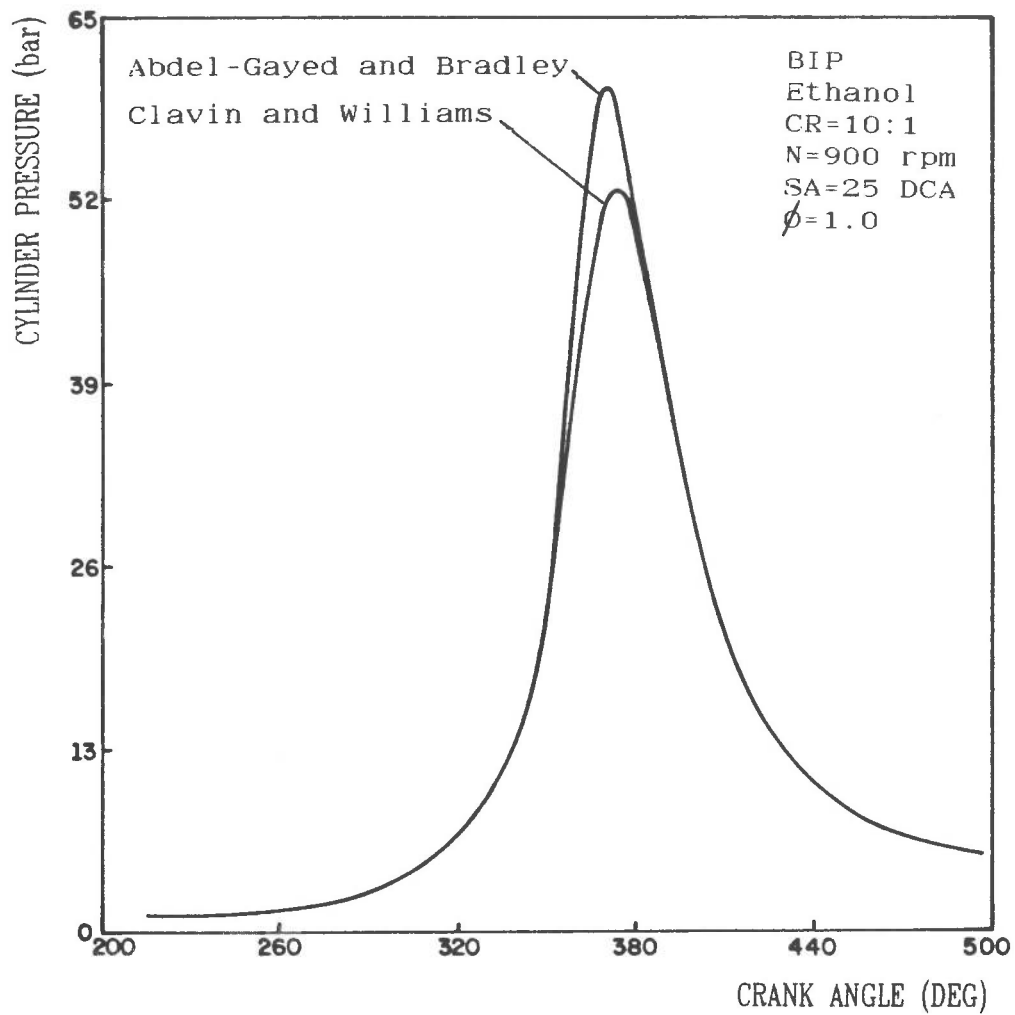


Fig. 5.42 Comparison between cylinder pressure curves using different turbulent burning velocity correlations.

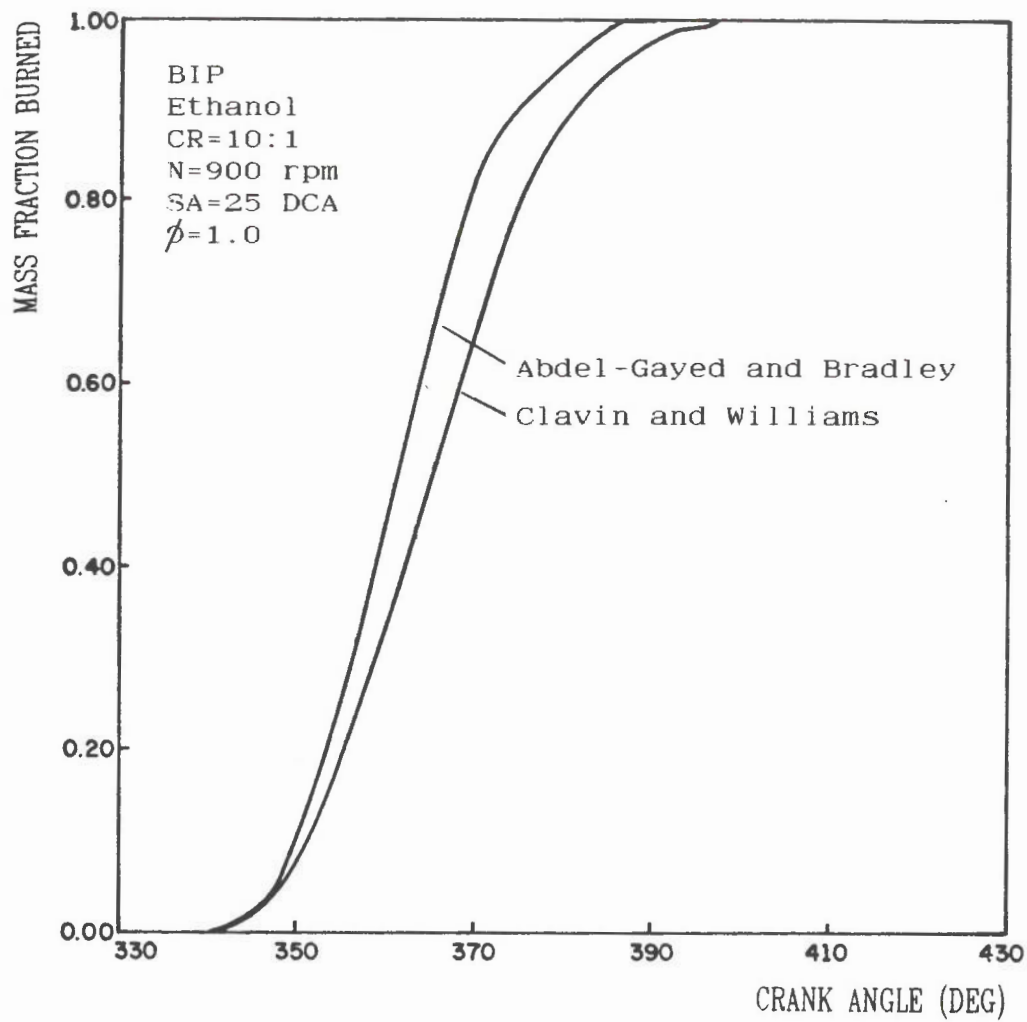


Fig. 5.43 Comparison between mass fraction burned curves using different turbulent burning velocity correlations.

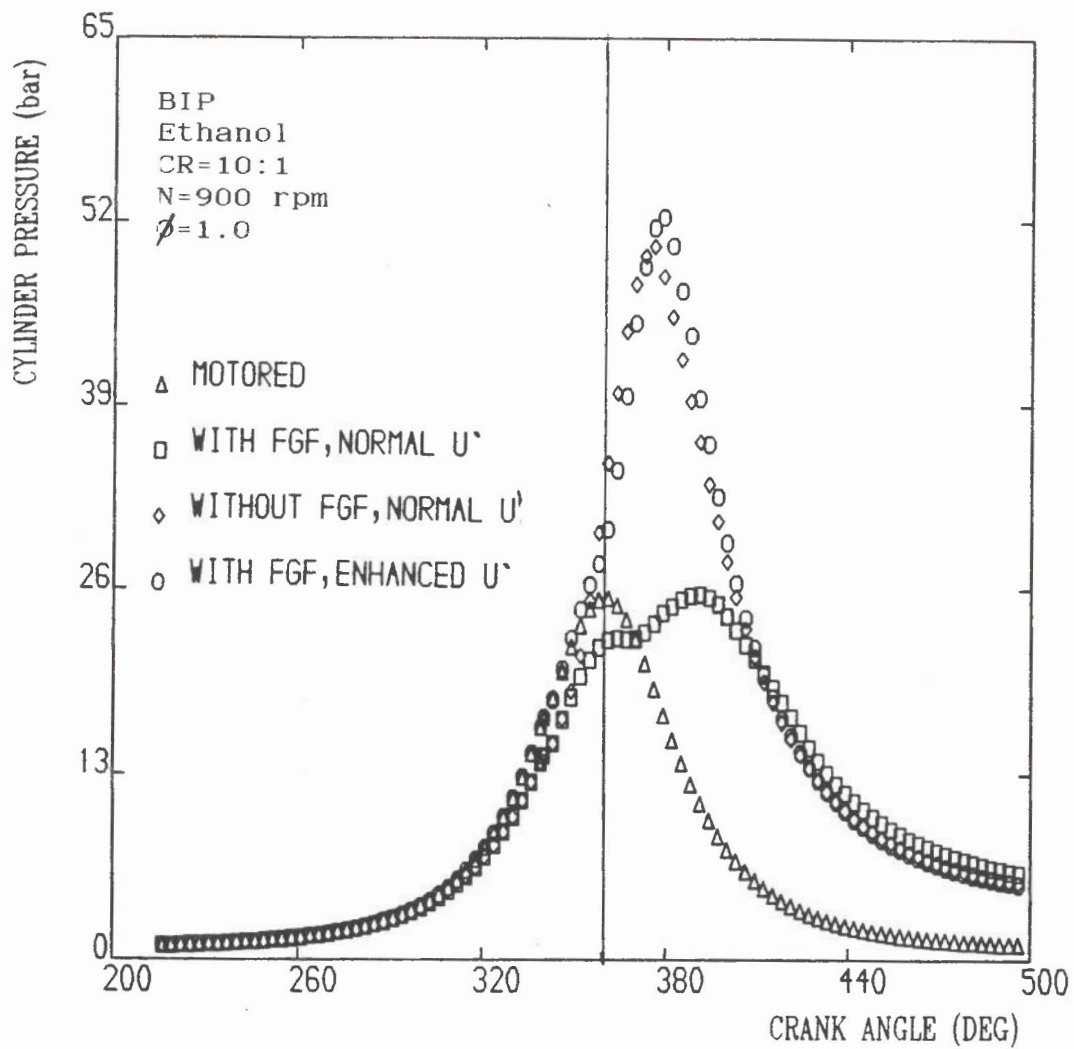


Fig. 5.44 Effect of RMS turbulent velocity on combustion; Cylinder pressure versus crank angle curves.

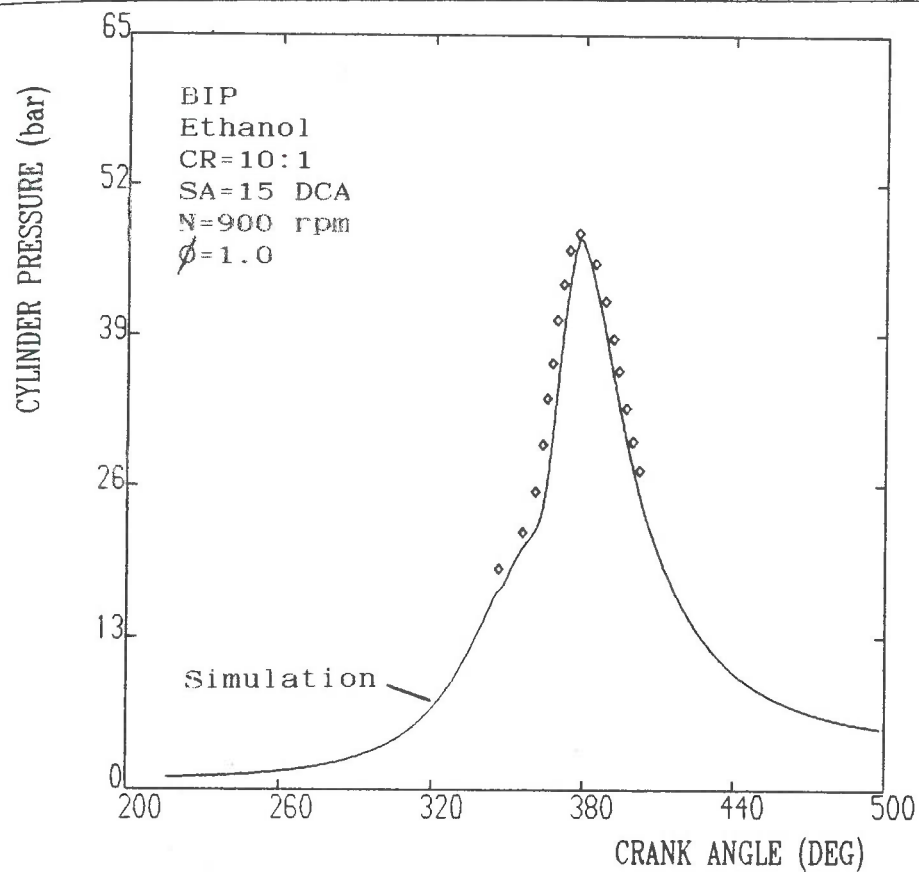
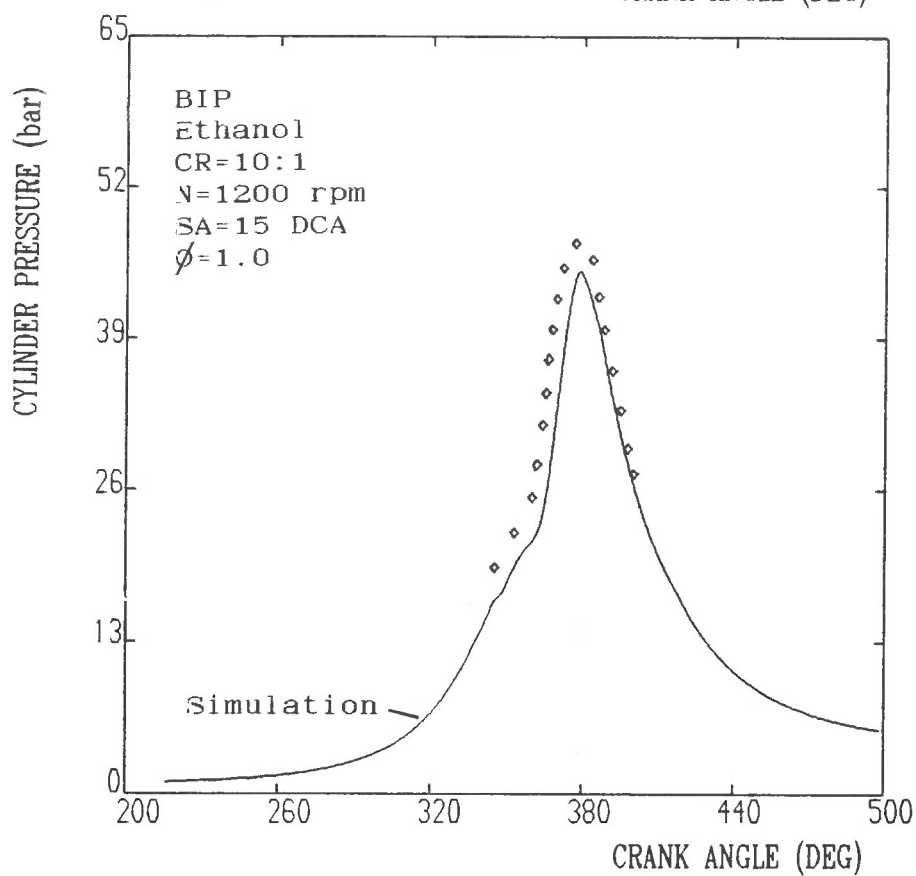


Fig. 5.45 and Fig. 5.46 Comparison between experimental and simulated cylinder pressure.

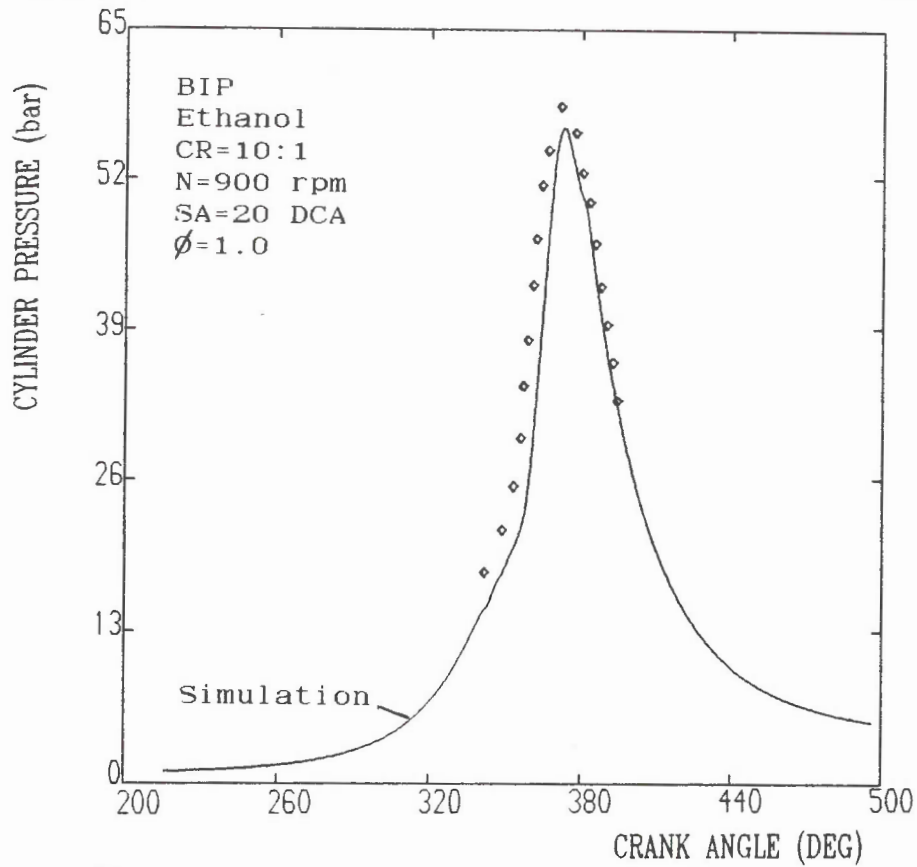
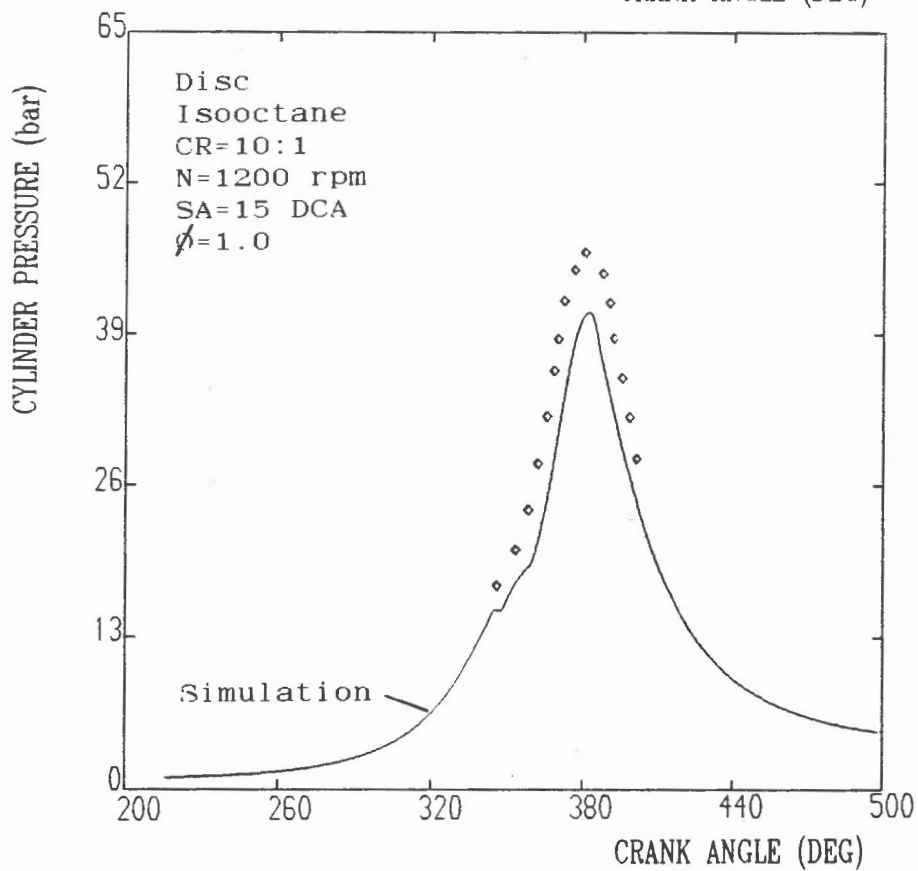
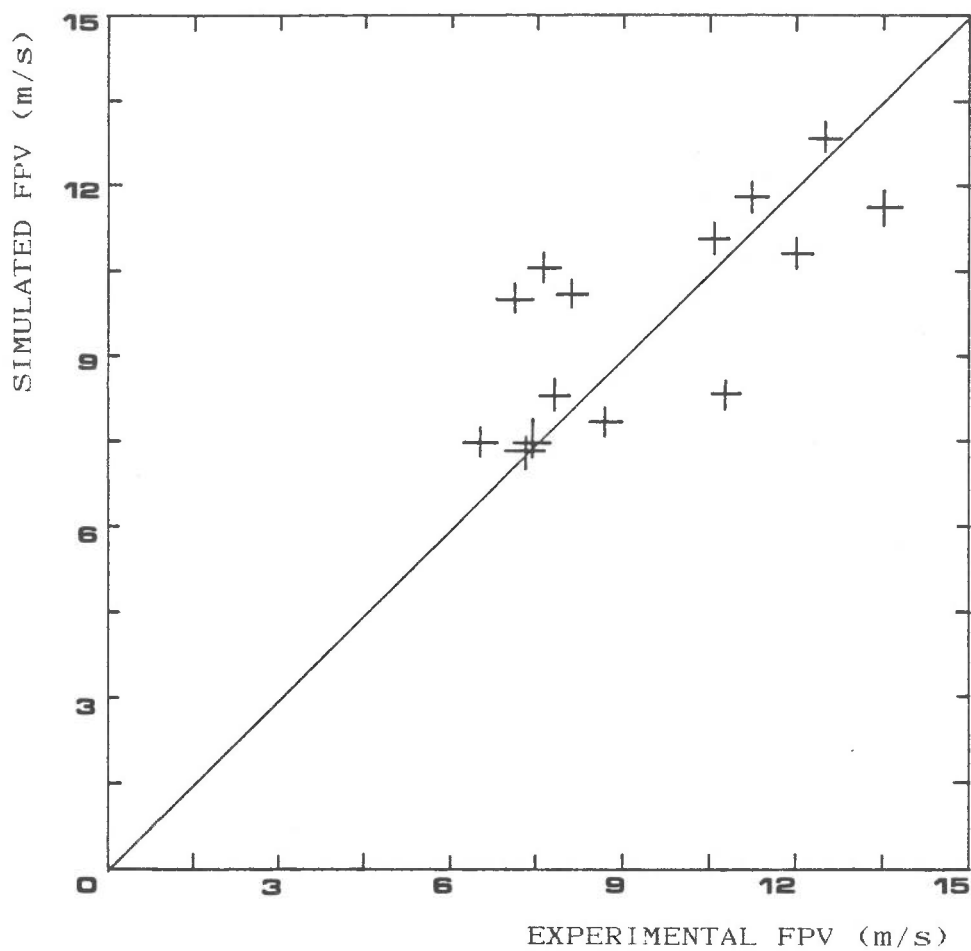


Fig. 5.47 and Fig. 5.48 Comparison between experimental and simulated cylinder pressure.





CORRELATION :  $u_{\text{e}}/u_{\text{e}} = 1.0 + 4.01 u'/u_{\text{e}}$

Fuel : Ethanol

Various conditions of engine speed,  
equivalence ratio, spark timing,  
compression ratio and chamber type.

Fig. 5.49 Theoretical versus experimental  
flame propagation velocities.

# Chapter 6

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## Conclusions and Recomendations for Further Work

## 6.1 Introduction

Detailed results from the experimental study have been presented in chapter 4. In chapter 5 the results of the computer simulation have been described and comparison between the experimental and theoretical data for cylinder pressure and flame speed reported.

In summary, the objectives of the present work were to develop a simulation programme for combustion in a spark ignition engine and investigate its validation against experiments in a single cylinder research spark ignition engine. The comparison with experiment focussed on turbulent flame propagation, which occupies a central role in the simulation, and a non-intrusive measurement technique was developed for this purpose based on laser beam refraction.

The programme establishes a methodology for evaluating engine performance, embracing both thermodynamic and geometrical effects.

## 6.2 Conclusions

1. The computer simulation programme, based on a zero-dimensional or phenomenological approach, has been developed which incorporates the following features:

- . a two-zone combustion model
- . the two zones comprised the burned and unburned gases which were at the same pressure and assumed to obey ideal gas laws.
- . the unburned charge was considered to be a homogeneous mixture of fuel vapour, air and residual gases, within which chemical reactions were absent.
- . the model also includes heat transfer from the gases to the combustion chamber walls but not between burned and unburned gases.
- . flame propagation was assumed to be spherical.

.to compute the turbulent burning velocity, a range of correlations based on experimental data have been introduced and compared.

.the simulation includes a geometrical model which calculates areas and volumes of both burned and unburned gases and their interactions with the combustion chamber walls during spherical flame propagation.

Several important influences may be observed from the simulations undertaken.

- 1.It is clear that combustion chamber geometry greatly influences the cylinder pressure and combustion rate. The geometrical parameters to have the greatest influence are spark plug position and compression ratio. Combustion chamber geometry also affects the rate of pressure rise and peak pressure.
- 2.The model predictions show that squish velocity may have an important effect on mixture turbulence, thereby affecting the combustion rate. Squish velocity is strongly affected by bumping clearance and squish ratio.
- 3.Engine performance can be greatly improved by a careful design of the combustion chamber, thus allowing leaner mixtures to be used, increasing the thermal efficiency. The choice of fuel also plays an important role. Both ethanol and isooctane have been investigated at the present study. Ethanol, allowing higher compression ratios and leaner mixtures, is shown to be an excellent substitute for gasoline. Ethanol in general presented a lower exhaust temperature as compared with isooctane indicating a higher combustion efficiency.
- 4.The model developed for the present work is revealed to be valuable tool for engine design and to a lesser extent for combustion research despite the shortcomings introduced by simplifications made in relation to spherical flame propagation. The effect that spark plug position has upon

engine performance can readily be demonstrated with great savings in time and money relative to more complex computational alternatives. These newer computational techniques, making use of finite difference methods, are increasingly being applied to detailed study of gas flows inside the engine, giving good spatial resolution. Their increasing application to engine research can be considered as directly related to recent developments in laser based techniques, which permit the gathering of detailed experimental information about the engine flow and combustion processes, inaccessible using less sophisticated methods.

Nevertheless, for general simulation of the spark ignition engine with a view to parametric investigations of alternative designs, the phenomenological models still constitute a better tool than the more complex models.

5. Burning velocity plays a crucial role in the development of turbulent combustion models applied to engine research. It is particularly important in zero-dimensional models since it links the two thermodynamically distinguished zones. The predictions performed using different correlations for turbulent burning velocity revealed that the problem of establishing a correlation which represents all the stages of combustion in an engine is far from resolved. The results suggest that the correlations based on fundamental principles do not immediately provide a satisfactory alternative to empirical correlations derived from engine experiments.

The correlations based on more fundamental quantities characterising the turbulent flow, are able to predict the turbulent burning velocity once the conditions of the turbulent field are determined. Some of the expressions do appear to be applicable only to smaller levels of turbulence than those found in internal combustion engines. It is not possible to compare them directly with those correlations specially tailored from engine experiments, which in effect

characterise both turbulent flow and flame propagation together.

6.The correlations based on more fundamental principles revealed that they can in principle be used through the whole combustion interval without the necessity of being corrected by the flame growth factor used in the empirical correlations to account for the period of initial flame kernel growth. The problem as to whether the radius of the fully developed flame is less than or greater than 30 mm is still open to question. Experimental results suggest that the subject should be examined in greater detail.

7.The laser beam technique developed to measure flame speed in the present work was successful in respect of the main objectives established during the design phase namely, reliability, simplicity and cost effectiveness. The results given by the data acquisition system showed relatively good agreement with the results given by the computer simulation as indicated in chapter 5. The comparison between experimental and theoretical flame propagation velocities, shows a slight scatter of the data but taking into account the observations regarding the uncertainties in evaluating turbulence intensity, the results are very encouraging.

### 6.3 Recommendations for further work

#### 6.3.1 Theoretical study

Despite the fact that the model has been shown to give relatively good agreement with experiments, there are several modifications that could be made in order to further generalise the model and thereby to improve the usefulness of the computational process and its accuracy.

1.The ever increasing concern with engine emissions in all countries makes this subject of prime importance to engine



researchers. The thermodynamic model could readily be changed from the present fully-mixed model to a stratified model, in order to simulate NO<sub>x</sub> emissions. In addition, a flame quench model could be introduced to address issues of wall boundaries and highly turbulent mixing. This would be useful in predicting HC emissions, another engine pollutant of great concern presently.

2.The actual model considers only that part of the cycle during which valves are closed ie, compression, combustion and expansion. Therefore it would be highly desirable to have the complete cycle simulated with the inclusion of the gas exchange process during intake and exhaust. Also, a more complete and detailed geometrical model, based on finite element techniques could be developed permitting in this way the simulation of any type of combustion chamber geometry, however complex.

3.Another desirable feature would be the introduction of a model for auto-ignition which permitted the study of the important phenomenon of spark knock. Modelling of spark knocking is becoming common practice as the use of higher compression ratios associated with leaner mixtures becomes more frequent.

#### 6.3.2 Experimental study

1.Optical access is limited in the E6 engine and therefore major improvements in the experimental data base are inhibited. Minor changes in technique are however possible. The laser based flame speed measurement system could be improved further. In the present work it is still in the phase of a prototype, developments being restricted in this study simply to achieving acceptable results. It is suggested that all components of the system could be assembled in a modular

form in order to permit easy assembling at the site of the experiments. A detailed study of lay-out and ergonomics would also be valuable.

2. The laser rig could be improved in order to allow easy movement of the laser system between two extremes of the combustion chamber, for example by installing larger windows. This would enable assessment of flame speed in a more comprehensive manner. Instrumentation and data acquisition could profitably be hardened and made more robust.

3. More ambitiously, the effects of squish generated turbulence could be examined in a bowl-in-piston combustion chamber using laser doppler anemometry, LDA. This would require some minor modifications to the spacer to provide access for another laser beam. It is suggested that small bumping clearances be used to enhance turbulence at TDC.

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# Appendix 1

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## Composition of the Mixture Before Combustion

Combustion is a general term applied to chemical reactions involving a fuel and an oxidizer, resulting in the release of energy. For internal combustion engines, hydrocarbon fuels are the most utilized, ie, fuels that contain the elements carbon and hydrogen. Hydrocarbon fuels for this particular application may be liquid or gaseous and the present work have considered only liquid fuels.

Gasoline is the most frequently used energy source for internal combustion spark ignition engines. Gasoline is a complex blend of several hydrocarbon compounds and additives, a particular composition being dependent on the source of the crude oil, the refiner and also the ultimate market of the gasoline.

Due to the complexity of the gasoline fuel, a common practice in research work is to simulate the gasoline properties by using a single compound called iso-octane, having a molecular formula  $C_8H_{18}$ .

Alcohols, in particular ethanol, are treated as pure substances having a single boiling point and therefore having specific properties as opposed to the range of properties of gasoline.

The oxidizer available for the combustion process in internal combustion engines is the atmospheric air. Dry air is composed by nitrogen, oxygen, small amount of argon and traces of carbon dioxide, neon, helium and hydrogen. The composition of air in a molar or volumetric basis can be assumed to be 20.95 percent nitrogen, 0.93 percent argon, 0.03 percent carbon dioxide and traces of hydrogen. The molecular weight is 28.96 allowing for traces of Ar and  $CO_2$ .

The term employed frequently to quantify the amount of fuel and oxidizer in the combustion process is the air/fuel ratio which in a molar basis represents the

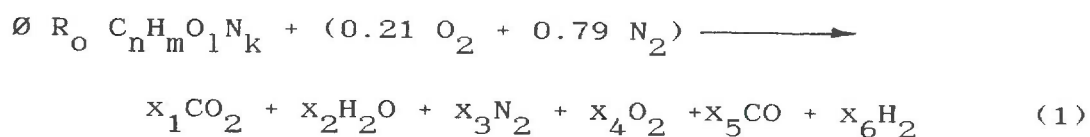
number of moles of air to the moles of fuel, or in a mass basis is the mass of air to the mass of fuel.

Another term highly used in combustion studies on internal combustion engines is the equivalence ratio  $\phi$ , defined as the ratio of the actual fuel/air ratio to the stoichiometric fuel/air ratio. Both terms are used in the present work.

In the conditions prevailing in the cylinder during the admission of the fresh charge, there is always the presence of a certain amount of residual gases which cannot be totally expelled from the combustion chamber. The total amount, however, may vary according to the valve timing setting. Therefore, the fresh charge is mixed with the remainder gases of the previous cycle.

Finally, the atmospheric air has always a certain amount of moisture present, but in the present work, when dealing with stoichiometry it is assumed that the air that reacts with the fuel is essentially dry air.

The general equation for hydrocarbon oxidation in a molar basis and in the absence of the chilled products of dissociation are given by the expressions



$$R_o = 0.21 / (n + 0.25 m + 0.5 l) \quad (2)$$

where  $R_o$  is the molar fuel/air ratio. The equation can be solved for  $R_o$  and the six coefficients  $x_i$  ( $i=1,2,3,\dots,6$ ), using traditional methods of atom balance. The equivalence ratio  $\phi$ , is defined by equation

$$\phi = F_a / F_s \quad (3)$$

and three alternatives may occur: if  $\phi < 1$  the mixture is said to be lean whilst if  $\phi > 1$  the mixture is said to be rich. For  $\phi = 1$ , the mixture is stoichiometrically correct.

When the mixture that reacts has excess air, ie, it is lean, the products will contain no carbon monoxide but instead will contain molecular oxygen that do not react with the fuel. For each of the cases above, convenient approximations for the combustion are given by expressions

$$\phi < 1 \quad x_{CO} = x_{H_2} = 0 \quad (4)$$

$$\phi > 1 \quad x_{O_2} = 0 \quad (5)$$

For the rich case, the water gas reaction



is in equilibrium, with the equilibrium constant given by

$$K = (x_{H_2O} x_{CO}) / (x_{CO_2} x_{H_2}) \quad (7)$$

The solution for the term  $x_{CO}$  is given by the positive root of the equation

$$x_{CO} = (-b + \sqrt{b^2 - 4ac}) / 2a \quad (8)$$

where the coefficients are as below

$$a = 1 - k \quad (9)$$

$$b = 0.42 - \phi R_O (2-1) + k[0.42 (\phi-1) + n \phi R_O] \quad (10)$$

$$c = -0.42 n \phi R_O (\phi - 1) k \quad (11)$$



Applying the atom balance method to the equation (7) for both cases of rich and lean mixture yields to the two sets of expressions for the number of moles  $x$  of each species, presented in Table A-1.

Table A-1

| Species              | $\phi < 1$              | $\phi > 1$                                 |
|----------------------|-------------------------|--------------------------------------------|
| $\text{CO}_2$        | $n \phi R_o$            | $n \phi R_o - x_{\text{CO}}$               |
| $\text{H}_2\text{O}$ | $m \phi R_o / 2$        | $0.42 - \phi R_o (2n - 1) + x_{\text{CO}}$ |
| $\text{N}_2$         | $0.79 + k \phi R_o / 2$ | $0.79 + k \phi R_o / 2$                    |
| $\text{O}_2$         | $0.21 (1 - \phi)$       | 0                                          |
| CO                   | 0                       | $x_{\text{CO}}$                            |
| $\text{H}_2$         | 0                       | $0.42 (\phi - 1) - x_{\text{CO}}$          |

## **Appendix 2**

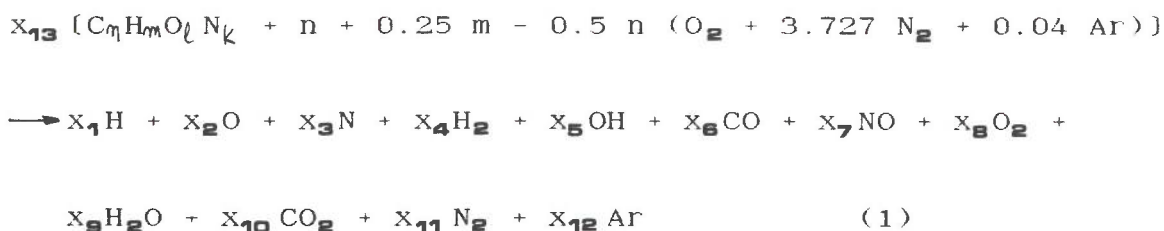
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### **Equilibrium Concentration of the Combustion Products**

At the conditions of pressure and temperature prevailing in the internal combustion engine, only 12 product species are produced in any appreciable concentrations.

In this work the products considered to be important are restricted to those 12 components, namely  $\text{CO}_2$ ,  $\text{H}_2\text{O}$ ,  $\text{N}_2$ ,  $\text{O}_2$ ,  $\text{CO}$ ,  $\text{H}_2$ ,  $\text{H}$ ,  $\text{O}$ ,  $\text{OH}$ ,  $\text{NO}$ ,  $\text{N}$ , and  $\text{Ar}$ . Therefore, to determine the number of moles of fuel necessary to produce one mole of product, there is the necessity to solve 13 equations, of which five are derived from atom balances and the sixth from the fact that the sum of all the mole fractions is equal to unit. The remaining seven equations are provided by the criteria of equilibrium among the products. All these equations have to be solved by the method of successive approximations.

Consider the general equation of a hydrocarbon fuel,  $\text{C}_m\text{H}_n\text{O}_l\text{N}_k$ , and let occur the reaction with air at equivalence ratio  $\phi$ . The products will attain the equilibrium at the temperature  $T$  and pressure  $P$ . The numbers  $n$  and  $m$  must be non-zero and  $l$  and  $k$  may or may not be zero.



In the above equation,  $x_1$  through  $x_{12}$  indicates the mole fractions of the product species while  $x_{13}$  represents the number of moles of fuel necessary to produce one mole of products.

The left side of equation (1) can be expressed as

$$x_{13} (n \text{C} + m \text{H} + r \text{O}_2 + r' \text{N}_2 + r'' \text{Ar}) \quad (2)$$

where

$$r = 1/2 + r_0$$

$$r' = k/2 + 3.727 r_0$$

$$r'' = 0.0444 r_0$$

$$r_0 = (n + 0.25 m - 0.5 l) / \phi$$

Balancing the equation for the various elements give

.carbon balance

$$x_6 + x_{10} = n x_{13} \quad (3)$$

.hydrogen balance

$$x_1 + 2x_4 + x_5 + 2x_9 = m x_{13} \quad (4)$$

.oxygen balance

$$x_2 + x_5 + x_6 + x_7 + 2x_8 + x_9 + 2x_{10} = 2r x_{13} \quad (5)$$

.nitrogen balance

$$x_3 + x_7 + 2x_{11} = 2r' x_{13} \quad (6)$$

.argon balance

$$x_{12} = r'' x_{13} \quad (7)$$

The other constraint is

$$\sum x_i = 1 \quad (i = 1, 2, \dots, 12) \quad (8)$$

To solve for the 13 unknowns, 7 more equations must be provided by the criteria of equilibrium among products, which are expressed by the 7 non-redundant hypothetical reactions.





where  $p$  is the pressure in atmospheres and  $K_i$ , the partial pressure equilibrium constants. These are curve fitted from data in JANAF Thermochemical Tables, second edition, (1971). The expressions for equilibrium constants are rearranged to express mole fractions of all the combustion products in terms of  $x_4$ ,  $x_6$ ,  $x_8$ ,  $x_{11}$ , the mole fractions of  $\text{H}_2$ ,  $\text{CO}$ ,  $\text{O}_2$ ,  $\text{N}_2$ , respectively.

$$x_1 = C_1 x_4^{0.5} \quad C_1 = K_1 / p^{0.5} \quad (10.1)$$

$$x_2 = C_2 x_8^{0.5} \quad C_2 = K_2 / p^{0.5} \quad (10.2)$$

$$x_3 = C_3 x_{11}^{0.5} \quad C_3 = K_3 / p^{0.5} \quad (10.3)$$

$$x_5 = C_5 x_4^{0.5} x_8^{0.5} \quad C_5 = K_5 \quad (10.4)$$

$$x_7 = C_7 x_8^{0.5} x_{11}^{0.5} \quad C_7 = K_7 \quad (10.5)$$

$$x_9 = C_9 x_4 x_8^{0.5} \quad C_9 = K_9 p^{0.5} \quad (10.6)$$

$$x_{10} = C_{10} x_6 x_8^{0.5} \quad C_{10} = K_{10} p^{0.5} \quad (10.7)$$

Reduction in number of equations

$$\text{from (3)} \quad x_{13} = 1/n (x_6 + x_{10})$$

from (7)

$$x_{12} = r x_{13} = r (x_6 + x_{10}) / n$$

which are used to eliminate  $x_{12}$  and  $x_{13}$

$$x_1 + 2x_4 + x_5 + 2x_9 - m/n (x_6 + x_{10}) = 0 \quad (11.1)$$

$$x_2 + x_5 + x_6 + x_7 + 2x_8 + x_9 + 2x_{10} - 2r (x_6 + x_{10})/n = 0 \quad (11.2)$$

$$x_3 + x_7 + 2x_{11} - 2r' (x_6 + x_{10})/n = 0 \quad (11.3)$$

$$\sum x_i + r'' (x_6 + x_{10})/n - 1 = 0 \quad (i = 1, \dots, 11) \quad (11.4)$$

Equations (10) can be used in equations (11) to eliminate all variables except  $x_4$ ,  $x_6$ ,  $x_8$ , and  $x_{11}$  and the resulting four non-linear equations with four unknowns may be expressed symbolically by

$$f_j (x_4, x_6, x_8, x_{11}) = 0 \quad (j = 1, 2, 3, 4) \quad (12)$$

The equations were linearized by expansion around a known vector as a Taylors series. Assuming that the solution of the equations is given by

$$\{ x_4^0, x_6^0, x_8^0, x_{11}^0 \}$$

and an initial estimate is

$$\{ x_4^*, x_6^*, x_8^*, x_{11}^* \}$$

$$\text{then } \Delta x_i = x_i^* - x_i^0 \quad (i = 4, 6, 8, 11)$$

Neglecting the partial derivatives of second and higher orders allows a set of linear equations with  $x$  as the approximate correction to be produced:

$$f_j + \left( \frac{\partial f_j}{\partial x_4} \right) \Delta x_4 + \left( \frac{\partial f_j}{\partial x_6} \right) \Delta x_6 + \left( \frac{\partial f_j}{\partial x_8} \right) \Delta x_8 + \left( \frac{\partial f_j}{\partial x_{11}} \right) \Delta x_{11} = 0 \quad (13)$$



$$(j = 1, 2, 3, 4)$$

where functions  $f_j$  and their partial derivatives are evaluated at the known vector. The equations were solved for  $\Delta x_4, \Delta x_6, \Delta x_8, \Delta x_{11}$ , using a gaussian elimination method.

The improved values are then

$$x_i^p = x_i^o + \Delta x_i \quad (i = 4, 6, 8, 11) \quad (14)$$

which can now being used to solve the partial derivatives and functions. The process of corrections can continue until the relative changes in all components are less than a specified value.

The method employed to solve equations (12) was the Newton-Raphson iteration and provided this method is not self starting an approximate solution, known from previous execution of the subroutine at conditions close to those presently encountered must be given to get the iteration started.

Suitable initial values were produced by the following method which assume that the product of combustion is limited to species  $H_2$ ,  $CO$ ,  $O_2$ ,  $H_2O$ ,  $CO_2$ ,  $N_2$ , and Ar. Equation (1) becomes

$$x_{13} (n C + m H + r O_2 + r' N_2 + r'' Ar)$$

$$x_6 CO + x_8 O_2 + x_9 H_2O + x_{10} CO_2 + x_{11} N_2 + x_{12} Ar$$

From the carbon balance and equation (10.7)

$$x_6 = n x_{13} / (1 + C_{10} x_8^{0.5}) \quad (15.1)$$

From the hydrogen balance and equation (10.6)

$$x_4 = 0.5 m x_{13} / (1 + C_9 x_8^{0.5}) \quad (15.2)$$

From the nitrogen, argon and oxygen balances

$$x_{11} = r^I x_{13} \quad (15.3)$$

$$x_{12} = r^{II} x_{13} \quad (15.4)$$

$$x_6 + 2x_8 + x_9 + 2x_{10} = 2r x_{13} \quad (15.5)$$

Substituting for  $x_4$ ,  $x_{11}$ ,  $x_{12}$ , and  $x_6$ , rearranging the terms dividing by  $x_{13}$  and letting

$$A = (2 C_{10} n x_8^{0.5} + n) / (1 + C_9 x_8^{0.5})$$

$$B = (0.5 C_9 m x_8^{0.5}) / (1 + C_9 x_8^{0.5})$$

results in the expression

$$A + B + 2x_8 / x_{13} - 2r = 0 \quad (15.6)$$

An initial estimate of  $x$  can be obtained with good accuracy from the requirement for the sum of the mole fractions of the products to be unity

$$x_4 + x_6 + \sum x_i = 1 \quad [i = 8, \dots, 12] \quad (15.7)$$

For  $0 \leq 1$  a good estimate of  $x$  is obtained from equation

$$x_{13} = 1 / (0.25 m + r + r^I + r^{II}) \quad (16)$$

For  $0 > 1$  the expression that gives an estimation for  $x$  is

$$x_{13} = 1 / (n + 0.5 m + r^I + r^{II}) \quad (17)$$

Substitution of the estimated value of  $x_{13}$  into equation (15.6) gives an equation in the single unknown  $x_8$  which can be solved by Newton's method. The other unknowns can

then be obtained by substitution in equations (15.1), (15.2), (15.3). The initial estimates can then be used to start the iterations of the main solution.

As mentioned previously, data for the calculation of the equilibrium constants were taken from JANAF Thermochemical Tables. The equilibrium constants for the reactions considered to be in equilibrium were calculated using the relation

$$\begin{aligned} \log K(\text{reaction}) = & \log K(\text{formation})\text{Products} \\ & - \log K(\text{formation})\text{Reactants} \end{aligned}$$

The constants were calculated using the relationship

$$\log K = A \ln T + B/T + C + D T + E T^2$$

where  $T$  is the absolute temperature and  $A$ ,  $B$ ,  $C$ ,  $D$ , and  $E$  are constants. The model was used to fit the tabulated data by means of a least squares technique. The constants are given in Table A.2, and are valid in the range of 600 to 4000 K.

Table A.2

|                                                                                | A                         | B                      | C                      | D                         | E                         |
|--------------------------------------------------------------------------------|---------------------------|------------------------|------------------------|---------------------------|---------------------------|
| $\frac{1}{2} \text{H}_2 \rightleftharpoons \text{H}$                           | .432168                   | $-.112464 \times 10^2$ | $.267269 \times 10^1$  | $-.745744 \times 10^{-1}$ | $.242484 \times 10^{-2}$  |
| $\frac{1}{2} \text{O}_2 \rightleftharpoons \text{O}$                           | .310805                   | $-.129540 \times 10^2$ | $.321779 \times 10^1$  | $-.738336 \times 10^{-1}$ | $.344645 \times 10^{-2}$  |
| $\frac{1}{2} \text{N}_2 \rightleftharpoons \text{N}$                           | .389716                   | $-.245828 \times 10^2$ | $.314505 \times 10^1$  | $-.963730 \times 10^{-1}$ | $.585643 \times 10^{-2}$  |
| $\frac{1}{2} \text{O}_2 + \frac{1}{2} \text{H}_2 \rightleftharpoons \text{OH}$ | -.141784                  | $-.213308 \times 10^1$ | .853461                | $.355015 \times 10^{-1}$  | $-.310227 \times 10^{-2}$ |
| $\frac{1}{2} \text{N}_2 + \frac{1}{2} \text{O}_2 \rightleftharpoons \text{NO}$ | $.150879 \times 10^{-1}$  | $-.470959 \times 10^1$ | .646096                | $.272805 \times 10^{-2}$  | $-.154444 \times 10^{-2}$ |
| $\text{H}_2 + \frac{1}{2} \text{O}_2 \rightleftharpoons \text{H}_2\text{O}$    | -.752364                  | $.124210 \times 10^2$  | $-.260286 \times 10^1$ | .259556                   | $-.162687 \times 10^{-1}$ |
| $\text{CO} + \frac{1}{2} \text{O}_2 \rightleftharpoons \text{CO}_2$            | $-.415302 \times 10^{-2}$ | $.148627 \times 10^2$  | $-.475746 \times 10^1$ | .124699                   | $-.900227 \times 10^{-2}$ |

# Appendix 3

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## Thermodynamic Properties

The thermodynamic properties of both the unburned gas and the burned gas are calculated assuming they behave as perfect gases.

The thermodynamic data for the individual species are available from the JANAF Thermochemical Tables, second edition (1971). However, for computer calculations, it is more interesting to have the data curve fitted to polynomials.

This have been done by the method of minimizing the least squares error and then the polynomials were normalized by the universal gas constant ( $R_0 = 8.3143 \text{ kJ/kgmol-K}$ ).

The expressions to calculate the specific heat at constant pressure, the enthalpy and the entropy of a mixture of perfect gases are then given by equations (1), (2), (3). The reference temperature is  $T = 298 \text{ K}$ , with the enthalpy of H , O , N , and C set to zero.

$$C_p / R_0 = a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4 \quad (1)$$

$$h / R_0 = a_1 + (a_2/2) T + (a_3/3) T^2 + (a_4/4) T^3 + (a_5/5) T^4 + a_6/T \quad (2)$$

$$S^0 / R_0 = a_1 \ln(T) + a_2 T + (a_3/2) T^2 + (a_4/3) T^3 + (a_5/4) T^4 + a_7 \quad (3)$$

$a_1$  and  $a_7$  are constants of integration determined by matching the enthalpy and entropy at the reference temperature. The values of all the constants are given in Table A.3 and they were extracted from Gordon and McBride (1971). Exception is made to the fuels constants which were calculated by a curve fitting subroutine developed for the present work.

The number of moles of each species is denoted by  $x$  and their calculation is presented in appendix 1 for the unburned mixture and in appendix 2 for the burned mixture. In this appendix both unburned and burned mixture are treated simply as mixture and the properties defined for both.

.The total number of moles of the mixture is given by equation

$$X = \sum x_i \quad (i = 1, 2, \dots) \quad (1)$$

.The mole fraction of each individual species is given by equation

$$y_i = x_i / X \quad (i = 1, 2, \dots) \quad (2)$$

.The molecular weight of the mixture is calculated from expression

$$M = \sum M_i \quad (i = 1, 2, \dots) \quad (3)$$

.The enthalpy of the mixture is estimated from equation

$$H = \sum x_i h_i \quad (i = 1, 2, \dots) \quad (4)$$

.The internal energy of the mixture is calculated from equation

$$U = \sum x_i u_i \quad (i = 1, 2, \dots) \quad (5)$$

.The gas constant of the mixture is calculated from equation

$$R = R_o / M \quad (6)$$



## Low temperature thermo data (300 &lt; T &lt; 1000 K)

|                                  |               |                |                |                |                |                |                |
|----------------------------------|---------------|----------------|----------------|----------------|----------------|----------------|----------------|
| CO <sub>2</sub>                  | 2.4007797E+00 | 8.7350961E-03  | -6.6078778E-06 | 2.0021862E-09  | 6.3274039E-16  | -4.8377527E+04 | 9.6951456E+00  |
| H <sub>2</sub> O                 | 4.0701275E+00 | -1.1084499E-03 | 4.1521180E-06  | -2.9637404E-09 | 8.0702101E-13  | -3.0279723E+04 | -3.2270047E-01 |
| N <sub>2</sub>                   | 3.6748261E+00 | -1.2081499E-03 | 2.3240102E-06  | -6.3217559E-10 | -2.2577253E-13 | -1.0611588E+03 | 2.3580425E+00  |
| O <sub>2</sub>                   | 3.6255984E+00 | -1.8782184E-03 | 7.0554543E-06  | -6.7635137E-09 | 2.1555993E-12  | -1.0475226E+03 | 4.3052778E+00  |
| CO                               | 3.7100928E+00 | -1.6190964E-03 | 3.6923593E-06  | -2.0319675E-09 | 2.3953344E-13  | -1.4356310E+04 | 2.9555349E+00  |
| H <sub>2</sub>                   | 3.0574450E+00 | 2.6765200E-03  | -5.8099163E-06 | 5.5210392E-09  | -1.8122739E-12 | -9.8890472E+02 | -2.2997055E+00 |
| C <sub>2</sub> H <sub>5</sub> OH | 2.2711000E+01 | -1.2751000E-02 | 1.0420000E-04  | -6.0841998E-08 | -9.4819996E-13 | -3.3912000E+04 | -7.7852997E+01 |
| C <sub>2</sub> H <sub>4</sub>    | 3.3210001E+00 | 6.1516198E-03  | 4.2551001E-05  | -5.6910000E-08 | 2.1223001E-11  | -2.9821000E+04 | 1.1742000E+01  |

## High temperature thermo data (300 &lt; T &lt; 4000 K)

|                  |               |                |                |                |                |                |                |
|------------------|---------------|----------------|----------------|----------------|----------------|----------------|----------------|
| CO <sub>2</sub>  | 4.4608040E+00 | 3.0981719E-03  | -1.2392571E-06 | 2.2741325E-10  | -1.5525955E-14 | -4.8961441E+04 | -9.8635983E-01 |
| H <sub>2</sub> O | 2.7167633E+00 | 2.9451374E-03  | -8.0224373E-07 | 1.0226682E-10  | -4.8472147E-15 | -2.0905826E+04 | 6.6305671E+00  |
| N <sub>2</sub>   | 2.8963194E+00 | 1.5154866E-03  | -5.7235275E-07 | 9.9807392E-11  | -6.5223557E-15 | -9.0586182E+02 | 6.1615148E+00  |
| O <sub>2</sub>   | 3.6219535E+00 | 7.3618261E-04  | -1.9652228E-07 | 3.6201559E-11  | -2.8945627E-15 | -1.2019825E+03 | 3.6150961E+00  |
| CO               | 2.9840696E+00 | 1.4891390E-03  | -5.7899683E-07 | 1.0364577E-10  | -6.9353550E-15 | -1.4245228E+04 | 6.3479156E+00  |
| H <sub>2</sub>   | 3.1001902E+00 | 5.1119464E-04  | 5.2644211E-08  | -3.4909974E-11 | 3.6945345E-15  | -8.7738043E+02 | -1.9629421E+00 |
| H                | 2.5000000E+00 | 0.0000000E+00  | 0.0000000E+00  | 0.0000000E+00  | 0.0000000E+00  | 2.5471627E+04  | -4.6011764E-01 |
| O                | 2.5420597E+00 | -2.7550619E-05 | -3.1028033E-09 | 4.5510674E-12  | -4.3680515E-16 | 2.9230803E+04  | 4.9203081E+00  |
| OH               | 2.9106426E+00 | 9.5931650E-04  | -1.9441703E-07 | 1.3756646E-11  | 1.4224542E-16  | 3.9353816E+03  | 5.4423447E+00  |
| NO               | 3.1889999E+00 | 1.3382281E-03  | -5.2899316E-07 | 9.5919335E-11  | -6.4847932E-15 | 9.8283291E+03  | 6.7458124E+00  |
| N                | 2.4502699E+00 | 1.0661500E-04  | -7.4653400E-08 | 1.8796501E-11  | -1.0259800E-15 | 5.6116000E+04  | 4.4487600E+00  |
| Ar               | 2.5000000E+00 | 0.0000000E+00  | 0.0000000E+00  | 0.0000000E+00  | 0.0000000E+00  | -7.4537500E+02 | 4.3660002E+00  |

Table A.3

## Appendix 4

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### Determination of the Equivalence Ratio

In the study of the performance of any internal combustion engine, the correct determination of the fuel/air ratio is a very important subject. In order to achieve an accurate determination of the fuel/air ratio, a precise measurement of the fuel and air flow rates is of capital importance.

The rate at which an engine consumes fuel may be determined in different ways, one of the most and accurate and certainly most convenient is the burette, the one that has been used in the present work. The equipment is well known among engine researchers and therefore requires no detailed description.

However, it is worth to mention that a modification has been made in the original equipment supplied with the Ricardo E6 engine in order to allow the system to be used with two different fuels.

The original cock was substituted by a five way valve which permitted to connect, according to the convenience one of the two fuel tanks, the burette and the engine. Fig A4.1 represents the 5 possible configurations. With the valve in the position to operate with a certain fuel, the connection between the fuel tank, the burette and the engine were established. When measuring the fuel flow, the passage between the fuel tank and the engine is blocked, therefore allowing the fuel from the burette to be consumed.

The burette method is fundamentally accurate but it may introduce some human error in the timing of the falling fuel column, specially when fuel flow rates are very high or very low. This human error can be minimized through very careful use of the equipment and using average time from various measurements.

In Fig A4.1, the full lines indicate fuel flow whereas the dashed lines indicate the blocked valve ports. The 5

possibilities allowed by the system are

- .position 1 - fuel from the ethanol tank flows to the engine and burette
- .position 2 - fuel from the burette feeds the engine allowing the consumption to be measured
- .position 3 - fuel from the burette is drained during the change of fuels.
- .position 4 - fuel from the iso-octane tank flows to the engine and burette
- .position 5 - fuel from the burette feeds the engine allowing the consumption to be measured

The air flow measurements have been performed using the Alcock viscous flow meter supplied with the engine. Due to the pulsating nature of the flow and the nature of air itself, the accurate measurement of air flow through engines is difficult to achieve.

The pulsating flow leads to the formation of standing pressure waves in the induction system and measuring equipment and this condition is specially important in the case of single cylinder engines running at wide open throttle and low speeds.

Most equipments have to use a damping system to minimize these effects which makes them very difficult to handle. However, the viscous flow meter is not adversely affected by pulsating flows and therefore does not require a damping capacity in series with it.

In the Alcock flow meter, air flows through a flame trap matrix of many triangular passages whose cross sectional areas are very small compared to their lengths. The total area of the flow meter is chosen in such a way to give a viscous flow in the range of the application. The air across this matrix creates a pressure differential which due to the viscous nature of the flow is proportional to the flow velocity.

This pressure difference is measured by means of a multi-slope manometer connected to the meter. The complete system, consisting of meter and multi-slope manometer, is claimed to have been accurately calibrated by Ricardo & Co Ltd and as the system had few hours of work in a clean environment, it was decided not to proceed to a recalibration of the system. Recalibration is recommended when the cells of the flow meter become partially obstructed due to dust accumulation, for example.

The calibration curve of the meter, as supplied by Ricardo is given in Fig A4.2. The meter is calibrated at 20°C and for temperatures differing from that value, a graph is given to determine the viscosity correction factor as a function of the temperature as depicted in Fig A4.3.

The procedure utilised to calculate the equivalence ratio for each condition of test is described in sequence. The volume of air leaving the meter is equal to the calibration constant times the manometer reading and times the correction factor for viscosity obtained from Fig A4.3, if temperature is different from 20°C.

$$V_a = CC \ MR \ F_v \quad (1)$$

It is convenient that this volume flow be reduced to some reference conditions ie, 20°C and 760 mm Hg. Therefore,

$$V_a = V_a \ P_e / P_r \left\{ (T_r + 273) / (T_e + 273) \right\} \quad (2)$$

where  $T_r = 20^\circ\text{C}$

$P_r = 760 \text{ mm Hg}$

$T_e = \text{temperature during the test}$

The density of the air at the reference conditions is  $\rho_a$ , therefore, the mass flow rate is given by



$$M_a = V_a \rho_a \quad (3)$$

The volume of fuel has been taken as 50 ml which gives a volume flow rate of

$$V_f = 50 \cdot 10^{-6} / t \quad (4)$$

The mass flow rate of fuel can be calculated from

$$M_f = V_f \rho_f \quad (5)$$

where  $\rho_f$  is the density of the fuel.

The fuel/air ratio is then determined from

$$F/A = M_f / M_a \quad (6)$$

and the equivalence ratio by

$$\phi = (F/A) / (F/A)_s$$

where the subscript s indicates a stoichiometric fuel/air ratio.

Combining all the above equations in only one, the equivalence ratio can be estimated from

$$\phi = \{P_e \rho_f V_f (T_e + 273)\} / \{CC \ h \ t \ F_v \ Pr \ \rho_a \ (F/A)_s \ (T_r + 273)\} \quad (7)$$

Substituting all the terms but h and t by their values during the time interval of testing, equation (7) may be simplified as follows

$$\phi = K / h \ t \quad (8)$$

ie, the equivalence ratio is equal to a constant, function of atmospheric conditions and fuel type, divided by the time measured to consume the specified volume of fuel and



divided by the height of the liquid column in the inclined manometer.

The value of the calibration constant as given by Ricardo for the manometer operating in the second inclined position and using blended paraffin of specific gravity 0.784 is equal to  $2.94 \cdot 10^{-4}$ .

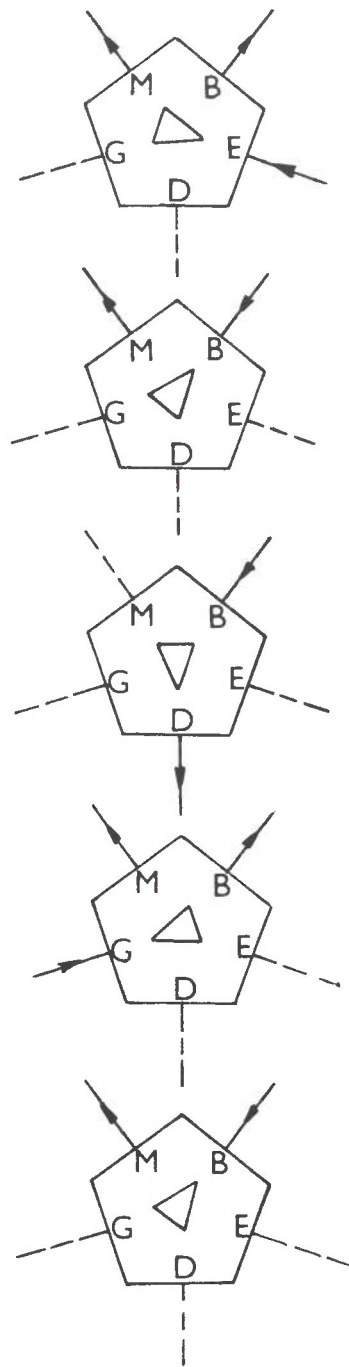


Fig. A4.1 Fuel system five way valve.

ALCOCK VISCOUS FLOW AIR METER No. 510V  
CALIBRATION ON STEADY FLOW CORRECTED  
TO 20°C. AIR TEMPERATURE

Drg. No. D29163

Date 16-10-75

FOR OTHER TEMPS. USE CORRECTION FACTOR FROM GRAPH No. 5653

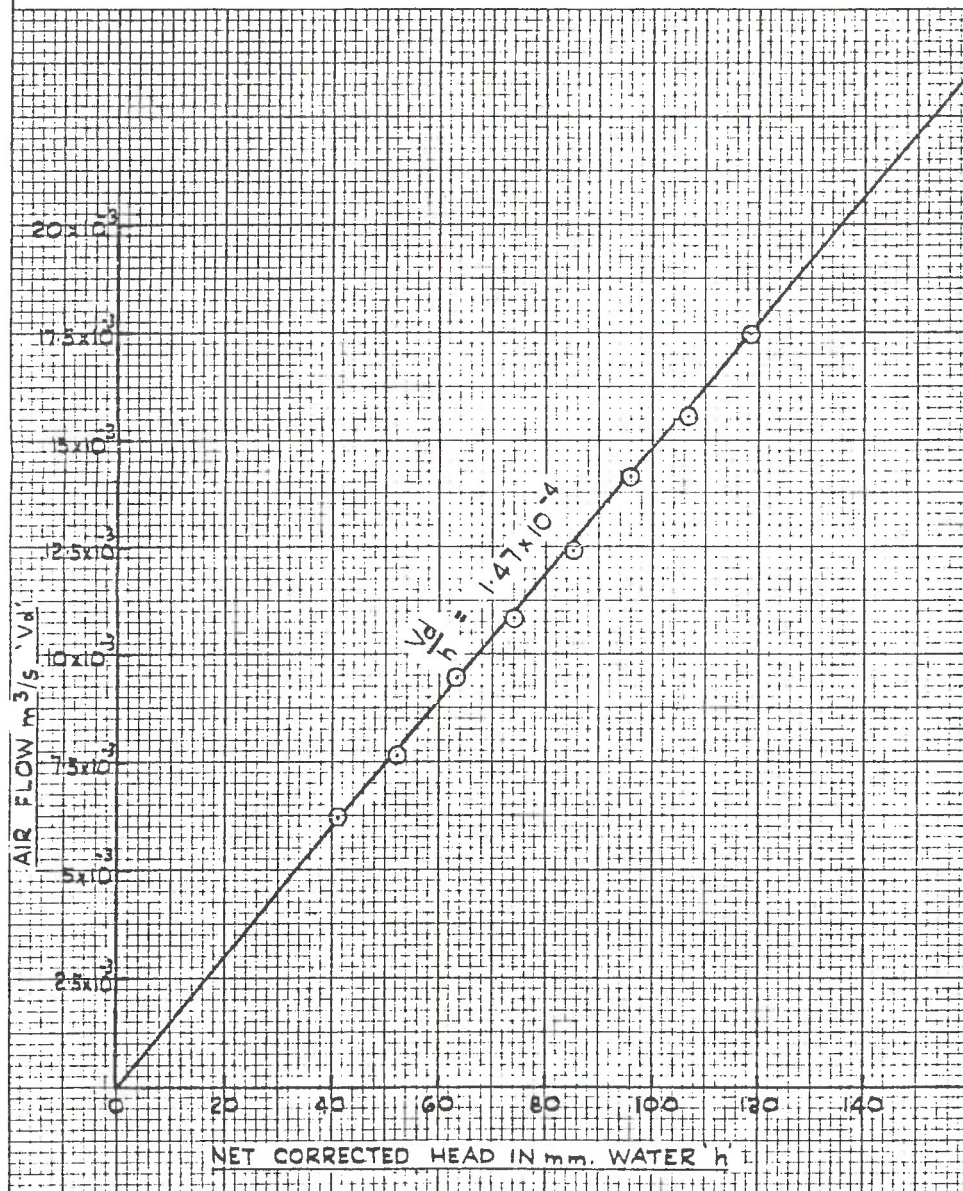


Fig. A4.2 Alcock flow meter calibration graph.

RICARDO & CO. *Engineers* (1927) LTD.  
BRIDGE WORKS ENGINE TESTING LABORATORY.

DRG.No. D.5653

VISCOUS FLOW AIR METER  
TEMPERATURE CORRECTION FOR METERS CALIBRATED AT 20°C.

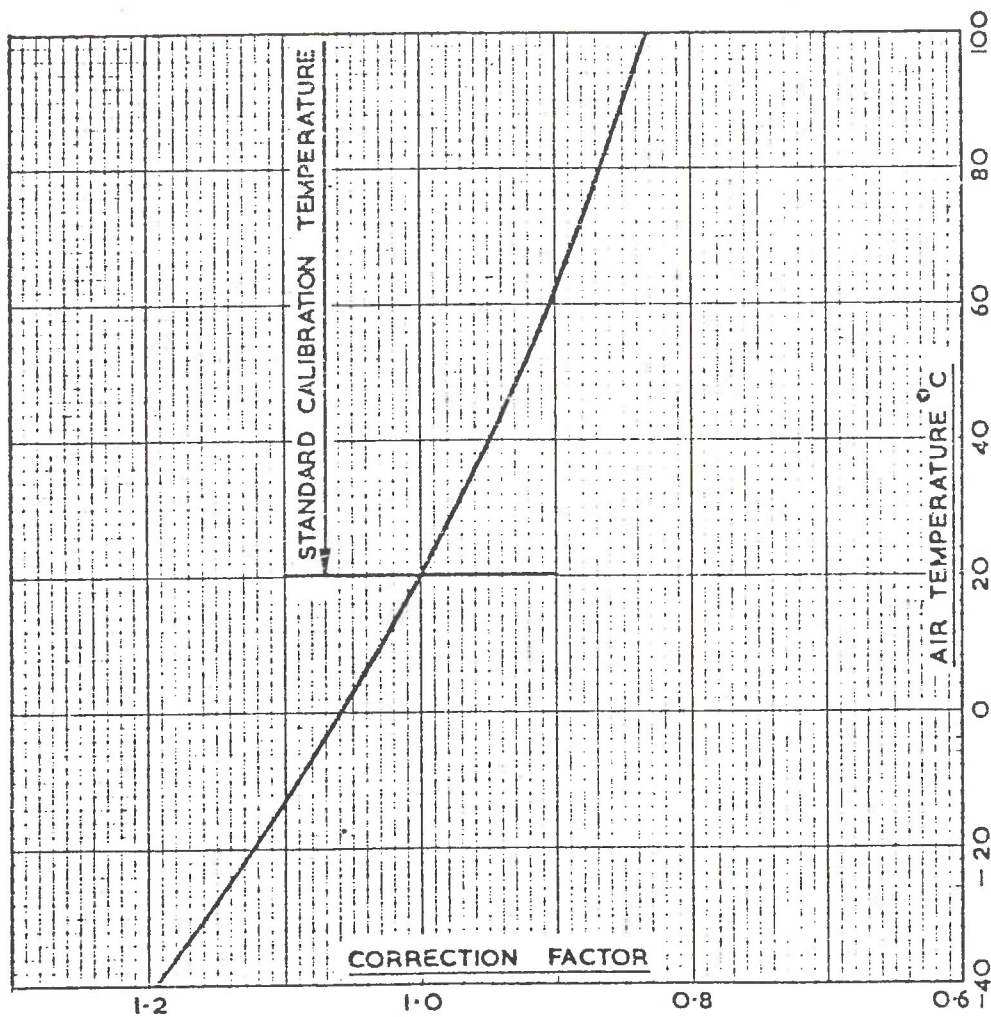


Fig. A4.3 Viscous flow air meter temperature correction graph

# Appendix 5

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## Combustion Chamber Dimensions



In this appendix, the calculation of the geometric dimensions of the combustion chambers are considered. A description of the combustion chambers utilized in the present work may be found in chapter 3 and chapter 4.

The procedure used to dimension these combustion chambers is described in sequence. The volumes of all the recesses in the cylinder head have been measured by filling these recesses with a thin oil contained in a graduated burette. In this way, the volume of oil necessary to fill the cavities can be read directly from the graduated burette, giving directly and accurately the volume of the recesses. The same procedure was adopted for the measurement of the recess volumes in the spacer and bowl-in-piston chamber. Fig A5.1 depicts the equipment employed, illustrating the determination of the recess volumes of a bowl-in-piston chamber.

The compression ratio and the clearance volume are related by the expression

$$V_{cl} = V_s / (CR - 1) \quad (1)$$

With the spacer assembled in the engine, two copper gaskets, having a thickness of 0.9 mm, should be used which produce additional dead volumes calculated by the expression

$$V_g = h (D_g^2 - D_c^2) \pi / 4 \quad (2)$$

Therefore, the total dead volume is calculated by

$$V_r = V_h + V_g + V_s \quad (3)$$

The total volume of the combustion chamber is defined by a disc of diameter  $D_c$  and height  $H$ , this volume including the volume of all recesses in the combustion chamber. Thus, to determine the height  $H$  of the disc, the volume of the



recesses should be subtracted from the total volume or clearance volume ie,

$$V_d = V_{cl} - V_r \quad (4)$$

$$H = 4 V_d / \pi D_c^2 \quad (5)$$

For the combustion chamber tested having a compression ratio equal to 10 : 1, the total recess volume was found to be equal to 6 cubic centimeters, giving a value for the total height equal to 10.44 mm. The symbols and their meaning are given in Fig A5.2.

The Heron chamber also follows the same principles but requires more extensive calculations. In Fig A5.3 are indicated the symbols utilised in this formulation. The Heron chamber worked with a bumping clearance of 1.0 mm in order to assure that a high degree of squish was attained.

It was found that a relief for the valves was necessary in order to avoid that both valves hit the piston crown during the period that they were open with the piston in the TDC. Furthermore, to provide a passage to the laser beams, a channel had also to be machined in the walls of the bowl, complicating still further the determination of the recess volumes.

In order to determine accurately the total volume of the bowl-in-piston combustion chamber, a procedure involving three steps was applied, as detailed in sequence. Applying the definition of squish ratio allowed to the calculation of the bowl diameter according to the following expression

$$SQR = (D_c^2 - D_b^2) / D_c^2 \quad (6)$$

The bowl diameter, according to the expression above, was

found to be equal to 53.88 mm. The height of the bowl was taken as 10 mm and the root radius 8 mm, and a prototype chamber machined to those dimensions. The volume of this prototype chamber was accurately measured and then the recesses for the valves and the channel for the laser were machined.

The volume was again measured and subtracted from the previous one, giving the total volume of the recesses. The final step consisted in the application of the information obtained above to determine the total height of the bowl according to the procedure below.

.dead volume due to gasket

$$V_g = h (D_g^2 - D_c^2) \pi / 4 \quad (7)$$

.dead volume due to bumping clearance

$$V_{bc} = D_c^2 BC \pi / 4 \quad (8)$$

.dead volume in the cylinder head  $V_h$ , is measured with the burette method

.for a given squish ratio, the volume of the lower portion of the bowl  $V_{ro}$ , is taken from measurements in the prototype.

.the total dead volume, including the volume of the valve recesses and laser channel  $V_r$ , is computed from

$$V_t = V_h + V_g + V_{bc} + V_r \quad (9)$$

.the clearance volume is given by

$$V_{cl} = V_s / (CR - 1) \quad (10)$$

.for a given SQR the diameter of the bowl is given by

$$D_b = D_c (1 - SQR) \quad (11)$$

.the volume of the upper cylinder (Fig A5.3) can be computed as

$$V = V_{cl} - V_t - V_{ro} - V_r \quad (12)$$

and the height of this cylinder is given by

$$H = 4 V / \pi D_b^2 \quad (13)$$

Finally, adding to this height, the radius of the root, the total height of the combustion chamber is determined. The sample calculation shown above was used to find the height of the bowl-in-piston combustion chamber having compression ratios of 10 : 1 and 12 : 1 and a squish ratio of 50 % utilised in the present work.

The total volume of the recesses was found to be 7.77 cubic centimeters and the total height of the bowl equal to 22 mm for CR = 10 : 1 and equal to 17.60 mm for CR = 12 : 1.

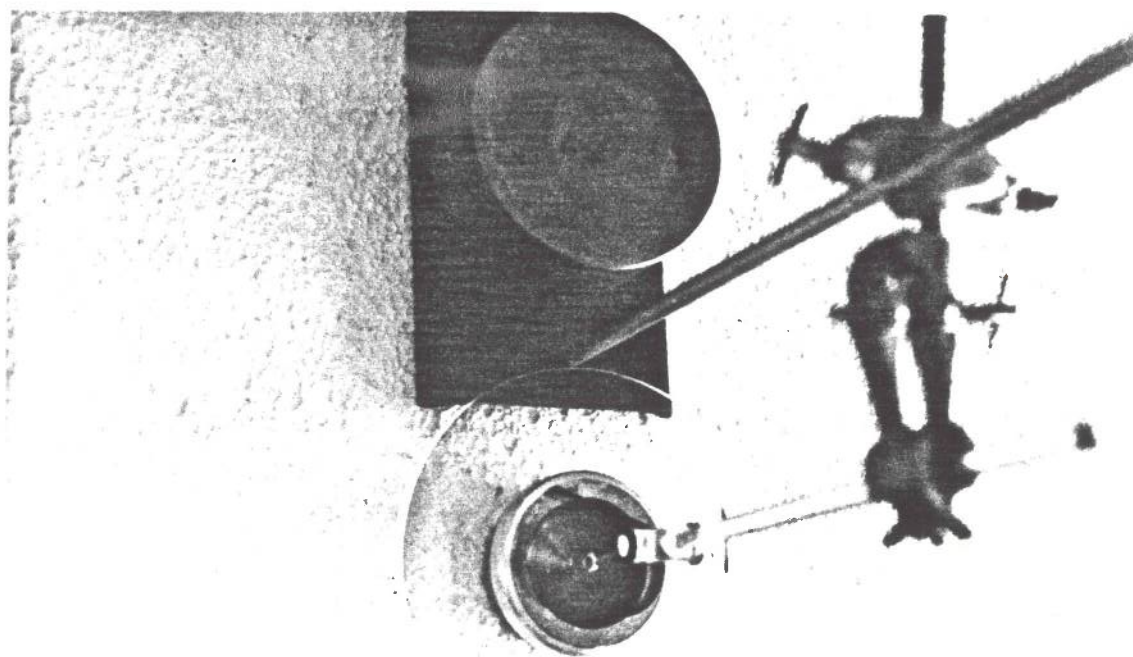


Fig. A5.1 Equipment for measuring combustion chamber volume.

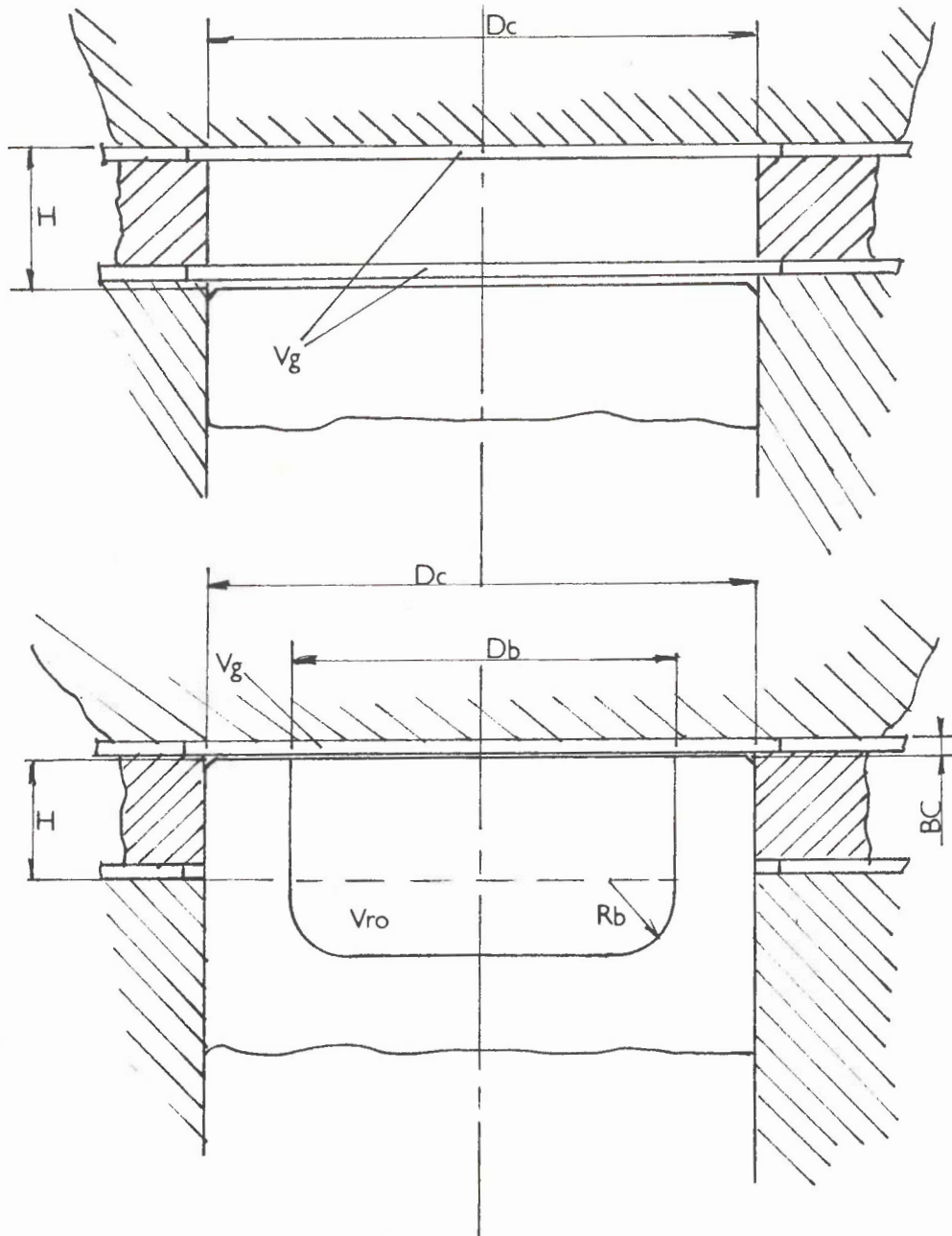


Fig. A5.2 (top) Symbols and dimensions of a DISC combustion chamber.

Fig. A5.3 (bottom) Symbols and dimensions of a BIP combustion chamber.

## Appendix 6

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### Graph to Determine Compression Ratio



The compression ratio in the Ricardo E6 can be varied, while the engine is running, by raising or lowering the cylinder head relative to the crank shaft, through a screw and nut mechanism. A micrometer, fixed in the crank case structure provides a means to evaluate the relative amount of travel of the cylinder head. The micrometer measures the distance between the top of the crank case and the cylinder head. A graph is supplied with the engine in which the micrometer readings are plotted against the compression ratio. In such a way, the compression ratio can easily be determined by knowing the reading of the micrometer.

However, when the spacer is mounted between the cylinder barrel and the cylinder head the graphic supplied no longer applies, therefore demanding the plotting of a new graph that take the spacer into account.

The recessed volumes in the cylinder head and spacer have been determined with the burette method and found to be equal to 3.1 and 2.8 cubic centimeters respectively. The micrometer readings corresponding to increments of 2.54 mm in the distance between piston crown and top face of spacer can be seen in Table A6.2.

The dead volumes left by the gasket and the spacer have been calculated from their geometric dimensions and are as follows

.for gasket - bore diameter,  $D_g = 79.00 \text{ mm}$   
 thickness,  $t_g = 0.90 \text{ mm}$

the volume can be calculated as

$$V_g = \frac{\pi}{4} D_g^2 t_g = 4.41 \text{ cm}^3$$

.for spacer - bore diameter,  $D_s = 76.20 \text{ mm}$   
 thickness,  $t_s = 12.00 \text{ mm}$

the volume being estimated from

$$V_s' = D_s^2 t_s / 4 = 54.72 \text{ cm}^3$$

taking into account the recessed volumes in the spacer, the total volume is

$$V_{sp} = V_s' + 2.8 = 57.52 \text{ cm}^3$$

In order to have the same compression ratio of the engine without spacer, the clearance volumes should be the same in both cases, therefore, knowing the clearance height that corresponds to a given CR without spacer, the same clearance height should be maintained when the spacer is fitted. The clearance heights when the spacer is not fitted are given in Table A6.1.

Using the data from Table A6.1 and Table A6.2, a graph that relates the clearance heights with the micrometer readings can be plotted which allows to find the compression ratio, given the micrometer reading when the spacer is fitted.

For example, to set the compression ratio  $CR = 10 : 1$  using the graph of Fig A6.1 the procedure is as follows

.find the clearance volume

$$V_{cl} = V_s / (CR - 1) = 56.31 \text{ cm}^3$$

.subtract all the recessed volumes from the clearance volume

$$V = V_{cl} - V_r - V_g = 45.90 \text{ cm}^3$$

.find the clearance height

$$H = V / A_p = 1.006 \text{ cm}$$

with  $A_p = \pi D_c^2 / 4$

From the graph, a vertical line by  $H = 10.06$  mm reaches the line B at point P. From there, a horizontal line to the vertical scale give a micrometer reading of 10.00 mm which is the value that the micrometer should be adjusted if a compression ratio equal to 10 : 1 is to be set.

Table A6.1

| Clearance<br>height | Micrometer<br>readings |
|---------------------|------------------------|
| 5.0                 | 22.71                  |
| 10.0                | 17.72                  |
| 15.0                | 12.72                  |
| 20.0                | 7.71                   |
| 25.0                | 2.72                   |

Table A6.2

| Clearance<br>height | Micrometer<br>readings |
|---------------------|------------------------|
| 2.54,               | 17.76                  |
| 5.08                | 15.75                  |
| 7.62                | 12.70                  |
| 10.16               | 10.13                  |
| 12.70               | 5.64                   |
| 15.24               | 5.05                   |
| 17.78               | 2.46                   |

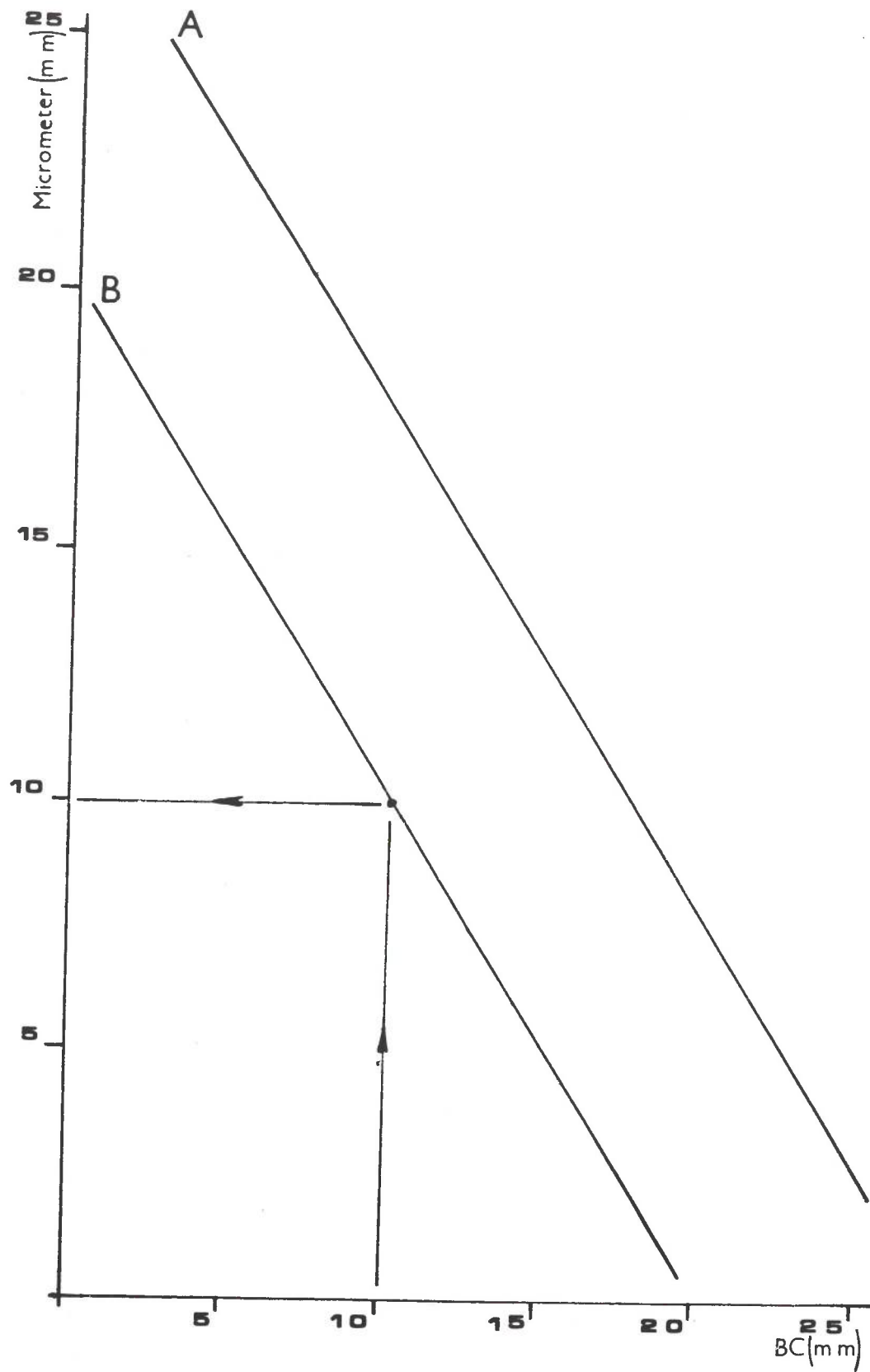


Fig. A6.1 Graph to determine the compression ratio.