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THE EFFECT OF COMBUSTION CHAMBER DESIGN ON THE COMBUSTION RATE IN AN S.I. ENGINE

by

M.F.J. BRUNT

A Doctoral Thesis

Submitted in partial fulfilment of the requirements for the award of Doctor of Philosophy of the Loughborough University of Technology

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SUMMARY

The effect of combustion chamber design on combustion rate has been investigated experimentally and theoretically.

The experimental work concentrated on the measurement of cylinder pressure and flame speed using a piezo-electric pressure transducer and multiple ionisation probes together with a data acquisition/processing system. A total of twenty one chamber designs of varying shape, compression ratio and spark plug arrangement were tested over a range of operating conditions on a single cylinder S.I. engine.

The pressure data were analysed to obtain values of pressure rise rate, cyclic dispersion and combustion (mass burn) rate whilst the ionisation data were processed to yield flame travel angles and flame dispersion. The results obtained show that for a given compression ratio, the flame speed is not significantly affected by chamber design. In contrast, the combustion rate and pressure parameters are highly dependent on the chamber design; more compact arrangements giving higher combustion rates and reduced cyclic dispersion.

A computer simulation model of the compression, combustion and expansion phases of the engine cycle was developed to predict the effects of the combustion chamber design parameters. Based on the experimental results, the model assumes that the ratio of laminar to turbulent burning velocity is independent of chamber design. The influence of chamber shape on the burnt volume, flame front surface area and heat transfer surface areas is modelled using a simple but effective geometric integration technique. This technique allows an infinite variation of the design parameters to be specified for a large range of chamber shapes with a minimum of input data being required.

The model predicts that chamber design does have a major effect on combustion rate and cylinder pressure but shows that the influence of individual design parameters is highly dependent on the setting of all other parameters. The effect of squish area is shown to be due to it changing the compactness of the chamber, optimum squish area being about 50% for conventional engines with higher areas being suited to higher compression ratio designs. Spark plug arrangement is predicted to be the most effective way of controlling the combustion
rate with a single centrally located spark plug or alternatively, dual spark plugs, giving large increases in combustion rate.

Computer model predictions have been compared directly with experimental results obtained in this study and with experimental results reported by two other independent workers. Good agreement was obtained thereby giving support to the assumption of the flame speed being unaffected by chamber design.

The model was also used to predict squish velocities in fired engines. The results show that the velocities and, in particular the reverse squish, can be significantly modified by the combustion process with a strong dependence on ignition timing being evident.

The predictive model has been modified to yield a heat release program capable of analysing experimental pressure time data to predict combustion rate, flame speed, turbulent burning velocity and many other variables. The predicted flame speeds were in good agreement with corresponding experimental values obtained from ionisation probes.

In conclusion, the study has confirmed the importance of combustion chamber design as a means of improving the combustion rate but has shown that the flame speed is not affected by chamber shape (i.e. squish). The semi-empirical simulation model has been shown to predict the effects of the chamber design parameters to an acceptable degree of accuracy.
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NOMENCLATURE

A  Area
b  Chord length
B  Engine bore
BC Bumping clearance
c  Capacitance
C  Constant
CEFF Combustion efficiency factor
CR Compression ratio
CL Connecting rod length
CP Specific heat at constant pressure
CV Specific heat at constant volume
d  Diameter
E  Activation energy
f  Flame arrival frequency
h  Height
hc Heat transfer coefficient
H Enthalpy, Piston to head distance
ΔH Pressure head
HV Valve recess depth
i  Index
k  Constant
K Dissociation (or equilibrium) constant
KT Turbulent multiplication constant
L  Length
M  Mass
M Mass transfer rate
MM Molecular weight
n Compression/expansion index, Number of moles
N  Engine speed
P  Cylinder pressure
\Delta P  Pressure rise rate \left( \frac{dP}{de} \right)
\dot{P}  Maximum cycle pressure
PH  Spark plug penetration
q  Number of moles of products assuming no dissociation
Q  Heat transfer rate
\dot{Q}  Volume flow rate
r  Radius
R  Specific gas constant, Resistance, Radius
R_e  Reynolds number
R_o  Universal gas constant
R_P  Reynolds parameter
RC  Cylinder radius \left( \frac{B}{2} \right)
RES  Residual gas concentration
RF  Flame radius
RR  Bowl/hemisphere radius
RS  Spark plug radial location
RSB  Bowl/hemisphere offset
RSP  Spark plug recess radius
RV  Valve recess radius
S  Stroke
SQ  Squish percentage area
SQA  Squish area
S_T  Stroke
T  time
\dot{t}  time interval
\dot{t}_{str}  time from B.D.C. to T.D.C.
T  Temperature
T'  Isentropic temperature
ΔT  Change in temperature
T_f  Adiabatic flame temperature
U   Velocity Internal energy
ΔU  Change in internal energy
U'  Turbulence intensity
v   Voltage
V   Volume
V_j  Mean intake jet velocity
Ψ   Work transfer rate, Load
x,y,z Distance or displacement along Cartesian co-ordinates
α  Flame contact angle relative to spark plug
β  Flame contact angle relative to cylinder axis
γ  Ratio of specific heats
θ   Crank angle
θ_b  Burn angle
θ_d  Delay angle
θ_p  Maximum pressure crank angle
σ  Dispersion
ϕ   Equivalence ratio
η_v  Volumetric efficiency
ρ   Density
ω   Specific humidity
μ  Absolute viscosity

Subscripts
b   for burnt charge
f   Relating to flame
h   Value for cylinder head
i   i\text{th} estimate
Subscripts (contd)

L  Laminar
n  Ionisation probe number
p  Relating to products or piston
r  Value for reactants
s  Relating to squish, spark plug or ignition
T  Turbulent
w  Value at combustion chamber wall
X  At position X
θ  Corresponding to θ crank angle

Abreviations

A.D.C.  Analogue to digital converter
A.T.D.C.  After top dead centre
B.I.P.  Bowl in piston
B.T.D.C.  Before top dead centre
C.R.O.  Cathode ray oscilloscope
D.G.  Dilute gas concentration
E.G.R.  Exhaust gas recirculation
E.V.C.  Exhaust valve closes
E.V.O.  Exhaust valve opens
F.S.  Flame shape
F.S.R.  Flame speed ratio
h.h.  Hemispherical head
H.U.C.R.  Highest usable compression ratio
I.V.C.  Inlet valve closes
I.V.O.  Inlet valve opens
L.M.L.  Lean misfire limit
M.B.T.  Minimum advance for best torque
S.I.  Spark ignition
T.D.C.  Top dead centre
W.O.T.  Wide open throttle
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CHAPTER 1

INTRODUCTION
CHAPTER 1

1.1 GENERAL

It is well known that the combustion (or mass burning) rate in a spark ignition (S.I.) engine has a major effect on the efficiency, exhaust emissions, smoothness, fuel octane requirements and lean misfire limit (L.M.L). Previous work has shown that optimum combustion rates exist for maximum efficiency (1)* and minimum oxides of nitrogen (2,3) with higher combustion rates increasing engine "roughness" (4,5), the highest useable compression ratio (1,5) and the L.M.L (6).

With present day conventional engines, maximum combustion rates of about the optimum value are relatively easy to obtain. However, in the case of the low emission engines based on either lean mixture operation or high rates of exhaust gas recirculation (E.G.R), the combustion rate may be greatly reduced to below the optimum value. The object of research and development into future engine designs is therefore to find ways of increasing the combustion rate.

For specified conditions (i.e. fuel type, equivalence ratio etc) the combustion rate may be increased by increasing the intake generated charge motion and/or modifying the combustion chamber design. The work detailed in this thesis is concerned with the latter approach.

Previous studies (1,2,3,7) have confirmed that large differences in combustion rate can occur between different combustion chamber designs. However, whilst the importance of chamber design is well accepted, the effect of individual parameters and the interaction between parameters is not very well understood and significant differences in empirical results have been reported. Also, the effect of chamber design on the burning velocity, flame speed and flame front profile have not been established.

* Numbers in parentheses designate References at the end of this thesis (pages 274 to 283).
This lack of detailed knowledge has prevented the effect of chamber design parameters from being predicted using either empirical correlations or theoretical models. Instead, the design of combustion chambers is mainly achieved using a philosophy based on experience and empiricism. Although this design approach has resulted in a steady improvement in engine performance over the years, it does suffer from being very slow and expensive and does not necessarily produce optimum designs. In addition, due to the rapid changes in design trends resulting from the introduction of emission and fuel economy legislation, the value of chamber "design guides" based on previous experience will be limited.

In recent years, a number of engine cycle simulation models have been reported (8, 9, 10, 11, 12), the objective of which is to reduce or eliminate the disadvantages associated with empirical engine development. Although these models have in general achieved some success, no model has been reported to date which can accurately predict the effect of all the chamber design variables. This failure is due mainly to the lack of detailed knowledge of the processes occurring prior to and during the combustion phase.

It is clear therefore that before a model can be produced to predict the effects of chamber design, further detailed measurements of flame speed and combustion rate are required to be obtained from a large range of combustion chambers. Although in the long term, this information could result in a fully predictive model being developed, it is felt that in the short term, empirically based correlations should be incorporated into a semi-empirical model.

The object of this study is to develop such a simulation model based on empirical measurements and to use this model to predict the effects of chamber design parameters. The objectives are formally defined in Section 1.3.

1.2 POTENTIAL EFFECTS OF CHAMBER DESIGN PARAMETERS

From mass continuity across the flame front, the combustion rate, \( \dot{M}_b \), is given by:

\[
\dot{M}_b = U_T \cdot A_f \cdot \rho_b
\]  

- (1.1)
Where \( U_T \) = Turbulent Burning Velocity
\( A_f \) = Flame Front Surface Area
\( \phi_b \) = Burnt Charge Density

The burnt charge density is a dependent variable which will not change significantly with combustion rate and therefore it is better to write:

\[
\dot{\dot{m}}_b \propto U_T \cdot A_f \quad - (1.2)
\]

Equation (1.2) shows that the combustion rate can be increased by increasing the turbulent burning velocity and/or enlarging the flame front area. Therefore, when considering the potential of chamber design as a means of increasing the combustion rate, we must investigate the probability of enhancing these two parameters.

**Turbulent Flame Speed \( U_T \)**

The burning velocity of a fuel/air mixture is normally determined under laminar flow conditions with the measured value being called the laminar burning velocity \( U_L \). Tests have shown that \( U_L \) for a given type of fuel is mainly a function of charge temperature and equivalence ratio.

\[
U_L = f(T_u, T_b, \phi, DG) \quad -(1.3)
\]

Where
- \( T_u \) = Unburnt Charge Temperature
- \( T_b \) = Burnt Charge Temperature
- \( \phi \) = Equivalence Ratio
- \( DG \) = Dilutant Gas Concentration

Burning velocities measured in engines are normally much higher than the corresponding \( U_L \) and this is due to the effect of turbulence which increases the burning velocity. (This phenomenon is described in Chapter 2). Therefore, the turbulent burning velocity \( U_T \) can now be expressed as follows:

\[
U_T = f(U_L, U') \quad -(1.4)
\]
which using equation (1.3) becomes

\[ U_T = f(T_u, T_b, \gamma, DG, U') \]  

\[-(1.5)\]

Where \( U' \) is the turbulence intensity.

**Flame Front Surface Area \( A_f \)**

If the combustion occurred within an infinite volume, the area of the flame front would only be a function of the flame radius and the flame shape. The shape of the flame would be spherical in the absence of spatial variations in charge properties but since spatial variations of turbulence intensity, charge velocity, equivalence ratio and temperature will occur, the flame shape will be distorted to some extent.

Flame propagation in engines is confined by the combustion chamber walls. The effect of this is to limit the area of the flame front and therefore reduce the combustion rate. Since the radius at which the flame contacts the chamber walls will be a function of chamber shape, size and spark plug location, we may write:

\[ A_f = f(\text{Shape}, B, S, CR, RS, X, FS) \]  

\[-(1.6)\]

Where \( B \) = Bore  
\( S \) = Stroke  
\( CR \) = Compression Ratio  
\( RS \) = Spark Plug Location  
\( X \) = Piston Location  
\( FS \) = Flame Shape

Therefore, it can be seen that the flame front area at a given time during the flame propagation period will be a function of many parameters. To simplify equation (1.6), it is better to think of "chamber compactness" which incorporates all of the geometric chamber parameters such that

\[ A_f = f(\text{Compactness}, \text{Flame Shape}) \]  

\[-(1.7)\]
A compact chamber is a design where the maximum flame radius of the main part of the combustion chamber perpendicular to the cylinder axis, divided by the flame radius parallel to the cylinder axis, is close to unity. An illustration of chamber compactness is shown in Fig. 1.1 where it can be seen that although a compact chamber may produce increased surface area over most of the flame propagation range, reduced surface area may result towards the latter stages of the range.

Equations (1.5) and (1.7) indicate the potential effects of chamber design. To investigate these in more detail, individual chamber design parameters are examined below.

**Compression Ratio**

Compression Ratio can affect the combustion rate by changing both the turbulent burning velocity and the flame front area. Increased compression ratio for example will increase the charge temperature and reduce the residual gas concentration, both of which will increase the laminar burning velocity. The effect on the flame front area will be more complex in that it will be dependent on the chamber shape but generally, increased compression ratio will reduce flame front area. However, this will be offset by some increase in charge density such that the effect of compression ratio is almost entirely due to the influence on burning velocity.

Although increased compression ratio will increase the combustion rate, the value adopted will be determined by the occurrence of detonation or knock. Increased compression ratio offers the advantage of increased thermal efficiency but has the associated disadvantages of higher HC emissions (13, 14) and increased fuel octane requirements (1). It should be noted that the H.U.C.R.* has been shown to be a function of chamber design (1, 5) and equivalence ratio (5, 15) so that higher compression ratios may be worthy of consideration for lean mixture, fast burn engines.

* Highest Useable Compression Ratio
Spark Plug Arrangement

Spark plug arrangement, which includes spark plug radial and axial locations in addition to the use of multiple spark plugs, will affect the combustion rate through its effect on the flame front area with negligible effect on the burning velocity. It is clear that a spark plug radially positioned in the centre of the chamber will produce larger flame front areas and shorter flame travel distances than a spark plug positioned on the outer edge of the chamber. Similarly, two spark plugs correctly positioned will give greater flame front areas and shorter maximum flame travel distances than those corresponding to a single spark plug. The effect of spark plug arrangement is illustrated in Fig. 1.2.

Although spark plug arrangement would not be expected to have a significant effect on burning velocity, small changes may result as a consequence of the charge temperatures and perhaps turbulence being dependent on the combustion rate. If M.B.T. (Minimum Advance for Best Torque) ignition timing was adopted, changes in burning velocity would occur due to changes in the properties with crank angle.

Also, it should be noted that due to spatial properties variations, spark plug location may produce effects which are not solely a function of geometry. This effect has been shown by Janeway(4).

Bore/Stroke Ratio

Although this ratio is more of an engine design rather than a chamber design parameter, it can have a significant effect on the combustion rate and duration due to its influence on "compactness". For present day engines operating with compression ratios in the region of 9:1, the bore to stroke ratio should be much less than unity for maximum compactness and minimum flame travel distance. The fact that engines have a ratio of about unity is due to other factors such as piston speed, engine height etc. To compensate for this, combustion chambers are made more compact by changing the shape so that the desirability for a small bore/stroke ratio is reduced.
Combustion Chamber Shape

Combustion chamber shape can affect the combustion rate due to changes in both flame front area and turbulent burning velocity. Many chamber shapes have been used and some common shapes are shown in Fig. 1.3.

The effect of chamber shape on flame front area is relatively easy to understand since the shape directly restricts the expansion of the flame front. An example of this effect has already been presented in Fig. 1.1. The effect of chamber shape on the turbulent flame speed is due to its influence on the turbulence intensity rather than any significant affect on the laminar burning velocity. This is due to the production of a piston generated charge motion known as "squish".

Squish is defined as the inward radial velocity produced by the piston as it approaches T.D.C. of the compression stroke. This is illustrated in Fig. 1.4. It is relatively easy to show\(^{(16,17)}\) that the squish velocity is a function of the piston speed and the combustion chamber shape.

Due to the fact that it is difficult to separate the effects on combustion rate of squish turbulence and compactness resulting from modifying the combustion chamber shape, much confusion and disagreement exists at present as to the relative importance of these two potential effects. Whilst it may be argued that this is only of academic interest, this is not the case since the uncertainty prevents design models from being produced. A major part of the experimental study reported in this thesis was aimed at solving this problem.

1.3 Objectives of this Study

The objectives of this study were as follows:

a) To determine experimentally, the effects that combustion chamber design parameters have on the combustion rate and flame propagation. Special attention to be given to the relationship between chamber design and flame speed.
b) To develop a computer simulation model for predicting the effects of the combustion chamber design parameters. The development effort to concentrate mainly on the modelling of chamber geometry and the correlation between burning velocity and chamber design.

c) To use the simulation model to predict the effect of, and the interaction between, combustion chamber design parameters. Having shown the effect of the parameters, to identify improved chamber designs and to illustrate the potential of combustion chamber design as a means of increasing and controlling the combustion rate.

To achieve (a), it was necessary to define some secondary objectives, these being:

i) To identify, procure, modify and develop a suitable test engine and associated test cell instrumentation and hardware.

ii) To design and have manufactured, the range of combustion chamber designs to be investigated.

iii) To design and develop the specialist rig instrumentation (such as that required for flame propagation and pressure measurement).

iv) To design, procure and develop equipment for high speed, multi-channel data acquisition.

v) To develop data reduction/processing hardware and software to analyse the flame propagation and cylinder pressure data.
Fig 1.1  **EFFECT OF CHAMBER DESIGN ON COMPACTNESS**

(a) Off-set Spark Plug

(b) Central Spark Plug

(c) Dual Spark Plugs

Compactness increased by chamber shape and spark plug location design modifications.

Fig 1.2  **EFFECT OF SPARK PLUG ARRANGEMENT ON FLAME FRONT AREA FOR DISC CHAMBER**
Fig 1.3 EXAMPLES OF COMBUSTION CHAMBER SHAPES

Fig 1.4 ILLUSTRATION OF SQUISH VELOCITY
CHAPTER 2

REVIEW OF PREVIOUS WORK
2.1 INTRODUCTION

Since it was first successfully operated over a century ago, the S.I. engine has been the subject of research and development (R & D) aimed at improving all aspects of its operation. With regard to the combustion process and the effect of combustion chamber design, many papers have been published over the last seventy years, the research effort having increased over the past twenty five years mainly due to the effects of increased government legislation and increased competition between manufacturers. Despite this amount of R & D, the combustion processes are still not fully understood, mainly due to the complexity of the chemical and physical processes involved and the difficulty of making detailed measurements in the hostile environment of the combustion chamber.

However, whilst the exact combustion mechanisms are not known, the theories proposed and the empirical measurements do give a reasonably good understanding of the way in which certain parameters affect the combustion rate. Therefore, in Section 2.2, flame propagation theories, models and empirical observations are described. The subjects covered include the ignition process, laminar and turbulent flame propagation theories and correlations and flame propagation in S.I. engine combustion chambers.

In Section 2.3, empirical measurements reporting the effect of combustion chamber design on various parameters are reviewed. The parameters covered include in-cylinder air motion, flame propagation, combustion rate, knock and emissions.

2.2 COMBUSTION IN S.I. ENGINES

The purpose of this section is to briefly review aspects of combustion which are related specifically to the objectives of this work. Therefore, it concentrates mainly on the physical rather than chemical combustion processes since it is the former which largely controls the rate of
combustion in engines using conventional liquid hydrocarbon fuels.

It must be emphasised that combustion in engines is a very extensive topic which can only be covered in the briefest of detail here. For readers wishing to obtain a more detailed description of both the physical and chemical combustion processes, texts such as Refs. 10, 17, 18 and 19 are recommended.

2.2.1 Ignition Process

Ignition of the flammable air fuel mixture in spark ignition engines is under normal conditions achieved by a spark discharge across the electrodes of a sparking plug. This discharge is usually made up of a high intensity short duration capacitive component and a long duration (approximately 1 mS) lower intensity inductive component(7). Although there is disagreement about the relative importance of these two components, it is accepted that the spark provides sufficient energy (in the form of heat and active particles) to initiate the chemical reactions necessary to achieve a self propagating flame. Flame propagation will only occur when the heat released from the reaction exceeds the heat losses to the cylinder walls, spark plug electrodes and unburnt bulk gas(19).

Due to this energy balance requirement, it is clear that there will be a minimum amount of spark energy which will be required to successfully ignite the mixture. This is known as minimum ignition energy and has been shown(19) to be a function of spark plug design, type of spark, air to fuel ratio, fuel composition, residual gas dilution, charge temperature and pressure and mixture motion. For the range of conditions normally encountered in S.I. engines, minimum ignition energies range from about 10 to 100 mJ(17). Increasing the ignition energy above the minimum value has been observed to have negligible effect on the establishment of combustion(20).
2.2.2 Laminar Flame Propagation

The speed at which the flame front propagates relative to the unburnt mixture under laminar flow conditions is the laminar flame velocity $U_L$. This can be measured using a variety of devices such as burner, tube, bomb and nozzle and a number of measurement techniques, details of the equipment and measurements being described in Ref. 19.

Laminar flame velocity measurements (19, 21, 22) show that the velocity is mainly a function of several parameters:

a) **Type of fuel.** For hydrocarbons, high hydrogen to carbon ratios generally produce high $U_L$ although other factors such as type of bond, bond strength etc are also important.

b) **Equivalence Ratio.** The maximum value of $U_L$ occurs at an equivalence ratio in the range 1.0 to 1.8 (depending on the fuel), the velocity reducing more rapidly for lean mixtures than for rich mixtures.

c) **Inert and Residual Gas Dilution.** The addition of inert and/or residual combustion gases will reduce the value of $U_L$ due mainly to reduction in charge temperature although transport properties will also be affected.

d) **Unburnt Charge Temperature.** This parameter has a considerable effect on $U_L$, increasing the initial temperature from about 300K to 450K doubling the velocity as shown in Figure 2.1.

Although several reports have been made of $U_L$ being dependent on mixture pressure (9), the measured dependence is small and within the range of measurement error. It is generally agreed that at pressures greater than atmospheric, the effect of pressure can be ignored (19).

Two limiting mechanisms for flame propagation have been proposed which are helpful in understanding the parameters affecting the propagation rate. The two mechanisms are:

a) the thermal mechanism

b) the diffusion mechanism.
The thermal mechanism is based on the assumption that heat transfer from the reaction zone to the unburnt gas raises the temperature of the unburnt gas sufficiently to cause it to ignite and hence release its chemical energy. The diffusion mechanism on the other hand assumes that active particles and radicals diffuse from the reaction zone into the unburnt gas and cause it to react explosively.

Based on the two mechanisms, a number of laminar flame propagation theories have been proposed. The most popular of these theories have been reviewed by James (10) and Phillipps and Orman (9). These include the diffusion theories of Tanford and Pease and Manson and the thermal theories of Mallard and le Chatelier and Semenov.

It is uncertain to what extent the two mechanisms account for the flame propagation rate in actual combustion processes although it is widely believed that both mechanisms occur (19). Therefore, the choice of flame propagation theory to predict the laminar flame velocity is normally based on how close the predictions agree with empirically measured values and on the complexity of the calculations required. Of all the theories, simplified versions of the Semenov equation are probably the most popular choice due to its reasonably good accuracy and because it avoids the need for dubious quantities such as ignition temperature and reaction zone thickness.

2.2.3 Turbulent Flame Propagation

Turbulence may be defined as a random velocity fluctuation superimposed on the mean fluid motion. The main effects of turbulence on flame propagation (10, 19, 23) are as follows:

a) It considerably increases the flame velocity

b) The reaction zone is modified from the uniform, very thin laminar profile to a much thicker, non uniform complex profile.

c) A reduction in the flammability limits when the intensity of turbulence is increased.
A number of theories have been proposed to account for the effects of turbulence on the flame velocity and some of these have been reviewed by James (10) and De Soete (23). All of these are based on two main theories:-

a) Surface Model or Wrinkled Flame Front.

b) Three Dimensional or Volume.

The former theory considers the behaviour of large scale turbulence on the laminar flame. Due to this, the laminar flame front is thought to undergo fluctuating deformations, thus becoming wrinkled as shown in Figure 2.2. It follows that the surface area of the wrinkled flame front is greater than that of the laminar flame and it is this which increases the flame velocity due to increased heat transfer and diffusion. The reaction zone would have the appearance of being thick due to the displacement corresponding to the erratic fluctuations of the wrinkled front.

Therefore the turbulence burning velocity $U_T$ is expressed by:

$$U_T = \frac{A_T}{A_L} \cdot U_L$$

Where $A_T$ and $A_L$ are the surface areas of the turbulent and laminar flame fronts respectively.

The volume or three dimensional theory is based on the eddy diffusion in turbulent flow increasing the transport properties of the thermal and diffusional mechanisms. This increases both the heat and mass transfer rates mainly due to the effects of turbulence on the heat transfer coefficient and eddy diffusivity. In this theory the reaction zone is considered to be similar to the laminar case but made thicker by the action of increased heat transfer and particle transportation.

It is probable that both the surface and volume mechanisms are responsible for the effects of turbulence on flame propagation although the importance of each mechanism is not known. In the review by James, it is summarised that the surface or wrinkled flame theory is fundamentally unsound and that the volume theory is the most plausible.
A different approach to the classic theories reviewed by James is the model proposed by Blizzard and Keck and later investigated and improved by McCuiston et al. and Tabaczynski et al. It attempts to overcome some of these objections by using fundamental quantities of eddy size and turbulence intensity. The model is based on the wrinkled flame front theory but deviates from other surface theories by assuming finite thickness of the reaction zone. The assumption is that the flame front propagates at a speed proportional to the turbulence intensity, entraining and igniting eddies. Each entrained eddy is then assumed to burn inward at a constant laminar flame speed.

Unfortunately, due to the problems associated with quantifying the turbulence parameters, the surface and volume theories proposed cannot be used at present to predict the turbulent burning velocity. Instead, empirically based expressions are used to predict the turbulent burning velocity. These are of the form

\[ U_T = K_T \cdot U_L \]  

- (2.2)

The dependence on the laminar burning velocity emphasises the surface mechanism. The turbulent multiplication factor, \( K_T \), allows for the many parameters affecting the turbulent flame velocity which are not catered for in the laminar flame velocity expression. The parameters affecting factor \( K_T \) in S.I. engine combustion chambers are discussed in the next sub-section.

2.2.4 Flame Propagation in S.I. Engines

The flame propagation rate in the combustion chamber measured relative to the chamber walls is known as the flame speed \( U_f \) and is the sum of two components, the turbulent burning velocity and the expansion velocity \( U_e \). The former is the actual rate of propagation into the unburnt portion and would be the velocity observed at a reference point situated a small distance into the unburnt charge. The expansion velocity is the rate of expansion of the burnt charge due to the change in density resulting from the combustion.
Flame propagation in the combustion chamber of a spark ignition engine has been the subject of a number of studies. A variety of techniques have been used to determine the flame speed and are briefly as follows:-

a) **Probe Measurement Methods**

i) **Ionisation Probe.** This is the most popular method and has been used by many workers (10, 27, 28, 29, 30). Curry (27) used 49 ionisation probes, installed in the head and piston to obtain three dimensional propagation maps whilst Harrow and Orman (28) used a combustion interval meter with ionisation probes to determine cyclic dispersion and mean flame travel angle.

ii) **Gas Sampling.** This was used by Withrow et al (31) to determine the presence of the flame front by detecting the oxygen concentration.

iii) **Fibre Optic Probe.** This method has been used by Karim and Badr (32) who employed 12 probes in conjunction with two photo-diodes to detect flame motion.

b) **Visual Measurement Methods**

Compared to probe methods, visual methods have the advantage of giving a two dimensional view of the flame front. The major disadvantages are the need to install a window into the combustion chamber walls, cost of equipment, qualitative results due to parallax error and flame curvature and operational problems due to fouling of window and vibration.

i) **Direct Photographic.** This method exploits the increase in light intensity which occurs during and following combustion. Rassweiler and Withrow (33) used black and white photography whilst Nakanishi et al (34) used colour photography. Both of these report that due to poor contrast in light intensity between the burnt and unburnt zones, additions of chemicals, namely sodium and copper oleate respectively, were required to give a clearly defined flame front.
ii) Schlieren Photography. This method has been used by Hamamoto et al(35) in preference to direct photography due to the weak luminosity of the flame at very weak mixtures. An extremely complex set up employing a transparent head and piston crown with an inclined mirror fitted in a slotted piston was used.

c) Pressure-Time Data Analysis Models

Rather than measure flame propagation directly, this method uses a theoretical technique to analyse measured pressure time data and hence obtain burnt volume. By making an assumption regarding the flame profile, the flame radius and speed can then be obtained. Rassweiler and Withrow(33) derived equations for calculating burnt volume and compared the predictions with estimated burnt volumes obtained from direct photography. Tidmarsh(36) and Mattavi et al(37) employed detailed computer programs to obtain flame speed predictions based on spherical propagation. The major error in these models is likely to be the assumptions made regarding the flame profile relative to the spark plug electrodes whilst the main restriction to their use is the need to accurately define the shape of the combustion chamber at all crank angles.

The techniques described above have been used to determine the way in which flames propagate in combustion chambers. These studies have shown that the flame speed ratio (FSR) is not constant both during propagation and from one cycle to the next.

The variation during combustion has been investigated by Lancaster et al(38). They concluded that combustion can be divided into four periods; ignition and kernel development, flame development, fully developed propagation and flame termination. The variation during combustion and the four periods are shown in Figure 2.3. The variation in flame velocity and FSR as a function of flame radius have been determined by Mattavi et al(37). A typical plot is shown in Figure 2.4. From Figures 2.3 and 2.4 it can be seen that the flame speed varies as a function of both time and position.
The variation from one cycle to the next is known as cyclic dispersion or cyclic variation. Harrow and Orman have shown that significant variations in flame travel angle to ionisation probes occur, the variation increasing with parameters which increase the mean flame travel angle. Patterson has studied cyclic dispersion of cylinder pressure and has concluded that mixture velocity variation near to the spark plug at the time of ignition is mainly responsible although poor fuel distribution also has an effect.

The flame profile has received attention by a number of workers. Curry obtained three dimensional plots using multiple ionisation probes and showed that spherical flame assumptions are reasonable for cases of low charge swirl although variations in turbulence and surface temperature caused some distortion. In the case of high swirl, generated by a 180° inlet valve mask, very large distortions were measured, this being shown in Figure 2.5. Photographs of combustion by a number of workers confirm the flame profile observations reported by Curry.

Due to the variations in flame velocity and flame shape described above, it is only possible to obtain approximate relationships between FSR and turbulence parameters or engine operating conditions by using mean values. Based on this approach, Lancaster et al conclude that the FSR is a linear function of turbulence intensity as shown in Figure 2.6. This relationship is of the form

\[ \text{FSR} = 1 + k \cdot u' \]  \hspace{1cm} -(2.3)

A number of workers have shown a linear dependence of flame speed ratio on engine speed, i.e.

\[ \text{FSR} = 1 + K_T \cdot N \]  \hspace{1cm} -(2.4)

Where \( N \) is the engine speed in RPM

This is in good agreement with the results of Lancaster et al since anemometry measurements indicate an approximately linear dependence of turbulence intensity on engine speed.
The value of $K_T$ in equation 2.4 has been determined by various workers using different engine designs. Typical examples are as follows:

<table>
<thead>
<tr>
<th>Reference</th>
<th>$K_T$ (eq. 2.4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>James (10)</td>
<td>0.00197</td>
</tr>
<tr>
<td>Phillipps and Orman (9)</td>
<td>0.002</td>
</tr>
<tr>
<td>Hodgetts (41)</td>
<td>0.0017</td>
</tr>
<tr>
<td>Tidmarsh (36)</td>
<td>0.0013 - 0.0016 (function of chamber shape)</td>
</tr>
</tbody>
</table>

An alternative approach suggested by Spalding (42) is to correlate FSR and Reynolds Number, $R_e$ which he defined as:

$$R_e = \frac{e \cdot \sigma^2}{t_{str} \tau}$$ \hspace{1cm} -(2.5)

Where $B$ is the engine bore and $t_{str}$ is the time for the piston to travel from B.D.C. to T.D.C.

Phillipps and Orman (9) show that based on the results of Bollinger and Williams, a relationship of the following form is obtained:

$$FSR = 1 + 0.002 \cdot R_e$$ \hspace{1cm} -(2.6)

Samaga and Murthy (11) used the following expression:

$$FSR = (1 + 2786 \cdot \left[ \frac{R_p}{U_L} \right]^2 )^{1/2}$$ \hspace{1cm} -(2.7)

Where $R_p = \text{Reynolds Parameter} = V_j \cdot e \cdot B / T^{0.67}$

$V_j = \text{Mean Intake Jet Velocity}$

$T = \text{Mean gas temperature}$

They suggest that although $R_p$ is based on intake velocity, allowances for variable piston generated motions such as squish could be incorporated.
2.3 THE EFFECT OF COMBUSTION CHAMBER DESIGN

The purpose of this section is to review the results of previous workers who have measured the effects of combustion chamber design on parameters such as charge motion, flame speed, combustion rate, knock and exhaust gas emissions. These findings can then either be compared with results from this study or taken into account when considering the overall effects of the design parameters. A further reason for the review is that it indicates areas where poor agreement exists between workers and/or areas where further work is required to establish the effect of chamber design.

It should be pointed out that some results reported are dubious due to the fact that engine design features which are outside of the area of study, but which are known to significantly affect the parameters being measured, are not maintained constant. The most obvious example is where one engine is compared with a completely different engine (of different manufacture) and the results attributed solely to chamber design differences as in Ref. 43. A further example is where one cylinder head is compared with a second design as in Ref. 37. The results are again attributed solely to chamber shape even though the inlet tract and valve have been extensively modified and the spark plug position changed. Many workers have established for example that inlet tract and valve design(3, 27) and carburation and ignition system design(44) can have major effects on the engine performance parameters and therefore these must be taken into account. Changing the relative position of the spark plug can affect the parameters due to spatial variations of turbulence (45, 46), charge composition (47) and temperature(18) within the chamber.

2.3.1 Charge Motion in Motored Engines

Alcock and Scott(48) used flow visualization to investigate the charge motion in a bowl in piston compression ignition engine. They found little evidence of squish motion although a reverse squish motion was much more evident. They suggest that the squish velocity was much lower than expected due to piston ring blowby and heat transfer effects.
James\(^{(49)}\) measured the air motion in a disc and a squish type chamber using a hot wire anemometry system. He concluded that the turbulence was higher in the squish chamber than in the disc chamber although there was negligible difference at frequencies greater than 1 KHz.

Anton\(^{(40)}\) used a similar arrangement to James with squish areas ranging from zero (disc) up to 20\%. He found negligible differences between the chambers regarding turbulence intensity. Eddy frequency and size increased with squish area whilst mean charge velocity actually decreased with squish areas of 15\% and 20\%, apparently due to interaction between swirl and squish.

Witze\(^{(45)}\) noted that the turbulence produced by a concentrated squish area was apparently insignificant when compared to the compression stroke enhancement of the intake generated turbulence. On the other hand, Dent and Derham\(^{(50)}\) found that the measured squish and swirl velocities were in good agreement with the theoretical predictions. Tidmarsh\(^{(36)}\) studied the flow in a number of combustion chambers and found large variations in fluctuating velocity with chamber configuration.

A different approach was used by Ghirlando\(^{(51)}\). He used an engine with no valves to measure the squish velocity in the absence of intake generated motion. Comparing a bowl in piston with a disc chamber design, he found that the squish velocities were much lower than predicted although better agreement was found as the bumping clearance was increased. Reverse squish was higher than forward squish. He thought that the measured values were lower than expected due to high piston ring blowby caused by the large piston crown to top piston ring distance caused by a piston crown extension.

The effect of compression ratio has been investigated by Semenov\(^{(52)}\) who showed that the velocity at TDC was slightly reduced with increased compression ratio whilst Molchanov et al\(^{(53)}\) found that the effect was negligible.

In summary, it is clear that there is no agreement concerning the effect of combustion chamber shape. However, it is generally agreed that squish velocity is small compared to intake generated motion.
and that the velocity is lower than predicted by theory whilst reverse squish exceeds the forward squish. It is probable that the differences reported are due to the interaction between the intake generated motion and squish although other factors such as the differences in detailed chamber designs and the spatial variation of charge motion will also have an effect.

2.3.2 Flame Speed

Compared to work on charge motion and combustion rate, the effect of chamber design on flame speed has received very little attention from an empirical measurement point of view. Nagayama et al(30) measured flame speed using two ionisation probes and compared four combustion chambers combining both squish area and swirl. They found that both squish and swirl increased the flame speed and that maximum speeds were achieved when the squish and swirl were combined. It is interesting to note that squish was effective in increasing the flame speed during both the initial and main combustion phases.

Mattavi et al(37) compared the flame speed ratio for wedge and open combustion chambers. The burning velocity was determined by using a computer model to analyse the measured pressure data. They found that although the measured turbulence intensity was greatest for the wedge chamber, the FSR was actually highest for the open chamber during the first 20 mm of flame travel. However, the FSR for the remainder of the propagation was highest for the wedge chamber. Possible reasons for the higher burning velocity in the open chamber during the first 20 mm were investigated but it was concluded by the authors that no consistent explanation could be offered. The theory that the turbulence was too high in the wedge chamber during the flame development phase was rejected since the FSR increased consistently with increased engine speed.

Tidmarsh(36) correlated flame velocity function, determined from pressure data analysis, with flow velocity, the latter determined from hot wire anemometer measurements. He found that although the flow velocity for a number of chambers varied by a factor of 2, the flame velocity function varied by only 20%. Furthermore, the best fit curve
drawn by Tidmarsh shows that whilst the flame velocity increases initially with flow velocity, a reduction occurs for higher flow velocities. The theory that flame speed can reduce at high turbulence levels is supported by De Soete\textsuperscript{(23)}. However, it should be noted that the squish chambers tested by Tidmarsh were very typical of production engines and would not be expected to produce extremely high levels of turbulence.

The results of Tidmarsh are shown in Figure 2.7. Taking into account errors which can occur in both flame velocity (via pressure analysis) and flow velocity (via hot wire anemometry) measurements, a constant flame velocity fit might have been a more realistic conclusion. This would then suggest that the flame velocity is not greatly affected by the chamber configuration.

To conclude on the three papers reviewed in this sub-section, it is evident that very little agreement exists. Further work is clearly required to establish the effect of combustion chamber shape on flame propagation.

### 2.3.3 Combustion Rate

The pioneering work of Ricardo\textsuperscript{(1)} demonstrated the importance of combustion chamber design as a means of controlling combustion rate. He showed that by modifying the cylinder head design of L head side valve engines as shown in Fig. 2.8, substantial reductions in combustion duration and octane requirement can be achieved. Ricardo firmly believed that these improvements were due mainly to the effects of turbulent air motion generated by the squish design and named it the turbulent head design. Since then, squish and open designs have been referred to as turbulent and quiescent even though these classifications do not agree with the results of air motion measurements.

The empirical findings of Ricardo were analysed theoretically by Janeway\textsuperscript{(4)} and Taub\textsuperscript{(54)} who both agreed that geometric parameters were probably more responsible than the piston generated turbulence for the improved performance. These authors described the potential effects of spark plug position and chamber shape and presented guide lines for controlling the combustion rate and thereby achieving
efficient and smooth engine operation. Ricardo has presented results showing that for maximum efficiency, the maximum rate of pressure rise should be about 2.3 Bar/deg (35 p.s.i./deg) whilst Janeway has shown that the gradient of the pressure rise rate curve, i.e. $\frac{d^2p}{d\theta^2}$ should be kept to a minimum for smooth operation. Based on these requirements, Taub suggests that the chamber design should be such as to promote the maximum flame area as possible during the first third of the flame travel, thereby reducing the "delay" period. The flame front area should then decrease steadily so that "roughness" can be avoided. Taub produced a volume distribution curve to aid designers and described an empirical method for determining burnt volume versus flame radius curves.

Obert(7) illustrates the use of the Taub method by analysing two combustion chambers which were known to produce "rough" and "smooth" combustion. The graphs produced are shown in Figure 2.9 and show that the "rough" design has a peak area at about 60% flame travel instead of the 30% value for the "smooth" chamber.

Andon and Marks(55) suggest a graphical method of plotting flame front area against flame travel (similar to but not the same as the method of Taub) comparing combustion chamber performance. They claim that the pressure rate difference between chambers can be predicted to within 10% although details of the calculation method are not given. They also show that reducing the cyclic dispersion reduces the subjective engine noise emissions and improves efficiency.

Mantey(56) compared the performance of zero squish hemispherical and wedge combustion chambers and concluded that the wedge was the best choice because of it having much smoother performance. The maximum rate of pressure rise for the "turbulent" wedge was about 1.9 Bar/deg compared to 2.8 Bar/deg for the "non-turbulent" hemispherical head design. Burt et al(43) compared the performance characteristics of three different engines using bowl in piston, bathtub and disc combustion chamber designs. They conclude that the bathtub chamber gave the best overall performance although the possible effect of other engine design differences were not considered.
Tanuma et al (44) compared disc, wedge, pan-cake (zero squish domed piston) and heron (bowl in piston) chambers and found that the heron and pan-cake chambers produced the highest combustion rates and the smoothest operation at weak mixtures. Specific fuel consumption and exhaust gas temperatures were also lower. The wedge chamber had the worst performance of all four chambers in all cases, although it should be pointed out that the spark plug location for the wedge chamber was not ideally positioned. The results made the authors conclude that "the squish area did not reveal a significant influence of the squish effect" and that for good lean mixture operation, the volume should be compact around the spark plug.

Mayo (3) studied the effects of, and the interaction between, intake generated swirl and velocity, squish area and spark plug location. The results obtained for ignition delay time (time for first 10% mass burn), burn time (time for 10% to 90% mass burn) and cyclic dispersion indicate that the interaction between parameters is large. For example, although 30% squish area reduced the burn time by 23% in one case, a reduction in the intake generated swirl rate produced a 24% increase for the same comparison.

It was concluded that the single factor resulting in the largest reduction in burn time in the absence of the other factors was the spark plug location. Increasing the compression ratio from 8.3 to 10.9:1 on an open chamber reduced the delay time by 22% but did not significantly affect the burn time.

Several papers have been published showing the effects of using dual spark plugs. Diggs (57) has presented results showing that dual spark plugs can give a considerable reduction in burn time whilst Quader (58) has confirmed that both spark plug location and dual spark plugs have a large effect on combustion duration and the lean limit. Oblander et al (98) show that a reduction in combustion duration and cyclic dispersion can be obtained with dual spark plugs.

Nakajima et al (59) compares the results from a compact wedge chamber with a "fast burn" engine of a zero squish, hemispherical head, dual ignition design. The results show that the dual ignition design is superior to the squish design on combustion rate, cyclic
dispersion and EGR (Exhaust Gas Recirculation) tolerance.

To conclude on the papers reviewed in this sub-section, it is clear that combustion chamber design can have a very large effect on combustion rate, thereby affecting the smoothness, efficiency and lean mixture operation of the engine. The interaction between parameters means that it is not possible to propose optimum values of chamber shape, spark plug location or number of spark plugs. The most important parameter is that of compactness where for high burn rates, the chamber area should be concentrated around the spark plug(s). Of all the design parameters, spark plug arrangement appears to be the most important with the squish turbulence effect being much less significant.

2.3.4 Knock

Knock is the term used to describe the condition where part of the cylinder charge, the so called end gas, reacts simultaneously rather than progressively. Although it is not known whether this is due to spontaneous ignition or a rapid acceleration of the flame front, knock must be avoided since it can lead to premature engine failure and results in reduced engine efficiency and "rough" operation.

It is well known that combustion chamber design has a major effect on the octane requirement of an engine at fixed compression ratio. Ricardo(1) demonstrated that reducing the volume of the end gas by using a squish design gave large improvements. Ricardo assumed that this was mainly due to the cooling of the end gas resulting from the heat transfer from the small mass of end gas to the cool cylinder walls, the heat transfer being enhanced by squish turbulence. Janeway(4) agreed with Ricardo and likened the end gas cooling to that of a cup (i.e. combustion chamber) of hot fluid being transferred to a saucer (i.e. squish area). He concluded that squish areas of up to 20% were effective in reducing knock and that bumping clearance reduction also reduced knock.

Caris et al(5) showed that combustion chambers having high combustion rates also have reduced octane requirements. Their results indicate that the octane rating can be reduced by either reducing
combustion duration (i.e. end gas exposure time) or by quenching the end gas.

Andon and Marks\(^{(55)}\) point out that increased cyclic dispersion will increase the octane requirement due to the more advanced M.B.T. ignition timing. Adams and Cantwell\(^{(60)}\) emphasise the point that it is the fast burn cycles which produce the knock but agree that reducing the cyclic dispersion will reduce octane requirement. Dye\(^{(61)}\) has put this theory into practice and operated an engine at a compression ratio of 15:1 using 98 RON fuel. He achieved this by using an inlet valve turbulence generator which reduces the combustion duration and cyclic dispersion.

The reduction in octane requirement with reduced combustion duration has been confirmed by the use of dual ignition systems. Digge\(^{(57)}\) found that dual ignition reduced octane requirement but that the improvement was less when deposits were present. Oblander et al\(^{(98)}\) also confirms that the tendency to knock is reduced with dual ignition.

Caris et al\(^{(5)}\) have shown that the octane requirement is greatly affected by equivalence ratio. The maximum value occurs slightly rich of stoichiometric and reduces significantly at weak mixtures. Bolt and Holkeboer\(^{(15)}\) have pointed out the advantages of operating lean burn engines at high compression ratios. May\(^{(62)}\) has combined the effects of lean mixture operation and fast burn rate in a high compression ratio, high squish area design. The 14.6:1 compression ratio "fireball" chamber has a "boost channel" which is reported to convert the squish into a swirling motion in the combustion space located below the exhaust valve.

To conclude on the subject of knock, there is not doubt that combustion chamber shape and spark plug arrangement do affect the octane requirement. The relative importance of end gas cooling, combustion duration and cyclic dispersion are not clear. The use of high compression ratio for lean burn designs should increase the combustion rate and hence improve the lean mixture performance.
2.3.5 Exhaust Emissions

The three main pollutants generated in the combustion process of an S.I. engine are carbon monoxide (CO), oxides of nitrogen (NO\textsubscript{x}) and unburnt hydrocarbons (HC). The CO emissions are mainly dependent on equivalence ratio and will not be considered here. The NO\textsubscript{x} emissions\textsuperscript{(63)} are a result of dissociation and are reduced by reducing the gas temperature, the time of exposure and the amount of excess oxygen. The HC emissions\textsuperscript{(64, 65, 66)} are mainly a result of the formation of a quench layer around the cylinder walls, some of which is entrained during the exhaust process.

For a given equivalence ratio, increasing the burn rate will increase the NO\textsubscript{x} emissions due to the increased temperature effect being more dominant than the reduced exposure time. Thring\textsuperscript{(2)} showed that reducing the combustion duration by the use of multiple spark plugs caused increases of up to 113% in NO\textsubscript{x}. Increased NO\textsubscript{x} with faster combustion has been observed by other workers\textsuperscript{(3, 64)} and implies that compact combustion chamber designs will have increased NO\textsubscript{x} emissions.

However, spark plug position, apart from modifying the combustion rate, is also known\textsuperscript{(2, 67)} to affect NO\textsubscript{x} emissions due to the spatial variation in charge temperature. The temperature of the charge burnt first may be 300 to 400\textdegree{}C\textsuperscript{(18)} hotter than the last portion of charge to burn and therefore large spatial variations in NO\textsubscript{x} occur. Due to the exhaust valve being much hotter than the remainder of the chamber wall, locating the spark plug near to the exhaust valve will increase the NO\textsubscript{x} emissions.

The opposite effect has been shown for HC emissions. Varde and Lucas\textsuperscript{(67)} have shown that minimum hydrocarbons are produced when the spark plug is closest to the exhaust valve because of the reduced quench effect at higher temperature. Dodd\textsuperscript{(65)} has shown that the quench layer near to the exhaust valve is more readily entrained and therefore makes a more significant contribution to the emissions than the quench layer around the inlet valve.
Since HC emissions are a function of combustion chamber surface area/volume ratio \( (66) \), increased squish area and compression ratio will both tend to increase the HC emissions. This effect has been found by a number of workers \( (3, 14, 59) \) although the difference between chambers is usually relatively small when operated at MBT ignition timing.

Many authors have shown \( (2, 14, 59, 98) \) that by operating fast burn engines at optimum equivalence ratio and EGR rate, the emissions of NO\(_x\) and HC are less than for slower burn engines. Therefore, in general, combustion chamber design parameters which increase the combustion rate will increase the emissions at constant operating conditions and reduce the emissions at optimised operating conditions.
Fig 2.1 EFFECT OF TEMPERATURE ON BURNING VELOCITY

Fig 2.2 WRINKLED TURBULENT FLAME IDEALISATION
**Fig 2.3 FLAME SPEED RATIO VS FLAME RADIUS**

- (1) Ignition and Kernel Growth
- (2) Flame Development
- (3) Fully Developed Propagation
- (4) Termination

From Ref No (38)

**Fig 2.4 FLAME SPEED RATIO VS FLAME RADIUS FOR DIFFERENT ENGINE SPEEDS**

From Ref No (37)
Fig 2.5 FLAME PROPAGATION PATTERNS SHOWING EFFECT OF SHROUDED INTAKE VALVE

Fig 2.6 EFFECT OF TURBULENCE INTENSITY ON FLAME SPEED RATIO

From Curry (27)

From Lancaster et al (38)
Fig 2.7 CORRELATION BETWEEN FLAME VELOCITY AND CHARGE TURBULENCE

Fig 2.8 RICARDO COMBUSTION CHAMBER DESIGN
Fig 29 VOLUME-DISTRIBUTION CURVES FOR TWO COMBUSTION CHAMBERS. OBERT (7)
CHAPTER 3

EXPERIMENTAL DETAILS
CHAPTER 3

3.1 The Engine

A Bermoter single cylinder air cooled 4 stroke engine was used for all the experimental work involved in this study. The engine, originally compression ignition, was converted to spark ignition operation by replacing the fuel injector with a spark plug, reducing the compression ratio and adding an electronic ignition system and petrol carburettor. The details of the modified engine are given in Table 3.1

<table>
<thead>
<tr>
<th>TYPE</th>
<th>BERMOTER W21</th>
</tr>
</thead>
<tbody>
<tr>
<td>bore mm</td>
<td>70</td>
</tr>
<tr>
<td>stroke mm</td>
<td>70</td>
</tr>
<tr>
<td>con-rod length mm</td>
<td>120</td>
</tr>
<tr>
<td>swept volume cc</td>
<td>268</td>
</tr>
<tr>
<td>i.v.o.</td>
<td>12° b.t.d.c.</td>
</tr>
<tr>
<td>i.v.c.</td>
<td>42° a.b.d.c.</td>
</tr>
<tr>
<td>e.v.o.</td>
<td>42° b.b.d.c.</td>
</tr>
<tr>
<td>e.v.c.</td>
<td>12° a.t.d.c.</td>
</tr>
<tr>
<td>fuel system</td>
<td>1½&quot; SJ CD CARBURETTOR</td>
</tr>
<tr>
<td>ignition system</td>
<td>CONTACTORLESS INDUCTIVE</td>
</tr>
</tbody>
</table>

This engine was chosen in preference to the others available mainly due to the fact that both the piston crown and cylinder head face were approximately flat and parallel to each other. This feature was desirable since it readily allowed the shape of the combustion chamber to be changed by sandwiching a suitable spacer plate between the cylinder head and block. Furthermore, it resulted in a similar configuration to that of a motored engine being used in another study in the department for the measurement of charge motion (40). Similarity was important since it was hoped that a correlation between the fired and motored results would be obtained.

This design arrangement greatly simplified the interpretation of the results since only the shape of the chamber is being changed, all other design features such as the inlet port and valve, the spark plug
location and the measurement probes remaining constant. This is considered to be of major importance when experimentally investigating the effects of combustion chamber design. (The reasons for this have been discussed in Section 2.3).

Another consideration was the fact that the engine was of a single cylinder air cooled design, reducing the amount of manufacturing and assembly work required for each chamber design. A single cylinder engine also allowed direct measurements of brake power, air/fuel flow rates, exhaust temperature and emissions etc to be made for the same cylinder as that used for flame position and pressure measurements.

The engine was mounted on a test bed alongside a Heenan and Froude Hydraulic Dynamometer which was used to load the engine and allow measurement of the power output. A photograph of the engine test rig is shown in Fig. 3.1.

3.2 Combustion Chamber Details

A large number of combustion chamber designs were tested and for convenience, these may be split up into 5 groups:

a) Disc (Cylindrical)
b) Concentrated Squish (1)
c) Bowl in Piston
d) Concentrated Squish (2)
e) "May Fireball"

The above groupings represent both the type of chamber design and the chronological order in which the test programme was executed. A total of 21 chamber designs were tested, 18 of these being different chamber shapes and the remainder being variations in the spark plug position. The full range of designs tested in this work are listed in Table 3.2.

a) Disc Chambers

This type of design was obtained by fitting cylindrical spacer plates between the head and block, the inside diameter being the same as the bore. Two spacers were manufactured in different thicknesses to give compression ratios of 8:1 and 10:1. The basic arrangement is shown in Fig. 3.2.
<table>
<thead>
<tr>
<th>REF No.</th>
<th>CHAMBER TYPE</th>
<th>SQUISH COVERAGE %</th>
<th>BUMPING CLEARANCE mm</th>
<th>COMPRESSION RATIO</th>
<th>IGNITION LOCATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>DISC</td>
<td>0</td>
<td>-</td>
<td>8.0</td>
<td>NORMAL</td>
</tr>
<tr>
<td>2</td>
<td>&quot;</td>
<td>0</td>
<td>-</td>
<td>8.0</td>
<td>CENTRAL</td>
</tr>
<tr>
<td>3</td>
<td>&quot;</td>
<td>0</td>
<td>-</td>
<td>10.0</td>
<td>NORMAL</td>
</tr>
<tr>
<td>4</td>
<td>TYPE 2</td>
<td>5</td>
<td>0.8</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>5</td>
<td>&quot;</td>
<td>10</td>
<td>0.8</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>6</td>
<td>&quot;</td>
<td>15</td>
<td>0.8</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>7</td>
<td>&quot;</td>
<td>20</td>
<td>0.8</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>8</td>
<td>&quot;</td>
<td>15</td>
<td>1.65</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>9</td>
<td>&quot;</td>
<td>15</td>
<td>0.4</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>10</td>
<td>TYPE 3</td>
<td>5</td>
<td>0.8</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>11</td>
<td>&quot;</td>
<td>10</td>
<td>0.8</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>12</td>
<td>TYPE 4</td>
<td>5</td>
<td>0.8</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>13</td>
<td>&quot;</td>
<td>10</td>
<td>0.8</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>14</td>
<td>&quot;</td>
<td>15</td>
<td>0.8</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>15</td>
<td>TYPE 5</td>
<td>5</td>
<td>0.8</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>16</td>
<td>BOWL IN PISTON</td>
<td>47.3</td>
<td>0.7</td>
<td>8.0</td>
<td>&quot;</td>
</tr>
<tr>
<td>17</td>
<td>CONCENTRATED (2)</td>
<td>48.0</td>
<td>0.8</td>
<td>8.0</td>
<td>As Fig. 3.7</td>
</tr>
<tr>
<td>18</td>
<td>&quot;FIREBALL&quot;</td>
<td>0.8</td>
<td>10.0</td>
<td>No. 1</td>
<td></td>
</tr>
<tr>
<td>19</td>
<td>&quot;</td>
<td>0.8</td>
<td>10.0</td>
<td>No. 3</td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>&quot;</td>
<td>0.8</td>
<td>10.0</td>
<td>No. 1 &amp; 3</td>
<td></td>
</tr>
<tr>
<td>21</td>
<td>&quot;</td>
<td>0.8</td>
<td>12.6</td>
<td>No. 1</td>
<td></td>
</tr>
</tbody>
</table>

**TABLE 3.2 RANGE OF COMBUSTION CHAMBER DESIGNS TESTED**
The majority of the tests using spacer plates were conducted using a spark plug located in place of the original fuel injector. This will be referred to as the "normal" spark plug location and was 19 mm from the centre of the chamber as shown in Fig. 3.2. In addition, a "central" spark plug location was tested for the 8:1 compression ratio version of the two disc chambers, this spark plug being made from a modified ionisation probe design.

b) Concentrated Squish (1)

A total of 12 of these squish designs were tested. The squish areas were arranged in either a single or double sector resulting in 4 basic designs, designated types 2 to 5 (type 1 being the disc chamber) and are shown in Figs. 3.3 and 3.4.

The squish areas were incorporated into squish plates, the thickness of the plates being varied with squish area to maintain a constant compression ratio of 8:1 for all of these tests. The bumping clearance (piston crown to squish face at TDC) was maintained constant at 0.8 mm for all the tests except two. In these cases, the bumping clearance was adjusted to 1.65 and 0.4 mm for the 15% type 2 design only. The full range of concentrated squish plates tested are shown in Table 3.3.

<table>
<thead>
<tr>
<th>TYPE (See Fig. 3.3)</th>
<th>% SQUISH AREA</th>
<th>BUMPING CLEARANCE mm</th>
<th>THEORETICAL SQUISH VELOCITY @ 1000 RPM &amp; 350 © (M/S)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>5, 10, 15, 20</td>
<td>0.8</td>
<td>1.37, 2.2, 2.92, 3.59</td>
</tr>
<tr>
<td>2</td>
<td>15</td>
<td>0.4, 1.65</td>
<td>4.175, 1.69</td>
</tr>
<tr>
<td>3</td>
<td>5, 10</td>
<td>0.8</td>
<td>1.37, 2.2</td>
</tr>
<tr>
<td>4</td>
<td>5, 10, 15,</td>
<td>0.8</td>
<td>2.2, 3.59, 4.83</td>
</tr>
<tr>
<td>5</td>
<td>5</td>
<td>0.8</td>
<td>2.2</td>
</tr>
</tbody>
</table>

TABLE 3.3 CONCENTRATED SQUISH DESIGNS

The % squish area is defined as the squish area/bore area and will be used throughout this thesis when referring to a chamber design, e.g. 10% type 2.
The theoretical squish velocities given in Table 3.3 were calculated using the expression derived in Appendix IV.

It should be noted that this group of chamber designs were limited to a maximum of 15% squish in one area or 20% squish in two areas. Also, the majority of the designs had the squish area adjacent to the spark plug rather than on the far side of the chamber. The reason for this was that the position of valves and spark plug (see Fig. 3.2) limited the maximum amount and position of the squish that could be incorporated.

c) **Bowl in Piston**

A single bowl in piston design was tested using the same cylinder head and spark plug location as for the previous designs (see Fig. 3.2). The bowl was obtained by machining out a standard Bermoter piston, which in its original form had a small dia. bowl that was plugged up for the flat piston designs. The general arrangement of the bowl in piston design is shown in Fig. 3.5 and 3.6.

The bowl diameter was 50.8 mm giving a squish coverage of 47.3%. The compression ratio was maintained at 8:1 and the bumping clearance was reduced slightly from the previous value to 0.7 mm (due to a machining error) giving a theoretical squish velocity of 5.87 m/s at 10° BTDC and 1000 RPM.

d) **Concentrated Squish (2)**

The previously mentioned concentrated squish designs were limited to relatively small squish areas positioned adjacent to the spark plug. Although this arrangement offered a number of advantages from a research point of view, it was not completely representative of production engines which would normally have larger squish areas opposite the spark plug.

Although such a design was therefore required to be tested, it was clearly not possible to achieve this using the same arrangement as used for the previous concentrated squish area designs. The final arrangement used is shown in Figs. 3.7 and 3.8. This single design still utilised the original cylinder head as used before thereby allowing direct comparisons with the previous results to be made.
The original spark plug was blanked off and a new 12 mm spark plug was installed into the side of the squish plate adjacent to the valves. The squish coverage of 48% was made up of 8% on the valve side and 40% on the opposite side to the spark plug, both of these being in the form of sectors. The compression ratio was again fixed at 8:1 with a bumping clearance of 0.8 mm.

e) May "Fireball"

The final chamber designs tested were broadly based on the "Fireball" design of Michael May reported in Ref. 68. This was considered an important addition to the range since it is a high squish area, high compression ratio engine which has been reported (62, 68) to give good performance with lean mixtures.

The fireball chamber was achieved by producing a cast aluminium insert which fitted inside a second modified Bermoter cylinder head. This design is shown in Figs. 3.9 and 3.10. Two of these designs were tested with compression ratios of 10:1 and 12.4:1. Bumping clearance was maintained at 0.8 mm.

Water cooling was employed by machining channels and casting passages in the head and insert. This feature was considered necessary to avoid excessive wall temperatures causing knock and pre-ignition at the high compression ratios. Another useful design attribute was the provision of three spark plug access positions. These allowed for the investigation of the effects of spark plug position and multiple spark plug and also gave access for measurements such as pressure, flame arrival, charge velocity etc.

A simple water coolant circuit was used which utilised a small centrifugal pump to circulate water through the cylinder head. The temperature was controlled by manually adjusting the flow rate of cold mains water into the coolant circuit resulting in a similar amount of hot water over-flowing from a header tank to waste.
3.3 Instrumentation

The main parameters to be measured during the experimental work were the cylinder pressure and the flame propagation rate. In addition, exhaust temperature and emissions, brake load, air flow rate, fuel flow rate, engine speed, and ignition timing were also required to be determined.

All the variables except pressure and flame propagation were continuously displayed and manually noted during the testing. This was possible since the time average values were required and therefore only one reading for each variable per test condition was required. In the case of pressure and flame propagation, measured using ionisation probes, the time average values were not suitable and either ensemble averaged or instantaneous values were required. Since cyclic dispersion is an important parameter in this work, instantaneous values were required in most cases.

A further requirement was that all instantaneous pressure and ionisation data be recorded simultaneously. This was necessary to allow accurate flame profiles and speed to be determined and to permit a comparison between measured flame arrivals and calculated values based on the measured pressure analysis.

It was decided that the best method for dealing with the pressure and ionisation data was to record this information on a multi-channel tape recorder together with crank angle marker pulses. Apart from allowing instantaneous values of a number of channels to be recorded simultaneously, it permits a significant reduction in engine running time due to the multi-channel capability and because of the fact that data reduction is performed after the tests. The reduction in running time results in increased reliability, improved control over the test conditions and reduced demand for laboratory facilities.

The instrumentation used is shown in Figs. 3.11 and 3.12. A brief description of the individual items of equipment is given below.

The cylinder pressure was measured using a Kistler Type 6121 piezo-electric transducer, flush mounted in the cylinder head, together with a Kistler type 566 charge amplifier. The 6121 was chosen because of its compactness, ability to operate at combustion chamber temperatures without water cooling and its very good
overall specifications. The transducer and charge amplifier were calibrated prior to starting the experimental test programme, details of the calibration are given in Appendix III.

The flame propagation parameters were determined by using a number of ionisation probes in conjunction with an ionisation amplifier. The number of probes used at any one time varied from a minimum of 2 to a maximum of 9. The ionisation instrumentation details are given in Appendix II whilst Fig. 3.2 and 3.9 show the ion probe locations.

A Sangamo 3500, 14 channel F.M. tape recorder was used to record the pressure, ionisation and crank angle marker pulses. A recording speed of 60 i.p.s. was used throughout the tests and the maximum input voltage was not allowed to exceed 1.5 volts.

Crank angle markers were produced at 2° intervals by using a slotted disc in conjunction with 2 optical switches. A 200 mm dia. disc with 180 slots equally spaced around the circumference to produce the interval markers was sandwiched between the flywheel and cooling fan disc. A further single slot to produce an absolute position marker pulse was positioned on the same disc but at a slightly greater radius than that used for the 2° slots.

The slotted switches, manufactured by R.S. Components, comprised of a light emitting diode and photo-transistor aligned and mounted in a plastic moulding. Two of these, one for the 2° interval markers and one for the absolute position marker, were fixed to the inside of the cooling fan casing. They were carefully positioned such that the interval markers corresponded to even numbers of crankangle degrees (e.g. T.D.C., 2°, 4° etc) and the absolute position marker pulse occurred at 159° B.T.D.C. The slotted disc/switch arrangement is shown diagramatically in Fig. 3.13.

The system described resulted in one absolute position marker per engine revolution whilst only one pulse per cycle (i.e. every 2 revs.) was required. This requirement was achieved by electronically gating out the pulse occurring in the exhaust stroke with a pulse derived from the electronic ignition slotted switch fitted to the camshaft. This arrangement was considered to be an improvement over using a pulse derived directly from the camshaft since in the former case, the pulse can be positioned more accurately.
A few runs were conducted using pulses of 1° interval instead of the normal 2° value. This was achieved using a pulse shaper which produced a second pulse at an adjustable delay period after the first pulse.

The air flow rate was determined with a viscous flowmeter in conjunction with an inclined manometer. This type of flowmeter was used since it is not significantly affected by the cyclic flow of a single cylinder engine. The flowmeter was calibrated at the start of the work and details of the calibration are given in Appendix 1.

The fuel flow rate was measured using a 50 ml pipette and 3 way valve arrangement in conjunction with automatic electronic timing. This piece of apparatus is described in detail in Ref. 69.

The engine speed was obtained by using an inductive probe mounted in close proximity to a 60 tooth wheel fitted to the dynamometer shaft. The resulting frequency in Hz therefore corresponded to the engine speed in RPM and was indicated on a digital display.

The brake load was determined using a spring balance and weight arrangement attached to the load arm on the Heenan and Froude Dynanometer. Brake power was then given by:

\[ \text{Brake Power} = \frac{W \cdot N}{6032} \text{ kW} \]

Where \[ W = \text{Brake Load in lbf} \]
\[ N = \text{Engine Speed in RPM} \]

The ignition timing was manually adjusted to the desired setting by rotating an optical slotted switch relative to the camshaft timing disc. A timing light illuminated crank angle marks engraved on the flywheel. The timing light was disconnected after each adjustment to avoid electrical interference to the instruments.

The exhaust emissions were analysed to determine the concentrations of CO, CO\(_2\), NO and unburned hydrocarbons (as Hexane) in the exhaust using 4 non-dispersive infra-red gas analysers manufactured by The Analytical Development Company Ltd. The exhaust sample, taken continuously from a tapping approximately 100 mm downstream of the exhaust valve was cooled, dried and filtered before being fed to the analysers. Span gases with a known concentration of the relevant gas were used to calibrate the analysers.
The exhaust temperature was measured using a Chromel/Alumel thermocouple in conjunction with a Comark Type 1601 electronic thermometer. The thermocouple was installed in to the exhaust pipe approximately 100 mm downstream of the exhaust valve.

3.4 Data Processing Equipment

The pressure and ionisation data were recorded on a magnetic tape recorder together with crank angle marker pulses as detailed in Section 3.3. Due to the large amounts of data produced, it was decided to use a digital computer to reduce and analyse the data.

It was therefore necessary to convert the analogue pressure and ionisation signals into digital form. This operation was performed using a Hewlett Packard (HP) 2100 A Digital Computer with a HP5465 Analogue to Digital Converter (A.D.C.) and two HP 7970E Digital Magnetic Tape Units. A HP high speed paper tape punch and reader and a Teletype Terminal were also used to input and output data.

An important feature of this A.D.C. is that the digitising location and rate may be controlled by the use of external trigger and external clock pulses respectively. Thus, in this work, the crank angle absolute position marker was used as the external trigger to enable the A.D.C. to operate and the crank angle interval marker pulses were used as the external clock to set the digitising rate. A block diagram of the digitising/processing system is shown in Fig. 3.14.

Once started by the external trigger pulse, the analogue signal was digitised at every external pulse until the specified block size was reached, at which point the A.D.C. was inhibited. For this work, a block size of 256 was used.

Therefore, it can be seen that since the external trigger pulse occurs at 159\(^\circ\) BTDC and the external clock pulses occur every 2\(^\circ\), the digitising will start at 158\(^\circ\) BTDC and continue every 2\(^\circ\) until 8\(^\circ\) BTDC on the exhaust stroke. The part of the cycle not digitised consisted almost entirely of the intake stroke, which was not of interest for this study.

The data thus digitised were stored on one of the digital magnetic tape units in blocks of 256. Each value in a block corresponded to a precise crank angle location and each block corresponded to a precise
engine cycle of a test run. These data were then either processed directly by the HP computer or output on either magnetic or paper tape to be processed by the University's I.C.L. 1904S computer.

3.5 Range of Operating Conditions Studied

Each of the chamber designs listed in Table 3.2 was tested over a range of engine speed, ignition timing and equivalence ratio. Early work on the concentrated squish designs was conducted at full (wide open) throttle conditions but the remainder of the tests were performed with a constant volumetric efficiency of 75%. This was done to ensure that the chamber designs were tested at as near identical conditions as possible so that comparison of the results would be simplified. In addition, a few tests were made over a range of throttle settings for several of the designs.

To allow the large range of operating conditions to be achieved whilst maintaining the experimental work within a practical time scale, each of the parameters was varied in turn whilst all other variables were fixed at a specified "datum" condition. The values of the datum conditions and the range over which the variables were tested are shown in Table 3.4.

<table>
<thead>
<tr>
<th>PARAMETER BEING VARIED</th>
<th>PARAMETER RANGE TESTED</th>
<th>VALUE OF OTHER PARAMETERS</th>
</tr>
</thead>
<tbody>
<tr>
<td>ENGINE SPEED (RPM)</td>
<td>1500 - 3000</td>
<td>-</td>
</tr>
<tr>
<td>IGN TIMING DEG BTDC</td>
<td>M.B.T. ±20°</td>
<td>2000</td>
</tr>
<tr>
<td>AIR/FUEL RATIO</td>
<td>10:1 TO L.M.L.</td>
<td>2000</td>
</tr>
</tbody>
</table>

TABLE 3.4 RANGE OF OPERATING CONDITIONS TESTED

* The 30° ignition timing was used for the 8:1 compression ratio tests only and values of 20° and 15° were used for the 10:1 and 12.4:1 compression ratio design respectively. A few tests were also done at M.B.T. timing.
A fixed rather than M.B.T. timing was used for the majority of the tests due to the fact that flame speed and hence combustion rate are greatly affected by ignition timing. If the ignition timing is not maintained constant, the measured changes in flame speed and pressure may be due more to the ignition change than due to the parameter being studied. For example, if M.B.T. timing is used when testing a combustion chamber with initially "central" ignition and later "side" ignition, the former would require less ignition advance than the latter and as such would almost certainly result in increased measured flame speeds. If on the other hand a fixed ignition timing had been employed, the change in flame speeds would have been quite different. This is a very important point which is apparently often neglected by researchers.

A further objection to testing at M.B.T. timing is that it is extremely difficult to determine exactly the optimum ignition setting. Apart from being a time consuming operation, it tends to result in increased scatter of measured ionisation and pressure data. M.B.T. timing is useful when comparing brake power related parameters such as power output and efficiency since production engines will operate at about this setting.

An air to fuel ratio of 13:1 was chosen for the datum condition since the parameters being measured are fairly insensitive to air to fuel ratio errors at this setting. Due to the instrumentation used for determining the air and fuel flow rates and the type of fuel system employed, errors in the true air to fuel ratio must be expected and allowed for. A more accurate exhaust gas analysis method would have been preferred but the necessary equipment was not available.

3.6 Experimental Procedures

Prior to testing each design, the spark plug gap and valve clearance were adjusted to 0.7 mm and 0.25 mm respectively and the air flow manometer and dynamometer spring balance were zeroed. The exhaust gas analysis equipment, having been switched on with the other equipment at least several hours before testing, was calibrated using the span gases and fresh ice, a new filter and if necessary fresh silica gel were provided.
All of the engine tests were made under "clean" chamber or minimum carbon deposit conditions since the presence of carbon deposits is known to greatly affect the measured flame speeds \( (28,70) \). The effect of carbon deposits was confirmed during this work when results obtained from a combustion chamber used for the instrumentation development work (run for approx. 100 hours) were compared with the same chamber soon after having the deposits removed.

Harrow and Orman \( (28) \) suggest that the mechanism through which deposits increase the flame speed is either due to increased turbulence resulting from increased surface roughness or a surface-catalysed preflame reaction whilst Downs \( (70) \) presents results which suggest that the deposits accelerate the end gas reactions. A further effect of deposits when using ionisation probes is to cause a change in probe impedance which may affect the response time (See Appendix II).

Whatever the reason for the carbon deposit effect, it is clearly desirable in the interests of accuracy to test under similar conditions (i.e. either "clean" or "dirty"). For this work, the following procedure was adopted to ensure minimum deposits:

a) Before each rebuild, the combustion chamber and ionisation probes were cleaned to remove the majority of carbon deposits.

b) The engine was run on slightly weak mixtures for about one hour prior to the start of testing to allow time for warm up and setting up/checking of instrumentation.

c) The testing time was reduced to about 90 minutes by using the multi-channel tape recorder system.

d) An identical test procedure was adopted for all the work and excessively rich mixtures were avoided or tested last.

During the tests, direct readings were made of fuel flow rate, air flow rate, brake load, engine speed, exhaust temperature and emissions. Air to fuel ratio and volumetric efficiency were determined during the tests to allow these to be adjusted to the desired values.

The other data obtained from the tests were recorded on the tape recorder and consisted of cylinder pressure, ionisation (up to 9 channels) and crank angle markers. Some special procedures were found to be necessary to ensure good accuracy and these are outlined below.
Prior to recording the ionisation data for each test condition, the amplifier input and output signals were checked using a Cathode Ray Oscilloscope (C.R.O). The ionisation supply voltage was adjusted to give an average input signal of 4Vpk to pk under all operating conditions to ensure approximately constant response time (See Appendix II).

The cylinder pressure was displayed continuously on a C.R.O. during the tests and prior to recording, the signal was checked for d.c. drift and if necessary the charge amplifier was grounded to give an acceptable signal (within about 0.1 V). Signal drift during the tests occurred at a slow rate and normally only one or two adjustments were necessary during the test since a correction for d.c. errors was made at the data analysis stage (See Section 4.3).

The crank angle marker pulses were continuously displayed on a C.R.O, during the test run and these were checked before each recording to ensure that no pulses were missing. It was found that no problems were encountered providing the pulse shaper was set up to produce a satisfactory output at the maximum frequency (i.e. 3000 RPM engine speed).

The tape recorder tape path components (i.e. record/erase head, idlers etc) were periodically cleaned using iso-alcohol. The accuracy of signal reproduction was checked periodically for all the channels being used and adjustments were made when necessary to give acceptable results. An exact reproduction was not required for the ionisation and crank angle marker signals since these swing rapidly between logic states.

An accurate determination of the pressure signal amplitude was required however and to obtain this a set of calibration signals of known voltages were recorded on the pressure channel at the start of each magnetic tape during the engine warm up period. This procedure is outlined in Appendix I and has the advantage of giving accurate signal reproduction (using a simple transfer function) even in the event of component or calibration setting changes which would probably occur over a period of time.

The data were recorded at 60 inches per second (i.p.s), each test occupying 100 ft (20 seconds record time/test condition) of the 3500 ft long tape, this corresponding to 500 engine cycles at 3000 RPM. Approximately 5 ft of tape was left unrecorded between the recordings so that the start and finish of a test run could easily be identified during playback.
The data on the tape recorder were later digitised on the A.D.C. system described in Section 3.4. Due to the maximum digitising frequency, each channel was digitised separately using a tape speed of 7½ i.p.s., i.e. a speed reduction of 8. A block size of 256 was used for the majority of the results with a sample size of 99 blocks (or engine cycles) being chosen since this allowed the data to be stored in blocks of 100 (1 block being used to write the block qualifier or calibration data) and represented a good compromise between accuracy and computer storage/processing time.

The information on each of the channels was digitised over approximately the same portion of the tape, starting the A.D.C. at about the same tape position counter reading each time. This would be expected to give slight variations in the exact engine cycles digitised which is not important if comparing ensemble averaged values.

In a few cases however, several ionisation and pressure values for identical individual cycles were required. This requirement was achieved by grounding the crank angle absolute position marker pulse during the first several seconds of recording and then on digitising, the A.D.C. was unable to start until the first trigger pulse was received. Since this was the same pulse each time a channel of data was digitised, each block number of each channel was for the identical engine cycle.

The data thus digitised were stored on a magnetic tape, each tape having a capacity of about 1.5 million data values equivalent to about 55 test runs of 100 cycles for a single data channel.

The ionisation data were generally ensemble averaged on the HP 2100 A computer and the results of interest were outputted either on paper tape (to be processed by the I.C.L. 1904 S computer) or on the terminal to be manually processed.

The pressure data were always (except for checking purposes) processed on the I.C.L. computer using the raw data for the individual engine cycles so that cyclic dispersion could be analysed. This information as stored on the A.D.C. throughput file was not I.C.L. compatible and was therefore translated onto a second magnetic tape loaded on the other digital tape deck.
This tape was then taken to the University Computer Centre where the data were re-formatted and stored on a I.C.L. data file. The information on this file was then available for the analysis procedure which is described in the following chapter.
Fig 3.1 GENERAL VIEW OF THE EXPERIMENTAL TEST RIG
FIG 3.2 COMBUSTION CHAMBER ARRANGEMENT
FULL SIZE

SPARK PLUG LOCATION

SQUISH AREAS

TYPE 2

TYPE 3

TYPE 4

TYPE 5

FOR DETAILS OF COMBUSTION CHAMBER ARRANGEMENT SEE FIG 3.2

FOR DETAILS OF RANGE OF DESIGNS TESTED SEE TABLE 3.2

FIG 3.3 SQUISH PLATE CONFIGURATIONS
Fig 3.4 PHOTOGRAPH OF TWO SQUIRH PLATES SHOWING 20% TYPE 2 AND 15% TYPE 4 (RIGHT)
Fig 3.5 PLAN VIEW OF PISTON USED FOR THE 47% SQUISH AREA B.I.P DESIGN
FIG 3.6  BOWL IN PISTON DESIGN

FIG 3.7  SQUISH PLATE(2) DESIGN (48% SECTOR)
Fig 3.8 PLAN VIEW OF 48% SQUISH AREA
TYPE 3 COMBUSTION CHAMBER
COOLANT OUTLETS

ION PROBES

MAIN COMBUSTION CHAMBER

3 SPARK PLUG LOCATIONS

CAST COOLANT PASSAGES

SCALE - FULL SIZE
MAT'L - CAST ALUMINIUM

STEEL VALVE SEAT INSERTS

SECTIONAL ELEVATION ON X-X

FIG 3-9 ‘FIREBALL’ TYPE COMBUSTION CHAMBER INSERT
Fig 3.10  CLOSE UP VIEW OF THE MAY 'FIREBALL' COMBUSTION CHAMBER
FIG 3.11 BLOCK DIAGRAM OF TEST BED INSTRUMENTATION
Fig 3.12 PHOTOGRAPH SHOWING ELECTRONIC INSTRUMENTATION USED DURING THE EXPERIMENTAL STUDY
FIG 3·13  SLOTTED SWITCH ARRANGEMENT

FIG 3·14  BLOCK DIAGRAM OF DATA PROCESSING SYSTEM
CHAPTER 4

ANALYSIS OF EXPERIMENTAL DATA
4.1 Introduction

The data obtained from the engine tests may be considered to fall into one of two categories. First, steady state or time independent measurements which include power output, exhaust temperature and exhaust emissions. These variables required only a single reading per test condition and derived parameters such as brake thermal efficiency are easily obtained using simple standard expressions. Details of the analysis of this type of data are given in most text books on the subject (e.g. refs. 1, 69) and therefore no details will be given here.

This chapter deals exclusively with the analysis of the second data group involving the instantaneous or time dependent parameters which covers the ionisation and pressure measurements. The time dependence of these parameters is due to both variations in the signal amplitude during the engine cycle and a variation between cycles due to cyclic dispersion (defined in Section 2.2). This resulted in large quantities of data and substantial data analysis was required to yield the information needed.

The majority of the data analysis for both the ionisation and pressure data was performed to obtain "mean" values based on a sample size of 99 consecutive engine cycles. The results obtained from this type of approach were representative of an equivalent dispersion free engine cycle and allowed comparisons to be made more readily between test conditions.

In addition, several test conditions were analysed using data from individual cycles. In this case, the variations in flame propagation and pressure development, both during combustion and from one cycle to the next were studied.

The analysis of the ionisation data to obtain flame propagation parameters was performed using a technique which is described in the next section. The pressure data was converted to absolute pressure values using a method described in Section 4.3 and these results were
then further analysed using the same program to give values of burnt mass fraction. In addition, selected samples of the pressure values were analysed using a heat release computer model, developed as part of this work, to yield burnt mass fraction, flame propagation rates and other information. Details of this program are given in Section 4.4 whilst a comparison between burnt mass fraction results and the effect of data errors are examined in Section 4.5.

4.2 Analysis of Ionisation Results

4.2.1 General Description

The ionisation data analysis system used for this work was different from that employed by previous workers (reviewed in Section 2.2) and offered several major advantages such as:

a) Consecutive cycle analysis
b) Multiple channel recording
c) Individual and multiple cycle analysis
d) Good accuracy
e) Greatly reduced engine running time
f) Computer data reduction, analysis, graphical output etc
g) Minimal rig hardware required

The method used was based on the F.M. Tape Recorder/H.P. ADC system described in the previous chapter. The tape recorder gave many of the above advantages and allowed the simultaneous recording of up to 9 ionisation channels, cylinder pressure signal and 2 crank angle marker pulse signals. (Although an on-line computer would have been preferable, the very high data generation rate (up to 90,000/second) was prohibitive). The H.P. ADC allowed the data to be digitised at exact crank angle locations (rather than at preset time intervals) and permitted rapid data manipulation.

The ionisation signals recorded were basically square waves, the +ve going edge corresponding to flame arrival. Fig. 4.1 shows a typical
ionisation signal, together with the crank angle marker pulse signals. The important features to note about the ionisation signal are the low and high voltage levels of approximately 50 mV and 1.0 V respectively, the pulse length of about 20 mS and the rapid change in signal from logic levels 0 (50 mV) to 1 (1.0 V) with negligible overshoot or transient. The 20 mS long pulse is equivalent to 360° at 3000 RPM or 180° at 1500 RPM. The pulse shape is only really important when determining the mean flame arrival angles and will be discussed further in Section 4.2.3.

The ionisation signal was digitised at either 1° or 2° intervals, the former value being used for the single cycle analysis only. Since the digitising process was initiated for each cycle by the absolute crank angle marker pulse, occurring at 159° BTDC, the digitised data could be identified with exact crank angle locations. These data were stored in blocks containing the data from one cycle and therefore each value could be identified with a specific engine cycle crank angle and data channel.

Having digitised the ionisation signals, the data were then analysed to determine the important flame propagation parameters. These parameters, together with their definitions are as follows:

- **Flame Arrival Angle** \( \theta_n \): This is defined as the crank angle at which ionisation is first detected at probe number n. For individual cycles, this is the point at which the ionisation signal changes state but for mean cycles, it is defined as the crank angle at which there is a 50% probability that a flame will have arrived at the probe. (See Section 4.2.3)

- **Flame Travel Angle** \( \theta_{n1,n2} \): This is the crank angle required by the flame to propagate between points 1 and 2 and is defined by

\[
\theta_{n1,n2} = \theta_{n2} - \theta_{n1}
\]

- (4.1)

Note that \( n1 \) may refer to a probe or an active spark plug. For example, \( \theta_{s,1} \) would refer to the flame travel angle between ignition (spark) and probe 1.
Flame Travel Time \( t_{n1,n2} \) This is simply the flame travel time angle expressed in terms of time rather than crank angle, i.e.

\[
t_{n1,n2} = \frac{\theta_{n1,n2}}{6 \cdot N}
\]

Where \( N = \) Engine speed in RPM

Flame Speed \( U_{n1,n2} \) This is the average speed at which the flame propagates spherically between points \( n1 \) and \( n2 \) and is defined by:

\[
U_{n1,n2} = \frac{x_{n1,n2}}{t_{n1,n2}} = \frac{x_{n1,n2} \cdot 6 \cdot N}{\theta_{n1,n2}}
\]

Where \( x_{n1,n2} \) is the spherical distance (origin at spark plug electrodes) between points \( n1 \) and \( n2 \).

Flame Dispersion \( \sigma_n \). This is only applicable to mean cycle analysis and is defined as the crank angle difference between 80% and 20% probabilities of flame arrivals. (See Section 4.2.3)

4.2.2 Individual Cycle Analysis

Individual engine cycles were studied for 2 main reasons, namely:

a) To provide data for the heat release computer program (described in Section 4.4) so that the accuracy of the model in predicting flame propagation could be determined for both individual and mean cycles.

b) To examine the cyclic dispersion in detail. In particular, the variation with consecutive cycles and the variation during the propagation across the combustion chamber.

It is clear from Fig. 4.1 that if the values at 1° intervals are printed out, the flame arrival angle can be determined by locating the point at which the signal changes from logic state 0 to 1. For example, consider the following values.
### Table 1

<table>
<thead>
<tr>
<th>CRANK ANGLE (Deg)</th>
<th>364</th>
<th>365</th>
<th>366</th>
<th>367</th>
<th>368</th>
<th>369</th>
<th>370</th>
</tr>
</thead>
<tbody>
<tr>
<td>AMPLITUDE (mV)</td>
<td>35</td>
<td>36</td>
<td>35</td>
<td>997</td>
<td>997</td>
<td>996</td>
<td>998</td>
</tr>
</tbody>
</table>

The flame arrival angle is clearly between $367^\circ$ and $368^\circ$ although the exact value cannot be determined using this method. (This resolution limitation does not occur for the multiple cycle analysis described later upon which the majority of the work was based). However, the $1^\circ$ resolution was considered acceptable for this work, being relatively small compared with both cyclic dispersion and flame travel angles. For the results given in Chapter 5, flame arrival was taken to be the crank angle which corresponded to the last "low". Therefore, for the data shown, a flame arrival angle of $366^\circ$ ($3^\circ$ ATDC) would have been assumed.

Although it is possible to determine the single cycle flame arrival angles by manually examining the data in the region of interest, this would have been very time consuming for the several thousand cycles examined. Therefore, the data was transferred to the I.C.L. computer for processing. A simple program read the data for each cycle and located the point at which the voltage exceeded a specified value (arbitrarily set at 300 mV). The preceding crank angle and cycle number were then printed and the process continued.

The analysis procedure used for the ionisation data is shown schematically in Fig. 4.2 for both the individual cycle and multiple cycle analysis. The latter is described below.

#### 4.2.3 Multiple Cycle Analysis

The majority of the data were processed to obtain parameters based on a mean engine cycle obtained by averaging 99 consecutive cycles. Mean values are necessary for comparing the performance at different operating conditions and engine designs.

To obtain the mean flame arrival angle and flame dispersion, it is necessary to plot a cumulative flame arrival frequency graph as shown in Fig. 4.3. Having constructed this type of graph, the required
values can be read off as indicated.

One method which has been used (28) to obtain the flame arrival frequency plot is to employ a Flame Frequency Counter as described in Appendix II. A second method is to determine the flame arrival angles for a number of consecutive cycles and to use these values to obtain the graph.

Clearly the latter method would have been suitable for this work since the flame arrival angles for individual cycles could be determined using the method described in Section 4.2.2. The percentage flame arrivals at each crank angle location could then be calculated and plotted.

However, since the object was to obtain a cumulative frequency plot, this can be obtained directly by ensemble averaging the "square wave" ionisation signals. The effect of ensemble averaging several dispersed square wave signals is illustrated in Fig. 4.4. This procedure would clearly give the same results as determining individual cycle values providing that the ionisation signal satisfies the following criteria:

a) the voltages for logic level 0 and 1 are constant with negligible overshoot and instantaneous response between logic states.

b) signals changing to logic level 1 must remain at this level for the total period of interest.

The reason for using the ionisation signal shown in Fig. 4.1 should now be apparent. The 20 mS long pulse (any value between about 10 and 30 mS would have been satisfactory) was necessary to ensure condition (b). Also, although the tape recorder caused a slight distortion of the signal, this was found to be negligible and therefore condition (a) was also satisfied.

The averaging technique has several advantages over calculating individual cycle flame arrival angles. First, ensemble averaging on the H.P. computer can be performed rapidly (approx. 30 S for 99 cycles) and requires only a simple keyboard program. Also, having obtained the
averaged signal, only a few values per test were required to be outputted and hence there was no need to transfer large amounts of data to the I.C.L. for processing.

The system as described gives the flame frequency curve in terms of voltage and since no attempt was made to preset the low and high values to exact values, the zero and 100% values would correspond to about 50 mV and 1.0 V respectively. (A pulse shaper to give voltage levels of zero and 1.0 V was considered but was not used for this work). Therefore to obtain the mean flame arrival angle (50%) and flame dispersion (80% - 20%), 3 alternatives were possible:-

a) To plot voltages and scale the graphs
b) To plot voltages and scale the 20%, 50% and 80% points
c) To convert all voltages into percentage values.

Of these three alternatives, the latter was used since it could be achieved easily and it simplified graph plotting, interpolation and presentation of the results. This was achieved by outputting the ensemble averaged data over the range of interest (about 30 values) on paper tape which was then used as data for a simple computer program run on the I.C.L. computer. The percentage flame arrival frequency $f$ was calculated using

$$f = \left( \frac{V_{100} - V_{0}}{V_{100} - V_{0}} \right) \times 100 \%$$

where $V_0$ = Voltage level for zero flame arrivals
$V_{100}$ = Voltage level for 100% flame arrivals
$V_\theta$ = Voltage level at $\theta$ crank angle.

Having calculated the percentage flame arrival values at $2^\circ$ intervals over the range of interest, the next step was to construct the flame frequency plot. (Note that it was not generally possible to calculate the 20, 50 and 80% points by interpolation to the desired degree of accuracy). Ideally, this operation should have been performed by the computer but unfortunately this was not practical due to the scatter in the data. A best fit technique could not be employed due to insufficient data points in most cases.
Therefore, the data was plotted on graph paper by hand and a best fit curve was drawn through the data points. In a few cases, the data were plotted by the computer and the curve was drawn later by hand. The mean flame arrival angle and flame dispersion were then read off the graphs as indicated by Fig. 3.3. It should be noted that the crank angle resolution limitation discussed for single cycle analysis does not occur for the multiple cycle analysis.

In summary, the mean cycle ionisation analysis procedure was as follows:

a) Ensemble average 99 consecutive ionisation signals
b) Output approximately 30 data values of interest on paper tape.
c) Convert the voltages into percentages
d) Plot flame frequency curves
e) Read off the mean flame arrival angle and flame dispersion values.

4.3 Cylinder Pressure

4.3.1 General

The cylinder pressure signals were digitised and then transferred to the University's I.C.L. 1904S computer for processing as detailed in Section 3.6. These data were then analysed to determine a number of parameters which can be used to indicate the combustion rate. These computations involved:

a) Calculation of pressure parameters such as ensemble averaged pressure diagrams, maximum pressures for individual cycles, cyclic dispersion and maximum pressure rise rates. These values were calculated from the voltages read from the magnetic tape files with corrections being made to allow for both sensitivity and zero errors.
b) Determination of the burnt mass fraction and associated variables. These were calculated from the pressure time
diagrams using an expression derived from an approximate thermodynamic analysis.

c) The pressure time diagram was used as data for a "heat release" combustion simulation program which calculated burnt mass fraction, flame speed, turbulent burning velocity etc.

(a) and (b) above were achieved using a single program which is described below whilst the heat release program of (c) was run separately using selected pressure data. This is described in Section 4.4.

4.3.2 Pressure/Mass Burnt Program Description

A flow chart showing the major operations in the pressure/mass burnt computer program, called PRESS, is shown in Fig. 4.5 and a listing of the same program is given in Appendix VIII. Both individual and multiple cycle analysis were performed using similar procedures. A brief description of the more important features of the analysis is as follows:

The pressure data voltages for 99 consecutive cycles were read off the magnetic tape files and ensemble averaged (multiple cycle analysis only). The maximum pressure voltage and the corresponding crank angle were also located during the averaging process. These values were then converted into relative pressures (in Bars) by multiplying by the calibration constants for the pressure transducer/data acquisition systems (see Appendix I for details of calibrations).

At this stage, the results were in terms of relative pressures and hence it was necessary to convert them into absolute values (the need for this correction together with some effects of pressure errors are discussed in Appendix III). Several methods were available for the accurate determination of absolute pressures and these are reviewed in Appendix III. However, these methods were all rejected due to either increased instrumentation requirements, increased data generation, cylinder head space limitations or poor accuracy compared to the actual method used.

Instead of these experimental methods, a numerical technique was used which provides good accuracy whilst avoiding the problems
listed above. This technique is described below.

**Absolute Pressure Correction Method**

The pressure data absolute pressure correction technique was based on the fact that the value of the compression index should be approximately constant during the compression process \(^{(71, 72)}\). If the pressure values are in error however due to a zero error, the calculated value of index is no longer constant with large errors occurring where the pressure values are small. The calculated variation in \( n \) for a \( \pm 0.2 \) Bar zero pressure error is illustrated in Fig. 4.6 for typical engine operating conditions and this shows that the changes in \( n \) can be substantial during the early part of the compression process.

The method employed in this work exploits this dependence, adjusting the pressure until the calculated value of \( n \) is acceptable. Basically, 2 approaches could be used:

a) Assume a value of \( n \) based either on theoretical considerations or previously published results, e.g. Refs. 33, 72, and adjust the pressure until this value is obtained. A small error between the assumed and "true" values of \( n \) would probably exist but providing the calculations are carried out for the early part of the process, the final error in pressure should be small.

b) Calculate the index for 2 points in the process and adjust the pressure until the 2 values agree. This approach has the advantage that an exact value of \( n \) need not be assumed.

For the program being described, the former method was used to ensure that the data were of the right order whilst the latter method was used to achieve a more accurate value. The computational procedure used to achieve the absolute pressure values was as follows:
1) Obtain a reasonable estimation of the absolute pressure by assuming that the ensemble averaged cylinder pressure at the end of the exhaust stroke was equal to 1.0 Bar absolute. (It was found that this was a typical value and provided a good start point for the subsequent calculations). The difference between the experimental and assumed pressure was called the pressure error and this value was then added to the pressure values throughout the cycle.

2) The value of the compression index was calculated for a crank angle location of about 100° BTDC using a least squares fitting technique applied to 10 consecutive pressure/volume values. (It was found necessary to use this type of approach due to slight scatter in the data).

3) The calculated value of \( n \) was then checked and if outside values of the range 1.24 to 1.38, the values of the pressures were adjusted by a constant amount (increased if \( n_{\text{calc}} \) high) and the calculation procedure repeated until this criterion was satisfied. An iterative process was used with the pressure being changed in increments of 0.08 bar.

4) Having satisfied the conditions of (3) above, the value of \( n \) was then calculated for a second position at about 10° before the ignition crank angle (read in as data) using the same method as before.

5) The 2 values of \( n_{\text{calc}} \) were compared and if the absolute difference was greater than 0.005, the pressures were changed (increased if \( n_{\text{calc}} \) early in the cycle was greater than the later value) using the iterative procedure used in (3). Convergence was ensured by decreasing the pressure increment by a factor of 2 each time the sign of the difference between the 2 values changed.

The accuracy of this method depends on several factors, such as:

a) the assumption that \( n \) is constant. In theory \( n \) will vary during the compression process due to variations in the specific heats and heat transfer rates and due to piston ring blowby. However, these effects are small and results obtained from experimental studies (33, 72) have shown that the index does not change significantly.
b) the accuracy of the pressure signal. Sensitivity errors would have an effect but providing the equipment is accurately calibrated (see Appendix I), the effect of this type of error should be small.

c) the accuracy of the cylinder volume calculations. This would be mainly due to compression ratio or crank angle determination errors. However, volume errors have little effect on the accuracy of the crank angle range used (volume errors mainly apparent near to TDC position (72) and it is doubtful if this would introduce significant errors.

d) the final difference between the calculated indices

In the program, a maximum difference of ± 0.005 was allowed which corresponds to a maximum zero error of approximately 0.01 Bar (0.15 p.s.i.). An error of this magnitude should have negligible effect on the results.

Generally, it was found that this method resulted in calculated values of compression index in the range \( n = 1.32 \pm 0.04 \). This agrees fairly well with the results of other workers (33, 72) and the values predicted by the computer simulation programs described later.

Having calculated the absolute pressures, the peak pressures (2 degree resolution) were determined by adding the pressure error to the previously calculated relative values for the 99 cycles in the sample. The percentage cyclic dispersion based on the peak values for the sample, was then evaluated using the formula suggested by Haile (73)

\[
\sigma_p = \sqrt{\frac{1}{N-1} \sum_{i=1}^{N} (p_i - \bar{p})^2 / \bar{p}^2}
\]  

(4.5)

where \( \sigma_p \) = % cyclic dispersion of the maximum pressure
\( N \) = Number of cycles in sample (99 for this work)
\( p_i \) = Maximum pressure for cycle No \( i \)
\( \bar{p} \) = Sample mean of the maximum pressures, defined as
\[
\bar{p} = \frac{1}{N} \sum_{i=1}^{N} p_i
\]
Mass burnt calculations

Although direct pressure parameters are useful in analysing the combustion process, it is often better to calculate mass burnt variables since these can indicate the combustion rate more clearly. In particular, it is useful to calculate the delay and burn angles which are defined in this work as follows:

\[ \theta_d = \theta_{2\%} - \theta_s \] \hspace{1cm} -(4.6)

\[ \theta_b = \theta_{90\%} - \theta_d \] \hspace{1cm} -(4.7)

where \( \theta_{2\%} \) = Crank angle for 2\% mass burnt  
\( \theta_{90\%} \) = Crank angle for 90\% mass burnt  
\( \theta_s \) = Ignition timing crank angle

It should be noted that these definitions do not agree exactly with those used by previous workers\(^{(3, 25, 74)}\) who have used values of 2-10% and 90-98% in the mass burnt definitions. In the present work, 2\% and 90\% were used since it was found that these points could be located accurately and consistently, on the burnt mass fraction VS crank angle plot. In particular, substantial scatter in the data at burnt mass fractions greater than about 95\% occurred in all cases and would have made the burn angle more dependent on interpretation if a value greater than 95\% had been defined. Also, errors in the determination of the crank angle for complete combustion (discussed later) would make the final several percent subject to some error.

The burnt mass fraction, \( m_b \), is calculated in the program using an approximate analysis method proposed by Harrington\(^{(74)}\). This method basically involves calculating the value of \( P.V^n \) which should be approximately constant before and after combustion but increases during combustion. The burnt mass fraction during combustion is then given by:
Harrington derived the equation given above by using a theoretical analysis based on the first law of thermodynamics and assuming that the cylinder volume, gas properties and heat losses were all constant during the combustion process. Details of this derivation are given in Refs. 25 and 74 and therefore are not given here.

The main source of error in the method is likely to be due to the assumption of constant volume combustion. For fast combustion conditions with MBT ignition timing, the errors due to this should be small since the volume changes near to TDC are small. For slower combustion rates (such as for weak mixtures) and/or for advanced/retarded ignition timing, the errors would be expected to be more significant.

Generally, the errors should be acceptable and are unlikely to be important when comparing designs or investigating trends since the errors should be common to all results. An analysis of typical errors and a comparison between the results from this method and from a heat release combustion simulation model (described in Section 4.4) are given in Section 4.5.

It is also interesting to note that this method should give similar results to those obtained from using the method suggested by Rassweiler and Withrow (33) which was based more on observation than theory. The latter work showed that the burnt mass fraction at any instant was approximately equal to the cumulative percentage increase in pressure due to the combustion alone.
Harrington, and later McCuiston et al. (25), used experimental set-ups to display the variation of the value of $P.V^n$ during the engine cycle on a C.R.O. The value of $n$ was taken as the arithmetic mean of the measured compression and expansion indices and the delay and burn angles were measured directly from the oscilloscope display.

For the work reported in this thesis, the burnt mass fraction was calculated using a computer program to analyse the pressure-time data. This removes the need for additional test bed instrumentation and allows the data to be analysed more accurately and consistently. However, a problem encountered with the program method which does not exist with the visual display of the experimental method is finding a way of clearly defining the value of $P.V^n$ corresponding to 100% burnt mass fraction.

This problem is illustrated in Fig. 4.7 which is a plot of $P.V^n$ and $M_b$ against crank angle for 2 values of $n$ corresponding to typical compression (1.33) and expansion (1.25) indices. The $P.V^n$ graph shows that increasing $n$ greatly increases the value of $P.V^n$ and that the end of combustion is only clearly defined if $n$ is set equal to, or lower than, the expansion index. The burnt mass fraction curves show that there is only a slight difference between the zero to 100% values predicted for the 2 different indices provided the correct value for the end of combustion is obtained. If however an incorrect value of $(P.V^n)_{\text{final}}$ is used, large errors can result as indicated by curve C.

Several methods for ensuring correct and consistent computation of the completion of combustion were tried and evaluated, these involved using different values for $n$ and the use of several end of combustion criteria. It was found that basing $n$ on the calculated value for the expansion did not give very consistent results and therefore a method involving only the compression index value was used for the analysis.

The end of combustion was determined by calculating $P.V^n$, with $n$ being set equal to the compression index less 0.05, and comparing the current and previously calculated values, continuing the process until a decrease in $P.V^n$ was detected. The crank angle at which this
occurred was taken to be the point at which the burnt mass fraction reached 100%. Having determined this crank angle, the values of $P.V^n$ and $M_b$ were then calculated for the range of interest using the compression index value. It was found that this procedure gave very consistent results, this being checked by visual inspection of the calculated $P.V^n$ - crank angle curve.

At the end of the calculations performed on the data for one test run, the results were outputted on the line-printer and also on the graph plotter in certain cases. Additionally, the card punch was used to output the ensemble averaged or individual cycle pressure data to be used by the heat release program described in Section 4.4.

The delay angle and burn angle values, defined by equations 4.6 and 4.7, were obtained from the print-out by interpolating over the $2^\circ$ (or $1^\circ$) interval results either side of the 2% and 90% values. The same procedure was also used to determine the maximum mean cycle pressure, pressure rise rate and the corresponding crank angles. These and other results are given in Chapter 5.

4.4 Heat Release Computer Program

As part of this work, a heat release pressure analysis program (called COMPARED) was developed. This program was capable of analysing the measured pressure - crank angle data to predict flame propagation parameters such as burnt mass fraction, flame radius and flame speeds.

A model of this type is very desirable for combustion analysis since it allows the full range of parameters to be considered, thereby permitting a detailed analysis of the results to be made and at the same time reducing the need for extensive test-bed instrumentation and measurements.

There were several reasons for developing the heat release model described in this Section. First, it provided a means of extending the flame arrival angles obtained at several locations using the ionisation probes to give complete coverage of the flame propagation process. Also, other parameters such as turbulent flame speed could be obtained which would not otherwise have been available. The
results obtained from the model enabled a comparison to be made between predicted and experimental results and also between burnt mass fractions calculated using this model and the much simpler method of Harrington (74) (described in the previous section).

The comparison between predicted and measured flame arrival angles was of particular interest since previously reported heat release models (36, 37, 75) have not been compared in detail with measured flame propagation data. The experimental data acquisition system used (described in Chapter 3) enabled the results to be compared for identical cycles so that an accurate determination of errors could be obtained. By knowing the magnitude of these errors, the relative merits of direct flame propagation measurements (e.g. ion probes, photography, etc) and indirect measurements, using the computer model to analyse pressure data, may be more easily assessed.

The heat release model is basically a modified version of the combustion simulation computer program COMPSIMP which is described in Chapter 6. In fact, all the subroutines and the majority of the algorithms in the master (or main) segment are common to both models, this arrangement giving three main advantages;

a) Reduction in program development time. The time needed to develop the two programs was only slightly longer than would have been required for either one of the models alone.

b) Simplified error analysis. Since the programs use the same algorithms, it follows that errors detected using one of the programs could usually be related to the other.

c) Determination of empirical constants. Analysis of the experimental data using the heat release model allows empirical constants to be determined which can then be used for the full simulation model.

In its present form, the model can be used over a wide range of operating conditions and for a variety of engine designs (note that a "turbulence factor" is not required). The geometrical modelling technique employed allows the model to be suited to a large range of chamber designs.
In addition, it should be noted that the model includes for the effects of the following:-

a) Residual Gas Dilution  
b) Dissociation  
c) Variable Specific Heats  
d) Heat Transfer

A further feature of the model is that it uses an "efficiency factor" to ensure that the whole charge is enflamed, thereby giving complete flame propagation predictions. This permits more meaningful results to be obtained and also makes the results less sensitive to assumptions regarding heat transfer coefficient, residual gas fraction etc, and experiment errors.

Program Description

The combustion simulation computer program COMPSIMP (COMPuter SIMulation Program) upon which the heat release program COMPARED (COMputed PARameters from Experimental Data) is based is described in detail in Chapter 6. Since the majority of the computer algorithms are common to both models, only the basic mode of operation together with details of the differences between the two programs will be described here.

The main differences between the two models are as follows

1) The heat release program considers just the combustion process, i.e. compression and expansion processes ignored.

2) Pressure data rather than the flame speed correlation dictates the rate of combustion, hence no need to calculate adiabatic flame temperature and laminar burning velocity.

3) The combustion process calculations in COMPARED do not start at ignition but are delayed until the pressure increase due to combustion exceeds a specified amount.

4) The combustion process calculations are terminated by an end of combustion criterion rather than the burnt volume fraction exceeding a specified amount as in COMPSIMP. An efficiency factor is then calculated and the whole process
calculations repeated as necessary until the burnt volume at the end of the combustion exceeds the specified amount.

A simplified flow diagram of the heat release model is illustrated in Fig. 4.8 whilst a listing of the Fortran computer program is given in Appendix VIII. A brief description of the computer segments is also included in Table 6.1.

The user specified input data required by the computer program includes operating conditions, engine specification, combustion chamber design details and pressure data values. The pressure data values must start at the specified ignition timing crank angle and continue at consecutive specified crank angle intervals (say 2°) for the whole of the combustion process. Further details of the input data required will be found in the program listing in Appendix VIII.

The composition and properties of the unburnt charge are calculated based on the specified volumetric efficiency, equivalence ratio and assumed residual exhaust fraction. Details of these calculations will be found in Appendices V and VI. Note that the fuel is assumed to be iso-octane.

The process calculations start at the crank-angle corresponding to ignition since the mass of charge burnt at this point is zero. The charge temperature is calculated from the state equation (6.1) knowing the charge mass, cylinder volume and pressure.

The calculations proceed assuming that no combustion is occurring, i.e. compression process only. The state properties at the end of the stage are determined using subroutine COMP, described in Section 6.2.2 and continue until the measured pressure exceeds the predicted compression pressure by 0.5 Bar. It is found that this normally corresponds to a flame radius of about 9 mm.

The purpose of this procedure is to ensure that the combustion calculations start after this "delay" period since computational problems can arise during this period (typically of the order of 15 degrees of crank angle). The problems arise due to the very small values of burnt mass fraction (typically < 1%). Small amounts of scatter in the pressure data during this period will therefore
cause large variations in the predicted combustion rate and may even cause negative rate predictions.

Having determined the end of the "delay" period, the combustion calculations start, the initial stage increment covering the whole of the delay period. The computational procedure during the stage is almost identical to that described in Section 6.2.1, the main difference being that the pressure at the end of the stage is now known.

However, major differences exist in determining the amount of charge burnt during the stage and in defining the end of combustion. The procedures used in this program are iterative and are described below.

**Charge Burnt during Stage**

The method used here is to estimate the burnt mass fraction, calculate the final stage pressure and then repeat the process calculations with a more suitable value of mass fraction until the difference between the predicted and measured pressures is within a specified limit.

For the first stage calculation, i.e. the delay period, a burnt mass fraction of 1% is initially assumed. For subsequent stages, the estimate is based on the values from the previous two stages and is calculated from:

\[
\left( m_{b2}\right)_j = \left( m_{b1}\right)_j + m_b' + m_b''
\]

where

\[
m_b' = (m_{b2} - m_{b1})_j - 1
\]

\[
m_b'' = (m_{b2} - m_{b1})_j - 1 - (m_{b2} - m_{b1})_{j-2}
\]

and \((m_{b1})_j, (m_{b2})_j\) are the burnt mass at the start and end of the \(j\)th stage.
This Euler type prediction is illustrated in the adjacent diagram, and uses both the gradient and rate of change of gradients of the mass burnt-time curve.

Whilst this method will only result in an approximate burnt mass at the end of the stage, it should reduce the total number of iterations required and hence reduce computer run time.

The estimated burnt mass is considered to be acceptable if the difference between measured and predicted pressure does not exceed 0.04 Bar. This order of error is considered acceptable since scatter in the experimental data due to noise, resolution limitations and crank angle interval errors would probably exceed this value. It should be noted that errors due to these iteration errors are not cumulative and therefore if the burnt mass is slightly too low on one stage, it is likely to be increased on the subsequent stage.

If the difference between the pressure exceeds 0.04 Bar, a more accurate estimate of burnt mass is then calculated using

\[ M_{b2} = M_{b2} + \sqrt{\frac{M_{b2}}{M}} \cdot \frac{(P_{\text{exp}} - P_{\text{pred}})}{P_{\text{exp}}} \cdot C \quad -(4.10) \]

where
- \( M \) is the total charge mass
- \( P_{\text{exp}} \) is the experimental cylinder pressure
- \( P_{\text{pred}} \) is the predicted cylinder pressure
- \( C \) is a multiplier which is initially 0.3 but is reduced by a factor of 3 each time the sign of the pressure difference changes, thereby ensuring rapid convergence.
End of Combustion Criteria

Two criteria are used to determine the completion of combustion. These are:

a) The burnt mass fraction exceeds 99.8% and the measured pressure exceeds the predicted pressure by at least 0.04 Bar.

b) The charge burnt for the stage is less than 0.5% and the predicted pressure exceeds the measured pressure by at least 0.04 Bar.

It was found that in every case initially considered, the calculations were terminated by the second of the two criteria and that the value of the burnt mass fraction at the end of the simulation was in the range of about 70-90% (depending on operating conditions and assumed data values). This order of burnt mass prediction has been found by other workers (75, 76) using more complex computer models and is probably due to the effects of incomplete combustion, cyclic variation, assumptions made etc. At weak mixtures, misfire, early flame termination and very slow rates of propagation would also cause a reduction in the predicted mass burnt fraction as shown by Peters (75).

Although it is possible in some cases to analyse results when the predicted values of burnt mass are less than 100%, it does make comparisons between experimentally measured or predicted results (using other methods) and these heat release predictions very difficult. Modifying the results by using a linear type multiplication factor is also not satisfactory.

This is particularly so when dealing with the predicted flame radii. In this work, a comparison between measured ionisation probe flame arrival angles and predicted flame arrival angles was required. Under these circumstances however, the maximum flame radii predicted were less than the distance from the spark plug to the probe farthest away.
It was decided therefore to modify the program to ensure that complete flame propagation was predicted in all cases (note that this may not be suitable when using very weak mixtures). An combustion efficiency factor, CEFF, was used to reduce the chemical energy of the charge by reducing the Internal Energy of Formation values

\[ i.e. \quad U_f = U_f \cdot CEFF \]  

where \( U_f \) is the Internal Energy of Formation

(see Appendix VI)

The value of CEFF is calculated at the end of the combustion process calculations using

\[ CEFF = CEFF \cdot \frac{M_{b2}}{M} \]  

The whole process calculations are then repeated as necessary using the current value of CEFF until the burnt volume fraction exceeds 99.4%. Normally, about three iterations are required.

At the end of the computations, predicted values for the process are printed out in tabular form. An example of the output is included in Appendix VIII. The important parameters predicted include turbulent burning velocity, flame speed, flame radius, burnt mass and volume fractions, temperatures and burnt gas composition.

4.5 Heat Release Model Prediction Error Analysis

The purpose of this section is to first compare the burnt mass fraction predictions of the heat release model and the Harrington method and then to consider the effect of both experimental errors and model assumptions on the accuracy of the heat release predictions. For this analysis, theoretical pressure-time data obtained from the simulation program described in Chapter 6 has been used since this should provide datum conditions free from experimental errors.

Fig. 4.9 shows the burnt mass fraction versus crank angle predictions obtained from the heat release model and the much simpler Harrington
method (described in Section 4.3) for two different engine operating
conditions and combustion chamber designs. The graphs show that the
differences in the predicted burnt mass fraction at a given crank
angle are very small over the full range of values. The greatest
difference occurs in the upper graph and is about 0.4 degrees of
crank angle at 90% burnt mass fraction. The maximum difference in
the lower graph, which represents a retarded ignition case, is only
about 0.2 degrees but in this example, the heat release model predicts
the highest burnt mass fraction.

These results are typical of several sets of pressure data
analysed in this way. It is clear from these results that the heat
release model and the calculation method of Harrington will produce
about the same results provided experimental and analysis errors are
eliminated. The small differences in the predictions obtained from
analysing the theoretical pressure time data could be fully accounted
for by the sensitivity of the Harrington method to the exact value
of compression index, n, assumed. This sensitivity has been
previously illustrated in Figure 4.7. Note that for the comparison
shown in Fig. 4.9, the compression index at ignition was used in
both cases.

In the case of analysing experimental pressure data, the difference
between the heat release model and the Harrington method would
probably be greater than indicated here due to experimental errors.
A comparison between the two methods under such circumstances was
made during the experimental study and the results are discussed in
Section 5.5.

However, before considering the actual experimental results, it is
possible to simulate possible experimental errors and to see the effect
of these on the predictions. Also, the effect of changes in the
assumed heat release model data can be investigated. The results of
these changes are shown in Table 4.1. Note that this analysis has
been restricted to the heat release model since the effect of
experimental errors on the Harrington method will be largely dependent
on the way in which the compression index is determined. Also, since
the heat release model has been developed as part of this work, the
accuracy of its predictions are of more interest.
Table 4.1 shows the effect of pressure data errors (crank angle, d.c and sensitivity), operating condition errors (volumetric efficiency and equivalence ratio) and the effect of model assumptions (heat losses and residual gas fraction) on the combustion efficiency factor (defined by equations 4.11 and 4.12), 2% and 90% burnt mass fraction angles and the 12 mm and 50 mm flame arrival angles. Note that a flame radius of 12 mm rather than 10 mm was used since the combustion calculations do not start until about 9 to 11 mm flame radius due to the "ignition delay" effect.

The effect of pressure errors has been simulated by modifying the datum pressure time/data whilst the operating parameter errors results shown used the datum pressure data with the values of the parameters read in to the computer program being modified. Note that the order of magnitude of the errors assumed are much greater than would be expected from a carefully conducted experimental study using reasonably accurate instrumentation and therefore represent extreme rather than typical errors.

The results show that the predicted burnt mass and flame arrival angles are not greatly affected by the errors and assumptions investigated. The ±4° crank angle errors result in the absolute errors of about ±4° but if the results are corrected for the original ±4° error, the actual changes are much less. Apart from the crank angle error, the only other error of any significance is the 1.0 Bar absolute pressure error which results in 0.8° and 1.3° errors in the 2% burnt mass fraction and 12 mm flame travel angles. The effect of the other errors and assumptions investigated are significantly less than this and are typically ±0.25°.

The greatest effect of the simulated errors is on the "combustion efficiency factor" CEF which varies between 0.69 and 1.04 for the assumed data. It is clear that apart from allowing for the simplified assumptions made in the model, this factor also allows for errors in the experimental data. Without the incorporation of this efficiency factor, the effect of the errors would have been much greater, especially for the 90% burnt mass fraction and 50 mm flame arrival angles. In fact, in several of the cases considered, these values would have been indeterminate since the burnt mass fraction would have been much less than 90%.
<table>
<thead>
<tr>
<th>SIMULATED DATA ERROR OF MODEL ASSUMPTION</th>
<th>COMB. EFF. FACTOR 'CEFF'</th>
<th>CRANK ANGLE FOR BURNT MASS FRACTION</th>
<th>FLAME ARRIVAL ANGLES AT FLAME RADI</th>
</tr>
</thead>
<tbody>
<tr>
<td>Datum Condition ( \phi=1.13, \gamma_{\text{VOL}}=75% ) ( \theta_0=30^\circ ) BTDC</td>
<td>0.69</td>
<td>348.3 372.1</td>
<td>348.9 371.5</td>
</tr>
<tr>
<td>+4° Crank Angle Error (i.e. TDC=364° Indicated)</td>
<td>1.02</td>
<td>352.1 (348.1)*</td>
<td>375.6 (371.6)</td>
</tr>
<tr>
<td>-4° Crank Angle Error (i.e. TDC=356° Indicated)</td>
<td>0.69</td>
<td>344.6 (348.6)</td>
<td>368.0 (372.0)</td>
</tr>
<tr>
<td>+ 1.0 BAR DC Error In Pressure Data</td>
<td>0.88</td>
<td>349.1</td>
<td>372.2</td>
</tr>
<tr>
<td>+ 1% Sensitivity Error In Pressure Data</td>
<td>0.88</td>
<td>348.4</td>
<td>372.2</td>
</tr>
<tr>
<td>( \gamma_{\text{VOL}} ) Assumed To Be 85% (( \gamma_{\text{VOL}} \uparrow 10% ))</td>
<td>0.69</td>
<td>348.4</td>
<td>372.3</td>
</tr>
<tr>
<td>( \gamma_{\text{VOL}} ) Assumed To Be 65% (( \gamma_{\text{VOL}} \downarrow 10% ))</td>
<td>1.04</td>
<td>348.2</td>
<td>371.7</td>
</tr>
<tr>
<td>( \phi ) Assumed To Be 1.23 (( \phi \uparrow 0.1 ))</td>
<td>0.89</td>
<td>348.3</td>
<td>372.1</td>
</tr>
<tr>
<td>Heat Transfer To Walls Assumed To Be Zero</td>
<td>0.81</td>
<td>348.0</td>
<td>371.8</td>
</tr>
<tr>
<td>Residual Gas Fraction Assumed To Be Zero</td>
<td>0.93</td>
<td>347.9</td>
<td>371.6</td>
</tr>
</tbody>
</table>

* Predicted Burnt Mass and Flame Arrival Angles Corrected For ±4° Crank Angle Errors.

TABLE 4.1 EFFECT OF EXPERIMENTAL ERRORS AND MODEL ASSUMPTIONS ON THE HEAT RELEASE MODEL PREDICTIONS
To conclude on the sensitivity of the model to errors and assumptions, it is evident that the effect on the predictions will be small for the order of magnitude of errors which would normally be expected. The effect of estimated parameters such as heat transfer to the cylinder walls and residual gas fraction also result in only a small difference to the predictions.
Fig 4.1  IONISATION AND CRANK ANGLE MARKER PULSES

Fig 4.2  BLOCK DIAGRAM OF IONISATION ANALYSIS SYSTEM
Fig 4.3  FLAME ARRIVAL GRAPH SHOWING MAIN PARAMETERS

Fig 4.4  ENSEMBLE AVERAGING OF TEN SQUARE WAVES
INITIALISE AND READ ENGINE DATA. READ MAG’TAPE BLOCK QUALIFIERS AND EVALUATE DIGITISING CONSTANT

READ IGNITION TIMING, INITIALISE VARIABLES AND DETERMINE CONSTANTS

READ PRESSURE DATA FROM MAGNETIC TAPE FILES FOR 99 CYCLES, ENSEMBLE AVERAGE AND CALC MAX’ VALUES

CONVERT RELATIVE PRESSURES INTO ABSOLUTE VALUES USING EXHAUST STROKE PRESSURE ASSUMPTION

INCREASE ABSOLUTE PRESSURES

CALCULATE VALUE OF CIL USING LEAST SQUARES METHOD

L.T.1.24

IS

C.I.L.?

G.T.1.39

G.E. 1.24 and L.E. 1.38

CALCULATE VALUE OF CII USING LEAST SQUARES METHOD

LT. - 0.005

IS

(CIL - CIU)?

GT. + 0.005

ABS. LT. 0.005

DECREASE ABSOLUTE PRESSURES

Determine value of $P.V^n_{\text{final}}$ for burnt mass calcs.

CONVERT RELATIVE MAX. CYCLE Pressures INTO ABSOLUTE VALUES, DETERMINE CYCLIC DISPERSION.

EVALUATE BURNT MASS FRACTION USING EQUATION 4.8

OUTPUT DATA AND CALCULATED VALUES ON LINE-PRINTER AND CARD PUNCH (IF REQUIRED)

MORE ENGINE CYCLES?

YES

STOP

NO

FIG. 4.5 FLOW DIAGRAM FOR PRESSURE ANALYSIS PROGRAM "PRESS"

NAMELIST

CIL Calculated Compression Index at about $100^\circ$ BTDC

CIU Calculated Compression Index at about $10^\circ$ before ignition
Fig 4.6 EFFECT OF A 0.2 BAR ERROR ON THE CALCULATED COMPRESSION INDEX
Fig 4.7 EFFECT OF COMPRESSION INDEX ON BURNT MASS PREDICTION
ESTIMATE FINAL CHARGE
TEMPERATURES AND HEAT TRANSFER SURFACE AREAS. CALCULATE PROPERTIES VALUES BASED ON THE ESTIMATED TEMPERATURES.

ESTIMATE BUM BT M ASS FRACTION, CALCULATE CORRESPONDING BUMT VOLUME.

START OF COMBUSTION PROCESS CALCULATIONS. INITIALISE PARAMETER VALUES.

COMBUSTION PROCESS CALCULATIONS AS FOR PROGRAM 'COMPSMIP' - SEE FIG. 6.1

REPEAT CALCULATIONS USING MORE ACCURATE ESTIMATES OF TEMP AND SURFACE AREAS.

EVALUATE COMBUSTION CHAMBER DIMENSIONS, CHANGE COMBUSTION AND MASS AS FOR PROGRAM COMPSMIP SEE FIG. 6.1 SET P = P_0 AND PDF = 0.5 BAR

EVALUATE IMPROVED ESTIMATE OF BUMT M ASS FRACTION AND BUMT VOLUME.

END OF COMBUSTION CRITERIA SATISFIED?

NO

INCREASE PDF BY 0.5 BAR
SET P = Q_2

REPEAT CALCULATIONS USING MORE ACCURATE ESTIMATES OF TEMP AND SURFACE AREAS.

NO

YES

CALCULATE FLAME RADIUS AND HEAT TRANSFER SURFACE AREAS USING FUNCTION 'FLAREAD'.

YES

DETERMINE FLAME SPEED FOR INTERVAL, STORE ALL INTERVAL DATA IN ARRAYS.

RESET ALL VALUES AND REPEAT CALCULATIONS USING MORE SUITABLE "COMBUSTION EFFICIENCY FACTOR"

STOP

FIG. 4.6 FLOW DIAGRAM FOR THE HEAT RELEASE MODEL (PROGRAM COMPSMIP)
Fig 4.9  COMPARISON BETWEEN BURNT MASS FRACTION PREDICTIONS
CHAPTER 5

EXPERIMENTAL RESULTS
CHAPTER 5

5.1 Introduction

Results obtained from the experimental part of the study are presented and discussed in this Chapter. Details of the experimental equipment, engine and combustion chamber designs and the experimental test programme are given in Chapter 3 whilst the methods used for the analysis of the data have been described in Chapter 4. A comparison between the experimental and computer simulation model predicted results can be found in Chapter 7.

The bulk of the experimental measurements involved determining flame propagation rates using ionisation probes and cylinder pressure related parameters including burnt mass fraction and cyclic dispersion. Details of these results are given in Sections 5.2 and 5.3 whilst section 5.4 provides some results for the steady state parameters such as brake power, optimum ignition timing and exhaust emissions and temperature.

Section 5.5 compares results obtained by analysing the measured cylinder pressure using the heat release program (described in Section 4.4) with either direct measurements or other analysis methods.

Section 5.6 presents results obtained from combustion chamber measurements for consecutive individual cycles showing the variation from one cycle to the next and also the variation during an individual cycle.

5.2 Ionisation Probe Results

5.2.1 General

The results given in this Section were obtained using several ionisation (up to nine) probes and a data analysis system which is described in Chapter 3. Also, to allow comparisons to be more readily made, only mean cycle results based on 99 consecutive cycles are presented here (see Section 5.6 for individual cycle results).
An exception to this are the results for the 48% concentrated sector type squish design (ref. no. 17). In this case, the results presented have been obtained using the heat release model since time and combustion chamber space restrictions did not permit ionisation probes to be fitted. Although it has been shown in Sections 4.5 and 5.5 that the heat release program predictions agree very well with measured results, the possibility of analysis errors should be borne in mind.

The majority of the results have been obtained using the same cylinder head and ionisation probe and spark plug locations, the only variable being the shape of the combustion chamber. This arrangement allows direct comparisons to be made and eliminates the majority of the errors associated with ionisation probe measurements.

Although the May "Fireball" (Ref. 18 to 21) designs employed a similar cylinder head, the spark plug(s) location, and ionisation probe positions were different from the designs employing the original cylinder head. Also the shape of the inlet tract was modified by the chamber insert. Therefore, direct comparisons are not advisable.

Note that for the 48% concentrated squish design mentioned previously, a direct comparison would have been suspect even if ionisation probes had been used due to probe positioning errors, flame speed variations with flame radius and the effects of spark plug location.

5.2.2 General Trends

For each ionisation probe and test run, a plot of flame radius arrival frequency versus crank angle was constructed and the values of mean flame arrival and flame dispersion (defined in Section 4.2.1) were determined. Fig 5.1 shows a typical plot of flame arrival frequency against crank angle for three probe positions and two engine speeds. The actual experimental data points have been plotted and a smooth curve giving a best fit to the data has been drawn. The graph shows that whilst some scatter is evident, this is not significant. It can be seen that the flame arrival is later for the higher engine
speed case and that the dispersion, as indicated by the slope of the
curve, increases as the probe to spark plug position is increased.

Having determined the flame arrival angle and flame dispersion
values for each of the probes and test conditions, the results were
then plotted against engine speed, ignition timing and equivalence
ratio. Typical plots for several chamber designs are shown in
Figs. 5.2 to 5.4.

The main purpose of presenting these graphs is to illustrate
the variation in the parameters with engine operating conditions and to
indicate the typical variation between the different chamber designs.

The effect of engine speed is to increase both the flame travel
angle and flame dispersion although the rate of increase is reduced
at the higher engine speeds. A minimum flame travel angle is shown
in Figs. 5.3 and 5.4 to occur at about 20° ignition timing and an
equivalence ratio of approximately 1.1. These trends agree well with
previously reported results (10, 28). In addition, Fig. 5.3
indicates that the flame dispersion at large flame radii is decreased
by over-advanced ignition timing.

The graphs show that the difference between the three combustion
chamber designs illustrated is generally relatively small compared with
the scatter in the data. Also, whilst the shape of the curves, drawn
to give a reasonable fit, are similar, some variations do occur. When
plotted against engine speed, the curves indicate that the concentrated
sector squish design has the greatest flame travel angle whilst there
is little difference between the bowl in piston and disc designs.
Fig. 5.3 shows that the disc design gives a minimum flame travel angle
at a more advanced ignition timing than for the other two designs with
the sector type giving the fastest flame speed. Fig. 5.4 suggests that
whilst the equivalence ratio for minimum flame travel angle is about
the same for all three designs, increased flame speed occurs at weak
mixtures for the squish type designs. However, it should be noted that
errors in the determination of the equivalence ratio at very weak
mixtures may have caused some of the differences.
The graphs also indicate that the flame dispersion for the bowl in piston is consistently below that of the disc chamber. However, it will be shown in Section 5.2.4 that the relative difference is highly dependent on flame radius and that at some radii, the disc has much lower dispersion than the bowl in piston.

The effect of volumetric efficiency or throttling is shown in Fig. 5.5 for the disc and concentrated sector type squish designs. This shows that whilst reduced volumetric efficiency increases the flame travel angles in all cases, the effect is more pronounced for the disc. The flame travel angles for the disc are smaller than those for the squish design in all cases.

The ionisation probe results obtained for the other chamber designs tested produced similar trends to those indicated in Figs. 5.2 to 5.4. However, no significant differences at weak mixtures could be detected with the small area concentrated sector squish designs (Refs. 4 to 15) although some improvement at weak mixtures was evident with the May "fireball" designs. Generally, the difference between chamber designs when considering all the data was small compared to the scatter with no clear indication as to which design gave the highest flame speed.

5.2.3 Quantitative Comparison

It is clear from the plots presented so far that it is difficult to compare the designs qualitatively due to data scatter and the different effect of operating parameters. However, qualitative results are needed to form conclusions and therefore a method was devised to obtain such results.

The method employed was to plot the results for each probe and chamber design against engine speed, ignition timing and equivalence ratio. A best fit curve was then drawn through the points, the shape of the curve being based on both the actual data points and on the typical shape for a large range of data.

Since the test conditions were varied around a datum condition of 2000 RPM, 13:1 air to fuel ratio and either 20° or 30° ignition timing (Depending on compression ratio) the three separate plots should give agreement at this condition.
Generally, this was not the case but by considering the differences, it was possible to modify the best fit curves to give agreement at the datum condition. The value obtained at this point was then considered to be the parameter value representing the chamber design and is used for the comparative results presented below. Whilst this method does not fully indicate the relative performance at say weak mixtures, it does give an indication of the relative merits at typical engine operating conditions based on all the experimental results.

The comparative results obtained using this method are presented in Tables 5.1 to 5.3. Table 5.1 compares the results for the disc and small area concentrated sector type squish designs. The results show that in all cases, except the 5% type 4 case where the change is negligible, the flame travel angle to probe 1 (situated 10 mm from the spark plug) was reduced with the addition of squish area, the largest reduction being about 11% for the 20% type 2 case. There appears to be a steady trend of reduced flame travel angle with increased squish area.

The flame travel angles between probes 1 and 2 (10 mm and 50 mm radius respectively) shows a general increase with squish area suggesting that the flame speed is reduced by the addition of squish area. This effect can be seen more clearly when the change in flame speed is considered, as shown at the bottom of the table. This indicates that the flame speed is increased in only two cases, the largest increase being only 2% whilst the largest decrease recorded was 22%. The most effective type of chamber would appear to be the type 2 with the least effective being types 3 and 5. It should be noted however that for the latter two types, errors may have been introduced due to the chamber wall to ionisation probe distance having been reduced, causing the probe to be situated in a region of low flame speed due to wall quenching.

The flame travel angle to probe 2 is affected by the improvement in the early flame travel and the detrimental effect over the main travel period. Therefore, in some cases the values are better whilst in others, the situation is worse. Generally, the type 2 designs gave the best results.
<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>COMBUSTION CHAMBER CONFIGURATION AND REFERENCE NUMBER*</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0/1 5/2 10/2 15/2 20/2 5/3 10/3 5/4 10/4 15/4 5/5</td>
</tr>
<tr>
<td>FLAME TRAVEL ANGLE TO PROBE 1 $\theta_{1,s}$ DEG</td>
<td>17.8 16.6 16.8 16.0 15.8 17.4 17.2 17.5 17.3 16.1 17.9</td>
</tr>
<tr>
<td>FLAME TRAVEL ANGLE TO PROBE 2 $\theta_{2,s}$ DEG</td>
<td>42.4 40.7 41.2 41.2 40.8 46.0 48.7 42.5 44.5 42.1 47.6</td>
</tr>
<tr>
<td>FLAME TRAVEL ANGLE BETWEEN PROBES $\theta_{2,1}$ DEG</td>
<td>24.6 24.1 24.4 25.4 25.0 28.6 31.5 27.0 27.2 26.0 29.7</td>
</tr>
<tr>
<td>MEAN APPARENT FLAME SPEED $U_{2,1}$ m/s</td>
<td>19.5 19.9 19.7 18.9 19.2 16.8 15.2 17.8 17.7 18.5 16.2</td>
</tr>
<tr>
<td>% IMPROVEMENT IN FLAME SPEED RELATIVE TO DISC</td>
<td>-    +2.0 +1.0 -3.0 -1.5 -13.8 -22.0 -8.7 -9.2 -5.1 -16.9</td>
</tr>
</tbody>
</table>

* : See Table 3.2

TABLE 5.1 COMPARISON OF EFFECT OF CHAMBER SHAPE ON FLAME PROPAGATION PARAMETERS AT 2000 RPM, 30° IGNITION TIMING, W.O.T. AND 13:1 AIR TO FUEL RATIO
<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>DISC 0/1 REF No 1</th>
<th>B.I.P. 47.3% REF No 16</th>
<th>SECTOR 48% REF No 17</th>
</tr>
</thead>
<tbody>
<tr>
<td>FLAME TRAVEL ANGLE</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TO PROBE 1 $\theta_{1,1}$</td>
<td>17.2</td>
<td>16.8</td>
<td>18.2</td>
</tr>
<tr>
<td>DEG</td>
<td></td>
<td>(+ 2.3%)</td>
<td>(- 5.8%)</td>
</tr>
<tr>
<td>FLAME TRAVEL ANGLE</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TO PROBE 2 $\theta_{2,2}$</td>
<td>45.8</td>
<td>47.2</td>
<td>47.5</td>
</tr>
<tr>
<td>DEG</td>
<td></td>
<td>(- 3.1%)</td>
<td>(- 3.6%)</td>
</tr>
<tr>
<td>FLAME TRAVEL ANGLE</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>BETWEEN PROBES $\theta_{2,1}$</td>
<td>28.6</td>
<td>30.4</td>
<td>29.3</td>
</tr>
<tr>
<td>DEG</td>
<td></td>
<td>(- 6.3%)</td>
<td>(- 2.5%)</td>
</tr>
<tr>
<td>MEAN APPARENT FLAME SPEED</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$U_{2,1}$ m/s</td>
<td>16.78</td>
<td>15.79</td>
<td>16.4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(- 6.0%)</td>
<td>(- 2.3%)</td>
</tr>
</tbody>
</table>

*: Improvement Relative To Disc Chamber

TABLE 5.2 EFFECT OF CHAMBER DESIGN ON FLAME PROPAGATION

PARAMETERS AT 2000 RPM, 30° IGNITION TIMING, 75% VOLUMETRIC EFFICIENCY AND 13:1 AIR TO FUEL RATIO

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>8:1 REF No 1</th>
<th>10:1 REF No 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>FLAME TRAVEL ANGLE</td>
<td></td>
<td></td>
</tr>
<tr>
<td>TO PROBE 1 $\theta_{1,1}$</td>
<td>17.2</td>
<td>16.1</td>
</tr>
<tr>
<td>DEG</td>
<td></td>
<td>(+ 6.4%)</td>
</tr>
<tr>
<td>FLAME TRAVEL ANGLE</td>
<td></td>
<td></td>
</tr>
<tr>
<td>TO PROBE 2 $\theta_{2,2}$</td>
<td>44.3</td>
<td>40.4</td>
</tr>
<tr>
<td>DEG</td>
<td></td>
<td>(+ 8.8%)</td>
</tr>
<tr>
<td>FLAME TRAVEL ANGLE</td>
<td></td>
<td></td>
</tr>
<tr>
<td>BETWEEN PROBES $\theta_{2,1}$</td>
<td>27.1</td>
<td>24.3</td>
</tr>
<tr>
<td>DEG</td>
<td></td>
<td>(+10.3%)</td>
</tr>
<tr>
<td>MEAN APPARENT FLAME SPEED</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$U_{2,1}$ m/s</td>
<td>17.71</td>
<td>19.75</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(+11.5%)</td>
</tr>
</tbody>
</table>

*: Improvement Relative To 8:1 Compression Ratio

TABLE 5.3 EFFECT OF COMPRESSION RATIO ON FLAME PROPAGATION

PARAMETERS AT 2000 RPM, 20° IGNITION TIMING, W.O.T., AND 13:1 AIR TO FUEL RATIO
Table 5.2 compares the results obtained for the disc, bowl in piston and 48% concentrated sector type squish design. It should be noted that the results for the latter were obtained using the heat release computer program and that the spark plug position was not the same as for the former two designs. Therefore, a direct comparison should be treated with some caution.

The flame travel angle to probe 1 (10 mm) is similar for all three cases, the B.I.P. being the best with the sector being the worst. The flame travel angle to probe 2 (50 mm) is greater for the two squish designs than the disc chamber by about 3% giving a 6% and 2% reduction in flame speed between probes for the B.I.P. and sector designs respectively. These results agree fairly well with those presented in table 5.1 for the small squish areas.

The effect of compression ratio on the measured flame travel angles is shown in Table 5.3. It can be seen that the 10:1 compression ratio improves the flame travel angles throughout the flame propagation range by about 10%.

5.2.4 Multiple Ionisation Probe Results

Although the early tests were performed using only two ionisation probes, later tests employed up to nine probes. The purpose of using a greater number of probes was to see how the chamber design affected the flame speed variation across the chamber and the flame front shape. A further reason for using multiple probes was that it provided data for comparisons with the heat release program predictions and allowed the assumptions regarding spherical flame propagation to be verified.

Figure 5.6 shows the position of the ionisation probes in the Bermoter engine cylinder head. Also shown are the flame arrival angles for the disc and B.I.P. chambers when operating under identical conditions. These results can be used to plot graphs showing the variation of parameters with flame radius and the variation in flame front profile. Such graphs are presented in Figs. 5.7 and 5.8.

It was found that there were significant variations between the profiles for different engine operating conditions. Therefore, the results presented in Figs. 5.7 and 5.8 have been based on the mean
value of 5 engine speeds in the range 2000 to 3000 RPM.

Fig. 5.7 shows the variation of flame travel angle and flame dispersion with flame radius (distance from spark plug) for a number of combustion chambers. A comparison indicates that the 10:1 May chamber gave the highest flame speed at larger flame radii whilst the 10:1 disc chamber was better during the early part of the flame propagation. The bowl in piston design, whilst starting off well produced a low flame speed during the middle of the flame propagation period with higher flame speed towards the end of combustion. It should be noted that this trend occurred for all the bowl in piston test results when compared to the corresponding disc chamber results.

The variation of flame dispersion with flame radius also produced some interesting results. Compared to the disc designs, the squish chambers produced a large value of dispersion at about 30 mm flame radius. Since this corresponds with about the TDC position, it is likely that this is associated with the change in direction of charge motion as the piston changes direction. As with the flame travel angle results, this trend occurred in all cases for the bowl in piston results.

Fig. 5.8 shows the deviation of the measured results from a spherical flame profile for several chamber designs. The graph shows the flame travel angles based on the ionisation data at two flame radii. At a flame radius of 21 mm, the results show that the flame speed along the centre-line of the chamber is significantly higher than in the other directions with some asymmetry probably due to swirl. This type of flame profile has been noted by other workers. The flame profile does not appear to be greatly affected by the chamber design.

At a flame radius of 35 mm, the flame travel angle on the exhaust valve side of the chamber is less than that for the centre which in turn is less than that for the inlet valve side. This asymmetry is probably due to swirl, the magnitude of which increases with distance from the centre of the chamber. A secondary cause could be the effect
of the hot exhaust valve causing some increase in the burning velocity.

The results shown on the graph are plotted above showing the distortion from the spherical profiles. Based on the limited data available, it would appear that the effect of the swirl is more dominant at the wide open throttle condition than at 75% volumetric efficiency. The deviation from spherical, whilst not being large, would tend to be a source of error in the analysis of the flame propagation results and in the computer models where spherical flame propagation is assumed.

5.3 Cylinder Pressure Measurement Results

5.3.1 Introduction

Cylinder pressure measurements were obtained for all of the combustion chambers tested and for each of the operating conditions. Details of the instrumentation, data acquisition and processing systems and analysis procedures used have been described in Chapters 3 and 4.

Cylinder pressure - time data are the most useful of all the information that can be obtained during an experimental investigation into combustion chamber design. This is because the data can either be used to determine the combustion rate (at which the charge is burnt) or can be used to estimate the engine smoothness and performance at lean mixtures.

Further information can be obtained by analysing the pressure-time measurements using a more detailed heat release computer model. Results of such an analysis performed on selected pressure data obtained during this study are reported in Section 5.5.

5.3.2 Graphical Comparison and General Trends

Figure 5.9 shows the pressure-time plots for a number of combustion chambers. This type of graphical representation of the measured data, whilst requiring the least data analysis, can provide a means of rapidly comparing the performance of each chamber design.
It can be seen from inspection of the upper of the two graphs that the bowl in piston yields the highest peak pressure, $P$, whilst there is little difference between the disc and sector chambers. The crank angle at which the maximum pressure is achieved, $\theta_p$, occurs at about $12^\circ$ ATDC for the two squish designs whilst the corresponding value for the disc chamber is retarded by about $4^\circ$.

The lower graph shows that very large differences in the pressure diagram can result from combustion chamber design. It is evident that chambers producing the highest peak pressure also tend to have reduced maximum pressure crank angles and higher pressure rise rates. Comparing the designs, it is clear that compression ratio has a major effect whilst dual ignition and more compact chamber shape also significantly increase the peak pressure.

The burnt mass fraction-time curves for the same data are shown in Fig. 5.10. The upper graph confirms the previous results showing that the bowl in piston and sector squish designs have a higher combustion rate than the disc chamber. Note that during the early part of combustion, both the sector and disc designs give a similar burnt mass rate which in turn is lower than that achieved by the bowl in piston.

The lower graph shows that the dual ignition gives the highest rate of combustion with the high compression ratio designs also giving high values. The 10:1 compression ratio May "Fireball" design resulted in a substantial increase in combustion rate when compared to the corresponding disc chamber. A further point to note is that all the curves are of similar profile, the relative ranking of the chambers being maintained throughout the combustion process.

Although the cylinder pressure and burnt mass plots can be informative, they do suffer from two major disadvantages:

a) They can be misleading when used in comparisons due to scatter in the data since the date for each plot are based on a single test.

b) They cannot be easily used to indicate the effect of operating variables and the relative performance of the chamber designs over the operating range.
A more suitable method of analysing results is to define several parameters and to plot these against the operating conditions. In this work, six parameters are used:-

1. Maximum Mean Cycle Cylinder Pressure, $P$
2. Maximum Cylinder Pressure Crank Angle, $\theta_p$
3. Maximum Pressure Rise Rate, $\Delta p$
4. Percentage Pressure Cyclic Dispersion, $\sigma_p$
5. Delay Angle (Crank Angle for 2% Burnt Mass Fraction), $\theta_d$
6. Burn Angle (Crank Angle for 90% Burnt Mass Fraction), $\theta_b$

Parameters (d) to (f) have been defined in Chapter 4 by equations 4.5, 4.6 and 4.7 respectively.

Typical variation of these parameters with operating conditions and several chamber designs are indicated in Figures 5.11 through 5.16. As with the ionisation results, the purpose of these graphs is to show typical trends and to illustrate the magnitude of scatter in the results since a quantitative comparison of chamber designs is made in sub-section 5.3.3.

Figures 5.11 through 5.14 shows the effect of engine speed on the six parameters for six different combustion chamber designs. The actual data points are shown and best fit curves have been fitted to the data points. It can be seen that in general, the amount of scatter is small compared to the magnitude of the difference between the chamber designs. Compared to the flame speed (ionisation) results, the differences are much greater.

Examining these graphs in more detail, it can be seen that the maximum cycle pressure and the maximum pressure rise rate reduce with engine speed whilst the crank angle at which the maximum pressure occurs tends to increase. These trends are to be expected since the ignition timing was maintained at a constant value instead of optimum (M.B.T.) timing, the latter would have required more ignition advance as the engine speed was increased.

Figures 5.12 and 5.14 indicate that the cyclic dispersion, delay angle and burn angle all tend to increase with engine speed. These trends are as expected. It should be noted however that M.B.T. rather
than constant ignition timing would not necessarily have changed these trends.

The effect of ignition timing on the derived pressure parameters is shown in Fig. 5.15. This illustrates that the parameter values are very sensitive to ignition timing with the delay angle increasing and the cyclic dispersion reducing with advanced timing whilst the burn angle increases either side of about $35^\circ$ ignition timing. These trends are as expected since advanced timing results in reduced charge temperature and hence burning velocity at ignition, thereby increasing the delay angle. The burn angle will be a minimum when the combustion period associated with the 2% to 90% change in burnt mass fraction occurs at about TDC. The percentage cyclic dispersion will tend to decrease with more advanced ignition timing since the maximum mean cycle cylinder pressure will be increased.

Fig. 5.16 shows the effect of mixture strength on the derived pressure parameters. As expected, minimum parameter values occur at an equivalence ratio of about 1.1 with large increases occurring at lean mixtures. This graph is interesting since the corresponding plot of ionisation data (Fig. 5.4) suggested that the flame speed at weak mixtures was substantially better for the squish chambers than for the disc chamber. Fig. 5.16 suggests that the magnitude of the differences between the chambers increase as the mixture strength is reduced such that the squish designs should be better at weak mixtures than the disc chamber.

It should be emphasised however that results obtained from the other chamber designs suggest that the improved weak mixture performance is associated with increasing the compactness of the chamber or charge temperature rather than a squish velocity effect. For example, similar improvements to those shown in Fig. 5.16 were obtained using a more central spark plug location (chamber Ref. No. 2), increased compression ratio (chamber Ref. No. 3) or dual ignition (chamber Ref. No 20). Generally, chambers having improved performance at slightly rich mixtures were also found to be better at lean mixtures.
5.3.3 Quantitative Comparison

To enable a quantitative comparison between the many chamber designs tested to be made, a similar procedure was adopted for the pressure data results as used for the ionisation results described previously in Section 5.2.3. Figures 5.11 through 5.16 have indicated that the relative merits of the designs as compared at the datum condition should be representative of the merits at other operating conditions.

Table 5.4 compares the performance of the disc and small area concentrated sector type squish designs. It should be emphasised that in the majority of these designs, the squish area is positioned such that the chamber compactness is slightly reduced. Therefore, for a constant burning velocity, the overall combustion rate should also be reduced slightly. The main purpose of presenting these data is to confirm the ionisation measurement results shown in Table 5.1.

The maximum mean cycle pressure results do not show any clear trends but the crank angle at which the maximum pressure occurs generally increases as the squish area is increased. The lowest value was achieved with the disc chamber, confirming that the combustion process as a whole is worse with the type of design.

The effect of squish area on the delay angle is in general a slight reduction and therefore this result agrees well with the ionisation results for probe 1. Although the values of $Q_d$ and $Q_{l,S}$ are of similar magnitude, the reduction in $Q_d$ is much less than the reduction in $Q_{l,S}$. This however would be expected since the squish area reduces the flame area at small flame radii.

Burn angle, $Q_b$, generally increases with squish area, this trend being in good agreement with the ionisation results. The table shows that the small squish area type 2 designs actually gave a small improvement (in exact agreement with ionisation results) whilst the worst case was the 10% type 3 with a 7.6% increase in burn angle. The percentage changes in burn angle are less than the corresponding changes in flame speed due to the increased height of the chamber actually increasing the flame area at the larger flame radii.
<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>COMBUSTION CHAMBER CONFIGURATION AND REFERENCE NUMBER *</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0/1</td>
</tr>
<tr>
<td>MAXIMUM CYCLE PRESSURE $\hat{P}$, BAR</td>
<td>52.0</td>
</tr>
<tr>
<td>MAXIMUM PRESSURE CRANK ANGLE $\theta_p$, DEG</td>
<td>374.0</td>
</tr>
<tr>
<td>DELAY ANGLE $\theta_d$, DEG</td>
<td>16.5</td>
</tr>
<tr>
<td>BURN ANGLE $\theta_b$, DEG</td>
<td>31.8</td>
</tr>
<tr>
<td>% IMPROVEMENT IN $% \text{IMPROVEMENT IN } \theta_b \text{ RELATIVE TO DISC}$</td>
<td>-</td>
</tr>
</tbody>
</table>

* : See Table 3.2

**TABLE 5.4** COMPARISON OF EFFECT OF CHAMBER SHAPE ON PRESSURE PARAMETERS AT 2000 R.P.M., 30° IGNITION TIMING, W.O.T. AND 13:1 AIR TO FUEL RATIO
By taking into account the effects of changes in chamber geometry, the results shown in Table 5.4 confirm the results obtained with the ionisation probes. Further details of this confirmation will be found in Chapter 7.

A comparison between the disc, B.I.P. and large squish area sector design is shown in Table 5.5. This shows that the maximum cycle pressure is highest for the B.I.P. design whilst the two squish designs yield approximately a 30° reduction in maximum pressure angle. The maximum pressure rise rate is greatest for the B.I.P., a 35% increase over the disc whilst the cyclic dispersion is halved for the squish chambers. The delay angle is reduced slightly for the B.I.P. but a 7% increase results for the sector design, probably due to the reduced flame area at small flame radii with the spark plug arrangement used. Finally, the burn angle is reduced by 10% and 12% for the B.I.P. and sector designs respectively, confirming that the combustion rate is substantially increased for these two cases.

Corresponding results are given for the 10:1 compression ratio designs in Table 5.6. These results show quite clearly that the combustion chamber design can have an extremely large effect on the parameters. The compact May design gives 12% and 22% reductions in delay and burn angles respectively with corresponding reductions of 17% and 45% for the dual ignition May design. The cyclic dispersion is halved by the squish designs whilst the maximum pressure rise rate is increased by nearly 200% for the dual ignition case.

The dual ignition design clearly gives the highest combustion rate proving that increased flame area or compactness are mainly responsible for the effects of chamber design. The single ignition May head shows that a very compact design can cause considerable increases in combustion rate.

Table 5.7 indicates the effect of compression ratio by comparing disc chamber designs of 8:1 and 10:1 compression ratio. These results show that substantial increases in combustion rate can be obtained by increasing the compression ratio. The delay and burn angles are reduced by 6% and 18% respectively for the higher compression ratio whilst the percentage cyclic dispersion is reduced by 57%, partly as a consequence of the 28% increase in peak pressure.
### Table 5.5 Effect of Chamber Design on Pressure Parameters at 2000 RPM, 75% Volumetric Efficiency, 30° Ignition Timing and 13:1 Air to Fuel Ratio

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>CHAMBER DESIGN AND REF. NO.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>DISC REF No 1</td>
</tr>
<tr>
<td>Maximum Pressure $P$</td>
<td>BAR</td>
<td>42.2</td>
</tr>
<tr>
<td></td>
<td>%</td>
<td>-</td>
</tr>
<tr>
<td>Max' Press' Crank Angle $\theta_p$</td>
<td>DEGREE</td>
<td>376.2</td>
</tr>
<tr>
<td>Max' Press' RISE Rate $\Delta p$</td>
<td>BAR/DEG</td>
<td>1.7</td>
</tr>
<tr>
<td></td>
<td>%</td>
<td>-</td>
</tr>
<tr>
<td>Cyclic Dispersion $\sigma_p$</td>
<td>%</td>
<td>6.0</td>
</tr>
<tr>
<td></td>
<td>%</td>
<td>-</td>
</tr>
<tr>
<td>Delay Angle $\theta_d$</td>
<td>DEGREE</td>
<td>17.5</td>
</tr>
<tr>
<td></td>
<td>%</td>
<td>-</td>
</tr>
<tr>
<td>Burn Angle $\theta_b$</td>
<td>DEGREE</td>
<td>28.7</td>
</tr>
<tr>
<td></td>
<td>%</td>
<td>-</td>
</tr>
</tbody>
</table>

*: % Improvement Relative to Disc

### Table 5.6 Effect of Chamber Design and Dual Ignition on Pressure Parameters at 2000 RPM, 10:1 Compression Ratio, W.O.T., 20° Ignition Timing and 13:1 Air to Fuel Ratio

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>CHAMBER DESIGN AND REF. NO.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>10:1 DISC REF No 3</td>
</tr>
<tr>
<td>Maximum Pressure $P$</td>
<td>BAR</td>
<td>47.4</td>
</tr>
<tr>
<td></td>
<td>%</td>
<td>-</td>
</tr>
<tr>
<td>Max' Press' Crank Angle $\theta_p$</td>
<td>DEGREE</td>
<td>381.3</td>
</tr>
<tr>
<td>Max' Press' RISE Rate $\Delta p$</td>
<td>BAR/DEG</td>
<td>1.55</td>
</tr>
<tr>
<td></td>
<td>%</td>
<td>-</td>
</tr>
<tr>
<td>Cyclic Dispersion $\sigma_p$</td>
<td>%</td>
<td>7.9</td>
</tr>
<tr>
<td></td>
<td>%</td>
<td>-</td>
</tr>
<tr>
<td>Delay Angle $\theta_d$</td>
<td>DEGREE</td>
<td>14.2</td>
</tr>
<tr>
<td></td>
<td>%</td>
<td>-</td>
</tr>
<tr>
<td>Burn Angle $\theta_b$</td>
<td>DEGREE</td>
<td>32.5</td>
</tr>
<tr>
<td></td>
<td>%</td>
<td>-</td>
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</table>

*: % Improvement Relative To Disc Chamber (Ref No 3)
<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>UNITS</th>
<th>8:1 DISC REF No 1</th>
<th>10:1 DISC REF No 3</th>
<th>MAXIMUM PRESSURE $\Delta p$ BAR</th>
<th>% *</th>
<th>+28.3</th>
</tr>
</thead>
<tbody>
<tr>
<td>MAXIMUM PRESSURE CRANK ANGLE $\theta_p$ DEGREE</td>
<td>374</td>
<td>368.9</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MAXIMUM PRESSURE RISE RATE $\Delta p$ BAR/DEG %</td>
<td>1.66</td>
<td>-</td>
<td>+57.2</td>
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<td></td>
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<tr>
<td>CYCLIC DISPERSION $\sigma_p$ %</td>
<td>7.0</td>
<td>-</td>
<td>+37.1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>DELAY ANGLE $\theta_d$ DEGREE %</td>
<td>16.5</td>
<td>-</td>
<td>+6.1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>BURN ANGLE $\theta_b$ DEGREE %</td>
<td>31.8</td>
<td>-</td>
<td>+18.2</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*: Improvement Relative to 8:1 Disc Chamber

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>UNITS</th>
<th>&quot;NORMAL&quot; REF No 1</th>
<th>&quot;CENTRAL&quot; REF No 2</th>
<th>MAXIMUM PRESSURE $\Delta p$ BAR</th>
<th>% *</th>
<th>+6.4</th>
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<tbody>
<tr>
<td>MAXIMUM PRESSURE CRANK ANGLE $\theta_p$ DEGREE</td>
<td>367.8</td>
<td>366.0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MAXIMUM PRESSURE RISE RATE $\Delta p$ BAR/DEG %</td>
<td>2.5</td>
<td>-</td>
<td>+24.0</td>
<td></td>
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</tr>
<tr>
<td>CYCLIC DISPERSION $\sigma_p$ %</td>
<td>2.2</td>
<td>-</td>
<td>+13.6</td>
<td></td>
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<td></td>
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<tr>
<td>DELAY ANGLE $\theta_d$ DEGREE %</td>
<td>19.0</td>
<td>-</td>
<td>+1.1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>BURN ANGLE $\theta_b$ DEGREE %</td>
<td>26.0</td>
<td>-</td>
<td>+6.9</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*: % Improvement Relative To "Normal" Ignition
The combustion process is approximately 5° of crank angle faster at 10:1 compression ratio as shown by the maximum pressure crank angle. This increased combustion rate is due mainly to higher burning velocity as a result of the increased charge temperature and reduced residual gas dilution. Increased measured flame speed results with higher compression ratio were presented earlier in sub-section 5.2.3 and Table 5.3.

The effect of spark plug location is demonstrated by the results shown in Table 5.8. This compares the 8:1 compression ratio disc chamber for the spark plug set at a radius of 19 mm (RS/RC = 0.54) with an alternative spark plug located at the centre of the chamber (RS/RC = 0). These results show that the central spark plug location gives a higher combustion rate compared to the normal spark plug location as verified by the 7% decrease in burn angle, 24% increase in maximum pressure rise rate and 2° reduction in maximum pressure crank angle. The cyclic dispersion was also reduced by 14%. Although the overall improvement is less than for the cases examined of increased compression ratio and dual ignition, the changes are only slightly less than for the B.I.P. and large area concentrated squish sector designs.

5.4 Steady State Parameter Results

Apart from in-cylinder measurements involving ionisation probes and a pressure transducer, measurements of other parameters were also made. These included power output, exhaust temperature and exhaust emissions. Although these parameters vary during the engine cycle, only mean or steady state measurements were made.

It was found that the differences between the power output of the many combustion chambers designs tested was small and generally exceeded by scatter in the results. The power output clearly increased as the compression ratio was raised but since the ignition timing was also reduced, comparisons are not justified.

As already stated, the vast majority of the tests were performed with the ignition timing fixed rather than using an M.B.T. setting since this method is considered to be far more applicable to the
in-cylinder measurements. However, when dealing with power output, it is more useful to consider M.B.T. timing and therefore a comparison between the disc and B.I.P. chambers at M.B.T. timing was made.

The results are shown in Fig. 5.17 which shows the optimum ignition timing and the corresponding power output as a function of engine speed. The graph indicates that the B.I.P. requires less ignition advance than the disc, varying between about $2^\circ$ at 1250 RPM to $5^\circ$ at 3000 RPM. This result confirms the combustion rate results presented earlier in Table 5.5 and Fig. 5.12.

The power output at the lower speeds is about the same for both chamber designs but at higher speeds, the B.I.P. results in a higher power output.

Exhaust gas temperature was measured since this parameter is sensitive to combustion rate; slower combustion resulting in higher temperature providing all other parameters are maintained constant. Two examples of exhaust gas temperature plots for the disc, B.I.P., and 48% sector designs are shown in Figs. 5.18 and 5.19. The former graphs indicate the effect of mixture strength and it can be seen that peak temperatures occur at air to fuel ratios of about 17:1, this being a compromise between high combustion temperature at rich mixtures and slow combustion at weak mixtures. The important point to note however is that the disc chamber results in the highest temperature with the B.I.P. second and the sector design having the lowest temperature. Of particular interest is the plateau occurring at air to fuel ratios greater than about 18.5:1 for the disc chamber suggesting that misfire is occurring. The temperatures for the other two designs continue decreasing down to air to fuel ratios of about 22:1 indicating that the compact squish designs would give improved lean mixture performance. The occurrence of misfire would partially explain the increase in flame travel angle at weak mixtures as shown in Fig. 5.4.

Fig. 5.19 shows the effect of engine speed on exhaust temperature for the same three chamber designs. The temperature increases with engine speed mainly as a consequence of the constant ignition timing. The important point to note is that the temperature for the disc design is consistently higher than for the other two chambers confirming the
fact that a more compact design has the highest combustion rate.

Exhaust emissions were also analysed to determine the concentrations of CO, CO$_2$, NO$_x$, and HC (as hexane). Problems experienced with signal drift and reliability, despite servicing by the manufacturers, reduced the amount of measurements made. Those chambers for which measurements were made did not show any significant differences compared to the level of scatter in the results.

5.5 Results from Heat Release Computer Model

The heat release computer model COMPARED which is described in Chapter 4 was developed in order to analyse experimental pressure data in detail. This model was used to obtain the flame speed given in Section 5.2 for the 48% sector squish design.

The effect of pressure data errors on the predicted values obtained using this model have been considered in Section 4.5. The purpose of this Section is to first illustrate the accuracy of the model by comparing measured flame propagation data, obtained from ionisation probes, with the predictions obtained from the model analysing pressure data recorded for the same engine cycles as the ionisation data.

Second, turbulent burning velocity predictions for several chamber designs are compared, thereby indicating the utility and limitations of the model.

Figures 5.20 through 5.22 compare the predicted flame radius/crank angle profiles with the ionisation data results for several chamber designs and different operating conditions. Inspection of these graphs shows that in general the heat release model predicts slightly lower flame speeds during the early part of the flame propagation. The accuracy of the model does not vary considerably over the range of different combustion chamber configurations and operating conditions. It should be emphasised that the difference between the results will be due to errors in the ionisation data, pressure data and heat release model. Bearing this in mind, it would appear that the model gives very good results since the flame radius/crank angle curves based on results
from several ionisation probes suggest some scatter between probes.

The model predictions have been compared with the ionisation data for a large number of operating conditions and chamber designs. In all cases, the results obtained suggest that the model is sufficiently accurate to be used as an analysis tool to enable quantitative results to be obtained. Comparisons between the burnt mass fractions predicted by this model and the Harrington(1974) method, described in Chapter 4, have shown that the difference is negligible in most cases.

The major source of error in the model when used for flame speed evaluation is the assumption of spherical flame profiles. Whilst this is in general a reasonable assumption, it is known that swirl, quench zones and probably squish will tend to distort the flame profile. Evidence of this type of distortion has been detected in this study and has been presented earlier in Fig. 5.8. It is clear that if large deviations from the spherical profile occur, errors may be significant. This is particularly the case for large squish area type chambers due to the large variation in chamber height (and hence volume) between the main chamber and squish zone.

On the other hand, if large flame profile deviation occur, flame speed measurements using several ionisation probes become meaningless since at best they can only indicate the mean flame speed or travel angle between two probes in the chamber and do not indicate the rate of volume enflamment. In such circumstances, comparison of flame speeds, obtained from whichever method can only be of limited usefulness. However, the heat release model has the advantage that it can predict a reasonably accurate burnt volume and burnt mass fraction.

Predicted turbulent burning velocities plotted against flame radius for several chamber designs are shown in Figs. 5.23 and 5.24. Apart from showing the way in which the burning velocity varies across the chamber, this type of plot can be used to compare the burning velocity rather than the total flame speed variation for the different chambers.

Fig. 5.23 suggests that the burning velocity for the three designs is similar during the first half of the propagation with large variations during the latter half. This is in contrast with 5.23 where the differences are much smaller. There is insufficient evidence to show
whether or not these variations do occur but it is thought likely that some of the differences with the large squish area designs are due to the errors discussed earlier regarding the spherical flame profile assumption.

In addition to the analysis of mean engine cycles, based on the ensemble average of 99 consecutive engine cycles, about forty individual engine cycles were also analysed using the heat release model. The purpose of doing this was to check on the agreement between the ionisation results and model predictions for individual cycles ranging from "fast" to "slow" combustion rates. A second reason was to determine whether or not fast cycles resulted in a higher combustion efficiency factor than the slow cycles.

For the individual cycle analysis, the mean measured values of equivalence ratio and volumetric efficiency were used as input data. The predictions obtained were found to be in good agreement with the ionisation probe results, the magnitude of the differences being similar to the mean cycle results already presented. The agreement was in general about the same for both fast and slow cycles.

The combustion efficiency factor, defined in Section 4.4 by equations 4.11 and 4.12, was found to vary between about 70% to 90% for mean cycles and about 60% to 100% for individual cycles. Although this parameter was plotted against operating parameters such as engine speed, ignition timing and equivalence ratio, no clear trends could be detected. With individual cycles, it was found that whilst the slow cycles in general gave slightly lower combustion efficiency factors than the fast cycles, the differences were very small and almost negligible compared with the scatter.

5.6 Individual Cycle Analysis

The ionisation and pressure data results reported in the previous sections of this Chapter were ensemble averaged values based on 99 consecutive engine cycles. Due to cyclic dispersion, ensemble averaged values are much more suitable for analysing combustion chamber performance than individual cycle values.
However, data from several hundred individual cycles were also analysed, there being three main reasons for doing this:

a) To demonstrate the ability of the data acquisition/processing system to readily provide data for consecutive individual cycles with synchronisation for up to twelve data channels.

b) To provide pressure and ionisation data so that the accuracy of the heat release model could be verified for individual as well as ensemble averaged cycles. Details of these results have been given in the previous section.

c) To exploit the advantages of the experimental set up so that information showing the variation of the flame propagation, both during a single cycle and from one cycle to the next, could be obtained. Whilst this information is not completely relevant to the objectives of this study, the results obtained are of general interest. This is particularly so since little information of this type has been published.

Fig. 5.25, the first of three graphs to be presented in this section, shows the flame arrival angles for 15 consecutive cycles as measured using five ionisation probes situated at different distances from the spark plug (see Fig. 3.2 for details of the probe positions). There are three major points to note from this graph. First, the cycle to cycle variation increases as the flame arrival angle and ionisation probe distances increase. Next, the cycle to cycle variation appears to be completely random with no dependence between consecutive cycles. Third, although in general the relative merits of an individual cycle are the same for all probes, some cases exist where this is not completely true. For example, cycles No. 6 and 11 are better than average throughout the propagation whereas cycle No. 12 starts off slower but finishes slightly faster than average.

Correlations between the flame arrival angles at ionisation probes situated at 10 mm and 50 mm and maximum cylinder pressure are shown in Fig. 5.26. Best fit lines show that the engine cycles which result in higher than average cylinder pressure also tend to be the cycles with minimum flame travel angles. The maximum deviation in flame travel angle from the best fit lines and from the mean flame travel angle is about 16% and 20% respectively for both sets of data.
This shows that although the magnitude of the flame dispersion increases throughout the flame propagation process, the percentage remains about constant. More important, the correlations suggest that the level of dispersion is dictated by the events occurring during the early part of the ignition process. This is supported by the good agreement between flame arrival angles at ionisation probe 1, situated at 10 mm from the spark plug, and the peak cylinder pressure.

Further correlations are presented in Fig. 5.27. The top graph shows that the correlation between flame travel angles for ionisation probes 1 and 2 is about ±20%. This confirms the importance of the ignition process with respect to dispersion. The lower graph shows that the crank angle at which maximum pressure occurs is related to the flame arrival angle at probe 2. It can also be seen that whilst the flame arrival and maximum pressure angles are about the same value for fast combustion cycles, slower cycles have less effect on the latter parameter than they do on the former. This would be expected since the peak pressure angle is a function of both the rate of combustion and piston movement.

Examination of the pressure time data for the individual cycles reveals that quite large variations in cylinder pressure occurred prior to and during the early part of the ignition process. Maximum variations of about ±0.8 Bar (approx. ±10%) were noted. Due to the evidence obtained suggesting the importance of the dispersion at the start of combustion, the cylinder pressure at ignition was plotted against peak cylinder pressure. However, this plot did not suggest any dependence between the two parameters, thereby eliminating the pressure variations at ignition as being a possible mechanism for the dispersion.

Therefore, it would appear that the variation is due to some other cause, the most likely candidates being charge motion effects or non-homogeneous variations in charge composition adjacent to the spark plug electrode. Whilst the exact mechanism is not known, the results presented in this Chapter have shown that design parameter changes which reduce the combustion duration also reduce the cyclic dispersion.
Fig 5.1 FLAME ARRIVAL FREQUENCY PLOT SHOWING TYPICAL PROFILES AND SCATTER

- Probe 1, R = 10 mm
- Probe 5, R = 34.5 mm
- Probe 2, R = 50 mm

\( \varphi = 1.1 \)
\( \theta_s = 30^\circ \)
\( \gamma = 75\% \)
Fig 5.2  TYPICAL FLAME PARAMETER PLOTS SHOWING
EFFECT OF ENGINE SPEED
Fig 5.3 EFFECT OF IGNITION TIMING ON FLAME TRAVEL AND DISPERSION ANGLES
Fig 5.4  EFFECT OF MIXTURE STRENGTH ON MEASURED FLAME PARAMETERS
Fig 5.5 EFFECT OF VOLUMETRIC EFFICIENCY ON FLAME TRAVEL ANGLES

LEGEND
1 DISC (Type 1) Ref No 1
2 SECTOR (48\% Type 3) Ref No 17

Fig 5.6 FLAME TRAVEL ANGLES FOR SIX ION PROBES AND TWO CHAMBER DESIGNS

N = 2000 RPM
\( \phi = 1.13 \)
\( \theta_s = 30^\circ \)
\( \eta_v = 75\% \)
Average of 5 Engine Speeds
2000—3000 RPM
\( \varnothing = 1.13 \)
\( \theta_s = 20^\circ(10:1) & 30^\circ(8:1) \)
\( \gamma = W.O.T(10:1) & 75\%(8:1) \)

**Legend**

- **DISC 8:1 C.R** 1
- **DISC 10:1 C.R** 3
- **B.I.P** 16
- **MAY 10:1 C.R** 18
- **SECTOR** 17

**Fig 5.7 VARIATION OF FLAME PARAMETERS WITH FLAME RADIUS**
Fig 5.8 PLOT OF IONISATION RESULTS SHOWING DEVIATION FROM SPHERICAL FLAME PROFILE
Fig 5.9 EFFECT OF CHAMBER DESIGN ON THE PRESSURE-TIME DIAGRAM
Fig 5.10 EFFECT OF CHAMBER DESIGN ON THE BURNED MASS FRACTION – TIME DIAGRAM
Fig 5.11  EFFECT OF ENGINE SPEED AND CHAMBER DESIGN ON MEASURED PRESSURE PARAMETERS

Legends:
1. DISC Ref No 1
2. B.I.P Ref No 16
3. SECTOR Ref No 17

\[ \phi = 1.13 \]
\[ \theta_e = 30^\circ \]
\[ \gamma_v = 75\% \]
Fig 5.12 EFFECT OF ENGINE SPEED AND CHAMBER DESIGN ON DERIVED PRESSURE PARAMETERS
Fig 5.13  EFFECT OF ENGINE SPEED AND CHAMBER DESIGN ON MEASURED PRESSURE PARAMETERS

LEGEND

- DISC Ref No 3
- MAY Ref No 18
- MAY DUAL Ref No 20

Max Pressure Rise Rate Bar/°
Max Pressure Crank Angle °
Max Cycle Pressure Bar

Engine Speed RPM

\( \phi = 1.13 \)
\( \theta_s = 20^\circ \)
\( \gamma_V = \text{W.O.T} \)
C.R = 10:1
Fig 5.14 EFFECT OF ENGINE SPEED AND CHAMBER DESIGN ON DERIVED PRESSURE PARAMETERS
LEGEND
1 • DISC Ref No 1
2 x B.I.P Ref No 16
3 ○ SECTOR Ref No 17

N = 2000 RPM
\( \varphi = 1.13 \)
\( \gamma_{V} = 75\% \)

Fig 5.15 EFFECT OF IGNITION TIMING AND CHAMBER DESIGN ON DERIVED PRESSURE PARAMETERS
Fig 5.16
effect of equivalence ratio and chamber design on derived pressure parameters

Legend
- DISC Ref No 1
- BIP Ref No 16
- SECTOR Ref No 17

\[ \Phi_s = 30° \]

\[ \eta_s = 75\% \]

\[ N = 2000 \text{ RPM} \]
\[ \eta_v = 75\% \]

\[ \varnothing = \text{Max Power Setting} \]

Fig 5.17 M.B.T. IGNITION TIMING AND POWER CURVES FOR DISC AND BOWL IN PISTON CHAMBERS
Fig 5.18 Measured Exhaust Gas Temperatures For Various Chamber Designs Plotted Against Mixture Strength

Fig 5.19 Effect of Chamber Design and Engine Speed on Exhaust Gas Temperature
Fig 5.20  COMPARISON BETWEEN MEASURED AND PREDICTED FLAME PROPAGATION PROFILES
DISC Ref 1

N = 2000 RPM

\( \phi = 1.13 \)

\( \gamma = 75\% \)

\( \theta_s = 40^\circ \)

\( \theta_s = 30^\circ \)

\( \theta_s = 20^\circ \)

Fig 5.21 Comparison between measured and predicted flame propagation profiles for different ignition timings.
Fig 5.22  COMPARISON BETWEEN MEASURED AND PREDICTED FLAME PROPAGATION PROFILES FOR DIFFERENT MIXTURE STRENGTHS
Predicted Turbulent Burning Velocity (m/s) vs. Flame Radius (mm)

- Sector Ref No. 17
- Disc Ref No. 1
- B.I.P. Ref No. 16

Predicted Values using Heat Release Model COMPARED

N = 2000 RPM
\( \theta_s = 30^\circ \)
\( \phi = 1.13 \)
\( \gamma = 75\% \)

Fig 5.23  PREDICTED TURBULENT BURNING VELOCITY PLOTTED AGAINST FLAME RADIUS
Fig 5.24 PREDICTED TURBULENT BURNING VELOCITY PLOTTED AGAINST FLAME RADIUS

$N = 2000\text{ RPM}$

$\Theta = 30^\circ$

$\phi = 1.13$

$\chi_v = \text{W.O.T}$

10:1 DISC Ref No 3

15% SECTOR TYPE 4 Ref No 14

8:1 DISC Ref No 1

Predicted Values using Heat Release Model COMPARED

Flame Radius mm
Fig 5.25 FLAME ARRIVAL ANGLES FOR 15 CONSECUTIVE ENGINE CYCLES AND 5 IONISATION PROBES
Fig 5.26  CORRELATION BETWEEN FLAME PROPAGATION RATE AND MAXIMUM CYLINDER PRESSURE
Fig 5.27  CORRELATIONS BETWEEN FLAME ARRIVAL AND MAXIMUM PRESSURE ANGLES
CHAPTER 6

DETAILS OF COMPUTER SIMULATION MODELS
CHAPTER 6

6.1 Introduction

Computer simulation of the internal combustion cycle offers many advantages to the engine research worker, the main ones being:-

a) Reduction in the need for experimental testing, producing results in a fraction of the time and at reduced cost.

b) The detailed cycle analysis provides a better understanding of the complex processes involved.

c) Parameters can be varied both independently (of other parameters) and outside of the experimental range.

d) The range of parameters obtained from an experimental study can be greatly extended by using a simulation model to analyse the experimental (e.g. pressure-time) data.

These advantages have resulted in the development of a large number of models of varying complexity and scope. Much success has been achieved to date using thermodynamic or zero dimensional models such as those reported for example in Refs. 8-12. A characteristic of these models is that the spatial and time variations of the charge properties are either ignored or modelled in a relatively simple fashion. Also, it is common for the model to concentrate on the aspects of the engine processes which are of particular interest to the objectives of the study for which it is being used. Despite this simplistic approach, good predictions can be obtained provided that empirical data (such as a burning velocity correlation) are used. As one might expect, these models are ideally suited for parametric studies.

More recently, multi-dimensional models (e.g. 77, 78) have been reported which model the spatial variations of parameters such as charge motion, composition and temperature within the combustion chamber and inlet and exhaust manifolds. These models tend to be very detailed in their description of the processes since their ultimate goal is to be totally predictive with respect to engine design. At present, these models suffer from being expensive to run and tend to give poor agreement with experimental results due to the lack of data and knowledge on the fundamentals of combustion.

For this study, a simulation program was required which could predict the values of the relevant parameters for a large range of idealised
consideration of the previously reported models indicated that none of these readily permitted the chamber design to be altered since large amounts of data in the form of either flame radius/burnt volume/surface area polynomials \((10, 79)\) or chamber wall co-ordinates \((36, 78)\) would be required for each design. Also, multi-dimensional type models were considered to be unsuitable due to their excessive computer storage and run time requirements.

It became clear that a model with the ability to predict the effect on engine performance of a large range of idealised chamber designs, approximating to typical present and future designs, whilst requiring minimal input data would be of considerable utility to engine researchers and designers. Such a model would allow optimum chamber designs to be obtained rapidly and with minimum expense, would indicate the relative importance of parameters and would simplify the analysis of experimental results.

Therefore, the main objective of the theoretical part of the work was to produce a computer simulation model capable of satisfying the requirements outlined above. For convenience, the development of this model can be considered in 2 parts, namely:

a) Thermodynamic Model. This was a relatively simple program based on a number of previously reported models (e.g. refs. 9, 10, 11, 79) and is described in Section 6.2.

b) Geometrical Model. This was an extension of the simple cylindrical model previously reported (24, 80) and calculated the burnt volume, burnt gas wall surface areas and flame front surface areas for a large range of chamber designs. This model is described in Section 6.3 and was either used as part of the combustion simulation programs or was used independently to calculate the variation of the geometrical parameters with flame radius for the different chamber designs.

The computer simulation model was developed in 2 versions, a predictive combustion program called COMPSIMP (COMputer SIMulation Program) and a heat release program called COMPARED (COMbustion PARameters from Experimental Data). These programs are very similar, sharing all subroutines/functions and many main program algorithms, the difference being that the former uses a flame speed correlation whilst the latter requires measured pressure-time data. Due to this
similarity, only the predictive program will be described in this Chapter, the parts of the heat release program which differ from this have been described in Section 4.4.

In the final part of this Chapter, Section 6.4, results predicted by the simulation program for a range of operating conditions are presented to verify the accuracy of the computer model. Computed results indicating the effect of combustion chamber design on both burnt volume/surface areas and combustion parameters such as combustion rate are presented and discussed in Chapter 7.

6.2 Thermodynamic Model

6.2.1 General

The purpose of the computer model is to predict the effects of combustion chamber design. To reduce the amount of work required to construct the computer program and the amount of computer resources required to use it, the general philosophy adopted was to avoid modelling parts of the engine cycle which were not significantly affected by chamber design.

In keeping with this philosophy, the complex processes occurring during the intake and exhaust processes are not simulated and instead, assumed values of volumetric efficiency, cylinder pressure at inlet valve closure (I.V.C.) and exhaust residual fraction are specified as data. Therefore, the part of the engine cycle modelled is from I.V.C. to exhaust valve opening (E.V.O.). For convenience, this period was divided into three separate processes, namely compression, combustion and expansion.

The simulation model, which was written in Fortran, was based on a forward stepping, iterative solution technique similar to that used by James (10) and Phillipps and Orman (9). This method was used in preference to the alternative numerical solution of non-linear multi-order differential equations due to its relative simplicity, flexibility, reduced computer resource requirement (10) and the good predictions previously obtained using this method.
In its present form, the model can simulate the effect of a number of parameters including:-

a) Engine Speed
b) Ignition Timing
c) Compression Ratio
d) Cylinder Dimensions (Bore and Stroke)
e) Combustion Chamber Shape
f) Spark Plug Location
g) Dual Spark Plugs.

In addition, the program includes several desirable features:-

a) Residual Gas Dilution
b) Dissociation
c) Variable Specific Heats
d) Heat Transfer

The main assumptions made in the model are:-

a) The fuel is iso-octane and the oxidant is moist air.
b) The charge behaves as a perfect gas.
c) The charge properties and species concentrations are completely homogeneous (i.e. no quench layers or pressure and temperature gradients) during the compression and expansion processes.
d) No pre-flame reactions occur.
e) During combustion, two sub-systems, each having homogeneous properties, exist, separated by an infinitely thin reaction zone.
f) The flame propagates spherically outwards about the ignition origin.
g) The products satisfy the equilibrium conditions (i.e. chemical kinetics are ignored).

A flow diagram showing the computational logic is shown in Fig. 6.1. This shows only the major steps. A full listing of the program is presented in Appendix VIII for readers wishing to know more precise details. A number of subroutines and functions
are used and a brief description of these is given in Table 6.1.

Basically, the model predicts the charge temperatures, pressures, flame speed, flame radius, heat and work transfer, emissions and other data during the cycle. In addition, the power output and heat transfer for the cycle is calculated allowing optimum ignition timings to be determined.

The cycle simulation uses a number of expressions, the major ones being as follows:-

**STATE EQUATION**

\[ P \cdot V = M \cdot R \cdot T \]  \hspace{1cm} - (6.1)

where

- \( P \) = Pressure
- \( V \) = Volume
- \( M \) = Mass
- \( R \) = Specific Gas Constant
- \( T \) = Temperature

**ENERGY BALANCE (1st LAW)**

\[ Q - W = \Delta U \]  \hspace{1cm} - (6.2)

where

- \( Q \) = Heat Transfer
- \( W \) = Work Transfer
- \( \Delta U \) = Change in Internal Energy

**POLYTROPIC PROCESS RELATIONSHIP**

\[ P \cdot V^n = \text{Constant} \]  \hspace{1cm} - (6.3)

where \( n \) is the compression or expansion index

**CONTINUITY EQUATION**

\[ M_1 = M_2 \]  \hspace{1cm} - (6.4)

where \( M_1 \) and \( M_2 \) refer to the mass at two points in time
<table>
<thead>
<tr>
<th>SEGMENT NAME</th>
<th>PURPOSE OF SEGMENT</th>
<th>REFER TO</th>
</tr>
</thead>
<tbody>
<tr>
<td>COMP</td>
<td>Calculates change in state properties and energy transferred for a polytropic process</td>
<td>6.2.2</td>
</tr>
<tr>
<td>CPMEAN</td>
<td>Determines the mean mixture specific heat over specified temperature range</td>
<td>Appendix VI</td>
</tr>
<tr>
<td>DIM</td>
<td>Evaluates the dimensions of the specified combustion chamber design</td>
<td>6.3</td>
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<tr>
<td>EBURN</td>
<td>Calculate equilibrium composition of products for given temperature and pressure</td>
<td>Appendix V</td>
</tr>
<tr>
<td>FLARAD</td>
<td>Computes the flame radius corresponding to a given burnt volume for a specified chamber</td>
<td>6.3</td>
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<tr>
<td>RESIDUL</td>
<td>Determines the composition of the unburnt mixture assuming a known residual fraction</td>
<td>Appendix V</td>
</tr>
<tr>
<td>SAVOL</td>
<td>Calculates the burnt volume and flame and heat transfer surface areas for a given chamber design</td>
<td>6.3</td>
</tr>
<tr>
<td>VALVE</td>
<td>Evaluates the additional volume and areas for cylindrical valve type recesses</td>
<td>6.3</td>
</tr>
</tbody>
</table>

**TABLE 6.1**

**DESCRIPTION OF COMPUTER SEGMENTS**
HEAT TRANSFER CORRELATION

\[ Q = A_w \cdot h_c \cdot (T - T_w) \]  - (6.5)

where

- \( A_w \) is the Heat Transfer Area
- \( T_w \) is the assumed Wall Temperature
- \( h_c \) is the Heat Transfer Coefficient

The heat transfer coefficient is calculated using Eichelberg expression:

\[ h_c = 0.523 \times 10^{-6} \left( \frac{S \cdot N}{300} \right)^{\frac{1}{3}} \cdot (P/T)^{\frac{1}{2}} \]  - (6.6)

where

- \( h \) is in \( W/mm^2\ ^{0}K \)
- \( S \) is the Stroke (mm)
- \( N \) is the Engine Speed (R.P.M.)
- \( P \) and \( T \) are in Bars and \(^{0}K\)

PISTON POSITION

The piston position relative to the T.D.C. position, \( x \), is calculated using the following expression:

\[ x = \frac{S}{2} \cdot (1 - \cos (\theta) + E - \sqrt{E^2 - \sin^2(\theta)}) \]  - (6.7)

where

- \( E = 2 \frac{C_L}{S} \)
- \( \theta \) is the Crank Angle
- \( C_L \) is the Connecting Rod Length

These are the main expressions upon which the program is based but a number of other equations are also used for the various processes. Details of some of these are given in the remainder of this section and in Appendicies V and VI.

Input data required by the program include details of the operating conditions, general engine specification, combustion chamber design and empirical constants. Details of the data which must be specified are included in the program listing in Appendix VIII.
The predicted results are printed out in tabular form at the end of the cycle simulation. An example of the results produced and the format used is also presented in Appendix VIII. In addition to the tabular output, selected results and warnings are printed out during the cycle calculations to help the user to identify any errors or problems which may arise.

The program has been run on both I.C.L. 19045 and Digital PDP 11 computers with reliable results. Note that the listing given in Appendix VIII is for the I.C.L. machine and some minor changes are required for the PDP 11. The run time is dependent on the specified data but is typically of the order of 20 - 30 seconds.

6.2.2 Compression/Expansion Processes

The calculations required for the compression and expansion processes were very similar and were performed using subroutine COMP, the flowchart of which is shown in Fig. 6.2. The major differences between the two processes, apart from the fact that the composition and properties of the charge are different, is that for compression, the composition is assumed constant (i.e. preflame reactions ignored) whereas for the expansion, the composition is varied as some of the dissociated products recombine. As with previous models (9, 10) it is assumed that at temperatures below $1600^\circ$K, the composition is frozen at the $1600^\circ$K value.

The pressure at the start of the compression process is specified as data and the temperature calculated using equation (6.1). The pressure and hence temperature for any given intake system will be a function of volumetric efficiency and residual fraction. An approximate empirical expression to give the correct type of dependence of pressure and temperature on throttling and residual fraction was used and is given by:

$$P_{IVC} = P_{DAT} \cdot \left( \frac{\rho_{VOL} \times 0.93/75.0 + RES/100}{\rho_{VOL} \times 0.93/75.0 + RES/100} \right) - (6.8)$$

where

- $P_{IVC}$ is the calculated pressure at the start of compression
- $P_{DAT}$ is the specified pressure constant (taken as 0.9 Bar in this work)
The object of the calculations is to establish the conditions at the end of the specified interval (values of 10° and 4° were used for compression and expansion respectively) knowing the conditions at the start. To achieve this, the process between end states, was idealised as being made up of two processes, namely an isentropic process from the initial to final cylinder volumes, followed by heat transfer and chemical recombination effects at constant volume. This idealisation is shown diagramatically in Fig. 6.2 and is similar to that used by James and Phillipps and Orman although the computational method used here is different from these previous works.

The calculations start by estimating the final conditions and then calculating the gas properties and heat transfer based on the estimated values. For the expansion process, the internal energy of reaction due to equilibrium composition changes is also computed providing the temperature exceeds 1600°K.

The final volume, \( V_2 \), is calculated using equation (6.7) and the final temperature, \( T_2' \), is then evaluated using equations (6.2) and (6.15)

\[
T_2 = T_2' - \frac{Q}{M \cdot \bar{C}_v} - \frac{\Delta U_p}{M} \tag{6.9}
\]

where
- \( m \) is the charge mass
- \( \Delta U_p \) is the change in internal energy of formation \((U_p_2 - U_p_1)\) for the products = 0 for compression process
- \( \bar{C}_v \) is the mean specific heat at constant volume
- \( T_2' \) is the isentropic temperature

Using equations (6.1) and (6.3), \( T_2' \) is given by:

\[
T_2' = \left( \frac{V_1}{V_2} \right)^{(\gamma - 1)} . T_1 \tag{6.10}
\]

where \( \gamma \) = Ratio of Specific Heats
The corresponding pressure is calculated using the equation of state (6.1). If the value of \( T \) is not within 0.2°C of the previously estimated value, the calculations are repeated until agreement is reached. This order of error will only have a slight effect on the calculated properties and heat transfer and should be insignificant.

Having calculated the final state values, the work transfer is computed using:

\[
W = \frac{P_2 \cdot V_2 - P_1 \cdot V_1}{1 - n}
\]

which using equation (6.3) becomes

\[
W = \frac{P_2 \cdot V_2 - P_1 \cdot V_1}{\log \left( \frac{P_2}{P_1} \right) / \log \left( \frac{V_1}{V_2} \right)}
\]

This process is repeated throughout the compression and expansion processes until either ignition or exhaust valve opening occur, each time the final values becoming the initial values for the next stage.

6.2.3 Combustion Process

During the combustion process, the cylinder contents are assumed to be divided into two sub-systems representing the unburnt (\( u \)) and burnt (\( b \)) fractions. These two zones are assumed to be separated by an infinitesimally thin flame front which propagates spherically from the spark plug electrodes.

In this model, the combustion process is idealised as involving six steps:

a) Constant pressure enflament

b) Adiabatic mixing of the burnt and enflamed charges

c) Combustion of burnt/enflamed mixture at constant volume
d) Expansion/compression of burnt/unburnt fractions to achieve pressure equilibrium

e) Piston movement resulting in expansion/compression of the charges to give uniform pressure

f) Heat transfer at constant volume resulting in further changes in pressure, temperature and sub-system volumes.

This type of sequence is similar to (but not the same as) that used by Patterson and Van Wylen (79). However, for the purposes of the combustion calculations, the six idealised steps were reduced to two computational steps as follows:-

1) "Combustion step" covering idealisation steps (a) - (c). Let the initial and final states be denoted by suffixes 0 and 1 respectively

2) "Piston Movement" step covering idealisation steps (d) - (f). Let the initial and final states be denoted by 1 and 2 respectively.

This two step combustion system is similar to that used by Phillips and Orman (9) and James (10) and is preferable to using a larger number of steps since it reduces program size, running time and the total magnitude of iteration errors.

The concept of the two step combustion model is illustrated diagramatically in Fig. 6.3. This shows the P-V diagrams for the burnt and unburnt sub-systems in addition to a diagrammatic illustration of the flame front and piston positions.

**Combustion Step**

At the start of the combustion step, the states of the sub-systems are fully defined, having been set equal to the values calculated for the end of the previous combustion or compression step.

Knowing the unburnt temperature and cylinder pressure, the turbulent burning velocity $U_T$ is calculated, the method and expressions used being described in Section 6.2.4. The calculated value of $U_T$ is then used to determine the flame radius at the end of the step, $r_f$. 
using:-

\[ r_{f1} = r_{f0} + U_r \cdot t_i \]  \hspace{1cm} (6.12)

Where \( r_{f0} \) is the flame radius at the start of the stage and \( t_i \) is the time (in seconds) which corresponds to the specified crank angle interval, \( \Theta_i \), taken to be 2° in this work, and defined by:-

\[ t_i = \frac{\Theta_i}{6 \times N} \]  \hspace{1cm} (6.13)

The corresponding volume of the burnt zone, \( V_{bl} \), is now calculated using subroutine SAVOL, which is described in Section 6.3.

The corresponding values of unburnt volume and the masses of the sub-systems are then calculated using

\[ V_{ul} = V_1 - V_{bl} \]  \hspace{1cm} (6.13)

\[ M_{ul} = V_{ul} \cdot \rho_u = \frac{V_{ul} \cdot P_{ul}}{R_u \cdot T_{ul}} \]  \hspace{1cm} (6.14)

where \( \rho \) is the density

\[ M_{bl} = M - M_{ul} \]  \hspace{1cm} (6.15)

Having established the mass and volume of the sub-systems, the mixture in the burnt zone is allowed to react under equilibrium conditions and at constant volume. The temperature of the burnt gas at the end of the combustion step is calculated by applying the first Law of Thermodynamics, equation (6.2). Since the combustion is assumed to occur at constant volume with no heat transfer, the internal energy must be constant.

\[ U_{b2} - U_{bl} = 0 \]

Using equation (VI.18) we obtain the following expression for \( T_{bl} \)-:
\[ T_{bl} = T_L + \left[ \frac{(T_{u0} - T_L) \cdot \bar{e}_v + (U_r - U_{p1})}{(U_{p0} - U_{p1}) + (T_{b0} - T_L) \cdot \bar{e}_v} \cdot \Delta m_b \right] / (\bar{e}_v \cdot m_{bl}) \] 

- (6.16)

where: 

\[ T_L = 298.15^\circ K (25^\circ C) \]
\[ \Delta m_b = m_{bl} - m_{b0} \]
\[ U \] is the Internal Energy of Formation (suffices r and p for reactants and products)

The internal energy of formation for the burnt and unburnt mixtures is calculated using the data and expressions given in Appendix VI. The values used in equation (6.16) were reduced by multiplying by an efficiency factor which is determined from experimental pressure-time data. In this work, an efficiency factor of 0.89 was used. (See Section 4.5 for further details).

The corresponding pressure \( P_{bl} \) is calculated using the equation of state (6.1).

**Piston Movement Step**

In this step, heat transfer and piston movement are allowed to occur and the sub-systems expand/compress to give pressure equilibrium at the end of the stage.

Heat transfer from the two sub-systems is calculated using equations (6.5) and (6.6) with the surface areas being calculated using subroutine SAVOL. Note that it was assumed that no heat transfer occurred between the two sub-systems. Mean values of surface area and temperature for the stage are used with the final values being estimated. These are then updated as necessary until the required agreement is achieved.

The solution is as follows:-
a) Estimate volumes at the end of the step

b) Calculate burnt and unburnt temperatures based on isentropic expansion/compression minus heat losses as for compression/expansion processes, i.e. equation (6.9)

c) Calculate pressures from equation (6.1)

d) Compare the calculated pressures for the two sub-systems. Generally, these will not agree and an estimate of the final pressure is obtained from:

\[
\frac{p_2}{2.0} = \left( \frac{M_{b2}}{M} + 0.5 \right) + \frac{p_{u2}}{2.0} \left( 1.5 - \frac{M_{b2}}{M} \right) \quad (6.17)
\]

e) Improved estimates of the final volumes are then obtained by expanding/compressing the smaller of the two volumes to the value given by equation (6.17) using equation (6.3).

f) (b) to (e) are repeated until agreement is reached.

The flame radius at the end of the stage corresponding to the burnt volume and piston position is calculated using function FLARAD, described in Section 6.3. The mean flame speed for the stage, \( U_f \), is calculated using:

\[
U_f = \frac{r_{f2} - r_{f0}}{t_i} \quad (6.18)
\]

The piston work for the interval was finally calculated using an approximate relationship:

\[
W = \left[ \frac{P_2 + P_0}{2.0} - P_{atm} \right] \cdot (V_2 - V_1) \quad (6.19)
\]

Although this is only strictly true for a constant pressure process, it should give acceptable results for the small stage incremental values used.

The combustion calculations are repeated until the calculated burnt volume fraction exceed 99.4%. At this point, the charge is assumed to be totally products and the expansion process calculations described in Section 6.2.2 continue.
6.2.4 *Prediction of Turbulent Burning Velocity*

A multitude of laminar and turbulent flame propagation theories and models have been proposed and some of these have been reviewed in Section 2.2. None of the turbulent models reported are capable of being totally predictive or universally applicable and therefore require empirical constants to obtain agreement with measured values.

For this study, the turbulent burning velocity $U_T$ was assumed to be related to the laminar burning velocity $U_L$ by:

$$U_T = U_L (1 + K_T \cdot N)$$  \hspace{1cm} (6.20)

Where $K_T$ is a constant determined from empirical data

$N$ is the engine speed

This type of expression has been used by many researchers (9, 10, 36) and gives fairly good agreement with measured results. It should be noted however that equation (6.20) implies that the ratio $U_T/U_L$ (i.e. F.S.R.) is:

a) Increases linearly with engine speed and is equal to unity at zero engine speed. This is in good agreement with experimental results (38).

b) Is independent of crank angle, throttle opening (or load) and combustion chamber design (shape, compression ratio, etc). This does not mean that $U_T$ is constant since $U_L$ will vary with temperature.

It is not clear from previous work whether or not the ratio $U_T/U_L$ should be made proportional to mean gas flow rate through the intake valve as suggested by some workers (11, 24) rather than the engine speed correlation used. The uncertainties arise due to throttling affecting the residual fraction and charge temperatures and pressures in addition to the turbulence effects, each one of which would modify the turbulent burning velocity. Fortunately, this was not a serious problem here since the majority of the predicted results given in Chapter 7 correspond to a constant volumetric efficiency of 75%.
This study concentrates on combustion chamber design and therefore it is important to point out that the turbulent to laminar flame speed ratio was assumed to be independent of chamber design (assuming that the intake valve and port arrangement is not modified). In the case of compression ratio, an increased value would reduce the residual fraction slightly and increase the compression temperature, the nett result being an increase in predicted flame speed.

In the case of chamber shape, the flame speed ratio was kept constant for 2 reasons.

a) Experimental results (reported in Chapter 5) obtained as part of this work, in addition to the results of other workers (36, 40, 46) indicate that the flame speed and turbulence are not greatly affected by chamber shape.

b) It allowed the geometrical effects of chamber shape to be studied independently of turbulence effects. The prediction could then be compared with experimental results, thereby indicating the relative effect of the chamber generated turbulence.

The value of $K_T$ in equation (6.20) was determined by comparing the predicted and experimentally measured flame propagation rates for the cylindrical combustion chamber operating over a range of speeds and equivalence ratios but at a fixed volumetric efficiency of 75%. As with previous workers (10, 11) it was found necessary to use a reduced value of $K_T$ during the early part of the flame travel although a constant value could be assigned for the remainder. This is thought to be due to the effect of a stagnant boundary layer and a thermal quench zone around the spark plug electrodes and walls. James (10) for example assumed laminar propagation for the first 3.5 mm of travel whilst Samaga and Murthy (11) assumed a linear increase from zero to full velocity during the first 10% of its travel and a linear decrease during the final 5%.

The value of $K_T$ used in this work was as follows:

\[
K_T = 0.00291 \text{ R.P.M.}^{-1} \text{ for } r_f > 10.0 \\
\text{or } K_T = 0.00291 \times \frac{r_f}{10.0} \text{ R.P.M.}^{-1} \text{ for } r_f \leq 10.0
\]
Where \( r_f \) = flame radius in mm.

These values were found to give good agreement with the multiple ionisation probe measurements obtained from the Bermoter engine and have been used for most of the results presented in this and the next chapter.

The laminar flame velocity, \( U_L \), was calculated using a simplified version of the Semenov (82) equation as reported by Samaga and Murthy (83). The values of \( U_L \) in cm/s is given by:

\[
U_L = \sqrt{\frac{k \cdot (n_1/n_2) \cdot T_u^2 \cdot T_f^{1.21} \cdot e^{-E/R_o \cdot T_F \cdot (1 - A \cdot Y)}}{E^2 \cdot (1 + T_u/T_F)^{0.375} \cdot (1 - T_u/T_F)^2}}
\]  

\[(6.21)\]

Where \( Y = 1 - \frac{R_o \cdot T_F}{E \cdot (1 - T_u/T_F)} \)

\( A = \varphi \) for \( \varphi \leq 1 \)

\( A = 1/\varphi \) for \( \varphi > 1 \)

and \( E \) is the activation energy = 40 Kcal/mol for iso-octane \((n_1/n_2)\) is the ratio of moles of reactants to moles of products in the stoichiometric equation = 0.945 for \( C_8 H_{18} \).

\( R_o \) is the universal gas constant = \( 1.985 \times 10^{-3} \) k cal/mol\(^0\)K

\( T_u \) = unburnt gas temperature \(^0\)K

\( T_F \) = adiabatic flame temperature \(^0\)K

\( K \) = is a constant and was given as 59.0 in Ref. 83.

The adiabatic flame temperature, \( T_F \), is calculated by assuming a constant pressure combustion process with no heat transfer across the "boundary". The first law of thermodynamics states that the enthalpy \( H \) is constant, i.e.

\[ H_2 = H_1 \]
Using equation (VI.16) we obtain:-

\[ T_F = \frac{\bar{H}_F + \bar{C}_p \cdot (T_u - 298.15)}{\bar{C}_p} \times \frac{[\bar{H}_F]_p + 298.15}{[\bar{C}_p]_p} \]  

-(6.22)

Where \( \bar{H}_F \) is the enthalpy of formation per unit mass of \( \bar{C}_p \) is mean specific heat at constant pressure sufficies r and p denote reactants and products respectively.

Since \( \bar{C}_p \) and \( \bar{H}_F \) for the products will be a function of the final temperature, the calculation is performed using an iterative technique.

Using equations (5.21) and (6.22), the predicted laminar flame speed was calculated for a range of equivalence ratio's for a fixed unburnt temperature of 700°K. The results are shown in Fig. 6.4 by the dotted curve.

This shows that the maximum value is predicted at an equivalence ratio of 1.15. Also, although the curve reduces either side of this, a discontinuity exists between equivalence ratios of between 0.95 and 1.1. This does not agree with experimental results as shown by Dugger and Graab (84) and therefore it was decided to modify the results in this region by using an alternative expression for \( \lambda' \) in equation (6.21).

The value of \( \lambda' \) was calculated using

\[
\begin{align*}
\lambda' &= \lambda' \quad \text{if } \phi \leq 0.9 \\
&= 0.9 + (\phi - 0.9)/2.0 \quad \text{for } 0.9 < \phi \leq 1.0 \\
&= 1.0/1.1 + ((1.1 - \phi) \times 0.126)\frac{1}{2} \quad \text{for } 1.0 < \phi < 1.1 \\
&= 1/\phi \quad \text{for } \phi \geq 1.1
\end{align*}
\]

The effect of this is shown by the solid line in Fig. 6.4 and it can be seen that a smooth curve is now predicted. However, the constant \( 'K' \) in equation (6.21) will need to be reduced since the data upon which the value of 59.0 was based was for \( \phi = 1.05 \) and \( U_L \) is now increased at this equivalence ratio. It was found that a value of...
of 55.0 gave the best agreement with the data for iso-octance given in Ref. 21 as shown below.

| Initial Flame Equivalence Laminar Flame Velocity cm/s |
|-------------------------------|------------|
|Temperatures | Equivalence | Laminar Flame Velocity cm/s |
| $T_u$ K | $T_f$ K | Ratio | | Experimental | Predicted |
| 311 | 2285 | 1.05 | | 34.9 | 34.94 |
| 422 | 2337 | 1.05 | | 56.8 | 56.91 |

The predicted laminar flame velocities for a range of unburnt temperatures and equivalence ratios are presented in Fig. 6.5.

6.2.5. Calculation of Squish Velocities during Combustion

Appendix IV gives details of the method used and the assumptions made for the calculation of squish velocities when the density of the cylinder charge is assumed to be spatially uniform. Although this is approximately true for motored engines, it is clearly a gross assumption for fired engines where large density variations exist between the burnt and unburnt charges. The effect of combustion on the squish velocity appears to have been either over-looked or ignored by previous workers studying this subject since no results have been published.

To help in understanding the effects of combustion chamber design, it would be helpful to have a more reasonable estimate of the squish velocities during the combustion. In this work therefore, an algorithm was written to calculate the squish velocity under firing conditions. This was incorporated into the computer simulation program and used the predicted values of charge densities for the squish determination.

Since the purpose of these calculations is to show the effect of combustion on the motored values, no attempt was made to calculate the velocities for all the chamber configurations which can be specified. Instead, the computations were restricted to the simple
case of a symmetrical squish chamber configuration with central
ignition, i.e., bowl in piston or hemi-spherical head.

The squish velocity is evaluated by calculating the change in
the mass of the charge in either the annulus or bowl during the stage
increment and then dividing by the mean flow area and density. The
squish velocity for a bowl in piston design of bowl radius $r_b$
is given by:–

$$u_s = \frac{M}{\tilde{e} \cdot \bar{h} \cdot 2\pi \cdot r_b \cdot t_i}$$

where:

For $r_f < r_b$

$$M = SQA \cdot (h_0 \cdot \epsilon_{uo} - h_2 \cdot \epsilon_{u2})$$

$$\tilde{e} = (\epsilon_{uo} + \epsilon_{u2})/2.0$$

For $r_f > r_b$

$$M = (V_2 - SQA \cdot h_2) \cdot \epsilon_{b2} - (V_0 - SQA \cdot h_0) \cdot \epsilon_{bo}$$

$$\tilde{e} = (\epsilon_{bo} + \epsilon_{b2})/2.0$$

and $r_b$ is the bowl radius
$t_i$ is the stage increment in seconds
SQA is the squish area
$h$ is the piston to head height = $x$ + bumping clearance.

The results obtained from using this expression are compared with the
corresponding motored values in Chapter 7.
6.3 Geometrical Simulation Model

6.3.1 Introduction

This section describes the geometrical simulation model which was written to predict the burnt volume/surface areas versus flame radius relationships for a number of combustion chamber designs. This model, the development of which was a major part of the work, was required for three reasons

a) To obtain plots of burnt volume and flame surface area against flame radius. These curves can then be compared and analysed to obtain a good idea of how the engine will perform with the combustion chamber under consideration (see Section 2. for details).

b) To be used as part of the heat release model COMPARED, so that the flame speed and heat transfer surface area could be determined.

c) To be used as part of the predictive combustion simulation model COMPSIMP, so that burnt volume, heat transfer surface area and flame speed could be determined.

The geometrical model consists of four computer segments, namely DIM, FLARAD, SAVOL and VALVE, a brief description of the purpose of these segments having been given in Table 6.1 whilst a program listing appears in Appendix VIII.

FLARAD calculates the flame radius knowing the burnt volume and is described in more detail in Section 6.3.4. The other three segments are complementary and calculate the burnt volume and flame/heat transfer surface area for a specified flame radius assuming a spherical flame front. A general description of these segments is presented in Section 6.3.2 whilst details of the computational method and expressions used are described in Section 6.3.3 and Appendix VII.
6.3.2 General Description

There are several ways in which the burnt volume and surface areas may be calculated in a combustion simulation model:

I) Input Data Interpolation. Burnt volumes and flame/heat transfer surface area data at a number of piston and flame front locations are input to the program in the form of array data (79) or polynomial equations (10). The data can be obtained either experimentally or using theoretical expressions. The burnt volume and surface areas at a specified flame radius and engine crank angle are then calculated by interpolating between the input values.

II) Approximation Formulae. Expressions relating the geometrical parameters to the flame radius and piston position are used in this method (80). These expressions can be obtained either from theory or a data fitting correlation.

III) Numerical Integration. An integration method is used to give exact integral expressions relating the geometrical parameters to flame radius and piston position (24, 80). These equations are then solved numerically.

IV) 3 Dimensional Mesh. The shape of the combustion chamber is specified in terms of the co-ordinates of a 3 dimensional mesh or grid (36). The burnt volume and surface areas can then be determined for any flame radius and piston position (providing the mesh dimensions are not fixed in the axial plane). The accuracy and computer requirements for this method are dependent on the size of mesh used.

The relative advantages and disadvantages of these methods are compared in table 6.2. It is clear from this comparison that each method has some favourable features and the method chosen will be dependent on the actual application.

For this work, a technique was required which would be capable of accurately simulating the effect of a large number of combustion chamber designs. Examination of the methods indicated that whilst
<table>
<thead>
<tr>
<th>METHOD</th>
<th>ADVANTAGES</th>
<th>DISADVANTAGES</th>
</tr>
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<tbody>
<tr>
<td>I. Input data</td>
<td>a) Simple computational procedure</td>
<td>a) Large amounts of data required for each chamber design</td>
</tr>
<tr>
<td>interpolation</td>
<td>b) Can assume various flame front profiles</td>
<td>b) Time consuming processes in generating input data</td>
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<tr>
<td>II. Approximation</td>
<td>a) Simple computational procedure</td>
<td>a) Restricted to very simple shapes</td>
</tr>
<tr>
<td>formulae</td>
<td>b) Minimum input data requirement</td>
<td>b) Errors can be large</td>
</tr>
<tr>
<td>III. Numerical</td>
<td>a) Minimum input data requirement</td>
<td>a) Restricted to chamber shapes which can be geometrically defined</td>
</tr>
<tr>
<td>integration</td>
<td>b) High order of accuracy possible</td>
<td></td>
</tr>
<tr>
<td>IV. 3 Dimensional</td>
<td>a) Can be used for any combustion chamber shape</td>
<td>a) Large amounts of data required for each chamber shape</td>
</tr>
<tr>
<td>mesh</td>
<td>b) Suited to multi-dimensional combustion models</td>
<td>b) Large computer storage and run times</td>
</tr>
<tr>
<td></td>
<td></td>
<td>c) Errors can be significant</td>
</tr>
</tbody>
</table>

**TABLE 6.2**

COMPARISON OF TECHNIQUES FOR THE DETERMINATION OF COMBUSTION CHAMBER/FLAME GEOMETRICAL PARAMETERS
all of these were far from ideal, the numerical integration method offered the most potential. The other methods were rejected due to excessive input data requirements, poor accuracy and excessive computer demands.

The numerical integration technique has been used by Blizzard and Keck (24) and Annand (80) but these workers have only considered a right circular cylinder (disc type) with the ignition origin on the upper flat face. For this simple case, the method can simulate the effects of piston position, spark plug radial location, compression ratio, and bore/stroke ratio.

Clearly, if this method was to be suitable for work on squish type chambers, modifications to the expressions would be required. An initial inspection revealed that many simple shapes could be catered for by this technique.

Initially, the method was applied to single and double sector type squish chambers (i.e. with squish area between chord and circumference) in addition to the disc type chamber. Later, this was extended to bowl in piston and hemi-spherical head designs and further additions were made to include for cylindrical valve recesses and a hemispherical spark plug recess. Finally, the spark location was made variable in the axial direction (in addition to the radial direction) and axi-symmetrical dual ignition was added.

The combustion chamber designs that can be modelled by the latest computer program are indicated in Figs. 6.6 to 6.8. (See Appendix VII for a more precise description of the range over which the design parameters shown can be varied). These show that there are ten basic combustion chamber configurations, referred to as types 1 to 10, the first six types corresponding to those already mentioned in Chapter 3 (see Fig 3.3). These ten types are really variations on five chamber shapes since types 4 and 5 are special cases of type 3 and types 7, 8 and 9 are the dual ignition versions of types 1, 6 and 10 respectively.

For each of these configurations, the following parameters may be varied.
a) Spark plug radial position (RS)
b) Spark plug penetration (PH)
c) Percentage squish area (SQ1, SQ2)
d) Bumping clearance (BC)
e) Bowl in piston corner or dome radius (RA)
f) Bowl/hemispherical head offset (RSB)

In addition, the spark plug hemispherical recess and inlet/exhaust valve cylindrical recesses which may be specified are shown in Fig 6.9. The valve recesses can be located anywhere on the cylinder head upper face and may be of specified diameters and depths. Note that the model assumes cylindrical rather than spherical flame propagation in the valve recesses. (The difference is normally small for typical valve dimensions).

These recesses may be combined with the main chamber types to give other shapes. For example, in the case of the engine used in the experimental part of this study, the spark plug and valves formed recesses in the flat cylinder head face. By specifying cylindrical valve and hemispherical spark plug recesses, together with types 1 to 6 configurations, the volume and area profiles for this cylinder head are accurately determined.

Other chamber shapes can also be modelled by this method and some of these are shown in Fig. 6.10. For example, the heart shape design which is common to many production engines can be approximated by specifying deep valve recesses combined with a flat top piston (e.g. BL 'A' series) or with a shallow bowl (e.g. Ford Kent). Another shape of interest that can be modelled is a spherical combustion chamber with central ignition (not a practical design but one which gives greatest flame front area and is therefore ideal for comparison purposes).

It should be noted that in its present form, the model, whilst being capable of simulating spherical flame /combustion chamber parameters for a large range of designs, cannot be used for wedge shape chambers or for shapes that do not conform to the relatively simple geometry indicated by Figs. 6.6 to 6.10.
However, the work done so far indicates that the method described here could be extended to include other common designs such as the wedge shape and to remove some of the symmetry restrictions placed on the model. The fact that production engine chambers of the concentrated squish type (types 2 to 5) tend to have a more rounded profile need not be restrictive since this would normally be a small order effect.

The input data required by the model is shown in the program listing in Appendix VIII. A total of twenty values are required to fully specify the combustion chamber design and size. Note that the model calculates the total chamber height to give the correct compression ratio after allowing for the specified recess volumes and squish areas.

6.3.3 Computational Method

The computational method used basically involves determining the relationship between the flame radius and combustion chamber dimensions and the cross sectional area/lengths in terms of axial displacement. The expressions obtained are then integrated over the total chamber height to give the required parameters.

Using simple trigonometry, i.e. Pythagorean Theorem, Sine and Cosine Rules and the nomenclature given in Fig. 6.11, we obtain the following equations:

\[ V_b = \int_0^H (\alpha \cdot R^2 + \beta \cdot R C^2 - R C \cdot R S \cdot \sin \beta) \, dZ \quad (6.24) \]

\[ A_f = 2 \cdot R C \cdot \int_0^H \alpha \cdot dZ \quad (6.25) \]

\[ A_b = 2 \cdot R C \cdot \int_0^H \beta \cdot dZ + A_h + A_p \quad (6.26) \]

where

\[ \alpha = \cos^{-1} \left( \frac{(R S^2 + R^2 - R C^2)}{2 \cdot R S \cdot R} \right) \quad (6.27) \]

\[ \beta = \cos^{-1} \left( \frac{(R S^2 + R C^2 - R^2)}{2 \cdot R S \cdot R C} \right) \quad (6.28) \]

\[ R = \sqrt{R F^2 - z^2} \quad (6.29) \]
\[ V_b = \text{Volume of Burnt Zone} \]

\[ A_f = \text{Surface Area of the Flame Front} \]

\[ A_b = \text{Total Burnt Zone Wall Area} \]

\[ A_h \] and \[ A_p \] are the surface areas of the Burnt Zone Wall Area corresponding to the flat cylinder head and piston faces respectively. These values can be evaluated using

\[ A = \alpha \cdot R^2 + \beta \cdot RC^2 - RS \cdot RC \cdot \sin \beta \] \hspace{1cm} -(6.30)

with the value of \( R^2 \) being either set to \( RF^2 - PH^2 \) or \( RF^2 - (H - PH)^2 \) for \( A_h \) and \( A_p \) respectively.

Equations (6.24) to (6.30) must now be solved using a numerical integration method. For this work, "Simpsons Rule" was used although several other well known techniques could have been employed.

Basically, Simpsons rule involves dividing the region over which the function is to be integrated into any even number of equal width strips as illustrated by Fig. 6.12. The value of the function at each of the ordinates is then evaluated. The value of the integral is then given by:

\[ \int_{x_1}^{x_2} f(x) \cdot dx = \frac{S}{3} \cdot \left[ (F + L) + 4E + 2R \right] \] \hspace{1cm} -(6.31)

where

\[ S = \text{strip width} = \frac{(x_2 - x_1)}{n} \]

\[ n = \text{number of strips} \]

\[ F = \text{Value of function at first ordinate} \]

\[ L = \text{Value of function at last ordinate} \]

\[ E = \text{Sum of even-numbered ordinates} \]

\[ R = \text{Sum of remaining odd-numbered ordinates} \]

The proof of Simpsons Rule is given by Stroud (85). The accuracy of the method will improve as the number of strips is increased and will be better for slowly varying functions. However, the computer requirements also increase as the number of strips are increased and therefore the number of strips chosen will be a compromise between accuracy and computer time.
Simpson's Rule, when applied to the combustion chamber problem, involves dividing the total chamber height (or flame radius if $RF < H$) into strips. The ordinate values to be determined are the burnt zone plan areas and the flame front and wall circumferential lengths for the burnt volume and surface areas respectively. These values are shown more clearly in Fig. 6.13, and are calculated using the $f(x)$ expressions in equations (6.24) - (6.26).

The method used for simulating other chamber shapes and dual ignition is similar to that described for the simple disc chamber. However, the equations used are different and these are given in Appendix VII, together with a description of the method and expressions used for calculating cylindrical valve recess burnt volume and surface areas.

A flow chart showing the main features of the solution logic used in this model is given in Fig. 6.14. Note that when possible (this depending on the chamber design and flame radius) exact geometrical formulae are used instead of the numerical integration method thus reducing computational time.

As previously mentioned the number of integration strips used is a compromise between accuracy and computer run time. It was found from experience that for zero or low squish areas, about 20 strips were sufficient but for larger squish areas, the number of strips required for good accuracy ($> 99.7\%$) was increased above 50. For the results reported in this thesis, 80 strips were used to ensure accurate results in all cases. Typical computer time for this number of strips is approximately 0.25 seconds.

6.3.4 Flame Radius Determination

The flame radius corresponding to specified burnt volume, piston position, engine specification and combustion chamber design is calculated using the computer function FLARAD. This segment is normally called at the end of the combustion stage calculations when it is necessary to determine the radius so that the flame speed, heat transfer surface area and initial flame radius for the next stage can be evaluated. In the heat release program, it is also used to
determine the turbulent burning velocity.

A flow chart showing the logic used in this function is shown in Fig. 6.15. It can be seen from this that the solution is based on a simple iterative technique which makes use of the burnt volume/surface area subroutine SAVOL. It should be noted that this technique could be used with any type of burnt volume calculation method (types shown in Table 6.2). Its main advantages are simplicity and the fact that it does not require any additional data to describe the chamber shape, thereby making it universally applicable.

Basically, the solution involves estimating the flame radius, calculating the corresponding burnt volume and then, depending on the difference between this and the actual volume, adjusting the radius by a fixed increment. The process is repeated until a change in the sign of the difference between the volumes is detected, thereby indicating the actual radius is somewhere between the latest and preceding estimates. A more accurate value is then calculated by using a linear interpolation.

A radial increment of 1.0 mm is initially used (to give rapid convergence) and then the whole of the process is repeated using a smaller increment of 0.1 mm (for improved accuracy). The amount of calculation required will depend on the accuracy of the initial estimate, but this will be reasonably accurate since the radius for the previous step is known. If the initial estimate is within say 2.0 mm, SAVOL will be called about five times.

The accuracy of the technique, assuming no errors in SAVOL, will be dependent on the flame radius/burnt volume relationship over the radius range at which the final linear interpolation is made. The worst condition would be during the early part of the flame propagation over which the burnt volume increases with radius to the third power. Specimen calculations made under these conditions show that the worst error at a flame radius of 10.0 mm is about 0.003 mm.

To ensure reliable operation, it was found necessary to make two modifications to the program.
First, at flame radii less than 1.0 mm, it is possible to get a condition where the radius is made negative due to the size of the radius increment. Therefore, if this occurred, the iteration is performed between zero and the previous iteration radius and the procedure is continued.

The second modification is needed for large flame radii corresponding to burnt volumes greater than about 99.8% and is due to small errors in SAVOL (see Section 6.33). In certain cases, these errors cause the predicted burnt volumes at complete flame propagation to be slightly less than the total cylinder volume. In this event, FLARAD would continue increasing the flame radius indefinitely.

To prevent such an occurrence, the flame radius is prevented from exceeding the maximum possible radius. If at this radius, the burnt volume predicted is still less than the known volume, the radius is set to the maximum value and a warning is printed.

6.4 Program Verification

The computer simulation model COMPSIMP described in the previous sections of this Chapter has been used to predict the effect of combustion chamber design parameters. These results are presented and discussed in Chapter 7.

The purpose of this section is to present results obtained from the same program showing the effect of operating conditions rather than chamber design. There are two reasons for doing this. First, such results would not otherwise have been included in this thesis since Chapter 7 deals specifically with combustion chamber design. Second, it provides an opportunity for demonstrating the accuracy and limitations of the model (since the results can be readily compared with predicted and empirical results reported in the literature.

To simplify the comparisons, a single combustion chamber design is assumed with the following specifications:-
Shape ........................................ Disc (Type 1), no recesses
Spark plug location ............... 19 mm radial, zero penetration
Compression ratio ....................... 8:1
Bore ........................................ 70.0 mm
Stroke ....................................... 70.0 mm
Con rod length ...................... 120.0 mm

Due to the fairly large range of operating conditions, the effect of each of the parameters is demonstrated whilst maintaining all other parameter values fixed at a datum condition. This datum is as follows:

Speed ................................. 2000 RPM
Ignition timing ....................... 30° BTDC
Equivalence ratio ................... 1.1 (i.e. Rich)
Volumetric efficiency .............. 75%

In addition, the cylinder wall temperature is assumed to be 440°K and a constant residual mass equivalent to 7% mass fraction at a volumetric efficiency of 75% is assumed. The values of the other data required are as specified in section 6.2.

The predicted values of the cylinder pressure and burnt and unburnt zone temperatures during the compression and expansion strokes are shown in Fig. 6.16. For the assumed conditions, a maximum pressure of about 47 Bar occurring at about 12° ATDC is obtained. Maximum temperatures are about 800 and 2500°K for the unburnt and burnt zone respectively. These results agree well with other published data (10, 83, 86).

Fig. 6.17 shows the work and heat transfer rates during the same part of the engine cycle. The former parameter varies between negative and positive values of approximately 11 and 53KW respectively. The heat transfer rate is negligible during most of the compression stroke due to the small temperature difference but increases rapidly to a maximum of about 26 KW at 14° ATDC. These trends agree well with the results published by Zeleznik and McBride (86).

Figs. 6.18 and 6.19 show the variation of the flame propagation parameters during the combustion phase. For the assumed conditions, the charge is completely burnt at a crank angle of 14° ATDC. The volume and mass fraction curves agree with previously reported
results (86) and indicate a "delay period" of about 15°. The flame radius at this crank angle is approximately 6 mm showing that the flame front is by no means stationary during the delay period.

This can be seen more clearly in Fig. 6.19 which shows the laminar and turbulent burning velocities in addition to the total (or measured) flame speed. The laminar and turbulent burning velocities increase continuously during combustion whilst the total flame speed increases rapidly during the early phase, reaching a maximum of about 24 m/s and then reducing steadily for the remainder. The value attained at the end of combustion is equal to the corresponding turbulent burning velocity. The reduction in flame speed during the latter part of the combustion process is due to the increasing burnt volume fraction and unburnt charge density restricting the expansion of the burning charge.

Fig. 6.20 shows the effect of equivalence ratio on the combustion rate and exhaust emissions. Flame travel angles, based on the interval between flame radii of 10 to 50 mm and mass burn angles, based on 2 to 90% burnt mass fractions give very similar values with maximum angles at about 1.1 equivalence ratio. This agrees well with experimental results and is a significant improvement on the predictions obtained by Phillipps and Orman (9) using the Mallard and le Chatelier laminar burning velocity theory.

The effect of equivalence ratio on emissions is in good agreement with measured experimental values. The nitric oxide emissions, based on peak concentration yields a maximum concentration of about 5300 p.p.m. at an equivalence ratio of 0.85 (approximately 17:1 air to fuel ratio).

The effect of engine speed on the flame propagation rate is shown in Fig. 6.21. It can be seen that the flame speed, based on propagation between 10 and 50 mm radii, increases linearly with engine speed. The flame travel angles also increase with engine speed although the rate of increase is less at higher engine speeds.
Fig. 6.22 indicates the effect of ignition timing on the power output (or i.m.e.p. or indicated thermal efficiency) and the flame travel angles. This graph shows that the optimum ignition angle for the three parameters is different as one would expect. For the power output, the optimum occurs at approximately 28° BTDC, this angle being dictated by the negative compression and positive expansion work values. The flame travel angles on the other hand will be at a minimum when the combustion period defining the parameter is near to TDC due to the increased charge temperature and hence higher laminar burning velocity.

The effect of ignition timing on the pressure time diagram is shown in Fig. 6.23. The trends predicted are as expected, showing an increased maximum pressure with more advanced ignition timing whilst with retarded ignition, a small reduction in pressure immediately after TDC is obtained due to the effect of the pressure delay period.

Finally, Fig. 6.24 shows the effect of volumetric efficiency on the pressure time diagram. Note that a constant residual mass has been assumed rather than a constant residual mass fraction since this is more representative of real engine operation. For this condition, the peak pressure is increased for a larger value of volumetric efficiency whilst the combustion period, indicated by the peak pressure crank angle, is reduced slightly as the volumetric efficiency is increased. These predictions agree well with empirical observations although it must be emphasised that a different assumption regarding the residual mass could significantly affect the results. Note also that for these results, a constant turbulent multiplier is used rather than a value proportional to volumetric efficiency (see Section 6.2.4).

In conclusion, the predictions obtained from the simulation program agree very well with theory and experimental and simulation results which have been reported by other workers. In particular, the flame propagation, ignition timing and equivalence ratio results suggest that the model should be sufficiently accurate for predicting the effect of combustion chamber design on the flame propagation and general performance parameters. These predictions are presented and discussed in the next chapter.
READ ENGINE DETAILS, OPERATING CONDITIONS AND OTHER DATA

INITIALISE VARIABLES, EVALUATE VALUE OF CONSTANT

CALCULATE COMBUSTION CHAMBER DIMENSION USING SUBROUTINE 'DIM'

CALCULATE MIXTURE COMPOSITION AND PROPERTIES USING SUBROUTINE 'COMPA'

CALCULATE BURN CHARGE COMPOSITION AND PROPERTIES USING SUBROUTINE 'COMPA'

CALCULATE BURNED VOLUME AND MASS FRACTIONS AND TURBULENT BURNING VELOCITIES AND FLAME SPEED

Determine state properties at end of constant volume combustion process

Determine state properties at end of interval using subroutine 'COMPA'

Evaluate laminar and turbulent burning velocities and flame speed

Evaluate burned volume at end of interval

Evaluate burned surface area using segments 'FLAM' and 'SAVL'

Evaluate flame radius and burnt surface area using segments 'FLAM' and 'SAVL'

Evaluate flame radius and burnt surface area using segments 'FLAM' and 'SAVL'

Evaluate burnt charge composition and properties using subroutine 'COMPA'

Determine state properties at the end of interval using subroutine 'COMPA'

Determine adiabatic flame temperature using energy balance

Determine adiabatic flame temperature using energy balance

Determine expansion values of variables at the end of interval using subroutine 'COMPA'

Determine expansion values of variables at the end of interval using subroutine 'COMPA'

Determine expansion values of variables at the end of interval using subroutine 'COMPA'

Output calculated results in tabular form on line printer

Another simulation?

Yes

No

Another simulation?

Yes

No

Another simulation?

Yes

No

Another simulation?

Yes

No

Another simulation?

Yes

No

Another simulation?

Yes

No

Another simulation?

Yes

No

Another simulation?

STOP

NAMELIST

A Area
T Temperature
V Volume
θ Crank Angle

SUBSCRIPTS

1 Start of Interval
2 End of Interval
b Burnt Zone
i E.V.O. Angle
l L.E.O. Angle
S Ignition Angle
t Total

FIG. 6.1 FLOW DIAGRAM FOR COMPUTER SIMULATION MODEL 'COMPA'
Fig 6.2 COMPRESSION PROCESS IDEALISATION
Fig 6.3  COMBUSTION PROCESS IDEALISATION
Fig 6.4 LAMINAR BURNING VELOCITY CORRELATION (EQU 6.21)

Fig 6.5 PREDICTED EFFECT OF $T_u$ ON LAMINAR BURN VELOCITY
**Fig 6.6** COMBUSTION CHAMBER DESIGNS MODELLED (TYPE 1-4)

**TYPE 1**
Cylindrical or Disc

**TYPE 2**
Vee Double Squish

**TYPE 3**
Parallel Double Squish

**TYPE 4**
Single Squish Plug Side
COMBUSTION CHAMBER DESIGNS MODELLED (TYPE 5-8)
Fig 6.8 COMBUSTION CHAMBER DESIGNS MODELLED (TYPES 9 & 10)

Fig 6.9 VALVE AND SPARK PLUG RECESSES MODELLED
Heart Shaped Chamber Approximation

Combined with Types 1 or 6

Fig 6.10 EXAMPLE OF CHAMBER SHAPES USING RECESS AND MAIN CHAMBER MODEL COMBINATION

Fig 6.11 SPHERICAL FLAME FRONT IN DISC TYPE CHAMBER
Fig 6.12 SIMPSONS RULE FOR NUMERICAL INTEGRATION

Length of wall contact A (B for $A_b$ ordinate)

Length of flame front A)B for $A_f$ ordinate

Burnt Zone

Area of burnt zone for $V_b$ ordinate

Divide into even number of strips

Fig 6.13 SIMPSONS RULE APPLIED TO A DISC TYPE CHAMBER
INITIALISE DATA. CALCULATE MAX. AND MIN. FLAME RADI.

FLAME PLAN VIEW PROFILE CIRCULAR?

CALCULATE VOLUME AND SURFACE AREA USING SPHERICAL EXPRESSIONS.

YES

RF > (H-PH)?

DIVIDE CHAMBER INTO STRIPS BASED ON H

DIVIDE CHAMBER INTO STRIPS BASED ON RF

YES

DETERMINE AXIAL ORDINATE DIMENSIONS AND SQUISH FACE LOCATION

EVALUATE RADIAL COMPONENT OF THE FLAME RADIUS.

SET RC = RB
RS = RSB

B.I.P / H.H CHAMBER DESIGN?

NO

YES

DETERMINE FLAME VOLUME AND AREA USING SIMPSONS RULE

CALCULATE RECESS VOLUME/AREAS USING SUBROUTINE VALVE.

YES

ALL AXIAL ORDINATES CONSIDERED?

CALCULATE FLAME AREA AND LENGTHS FOR THE ORDINATE LOCATION

YES

EVALUATE BURNT SURFACE AREA FOR SQUISH FACE.

RETURN

START

* SYMBOLS USED ARE AS DEFINED IN SECTION 6.3

FIG. 6.14 FLOW CHART FOR GEOMETRICAL SIMULATION MODEL
(SUBROUTINE "SAVOL")
INITIALISE DATA SET
RI = 1.0 mm AND
RFC = RFI

CALCULATE VR USING SUBROUTINE 'SAVOL'

- Ve

FB - VR

0

Ve

RFC = RFC - RI

CALCULATE VR USING SUBROUTINE 'SAVOL'

FB - VR

0

Ve

RFC = RFC + RI

CALCULATE VR USING SUBROUTINE 'SAVOL'

FB - VR

0

Ve

COMPUTE MORE ACCURATE RFC USING A LINEAR INTERPOLATION

RI = 0.1 mm?

YES

RF = RFC

NO

RETURN

FIG. 6.15 FLOW CHART SHOWING LOGIC FOR EVALUATING FLAME RADIUS (FUNCTION 'FLARAD')

NAMELIST
RF ACTUAL FLAME RADIUS
RFC CURRENT ESTIMATE OF RF
RFI INITIAL ESTIMATE OF RF
RI RADIUS ITERATION_INCREMENT
Vb SPECIFIED BURNED VOLUME
VR CALCULATED BURNED VOLUME BASED ON RFC
Fig 6.16 PREDICTED STATE PROPERTIES DURING COMP/EXPANSION

Fig 6.17 WORK AND HEAT TRANSFER RATES DURING CYCLE
**Fig 6.18** BURNT VOLUME AND MASS FRACTION CURVES

**Fig 6.19** PREDICTED FLAME SPEEDS DURING COMBUSTION
Fig 6.20 EFFECT OF EQUIVALENCE RATIO ON COMBUSTION RATE AND EMISSIONS
Fig 6.21 EFFECT OF ENGINE SPEED ON FLAME SPEED AND FLAME TRAVEL ANGLES

Fig 6.22 EFFECT OF IGNITION TIMING ON POWER OUTPUT AND FLAME TRAVEL ANGLES
Fig 6.23 PREDICTED EFFECT OF IGNITION TIMING

Fig 6.24 P-θ DIAGRAM FOR VARIOUS VOLUMETRIC EFF'S
CHAPTER 7

COMPUTER MODEL PREDICTIONS
CHAPTER 7

7.1 Introduction

In the previous chapter, computer simulation models have been described which allow the effects of the major design variables to be readily assessed. In this Chapter, predictions obtained from these models are presented, indicating the effect of the design variables on the rate of combustion and other performance parameters. In addition, the results presented illustrate the utility of the models as combustion chamber design tools.

Previous work, reviewed in Section 1.2, has shown that a great deal of information regarding the overall performance of a combustion chamber design can be obtained from considering purely geometrical parameters such as burnt volume and flame surface area plots. A number of plots of this type generated using the geometrical model described in Section 6.3, are therefore presented and discussed in Section 7.2.

Section 7.3 shows the effect of the chamber design on the combustion parameters as modelled by the simulation program COMPSIMP. Several "optimised" fast burn designs are compared to indicate the relative potential of combustion chamber design improvements on the weak mixture performance.

Section 7.4 presents results which indicate the magnitude of the squish velocities which occur under actual firing conditions. These are compared to motored engine (non-firing) predictions thereby showing the effect of the combustion.

A comparison between the results obtained from the simulation model and those obtained experimentally appears in Section 7.5. In addition to the experimental results obtained in this study (reported in Chapter 5), the experimental results reported in the literature by two other independent workers are also analysed. Note that this type of comparison is of special interest since a major assumption in the combustion model is that the squish velocities do not have any effect on the turbulence or flame profile.
Before presenting the results obtained from the models, it should be pointed out that the bore/stroke ratio and the number of cylinders for a given engine capacity are assumed to be constant for the purposes of this chapter. In fact, a "square design" giving a bore/stroke ratio of 1.0 is assumed.

The reason for ignoring these parameters is that their values are mainly determined by other factors such as piston speed, power output, engine dimensions etc. However, bore/stroke ratio and the number of cylinders have a major effect on combustion rate, general performance, emissions and heat transfer and therefore this should be borne in mind.

It is easy to show, using simple geometrical expressions, that for reduced flame travel distance and chamber surface area, the bore/stroke ratio should be significantly less than 1, i.e. large strokes. Also, a large number of cylinders should be used for minimum flame travel distance whilst the opposite is true for minimum chamber surface area. Whilst there may be a gradual move towards smaller bore/stroke ratios in the coming years, the reduction is likely to be small.

7.2 Geometrical Model Predictions

Previous work which has been reviewed in Section 1.2 has shown that a guide to the suitability of a combustion chamber design can be obtained by considering geometrical parameters such as burnt volume and flame surface area.

Basically, for good overall performance, the burnt volume during the early part of the flame propagation should be as high as possible and the flame front area/flame radius curve should be as smooth as possible to ensure smooth engine running.

In addition, by plotting burnt volume and flame front area curves for various chamber designs, the relative merits of each of the designs can be assessed. By assuming similar flame speeds for each of the chambers, the burnt volume/flame radius curves can be considered to be a scaled version of the burnt volume/time curve and hence burnt mass...
fraction/time curves. Therefore, whilst the shape of the latter curves will be different, the relative order of the chambers will not be altered.

Figs. 7.1 and 7.2 shows the burnt volume and flame front area plotted against flame radius for a disc chamber and a 50% squish area bowl in piston design for several radial spark plug locations. As one would expect for these solid of revolution designs, moving the spark plug from the centre (RS = 0) towards the cylinder wall reduces the burnt volume for a given flame radius. By comparing the disc with the bowl in piston, it is clear that the burnt volume is highest for the latter due to the steeper combustion chamber allowing the flame front to retain a hemispherical profile during the first half of its travel.

The flame front area/flame radius curves show a greater sensitivity to chamber shape since unlike the smooth burnt volume plots, large changes in curvature occur when the flame contacts the chamber wall or piston face. Whilst the maximum flame front areas are similar for both designs, the curves for the bowl in piston are much smoother.

Fig. 7.3 shows the burnt volume/flame radius curves for a 50% sector type 5 design and a disc with dual ignition for a range of spark plug locations. In contrast to the previous cases, the highest burnt volume does not occur for central ignition but for an ignition location at about 15 mm from the centre for the designs being considered.

A major improvement for the dual ignition design is that the volume burnt during the early combustion phase is twice that for the single ignition case which would result in reduced "delay" period.

Fig. 7.4 shows the burnt zone heat transfer surface area as a function of flame radius for the disc type combustion chamber. The figures indicate the contact area between the burnt zone and the cylinder wall, piston crown, cylinder head and the % total surface area. Whilst the profile of these curves is very dependent on chamber design, the total surface area of the combustion chamber (at 100% burnt volume) is less affected.
Table 7.1 shows the total surface area of several combustion chambers with the piston at TDC. For the cases considered, compared to a cylindrical disc chamber, a 50% squish hemispherical head or bowl gives about a 3% reduction whilst a 50% squish cylindrical bowl in piston gives an 8% increase, a 50% sector type squish results in a 13% increase and a deep valve recess design gives an 11% increase.

<table>
<thead>
<tr>
<th>CHAMBER DESIGN, BORE = STROKE = 70 mm, COMPRESSION RATIO = 8:1</th>
<th>TOTAL SURFACE AREA cm²</th>
<th>% CHANGE REL' TO DISC</th>
</tr>
</thead>
<tbody>
<tr>
<td>DISC CHAMBER TYPE 1</td>
<td>98.96</td>
<td>-</td>
</tr>
<tr>
<td>50% SQUISH B.I.P. TYPE 6 WITH CYLINDRICAL BOWL</td>
<td>107.21</td>
<td>+ 8.34</td>
</tr>
<tr>
<td>50% SQUISH HEMISPHERICAL HEAD TYPE 10</td>
<td>95.62</td>
<td>- 3.38</td>
</tr>
<tr>
<td>50% SQUISH SECTOR TYPES 4 OR 5</td>
<td>111.56</td>
<td>+12.73</td>
</tr>
<tr>
<td>50% SQUISH VALVE RECESS DESIGN, 35 mm DIA. x 10 mm DEEP RECESSES</td>
<td>109.96</td>
<td>+11.12</td>
</tr>
</tbody>
</table>

TABLE 7.1 COMPARISON OF TOTAL SURFACE AREA AT TDC FOR SEVERAL COMBUSTION CHAMBER DESIGNS

To reduce heat losses and hydrocarbon emissions, the surface area should be reduced to a minimum. Although the differences between the chambers would be reduced as the piston moves away from the TDC position, it is clear that a disc or hemispherical type design is more desirable than the concentrated squish designs. (Note that this simple analysis does not take into account the effect of squish velocities and thermal quench layers).
Of the four designs considered so far, assuming optimum spark plug positioning, the dual ignition disc chamber would give the highest combustion rate with the bowl in piston in second place. However, other parameters such as the percentage squish area, pumping clearance, compression ratio and spark plug axial location would tend to alter the relative merits of each type of combustion chamber. The effect of these parameters including their interaction are shown in Figures 7.5 through 7.12.

Figure 7.5 illustrates the effect of spark plug axial location for a 50% B.I.P. design with central ignition. For this case, increasing the spark plug projection increases the burnt volume for a given flame radius. However, it should be noted that this is an ideal case for increased spark plug projection due to the deep bowl (c 18 mm) and the central ignition. For offset ignition or reduced squish area, the effect would have been reduced. For a disc chamber for example, the effect is negligible except at very small flame radii where a small improvement occurs. Generally, the spark plug projection would be determined by other factors such as electrode temperature, avoiding quench zones etc.

Figure 7.6 shows the importance of bumping clearance for the same chamber design. It can be seen that increased bumping clearance reduces the burnt volume at the larger flame radii. Whilst large bumping clearances greatly affect the burnt volume, small increases in the normal range of 1 to 2 mm would be less significant. The effect of bumping clearance at crank angles other than TDC would be less than that indicated by Fig. 7.6.

The graphs presented so far have assumed that the piston is at TDC. Fig. 7.7 shows that the piston movement can cause substantial increases in the flame radius to achieve a given burnt volume. However, it should be pointed out that the central ignition B.I.P. case shown would be more affected by piston position than say a disc chamber or a B.I.P. with offset ignition.

Therefore, whilst analysing the burnt volume and flame front surface area curves at TDC is a reasonable approach, the effect of piston movement should not be completely ignored. However, the piston movement prior to TDC would not normally affect the analysis whilst for
M.B.T. ignition timing, the combustion would normally be completed at about 12\textdegree\ ATDC where the piston movement is almost negligible.

It is clear from Figs. 7.1 through 7.7 that it is difficult to compare combustion chambers directly using burnt volume versus flame radius plots due to the large effect of spark plug radial location. Therefore, to allow comparisons to be more readily obtained, it was arbitrarily decided to compare the designs based on the flame radius at which 80\% burnt volume is achieved. Whilst such a criterion is not ideal due to the variation in burnt volume/flame radius profiles, it does give a reasonable indication of the relative merits of the designs being considered.

Figures 7.8 through 7.12 all use this criterion. The effect of spark plug radius for six different designs is shown in Fig. 7.8, indicating that the dual ignition designs with optimum ignition timing give the minimum value of flame radius and hence the fastest combustion.

Figures 7.9 to 7.11 show the effect of squish area coverage for different values of spark plug radial position and for four different chamber designs. It can be seen that the amount of squish area which results in minimum flame radius is in the range of 40 to 65\% in all cases, this amount of squish area being typical of production engine combustion chambers.

The interaction between squish area and spark plug radius is shown by the minima locus drawn on Figs. 7.9 and 7.10. This shows that the "optimum" squish area increases with spark plug radius.

The effect of increasing the squish area coverage is a fairly gradual change for squish areas between zero and the "optimum" with a much greater rate of increase at values above 60\% due to the undesirable increase in chamber height. The hemispherical head designs shown in Fig. 7.11 are the least sensitive of those considered to increased squish area between zero and about 50\% but have the largest rate of increase in the range above 60\%. The reason for this is that the spark plug to piston crown height is greater for the hemispherical head which makes increases in chamber heights less desirable than for many other types of chamber.
The effect of compression ratio on the radius for 80% volume burnt is shown in Fig. 7.12 for a dual ignition B.I.P. design. This graph shows quite clearly that the "optimum" squish area is increased as the compression ratio increases as one would expect since the controlling factor is the chamber height which is less for higher compression ratios. This type of graph helps to understand why the May "Fireball" (62) combustion chamber combines a large squish area with a high compression ratio.

To conclude this section, it has been shown that the factors mainly affecting the burnt volume and flame front area are the spark plug radial location, the chamber shape and the squish area. There is a very strong interaction between design parameters which suggests that an iterative approach is needed to obtain an optimum chamber design.

The optimum amount of squish, based purely on geometrical considerations (i.e. ignoring squish velocities, quench layers, detonation etc) is dictated by the need to make the chamber depth approximately equal to its effective radius, i.e. to make it more compact. The squish area to achieve this is normally in the region of 50%. However, a reduction in compression ratio, bore/stroke ratio, spark plug radius or projection would reduce the optimum amount.

7.3 Combustion Model Predictions

7.3.1 Introduction

The effect of combustion chamber design parameters on the flame front area and burnt volume has been described in the previous section. In this section, predictions showing the effect that these parameters have on the combustion rate and other performance variables are presented. The predictions presented have been obtained using the computer simulation model COMPSIMP described in Chapter 6.

To allow comparisons to be made, the engine parameters and operating conditions have been maintained at fixed values for the majority of the predictions. The values chosen were as follows:
Engine

Bore ........................................ 70 mm
Stroke ........................................ 70 mm
Con Rod Length .............................. 120 mm
Compression Ratio ........................... 8.0:1
I.V.C./E.V.O. ................................. 42° ABDC/42° BBDC

Operating Conditions

Engine Speed ................................. 2000 RPM
Ignition Timing .............................. 30° BTDC
Equivalence Ratio ............................ 1.1
Volumetric Efficiency ....................... 75%
Residual Gas Mass Fraction .................. 7%
Atmospheric Pressure ......................... 1.0 Bar
Atmospheric Temperature ..................... 292 K
Relative Humidity ............................ 0.7%

7.3.2 Effect of Combustion Chamber Design Parameters

In this subsection, results are presented to show the effect of the major combustion chamber design parameters. In most cases, these results can be compared directly with those presented in the previous section.

Figures 7.13 and 7.14 show the predicted effect that spark plug radial location has on the cylinder pressure diagram for the disc and bowl in piston design respectively. It can be seen that moving the spark plug towards the cylinder wall (i.e. increasing RS) reduces both the peak pressure and peak rate of pressure rise. Also the crank angle at which these maximum values occur increases with increasing values of RS.

Ricardo (1) has shown that for high efficiency, the pressure rise rate should not exceed about 2.4 Bar (35 p.s.i.) per degree at 2000 RPM. The work of Janeway (4), and Taub (54) has indicated that to avoid engine "roughness", the rate of pressure rise \( \frac{dP}{d\theta} \) and the rate of change of pressure rise rate \( \frac{d^2P}{d\theta^2} \), should be kept as low as possible. Therefore, taking these factors into account, it is clear
that central ignition, whilst giving a high combustion rate, would probably result in a very rough running engine of less than optimum efficiency (at the assumed operating conditions).

Comparing the two designs, the bowl in piston clearly gives the highest rate of combustion for any given value of RS. However, the bowl in piston also gives the highest values of \( dP/d\theta \) and \( d^2P/d\theta^2 \). To achieve maximum pressure rise rates of about 2.5 Bar/deg, the radial spark plug position would be about 10 mm for the disc and about 20 mm for the B.I.P. However, for the conditions considered, the ignition timing of \( 30^\circ \) BTDC is slightly over advanced. This would increase the pressure rise rate. An optimised ignition timing would therefore reduce the rate which in turn would allow more central locations to be considered. Note also that the B.I.P. would require less ignition advance than the disc chamber.

It must be emphasised however that the computer model would tend to over-estimate the peak pressure rise rate since a spherical flame front is assumed. In practice, distortion of the flame front would tend to smooth out the high rates of change of pressure rise which occur with the central ignition cases shown.

Figures 7.15 and 7.16 show burnt mass fraction plotted against crank angle for four chamber designs and four values of spark plug radial location. These graphs confirm the previous results by showing that the spark plug location has a major effect on the rate of combustion. For the solid of revolution type 1 and 6 designs, central ignition gives the faster combustion rate as expected, the difference between central and side ignition being about 18 degrees at the end of combustion (100% burnt mass fraction). Fig. 7.15 also shows that the shape of the curves is different for the disc and B.I.P. designs, the disc giving the highest mass burn rate at the end of the combustion period.

Fig. 7.16 shows that for a sector type squish design and dual ignition disc chamber, a spark plug radial location of about 10 mm would give the faster rate overall. A larger value of RS would give the fastest rate during the early part of the combustion process and it is clear from this graph that unlike the previous cases, the optimum spark location is more of a compromise.
Comparing the four designs, it can be seen that the dual ignition design gives the highest combustion rate with the B.I.P. in second place. These results agree qualitatively with the geometrical results presented in Figs. 7.1 through 7.3.

Fig. 7.17 indicates the effect of squish area on both the pressure and burnt mass fraction curves for a B.I.P., central ignition, design. This shows that the fastest combustion rate is achieved with a squish area of about 60%, whilst a large squish area of 80% results in a large decrease in combustion rate. As expected, a squish area of about 60% gives a substantial improvement in the combustion rate during the first half of combustion.

The effect of compression ratio on the pressure and burnt mass fraction curves is indicated by Fig. 7.18. This graph shows that compression ratio has a large effect on both the cylinder pressure and burnt mass fraction. For example, increasing the compression ratio from 8:1 to 10:1 results in about a 3 degree reduction in the combustion period and 15 Bar increase in maximum cylinder pressure.

The graphs presented so far have been plotted against crank angle, showing the variation in the parameters during the combustion phase. However, this type of presentation is not ideal for indicating the effect of design parameters or for comparing chamber designs. As with previous results, it is better to plot specified performance parameters against the design parameters. Such plots are shown in figures 7.19 through 7.21.

The first two of these graphs shows the effect of spark plug radial location for seven chamber designs. Fig. 7.19 indicates the effect on the 50% burnt mass fraction crank angle, confirming the optimum spark locations discussed previously. The graph shows that the fastest rates of combustion are achieved using dual ignition with the B.I.P. giving the fastest rate of combustion for all seven chambers. Considering only the single ignition designs, the bowl in piston has the fastest combustion rate for spark plug radii less than about 15 mm with the sector type squish design being best for larger values of RS.
Fig. 7.20 shows the effect of the same variables on the maximum pressure rise rate. It can be seen that whilst in general, chambers giving the highest rate of combustion have the highest pressure rise rate, this is not always the case. Also, the dual ignition designs have the higher pressure rise rates in most cases. A chamber design worthy of special comment is the dual ignition hemispherical head which has the second highest rate of combustion but a relatively low pressure rise rate for spark plug radii of less than about 20 mm.

The effect of squish area and chamber design on the rate of combustion is shown in Fig. 7.21. Since increasing the squish area tends to improve the combustion rate during the early part of the process and reduce the rate during the latter part, both 50% and 90% burnt mass fraction crank angle values are plotted.

This graph shows that the amount of squish area resulting in minimum mass burnt crank angle is in the range of 40% to 60% based on 50% burnt mass whilst squish areas of about 20% to 60% give minimum crank angle at 90% burnt mass. It is clear that the bowl in piston arrangement is the most suited to large amounts of squish area with the hemispherical head preferring no squish area. The dual ignition bowl in piston gives the highest combustion rate of all the five designs considered with the sector type squish being the slowest. The results of Fig. 7.21 are in good agreement with the geometrical results of Figs. 7.9 through 7.11 presented earlier.

7.3.3 Comparison of Specific Combustion Chamber Designs

In the previous sub-section, results were presented showing the general effect of combustion chamber design parameters. However, it should be emphasised that the design parameters are highly interactive such that an optimum design can only be chosen by using an iterative approach. For example, the optimum squish area is affected by spark plug location, chamber shape, compression ratio, bore/stroke ratio etc.

In a real engine, the spark plug cannot be located anywhere within the chamber due to valves, squish areas etc and its position is constrained by other considerations such as residual gas scouring, electrode cooling etc. Also, the compression ratio is limited by
detonation and fuel octane requirement considerations although it should be noted that the chamber design can have a considerable effect on the highest usable compression ratio. The chamber shape is also dictated by the overall engine design such that valve positions and orientation may restrict the chamber shapes that can be considered.

Bearing these points in mind, five chamber designs have been selected for the purpose of a comparison to indicate the relative merits. The specification of the five designs is shown in Table 7.2.

The disc chamber was chosen since it is the simplest shape whilst the sector type, 30% squish design is typical of many production engines. Three fast burn designs consisting of a 50% squish B.I.P., together with a dual ignition disc and a dual ignition, zero squish area hemispherical head arrangement complete the range of designs.

A compression ratio of 9:1 has been chosen since this is typical of present day production engines. Although higher compression ratios may be possible with lean mixtures and increased combustion chamber compactness, pressure to decrease the lead content and hence octane rating will tend to offset such increases.

Spark plug radial positions of 10 mm have been specified for all the chambers with the exception of the sector type squish design where 20 mm has been chosen. The reason for using 10 mm is that it is possible to achieve this value in practice whereas a more central position would be difficult to achieve due to the position of the valves. Also, it has been shown that a central position is not desirable due to the high rates of change of pressure and pressure rise rates which would cause unacceptable levels of roughness. A value of 10 mm is about optimum for the dual ignition cases although in practice it may be necessary to use slightly larger values of RS. A value of 20 mm was chosen for the sector type since for this design, the valve positions are more restricted and would need to be moved towards the spark plug side of the chamber.

The five chamber designs have been designated by the letters A to E to permit easy recognition of results. Before presenting results comparing the chambers under typical operating conditions, it must be
<table>
<thead>
<tr>
<th>CHAMBER DESIGN DESIGNATION</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
</tr>
</thead>
<tbody>
<tr>
<td>CHAMBER TYPE (See Fig. 6.6-6.8)</td>
<td>1 DISC</td>
<td>5 SECTOR</td>
<td>6 B.I.P.</td>
<td>7 DISC</td>
<td>9 HEMI' HEAD</td>
</tr>
<tr>
<td>SQUISH AREA SQ %</td>
<td>0</td>
<td>30</td>
<td>50</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>TYPE OF IGNITION</td>
<td>SINGLE</td>
<td>SINGLE</td>
<td>SINGLE</td>
<td>DUAL</td>
<td>DUAL</td>
</tr>
<tr>
<td>SPARK PLUG RADIAL LOCATION RS mm</td>
<td>10</td>
<td>20</td>
<td>10</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>SPARK PLUG AXIAL LOCATION PH mm</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>BUMPING CLEARANCE BC mm</td>
<td>-</td>
<td>1.0</td>
<td>1.0</td>
<td>-</td>
<td>1.0</td>
</tr>
<tr>
<td>COMPRESSION RATIO CR</td>
<td>9.0</td>
<td>9.0</td>
<td>9.0</td>
<td>9.0</td>
<td>9.0</td>
</tr>
<tr>
<td>BOWL RADIUS RR mm</td>
<td>-</td>
<td>-</td>
<td>5.0</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>BOWL AXIS TO SPARK PLUG RSB mm</td>
<td>-</td>
<td>-</td>
<td>10</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>VALVE AND SPARK PLUG RECESSES</td>
<td>NONE</td>
<td>NONE</td>
<td>NONE</td>
<td>NONE</td>
<td>NONE</td>
</tr>
<tr>
<td>BORE mm</td>
<td>70</td>
<td>70</td>
<td>70</td>
<td>70</td>
<td>70</td>
</tr>
<tr>
<td>STROKE mm</td>
<td>70</td>
<td>70</td>
<td>70</td>
<td>70</td>
<td>70</td>
</tr>
</tbody>
</table>

**TABLE 7.2** SPECIFICATION OF COMBUSTION CHAMBERS USED FOR COMPARISON
emphasised that the chambers are not "optimised" designs but have been chosen because they represent different alternatives which are practically suitable and are sufficiently different to show the importance of chamber shape. In practice, an optimised design can only be achieved by knowing all the constraints and by considering the full range of operating conditions.

Fig. 7.22 shows the relative performance of the chamber designs in terms of the cylinder pressure and burnt mass curves. The upper plot reveals that the dual ignition designs (D and E) give the highest pressures although the bowl in piston (C) also results in a significant increase over the disc (A) and sector chambers (B). The burnt mass fraction plot shows good agreement with the pressure diagram and indicates that the combustion period for the dual ignition designs is approximately 5° faster than for the disc and sector whilst the bowl in piston is about 3° faster. The sector designs give a slight improvement over the disc chamber during the main combustion period but requires longer to consume all the charge due to the increased spark plug radii.

Figures 7.23 and 7.24 show the effect of chamber design over the range of equivalence ratios. Note that a constant ignition timing of 30° is assumed throughout (this being over-advanced except for the weak mixtures range). The plot of maximum pressure agrees with the previous graph regarding the relative effect of the chambers. The plot of pressure rise rate shows that the dual ignition disc results in the highest rate of about 5 Bar/deg at an equivalence ratio of 1.1. Although the pressure rise rate with dual ignition would be too high for engines operating on slightly rich mixtures, it can be seen that the pressure rise rate at weak mixtures for this design is of the right order.

Fig. 7.24 shows the predicted crank angles for 50% and 90% burnt mass fractions over a range of equivalence ratios. This graph shows that the dual ignition hemispherical head has the highest combustion rate. For the same crank angle of 360° for 50% burnt mass fraction the difference in equivalence ratio between the designs varies from about 0.77 to 0.91 (19.1 to 16.2 air to fuel ratio). This suggests that the fast burn designs should be much more suitable when operated at lean mixtures.
Fig. 7.25 shows a plot of indicated power output versus ignition timing for the five chamber designs. This graph reveals that the fast burn designs require less ignition advance for maximum power output, i.e. fast burn results in smaller M.B.T. settings. As shown by the graph, the dual ignition hemispherical head requires least ignition advance at $21^\circ$ BTDC with the single ignition disc chamber needing $25^\circ$ advance.

Although the difference is only $4^\circ$ for the 0.95 equivalence ratio, 2000 RPM case being considered, this is still significant since the difference should be greater at weaker mixtures and higher engine speeds. Reduced ignition advance is a very important requirement for lean burn engines since, due to the higher charge temperature, the charge is more readily ignited. This effect has been shown by Quader (6) and as a consequence, the fast burn designs should have a reduced probability of complete misfire (i.e. no ignition).

Fig. 7.25 also reveals the difference in power output between the five chambers. It can clearly be seen that whilst the difference between designs A to C is small, the hemispherical head (design E) is substantially better. In terms of percentage improvement, design E is approximately $5\%$ better than designs A and B and $4\%$ better than C and D.

The main reason why the hemispherical head is superior to the other designs is mainly due to the fact that the heat losses are much less as a result of the reduced chamber surface area. (A comparison between surface areas for several designs has been presented in Table 7.1). A second, less significant reason is that the peak power output is increased slightly as the combustion rate is increased (at the specified operating conditions) as proved by the results for the single and dual ignition disc designs.

Table 7.3 compares the performance of the five chamber designs at a fixed ignition timing of $30^\circ$ and at M.B.T. ignition timing. The operating conditions are for an engine speed of 2000 RPM, equivalence ratio of 0.95 and 75% volumetric efficiency.

Examining the cylinder pressure results, it can be seen that the fast burn designs (C,D,E) give the highest pressures with design B being the lowest. At M.B.T., the maximum pressure occurs at about $12^\circ$ ATDC, this
<table>
<thead>
<tr>
<th>PARAMETERS</th>
<th>UNITS</th>
<th>IGNITION TIMING SET AT FIXED VALUE OF 30° BTDC</th>
<th>IGNITION TIMING SET AT MINIMUM ADVANCE FOR BEST TORQUE (M.B.T.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ignition Timing $\theta_s$</td>
<td>Deg BTDC</td>
<td>30 30 30 30 30</td>
<td>25 24 24 22 21</td>
</tr>
<tr>
<td>Max' Cylinder Pressure $P$</td>
<td>Bar</td>
<td>54.6 53.0 57.1 58.4 58.7</td>
<td>50.3 47.4 52.5 53.3 53.1</td>
</tr>
<tr>
<td>Max' Pressure Crank Angle $\Theta_p$</td>
<td>Deg CA</td>
<td>369.5 368.5 366.5 364.5 364.0</td>
<td>374.0 374.0 371.0 371.5 371.0</td>
</tr>
<tr>
<td>Max' Pressure Rise Rate $\Delta p$</td>
<td>Bar/Deg</td>
<td>2.75 3.0 3.6 4.3 4.0</td>
<td>2.3 2.55 3.1 3.95 3.4</td>
</tr>
<tr>
<td>Max' Burnt Gas Temp. $T_b$</td>
<td>$^\circ$K</td>
<td>2495 2500 2527 2534 2544</td>
<td>2456 2461 2496 2499 2505</td>
</tr>
<tr>
<td>Charge Temp. At E.V.O. $T_{EVO}$</td>
<td>$^\circ$K</td>
<td>1225 1216 1212 1209 1281</td>
<td>1240 1241 1231 1230 1298</td>
</tr>
<tr>
<td>2% Mass Burnt Angle $m_{2,s}$</td>
<td>Deg</td>
<td>17.0 17.0 16.9 16.8 17.0</td>
<td>16.8 16.4 16.0 15.0 15.0</td>
</tr>
<tr>
<td>50% Mass Burnt Angle $m_{50,s}$</td>
<td>Deg</td>
<td>31.4 30.6 29.2 27.2 27.7</td>
<td>21.0 20.4 27.8 26.7 25.0</td>
</tr>
<tr>
<td>90% Mass Burnt Angle $m_{90,s}$</td>
<td>Deg</td>
<td>38.3 39.6 34.2 32.8 32.5</td>
<td>37.4 39.0 33.6 31.5 30.4</td>
</tr>
<tr>
<td>10 mm Flame Travel Angle $\theta_{10,s}$</td>
<td>Deg</td>
<td>18.8 18.7 18.5 18.5 18.7</td>
<td>17.8 17.7 17.7 17.3 17.3</td>
</tr>
<tr>
<td>30 mm Flame Travel Angle $\theta_{30,s}$</td>
<td>Deg</td>
<td>30.9 31.0 31.4 31.4 31.4</td>
<td>31.6 29.3 29.3 30.0 30.0</td>
</tr>
<tr>
<td>Power Output $W$</td>
<td>KW</td>
<td>3.87 3.86 3.87 3.82 4.00</td>
<td>3.88 3.87 3.91 3.90 4.06</td>
</tr>
<tr>
<td>Heat Loss to Chamber $Q$</td>
<td>KW</td>
<td>2.49 2.54 2.47 2.57 1.98</td>
<td>2.34 2.38 2.35 2.38 1.83</td>
</tr>
<tr>
<td>Maximum Nitric Oxide NO</td>
<td>P.P.M.</td>
<td>5006 5070 5323 5366 5567</td>
<td>4800 4711 5044 5021 5144</td>
</tr>
</tbody>
</table>
figure agreeing well with experimental observations. The pressure rise rates are less at M.B.T. than for the over-advanced 30° case as expected, the highest value being 3.95 Bar/Deg for the dual ignition disc.

The maximum burnt gas temperatures are predicted to be in the region of 2500 K, the fast burn designs showing a small increase. The charge temperatures at exhaust valve opening (E.V.O.) are interesting since apart from design E, the differences are negligible. It had been expected that the fast burn designs would have had lower temperatures but it would appear that the effect of the faster combustion is offset by the higher peak gas temperature. Design E has a higher charge temperature at E.V.O. due to the reduced heat losses as discussed earlier.

The crank angle from ignition to reach 2%, 50% and 90% burnt mass fraction is shown in the next three rows. The predictions show that the dual ignition designs have the fastest combustion rate at both fixed and M.B.T. ignition timing. As expected, reduced ignition advance tends to increase the combustion rate.

The flame travel angles for 10 mm and 30 mm flame radii are also shown in the table. These predictions suggest that the flame speed is slightly lower for the chambers having the greatest mass burn rates. However, it must be emphasised that the differences are very small and could partly be accounted for by both program errors (note that 2° crank angle intervals were used) and interpolation errors. Other factors which would account for the differences are the different charge temperatures affecting the laminar burning velocity and the increased unburnt gas density with the fast burn designs (and in particular with the dual ignition chambers) which reduce the burnt gas expansion.

Comparing the heat transfer rates from the charge to the chamber walls, it can be seen that the dual ignition hemispherical head chamber offers a 20% reduction from about 2.35 KW to 1.83 KW at M.B.T. This reduction is mainly due to the reduced heat transfer surface area and is largely responsible for the 5% improvement in thermal efficiency discussed earlier.
Finally, the peak nitric oxide concentrations predicted by the model indicate that due to the higher burnt gas temperatures, the fast burn designs would have the highest NO emissions, although the differences are quite small (circa. 7%). In terms of indicated specific nitric oxide, the differences are almost negligible.

7.4 Fired Engine Squish Velocity Predictions

Appendix IV gives details of previous work which has been reported on the prediction of squish velocities in non-fired motored engines. Theoretical squish velocities are presented showing the effect of the important parameters such as squish area, bumping clearance and chamber shape.

Although previous researchers have investigated the effect of variables such as gas inertia, friction and heat transfer, the work on this subject appears to have been restricted to the case of non-fired motored engines only. One reason for this is the fact that velocity measurements are difficult to achieve in fired engines and as such, squish velocities have only been measured in motored engines. Therefore, motored engine prediction formulae have been derived so that comparisons between the measured and predicted velocities could be made.

A second reason may be that squish velocities are more difficult to predict for fired engines. However, considering the importance of knowing the magnitude of the squish velocities, it does seem curious that no attempt has been made (to the authors knowledge) to determine the effect that combustion has on the motored values.

The motored and fired engine squish velocities will be different due to the large density variation in the latter case. As the charge is burnt, the increase in temperature causes the gas to expand, compressing both the unburnt and previously burnt fractions. If the normal case is considered where combustion is occurring in the main part of the chamber away from the squish zone, the expanding gas will compress the charge in the squish zone producing a reverse squish motion.
Although it is possible to accept this effect qualitatively, the magnitude of the differences between the motored and fired velocities is required to be determined before the importance of this effect can be evaluated. To allow this to be done, an algorithm was added to the simulation model to compute the squish velocity for the simplest case of the bowl in piston with central ignition. Details of the equations used have been presented in Section 6.2.5. Typical results obtained from this algorithm are presented in the remainder of this section.

The variation of the predicted squish velocities over a range of crank angles either side of TDC is shown in Figures 7.26 through 7.28 for a 50% squish area bowl in piston design. The first two of these figures shows the effect of ignition timing on the predicted velocities for the two cases of rich (fig. 7.26) and weak (fig. 7.27) combustion.

It can be seen that following ignition, the predicted fired engine squish velocity is equal to the motored velocity for about 16 degrees before significant differences can be detected. This is due to the effect of the pressure delay period where the pressure rise due to combustion is negligible. The agreement between the fired and motored predictions during the pre and post combustion periods verifies the accuracy of the approximate fired engine algorithm.

During the combustion phase, the squish velocities are either reduced or made more negative (i.e. increased reverse squish) by the action of combustion. Over-advanced ignition causes the magnitude of the peak forward squish velocity to be reduced, the peak occurring earlier in the cycle. Retarded ignition on the other hand has negligible effect on the forward squish velocity but greatly increases the magnitude of the reverse squish velocity. Comparing the results of Fig. 7.26, an ignition timing of 20° BTDC more than doubles the peak reverse squish velocity.

It is clear from Figs. 7.26 and 7.27 that the theoretical squish velocity in a fired engine is highly dependent on the ignition timing. Advanced timing reduces the forward squish velocity whilst retarded timing greatly increases the magnitude of the reverse squish. However, since production engines are required to operate at about the M.B.T.
ignition setting, it is of special interest to determine the combustion effect at the M.B.T. value.

This is shown in Fig. 7.28 for an equivalence ratio of 0.75. The hatched area indicates the difference between the motored and fired engine model predictions. It can be seen that at M.B.T., the reduction in peak forward squish velocity is small whereas the reverse squish is greatly increased to about twice the forward squish magnitude. Also to be noted is the position of the zero squish velocity point which is displaced from TDC to about $4^\circ$ BTDC by the combustion.

The effect of combustion on the squish velocity for several other operating conditions and squish areas was also investigated and it was found that the trends were similar to those already presented. In all cases when the timing was set at M.B.T., the combustion effect was similar to that shown in Fig. 7.28 with only a negligible difference in the forward squish velocity.

To summarise these results, it is clear that combustion can have a significant effect on the squish velocities, the forward velocity being reduced and the reverse squish being increased. Since this effect is very dependent on the ignition timing, one would expect the burning velocity versus ignition timing relationship to be different for a squish chamber than for a disc chamber providing the squish velocity does affect the turbulence.

Although the results presented have been restricted to a symmetrical bowl in piston chamber design, the trends should be similar for most other chamber shapes. An exception would be if the burnt zone was situated in the squish area, giving the opposite effect of increasing the squish velocity. However, this would not normally occur until after T.D.C. and would not significantly affect the main combustion phase.
7.5 COMPARISON BETWEEN PREDICTED AND MEASURED RESULTS

7.5.1 Introduction

The purpose of this section is to compare the predicted effect of combustion chamber design with experimental observations. The intention here is to verify the accuracy of the computer simulation model COMPSIMP with regard to the assumptions made concerning the effects of combustion chamber shape. Results verifying the accuracy of the model in predicting the effects of engine operating conditions for a disc chamber have been presented in Section 6.4.

In addition to a comparison between the computer predictions and the experimental results obtained in this study, further comparisons are made using experimental results reported by two other workers. Using the additional results reported in the literature for two different engine designs has the advantage that it extends the comparison to other chamber and engine designs. Apart from increasing the amount of data and so reducing the probability of errors, this is important in that it helps to cover the possibility of parameter interaction distorting the conclusions.

7.5.2 Comparison With Experimental Results Obtained In This Study

Table 7.4 shows the range of experimental designs simulated together with the predicted effect on the combustion and performance parameters. It can be seen that in addition to the seven chambers designs, an additional set of results is included to indicate the effect of a 5% increase in turbulence intensity.

The results show that in all cases, the maximum cylinder pressure is higher than that for the datum chamber, the largest increases being 34% and 16% for the increased compression ratio and central ignition designs respectively. The crank angle at which the maximum pressure occurs is reduced most by the central ignition (70°) with the large
squish area and high compression ratio chambers giving about a 4° reduction. The maximum pressure rise rate is predicted to be greatly increased by the combustion chamber modifications with a maximum increase of 91% for the central ignition location.

The delay angle, \( \theta_d \), is in general only slightly reduced by the changes in design with the exception of the higher compression ratio where a 9% reduction occurs. These trends agree well with the experimental observations shown in Tables 5.4 through 5.8. In contrast to the small variation in delay angle, the burn angle, \( \theta_b \), is increased by up to 36% (central ignition) by the modifications. The bowl in piston and large squish area type 3 squish designs give 15% and 18% increases in \( \theta_b \) with the small area squish type 2 and 4 designs producing negligible improvement.

The flame travel angles over the first 10 mm of flame propagation are predicted to be reduced slightly by the modifications with a much larger reduction of 8% for the 10:1 compression ratio case. Similar trends are predicted for the flame travel angle over the range 10 mm to 50 mm. Note that a 5% increase in turbulence intensity does not necessarily result in a 5% increase in flame speed due to the expansion effect of the burnt charge.

Finally, the indicated power output predictions indicate that this parameter is not greatly affected by the chamber shape or the spark plug location. The design parameter having a substantial effect however is compression ratio with the increase from 8:1 to 10:1 resulting in an 8% increase in indicated power output (or i.m.e.p. or indicated thermal efficiency).

Of the combustion chamber arrangements compared in Table 7.4, it is clear that the high compression ratio and central spark plug location result in the greatest improvements overall with the large squish area designs giving smaller but still significant increases in combustion rate. The small squish area type 2 and 4 designs have negligible effect overall.
The 5% increased turbulence intensity is predicted to have a relatively small effect relative to the more dominant design modifications, giving a small improvement in all of the parameters. However, although the changes are not large, they are significant and would be detectable in an experimental study of the type performed in this work. This suggests therefore that an increase in turbulence intensity of approximately 5% or more due to combustion chamber generated charge motion (e.g. squish) should produce increases in the flame speed and combustion rate which can be readily measured.

The predictions shown in Table 7.4 agree fairly well with the experimental results which have been reported in Chapter 5. To allow a direct comparison to be made between predicted and experimentally observed trends, Table 7.5 has been compiled. This shows the changes in the more significant parameters relative to the datum or disc chamber for the six modified combustion chambers. Note that the results are presented as changes relative to the datum rather than in absolute values since exact agreement does not exist between the predicted and experimental results for the datum chamber due to the empirical modelling constants having been based on giving good agreement over a large range of operating conditions. Although it would have been possible to change the constants to obtain much better agreement, the relatively small differences would not have justified this change.

Note also that the predicted results are based on same operating conditions as for the experimental results, the latter having been obtained from Tables 5.4 to 5.8.

Comparing the results, it can be seen that in general, the agreement is good with the correct trends being predicted in all cases except three (out of a total of 24). The three cases all occur for the concentrated sector type squish designs with the 15% type 2 and 4 designs predicting about a 4% reduction in burn angle compared to a 0.6% and 5.4% increase respectively by experiment and the type 3 design predicting a 4.5% increase in maximum pressure against a 4.3% reduction from experiment.
<table>
<thead>
<tr>
<th>EXPERIMENTAL CHAMBER DESIGN</th>
<th>CHAMBER REF. NO.</th>
<th>MAX. CYL. PRESSURE ΔP BAR</th>
<th>MAX. PRES. CRANK ANGLE θp DEG</th>
<th>MAX. PRES. RISE RATE Δp BAR/DEG</th>
<th>DELAY (2%) BURN (90% -2%) ANGLE</th>
<th>10 mm FLAME TRAVEL ANGLE θ10,5 DEG</th>
<th>50 mm FLAME TRAVEL ANGLE θ50,10 DEG</th>
<th>INDICATED POWER OUTPUT KW</th>
</tr>
</thead>
<tbody>
<tr>
<td>8:1 CR Disc Type 1</td>
<td>1</td>
<td>46.44</td>
<td>374.0</td>
<td>1.7</td>
<td>18.3</td>
<td>24.1</td>
<td>18.7</td>
<td>23.3</td>
</tr>
<tr>
<td>15% V Type Squish Type 2</td>
<td>6</td>
<td>47.37 +2%*</td>
<td>374.0</td>
<td>2.55 +50%</td>
<td>18.0</td>
<td>23.7 +1.7%</td>
<td>18.5</td>
<td>23.0</td>
</tr>
<tr>
<td>48% Sector Squish Type 3</td>
<td>17</td>
<td>48.51 +4.5%</td>
<td>369.5</td>
<td>2.95 +74%</td>
<td>17.8</td>
<td>19.7 +18.0%</td>
<td>18.3</td>
<td>22.8</td>
</tr>
<tr>
<td>15% Sector Squish Type 4</td>
<td>14</td>
<td>46.64 +0.4%</td>
<td>374.0</td>
<td>2.05 +21%</td>
<td>18.3</td>
<td>24.0 +0.4%</td>
<td>18.2</td>
<td>23.4</td>
</tr>
<tr>
<td>47% Bowl In Piston Type 6</td>
<td>16</td>
<td>49.24 + 6%</td>
<td>370.5</td>
<td>2.75 +62%</td>
<td>18.0</td>
<td>20.4 +15.0%</td>
<td>18.3</td>
<td>23.3</td>
</tr>
<tr>
<td>8:1 CR Disc With Central Ignition</td>
<td>2</td>
<td>53.96 +16%</td>
<td>367.0</td>
<td>3.25 +91%</td>
<td>18.3</td>
<td>15.4 +36.0%</td>
<td>18.5</td>
<td>N/A</td>
</tr>
<tr>
<td>10:1 CR Disc Type 1</td>
<td>3</td>
<td>62.3 + 34%</td>
<td>370.5</td>
<td>2.9 +71%</td>
<td>16.9 +9%</td>
<td>22.3 +7%</td>
<td>17.2</td>
<td>21.3 +9%</td>
</tr>
<tr>
<td>REF No. 1 With 5% Increased Turbulence</td>
<td>1</td>
<td>47.34 + 2%</td>
<td>373.0</td>
<td>2.0 +18%</td>
<td>18.0</td>
<td>23.0 +5.0%</td>
<td>18.0</td>
<td>22.5 +3.4%</td>
</tr>
</tbody>
</table>

* : Figures Indicate Percentage Improvement Relative To Disc Chamber Ref. No. 1

TABLE 7.4 PREDICTED PERFORMANCE OF EXPERIMENTAL COMBUSTION CHAMBER AT 2000 RPM, 75% VOLUMETRIC EFFICIENCY AND 1.13 EQUIVALENCE RATIO
<table>
<thead>
<tr>
<th>EXPERIMENTAL COMBUSTION CHAMBER DESIGN</th>
<th>CHAMBER REFERENCE NUMBER (SEE TABLE 3.1)</th>
<th>EXPERIMENTAL AND SPECIFIED (E AND P) IGNITION TIMING (DEG BTDC)/VOLUMETRIC EFFICIENCY (%)</th>
<th>MAX. CYLINDER PRESSURE P % INCREASE RELATIVE TO DISC (Ref. No. 1)</th>
<th>MAX. PRESSURE CRANK ANGLE $\Delta_p$REDUCTION REL. TO DISC (Ref. No. 1), DEG</th>
<th>MAX. PRESSURE RISE RATE $\Delta_p$ % INCREASE REL. TO DISC (Ref. No. 1)</th>
<th>BURN (90% - 2%) % REDUCTION REL. TO DISC (Ref. No. 1)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15% V Squish Type 2</td>
<td>6</td>
<td>30/85</td>
<td>1.2</td>
<td>0</td>
<td>-1.5</td>
<td>-1.0</td>
</tr>
<tr>
<td>15% Sector Squish Type 4</td>
<td>14</td>
<td>30/85</td>
<td>0.8</td>
<td>0</td>
<td>-1.0</td>
<td>-0.5</td>
</tr>
<tr>
<td>48% Sector Squish Type 3</td>
<td>17</td>
<td>30/75</td>
<td>-4.3</td>
<td>4.5</td>
<td>2.6</td>
<td>4.5</td>
</tr>
<tr>
<td>47% Bowl In Piston Type 6</td>
<td>16</td>
<td>30/75</td>
<td>8.3</td>
<td>6.0</td>
<td>3.1</td>
<td>3.5</td>
</tr>
<tr>
<td>8:1 CR DISC With Central Ignition</td>
<td>2</td>
<td>40/85</td>
<td>6.4</td>
<td>4.0</td>
<td>1.8</td>
<td>4.0</td>
</tr>
<tr>
<td>10:1 CR DISC Type 1</td>
<td>3</td>
<td>20/85</td>
<td>28.3</td>
<td>32.0</td>
<td>5.1</td>
<td>4.0</td>
</tr>
</tbody>
</table>

E : Experimental Results  
P : Simulation Model Predictions

**TABLE 7.5** COMPARISON BETWEEN PREDICTED AND EXPERIMENTAL RESULTS FOR SEVERAL COMBUSTION CHAMBER DESIGNS
As one would expect, the largest variation in the magnitude of the differences between experiment and prediction occur for the maximum pressure rise rate since this parameter would be highly sensitive to errors in the computer model resulting from the spherical flame front assumption and the neglect of quench zone effects. These assumptions would account for the predicted rise rates being too high.

Of the six combustion chambers, the poorest agreement occurs for the 48% sector type squish and the central ignition disc designs. It is believed that in the case of the former chamber, this is partly due to the quenching of the flame in the squish zone. A second reason applying to both chambers is the effect of the non-spherical propagation as already discussed which would be more significant with these two design than for the less concentrated squish arrangements or for a more offset ignition design. This suggests that the increased combustion rate resulting from a more central ignition location or from large squish areas would be less than predicted by the simulation model.

7.5.3 Comparison With Experimental Results Reported by Ziv

In this paper, Ziv compares four hemispherical head combustion chambers in terms of cylinder pressure, M.B.T. timing, power output etc. No flame speed or mass burn rates were determined during the study. The hemispherical heads had a central ignition location with the squish area being varied from zero up to about 60% (estimated) or in terms of theoretical maximum squish velocity, from zero to 33.5 m/s. The engine used for this study was a high speed two stroke unit, which, due to the absence of valves is ideally suited to a hemispherical head arrangement.

This paper was chosen for analysis and comparison for several reasons. First, the experimental results reported in this thesis had not covered a hemispherical head arrangement. Second, the engine was vastly different from the engine used in this work. Third, the author of the paper was clearly convinced that the observed
differences between the chambers was totally due to squish generated turbulence with no reference (except in the introduction) to the effect of compactness, flame area, or flame travel distances. Finally, the dimensions of the experimental chambers were specified in sufficient detail to allow the comparison to be carried out.

Details of the combustion chamber designs assumed for the comparison are shown in Table 7.6 and Fig. 7.29. In addition to the four cylinder heads tested by Ziv, an additional two designs consisting of 70% and 80% squish were added to illustrate the predicted trends more clearly. Note that only the squish velocities were specified in the paper and therefore the assumed squish areas (calculated from the equation given by Ziv) may be subject to some error.

<table>
<thead>
<tr>
<th>SQUISH AREA %</th>
<th>THEORETICAL SQUISH VELOCITY m/s (ft/s)</th>
<th>CORRESPONDING TO DESIGN IN REF</th>
</tr>
</thead>
<tbody>
<tr>
<td>80</td>
<td>62.5 (205)</td>
<td>A</td>
</tr>
<tr>
<td>70</td>
<td>43.3 (142)</td>
<td>B</td>
</tr>
<tr>
<td>60.4 *</td>
<td>33.5 (110)</td>
<td>C</td>
</tr>
<tr>
<td>48.7 *</td>
<td>24.4 (80)</td>
<td>D</td>
</tr>
<tr>
<td>34.8 *</td>
<td>15.2 (50)</td>
<td></td>
</tr>
<tr>
<td>0</td>
<td>0</td>
<td></td>
</tr>
</tbody>
</table>

BUMPING CLEARANCE = 1.524 (0.060")
CENTRAL SPARK PLUG LOCATION WITH 2.0 mm PROJECTION
* ESTIMATED VALUES BASED ON EQ (3) IN REF 87

TABLE 7.6 COMBUSTION CHAMBER DETAILS

The engine data required by the program were taken from the details supplied by Ziv in Table 1 of the paper. A connecting rod length of 95 mm was assumed. The most important assumption however was that the spark plug projection was constant at 2.0 mm. This meant that as the squish area was increased, the total chamber height also increased resulting in the spark plug being moved further from the piston.
For the purpose of the comparison, a single operating condition corresponding to 6000 RPM, 75% volumetric efficiency and 1.1 equivalence ratio was assumed.

In an attempt to achieve reasonable agreement between the predicted and reported results, the values of the turbulence multiplication factor and turbulent flame radius were adjusted to $3.5 \times 10^{-3}$ and 0.1 mm respectively (from $2.91 \times 10^{-3}$ and 10 mm). This meant that for the interval immediately following ignition, the burning velocity was set equal to the calculated laminar velocity and for all later intervals, the fully turbulent value was used. Whilst this is difficult to justify, it agrees fairly well with the results reported by Ziv which indicate only about a $4^\circ$ increase in M.B.T. ignition timing resulting from increasing the engine speed from 3500 to 7000 RPM.

Comparisons between the reported and predicted results are shown in Figures 7.30 through 7.32. All of these graphs have been plotted against maximum theoretical squish velocity (using equation 3 in Ref. 87) since this shows the trends with squish area and is the method by which the chambers were compared in the paper. Note also that whilst the measured data was based on four values with a maximum squish area of about 60%, the predictions are for six values with a maximum of 80% squish area.

The curve shown in Fig. 7.30 for the measured data has been copied directly from Fig. 13 of the paper. The graph shows that the predicted and experimental trends are similar although the measured curve indicates a greater sensitivity to increasing squish velocity. This difference could be partly due to flame quenching effects with the high squish area designs causing the predicted M.B.T. timing to be less affected by an increase in squish area.

Allowing for the possibility of quenching effects, model assumptions and assumed design parameter values, the difference between the predicted and experimental results are quite small and could be accounted for by experimental error. Accurate M.B.T. ignition timing is very difficult to determine experimentally, a problem which was experienced by Mayo (3). The results presented by Ziv at various engine speeds show quite a lot of scatter and whilst the general trend shown in Fig. 7.30 was evident, the relative differences varied considerably.
Figures 7.31 and 7.32 show plots of maximum pressure rise rate and maximum cylinder pressure plotted against squish velocity. The experimental results shown are the mean of the five sets of results presented by Ziv in Tables 9 and 10 for the engine speed range 4000 - 7000 RPM.

The model predictions indicate a slight increase in pressure rise rate for initial increases in squish velocity with a very large decrease occurring at values above about 40 m/s (approx. 65% squish area). With the maximum pressure, the predicted trend is an approximately linear decrease with squish.

The measured results agree fairly well with these trends as shown if allowances are made for scatter. The experimental pressure rise rates show a greater sensitivity to increasing squish area than predicted although it should be noted that this parameter is very sensitive to slight changes in ignition timing. The maximum cylinder pressure for the zero squish is less than the predicted value whilst the other three values are slightly higher. It is believed that these small differences are well within experimental error and could be partly accounted for by variations in M.B.T. ignition timing.

Overall, the predicted trends with increasing squish area show good agreement with the measured results. Since the model assumes constant turbulence intensity, the assumption by Ziv that the observed trends are due to squish generated turbulence is not justified. The experimental results do suggest however that the performance of high squish area chambers will be less than predicted by the model due to the effect of wall quenching in the squish zone.

7.5.4 Comparison With Experimental Results Reported by Mayo

In this paper, Mayo presents results obtained from an eight cylinder engine, showing the effects of and the interaction between intake generated charge motion, compression ratio spark plug location and combustion chamber shape. The results were presented mainly in terms of 10% and 90% burnt mass fraction angles, although some results for cyclic dispersion and exhaust emissions were also included.
This paper is very interesting in that it shows a very strong interaction between parameters such that the effect of a design variable is very dependent on the other design variables. However, for the purposes of this Section, the effects of intake charge motion are ignored and therefore the comparison is restricted to the other three parameters studied; namely, compression ratio, spark plug location and squish area.

A total of six combustion chamber designs were simulated for the comparison, the required data either being given in the paper, estimated from the sketches or assumed. Details of the six designs are specified in Table 7.7.

<table>
<thead>
<tr>
<th>CHAMBER PARAMETER</th>
<th>PRODUCTION</th>
<th>NORMAL SPARK OPEN</th>
<th>CENTRAL SPARK OPEN</th>
<th>NORMAL SPARK SQUISH</th>
<th>CENTRAL SPARK SQUISH</th>
<th>B.I.P</th>
</tr>
</thead>
<tbody>
<tr>
<td>STROKE mm</td>
<td>88.9</td>
<td>88.9</td>
<td>88.9</td>
<td>88.9</td>
<td>88.9</td>
<td>98.04</td>
</tr>
<tr>
<td>BORE mm</td>
<td>101.6</td>
<td>101.6</td>
<td>101.6</td>
<td>101.6</td>
<td>101.6</td>
<td>110.7</td>
</tr>
<tr>
<td>CON ROD LENGTH mm</td>
<td>146.7</td>
<td>151.1</td>
<td>151.1</td>
<td>146.7</td>
<td>146.7</td>
<td>161.8</td>
</tr>
<tr>
<td>COMP RATIO</td>
<td>8.3</td>
<td>10.9</td>
<td>10.9</td>
<td>10.4</td>
<td>10.4</td>
<td>10.6</td>
</tr>
<tr>
<td>SPARK PLUG RADIUS mm</td>
<td>31</td>
<td>31</td>
<td>14</td>
<td>31</td>
<td>14</td>
<td>3</td>
</tr>
<tr>
<td>SPARK PLUG PROJECTION mm</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>SQUISH AREA %</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>30</td>
<td>30</td>
<td>75</td>
</tr>
<tr>
<td>BUMPING CLEARANGE mm</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>1.0</td>
<td>1.0</td>
<td>0.7</td>
</tr>
<tr>
<td>CHAMBER TYPE</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>3</td>
<td>3</td>
<td>6</td>
</tr>
</tbody>
</table>

**TABLE 7.7 EXPERIMENTAL COMBUSTION CHAMBER AND ENGINE DATA ASSUMED FOR MODEL SIMULATION**

The spark plug radii specified above were estimated from scaling photographs presented in the paper (Figs. 1 and 2) and from knowing that the spark plug had been moved 17 mm towards the chamber centre. The shape of the chamber was assumed to be either a disc (open), a bowl in piston...
with flat head (B.I.P) or a type 3 squish design with 20% of the squish area on the opposite side of the chamber to the sparking plug (squish). Although these assumptions are reasonably accurate for the open and BIP chambers, the 30% squish chamber simulation is only a fairly crude simulation.

The operating conditions chosen for the simulation were 1500 RPM, 85% volumetric efficiency, 1.1 equivalence ratio and MBT ignition timing, these being similar to the actual experimental settings. A 7% residual gas concentration was assumed for the 8.3 compression ratio chamber, this being reduced for higher compression ratios in proportion to the change in clearance volume at TDC.

As with the simulation reported in the previous sub-section, an attempt was made to obtain about the same combustion rate and MBT timing as those obtained experimentally. It was found that reducing the fully turbulent flame radius from 10 mm to 5 mm without any change to the turbulent multiplier (set at $2.91 \times 10^{-3}$) gave about the same MBT setting as reported. However, the predicted delay and burn angles were larger and smaller respectively than those obtained from the analysis of the pressure-time data using the Rassweiler and Withrow (33) method. Although it would have been possible to have achieved better agreement by reducing both the turbulent flame radius and multiplier, this was not considered to be justified since it would not have made a significant change to the relative differences between chambers.

The predictions obtained for the range of combustion chambers are presented in Table 7.8, both in terms of absolute values and the percentage changes due to the central spark plug location and the addition of squish area. Also shown for comparison purposes are the reported experimental results. It must be noted that the BIP design experimental results are not directly comparable with the others since this was a completely different engine (in contrast to a cylinder head modification) and had a very high velocity, high swirl inlet port arrangement. Therefore, one would not expect good agreement between the predicted results (with the empirical constants based on the production engine design) and the BIP experimental results.
## Predicted and Experimental Results for the Six Combustion Chambers of Mayo

<table>
<thead>
<tr>
<th>Chamber Design</th>
<th>M.B.T. Ign. Timing (°BTDC)</th>
<th>Delay Angle (0-10% deg)</th>
<th>Burn Angle (10-90% deg)</th>
<th>Delay + Burn Angles deg</th>
</tr>
</thead>
<tbody>
<tr>
<td>8.3:1 CR Production</td>
<td>33 (30)*</td>
<td>24.7 (23.9)</td>
<td>26.7 (29.7)</td>
<td>51.4 (53.6)</td>
</tr>
<tr>
<td>10.9:1 CR Normal Open</td>
<td>27 (27)</td>
<td>20.9 (18.7)</td>
<td>23.3 (29.5)</td>
<td>44.2 (48.2)</td>
</tr>
<tr>
<td>10.9:1 CR Central Open</td>
<td>24 (?)</td>
<td>19.3 (17.8)</td>
<td>16.9 (23.6)</td>
<td>36.2 (41.4)</td>
</tr>
<tr>
<td>10.4:1 CR Normal Squish</td>
<td>28 (?)</td>
<td>21.4 (17.5)</td>
<td>18.9 (26.2)</td>
<td>40.3 (43.7)</td>
</tr>
<tr>
<td>10.4:1 CR Central Squish</td>
<td>21 (22)</td>
<td>18.3 (14.5)</td>
<td>13.4 (22.6)</td>
<td>31.7 (37.1)</td>
</tr>
<tr>
<td>10.6:1 CR Bip</td>
<td>17 (7)</td>
<td>16.8 (11.1)</td>
<td>13.2 (10.7)</td>
<td>30.0 (21.8)</td>
</tr>
</tbody>
</table>

* Experiment Results, Reported By Mayo(3) Shown in Parenthesis

## Percentage Change in Parameters Due to Using a More Central Spark Location

<table>
<thead>
<tr>
<th>Combustion Chamber Shape</th>
<th>% Reduction by Use of &quot;Central&quot; Spark</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Delay Angle</td>
</tr>
<tr>
<td>OPEN</td>
<td>7.6 (4.8)*</td>
</tr>
<tr>
<td>SQUISH</td>
<td>14.5 (17.1)</td>
</tr>
</tbody>
</table>

## Percentage Change in Parameters Due to Addition of 30% Squish Area

<table>
<thead>
<tr>
<th>Spark Plug Location</th>
<th>% Reduction by Addition of 30% Squish Area</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Delay Angle</td>
</tr>
<tr>
<td>BASE (NORMAL)</td>
<td>2.4 (6.4)*</td>
</tr>
<tr>
<td>CENTRAL</td>
<td>5.2 (18.5)</td>
</tr>
</tbody>
</table>

**Table 7.8: Comparison Between Predicted and Reported Experimental Results for the Combustion Chamber Designs of Mayo(3)**
Inspecting the results in Table 7.8, it can be seen that there is fairly good agreement (excluding the BIP design) between the predicted and experimental results for the M.B.T. timing and the crank angle required to burn 90% of the charge. As mentioned previously, the predictions are on average about three degrees longer for the delay angle (defined as 10% burnt mass fraction) and about 5° shorter for the burn angle. The poorest agreement exists for the BIP results as expected although the burn angle for the central ignition squish design is predicted to be about nine degrees less than that obtained experimentally. Also, the experimental results suggest that compression ratio reduces the delay angle with no effect on the burn angle whilst the model shows improvements to both parameters.

The percentage changes due to the spark plug location and incorporation of squish area are shown in (b) and (c) of Table 7.8. The agreement for the effect of spark plug location is very good, especially for the open chamber although the measured percentage improvement in the burn angle with the squish chamber is only about half that which is predicted. The percentage reduction due to the addition of the 30% squish area is in good agreement for the normal spark plug location (considering the crudeness of the squish area simulation) with the central spark plug location predictions having mixed success. The agreement for the overall time to burn 90% of the charge is very good, the difference being due to the prediction under-estimation of the delay period which therefore results in an over-estimate of the burn angle.

The experimental results show an 18.5% improvement in delay angle for the addition of the squish area in the central ignition case, compared with a predicted 5.2% increase. This difference cannot be attributed to the crudeness of the model since at 10% burnt mass fraction, the flame front will not have contacted the squish area (assuming spherical flame propagation). Therefore, the predicted reduced delay angle of 5.2% is due mainly to the increased chamber height. The much larger experimental value (excluding experimental errors) suggests either a highly distorted flame profile or an enhanced charge turbulence/velocity structure in the region of the spark plug.
However, if the reduced delay angle does occur, the results suggest that the burn angle is greatly reduced by the squish area. This implies either a distorted flame front profile, a deteriorated turbulence structure or the possibility of flame quenching. Whatever the reason, the difference between predicted and experimental results for the overall delay and burn angles is small.

To conclude this sub-section, the agreement between predicted and experimental results is acceptable. This agreement verifies the accuracy of the simulation model. The predictions appear to be most accurate for the open chamber with the normal offset ignition and least accurate for the central ignition squish case.
Fig 7.1 BURNT VOLUME AND FLAME AREA CURVES FOR A DISC CHAMBER SHOWING EFFECT OF SPARK PLUG LOCATION
Fig 7.2 BURNT VOLUME AND FLAME AREA CURVES FOR BOWL IN PISTON DESIGN SHOWING EFFECT OF SPARK PLUG LOCATION.
Fig 7.3 BURNT VOLUME CURVES FOR TYPES 5 & 7 DESIGNS SHOWING EFFECT OF SPARK PLUG LOCATION
Fig 7.4  BURNT ZONE HEAT TRANSFER SURFACE AREA CURVES FOR DISC CHAMBER

Fig 7.5  EFFECT OF SPARK PLUG AXIAL LOCATION ON BURNT VOLUME CURVES
Fig 7.6  EFFECT OF BUMPING CLEARANCE ON BURNT VOLUME CURVES

Fig 7.7  EFFECT OF PISTON POSITION ON BURNT VOLUME CURVES
LEGEND
1 TYPE 1 DISC
2 50% Type 6 B.I.P
3 50% Type 5 SECTOR
4 Type 7 DISC DUAL
5 50% Type 8 B.I.P DUAL
6 30% Type 9 HEMI DUAL

(\text{PH} = 4.0\,\text{mm})

\text{PH} = 0.0
\text{TDC}(X = 0.0) \text{ Dimensions}
\text{RR} = 0.0 \text{ in mm}
\text{RC} = 35.0
\text{CR} = 8:1

\text{Fig 7.8 EFFECT OF CHAMBER DESIGN AND SPARK PLUG POSITION ON RADIUS FOR 80\% VOLUME BURNT}
Fig 7.9 EFFECT OF SQUISH AREA ON RADIUS FOR 80\% BURNT VOLUME—TYPE 5 CHAMBER

Fig 7.10 EFFECT OF SQUISH AREA ON RADIUS FOR 80\% BURNT VOLUME—TYPE 6 CHAMBER
Fig 7.11 EFFECT OF SQUISH AREA ON RADIUS FOR 80% VOLUME BURNED — TYPES 9 AND 10

Fig 7.12 EFFECT OF COMPRESSION RATIO ON RADIUS FOR 80% VOLUME BURNED
Fig 7.13 EFFECT OF SPARK PLUG RADIAL LOCATION ON CYLINDER PRESSURE — DISC CHAMBER

TYPE 1 DISC  PH = 0
N = 2000 RPM
θ₀ = 30°

BAR = 1.1
γ₀ = 75 %
CR = 8:1
BORE = STROKE = 70 mm
Fig 7.14  EFFECT OF SPARK PLUG RADIAL LOCATION ON CYLINDER PRESSURE — B.I.P.
TYPE 1 DISC PH=0
Data as Fig 7.13

Fig 7.15 BURNT MASS PLOTS FOR DISC AND BOWL IN PISTON CHAMBERS SHOWING EFFECT OF SPARK PLUG RADIAL LOCATION

TYPE 6 BIP PH=0
50% Squish Area
Data as Fig 7.14
Fig 7.16  BURNT MASS PLOTS FOR 50% SECTOR AND DUAL IGNITION DISC SHOWING EFFECT OF SPARK PLUG RADIAL LOCATION
Fig 7.17  **EFFECT OF SQUISH AREA ON CYLINDER PRESSURE AND BURNT MASS FRACTION FOR BOWL IN PISTON**
Fig 7.18  EFFECT OF COMPRESSION RATIO ON CYLINDER PRESSURE AND BURNT MASS FRACTION FOR THE DISC CHAMBER
N = 2000 RPM
θs = 30°
Ø = 1.1
γ = 75%
CR = 8:1
BORE = STROKE = 70mm

LEGEND
1 Type 1 DISC
2 Type 5 SECTOR 50% SQ
3 Type 6 B.I.P 50% SQ
4 Type 7 DUAL DISC
5 Type 8 DUAL B.I.P 50% SQ
6 Type 9 DUAL H.H 30% SQ
7 Type 10 H.H 30% SQ

Fig 7.19 EFFECT OF SPARK PLUG RADIAL LOCATION ON 50% MASS BURNT CRANK ANGLE FOR SEVERAL COMBUSTION CHAMBER DESIGNS.
Legend and assumed operating conditions as for fig 7.19

Fig 7.20  EFFECT OF SPARK PLUG RADIAL LOCATION ON MAXIMUM PRESSURE RISE RATE FOR SEVERAL COMBUSTION CHAMBER DESIGNS
Fig 7.21  EFFECT OF SQUISH AREA ON 50% AND 90% MASS BURNT CRANK ANGLES FOR SEVERAL COMBUSTION CHAMBER DESIGNS
Fig 7.22 EFFECT OF COMBUSTION CHAMBER DESIGN ON PRESSURE AND BURNT MASS DIAGRAMS

A to E refers to Chamber designs specified in Table 7.2

N = 2000 RPM
\( \phi = 0.95 \)
\( \Theta_s = 30^\circ \)
\( \gamma_v = 75\% \)
A to E refers to Combustion Chamber designs specified in Table 7.2

Fig 7.23 COMPARISON BETWEEN COMBUSTION CHAMBER DESIGNS OVER A RANGE OF MIXTURE STRENGTHS

N = 2000 RPM
θₜ = 30°
ν = 75%
Fig 7.24 COMPARISON BETWEEN COMBUSTION CHAMBER DESIGNS OVER A RANGE OF MIXTURE STRENGTHS
A to E refer to Combustion Chamber designs specified in Table 7.2

Fig 7.25 COMPARISON OF POWER CURVES FOR FIVE DIFFERENT CHAMBER DESIGNS
Fig 7.26  EFFECT OF IGNITION TIMING ON SQUISH VELOCITY FOR FIRED ENGINE - 50% B.I.P AT $\varnothing = 1.1$
Fig 7.27 EFFECT OF IGNITION TIMING ON SQUISH VELOCITY FOR FIRED ENGINE – 50% B.I.P

Conditions as for fig 7.26 except for
$\phi = 0.75$ for this case

$\theta_s = 50^\circ$

$\theta_s = 40^\circ$

$\theta_s = 30^\circ$

$\theta_s = 20^\circ$
Fig 7.28  EFFECT OF COMBUSTION ON PREDICTED SQUISH VELOCITY AT M.B.T IGNITION TIMING — 50% B.I.P AT $\theta = 0.75$
Fig 7.29 CROSS SECTION OF COMBUSTION CHAMBERS ASSUMED FOR PREDICTIONS

Fig 7.30 COMPARISON BETWEEN PREDICTED AND MEASURED RESULTS FOR M.B.T. TIMING
Fig 7.31  EFFECT OF SQUISH AREA ON PREDICTED AND MEASURED PRESSURE RISE RATES

Fig 7.32  COMPARISON BETWEEN PREDICTED AND MEASURED MAXIMUM CYLINDER PRESSURES
CHAPTER 8

DISCUSSION OF RESULTS
CHAPTER 8

The findings of the experimental and theoretical studies have been presented and discussed in Chapters 5 and 7 respectively. The experimental results have shown that for the range of designs and operating conditions covered, combustion chamber design can have a significant effect on the combustion rate and pressure parameters. On the other hand, a relatively detailed ionisation probe study has failed to reveal any significant effect of combustion chamber shape on flame speed.

Based on the experimental observations, the combustion model, described in Chapter 6, assumes that the flame speed ratio is independent of chamber shape. The predictions obtained from the model have shown the importance of compactness and have confirmed the experimental results. By comparing the model predictions with experimental results obtained in this study and by two other workers, the constant flame speed ratio assumption has been shown to be reasonable.

A survey of previous experimental work on this subject, presented in section 2.3, has revealed that there is much disagreement and confusion regarding the mechanisms involved and the relative importance of combustion chamber design parameters. However, it is generally recognised that combustion chamber design does have a major effect on engine performance and combustion rate and that compact chamber designs do give increased combustion rate. The results of this study regarding combustion rate are therefore generally in agreement with the findings of previous studies.

On the effect of combustion chamber design on flame propagation, the situation is much more complex and controversial. This is because very few studies have been carried out to determine this effect and those that have, reviewed in section 2.3.2, have failed to agree on the importance of combustion chamber shape. The situation is not helped by the fact that in-cylinder charge motion measurements, reviewed in section 2.3.1, have also failed to agree. Also, despite the experimental evidence to the contrary, there is a wide belief
that squish area combustion chambers produce intense charge turbulence during the combustion period, resulting in substantial increases in flame speed.

The results of this study have shown quite convincingly that for the combustion chamber designs considered, the flame speed is not significantly affected by combustion chamber shape. Although this does not agree with the popular view, which is based on intuition rather than actual measurement, it does however agree very well with the results of Tidmarsh (36) and to some extent with the findings of Mattavi et al (37). The increased combustion rate resulting from the use of squish designs has been shown to be due to the effect on flame front area rather than flame speed.

The conclusion that flame speed is not significantly affected by combustion chamber shape is of considerable importance since it greatly simplifies the prediction of the effect of combustion chamber design. Based on this assumption, relatively simple zero-dimensional computer models can be used to achieve optimum chamber designs and reduce the amount of empirical testing.

The computer model developed in this work is based on this philosophy. The geometric integration modelling technique has been found to be very effective since it allows the chamber design parameters for a range of combustion chamber types to be infinitely varied. Further development effort could extend the range of combustion chamber types which can be simulated by this method.

The predictions obtained from the computer model have shown that there is a strong interaction between the design parameters such that optimum designs must be achieved using an iterative process. Optimum squish area for a given combustion rate for example is dependent on bore/stroke ratio, compression ratio, chamber shape and spark plug arrangement. For high combustion rates, multiple spark plugs or a single centrally located about plug with
50% squish area should be employed. The best design for a lean burn engine is predicted to be a dual ignition, zero squish hemispherical head since this combines high combustion rate with low combustion chamber surface area. It is interesting to note that a similar design has been reported recently by Nakajima et al\(^{(59)}\).

The main errors associated with the computer model are due to the assumption of spherical flame profile and the fact that chamber wall quenching has been ignored. With large squish area and/or central ignition designs, these two factors can produce significant errors. Wall quenching is particularly important when small bumping clearance are being considered since two plate quenching distances are typically five times\(^{(88)}\) the corresponding value for a single plate. Comparison between predicted and measured results show that for large squish area designs, the combustion rate is over-predicted, suggesting that for these designs the flame is either slowed down or prematurely extinguished.

The heat release model developed in this work has shown that this type of model can provide a great deal of additional and useful information. Flame speed data can be obtained much more easily using this method than by direct measurement. Comparison between predicted flame speed (using the model) and measured flame speed (using ionisation probes) for identical engine cycles has shown that good agreement can be obtained in most cases. As for the previous model, the main errors are likely to be due to non-spherical flame propagation and flame quenching. However, under such circumstances, direct flame measurements would be of limited value and would probably be less meaningful than the heat release model predictions. Analysis of some typical data errors has shown that the predictions are not very sensitive to such errors.
CHAPTER 9

CONCLUSIONS
CHAPTER 9

CONCLUSIONS

1. The effect of squish on flame speed is small and is considered to be negligible when compared to the other effects associated with chamber shape. This conclusion is based on the measured results obtained in this work from a large number of combustion chamber designs and from analysing experimental results reported in the literature. Flame speed is increased however by increasing the compression ratio. The flame front profile is not affected by chamber design.

2. The cylinder pressure and mass burn (combustion) rate can be significantly altered by changing the chamber design. The design parameters having the greatest effect are number of spark plugs, spark plug location, compression ratio and chamber shape. Chambers producing high combustion rates were found to have reduced cyclic dispersion.

3. It has been shown that the geometric integration technique can be used to simulate a large number of combustion chamber designs using a minimum of input data. This model has been combined with a semi-empirical engine cycle simulation model to produce a computer program capable of readily predicting the effect of chamber design parameters on the engine performance.

4. The simulation model predicts that the chamber design can have a large effect on the combustion rate, peak pressure, pressure rise rate and optimum ignition timing. The model shows that the interaction between squish area and the other design parameters is high and that an optimum amount of squish area exists. Large squish areas are suited to high compression ratio, high bore/stroke ratio designs where the chamber height at TDC is small compared to the cylinder radius.

5. For lean burn engines, dual ignition designs are shown to give improved performance over single ignition designs. With single ignition, a bowl in piston arrangement of about 60% squish area using a centrally located spark plug should give reasonable lean mixture performance.
6. Experimental results obtained in this work and from the other sources have been compared with predictions from the model for the same chamber designs. The agreement has been found to be satisfactory in most cases. The greatest errors are for either large squish areas or central ignition designs where the model over-predicts the rate of combustion. It is believed that this is due to the effects of non-spherical combustion and wall quenching effects.

7. Squish velocities under fired engine conditions have been compared with the corresponding motored values for the case of a bowl in piston design with central ignition. The results show that the effect of combustion is to reduce the forward squish slightly and increase the reverse squish by a substantial amount, the effect being highly dependent on ignition timing.
CHAPTER 10

FUTURE WORK
CHAPTER 10

10.1 Experimental Studies

To extend this work and to confirm the results obtained, it is recommended that the following experimental studies be carried out:

a) Extend the experimental measurements and procedures adopted in this study to several other engines and combustion chamber designs, particular attention being paid to the measurement of flame speed and profile.

b) A detailed investigation of the flame propagation characteristics in the squish zone on a high squish area, small bumping clearance design such as a May Fireball. This should establish the effects of wall quenching and squish generated charge motions.

c) To study the influence of combustion chamber design on the detonation (knock) characteristics of an engine. The object of this proposal would be to establish how much of the reduced octane requirements with fast burn engines is due to end gas cooling rather than the reduced time exposure resulting from the fast burn.

d) To establish the lean misfire limit and lean mixture performance of several combustion chamber designs. It is suggested that particular attention be paid to dual or multiple ignition designs, especially the zero squish area hemispherical head arrangement for which good performance has been predicted.

10.2 Simulation Model Improvements

Although the model has been shown to give satisfactory agreement with experimental observations, there are several major improvements which could be made, both to improve the accuracy and to extend the range of chamber designs which can be considered. These are as follows:
a) To modify the thermodynamic model to incorporate the effects of wall quenching in the squish zone between the piston cylinder head. A simplified quenching model which assumes that the local flame front is extinguished when the distance between the piston and head is less than a critical value, based on measured data reported in the literature (88, 89) should give improved predictions.

b) To incorporate an auto-ignition correlation into the program so that the probability of detonation could be predicted for a range of chamber designs. The auto-ignition correlation of Johnson et al (90) is recommended.

c) The thermodynamic model should be extended to give an empirically based simulation of the intake and exhaust processes so that the effects of throttling and compression ratio can be more accurately determined. The method as described by Sangama and Murthy (83) could be adopted for this simulation.

d) Extend the geometric model to include other common designs such as the wedge type chamber. The symmetry restrictions now placed on the model (e.g. dual ignition) should be relaxed where possible.
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APPENDIX I

CALIBRATION OF EXPERIMENTAL EQUIPMENT

I.1 Air Flowmeter
I.2 Pressure Transducer
I.3 Data Acquisition System
APPENDIX I

I.1 Air Flowmeter

An accurate determination of the air flow rate into the engine was required for the experimental work since this was needed to calculate both the volumetric efficiency and the equivalence ratio.

The flowmeter used for this work was of the viscous type (see Sec. 3.3 for details) and would therefore be expected to give an approximately linear relationship between flow rate and corresponding pressure drop (69). This relationship may be expressed by:

\[ Q = C \cdot \Delta H \]

where

- \( Q \) = volume flow rate (l/s)
- \( C \) = flowmeter constant
- \( \Delta H \) = pressure head (mm H\(_2\)O)

The object of the calibration was therefore to determine the value of the constant \( C \).

The flowmeter was calibrated prior to engine testing against a calibrated venturi, manufactured to comply with the recommendations of B.S. 1042 (91). The two flowmeters were connected in series and the desired flow rate was produced by a Roots Blower compressor unit. A schematic diagram of the set up is shown in Fig. I.1.1.

The flow rate was varied during the test by adjusting the two control valves, minimum flow being achieved with valve A fully open and valve B partially closed. Readings were taken at about 12 flow rates over the flow range equivalent to a pressure head of between 20 and 200 mm of water. Variables measured during the tests were the pressure drops across the two flow meters and the temperature and pressure of the air at both ambient and venturi upstream conditions.

The air flow rates were calculated from the test results using an expression given in Ref. 91 and the resulting values are shown plotted against \( \Delta H \) in Fig. I.1.2. The graph shows that the relationship between \( Q \) and \( \Delta H \) was not quite linear over the full range but since the flowmeter was only required to operate over the first part of this
range, it was possible to fit a straight line to the data as shown.

The resulting slope is equivalent to the flow meter constant by definition and measurement gave a value of 0.132 \( \text{l/s} / \text{mm} \ \text{H}_2\text{O} \). Therefore the flow rate for this flow meter was given by:

\[
Q = 0.132 \cdot \Delta H
\]

### I.2 Pressure Transducer

The Kistler 6121 piezo-electric pressure transducer used in this study was supplied by the manufacturers complete with calibration data. The Kistler 566 charge amplifier however was not calibrated and although a facility for calibration is provided on the amplifier, it was decided that it would be better to calibrate the transducer and amplifier as a single system, having a calibration constant of Volts/Bar.

The most likely problem which is going to occur when calibrating piezo-electric transducers is that of signal drift. Charge amplifiers normally have a choice of time constants (this being a measure of the rate of decay rather than the response to a signal) and for static calibration measurements, long time constant should be used. If drift is a problem, even on long time constant, this can usually be rectified by cleaning the electrical connections between transducer and amplifier and/or by drying out the amplifier by either heating (in a low temperature oven) or by leaving the unit connected to its power supply for a time.

Having eliminated any drift problems, the system can then be calibrated. A dead weight tester is normally used for this but in this work, the calibration was made against a reference pressure (Bourdon Tube) gauge using a bottled inert (Nitrogen) gas to pressurise the system, the set up being shown in Fig. I.3. This system was used in preference to a dead weight tester due to its availability and because it allows the pressure to be changed rapidly, thereby eliminating the effect of signal drift.
The calibration procedure adopted was as follows:-

a) Set charge amp to long time constant and select appropriate sensitivity setting (1 mV/pC in this case)
b) Zero charge amplifier
c) Close valve B and open valve A
d) Adjust regulator to required pressure
e) Close valve A
f) Note voltage on Digital Voltmeter (D.V.M.)
g) Vent pressure to atmosphere by opening valve B
h) Note voltage on D.V.M.
i) Repeat steps (b) to (h) about 10 times for different values of pressure over the full range of interest.

The results obtained from the calibration were then plotted as shown in Fig. 1.4. This graph shows that the relationship between pressure input and voltage output is linear with no significant difference between increasing and decreasing pressure. Measurement of the slope of the best fit straight line gave a value of 15.0 mV/Bar and this constant was used throughout the work to calculate the pressure from the voltage signal via:

\[ p = \frac{V}{0.015} \]

where \( p \) = cylinder pressure (Bar) 
\( V \) = charge amp output (Volts)

I.3 Data Acquisition System

The cylinder pressure data was recorded on an FM tape recorder and later digitised using an Analogue to Digital Converter (ADC). Since it was important to minimise errors in the pressure signal, it was found necessary to use a calibration technique which involved recording a number of signals on the same channel as that used to record the pressure signal. The calibration signals were d.c. voltages which were supplied by a reference source and checked during recording with a D.V.M. Approximately 8 different voltages in the
range +1.0 to -0.5 volts were recorded at the beginning of each tape prior to carrying out the engine test.

At a later date, the data were digitised on the ADC and it was at this stage that the calibration signals were also digitised and the resulting values printed out. A graph of the tape recorder input and ADC output voltages was then obtained and such a graph is shown in Fig. I.5.

Inspection of the plot shows that both a d.c. shift and a sensitivity error has occurred but since the pressure is converted to absolute by a numerical technique (see Appendix III), only the latter was used in the subsequent calculations to obtain the correct pressure.

The main advantage of recording calibration signals on to the data tape is that if the tape recorder settings are adjusted, after having recorded the data and before being digitised, a correct value can still be determined using the calibration constant calculated at playback. Also, the need to frequently check (and adjust if necessary) the accuracy of the tape recorder and ADC is reduced since errors will be accounted for during the data processing stage.
FIG 1.1 - BLOCK DIAGRAM OF CALIBRATION EQUIPMENT

FIG 1.2 - CALIBRATION GRAPH FOR VISCOSF FLOWMETER
FIG 1.3 - PRESSURE SYSTEM CALIBRATION EQUIPMENT

FIG 1.4 - CALIBRATION GRAPH FOR PRESSURE TRANSUDER / AMPLIFIER
Fig I-5  CALIBRATION GRAPH FOR DATA ACQUISITION SYSTEM

\[ V_i = \frac{V_o}{0.985 + 0.03} \]
APPENDIX II

IONISATION PROBE MEASUREMENT

TECHNIQUE AND EQUIPMENT

II.1 General
II.2 The Ionisation Probe Technique
II.3 Details of Equipment
II.4 Error Analysis
II.1 General

A major part of the experimental work in this study involved the determination of flame propagation rates in the combustion chamber of a fired engine. Techniques which may be used for measuring flame speed (reviewed in Section 2.2.4) were considered and it was finally decided to use the ionisation probe technique since this offers the following advantages over other methods:

a) quantitative results.
b) small dimensions of probe allow multiple probes to be used
c) electrical signal output simplifies data collection and processing.
d) simple and inexpensive construction of probe.
e) no major modifications required to engine.
f) individual cycles may be studied.

The purpose of this Section is to describe the basic mode of operation of the ionisation measurement system and to provide details of the ionisation equipment used in this study. Finally the source of detection errors are examined and ways in which the effect of such errors may be minimised are discussed.

II.2 The Ionisation Probe Technique

II.2.1 Description of Operation

The basic components of an ionisation probe flame arrival detection system are shown in Fig. II.1. It can be seen that this consists of a d.c. power supply, connected to the ionisation probe (or ion gap) via a resistor.

Under normal non combustion conditions, the resistance of the probe is very large and therefore the current flow and the voltage drop across the resistor are negligible. If the probe is now placed into a flame reaction zone, it will be found that the circuit current and hence the voltage drop across the resistor will increase. This change is due to
the fact that during combustion, ions, electrons and radicals are produced whilst the reactants are transformed into products. The negatively charged electrons thus produced will be attracted to the positive electrode of the probe, thereby reducing the effective probe resistance and causing an increase in current.

The signal produced by the flame moving past the probe is measured between the probe and resistor and a typical signal profile is shown in Fig. II.2. This indicates that at ionisation, assumed to be the point where the flame contacts the probe, there is a sudden reduction in voltage, followed by an exponential type increase back to the original level prior to combustion. The signal amplitude and pulse width are functions of the reaction zone thickness, the probe dimensions, the circuit design, the supply voltage etc in addition to the localised operating conditions in the region of the probe.

Generally, conditions which result in rapid combustion (e.g. maximum power A/F ratio, high turbulence, high compression ratio etc) will produce a large voltage swing at the probe but previous research work (92) aimed at quantitatively relating ionisation signal profile to combustion parameters was not successful.

II.2.2 Simple Electrical Analogy of Ionisation

The operation of the ionisation probe and the effect of combustion on the signal can be represented by an electrical analogy as shown in Fig. II.3. Here, the probe is represented by a capacitor $C_p$ and resistor $R_p$ which would correspond to the measured values under non-combustion conditions. The effect of the flame is represented by a switch $S_1$ and resistor $R_1$.

With switch $S_1$ open, only a small amount of current will flow due to the high resistance $R_p$ but during combustion, switch $S_1$ closes, thereby lowering the total circuit resistance by placing resistor $R_1$ in parallel with resistor $R_p$. This produces the voltage drop across supply resistor $R_s$ and thus causes capacitor $C_p$ to discharge at a finite rate, causing the voltage drop to increase with time.

As the reaction zone passes the probe, switch $S_1$ opens, increasing the circuit resistance and causing the voltage to increase, the rate of
increase being exponential and dependent on the relative value of
the resistances $R_p$ and $R_s$ and capacitance $C_p$.

Therefore, the signal obtained from this simple analogous circuit
would be similar to that produced by the actual ionisation circuit. This
type of analogy is very useful when optimising circuit design or when
considering the effect of operating conditions on gap resistance.

II.3 Details of Experimental Ionisation Equipment

II.3.1 Probe Construction

A diagram of a typical ionisation probe used for this study is
shown in Fig. II.4. These probes were made up in the department using
0.94 mm diameter nichrome wire for the central electrode, ceramic tube
of 1.96 and 1.0 mm outer and inner diameters respectively for the
insulator and a stainless steel tube of 2.7 mm outer and 2.1 mm inner
diameters for the outer electrode.

To increase durability of the probe, a spherical bead was formed
at the end of the inner electrode by heating the wire under an oxy­
acetylene torch. The three components were roughened on the mating
surfaces and then cleaned thoroughly in solvent before being bonded
together using Araldite epoxy resin strain gauge cement.

The electrical connections to the probe were obtained by either
the use of a B.N.C. connector and bracket arrangement or by soldering a
cable directly onto the inner electrode, the former arrangement only
being suitable where space permitted. The electrical resistance of the
probes were checked before installation using a 500 V insulation tester
(Mega), the resulting value normally being greater than 10 M.$\Omega$.

The ionisation probes were located in the combustion chamber by
drilling a 2.8 mm dia. hole in the cylinder head or squish plate and
cementing the probe in position using the Araldite cement. The end of
the probes were normally positioned 4 mm from the combustion chamber
wall to avoid undesirable quench zone effects.
II.3.2 Ionisation Amplifier

The purpose of the ionisation amplifier is to provide the supply voltage for the probe, to detect the change due to the ionisation and to provide a suitable output signal in phase with the occurrence of the flame arrival.

A block diagram of the main components of a single channel of the ionisation amplifier is shown in Fig. II.5 and the corresponding detailed circuit diagram is shown in Fig. II.6. A brief description of the main features of the ionisation amplifier is as follows:

A d.c. voltage, infinitely variable between 0 and 300 volts is supplied to the probe through a 330 KΩ resistor. The ionisation signal is smoothed to avoid spurious output signals by using a smoothing capacitor whilst the d.c. component of the signal is removed by employing an a.c. coupling capacitor.

A 741 operational amplifier was used as a comparator to detect any change between the ionisation signal, connected to the inverting input and a variable reference signal connected to the non-inverting input. A potentiometer was used to adjust the voltage level of the latter signal such that it was slightly lower than the noise level on the ionisation channel. A feedback resistor was chosen to give a large amplifier gain, thereby causing the output to rapidly change for a small decrease in the ionisation signal below the reference voltage.

The output signal obtained is basically a square wave with a pulse width corresponding to the width of the ionisation pulse at the reference voltage level, the leading edge of which is assumed to correspond to the time at which the flame front reached the probe.

Due to the fact that only a single supply of 5 V was provided for the electronic circuitry (rather than ±5 V) in an attempt to reduce the size of the amplifier unit, the output signal from the 741 varied between +2 V and +5 V which is not compatible with TTL (Transistor - Transistor Logic) circuitry. Therefore, interfacing was required between the 741 and the subsequent TTL, this being achieved using a BC108 transistor in conjunction with a decoupling capacitor and biasing bleed resistor. This now gives a voltage swing of 5 V to 0.6 V which is suitable for
the 74121 monostables which are used to give either a sharp 15 $\mu$s TTL pulse or a variable amplitude and pulse width signal suitable for recording on a magnetic tape recorder.

The circuitry described above was developed over a period of many months and represents a very reliable and accurate system. The latest unit built offers 8 channels powered from a single ionisation voltage supply, the problems being initially encountered with "cross talk" being successfully eliminated by using the smoothing capacitor.

II.3.3 Ionisation Processing Systems

The ionisation amplifier described above provides output signals with leading edges corresponding to ionisation. The next stage is to use some means of displaying, recording and/or analysing these signals. A number of methods for doing this have been reported by previous workers, notable Curry (27) and Harrow and Orman (28).

For this study, an original method employing a tape recorder and A.D.C. was used to obtain all the experimental results reported and this method is described in detail in Section 4.2. In addition, a "Flame Frequency Counter" was developed and used as a back-up system to the tape recorder/A.D.C. method to provide both a check on the equipment during testing and also as a means of cross-checking results.

The counter is similar to the Flame Arrival Frequency Counter first described by Harrow and Orman (28) and later used by Ma (33). A block diagram of the counter system used in this work is shown in Fig. II.7. Briefly, the unit detects, counts and displays the number of ionisation signals which occur within a specified crank angle interval and for a known number of cycles (either 100 or 1000). The crank angle interval is obtained from a crank shaft driven encoder providing a gate opening pulse at 90° BTDC, a reset pulse at 90° ATDC and a gate closing pulse at an adjustable crank angle.

The flame frequency counter can be used in several different ways depending on the information required:

a) To measure mean flame arrival angle directly. This can be quickly established by adjusting the position at which the variable pulse occurs until a 50% count is achieved. Due to cyclic dispersion
and crank angle resolution of the variable pulse, this method may be subject to significant errors.

b) To determine a flame arrival frequency/crank angle plot. This is achieved by adjusting the variable pulse in small steps, each time recording the flame arrival count, for about 10 values between say 5 and 95%. A typical plot is shown in Fig. II.8. Having constructed this plot, the mean flame arrival angle and flame dispersion can be established to a good degree of accuracy.

c) To determine the Lean Misfire Limit (L.M.L.). This could be arbitrarily defined as say 2% misfire at a crank angle of 45° ATDC and therefore with the variable set at this crank angle, the mixture would be weakened until a count of 98% was achieved. The unit shown in Fig. II.7 was provided with a facility for counting "Misses" as well as "Hits" and therefore for the above case, a miss count of 2% would correspond to the L.M.L.

II.4 Ionisation Probe Response Time Errors

Although a number of errors may be introduced when measuring flame propagation rates (e.g. crank angle errors) this section considers only the errors due to the ionisation detection circuitry. Curry (27) has shown that there is a definite time interval, called the response time, between the flame arrival and the time when the signal strength is sufficient to be detected.

The response for a typical signal is shown in Fig. II.9.

The response time is given by

\[ t_r = \frac{V_t}{V_s} \cdot t_t \]

where \( t_r \) = response time
\( t_t \) = total response time
\( V_t \) = trigger voltage offset.
\( V_s \) = total voltage swing

FIG II.9 Response Time Errors
The response time $t_r$ can therefore be reduced by:

a) Reducing $V_t$. This value is mainly determined by the "noise" level which should therefore be reduced to a minimum.

b) Increasing $V_s$. This is mainly affected by supply voltage, circuit component values and the combustion characteristics. The maximum value of $V_s$ will be restricted by the maximum voltage rating of the detection circuitry.

c) Reducing $t_t$. This is mainly a function of the combustion characteristics although it will also be affected by circuit component values.

Therefore, to minimise detection errors, the voltage swing should be kept to a maximum and the noise level should be minimised.

At the start of the work, tests were carried out to investigate the effect of fitting various smoothing capacitors across the ionisation probe, the object being to minimise the noise level and therefore reduce both the response time and the probability of spurious signals. The effect of a partially defective probe was also studied separately by soldering a 330 KΩ resistor in parallel with the probe. The effect of changing the effective resistance and capacitance of the probe are shown in Fig. II.10.

Inspection of the graphs shows that the value of these components can greatly affect the signals obtained. The reduced probe resistance, typical of the effect of carbon deposits, reduced the signal amplitude and also increase the noise level. A reliable output, free from spurious signals could only be obtained for this case by reducing the trigger voltage level, i.e. increasing $V_t$.

Graphs c to f of Fig II.10 show the effect of increasing the effective probe capacitance by various amounts. It was found that large values of parallel capacitance gave very smooth signals, free from noise but was unacceptable due to increased total response time $t_t$, reduced signal amplitude and very slow rise time following ionisation. Reducing the capacitance improved the signal response whilst giving a noise free signal. Very small values of capacitance resulted in noise spikes re-appearing.
The optimum value of capacitance tested was 22 nF, this addition resulting in good noise suppression, fast response and minimum signal attenuation. This allowed the trigger level voltage to be increased compared to the unmodified case and completely eliminated spurious signals caused by the ignition system or by "cross-talk" between the ionisation channels. A 22 nF smoothing capacitor was permanently soldered onto all channels of the ionisation amplifiers used to obtain the results presented in this thesis.

It was pointed out earlier that the ionisation signal is affected by operating conditions, in particular equivalence ratio and load. The effect of this is shown in Fig. II.11, the "slow" combustion conditions, resulting in a signal of reduced amplitude and increased total response time. This condition would therefore result in either increased response time, relative to "rapid" combustion condition or may even result in no output pulse, the latter condition is important since this could be interpreted as misfire.

To reduce the probability of this happening, it is best to adjust the probe supply voltage before each test to maintain a constant signal amplitude. The variation in supply voltage to maintain a swing of 4 volts is shown in Fig. II.12.

An alternative method used by Curry (27) is to measure the actual response time and to adjust the supply voltage to give a constant value. Since the response time is now known, a correction can be made to the measured flame arrival times to eliminate this source of error. The major drawback of this method is that it is difficult to measure to any degree of accuracy during an engine test, the small values of response times that would be encountered if the major sources of error (discussed above) have been reduced to a minimum.
**FIG II.1** BASIC COMPONENTS OF ION PROBE SYSTEM

**FIG II.2** TYPICAL IONISATION OUTPUT SIGNAL

**FIG II.3** ELECTRICAL ANALOGY OF FLAME IONISATION
Combustion End

Ceramic Tube 1.0mm i.d, 1.96mm o.d

Stainless Steel Tube 2.1mm i.d, 2.7mm o.d

Nichrome Wire Ø 0.94mm

FIG II.4 DETAILS OF IONISATION PROBE CONSTRUCTION

Ion Probe

330 kΩ Resistor

a.c Coupling Capacitor

Variable d.c Power Sup'y

15μs TTL output

20ms variable output

Compar't (op'amp) Mono-stable/ Interface Mono-stable/ P. Shaper

FIG II.5 BLOCK DIAGRAM OF ION AMP CIRCUITRY
FIG. 11-6 CIRCUIT DIAGRAM FOR IONISATION AMPLIFIER
FIG II-7  BLOCK DIAGRAM OF FLAME ARRIVAL FREQUENCY COUNTER
FIG IV.8 TYPICAL FLAME ARRIVAL GRAPHS FOR SHORT AND LONG DURATION FLAMES
Noise spikes superimposed on ionisation signal

Increased noise and reduced signal amplitude

Noise eliminated but small amplitude and slow response unacceptable

Some improvement over (c) but still not satisfactory

Smooth signal with good response and reasonable amplitude

Fast response but some noise interference present

FIG II.10 EFFECT OF ADDITIONAL R&C ON V-T ION SIGNALS
FIG II·11 CHANGE IN RESPONSE TIME DUE TO VARYING OPERATING CONDITIONS

FIG II·12 GRAPH SHOWING VARIATION OF SUPPLY VOLTAGE (FOR CONSTANT 4V SWING) WITH A/F RATIO
APPENDIX III

PIEZO-ELECTRIC TRANSDUCERS FOR THE
MEASUREMENT OF ABSOLUTE CYLINDER PRESSURE

III.1 Introduction
III.2 The need for accurate determination
   of absolute engine cylinder pressures
III.3 Techniques for achieving absolute
   pressures
APPENDIX III

III.1 Introduction

Piezo-electric pressure measurement systems have many advantages over other types of transducers (e.g. resistive, capacitive, inductive etc). When used for the measurement of pressures within the combustion chamber of fired engines. Typical advantages of piezo-electric systems are:-

I) Good linearity
II) High signal to noise ratio
III) Low temperature coefficient of sensitivity
IV) Good durability at high operating temperatures
V) Compactness
VI) High natural frequency
VII) Low hysteresis
VIII) High sensitivity adjustable to suit pressure range
IX) Good signal resolution.

These characteristics have resulted in the piezo-electric system having been used almost exclusively in recent engine studies.

The piezo-electric element, which is usually made from quartz, produces an electrical charge which is proportional to applied pressure. The charge produced is very small, and a charge amplifier with very high input impedance \( \approx 10^{13}\Omega \) is required to prevent the charge from being rapidly leaked away. However, some leakage will always occur resulting in signal drift. In addition, drift may be caused by electrical "noise", common sources of which are the ignition system and mains earth loops.

Signal drift is a major disadvantage of piezo-electric transducers and results in their being unsuitable for the measurement of static and slowly varying pressures. In dynamic situations, such as engine cylinder pressures, the dynamic or a.c. component of the signal can be measured very accurately but the absolute or d.c. component will not be known accurately due to the effect of drift.
The purpose of this Appendix is to first examine the need for accurate pressure data and then to review some techniques which have been used in the past to achieve the desired results.

III.2 The Need For Accurate Determination of Absolute Engine Cylinder Pressure

In some cases, for example when comparing the effects of engine modifications on the pressure diagram (e.g. i.m.e.p., peak pressures, rate of pressure rise etc) a small error in absolute pressure would make little difference to the results and an approximation could easily be made to give the desired results. In other cases, for example, when analysing the pressure-time diagram to calculate combustion rate, small errors may have a significant effect on the derived parameters.

Lancaster et al(72) has shown that a major effect of a constant error is to cause a change in the calculated value of the compression index $n$ which is more pronounced near to the start of compression. The reason for this is best demonstrated by considering the polytropic relationship

$$ P \cdot V^n = \text{Constant} $$

which between points 1 and 2 in a process may be re-written as:

$$ P_1 \cdot V_1^n = P_2 \cdot V_2^n $$

or in terms of $n$

$$ n = \frac{\log \left( \frac{P_1}{P_2} \right)}{\log \left( \frac{V_2}{V_1} \right)} $$

Therefore, an error in the pressure would result in a change in the pressure ratio and hence the calculated value of $n$.

The above expression has been used to calculate the effect of a $\pm 0.2$ Bar (3 p.s.i.) error on the compression index and the results are shown in Fig. III.1. The graph shows that the errors are greatest during the early stage of compression and for the throttled
conditions where the pressures are low, the value of $n$ varying between 0.92 and 2.1 for the worst conditions. The error in $n$ is much less near to TDC but average errors of about $\pm 0.06$ ($\pm 5\%$) may still occur due to the $\pm 0.2$ Bar pressure error.

Analytical methods for calculating mass burnt and heat release data (33, 74) using experimental pressure data determine the pressure rise due to combustion by subtracting the pressure charge due to piston motion, calculated using the polytropic relationship, from the measured pressure. Errors in either compression index or pressure would result in a change in the calculated value of mass burnt. The magnitude of such errors is discussed in Section 4.5.

To reduce such errors to a minimum, it is necessary to use some means of accurately locating the pressure signal relative to a known pressure. Methods that can be used to do this are discussed in the following subsection.

III.3 Techniques For Achieving Absolute Pressures

The object of these techniques is to determine the absolute pressure at a single point in the cycle since if this can be done, the absolute pressure for the whole cycle will be known.

The simplest method that may be employed is to assume a value of pressure during either the intake or exhaust strokes. Lancaster et al (72) for example assumed that the cylinder pressure at BDC of the intake stroke was equal to the mean intake pressure measured during the test whilst Brown (94) estimates that this assumption should be within about 0.07 to 0.14 Bar (1 to 2 p.s.i.) of the actual pressure for good breathing engines. Harrington (74) assumed that the pressure at the end of the exhaust stroke was equal to a "known value" but no details were given as to how this was measured.

Another technique which may be used is to install a valve arrangement which allows the transducer to periodically measure a known pressure instead of the cylinder pressure. The simplest form of this is shown in Fig. III.2. With valve A closed and valve B open,
either atmospheric or a known pressure will be measured, allowing
the charge amplifier to be zeroed (grounded) to this pressure or
alternatively, a reference pressure signal to be obtained.

The main disadvantage of this system is that errors may
be introduced due to the connecting passage between the combustion
chamber and pressure transducer. Brown (94) and Van Aken (95)
have shown that this type of arrangement can have an adverse effect
on the resonant frequency, the signal amplitude and the distortion
(apparent delay) and may produce transients in addition to modifying
the compression ratio.

The problems associated with the long connecting passage
are practically eliminated by using a switching adaptor. Kistler
manufacture a range of these devices and one of these, the type
642 which is designed for the measurement of pressures above an
adjustable reference pressure, is shown in Fig. III.3.

If the cylinder pressure is below the reference value, the
spool valve moves shutting off the cylinder port and causing the
transducer to measure reference pressure. If the cylinder pressure
exceeds the reference, the spool moves in the opposite direction
allowing cylinder pressure to be measured.

The effect of this on the pressure diagram is shown in
Fig. III.4. Since the clipping action occurs on each cycle, the
absolute pressure may be accurately determined (within the resolution
of the equipment) even in the presence of signal drift or d.c. shift
occurring in the data acquisition processing stages. A further
advantage is that the transducer may be "turned-off" when not required
by supplying a reference higher than the maximum cylinder pressure,
thereby increasing the life of the transducer. The main disadvantage
of the adaptor is the increased dimensions required for its installation.

A further method that may be employed is to use more than
one pressure transducer. The additional transducer(s) could be of
various designs and located in a number of positions in the engine.
One such set-up proposed by Brown (94) is to use a free disc balanced pressure indicator in addition to a piezo-electric transducer, the former device giving an electrical pulse each time the cylinder pressure changes sign relative to a known reference pressure. Another arrangement that could be used is shown in Fig. III.5 and uses a second transducer mounted in the cylinder wall which measures cylinder pressure near to BDC of each stroke. If this transducer is of a type which gives absolute pressures directly, no further information will be required but if a piezo-electric transducer is used, a reference to crankcase pressure can be obtained by drilling a hole in the piston as shown in Fig. III.5. If the crankcase pressure is well vented, atmospheric pressure may be assumed, otherwise, a further measurement device will be required.

Benson and Baruah (97) used three transducers mounted in the cylinder head, sleeve and sump to determine the absolute cylinder pressure. Although this type of system should give accurate results, it does have the disadvantage of increased instrumentation and data analysis.

A fourth method, proposed by Brown (94) is to use a solenoid operated miniature poppet valve mounted in the engine cylinder and connected to a manometer. This may be timed to open for a short period at the mid-point of the intake stroke where the pressure is fairly steady. The manometer would therefore register the mean pressure over the valve opening period for a large number of engine cycles.

To summarise, several methods are available for the measurement of absolute pressure, each method having advantages and disadvantages. The actual method used for a given situation would depend on the accuracy required, the amount of additional complexity that could be tolerated and the amount of space available for the installation of adaptors and additional transducers. Details of the method employed for the experimental part of this study are given in Section 4.3.
FIG III.1 EFFECT OF A 0.2 BAR ERROR ON THE CALCULATED COMPRESSION INDEX

Comp Ratio = 8:1
IVC = 40° ABDC

\[ n = \log\left(\frac{P_1}{P_2}\right) / \log\left(\frac{V_2}{V_1}\right) \]

\[ n = 1.28 \]

- \( P = 0.5 \text{ Bar at IVC} \)
- \( P = 1.0 \text{ Bar at IVC} \)
- \( P_{\text{err}} = 0.2 \text{ Bar} \)
- \( P_{\text{err}} = +0.2 \text{ Bar} \)
FIG III·2 METHOD OF PRESSURE REFERENCING USING REMOTE TRANSDUCER WITH 2 VALVES

FIG III·3 KISTLER 642 AUTOMATIC CLIPPER ADAPTOR

FIG III·4 EFFECT OF CLIPPER ON PRESSURE DIAGRAM

REFERENCE PRES

ACTUAL CYL PRESSURE
FIG III.5 ABSOLUTE PRESSURE MEASUREMENT USING TWO TRANSDUCERS

MAIN TRANSDUCER T1

REFERENCE TRANSDUCER T2

PISTON AT BDC
T2 MEASURING CYL' PRESSURE

PISTON AT TDC
T2 MEASURING CRANKCASE PRESSURE
APPENDIX IV

THE CALCULATION OF THEORETICAL SQUISH VELOCITIES IN MOTORED ENGINES
APPENDIX IV

IV.1 Introduction

The object of this Appendix is to give details of the calculation of the theoretical squish velocity in motored engines and to illustrate how such velocities are affected by the combustion chamber shape.

Calculations of this type serve the following purposes:

a) Allows the possible implications of chamber design to be assessed.

b) Allows the combustion chamber design to be classified according to the calculated charge velocities. This permits the performance of different designs to be compared in terms of the theoretical squish velocities.

The first part of this section gives details of the calculation of squish motion, including a review of some of the work reported in the literature. Calculated results showing the effect of the more important design parameters are then presented and discussed.

It must be emphasised that the calculations that follow are only applicable for non-combustion conditions since the expansion of the burning charge will greatly affect the motion in the squish zone. This effect, which does not appear to have been considered by previous workers, is illustrated and discussed in Chapter 7.4.

IV.2 Calculation of Squish Velocity

IV.2.1 Review of Previous Work

In 1952, Fitzgeorge and Allison (16) presented a paper outlining a method for calculating the radial velocity produced by a bowl in piston (B.I.P.) chamber design. By assuming uniform spatial density of the charge and zero blow-by, they showed that the squish velocity (the radial velocity at the inside edge of the bowl) is given by:

\[ U_s = \frac{(d^2/4 - r_b^2) \cdot V_b \cdot U_p}{2 \cdot r_b \cdot h \cdot (\pi \cdot d^2 \cdot h/4 + V_b)} \]
where

- $U_s$ = Squish velocity
- $U_p$ = Piston velocity
- $d$ = Bore
- $r_b$ = Bowl radius
- $V_b$ = Bowl volume
- $h$ = Piston to head distance

Ghirlando (51) calculated the squish velocity by considering the annular space outside the bowl as a one dimensional duct whose cross sectional area varied as a function of crankangle. His theory allowed for the effects of gas inertia, friction, heat transfer and piston ring blow-by, the resulting differential equations being solved numerically on a computer. The results obtained suggest that these effects for practical cases are, with the exception of blow-by, insignificant. Blow-by however had a major effect on the calculated velocities but the problem is knowing the amount of blow-by that is normal during the piston movement a few degrees either side of TDC of the compression stroke.

A computer simulation model developed by Gosman and Johns (98) solves the differential conservation equations governing the flow and heat transfer in an axi-symmetrical bowl in piston combustion chamber. Their results predict that the squish motion produces a vortex in the bowl during compression whilst on the expansion stroke, a vortex is generated above the bowl annulus. They also present results which suggest that the flow pattern will be drastically modified in the presence of swirl. In common with Ghirlando, they found that the squish velocities predicted by the simple Fitzgeorge and Allison formula were almost identical to the values given by far more complex and comprehensive models.

The previous work reviewed so far has considered the specific case of a bowl in piston design, which has received the most attention due to its popularity in compression ignition engines and due to the expected one dimensional flow in the squish zone. However, the majority of spark ignition engines have concentrated squish areas (e.g. wedge, bathtub etc) and it is desirable to be able to calculate squish velocities for these shapes as well. Lichty (17) derives an
expression for a concentrated squish area by using a similar approach to that used by Fitzgeorge and Allison. He obtained the following relationship:

\[
U_s = U_p \cdot \frac{A_s \sqrt{1 - (S + C_L)/(S + S_T/(C_R - 1))}}{b \cdot (S + C_L)}
\]

where
- \(U_s\) = Squish velocity
- \(U_p\) = Piston velocity
- \(A_s\) = Squish area
- \(S\) = Piston displacement
- \(b\) = Chord length
- \(C_L\) = Bumping clearance
- \(C_R\) = Compression ratio
- \(S_T\) = Stroke

The value of squish velocity given by this equation is the mean velocity based on the volume flow rate divided by the flow area. Although this equation appears to be different from that of Fitzgeorge and Allison, it is easy to show that they would both produce the same result for the same design.

IV.2.2. Derivation of a General Expression for the Determination of the Mean Radial Velocity

A more general expression than those already given can easily be derived for the mean radial velocity, which apart from being applicable to all squish designs also allows the mean velocity at any location to be calculated. (The previous expressions would give the maximum radial or squish velocity).

The derivation is based on uniform pressure distribution with no blow-by and proceeds by considering the volume change due to piston motion rather than the usual mass transfer approach since it is simpler and more direct.
Consider the combustion chamber shown in Fig. IV.1. The object is to derive an expression for the mean radial velocity at position \( X \).

Let the cylinder volume at this position be \( V_T \), the area of the section to the right of the chord at \( X \) be \( A_X \), the chamber height at \( X \) be \( h_X \) and the bore area be \( A_B \).

Let the piston move a very small distance \( dh \) in time \( dt \). The volume change in the sector at \( X \) is given by \( A_X \cdot dh \).

The volume ejected however is less than this due to a reduction in total cylinder volume and therefore the volume ejected is given by:

\[
A_X \cdot dh - A_X \cdot h_m \cdot \frac{A_B \cdot dh}{V_T} \quad \text{where:} \quad h_m = h_s \quad \text{if} \quad h_X = h_s
\]

\[
= A_X \cdot dh \cdot (1 - \frac{A_B \cdot h_m}{V_T})
\]

This is the volume of charge that is transferred from the squish side across the chord at \( X \) (in time \( dt \)).

The mean radial velocity is therefore the volume flow rate divided by the effective flow area.

\[
U_r = \frac{A_X}{A_{\text{eff}}} \cdot \frac{(1 - A_B \cdot h_m)}{V_T} \cdot \frac{dh}{dt}
\]

i.e.

\[
U_r = \frac{A_X}{A_{\text{eff}}} \cdot U_p \cdot \frac{(1 - A_B \cdot h_m)}{V_T}
\]
where \( U_r \) = Mean radial velocity
\( U_p \) = Piston velocity
\( A_X \) = Area on squish area side of \( X \)
\( A_B \) = Area of bore
\( h_m \) = Mean height as defined
\( A_{eff} \) = Effective flow area
\( = h_X \times \) chord length

This expression can be used to calculate the mean radial velocity at any location in the chamber. If, for example, the squish velocity is required, we would replace \( U_r \) by \( U_s \), \( h_m \) by \( h_s \) and \( A_X \) by \( A_B \) to give:

\[
U_s = \frac{A_B \cdot U_p}{A_{eff}} \left( 1 - \frac{A_B \cdot h_s}{h_X} \right)
\]

where \( U_s \) = Squish velocity
\( A_B \) = Squish area
\( h_s \) = Squish height

If we want to calculate the squish velocity for a bowl in piston design then, \( A_{eff} \) is replaced by \( 2\pi r_B \cdot h_s \).

**IV.3 Calculated Radial Velocities - Results and Discussion**

The expressions derived above were used to calculate the theoretical radial and squish velocities for both bowl in piston and concentrated squish type (in the form of a sector of a circle and referred to as sector type squish) chambers. The results are presented and discussed with the object of illustrating the effect of design and operating parameters on the squish motion.

Fig. IV.2 shows the variation of squish and piston velocity and volume flow rate (out of squish zone into bowl) with crank angle for a 50\% squish area B.I.P. design with 1 mm bumping clearance and at an engine speed of 1000 rpm. Although the results are only plotted up to TDC, the magnitude of these variables will be the same after TDC as those shown for crank angles before TDC, although of course they will be of opposite sign.
It can be seen that the squish velocity reaches a maximum value of 5 m/s at about 10 deg. BTDC and then reduces rapidly to zero at TDC due to the zero piston speed. It is helpful to compare the squish velocity values with the maximum piston speed of 3.8 m/s occurring at about 80 deg. BTDC. Also, although the squish velocity reaches a maximum value at about 10 deg. BTDC, the maximum volumetric flow rate occurs at about 32 deg. BTDC as shown, the reason for the difference being the effect of reducing squish height. Since the maximum charge activity will probably be affected by both the intensity and quantity of air movement, it is reasonable to assume that the maximum squish effect will occur at about 20 deg. either side of TDC. Therefore, the maximum charge activity due to squish approximately coincides with the crank angle at which combustion would normally be expected to occur.

Fig. IV.3 shows the variation in mean radial velocity with radius for both B.I.P. and sector type squish chambers. Inspection of the graphs shows that the calculated velocities in the squish area increase rapidly from zero at the chamber wall to a maximum value at the squish edge face. This value is the squish velocity.

The mean velocities calculated for the main part of the combustion chamber are much lower than the squish velocity and reduce to zero, either at the cylinder wall for the sector type or in the centre for the bowl in piston. The large reduction does not imply a sudden reduction in the charge velocity but suggests that a reverse flow will occur on the opposite face, therefore establishing a vortex type motion as shown in the small sketches in Fig. IV.3.

The variation of squish velocity with squish area and bumping clearance is shown in Figs. IV.4 and IV.5 respectively. As one would expect, increasing the squish area results in an almost proportional increase in the calculated squish velocities and (although not shown) a directly proportional increase in volumetric flow rate. For the two designs considered, a sector type squish design results in higher mean velocities than a B.I.P. chamber due to the smaller flow area for the former design.

Large increases in squish velocity are also predicted for reduced bumping clearance, especially for values less than about 1 mm. Fig. IV.5 shows that reducing the bumping clearance from 1.5 mm to 0.5 mm would about double the theoretical squish velocity for the same crank angle.
Another effect of reduced bumping clearance is that the crank angle for maximum squish velocity moves nearer to TDC.

Although reduced bumping clearance results in much higher velocities, the value of the volumetric flow rate is only slightly increased. Therefore it is likely that a large squish area/bumping clearance combination would result in higher charge activity than a small squish area/bumping clearance design. Furthermore, production engines would not normally be considered with bumping clearances of less than 1 mm due to manufacturing tolerances.

Other design features such as compression ratio, connecting rod length and bore to stroke ratio have little effect on the squish velocities for the practical range of values considered.

Finally, it should be emphasised that the results presented above assumes no piston ring blow-by and ignores "parasitic" volumes such as the crevise volume above the top piston ring. These effects will tend to reduce the squish velocities prior to TDC but a method of accurately predicting blow-by during the combustion period is not available.
FIG IV.2 EFFECT OF CRANK ANGLE ON SQUISH

PISTON VELOCITY

1000 RPM
3.0 BOWL IN PISTON
50% SQUISH AREA
1.0 MM BUMP CLEARANCE

VOLUME FLOW RATE

SQUISH VELOCITY

CRANK ANGLE (DEG)
Fig IV.3 PLOT OF RADIAL VELOCITY VS RADIUS

SQUISH VEL——

BOWL IN PISTON

SECTOR TYPE

1000 RPM
50% SQUISH AREA
1.0 BUMP CLEARANCE
10 DEG BTDC

MEAN RADIAL VELOCITY (M/S)

RADIUS FRACTION (R/RC)
**FIG IV.4 PLOT OF SQUISH VELOCITY VS SQUISH AREA**

1200 RPM
1.0 BUMPING CLEARANCE
10 DEG BTDC

**FIG IV.5 PLOT OF SQUISH VELOCITY AND VOL FLOW VS BUMP CLEARANCE**

1200 RPM
50% SQUISH AREA
10 DEG BTDC
### APPENDIX V

**CALCULATION OF CYLINDER CHARGE COMPOSITION**

<table>
<thead>
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<th>Section</th>
<th>Title</th>
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<td>V.2</td>
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<td>V.3</td>
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APPENDIX V

V.1 Introduction

This appendix gives details of the expressions and procedures used to calculate the composition of the burnt and unburnt charge compositions. Two computer program segments, namely EBURN and RESIDUL (listed in Appendix VIII), are used for these calculations and form part of the computer simulation programs described in Chapter 6.

For this work, the fuel was taken to be 150-octane \((C_8H_{18})\) and the working fluid was assumed to be fully represented by 12 constituents, namely \((CO_2, CO, N_2, H_2O, O_2, H_2, NO, OH, H, O, N \text{ and } C_8H_{18})\) - latter term for reactants only. In addition, several assumptions were made as follows:

a) Chemical equilibrium is achieved for all the constituents at gas temperatures exceeding 1600°K. At temperatures below 1600°K, the composition remains frozen at the value corresponding to 1600°K.

b) The state properties and species concentrations of the charge are completely homogeneous (i.e. no flame quenching and temperature variations).

c) The constituents behave as ideal gases.

These assumptions, whilst not being strictly true, considerably simplify the computations and should allow reasonably accurate results to be obtained for the purposes of this study.

V.2 Calculation of Equilibrium Composition

It is found that at high temperatures, chemical reactions can proceed in both directions, being both exothermic and endothermic. Chemical equilibrium is achieved when the forward and backward reactions are equal.

Consider the reaction

\[ a \text{ moles } A + b \text{ moles } B \rightleftharpoons c \text{ moles } C \text{ and } d \text{ moles } D \]
It can be shown (e.g. ref 71) that the ratio of the constituents at equilibrium is given by:

\[ K = \frac{c}{d} = \frac{(p_c)(p_B)}{(p_A)(p_D)} \quad -(V.1) \]

Where: \( K \) is the equilibrium or dissociation constant and for a perfect gas is a function of the temperature only.

\( p \) is the partial pressure of the constituent being considered and is given by:

\[ p_i = \frac{n_i}{n_t} \cdot P \quad -(V.2) \]

Where: \( P \) and \( n_t \) are the total pressure and total number of moles respectively.

In most cases, and certainly when a large number of chemical species are being considered, it is necessary to use a computer to solve the non-linear equations. In this work, the computational method developed by Brinkley (99) and later described by Phillips and Orman (9) and James (10) was used.

The method basically involves selecting "independent components" of the mixture with the remaining species being then regarded as "derived constituents". The independent components must fully define the overall system composition and will generally be equal in number to the chemical elements present in the system (in this case four).

The solution of the equilibrium equations is considerably shortened if the species with the greatest probable concentrations at equilibrium are selected. Three species which can be readily identified are CO\(_2\), N\(_2\) and H\(_2\)O whilst the fourth species would be CO or O\(_2\) for rich and weak combustion respectively.

Since the model was required to cover the full range of equivalence ratios, it was decided to follow the example of James and use 2 sets of equations, applicable to either rich or weak mixtures. Numbering the species 1-4 for the independent components and 5-11 for the derived
constituents, we obtain the species numbering system shown in Table V.1. This system will be used throughout the rest of this section.

<table>
<thead>
<tr>
<th>SPECIES NO</th>
<th>$\phi \geq 1.0$ (RICH)</th>
<th>$\phi &lt; 1.0$ (WEAK)</th>
<th>&quot;TYPE&quot;</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>CO$_2$</td>
<td>CO$_2$</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>CO</td>
<td>O$_2$</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>N$_2$</td>
<td>N$_2$</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>H$_2$O</td>
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<td></td>
</tr>
<tr>
<td>5</td>
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<td></td>
</tr>
<tr>
<td>6</td>
<td>H$_2$</td>
<td>H$_2$</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>NO</td>
<td>NO</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>OH</td>
<td>OH</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>H</td>
<td>H</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>O</td>
<td>O</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>N</td>
<td>N</td>
<td></td>
</tr>
</tbody>
</table>

**TABLE V.1**

**SPECIE NUMBERING FOR RICH AND WEAK MIXTURE RANGES**

The chemical equations by which each of the derived constituents may be formed from the independent components can now be written.

For rich mixtures we obtain:

a) $O_2 \rightarrow 2CO_2 - 2CO$

b) $H_2 \rightarrow H_2O + CO - CO_2$

c) $NO \rightarrow \frac{1}{3}N_2 + CO_2 - CO$

d) $OH \rightarrow \frac{1}{3}H_2O + \frac{1}{2}CO_2 - \frac{1}{2}CO$ - (V.3)

e) $H \rightarrow \frac{1}{3}H_2O + \frac{1}{2}CO - \frac{1}{2}CO_2$

f) $O \rightarrow CO_2 - CO$

g) $N \rightarrow \frac{1}{3}N_2$
Whilst for \textit{weak mixtures}

a) \[ \text{CO} \rightleftharpoons \text{CO}_2 - \frac{1}{2} \text{O}_2 \]

b) \[ \text{H}_2 \rightleftharpoons \text{H}_2\text{O} - \frac{1}{2} \text{O}_2 \]

c) \[ \text{NO} \rightleftharpoons \frac{1}{2} \text{N}_2 + \frac{1}{2} \text{O}_2 \]

d) \[ \text{OH} \rightleftharpoons \frac{1}{2} \text{H}_2\text{O} + \frac{1}{2} \text{O}_2 \] \quad -(V.4)

e) \[ \text{H} \rightleftharpoons \frac{1}{2} \text{H}_2\text{O} - \frac{1}{2} \text{O}_2 \]

f) \[ \text{O} \rightleftharpoons \frac{1}{2} \text{O}_2 \]

g) \[ \text{N} \rightleftharpoons \frac{1}{2} \text{N}_2 \]

The amount of derived constituents (\(n_5\) to \(n_{11}\)) can now be expressed in terms of the independent components using the general expressions for the equilibrium constant \(K(V,1)\) and for partial pressure \((V.2)\).

\textbf{For rich mixtures}

\begin{align*}
a) \quad n_5 &= K_5 \cdot (P/n_t)^{-1} \cdot n_1^2 \cdot n_2^{-2} \\
b) \quad n_6 &= K_6 \cdot n_1^{-1} \cdot n_2 \cdot n_4 \\
c) \quad n_7 &= K_7 \cdot (P/n_t)^{-\frac{1}{2}} \cdot n_1 \cdot n_2^{-1} \cdot n_3^{\frac{1}{2}} \\
d) \quad n_8 &= K_8 \cdot (P/n_t)^{-\frac{1}{2}} \cdot n_1^{\frac{1}{2}} \cdot n_2^{-\frac{1}{2}} \cdot n_4^{\frac{1}{2}} \\
e) \quad n_9 &= K_9 \cdot (P/n_t)^{-\frac{1}{2}} \cdot n_1^{-\frac{1}{2}} \cdot n_2^{\frac{1}{2}} \cdot n_4^{\frac{1}{2}} \\
f) \quad n_{10} &= K_{10} \cdot (P/n_t)^{-1} \cdot n_1 \cdot n_2^{-1} \\
g) \quad n_{11} &= K_{11} \cdot (P/n_t)^{-\frac{1}{2}} \cdot n_3^{\frac{1}{2}} \\
\end{align*}
and for weak mixtures

\begin{align*}
a) \quad n_5 &= K_5 \cdot (\frac{P}{n_t})^{\frac{3}{2}} \cdot n_1 \cdot n_2^{-\frac{1}{2}} \\
b) \quad n_6 &= K_6 \cdot (\frac{P}{n_t})^{\frac{1}{2}} \cdot n_2^{-\frac{1}{2}} \cdot n_4 \\
c) \quad n_7 &= K_7 \cdot n_2^{\frac{1}{2}} \cdot n_3^{\frac{1}{2}} \\
d) \quad n_8 &= K_8 \cdot (\frac{P}{n_t})^{-\frac{1}{2}} \cdot n_2^{\frac{1}{2}} \cdot n_4^{\frac{1}{2}} \\
e) \quad n_9 &= K_9 \cdot (\frac{P}{n_t})^{-\frac{3}{2}} \cdot n_2^{-\frac{1}{2}} \cdot n_4^{\frac{1}{2}} \\
f) \quad n_{10} &= K_{10} \cdot (\frac{P}{n_t})^{-\frac{1}{2}} \cdot n_2^{\frac{1}{2}} \\
g) \quad n_{11} &= K_{11} \cdot (\frac{P}{n_t})^{-\frac{1}{2}} \cdot n_3^{\frac{1}{2}} \\
\end{align*}

Where:

- $P$ is the total pressure of the system
- $n_t$ is the total number of moles of gas in the equilibrium mixture
- $K_5$ to $K_{11}$ are the equilibrium (dissociation) constants.

For the purpose of these calculations, perfect gas behaviour is assumed, resulting in the equilibrium constants being a function of temperature only. The value of $K$ is not dimensionless and the actual values depend on the actual partial pressure units used, these usually being expressed in terms of atmospheres. Also, the value of $K$ is dependent on the exact form of the chemical equation. For instance:

\[
H_2 + \frac{1}{2} O_2 \rightleftharpoons H_2O
\]

and

\[
2H_2 + O_2 \rightleftharpoons 2H_2O
\]

would produce different equilibrium constants.

The equilibrium constants used in this study are given in Table V.2 and are expressed in units of atmospheres. These values were obtained from the data given by Lichty (17) which for this work has been expressed in the form:

\[
\log_{10} K = a - \frac{b}{T}
\]
### PARTIAL Pressures IN Atmospheres

*(FROM LICHTRY-REF 17)*

\[
\log_{10} K = a - \frac{b}{T}
\]

<table>
<thead>
<tr>
<th>SPECIES No</th>
<th>SPECIE</th>
<th>(a)</th>
<th>(b)</th>
<th>SPECIES</th>
<th>(a)</th>
<th>(b)</th>
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<td>29055</td>
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<td>14528</td>
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<td>*</td>
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<tr>
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<td>9</td>
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<td>18413</td>
</tr>
<tr>
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<td>(O)</td>
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</tr>
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<td>(N)</td>
<td>3.484</td>
<td>19065</td>
<td>(N)</td>
<td>3.4840</td>
<td>19065</td>
</tr>
</tbody>
</table>

* For \(K_6\) (rich) use \(K = (0.102 + e (21.77 - 3.158 \times \ln(T)) \)

**TABLE V.2**

**Equilibrium Constant (K) Equation Coefficients**
\( T \) is the temperature, \( a \) and \( b \) are constants and this equation is applicable over the temperature range \( 1500 - 3000^\circ K \).

An exception to this was the value of \( K \) for \( H_2 \) in the rich mixture range defined by equation (V.5b). For this, \( K \) was given by

\[
K = 0.102 + e^{(21.77 - 3.158 \cdot \ln(T) + 0.102)}
\]

this being obtained from Ref.

A further set of equations based on the Conservation of Mass for the independent components may also be written. For rich mixtures, using equations (V.3a) - (V.3g), we get

a) \( n_1 = q_1 - 2n_5 + n_6 - n_7 - \frac{1}{2}n_8 + \frac{1}{2}n_9 - n_{10} \)
b) \( n_2 = q_2 + 2n_5 - n_6 + n_7 + \frac{1}{2}n_8 - \frac{1}{2}n_9 + n_{10} \)
c) \( n_3 = q_3 - \frac{1}{2}n_7 - \frac{1}{2}n_{11} \)
d) \( n_4 = q_4 - n_6 - \frac{1}{2}n_8 - \frac{1}{2}n_9 \)

Whilst for weak mixtures, using equations (V.4a) - (V.4g), we obtain

a) \( n_1 = q_1 - n_5 \)
b) \( n_2 = q_2 + \frac{1}{2}n_5 + \frac{1}{2}n_6 - \frac{1}{2}n_7 - \frac{1}{2}n_8 + \frac{1}{2}n_9 - \frac{1}{2}n_{10} \)
c) \( n_3 = q_3 - \frac{1}{2}n_7 - \frac{1}{2}n_{11} \)
d) \( n_4 = q_4 - n_6 - \frac{1}{2}n_8 - \frac{1}{2}n_9 \)

Where \( q_1 - q_4 \) represent the number of moles of each independent component in the absence of all derived constituents, these values being readily calculated from the known composition of the unburnt mixture. Details of the calculation of the unburnt mixture composition including the expressions for calculating the \( q \) values are given in Section V.3.
We have 12 unknowns \((n_1 - n_{11} \text{ and } n_t)\) and so far have only obtained 11 equations. A further equation is therefore required and this is obtained from the definition of \(n_t\):

\[
\begin{align*}
  n_t &= n_1 + n_2 + n_3 + n_4 + n_5 + n_6 + n_7 + n_8 + n_9 + n_{10} + n_{11} \quad (V.9)
\end{align*}
\]

These equations are solved using subroutine EBURN, a listing of which is given in Appendix VIII. Basically, the solution procedure is as follows:

a) The equilibrium constants for the specified mixture temperature are calculated using the values given in Table V.2.

b) A first estimate of the number of moles of each of the four components \(n_1 - n_4\) and total number of moles \(n_t\) is then made. (The initial values were set equal to the previously calculated mole fractions).

c) The values of the derived constituents, \(n_5 - n_{11}\), are then calculated, based on the estimated independent components, using equations (V.5) or (V.6).

d) The number of moles of the independent components are then calculated from equations (V.7) or (V.8), using the values calculated in step (c) above.

e) A check on the accuracy of the assumed values of the independent components used in step (c) is then made. If these had been correct, agreement with the number of moles calculated in step (d) would result. Generally, this would not be the case and therefore steps (c) to (e) are repeated, using a more accurate estimate of the independent components until the results agree to within 0.05\%. 
V.3 Calculation of Unburnt Charge Composition

The unburnt charge is regarded as a homogeneous mixture of humid air, gaseous octane fuel and residual exhaust gases. The specific humidity of the air, denoted by \( \omega \), is specified as data and was usually taken to be 0.8%, corresponding to a relative humidity of 55% at an air temperature and total pressure of 20°C and 1 Bar respectively.

The exhaust gas residual fraction \( r_f \) is specified as data and was normally taken to be 7%, a value used by Ref. 10. (This simplification is acceptable for the present study since parameters such as throttle opening, valve timing etc which greatly affect the exhaust residual fraction were not considered. Although compression ratio, equivalence ratio, combustion rate etc would cause a change in \( r_f \), this would tend to be relatively small over the range considered. The predicted effect of exhaust residual fraction is illustrated in Section 6.4).

The mass fractions of the fresh charge constituents are given by:

\[
M_a = \frac{(1 - r_f)}{(1 + \omega + \phi/15.06)} \quad (V.10)
\]
\[
M_f = M_a \times \phi/15.06 \quad (V.11)
\]
\[
M_v = M_a \times \omega \quad (V.12)
\]
\[
M_{o2} = M_a \times 0.233 \quad (V.13)
\]
\[
M_{n2} = M_a \times 0.767 \quad (V.14)
\]

Where:
- \( M_a \) = mass fraction of dry air
- \( M_f \) = mass fraction of fuel (C_8H_18)
- \( M_v \) = mass fraction of water vapour
- \( M_{o2} \) = mass fraction of oxygen (O_2)
- \( M_{n2} \) = mass fraction of nitrogen (N_2)
- \( r_f \) = mass fraction of residual
- \( \omega \) = specific humidity
- \( \phi \) = equivalence ratio
The residual gas composition is calculated using the equilibrium subroutine EBURN, described in V.2, assuming a temperature and pressure of 1600°K and 1 Bar respectively. Before this calculation can proceed, it is necessary to assign values for the number of moles of the independent components in the absence of the derived constituents, namely $q_1 - q_4$ as shown in equations (V.7) and (V.8)

The combustion equation for octane, air and moisture in terms of $q_1$ to $q_4$ is:

$$\left( C_8H_{18} \right) + 12.5 (O_2) + 12.5 \times 79.0 (N_2) + 12.5 \times 32.0 \times \omega (H_2O)$$

\[
\begin{aligned}
q_1 \cdot (CO_2) + q_2 (CO \text{ (rich or) or}) + q_3 (N_2) + q_4 (H_2O)
\end{aligned}
\]

- (V.15)

The values of $q_1 - q_4$ for rich mixtures, are then given by:

a) $q_1 = 2n_{O_2} - 17n_{C_8H_{18}}$

b) $q_2 = -2n_{O_2} + 25n_{C_8H_{18}}$

c) $q_3 = n_{N_2}$

d) $q_4 = n_{H_2O} + 9n_{C_8H_{18}}$

Whilst for weak mixtures:

a) $q_1 = 8n_{C_8H_{18}}$

b) $q_2 = n_{O_2} - 12.5n_{C_8H_{18}}$

c) $q_3 = n_{N_2}$

d) $q_4 = n_{H_2O} + 9n_{C_8H_{18}}$
Where:

\[ n_{N_2} = \frac{12.5 \times 79.0}{\varphi \times 21.0} \]

\[ n_{H_2O} = \frac{12.5 \times 32.0 \times \omega}{\varphi \times 18.0 \times 0.233} \]

\[ n_{O_2} = \frac{12.5}{\varphi} \]

\[ n_{C_6H_{18}} = 1.0 \]

Note that the values of \( q_1 - q_4 \) as given by equations (V.16) and (V.17) are in terms of moles of the constituent per mole of fuel.

These values are used by subroutine `EBURN` to calculate the residual exhaust gas mass and molar fractions. If \( M_p \) and \( M_R \) denote the mass fractions for the residual and unburnt charge respectively, we then obtain:

\[ a) \quad M_{R1} = M_R \cdot M_{p1} \]
\[ b) \quad M_{R2} = M_R \cdot M_{p2} \]
\[ c) \quad M_{R3} = M_R \cdot M_{p3} + n_{n2} \]
\[ d) \quad M_{R4} = M_R \cdot M_{p4} + M_v \]
\[ e) \quad M_{R5} = M_R \cdot M_{p5} + M_{O2} \]
\[ f) \quad M_{R6} = M_R \cdot M_{p6} \]
\[ g) \quad M_{R7} = M_R \cdot M_{p7} \]
\[ h) \quad M_{R8} = M_R \cdot M_{p8} \]
\[ i) \quad M_{R9} = M_R \cdot M_{p9} \]
\[ j) \quad M_{R10} = M_R \cdot M_{p10} \]
\[ k) \quad M_{R11} = M_R \cdot M_{p11} \]
\[ l) \quad M_{R12} = M_R \]

Where: \( M_R, M_{n2}, M_v, M_{O2} \) and \( M_f \) are as previously defined, and species (2) and (5) correspond to CO and O₂ respectively.

Having obtained the mass fractions, these can be readily converted to mole fractions using the molecular weights specified in Appendix VI.
The calculation of the unburnt charge composition using the expressions given is performed using subroutine RESIDUL, a listing of which is given in Appendix VIII.
APPENDIX VI

CALCULATION OF THE CYLINDER CHARGE PROPERTIES
APPENDIX VI

The working fluid in the combustion chamber of an I.C. engine is a complex mixture of chemical constituents, the composition and properties of which are continually changing. For the purposes of the computer simulation model described in Chapter 6, a means of evaluating the mean properties of the mixture during the cycle was required. The purpose of this Section is to present the thermodynamic data and expressions used to obtain the charge properties.

The mixture was assumed to be fully represented by 12 constituents (namely, CO₂, CO, N₂, H₂O, O₂, H₂, NO, O₃, H, O, N and C₈H₁₈ - latter term for reactants only), the actual molar/mass fractions being calculated using computer segments EBURN and RESIDUL which are described elsewhere. A further assumption was that the constituents behave as ideal gases with properties independent of pressure, this assumption being reasonably true for most gases.

It should also be noted that mixture properties can be evaluated using different approaches and that the properties can be expressed in terms of molar and/or mass units. The expressions presented here are representative of those used in the computer program (listed in Appendix VIII), for alternative expressions, text books such as ref. should be consulted.

All properties are calculated relative to the standard reference condition of 298.15₀K (25₀C).

Specific Heat

The molar specific heat at constant pressure, C_p, for each species is approximated by a pair of sixth order polynomials over the temperature ranges 273 - 200₀K and 2000 - 6000₀K. Hence,

\[ C_{pk} = a_0 + a_1 \cdot x + a_2 \cdot x^2 + a_3 \cdot x^3 + a_4 \cdot x^4 + a_5 \cdot x^5 + a_6 \cdot x^6 \]

which is equivalent to

\[ C_{pk} = \sum_{i=0}^{6} a_i \cdot x^i \]  - (VI.1)
Where \( a_i \) are the relevant polynomial coefficients

\[
x = T \times 10^{-3}
\]

\( k = \) Species number (1 to 12)

The polynomial coefficients used for the 2 temperature ranges are listed in Tables VI.1 and VI.2 and have been obtained from Ref. The coefficients shown would give \( C_p \) in units of kcal/kmol K and therefore to obtain SI units of kJ/kmol K, the expression should be multiplied by a conversion factor of 4.1868 as shown at the top of the tables.

For the majority of the calculations, a mean value of \( C_p \) over a temperature range \( T_1 \) to \( T_2 \) is required. The mean specific heat, \( \bar{C}_p \), is given by

\[
\bar{C}_p = \frac{1}{(T_2 - T_1)} \int_{T_1}^{T_2} C_p \cdot dT
\]

Since \( C_p \) is expressed as a function of \( T \) in terms of a polynomial shown in equation (VI.1), it is easy to integrate to give

\[
C_p = \sum_{i=0}^{i=6} a_i \cdot \chi^{(i+1)} + \text{Constant} \quad \text{-(VI.3)}
\]

If the upper and lower temperature limits, \( T_2 \) and \( T_1 \) respectively, both correspond to the same polynomial temperature range, then the value of \( \bar{C}_p \) can be evaluated using (VI.2) and (VI.3) directly. If however this is not the case, \( \bar{C}_p \) must be evaluated in 2 stages viz.

\[
\bar{C}_p = \frac{1}{(T_2 - T_1)} \left( \int_{T_1}^{T_2} C_{p_2} \cdot dT + \int_{T_1}^{2000} C_{p_1} \cdot dT \right) \quad \text{-(VI.4)}
\]

where \( C_{p_2} \) and \( C_{p_1} \) correspond to the upper and lower temperature range polynomial equations.

Although \( \bar{C}_p \) can be evaluated directly using (VI.4) a reduction in the computation necessary can be obtained by using the following approach:-
### TABLE VI.1
#### SPECIFIC HEAT (C_p)

**POLYNOMIAL EQUATION COEFFICIENTS**

\[
C_p = (a_0 + a_1 \cdot x + a_2 \cdot x^2 + a_3 \cdot x^3 + a_4 \cdot x^4 + a_5 \cdot x^5 + a_6 \cdot x^6) \times 4.1868 \text{ kJ/kmol K}
\]

<table>
<thead>
<tr>
<th>No</th>
<th>SPECIES</th>
<th>(a_0)</th>
<th>(a_1)</th>
<th>(a_2)</th>
<th>(a_3)</th>
<th>(a_4)</th>
<th>(a_5)</th>
<th>(a_6)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>CO₂</td>
<td>4.324933</td>
<td>20.808950</td>
<td>-22.945910</td>
<td>16.844800</td>
<td>-7.935670</td>
<td>2.121670</td>
<td>-0.240087</td>
</tr>
<tr>
<td>2</td>
<td>CO</td>
<td>7.812249</td>
<td>-6.668293</td>
<td>17.2629600</td>
<td>-17.287100</td>
<td>8.860130</td>
<td>-2.314820</td>
<td>0.244780</td>
</tr>
<tr>
<td>3</td>
<td>N₂</td>
<td>7.709929</td>
<td>-5.503897</td>
<td>13.121360</td>
<td>-11.679600</td>
<td>5.234000</td>
<td>-1.173190</td>
<td>0.103880</td>
</tr>
<tr>
<td>4</td>
<td>H₂O</td>
<td>7.988860</td>
<td>-1.506270</td>
<td>6.661376</td>
<td>-4.655970</td>
<td>1.696460</td>
<td>-0.370620</td>
<td>0.039920</td>
</tr>
<tr>
<td>5</td>
<td>O₂</td>
<td>7.361141</td>
<td>-5.369589</td>
<td>20.541790</td>
<td>-25.905300</td>
<td>15.945700</td>
<td>-4.858890</td>
<td>0.596150</td>
</tr>
<tr>
<td>6</td>
<td>H₂</td>
<td>6.183043</td>
<td>4.710657</td>
<td>-10.921360</td>
<td>12.540900</td>
<td>-7.016260</td>
<td>1.923400</td>
<td>-0.208410</td>
</tr>
<tr>
<td>7</td>
<td>NO</td>
<td>8.462334</td>
<td>-10.406690</td>
<td>27.548760</td>
<td>-30.281200</td>
<td>17.185100</td>
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<td>0.575530</td>
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<tr>
<td>8</td>
<td>OH</td>
<td>7.615100</td>
<td>-1.936000</td>
<td>0.877000</td>
<td>2.615300</td>
<td>-2.690900</td>
<td>0.977890</td>
<td>-0.126950</td>
</tr>
<tr>
<td>9</td>
<td>H</td>
<td>4.968000</td>
<td>0.000000</td>
<td>0.000000</td>
<td>0.000000</td>
<td>0.000000</td>
<td>0.000000</td>
<td>0.000000</td>
</tr>
<tr>
<td>10</td>
<td>O</td>
<td>5.974134</td>
<td>-4.241883</td>
<td>7.931254</td>
<td>-7.944230</td>
<td>4.403360</td>
<td>-1.271340</td>
<td>0.149140</td>
</tr>
<tr>
<td>11</td>
<td>N</td>
<td>4.966526</td>
<td>0.011505</td>
<td>-0.035335</td>
<td>0.04617</td>
<td>-0.03242</td>
<td>-0.01095</td>
<td>-0.00137</td>
</tr>
<tr>
<td>12</td>
<td>C₈H₁₈</td>
<td>2.547740</td>
<td>153.635900</td>
<td>-13.338200</td>
<td>-93.770700</td>
<td>70.979100</td>
<td>-16.495900</td>
<td>0.000000</td>
</tr>
</tbody>
</table>

**TEMPERATURE RANGE:** 273 - 2000°K
TABLE VI.2
SPECIFIC HEAT (Cp)

<table>
<thead>
<tr>
<th>No</th>
<th>SPECIES</th>
<th>$a_0$</th>
<th>$a_1$</th>
<th>$a_2$</th>
<th>$a_3$</th>
<th>$a_4$</th>
<th>$a_5$</th>
<th>$a_6$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>CO$_2$</td>
<td>8.153021</td>
<td>8.411419</td>
<td>-4.795179</td>
<td>1.54313</td>
<td>-0.28312</td>
<td>0.02766</td>
<td>-0.00111</td>
</tr>
<tr>
<td>2</td>
<td>CO</td>
<td>5.966461</td>
<td>3.288911</td>
<td>-1.660467</td>
<td>0.47645</td>
<td>-0.07854</td>
<td>0.00696</td>
<td>-0.00026</td>
</tr>
<tr>
<td>3</td>
<td>N$_2$</td>
<td>5.649167</td>
<td>3.579035</td>
<td>-1.794312</td>
<td>0.51213</td>
<td>-0.08419</td>
<td>0.00744</td>
<td>-0.00027</td>
</tr>
<tr>
<td>4</td>
<td>H$_2$O</td>
<td>3.401967</td>
<td>9.433046</td>
<td>-4.067415</td>
<td>1.04985</td>
<td>-0.16202</td>
<td>0.01377</td>
<td>-0.00049</td>
</tr>
<tr>
<td>5</td>
<td>O$_2$</td>
<td>8.439106</td>
<td>-0.376523</td>
<td>0.621716</td>
<td>-0.19235</td>
<td>0.02900</td>
<td>-0.00236</td>
<td>0.00009</td>
</tr>
<tr>
<td>6</td>
<td>H$_2$</td>
<td>4.103273</td>
<td>3.981784</td>
<td>-1.426509</td>
<td>0.26927</td>
<td>-0.01866</td>
<td>-0.00081</td>
<td>0.00012</td>
</tr>
<tr>
<td>7</td>
<td>NO</td>
<td>6.590193</td>
<td>2.604241</td>
<td>-1.291210</td>
<td>0.36511</td>
<td>-0.05937</td>
<td>0.00519</td>
<td>-0.00019</td>
</tr>
<tr>
<td>8</td>
<td>OH</td>
<td>4.946400</td>
<td>3.264500</td>
<td>-1.202600</td>
<td>0.25849</td>
<td>-0.03184</td>
<td>0.00207</td>
<td>-0.00005</td>
</tr>
<tr>
<td>9</td>
<td>H</td>
<td>4.968000</td>
<td>0.000000</td>
<td>0.000000</td>
<td>0.00000</td>
<td>0.00000</td>
<td>0.00000</td>
<td>0.00000</td>
</tr>
<tr>
<td>10</td>
<td>O</td>
<td>4.743426</td>
<td>0.480906</td>
<td>-0.364666</td>
<td>0.12739</td>
<td>-0.02117</td>
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<td>-0.00005</td>
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<td>11</td>
<td>N</td>
<td>4.845706</td>
<td>0.080796</td>
<td>0.087980</td>
<td>-0.10425</td>
<td>0.03658</td>
<td>-0.00487</td>
<td>0.00023</td>
</tr>
</tbody>
</table>

$C_p = (a_0 + a_1 \cdot x + a_2 \cdot x^2 + a_3 \cdot x^3 + a_4 \cdot x^4 + a_5 \cdot x^5 + a_6 \cdot x^6) \times 4.1686 \text{ kJ/kmol K}$

TEMPERATURE RANGE: 2000 - 6000°K
Let \[ \int C_p dT = f(T) \quad - (VI.5) \]
and \[ \int C_p dT = g(T) \quad - (VI.6) \]

From (VI.4)

\[
\bar{C}_p = \frac{1}{(T_2 - T_1)} \left( \left[ f(T) \right]_{T_2}^{T_1} + \left[ g(T) \right]_{T_2}^{T_1} \right)
\]

\[
\bar{C}_p = \frac{f(T_2) - g(T_1) - (f(2000) - g(2000))}{(T_2 - T_1)} \quad - (VI.7)
\]

The term \((f(2000) - g(2000))\) is a constant value, denoted by \(C_1\) and is easily calculated for the different species. The values obtained are shown in Table VI.3.

A general equation for \(\bar{C}_p\) can now be given

\[
\bar{C}_p = \left[ \sum_{i=0}^{6} a_i \cdot \frac{x(i+1)}{(i+1)} - C_1 \right] x_2
\]

\[
x_2 - x_1
\]

where \(a_i\) are polynomial coefficients corresponding to the range applicable to \(x\)

\[ x = T \times 10^{-3} \]

\(C_1\) = Constant as given in Table VI.3

= 0 for lower temperature polynomial

The specific heat for a mixture of gas calculated from:

\[
C_p = \sum C_{pk} \cdot \frac{n_k}{n_t} \quad - (VI.9)
\]

where \(\frac{n_k}{n_t}\) is the molar fraction of the \(k^{th}\) species.
### TABLE VI.3
SPECIFIC HEAT POLYNOMIAL INTEGRATION CONSTANTS

<table>
<thead>
<tr>
<th>No</th>
<th>SPECIES</th>
<th>$C_i$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>CO$_2$</td>
<td>1.08078</td>
</tr>
<tr>
<td>2</td>
<td>CO</td>
<td>-0.16134</td>
</tr>
<tr>
<td>3</td>
<td>N$_2$</td>
<td>-0.31135</td>
</tr>
<tr>
<td>4</td>
<td>H$_2$O</td>
<td>-1.61502</td>
</tr>
<tr>
<td>5</td>
<td>O$_2$</td>
<td>0.93325</td>
</tr>
<tr>
<td>6</td>
<td>H$_2$</td>
<td>-1.31135</td>
</tr>
<tr>
<td>7</td>
<td>NO</td>
<td>-0.00303</td>
</tr>
<tr>
<td>8</td>
<td>OH</td>
<td>-0.97955</td>
</tr>
<tr>
<td>9</td>
<td>H</td>
<td>0.00000</td>
</tr>
<tr>
<td>10</td>
<td>O</td>
<td>-0.31740</td>
</tr>
<tr>
<td>11</td>
<td>N</td>
<td>0.15448</td>
</tr>
<tr>
<td>12</td>
<td>C$<em>8$H$</em>{18}$</td>
<td>0.00000</td>
</tr>
</tbody>
</table>

### TABLE VI.4
MOLECULAR WEIGHT AND ENTHALPY OF FORMATION DATA (FROM REF. 10)

<table>
<thead>
<tr>
<th>No</th>
<th>SPECIES</th>
<th>$M_m$</th>
<th>$H_f(298.15K)$ (kJ/k mol K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>CO$_2$</td>
<td>44.011</td>
<td>-393785.0</td>
</tr>
<tr>
<td>2</td>
<td>CO</td>
<td>28.011</td>
<td>-110603.0</td>
</tr>
<tr>
<td>3</td>
<td>N$_2$</td>
<td>28.016</td>
<td>0.0</td>
</tr>
<tr>
<td>4</td>
<td>H$_2$O</td>
<td>18.016</td>
<td>-241989.0</td>
</tr>
<tr>
<td>5</td>
<td>O$_2$</td>
<td>32.000</td>
<td>0.0</td>
</tr>
<tr>
<td>6</td>
<td>H$_2$</td>
<td>2.016</td>
<td>0.0</td>
</tr>
<tr>
<td>7</td>
<td>NO</td>
<td>30.008</td>
<td>90351.0</td>
</tr>
<tr>
<td>8</td>
<td>OH</td>
<td>17.008</td>
<td>39490.0</td>
</tr>
<tr>
<td>9</td>
<td>H</td>
<td>1.008</td>
<td>218132.0</td>
</tr>
<tr>
<td>10</td>
<td>O</td>
<td>16.000</td>
<td>249362.0</td>
</tr>
<tr>
<td>11</td>
<td>N</td>
<td>14.008</td>
<td>472962.0</td>
</tr>
<tr>
<td>12</td>
<td>C$<em>8$H$</em>{18}$</td>
<td>114.232</td>
<td>-224287.0</td>
</tr>
</tbody>
</table>
Equations (VI.8) and (VI.9) are combined into a single function called CPMEAN which gives the mean molar specific heat for a specified temperature range and mixture composition.

The molar specific heat at constant volume, $C_v$, is calculated from

$$C_v = C_p - R_0$$  \hspace{1cm} -(VI.9)

$R_0$ is the Universal Gas Constant and has a value of 8.3143 kJ/kmol K

The ratio of the specific heats is denoted by $\gamma$ and is defined as

$$\gamma = \frac{C_p}{C_v}$$  \hspace{1cm} -(VI.10)

It is often more convenient to deal in units of mass rather than molar quantities. In this case the specific heat is denoted by $c$ and is defined as

$$c = \frac{C}{M}$$  \hspace{1cm} -(VI.11)

where $M$ is the molecular weight of the substance

**Molecular Weight**

The molecular weight values assumed for the 12 chemical species considered are given in Table VI.4. These values may then be used to obtain a molecular weight for the mixture using:

$$M_m = \sum n_m k \cdot \frac{n_k}{n_t}$$  \hspace{1cm} -(VI.12)

This can then be used to determine the specific gas constant for the mixture using

$$R = \frac{R_0}{M_m}$$  \hspace{1cm} -(VI.13)
Enthalpy and Internal Energy

Enthalpy values per unit mass, $H$, are calculated from the expression

$$H = \bar{c}_p(T - 298.15)$$  \hspace{1cm} -(VI.14)

where $\bar{c}_p$ is the mean specific heat for the temperature range 298.15 $-$ $T$ calculated from (VI.2) and (VI.11)

Similarly for internal energy, $U$

$$U = \bar{c}_v(T - 298.15)$$  \hspace{1cm} -(VI.15)

However, when chemical reactions occur, it is necessary to incorporate a Heat of Formation term, $H_f$, to allow the overall change in energy to be calculated. For this case, the following expression is used

$$H = H_f/M_m + \bar{c}_p(T - 298.15)$$  \hspace{1cm} -(VI.16)

$H_f$ is the enthalpy of formation of the substance under consideration at 298.15$^\circ$K and has units of kJ/kmol K. The values of $H_f$ used in this work were obtained from Ref. and are given in Table VI.4.

The enthalpy of formation for a mixture can be evaluated from the species values using

$$H_f = \sum H_{f_k} \cdot \frac{n_k}{n_t}$$  \hspace{1cm} -(VI.17)

Similar expressions can be used for calculating the Internal Energy and Internal Energy of Formation, $U_f$

$$U = U_f/M_m + \bar{c}_v(T - 298.15)$$  \hspace{1cm} -(VI.18)

and

$$U_f = \sum U_{f_k} \cdot \frac{n_k}{n_t}$$  \hspace{1cm} -(VI.19)
The $U_f$ values for the constituents can be obtained directly from the $H_f$ values given in Table VI.4, using

$$U_f = H_f - 298.15 \times R_0$$

i.e. $U_f = H_f - 2478.9$ -(VI.20)

Note that for most substances, the difference between $U_f$ and $H_f$ is small.
APPENDIX VII

CALCULATION OF COMBUSTION CHAMBER GEOMETRICAL PARAMETERS

VII.1  General
VII.2  Concentrated Sector Type Squish Designs
VII.3  Bowl in Piston and Hemispherical Head
VII.4  Dual Ignition
VII.5  Valve Recesses
APPENDIX VII

VII.1 General

The purpose of this appendix is to provide additional details of the geometrical simulation model described briefly in Section 6.3. In particular, the method and some of the expressions used for determining the burnt volume and surface areas for a range of combustion chamber designs, are described. Additional details of the expressions can be obtained from the program listing presented in Appendix VIII.

The values of the required parameters are usually determined by using a numerical integration method. This allows a large range of designs to be modelled and is relatively accurate and efficient in terms of computer requirements.

A general description of this method applied to a cylindrical disc chamber is presented in Section 6.3.3. Sections 2 to 4 of this appendix describe how this technique has been adapted to model squish type combustion chambers and dual ignition designs.

A further extension of the model was to include for the effects of cylindrical shaped valve type recesses. Details of how these can be simulated are given in Section VII.5.

Finally, it should be pointed out that in an effort to reduce computer time to a minimum, exact spherical relationships are used wherever possible. These expressions can be employed providing the flame front is circular at all cross-sections about a central axis, i.e. a solid of revolution.

Fig. VII.1 shows the general case of a parallel slice through a hemisphere of height h. The radius of the sphere is R and the radii of the top and bottom flat surface areas denoted by a and b respectively. The enclosed volume V and curved surface area S are then given by

\[ V = \frac{\pi \cdot h}{6} \left[ h^2 - 3a^2 + 3b^2 \right] \quad -(VII.1) \]

\[ S = 2\pi Rh \]
It can be seen from Fig. VII.1 that these expressions can be used in all cases where the flame front is solely restricted by the flat perpendicular faces of the cylinder head and/or piston crown. They cannot however be used where the flame front has either come into contact with a vertical cylinder or piston wall or alternatively has intersected with a second flame front (dual ignition case only).

In addition to calculating burnt volumes and surface areas, equations (VII.1) and (VII.2) are also used to permit hemispherical spark plug recesses to be modelled. These recesses are positioned with the spark plug electrodes at the centre of curvature and allowances are made for spark plug axial location.

### VII.2 Concentrated Sector Type Squish Designs

These designs are characterised by having either one or two squish areas built into the cylinder head in the form of sectors of a circle. Four designs, namely types 2 to 5 have been modelled and the general arrangements of these are shown in Figs. 6.6 and 6.7.

The types 4 and type 5 designs are specific cases of the types 3 design. Since the type 3 can be used to simulate the full range of parameters, there is strictly no need to bother with the type 4 and 5 designs. However, the latter two designs were developed prior to the type 3 and therefore these have been allowed to remain in the model. An additional reason for leaving these in the model is that they should be slightly more efficient than type 3 regarding computer run time.

The type 2 design has two sectors at 90° to each other in the form of a Vee. This design was modelled mainly due to the fact that this shape corresponds to some of the experimental squish plates used in this and other studies at Loughborough. Of the four types, this was the most difficult to model.

The general arrangement and important parameters defining the shape of a single sector is shown in Fig. VII.2. The size of the sector is specified in terms of percentage squish area and therefore it is necessary to calculate the angle \( \theta \) so that all other dimensions can then be determined.
From simple geometry, referring to Fig. VII.2 it is possible to write:

\[ A_s = R_c^2 \left( \theta - \frac{1}{2} \sin 2\theta \right) \]

Since the percentage squish area \( SQ \) is defined as:

\[ SQ = 100 \times \frac{A_s}{\pi R_c^2} \]

we finally obtain

\[ SQ = 100 \times \left( \theta - \frac{1}{2} \sin 2\theta \right) / \pi \]  

Equation (VII.3) is an implicit equation in \( \theta \) and can only be solved using a numerical method. In the model described, \( \theta \) is solved using an iterative process.

Having calculated \( \theta \), the other dimensions can then be readily determined. Referring to Fig. VII.2, we can write

\[ \text{PERP} = R_c \cos \theta - R_s \]  
\[ \text{PERPD} = R_c \cos \theta + R_s \]  
\[ \text{SPY} = R_c \sin \theta \]  
\[ \text{RM} = \sqrt{\text{PERP}^2 + \text{SPY}^2} \]  
\[ \text{RMD} = \sqrt{\text{PERPD}^2 + \text{SPY}^2} \]

The above values are calculated using subroutine DIM. Note that for the type 2 chamber, the value of \( SQ \) used in equation (VII.3) is only one half of the total specified percentage squish area.

The burnt volume and area parameters can now be calculated using subroutine SAVOL. The method used is basically the same as that described in Section 6.3.3 for the disc type chamber although the expressions used are different and less general.

To illustrate the computational method employed for concentrated squish type chambers, two cases corresponding to the type 4 and type 5 designs will be considered. For details of the other cases, the reader is referred to the program listing in Appendix VIII.
Type 4

Fig. VII.3 shows the general arrangement of the type 4 design at a specific axial location. For any given radius $R$, three distinct zones are encountered:

$$R < \text{PERP}$$ Proceed as for the disc chamber (See Section 6.3.3)

$$\text{PERP} < R < \text{RM}$$ For this region

$$A_b = \alpha \cdot R^2 + \text{PERP}^2 \times \tan \gamma$$ \hspace{1cm} (VII.9)

$$L_f = 2 \cdot \alpha \cdot R$$ \hspace{1cm} (VII.10)

$$L_b = 2 \cdot \tan \gamma$$ \hspace{1cm} (VII.11)

where

$$\gamma = \cos^{-1} \left( \text{PERP}/R \right)$$ \hspace{1cm} (VII.12)

$$\alpha = \pi - \gamma$$ \hspace{1cm} (VII.13)

and

$$A_b = \text{Burnt Area for ordinate}$$

$$L_f = \text{Flame Front Length for ordinate}$$

$$L_b = \text{Burnt Zone Heat Transfer Length for ordinate}$$

$$R > \text{RM}$$

For this region, simply calculate the values for a disc chamber and modify for the sector

$$A_b = A_b (\text{disc}) - A_s$$ \hspace{1cm} (VII.14)

$$L_f = L_f (\text{disc})$$ \hspace{1cm} (VII.15)

$$L_b = L_b (\text{disc}) + 2SPY - 29 \cdot RC$$ \hspace{1cm} (VII.16)

Type 5

Fig. VII.4 shows the general arrangement of the type 5 design at a specific axial location. For any given radius $R$, three distinct zones are encountered:
Proceed as for disc chamber
(See Section 6.3.3)

For this region, calculate the values for a disc chamber and modify for the sector

\[ A_b = A_b(disc) - \Psi \cdot R^2 + \text{PERPD}^2 \cdot \tan \Psi \]  
\[ L_f = L_f(disc) - 2\Psi \cdot \text{RC} \]  
\[ L_b = L_b(disc) + 2 \cdot \text{PERPD} \cdot \tan \Psi \]  

Where \[ \Psi = \cos^{-1} \left( \frac{\text{PERPD}}{R} \right) \]

As before, the disc chamber values are modified

\[ A_b = A_b(disc) - A_s \]  
\[ L_f = 0 \]  
\[ L_b = L_b(disc) + 2 \cdot \text{SPY} - 2 \cdot \Theta \cdot \text{RC} \]

The equations given above are used to calculate the area/length values at each of the axial ordinates as described in Section 6.3.3. Note that for axial locations greater than the height of the squish face, the values are calculated as for the disc chamber. Having considered all the axial ordinates, the burnt volume, flame surface area and the heat transfer vertical surface area are calculated using equation (6.31).

The burnt zone heat transfer surface area is made up of four areas, namely the vertical, head, piston and squish faces. The first of these is calculated using the method described above whilst the second and third are calculated using the area values at the first and last ordinates. The burnt surface area of the squish face parallel to the piston is calculated by determining the difference in the ordinate areas either side of this face.
VII.3 Bowl in Piston and Hemispherical Head

A major advantage of the numerical integration technique is that it allows any solid of revolution cross-sectional profile which can be defined as a function of axial location to be readily modelled. Furthermore, the bowl or hemisphere axis does not have to be co-incident with the cylinder axis.

In this work, both bowl in piston (B.I.P.) and hemispherical head (h.h) designs have been modelled. Both single and dual ignition cases have been included and for the single ignition designs, bowl/hemisphere offset can be specified.

For each of the b.i.p. and h.h designs, two types of profile can be simulated in the present model. These profiles are shown in Fig. VII.5, and are as follows:

a) B.I.P. dome. If the specified bowl corner/dome radius RR exceeds the calculated bowl radius RB, (based on specified squish area), a domed design is achieved. In this case, the bumping clearance is adjusted to give the correct compression ratio.

b) B.I.P. deep bowl. If RR is less or equal to the calculated value of RB, a cylindrical bowl with corner radius RR is modelled. In this case, the total bowl depth is adjusted to give the correct compression ratio.

c) h.h dome. This shape is achieved for low squish area and/or high compression ratio's when the required dome recess is less than that for a true hemisphere i.e. \(\frac{4}{5} \cdot \pi \cdot RB^2\). For this design, unlike the corresponding B.I.P, the specified bumping clearance is fixed and the dome radius RR and dome height SH are calculated using an iterative method to give the required recess volume.

d) h.h deep hemispherical recess. In cases where the required hemispherical recess volume exceeds the value for a hemisphere based on RB, the chamber shape becomes a full hemisphere on top of a cylinder. The height of the cylinder is varied to give the required compression ratio.
The designs described above were chosen since they represent common shapes and because they correspond to certain shapes which were needed to be modelled in this work (e.g. the B.I.P. used in the experimental study). Note that a true hemispheric recess can be achieved in both cases by specifying the correct squish area and bumping clearance.

Having calculated the chamber dimensions using subroutine DIM, the volume and surface areas can then be determined using subroutine SAVOL. The method used for these solid of revolution designs is basically the same as for the disc chamber described in Section 6.3.3. The main difference is that the actual chamber radius at the axial location under consideration is used rather than just the cylinder radius.

For example, consider the simplest case of a deep B.I.P. design with zero corner radius and bumping clearance $BC$ as shown in Fig. VII.6. For axial ordinates less than $BC + X$ from the head face, the bore radius $RC$ is used whereas for greater axial distances, the bowl radius $RB$ is substituted for $RC$ in equations 6.24 to 6.30.

In the present model, a bowl/hemisphere off-set can be specified so that the bowl axis and cylinder axis are not co-incident. This is achieved by specifying the off-set $RSB$ in terms of the distance between the spark plug and bowl/hemisphere axis, thereby permitting the off-set to be in any direction. This is then allowed for during the integration calculations by using $RSB$ rather than $RS$ (equations 6.24 to 6.30) whenever the ordinate being considered is inside the bowl/hemispherical recess.

Additionally, the spark plug location can be at any radius within the cylinder regardless of the profile of the bowl or hemispherical recess. This means that at many axial ordinate locations, the spark plug radial position may be outside of the combustion chamber boundary. In this case, the flame front/combustion chamber relationship is similar to the case of a cylindrical valve recess where the flame origin is outside of the recess volume. This type of situation is modelled for valve recesses using subroutine VALVE and therefore this same subroutine is also used for the solid of revolution situation described above. The expressions and method
used for valve type cylindrical recesses are described in Section VII.5.

The only other major difference between the B.I.P/h.h designs and the disc chamber is the additional surface area term due to the squish area annulus. This area is evaluated by determining the difference in the ordinate areas either side of the squish face.

VII.4 Dual Ignition

With multiple ignition, the flame fronts will initially be of the same shape as for single ignition but a point will be reached where the flame fronts intersect. As the flame radius increases, the shape of the flame front will be distorted by both the combustion chamber walls and by the other advancing flame fronts. It would therefore be very difficult to produce a model which was capable of simulating the geometrical parameters for a completely general multiple ignition design.

For the model described here, the simplest case of symmetrically located dual ignition is simulated. The assumption made is that the combustion chamber is completely symmetrical about a diametrical axis, thus restricting dual ignition to the disc, bowl in piston and hemispherical head (i.e. solid of revolution) designs. Also, valve recesses must be placed on the axis of symmetry and bowl/hemisphere off-sets are not permitted.

The symmetry assumption greatly simplifies the model since the flame fronts will always intersect on the axis of symmetry as shown in fig. VII.7. Due to the symmetry, it is possible to consider only one half of the chamber and then multiple the calculated values by a factor of two to obtain the correct results.

Since only one half of the chamber is being considered, it can be seen that the same ordinate areas and lengths would be obtained if a 50% squish area of the sector type had been assumed. Since this type of chamber had already been successfully modelled (i.e. type 5), it was relatively easy to make use of the same equations to cover the dual ignition cases.
Therefore, the method employed was basically the same as used for the type 5 chamber and this method and the expressions used are described in section VII.2. Only minor modifications to the type 5 model were required to incorporate the dual ignition simulation. Bowl in piston and hemispherical head designs with a full range of squish area and cross-sectional profiles (see Section VII.3) can be modelled by this method since these shapes only affect the value of the effective cylinder radius  in equations 6.24 to 6.30.

VIII.5 Valve Recesses

There were three reasons for wishing to extend the combustion chamber simulation to include for the case of valve type cylindrical recesses:

a) The experimental pressure data was obtained from an engine having a significant fraction of the clearance volume (approx. 9%) contained within the recesses. In the interest of accuracy, it was desirable to take this into account during the subsequent heat release analysis.

b) The incorporation of valve recesses into the model allows a considerable extension of the chamber shapes that can be simulated (see Figure 6.10).

c) The algorithm for calculating cylindrical recess parameters can be used for the case of bowl in piston and hemispherical head design where the radial spark plug location is outside of the bowl or hemisphere at the axial ordinate being considered. (See VII.3). The use of this algorithm allowed the spark plug position to be specified at any radius within the cylinder.

The valve recess volume and area parameters are determined using subroutine VALVE. A cylindrical rather than spherical flame profile is assumed for these calculations since this considerably simplifies the model (no need for numerical integration) and should only result in minor differences in most cases. The flame radius used is the radial component of the flame radius at the cylinder head which will only be equal to the actual flame radius if the spark plug penetration is zero.
Two cases have been considered in this model, namely single ignition and dual ignition. For both of these cases, recess diameter, depth and location can be specified but for dual ignition, symmetry about a centre-line is assumed. The expression used for the two cases are different and are described below.

**Single Ignition**

Fig. VII.8 shows the valve/flame relationship, together with the nomenclature used, for a typical flame radius RF where \((X - RV) < RF < (X + RV)\). The problem here is to calculate the area and curve lengths of the figure bounded by two arcs of known radii.

These parameters may be easily determined once the contact angles \(\theta\) and \(\beta\) have been determined. These are given by:

\[
\theta = \sin^{-1}\left(\frac{Y}{RF}\right) \quad -(VII.24)
\]

\[
\beta = \cos^{-1}\left(\frac{W}{RV}\right) \quad -(VII.25)
\]

The burnt volume \(V_b\), the flame front surface area \(A_f\) and the burnt zone heat transfer surface area \(A_b\) are then calculated using:

\[
V_b = \left[(RF^2 \cdot \theta - Z \cdot Y) + (RV^2 \cdot \beta - W \cdot Y)\right] \cdot HV \quad -(VII.26)
\]

\[
A_f = 2 \theta \cdot RF \cdot HV \quad -(VII.27)
\]

\[
A_b = 2 \beta \cdot RV \cdot HV + \frac{V_b}{HV} \quad -(VII.28)
\]

Note that the burnt zone surface area for the flat face, given by \(V_b/HV\) in equation (VII.28) is not normally required since this value will already have been accounted for in the main chamber calculations.

**Dual Ignition**

For the dual ignition case (described in Section VII.4) the symmetry assumption means that the valve recesses must be on a diametrical axis. This situation is shown in Fig. VII.9. It can be seen that if the flame radii are less than a certain value, denoted by \(RF_{min}\), equations (VII.24) to (VII.28) multiplied by a factor of 2 can be used. If on the other hand, \(RF\) is greater than
a maximum value, denoted by $RF_{\text{max}}$, the parameter values can be calculated from simple formula for a cylinder. The values of these critical radii are given by:

$$RF_{\text{min}} = \sqrt{RS^2 + (CV - RV)^2}$$  \hspace{1cm} -(\text{VII}.29)$$

$$RF_{\text{max}} = \sqrt{RS^2 + (CV + RV)^2}$$  \hspace{1cm} -(\text{VII}.30)$$

The problem here therefore is to determine the equations for the range $RF_{\text{min}} < RF < RF_{\text{max}}$. These are given by:

$$\theta = \left[ \theta_s + \cos^{-1}(RS/X) - \cos^{-1}(RS/RF) \right] / 2.0$$  \hspace{1cm} -(\text{VII}.31)$$

$$\beta = \left[ \beta_s + \cos^{-1}(CV/X) \right] / 2.0$$  \hspace{1cm} -(\text{VII}.32)$$

$$PV = CV - \sqrt{(RF^2 - RS^2)}$$  \hspace{1cm} -(\text{VII}.33)$$

Where $\theta_s$ and $\beta_s$ are the single ignition values calculated from equations (VII.24) and (VII.25).

The volume and areas are now given by:

$$V_b = \left[ (\theta \cdot RF^2 - \sin \theta \cdot \cos \theta \cdot RF^2) \right. \nonumber$$

$$+ \left. \beta \cdot RV^2 - PV \cdot RV \cdot \cos \theta \cdot (2\beta - \pi/2)^2 \right] 2HV$$  \hspace{1cm} -(\text{VII}.34)$$

$$A_f = 4 \theta \cdot RF \cdot HV$$  \hspace{1cm} -(\text{VII}.35)$$

$$A_b = 4 \beta \cdot RV \cdot HV + V_b/HV$$  \hspace{1cm} -(\text{VII}.36)$$

Where as in the single ignition case, $V_b/HV$ will normally be deleted.
Fig VII.1 SLICE THROUGH SPHERE OF RADIUS R

Fig VII.2 CONCENTRATED SQUISH DESIGN
Fig VII.3  TYPE 4 CHAMBER

Fig VII.4  TYPE 5 CHAMBER

a) BIP DOME  RR=RB
b) BIP DEEP BOWL RR=RB
c) HH DOME
d) HH DEEP RECESS

VII.5  B.I.P AND H.H DESIGNS SIMULATED
For $Z=(BC+X)$, substitute $RB$ for $RC$ in equations 6.24-6.30

**Fig VII.6** SIMULATION OF SOLID OF REVOLUTION DESIGNS

**Fig VII.7** INTERSECTION OF DUAL IGNITION FLAME FRONTS SHOWING SYMMETRY ABOUT CENTRE
Fig VII·8  CALCULATION OF VALVE RECESS PARAMETERS

Fig VII·9  CALCULATION OF VALVE RECESS PARAMETERS
FOR DUAL IGNITION
APPENDIX VIII

COMPUTER PROGRAM LISTINGS
**MASTER CONSIMP**

**COMPSIMP SIMULATES THE COMPRESSION/COMBUSTION AND EXPANSION**

**PROCESSES IN A SPARK IGNITION ENGINE OPERATING ON ISO-OCTANE**

**REAL N, LKM**

- **DIMENSION MDF(12), DHOL(12), CSA(30, 80)**
- **COMMON/BLOCK: 1, 73, RH, DOL, IS, 3A, JTCC, PERD, PEP D, RMAM, RMA D, B, PI**
- **COMMON/BLOCK: 2, GAIN, C, RFX, RIN, KEY, XIN, ST, RSH, SAW, SAH, SAI, SAS, SAF**
- **COMMON/BLOCK: 3, SITH, SPTX, CV, CIV, CIV, REX, HEA, HIN, NI**
- **COMMON/BLOCK: 4, PH(11), PH(11), RM(12), RM(12), MCEL(12), Q(4), PHI, JCM**
- **COMMON/BLOCK: 5, VR, RL, RS, ERS, RC, PH, SH, HDEF(12), THE**
- **COMMON/BLOCK: 8, CINV, MSA, HTM, THE, GLOSS, TV, WOLT, ERU, CEFF**

**DATA DHOI/4.011, 28.1, 24.1, 32.0, 2.016, 30.000, 17.008, 1.1008, 16.000, 1.14.232, HOF=-393785, 0, -110603, 0.0, -241969, 0.0, 20.0, 90351, 0.39400, 0.213132, 0.264362, 0.472962, 0.224287**

**READ DATA/ENGINE DETAILS, OPERATING CONDITIONS AND EMPIRICAL CONSTANTS**

**3 READ(1,730), PHI, VOLEFF, THEF**

- **IF(0.0, 0.0) GO TO 999**
- **READ(1,730), CRF, ST, CR, CII, TIVC, THEF, VOLEFF**
- **READ(1,740), QA, SOC, AC, RS, RSH, ER, PH, JTCC**
- **READ(1,770), KEY, ET, HEX, HIN, SPC, CIV, CIV, VCF**
- **READ(1,780), RES, EF, PAT, TAT, DTE, Tv**
- **READ(1,770), I, JCD**

**READ(1,750) TURF, CEFF, RND**

**N = ENGINE SPEED(REV/MIN)**

- **PHI = EQUIVALENCE RATIO**

**VOLEFF = VOLUMETRIC EFF(%)**

- **THEF = IONITION TIM1G(0EG ATDC)**

**BOM = CYL BORE (MM)**

- **ST = STROKE (MM)**

**CR = COMP PATIC**

- **CRI = CON ROD LENGTH (MM)**

**THEIVC = 1.V, C, A(DEG)**

- **THEF = T1, V, C, A(DEG)**

**SAQ = SOLID AREA (%)**

- **DQ = DISSOL AREA (%)**

**BC = BUNFING CLEARANCE (MM)**

- **PS = SPARK PLUG/CYL AXIS DIST (MM)**

**PR = PISTON BOIL RADIUS(MM)**

- **PH = PLUG PENTRATION (MM)**

**JTC = TYPE OF CHAMBER**

- **DF = VALVE OFFSET (MM)**

**REX = EX VALVE RECESS RADC(MM)**

- **TV = IN VALVE RECESS RAD(MM)**

**CEV = EX VALVE/AXIS DIST(MM)**

- **CIV = IN VALVE/AXIS DIST (MM)**

**HE = DEPTH OF VALVE DEC(MM)**

- **RS = RADIUS OF SPARK PLUG RECESS (MM)**

**RES = RESIDUAL GAS FRACTION**

- **UF = SPECIFIC HUMIDITY**

**PAT = ATMOSPHERIC PRESS(BAR)**

- **TAT = ATMOSPHERIC TEMP(DEG K)**

**TW = CYL WALL TEMP (DEG K)**

**NI = NO OF ITERATIONS**

- **JCM = MAX LOOP COUNT**

**TURF = TURB FLAME MULTIPLIER**

- **CEFF = COMB EFF COEFFICIENT (%)**

**RND = FULLY TURBULENT RAD(MM)**

**PI = 3.14159**

**TL = 298.15**

**RZ = 9.3143 / 4.1868 / 1000.0**

**E1BV = 40.0**

**AKCBV = 55.0**

**WP, 00 = 0.0**

**DFLP = 1.0**

**ANR = 0.948**
**CALCULATE CYLINDER WFAA DIMENSIONS**

\[
Fe = CR / (ST / 2, 0)
\]

\[
If (JTOCC, GE, 7) VOF = 0, 0
\]

\[
Ba = Pi * RC * RC
\]

\[
CIVOL = Ba + ST / (CR - 1, 0)
\]

\[
SR = RS * VOF
\]

\[
XIN = SQRT (SV * 2 + CV * 2)
\]

\[
XFX = SQRT (SV * 2 + CEV * 2)
\]

**CALL DIM**

\[
SO = (SOA + SOB) * 100 0 / BA
\]

\[
W = ASAE + SAP + SAB + SAS
\]

**CALCULATE MIXTURE COMPOSITION**

\[
Dn 22 1 = 1, 12
\]

\[
UNF (1) = VOF (1) - 2487, 9
\]

\[
WMOL (1) = MOI (1)
\]

\[
AFR = 15, 1 / PHF
\]

\[
RAT = PAT / PAT / 0, 2871 / 10000.0
\]

\[
RHA = PAT * 1, 0 * 04
\]

**CALL RESIDUAL (RES, WEF)**

\[
Un, VJ, HR, WJ, WMOLR = 0, 0
\]

\[
Dn 60 1 = 1, 12
\]

\[
WRITE (2, 790) RN (1), RN (1)
\]

\[
U2 = U1 * UNF (1) / WMOL (1) * RM (1)
\]

\[
U3 = HR + VOF (1) / WMOL (1) * RM (1)
\]

\[
WMOLR = WMOLR + RN (1) + WIOL (1)
\]

\[
RJ = 8, 3143 / WMOLR
\]

\[
WA = VOFEFF * ST + BA * RAT / 100.0
\]

\[
UB = WAI + (1.0 * WF + 1.0) / AFR / (1.0 - RES)
\]

\[
UT = WUB
\]

\[
WRITE (2, 785) VOLEFF, WUB, RJ, WMOLR, UR
\]

\[
USE = URF * CEFF
\]

\[
HF = (ST / 5.0 + N / 60.0) * 0.333 + 1.25E-07 * 4.186
\]

**SET CONDITIONS AT START OF COMPRESSION PERIOD**

\[
TRH = THE / 57.296
\]

\[
X1 = (ST / 2.0) * (1.0 - COS (RH) + ER - SORT (FR * FR - SIN (RH) * 2))
\]

\[
VOL = BA * X1 * CIVOL
\]

\[
P1 = PI V (VOL, EFF * 0.93 / 75.0 + RESG / 100.0)
\]

\[
TUB1 = P1 * VOL / WT / RII / 10000.0
\]

\[
THE = THEC1 / 6, 0
\]

**NEXT INTERVAL - CALCULATE VALUES AT END OF INTERVAL**

\[
J = 0
\]

\[
65 J = J + 1
\]

\[
OCA (1, J) = TRH
\]

\[
OCA (2, J) = VOL / 1000.0
\]

\[
OCA (3, J) = P1
\]

\[
OCA (4, J) = TUB1
\]
OSA(5, J) = P1/R1/TUB1*100.0
OSA(6, J) = GLOSS/THE/1000.0
OSA(7, J) = GAMMAR
OSA(R, J) = WD1/TTHF/1000.0
QR = QB + QLOSS
WD = WD + WD1
IF(THE < THE1) 0.150, 0
THE = THE + THEC1
IF(THE < THE1) 0.80
THE = (THEC1 - (THE - THES))/6.0/N
THF = THE
80 CALL COMP(TUB1, P1, VOL1, X1, 0.0, 0, 0)
GO TO 65

C

C SET INITIAL CONDITIONS AT IGNITION

C

150 JC = J
THEC1 = 2.0
DF1 = SAB1, W1, WB1, SAR1, VB1 = 0.0
DF1TR = 2.0
DF1TUB = 10.0
WB1 = W1
VB1 = VOI1 - VR1
RF2 = 0.2
RF4 = 0.3
TF0, TF, TBA = 2300.0
SA1 = USA + PI*X1*BORE

C

C NEXT INTERVAL - CALCULATE VALUES AT END OF INTERVAL

C

C

523 J = J + 1
THE = THE + THEC1
THEC1 = THEC1/6.0/N
TUB = TUB1
TF = TF + DF1TB

C

C CALCULATE ADIABATIC FLAME TEMPERATURE

C

C

1A0 TF = 0.5 + TF0 + 0.5*TF0
TF0 = TF
P = P1
CALL EIRVR( TF, P )
CDMP = CDMPFANT(T1, TF, 2)
WMOLP, WP = 0.0
DO 140 I = 1, 11
WP = WP + E1FC(I) / WMOL(I)*P1M(I)
140 WMOL = WMOLP + PNI(I) + WMOL(I)

C

C CALCULATE LAMINAR AND TURBULENT BURNING VELOCITIES

C

C

TF = (P1*CDMP/WMOL + (TUB - T1) - 4*P) / (CDMP/WMOL*P + T1)
IF(ABS( TF - TF0 ), GT 10.0 ) GO TO 160
VI1R = 1.0 + RF*TRF / (1.0 - TUR/TF)
IF(PHI, GT 1.0 ) ALBV = PHI
IF(PHI, GT 1.0 ) ALBV = 1.0/PHI
IF(PHI, GT 1.0 ) ALBV = 1.0/1.1 + SQRT(1.1 - PHI)*0.126
IF(PHI.IF.1.0.AND.PHI.GT.0.0) ALRV=0.0*(PHI-0.0)/2.0
VI=V0T((ALRV+TF*0.21*(1.0-ALRV*YLVBU)*EXP(-E1BV/RZ/TF)/((1.0+TUB/TF)))*0.375)*ANR*TUR/RBV/((1.0-TUB/TF))
VF=VI*10.0
RI=R1R

C ** ASSUME FINAL TEMPERATURES **
C
TP4=TP1+DELTP
TH4=TUR1+DELTUB
GO TO 214
214 TR4=0.0*TB4+0.4*TR4
TH4=0.0*TB4+0.4*TH4
211 TRF4=TR4
TH4=TH4
TRAV=(TR4+TP4)/2.0
THAV=(TH4+TH4)/2.0
C ** CALCULATE RAINY CHARGE COMPOSITION BASED ON MEAN VALUES FOR INTERVAL **
C
C COVR1=CPMFA:TL,TUR1,0
C COVR2=CPMFA:TL,TH4,0
C CPMP1=CPMFA:TL,TRAV,2
C CPMP2=CPMFA:TL,TR4,2
C MVRF1=CMRF1-8.3143
C MVRF2=CMRF2-8.3143
C MVMP1=CMMP1-8.3143
C MVMP2=CMMP2-8.3143
C GAMMAR=CMRR2/CMMR2
C GAMMAP=CMMP2/CMMP2
C WOLP,U4=0.0
C DO 180 1=1,11
WOLP=WOLP+1.0*(PN(I)+WOL(I))
180 UD4=UP4*UNF(I)*PN(I)/WOL(I)
UD4=UP4*CEF
R=8.3143/U4
THAV=100.0/TURAV/TH
WITF(2,795)RHO,TR4,CMRF1,CMRF2,CMMP1,CMMP2,WOLP
WRITE (7, 795) GAMMA, GAMMAR, QLOSSB, QLOSSUB, RP, WMOLP, SAB4, SABU4, SAB4=SAB4
RF4=FIAPAD(RF4, X4, V4)
SAR4=SAR4+SAD+SAR4+SAS
SAUB4=SHA+DI+ROF*X4=SAR4
IF (SAB4.LT.1.0) SUB4=1.0
SAVRE=(SAUB4+SAB4)/2
SAVB=(SAB1+SAR4)/2
JTIME=JTIME+1
IF (JTIME.GE.JCM) GO TO 197
C
**********
C CHECK INITIAL ASSUMPTIONS, IF UNACCEPTABLE, REPEAT CALCULATIONS
C
**********
IF (ABS(RA4-TBF4).GT.20.0) GO TO 210
IF (ABS(SAB4-SAB4).LT.0.01) GO TO 188
GO TO 197
CONTINUE
IF (JTIME.GE.JCM) WRITE (2, 940)
DEFRA=UX=2-VB1
DEFTR=TR4-TR1
DEFTR=TR4-TR1
DEFO=54-P4
S1=1.2-0.2-PAT)*R14.VOL4-VOL1/10000.0

C
**********
C CALCULATE SQUISH VELOCITY FOR TYPE A CENTRAL IGNITION CASE
C
**********
SVEA=0.0
IF (JTIME.GE.6.0) GO TO 638
IF (RF4.GT.8.0) GO TO 639
RHOU1=PB4/RU1/THU1+100.0
RHOU4=PB4/RU4/THU4+100.0
SVEA=(S1/100.0*BA*((X1+RC)*RHOU1-(X4+RC)*RHOU4)/((X1+X4+2.0+BC)*2
10+PL+RR*(RHOU1+RHOU4)*TTFH*250.0)
GO TO 638

RHO1=PB4/RU1/TH1+100.0
RHOU4=PB4/RU4/THU4+100.0

SVEA=((X1*X4+2.0+BC)*2.0*PL+RR*(RHOU1+RHOU4)*TTFH*250.0)
SVEA=(VOL1-SQ/100.0*BA*(X4+RC))*RHOU4-(VOL1-SQ/100.0*BA*(X1+RC))
GO TO 638
CONTINUE
UP1=UP4
TUB1=44
T61=TB4
VIR1=VIR4
VR1=VR4
PL=P4
SA1=SA1
SAUR4=SAUB4
UR1=UR2
WIR1=WIR2
FSPEED=(RF4-RF1)/TTFH
TF=(RF4-RF1)/TTFH
RF4=RF4
X1=X4
VOL1=VOL4
OSA(1, J)=THF
OSA(2, J)=VOL4/1000.0
OSA(3, J)=P4
OSA(4, J)=TB4
OCA(5, J) = TUB4
OCA(6, J) = WP/WT*100.0
OCA(7, J) = VR4/VOL4*100.0
OCA(8, J) = FSPEED/1000.0
OCA(9, J) = BF4
OCA(10, J) = TF
OCA(11, J) = VJ/1000.0
OCA(12, J) = PG/RJ/TUB4*100.0
OCA(13, J) = PL/RP/TR4*100.0
OCA(14, J) = (LOSSR + QLOSSUR) / TTHE/1000.0
OCA(15, J) = WDI/TTHF/1000.0
OCA(16, J) = WO/OLP
OCA(17, J) = TFV/1000.0
TO(SOVEL.NE.0.0) OCA(16, J) = SOVEL
DO 556 J = 18, 28

556 OCA(J, J) = ON(J-17)*1.0E02
WDP = TF(2, 795) P4, TR4, TUB4, VR4, VR2, RF4, FSPEED, THE
IF(VR1.LT.0.99*VOL4) GO TO 523

C EXPANSION PROCESS
C
620 THFCI = 6.0
WP = VT
JCOM = J
K = 0

625 THE = THEF + THECI
IF(X-1) .GT. 666, 666
IF(THF-EFFVN) 630, 0, 0
THF = (THECI - (THE - THEEVN)) / 6.0/N
THF = THEE/VN
K = 1
GO TO 635

630 THE = THFCI / 6.0/N

C NEXT INTERVAL - CALCULATE VALUES AT END OF INTERVAL
C
635 J = J + 1
CALL COMP(TR1, P1, VOL1, X1, UP1, 2)
OCA(1, J) = TF
OCA(2, J) = VOL1/1000.0
OCA(3, J) = P1
OCA(4, J) = TB1
OCA(5, J) = P1/RP/TR1*100.0
OCA(6, J) = QLOSS/TTHE/1000.0
OCA(7, J) = GAMMAR
OCA(8, J) = WDI/TTHF/1000.0
DO 654 J = 18, 28

654 OCA(J, J) = ON(J-17)*1.0E02
Wn = Wn + WP
Qn = Qn + QLOSS
GO TO 625

665 JF = J
Jn = JC + 1
JCOM = JCOM + 1
ON = ON*N/1220000.0
Pn = Pn*N/120000.0

C END OF CYCLE CALCULATIONS, OUTPUT RESULTS ON LINE PRINTER
C
WRITE(2, 800) N, ROPE, SPARK, ST, VOICE, CFI
WRITE(2, 824) DH, PO, AN, PE, SUG, IT0C, TAT
WRITE(2, 836) SO, PAT, SG, PMA, RSP, RNOAT, RIN, RFX, XIN, XEX
WRITE (2, 850) HV, RSD, UT, RII, UMCP
WRITE (2, 951)
WRITE (2, 955)
WRITE (2, 956)
WRITE (2, 952)
WRITE (2, 957)
WRITE (2, 958)
WRITE (2, 920) ((OSA (L, K), I = 1, A), K = 1, J)
WRITE (2, 952)
WRITE (2, 957)
WRITE (2, 924) ((OSA (L, K), I = 1, 9), K = 9, JCOM)
WRITE (2, 959)
WRITE (2, 960)
WRITE (2, 924) ((OSA (1, K), (OSA (1, K), I = 10, 17), K = JD, JCOM)
WRITE (2, 953)
WRITE (2, 955)
WRITE (2, 956)
WRITE (2, 920) ((OSA (L, K), I = 1, A), K = DOM, JE)
WRITE (2, 954)
WRITE (2, 961)
WRITE (2, 962)
WRITE (2, 938) ((OSA (1, K), (OSA (1, K), I = 18, 2A), K = JD, JE)
WRITE (2, 970)
WRITE (2, 980)
GO TO 3
710 FORMAT (F90.0)
720 FORMAT (F90.0)
730 FORMAT (F90.0)
740 FORMAT (7F0.0, 10)
750 FORMAT (3F0.0)
760 FORMAT (210)
780 FORMAT (1H, OF12.3)
785 FORMAT (1H, 5X, SF15.3)
790 FORMAT (1H, 2F15.4)
795 FORMAT (1H, 5X, 7F14.4)
800 FORMAT (1H1, ?), 'COMPUTER SIMULATION OF COMBUSTION IN A SPARK IGNITION ENGINE', 21X, 1G ('=1')//
21X, 'TEST CONDITIONS ARE', 30X, 'ENGINE DETAILS ARE', 20X, 18X, 5=/
41X, 'ENGINE SPEED (RPM)', 13X, '=' F7, 10X, 'BURF (MM)', 28X, '=' F6, 2/5
61X, 'IGNITION TIMING (DEG BTDC)', 9X, '=' F5.1, 12X, 'STROKE (MM)', 26X, 7=' F6.2//
81X, 'VOLUMETRIC EFFICIENCY (%)', 6X, '=' F5.1, 12X, 'ICON ROD LENGTH (MM)
90X, 11X, '=' F6.1)
824 FORMAT (1H0, 'EQUIVALENCE RATIO', 14X, '=' F5.2, 12X, 'COMPRESSION RATIO
11, 20X, '=' F5.2//
21X, 'AIR TO FUEL RATIO', 14X, '=' F6.2//
31X, 'WALL TEMPERATURE (DEG K)', 7X, '=' F6.1//
41X, 'RESIDUAL MASS FRACTION (%)', 9X, '=' F5.1//
51X, 'COMBUSTION CHAMBER DETAILS ARE', 19X, 'ATMOSPHERIC CONDITIONS ARE
6F13 (1=-1), 19X, 25 (1=-1)///
71X, 'SQUISH PLATE TYPE', 14X, '=' 13, 14X, 'TEMPERATURE (DEG K)', 19X, '='
81, 6X, 'F6.1)
836 FORMAT (1H0, 'SQUISH COVERAGE (%)', 12X, '=' F5.1, 12X, 'PRESSURE (BAR)
11, 25X, '=' F5.7//
21X, 'BUMPING CLEARANCE (MM)', 9X, '=' F5.2, 12X, 'SPECIFIC HUMIDITY (%)
31, 16X, '=' F5.1//
41X, 'SPARK PLUG/OHub AXE (MM)', 6X, '=' F5.1, 12X, 'DENSIty (KG/MM**3)
5, 20X, '=' F5.2//
61X, 'INTake VALVE RADIUS (MM)', 4X, '=' F5.1//
71X, 'EXHAUST VALVE RADIUS (MM)', 6X, '=' F5.1//
81X, 'INTake VALVE/SPARK PLUG (MM)', 4X, '=' F5.1//
91X, 'EXHAUST VALVE/SPARK PLUG (MM) =', F5.1)
850 FORMAT(1HO, 'VALUE RECESS DEPTH (MM)', RX, ' = ', FS, 2/1)
11X, 'SPARK PLUG RECESS RADIUS (MM)' = ', FS, 2/1
21X, 'CALCULATED MIXTURE PARAMETERS (/30) = ', FS, 2/1
31X, 'TOTAL MASS OF CHARGE (G)' = ', FS, 4/1
41X, 'GAS CONSTANT (KJ/KG DEG K)' = ', FS, 6.3/1
51X, 'MOLECULAR WEIGHT', 15X, ' = ', FS, 2/1
940 FORMAT(1HO, '***WARNING---MAXIMUM LOOP COUNT EXCEEDED AT LINE 197
114 COMPRESS***')
951 FORMAT(1HO, 30X, 'COMPRESSION PERIOD')
952 FORMAT(1HO, 30X, 'COMBUSTION PERIOD')
953 FORMAT(1HO, 30X, 'EXPANSION PERIOD')
954 FORMAT(1HO, 30X, 'COMBUSTION/EXPANSION PERIOD')
955 FORMAT(1HO, 'CRANK ANGLE', 7X, 'VOLUME', RX, 'PRESSURE', 6X, 'TEMPERATURE
1E1, 7X, 'DENSITY', 7X, 'HEAT LOSS', 7X, 'GAMMA', 7X, 'POWER')
1', 11X, 'KW', 28X, 'KU')
957 FORMAT(1HO, 'CRANK ANGLE', 4X, 'VOLUME', 3X, 'PRESSURE', 11.2X, 'BURNT TEMP',
11.2X, 'UNBURNED MASS BURNT FLAME SPEED FLAME RAD')
1, 'X', 11X, 'X', 10X, 'M/S', 10X, 'MM')
959 FORMAT(1HO, 'CRANK ANGLE', 4X, 'AD FL TEMP FLAME SP', 4X, 'UNB DENS',
2 fl Sd1')
960 FORMAT(1HO, 3X, 'DEGREES', 6X, 'DEG K', 8X, 'K/S', RX, 'KG/M+3', 6X, 'KG/M+3
1', 11X, 'X', 8X, 1, 'X', 19X, 'K/S')
961 FORMAT(1HO, 'CRANK ANGLE', 2X, 'DEG K', RX, 'DEG K', 21X, 'H2O', RX, 32
970 FORMAT(1HO, 10X, 'POWER OUTPUT = ', FS, 2/1)
980 FORMAT(1HO, 10X, 'HEAT TRANSFER = ', FS, 3.3, 'KW')
920 FORMAT(1HO, 'F11.3', 'F15.3')
924 FORMAT(1HO, 'F11.3', 'F12.3')
926 FORMAT(1HO, 'F11.3', 'F12.3')
938 FORMAT(1HO, 'GPE 10.1, 5F10.2, 4D6F10.2')
900 STOP
END
MASTER COMPARED

** *COMPARED CALCULATES COMBUSTION PARAMETERS (FLAME SPEED, **
** MASS BURNT ETC) FROM EXPERIMENTAL PRESSURE-CRANK ANGLE DATA **
**
**
REAL N,LKM
DIMENSION OSA(27,50),PACT(50)
DIMENSION OOF(12),DMOL(12)
COMMON/BLOCK 1/RR,PR,SQA,SRG,BA,JTOCC,PERP,PERPD,RHAM,RHAMD,B,PI
COMMON/BLOCK 2/GMIN,C,REX,RIN,XEX,XIN,ST,RS,SAW,SAN,SAP,SAS,SAF.
COMMON/BLOCK 3/SPTH,SPTHX,SV,CIV,CR,HEX,HIN,NI
COMMON/BLOCK 4/PN(11),PH(11),RN(12),RM(12),WMOL(12),Q(4),PHI,JCM
COMMON/BLOCK 5/VB,PC,RS,RSB,BC,PH,SH,UOF(12),TTH
COMMON/BLOCK 6/GRN,C,REX,RIN,HEX,HIN,RS,CEV,CIV,VOF
COMMON/BLOCK 7/HF,THE,THEV,THEEO
COMMON/BLOCK 8/TV,TVF,TVFSET,MR
COMMON/BLOCK 9/PAT,GAMBL,WDI
DATA DMOL/44.011.28.011,28.016,18.00,2.016,0.008,17.008,1.1008,14.100,16.100,14.100,
DATA(ENGINE DETAILS,OPERATING CONDITIONS AND PRESSURE DATA)
READ(1,730)\,PHI,VOEFF,THES
READ(1,720)\,PORE,ST,CR,CRL,THEIVC,THEEO
READ(1,740)\,SQA,SRG,BC,RS,RSR,PR,PHJ,JTOCC
READ(1,710)\,REX,RIN,HEX,HIN,SP,CEV,CIV,VOF
READ(1,720)\,ES,UF,PAT,TAT,PIVC,TW
READ(1,770)\,I,JCM
READ(1,750)\,CAINT
READ(1,760)\,(PACT(K),K=1,50)
A = ENGINE SPEED (REV/MIN) PHI = EQUIVALENCE RATIO
VOEFF = VOLUMETRIC EFF(%) THES = IGNITION TIMING (DEG ATDC)
BORE = CYL BORE (MM) ST = STROKE (MM)
CR = COMP RATIO CRL = CON ROD LENGTH (MM)
SQA = SQUISH AREA 1 (%) SQA = SQUISH AREA 2 (%)
BC = BUMPING CLEARANCE (MM) RS = SPARK PLUG/CYL AXIS, DIST (MM)
FR = PISTON BOWL RADIUS (MM) PH = PLUG PENETRATION (MM)
JTOCC = TYPE OF CHAMBER VOF = VALVE OFFSET (MM)
REX = EX VALVE RECESS RAD (MM) RIN = IN VALVE RECESS RAD (MM)
CEV = EX VALVE/AXIS DIST (MM) CIV = IN VALVE/AXIS DIST (MM)
HV = DEPTH OF VALVE FEC (MM) RSP = RADIUS OF SPARK PLUG RECESS (MM)
RES = RESIDUAL GAS FRACTION UF = SPECIFIC HUMIDITY
PAT = ATMOSPHERIC PRESS (BAR) TAT = ATMOSPHERIC TEMP (DEG K)
TW = CYL WALL TEMP (DEG K)
NI = NO OF ITERATIONS JCM = MAX LOOP COUNT
PACT(K) = CYL PRESS STARTING AT IGN AT CAINT DEG INTERVALS
THECI = CAINT
CEFF=1.0
PI=3.14159
TL=298.15
PDF=0.5
SPARK=360.0-THF
PRESS=RES*100.0
IF(JTOCC,GE,7)VOF=0.0
IF(JCM,LT,20)JCM=20
PMA=WF*100.0
RAT=PRAT/TAT/0.2871/10000.0
RHOAT=RAT*1.0E+04

C CALCULATE CYLINDER HEAD DIMENSIONS
RC=BORE/2.0
ER=CRL/(ST/2.0)
BA=PI*RCSR
CLVOL=BA*ST/(CR-1.0)
SV=RS+VOF
XIN=SQRT(SV**2+CIV**2)
XFX=SQRT(SV**2+CEV**2)
CALL DIM
SI=(SGA+SGB)*100.0/BA
HSA=SAS+SAP+SAH+SAS
HTF=(ST/5.0*N/60.0)**0.333*1.0E-07*4.1868

C CALCULATE MIXTURE COMPOSITION
AFR=15.1/PHI
DO 22 I=1,12
UOF(I)=HOF(I)-2478.9
WMOL(I)=DMOL(I)
CALL RESIDUL(RGE,WF)
UR=WMOL=0.0
DO 60 I=1,12
WRITE(2,790)RNC(I),R~(1)
UR=UR+UOF(I)/WMOL(I)*RM(I)
60 WMOL=WMOL+WMOL(1)*WMGL(1)
RU=8.3143/WHOL
WAI=WOLFF-ST*BA*RAT/100.0
WU=WATK*(1.0+WF+1.0/AFR)/(1.0+RES)
W=UB
WRITE(2,785)VOLFF,WUB,RU,WMOL,UR

C SET INITIAL CONDITIONS AT IGNITION
263 THE=THES
K=1
RH=THE/57.296
XIS=(ST/2.0)**(1.0-COS(PH)+ER-SCRTR(ER*FR-SIN(RH)**2))
X1=XIS
VOL1=BA*X1*CLVOL
P1=PACT(1)
TUB1=P1*VOL1/WT/RU/10000.0
TSS=TUM1
THE=THECi/6.0/N
WRITE(2,785)P1,TUB1,VOL1,QLSS,THE

C CALCULATE PRESSURE AT THE END OF INTERVAL FOR NO COMBUSTION
65 K=K+1
THE=THE+THECI
CALL COMP(TUB1,P1,VOL1,Y1.0,0,0)
IF(P1=PACT(Y)+PDF)0,0,65
K=K

C MEASURED PRESSURE EXCEEDS CALCULATED PRESSURE BY PDF BAR
C START COMBUSTION CALCULATIONS AT THIS CRANK ANGLE

174 UR=UR*CEFF
JFD0=0
DELWB=0.01*WT
THECI=(X-1)*2.0
THE=THES
RH=THE/57.296
X1=XS
VOL1=BA*X1+CLVOL
WUB1=WT
DELTUB=20.0
DELTB=40.0
T81=2200.0
TUB1=TUB8
VU81=VOL1-V81
RF2=3.0
RF4=5.0
SAUB1=HSA+P1*X1*RE
P1=PACT(1)
JH=K
K=K-1
C

C NEXT INTERVAL - CALCULATE VALUES AT END OF INTERVAL
C
523 K=K+1
THE=THE+THECI
NC, ND, JCT1=O
THE=THECI/6.0
W82=WB1+2.0*DI16R-DELBW
IF(WB2.GT.WT)WB2=WT-999*WT
WUB2=WT-WB2
DELBW=WR2-WR1
VR2=VR1+DELB/WR1/VUP1
IF(VR2.GT.VCL1)VR2=VCL1-1.0
VUB2=VOL1-V82
RH=THE/57.296
XR=(ST/2.0)*((1.0-COS(PH)+ER-SQRT(ER*ER-SIN(RH)**2))
VOL4=BA*X4+CLVOL
WRITE(2,785)X4,VOL4,WR2,VR2,VUB2
SAB4=SAB1+DELSAB
SAUB4=HSA+P1*RE+X4-SAB4
SAVUB=(SAUB1+SAUB4)/2.0
PAV=(P1+P4)/2.0
C

C ASSUME FINAL TEMPERATURES
C
C
210 T84=T81+DELTB
TUB4=TUB1+DELTUB
GO TO 211
211 T84=T84
TUB4=TUB4
C

C CALCULATE CHARGE COMPOSITION BASED ON MEAN VALUES FOR INTERVAL
C
C
CALL EBURN(TBAV,PAV)
CPMR1=CPMEA(TL,TUB1,0)
193 \text{IF} (\text{ABS}(VU4-VU84)/VU4, LT. 0.001) \text{GO TO 222}

194 \text{IF}(\text{ABS}((VU4-VU84)/VU4, LT. 0.001) \text{AND} \text{ABS}((VU4-VU84)/VU84, LT. 0.003)) \text{GO TO 222}

C \text{CONTINUE LOOPING UNTIL DIFFERENCE IS ACCEPTABLY SMALL}

C \text{COMPARE CALCULATED PRESSURE WITH MEASURED PRESSURE}

C \text{ADJUST MASS BURNED FOR INTERVAL TO GIVE MORE ACCURATE CALCULATED PRESSURE}

C \text{CALCULATED PRESSURE OK, CHECK INITIAL ESTIMATES AND REPEAT IF NECESSARY}

562 \text{SAVE4=SAVE4}

RF4=FLARAD(WF4,X4,VB4)
SA4=SAA1+SAP1+SAH1+SAS
SAUB4=HSA1+P1*BF0*X4-SAB4
SAVB=(SAR1+SAR4)/2.0
SAUB=(SAUB1+SAUB4)/2.0
JCT1=JCT1+1
\text{IF(JCT1.GT.JCM)GO TO 221}
\text{IF(ABS(ST4-ST46), GT. 20.0, OR, ABS(TB4-TUB4), GT. 10.0) GO TO 210}
\text{IF(ABS(SAB4-SARE4), GT. 0.01*(SAB4+SAUB4)) GO TO 188}
\text{GO TO 223}

221 \text{NLINE=221}
\text{WRITE(2,798)NLHd:

C \text{CONDITIONS AT END OF INTERVAL OK, RESET INITIAL VALUES}

C
CHANGING AT CONSTANT VOLUME

DO 180 I=1,11
WHOLP=WHOLP+PN(I)*WOLC(I)
180 UP4=UP4+UPF(I)*PM(I)/WOLC(I)
UP4=UP4+CEFF
RP=8.3143/WHOLP
RHO=PAV*100/STURAV/RU
WRITE(2,795)RHO,UP4,CVMR1,CVMR2,CVMP1,CVMP2,WHOLP
501 ND=ND+1

COMBUSTION STEP AT INITIAL PISTON POSITION

ENERGY BALANCE FOR BURN/UNBURNED CHARGE AT CONSTANT VOLUMES

TR2=TL+((TR2-1L)*CVMR1/LMOR+(UR-UP4))*DELWB*((UP1-UP4)+(TRB-1L)*CVM1/LMOLP/WB2
TR2=TR2+PB
PB2=UP2*RP/VB2*TP2*16000.0
WRITE(2,790)TR2,PP2

PISTON MOVEMENT AND HEAT TRANSFER STEP

EXPANSION/COMPRESSION OF CHARGE TO ACHIEVE PRESSURE EQUILIBRIUM

IF(VB2.GT.VUB2)GO TO 192
VB4=(PB2/PP)**(1.0/GAMMAP)*VB2*VOL4/VOL1
IF(VB4.LT.0.001)VB4=0.001
IF(VB4.GT.0.9*VOL4)VB4=0.9*VOL4
VUB4=VOL4-VB4
GO TO 138

192 VUB4=VUB2*(PB2/PP)**(1.0/GAMMAP)*VOL4/VOL1
IF(VUB4.LT.1.0)VUB4=1.0
VBR=VOL4-VUB4

CALCULATE HEAT TRANSFER BASED ON MEAN VALUES FOR INTERVAL

1AA HTCB=HTF*SQRT(PAV*TBAV)
HTCB=HTF*SQRT(PAV*TUPAV)
GLOSSB=(TRB-TL)*HTCB*SAB*TTHE
GLOSSUB=(TUB+AV-TU)*HTCB*SBUB*TTHE
DO 194 JV=1,JCM
VB4=VBR
VUB4=VUB2
IF(VUB4.LT.10.0)GLOSSUB=0.0
TR4=TUB2*(VUB2/VB4)**(GAMMAP-1.0)-GLOSSB/VB2/CVM2*WHOLP
TUB4=TUB2*(VUB2/VUB4)**(GAMMAP-1.0)-GLOSSUB/WB2/CVM2*WMOR
IF(TUB4.LT.400.0)TUB4=400.0
PB4=UP2*RP*TUB4/VB4*10000.0
PUB4=WUB4*UW*SUN4/TUB4*10000.0
PUBL4=WUBL4*(TUB4/WUB4*10.0)**2.0*PUBL4*(1.5-WB2/WT)/2.0
IF(VUB4.LT.10.0)GO TO 193
VUB4=VUB4*(PUBL4/PUBL4)**(1.0/GAMMAP)
IF(VUB4.LT.1.0)VUB4=1.0
223  DEF UBO=DEW8  
    DEF W8=W82-W91  
    DEF TB4=TB4-TB1  
    DEF TUB4=TUB4-TUR1  
    DEF SAB=SAR4=SAR1  
    UP1=UP4  
    TUB1=TUB4  
    TR1=TB4  
    VIB1=VUB4  
    VB1=V84  
    P1=P4  
    SAB1=SAB4  
    SAUB1=SAUB4  
    WB1=WB2  
    WIUB1=UW82  

590  J=K  
    FSPEED=(RF4-RF1)/TTHE  
    RF2=FLARAD(RF1,X1,VB2)  
    TFV=(RF2-RF1)/TTHE  
    RF1::RF4  
    X1=X4  
    VOL1;vOL4  

C  **********************************************************************  
C  STORE CALCULATED VALUES IN ARRAY FOR FINAL OUTPUT  
C  **********************************************************************  
    OSA(1,K)=THE  
    OSA(2,K)=VOL4/1000.0  
    OSA(3,K)=P4  
    OSA(4,K)=TB4  
    OSA(5,K)=TUB4  
    OSA(6,K)=V62/WT*100.0  
    OSA(7,K)=VB4/VOL4*10C.0  
    OSA(8,K)=FSPEED/1000.0  
    OSA(9,K)=RF4  
    OSA(10,K)=SAB4  
    OSA(11,K)=P4/RP/TUB4*100.0  
    OSA(12,K)=P4/RP/TE4+100.0  
    OSA(13,K)=CLOSES+QLOSSUB)/TTHE/1000.0  
    OSA(14,K)=RP  
    OSA(15,K)=WMOLP  
    OSA(16,K)=TFV/1000.0  
    DO 554 L=17,27  

554  OSA(L,K)=P4*(L-16)*1.E02  
    WRITE(2,795)P4,TE4,TUB4,VB4,WB2,RF4,FSPEED,THE  
    IF(JEND.EQ.1)GO TO 592  
    THECI=CAINT  
    GO TO 523  

561  WRITE(2,796)  
    GO TO 574  

565  WRITE(2,797)  
    PDF=PDF+0.5  
    IF(WB2.LT.0.1*WT)GO TO 263  

574  RF4=FLARAD(RF2,X4,VB4)  

578  JEND=1  
    GO TO 590  

C  **********************************************************************  
C  END OF CYCLE CALCS,OUTPUT RESULTS IN TABULAR FORM ON LINE PRINTER  
C  **********************************************************************  

592  JCOME=X  
    WRITE(2,800)B,BOPE,SPARK,ST,VOLEFF,CRL  
    WRITE(2,824)PHI,CP,AFR,TW,PRES,ITOC,TAT  
    WRITE(2,836)SQ,PAT,BC,PHA,RS,RHOAT,RIN,REX,XIN,XEX
91X,'EXHAUST VALVE/SPARK PLUG (MM) =',F5.1
850 FORMAT(1HO,'VALVE RECESS DEPTH (MM) =',F5.2///
11X,'SPARK PLUG RECESS RADIUS (MM) =',F5.2///
21X,'CALCULATED MIXTURE PARAMETERS: \( \frac{30}{\%} \)///
31X,'TOTAL MASS OF CHARGE (G) =',F7.4///
41X,'GAS CONSTANT (KJ/KG DEG K) =',F6.3///
51X,'MOLECULAR WEIGHT =',F6.2)
926 FORMAT(1HO,F11.1,8F12.3)
938 FORMAT(1H,'\%PF10.1,5F10.2,4F6F10.2)
957 FORMAT(1H,'CRANK ANGLE',4X,'VOLUME',5X,'PRESSURE',3X,'BURNT TEMP
11,2X,'UNBURNED TEMPERATURE',5X,'BURNT VOLUME',VOL BURNED FLAME SPEED FLAME RAD')
958 FORMAT(1H,'\%',3X,'DEGREES',9X,'\%CC',9X,'BAR',8X,'DEG K',9X,'DEG K',8X
1,\%',11X,\%',10X,'M/S',10X,'MM')
959 FORMAT(1H,'CRANK ANGLE',4X,'MEAS PRESS SURF AREA',4X,'UNB DENS',
12X,'BURNT DENS',3X,'HEAT LOSS',3X,'GAS CONST',3X,'MOL WT',4X,'TURB
2 FL SD')
960 FORMAT(1H,'\%',3X,'DEGREES',9X,'BAR',6X,'MM**2',7X,'KG/M**3',6X,'KG/M**3
1,8X,'KW',8X,'KJ/KG K',19X,'M/S')
961 FORMAT(1H,'\%',3X,'\%',7X,'CO2',8X,'\%',8X,'H2',7X,'H2O',8X,0
12',8X,'H2',8X,\%NO',8X,'\%O2',9X,'O1',9X,'N')
962 FORMAT(1H,'\%',3X,'DEGREES',6X,\%',9X,\%',9X,\%',9X,'\%',8X,'PPM
1,7X,'PPM',7X,'PPM',7X,'PPM',7X,'PPM')
STOP
END
**SUBROUTINE DIM**

**DIM CALCS THE DIMENSIONS OF THE SPECIFIED COMBUSTION CHAMBER**

**COMMON/BLOCK 1/RA,PR,SOA,SQR,BA,JTOCC,PERP,PERPD,RMAM,RMAMD,B,PI**

**COMMON/BLOCK 2/GMIN,C,RFX,RIN,RFX,RIN,ST,RSP,SAY,SAM,SAP,SAS,SAP**

**COMMON/BLOCK 3/SPTHX,SPTHX,SV,CEV,CEV,CR,HEX,HEX,HEX,HEX**

**COMMON/BLOCK 5/VR,RC,RS,RS,PH,SH,UDF(12),TTHE**

**JC0=30**

Cl Vol=(BA-SR)/(CR-1.0)

SPTHE,SPTHE,SPF,RMAM=0.0

IF(JTOCC.EQ.3)SQB=0.0

IF(JTOCC.EQ.7 .OR.JTOCC.EQ.7)SOA=0.0

IF(RSP.LT.(PH+0.5))RSP=PH+0.5

PFR,P=PERPD=500.0

**KD1=1**

IF(JTOCC.EQ.7.AND.JTOCC.LT.10)E=2.0

SPVOL=PI*(RFX**2*HEX**2*HEX**2*HEX**2*(RSP-PH)/6.0*KDI*(3.0*(RSP**2-PH

12)**RSP-PH)**2)

IF(JTOCC.EQ.9)GO TO 20

IF(JTOCC.EQ.9.OR.JTOCC.EQ.6)GO TO 29

SH=(CLVOL-SPVOL-BA*BC)*100.0/(100.0-(SOA+SQB))/BA

JTOCC = 1 FOR ZERO SQUISH, 2 FOR VEE, 3 FOR DOUBLE, 4 FOR SINGLE ON

PLUG SIDE, 5 FOR SINGLE ON VALVE SIDE, 6 FOR BOWL IN PISTON

7 FOR DUAL SPARK/DISC, 8 FOR DUAL SPARK/BOWL IN PISTON

9 FOR DUAL SPARK/HEMISPHERICAL, 10 FOR HEMISPHERICAL

**COMMON/BLOCK 7/SPTHE,SPTHF,SPVOL,PI**

**CTO CALCULATE SQUIZH PLATE ANGLE SPTHE**

19 IF(SQ.F.EQ.0.0)GO TO 24

SPTHE=0.0

**STM=STM**

20 SPTHE=SPTHE

SPTHE=SQ*pi/100.0+SIN(2.0*SPTHE)/2.0

IF(ABS(SPTHE1-SPTHF).LT.0.001)GO TO 24

SPTHE=(SPTHE+SPTHE1)/2.0

GO TO 22

C **CALCULATE COMBUSTION CHAMBER DIMENSIONS**

**C**

24 RMAM,B,C,GMIN=0.0

PERPD=300.0

SPX=RC*COS(SPTHEX)

SPY=RC*SIN(SPTHEX)

IF(JSP.EQ.0)SPTHF=SPTHE

IF(JSP.EQ.0)SPTHE=0.0

IF(JSP.EQ.1)GO TO 26

GO TO (0,25,27,27),JTOCC
C TYPES 3 (SIDE 2) AND 5 (SECTOR TYPE SQUISH ON OPPOSITE SIDE TO SPARK PLUG)

26 PFRP=SPX*RS
RMAM=SQRT(PERP**2+SPY**2).
IF(JTOCC.EQ.5) PERP=PERP
GO TO 28

C TYPE 7 (DOUBL EE VEE SECTOR TYPE SQUISH AREA)

25 PFRP=SPX*RS/2.0*SQRT(2.0)
RMAM=SQRT(PERP**2+(SPY-RS/2.0*SQRT(2.0))**2)
RMAM=ACOS(PERP/RMAM)
IF(RMIN.GT.PI/4.0) GMIN=PI/4.0
VANG=PI/4.0-GMIN
R=RC*2.0*ATAN(SIN(VANG)*PERP/COS(GMIN)/(COS(VANG)*PERP/COS(GMIN)+RS)
C=(SPY-RS/SQRT(2.0))**2.0+RC*2.0*ATAN(SIN(VANG)*PERP/COS(GMIN)/(COS
VANG)*PERP/COS(GMIN)+RS))
GO TO 28

C TYPES 3 (SIDE 1) AND 4 (SECTOR TYPE SQUISH ADJACENT TO SPARK PLUG)

27 PFRP=SPX*RS
RMAM=SQRT(PERP**2+SPY**2)
GO TO 28

C TYPES 9 AND 10 (HEMI-SPHERICAL HEAD)

20 SOA=SQA*BA/100.0
RR=SQRT((RA-SOA)/PI)
IF(JTOCC.EQ.9) RS=RS
WFX, WY=0.0
VOLB=CLVOL-BA*RC
SH=VOLB/2.0/PI/RR**2
GO TO 33
1=1, JCM
32 VOLC=PI*SH/6.0*(3.0*RB**2+SH**2)
IF(SH, GF, RB, AND, VOLB, GT, VOLC) GO TO 34
IF(ABS((VOLC-VOLB)/VOLB), LT, 0.0002) GO TO 35
SH=SH*SQRT(VOLB/VOLC)
33 IF(SH.GE.RC) RR=RR
WRITE(2,690) RR,RB
GO TO (21,28), JTO

910 FORMAT(1H*, OUTSIDE SPECIFIED LIMITS AT LINE 34 ***)
GO TO 35
34 SH=RB*(VOLB-2.0/3.0*PI*RR**3)/(PI*RR**2)
35 JTO=JTOCC-8
RR=RR*2+SH**2)/2.0/SH
IF(SH, GF, RR) RR=RR
WRITE(2,690) RR,RB
GO TO (21,28), JTO

C TYPES 6 AND 8 (BOWL IN PISTON WITH SINGLE OR DUAL IGNITION)

29 SOA=SQA*BA/100.0
RR=SQRT((RA-SOA)/PI)
IF(RR, GT, RB) GO TO 38
SH=((CLVOL-SPVOL-PA*BC)/PI-(RR-RB)**2RR=2.0/3.0*RR**3-PI*(RR-RB)
RR=RR/2.0)/RR/RR+PR
GO TO 39
38 SH=RR*SQRT(RR**2-RR**2)
BC=((CLVOL-SPVOL-PI*SH/6.0*(3.0+RR**2+SH**2))/BA)
WRITE(2,690) BC, RR
690 FORMAT(1H *, F20.4)
39 IF(JTOCC.NE.8) GO TO 28

C TYPES 7 AND 9 (DUAL IGNITION SYMMETRICAL SQUISH DESIGNS)
21 PFRPD=RS
R<BR<
RMAMD=SQR(PERP**2+RC**2)
PFRP=PERPD
SPY,SPTHE=0.0
28 JSP=JSP+1
SB=SQB
IF(JTOCC.EQ.3.AND.JSP.EQ.1)GO TO 19
SB=SB*BA/100.0
WRITE(2,700)RMAMD, RMAM, PERPD, PERP, GMIN, B, C, SPTHE, SPTHX, SH
C
C CALCULATE CYL HEAD SURFACE AREA AT TDC BY CALLING SAVOL WITH RF =100
CALL SAVOL(100.0,0.0)
700 FORMAT(1H,10F11.3)
RETURN
END
SUBROUTINE SAVOL(RF,X)

C *********************************************
C ** SAVOL CALCULATES THE VOLUME AND SURFACE AREAS CORRESPONDING**
C ** TO A GIVEN FLAME RADIUS, PISTON POSITION AND CHAMBER DESIGN **
C *********************************************

DIMENSION Z(99),VF(99),SF(99),FF(99)
COMMON/BLOCK 1/RR,RR,SQA,SQR,RA,JTOCC,PERP,PERPD,PMANN,PMANH,B,PI
COMMON/BLOCK 2/GMIN,C,REX,RIN,XX,E,ST,RSP,SAM,SAMH,SAM,SFD
COMMON/BLOCK 3/STHE,SPTHX,SV,CEV,CIV,CR,HEX,HIN,NI
COMMON/BLOCK 5/VB,K,RS,RSB,BC,PH,SH,UNF(12),TTHE
COMMON/BLOCK6/KDI
IF(RF.LE.0.0)GO TO 300

INITIALISE VALUES

VFEX,VIN,SAM,SAMH,SAMH,SAMH,SAEX,SAIN,FAEX,FAIN,S,VF(1),SSP,RT=0.0
KDI=1
NL=(NI/2)+1
IF(NI.GT.99)NI=99
IF(RF.GT.PF)RT=SQRT(RF**2-PF**2)
IF(JTOCC.GE.7.AND.JTOCC.LT.10)KDI=2
SPY=RC*SIN(SPTHF)
SPYD=SPY*RC*SIN(SPTHX)
SPYTHD=SPYTH+SPTHX

JTOCC = 1 FOR ZERO SQUISH, 2 FOR VFE, 3 FOR DOUBLE, 4 FOR SINGLE ON
PLUG SIDE, 5 FOR SINGLE ON VALVE SIDE, 6 FOR BOWL IN PISTON
7 FOR DUAL SPARK/DISC, 8 FOR DUAL SPARK/BOWL IN PISTON

IF(JTOCC.NE.3)SPYTHD=SPYTH

V=SPYD*2.0
VYD=SPYD*2.0
S=SSQ
H=H+SH+BC
RMN=RC+RS
RMN=RC+RS

IF(RF.LT.1.0)GO TO 5
IF(JTOCC.GE.6)GO TO 40
IF(RF,GT,RMIN.OR.RF.GT,PERP.OR.RF,GT,PERPD)GO TO 40

ISH=500
IF(RSP.LT.1.0)RSP=1.0
HS=H-PH
IF(HS.GT.RF)HS=RF
AS=RF**2-HS**2
VF(NI)=PI*AS*KDI
IF(RF.GT.RS)GO TO 15

VSP=PI/6.0*(HS*(3.0*AS+HS**2+3.0*RF**2)+4.0*RF**3)*KDI
FSP=PI*2.0*RF*(HS+RF)*KDI
GO TO 131
15 ARS=RF+2-PH+2
    APS=RSP+2-PH+2
    HPS=RSP-PH
VSP=PI/6.0*(HS*(3.0*ARS+HPS**2)+PH*(3.0*ARS+PH**2)+3.0*RF;
12) + HPS*(3.0*APS+HPS**2)) * K01
FS=2.0*PI*(RF*(HS+PH)+RSP*HPS) * K01
SSP=PI*(ARS-APS+2.0*RSP*HPS) * K01
GO TO 131
C
C Calculate Axial Interval 'S' and Corresponding Distance from Head!
C
40 IF (H-RF-PH) 0.0, 80
   S=H/(NI-1)
   IF (RF.LT.PH) S=(H-PH+RF)/(NI-1)
   GO TO 90
80 S=(RF+PH)/(NI-1)
   IF (RF.LT.PH) S=2.0*RF/(NI-1)
90 ISH=INT(SH/S)+2
   HH=SH
   IF (JTOCC.EQ.6.0 OR JTOCC.EQ.8) ISH=INT(HH/S)+1
   Z(1)=PH
   IF (RF.LT.PH) Z(1)=-RF+0.00001
   DO 100 I=1,(NI-1)
100 Z(I+1)=Z(I)+S
   IF (Z(NI).GE.(H-PH)) Z(NI)=-H-PH=0.00001
C
C Calculate Length/ARFAS at 'Z' for NI Intervals
C
DO 150 I=1,NI
   V=Z(I)+PH
   FPERP=PERP
   R=SORT(RF**2-Z(I)**2)
   IF (R.LE.0.0) R=0.0001
   RN=RC
   RF=RS
   IF (JTOCC.NE.6.0 AND JTOCC.NE.8) GO TO 220
C
C Bowl in Piston
C
   IF (V.LT.(BC+X)) GO TO 222
   RF=RSB
   IF (RR.GT.RA) GO TO 223
221 RR=RB
   IF (V.LT.(H-RR)) GO TO 222
   YR=SORT(RR**2-(RR-H+V)**2)
   RR=RB-RR+YR
   GO TO 222
223 IF (V.LE.(BC+X)) GO TO 220
   RR=SORT(RR**2-(RR-H+V)**2)
220 IF (JTOCC.LT.9) GO TO 222
C
C Hemi-Spherical Head
C
   IF (V.LE.SH) GO TO 222
   RR=RB
   IF (V.LT.RA) RD=SORT(RR**2-(RR-V)**2)
   RF=RSB
222 RMAX=RD+RE
   RMIN=RD-RE
   IF (RMIN.LT.0.0) GO TO 104
   IF (RF.LE.0.0) GO TO 105
C
C Calculate Angles of Intersection between Flame and Chamber Wall
C

COSA = (RF + 2 * R + 2 - RD - 2) / 2.0 / RF / R
COSB = (RF + 2 * RD + 2 - R * 2) / 2.0 / RE / RD

GO TO 105

104 VF(I) = SF(I) = FF(I) = 0.0
IF (R, LT, (RE-RD)) GO TO 150
CALL VALVE (R, RD, RE, 0.0, 1.0, VF(I), SF(I), FF(I))
GO TO 150

105 IF (R, GT, RMAX) COSA = 1.0
IF (R, GT, RMAX) COSB = -1.0
IF (R, LT, RMIN) COSA = -1.0
IF (R, LT, RMIN) COSB = 1.0

ARAD = ACOS (COSA)
BRAD = ACOS (COSB)

DRA = ARAD
PRA = 0.0

J1 = 1
J2 = 3, J4, J5, J6, J7 = 0.0
IF (R, LE, PERP) GO TO 149
IF (JTOCC, GE, 7, AND, JTOCC, LT, 10) GO TO 141
IF (Z(I), GT, (SH-PH)) GO TO 149
IF (R, GT, PERPD, AND, R, LT, RHAM) GO TO 146
IF (R, GT, PERPD) GO TO 141
IF (R, GT, RHAM) GO TO 147

C

RG T PERP AND LT RMAM

J1 = 0
J5 = 1

GRAD = ACOS (PERP / R)
ORAD = PI - GRAD
DRA = PI - GRAD

IF (JTOCC, NE, 2) GO TO 149

C

TYPE 2

J5 = 1

PRA = PI / 4.0 - GRAD
ORAD = PI / 0.75 - GRAD
DRA = PI - 2.0 * GRAD

IF (PERP, LT, (RC = RE), OR, GRAD, LT, GMIN) GO TO 149
J3 = 0
J4 = 1

DRA = PI - GRAD - GMIN
GO TO 149

C

RG T PERPD AND LT RHAMD

141 IF (JTOCC, EQ, 8, OR, JTOCC, EQ, 9) RHAMD = SQRT (RD * RD + RE * RE)
IF (R, GT, RHAMD) GO TO 142

GRAD = ACOS (PERPD / R)
CRAD = GRAD

IF (JTOCC, EQ, 3) J2 = 1
J5, J6 = 1

PERP = PERPD
IF (JTOCC, GE, 7, AND, JTOCC, LT, 10) FPERP = 0.0

DRA = ARAD - GRAD
GO TO 149

C

RG T RHAMD

142 VF(I) = BA * SQA * SQR
IF (JTOCC, EQ, 8, OR, JTOCC, EQ, 9) VF(I) = PI * RD * R
IF (JTOCC, EQ, 2) VF(I) = BA - 2.0 * SQA
SF(I) = 2.0 * RD * (PI - SPTHD) + SYD
IF (JTOCC, EQ, 2) SF(I) = 2.0 * RD * (PI - 2.0 * SPTHD) + 2.0 * SYD
**FF(I)=0.0**
**GO TO 150**

**C**

**RGY PERPD AND LT RMAM (TYPE 3 ONLY)**

146

**J1=0**

**J5, J6, J7=1**

**GRAD=ACOS(PERP/R)**

**ORAD=PI-GRAD**

**CRAD=ACOS(PERPD/R)**

**DRAD=PI-GRAD-GRAD**

**GO TO 149**

**C**

**RGY RMAM AND LT PERPD**

**IF(JTOCC.EQ.2)J2=2**

147

**J2=1**

**C**

**CALCULATE THE AREA AND LENGTH VALUES FOR THE AXIAL LOCATION 'Z'**

149

**VF(I)=(ARAD*R*R+Rпад*Рпад+2*Е+sin(BRAD)+R)*J1-SQ+J2+(ORAD+K+PEK)**

**1P**

**+(tan(Grad))+(1.0+J3)+(PRAD*R+J3+R+J4)*J5-(R+GRAD-GRAD)*2**

**J6**

**SF(I)=2.0*Р*Р*(RAD*J1-SPTHE*J2)+S*J2+2.0*FF*PP*F*TAN(Grad)**

**J7**

**FF(I)=2.0*DRAD*RF**

**I=C(JTOCC,LT.7.OR.JTucc.EQ.1)GO TO 150**

**VF(I)=VF(I)+2.0**

**SF(I)=SF(I)+2.0**

**FF(I)=FF(I)+2.0**

150 **CONTINUE**

**C**

**CALCULATE SUM OF EVEN/UNEVEN ORDINATES FOR SIMPSONS METHOD**

**SUME,SUMR,SUMSE,SUMSR,SUMEF,SUMRF=0.0**

**DO 120 I=2,(NI-1),2**

**SUMEF=SUMEF+FF(I)**

**SUMSE=SUMSE+SF(I)**

120 **DO 130 I=3,(NI-2),2**

**SUMRF=SUMRF+FF(I)**

**SUMSR=SUMSR+SF(I)**

130 **SUM=SUM+VF(I)**

**C**

**CALCULATE VOLUME AND SURFACE AREAS FOR SPARK PLUG/VALVE RECESES**

**RTS=RF**

**IF(RF,GT,RE)RTS=RSP**

**HPS=RTS-PH**

**IF(HPS,LT,0.0)HPS=0.0**

**APS=RTS*2-PH**

**VSP=PI*6.0*HPS*(3.0*APS+HPS+2)**

**FSP=PI*2.0*RTS*HPS*KDI**

**SPF=PI**

**IF(RF,LT,RSP)GO TO 131**

**FSP=0.0**

**SDF=PI*(HPS+RSP-2.0-APS)**

**C**

**CALCULATE INLET AND EXHAUST VALVE RECESES VOLUMES/SURFACE AREAS**

131 **CALL VALVE(RT,REX,XEY,CEY,HEX,VEY,SAFX,FAFX)**

**C**

**INTEGRATE USING SIMPSONS METHOD OF APPROXIMATE INTEGRATION**
**C**

```
140 VR=S/3.0*(VF(1)+VF(NI)+4.0*SUMF+7.0*SUMR)+VSP+VIN+VEX
SAW=S/3.0*(SF(1)+SF(NI)+4.0*SUMSF+2.0*SUMSR)
SAF=S/3.0*(FF(1)+FF(NI)+4.0*SUMEF+2.0*SUMRF)+FAIN+FAEX+FSP
SAP=VF(NI)
GO TO (300,0,0,0,229,300,220)JTOCC
IF(ISH,GT,NI)GO TO 300
SAS=VF(ISH)-VF(ISH-1)*2.0+VF(ISH-2)
GO TO 230
229 IF(ISH,GT,(NI-1))GO TO 300
SAS=VF(ISH)-VF(ISH+1)
230 IF(SAS,LT,0.0)SAS=0.0
300 RETURN
END
```
SUBROUTINE VALVE(RF, R, X, C, H, VBV, SAV, FAV)

** SUBROUTINE VALVE CALCULATES THE BURNED VOLUME AND SURFACE AREAS **
** OF A VALVE RECESS FOR A GIVEN FLAME RADIUS AND VALVE DIMENSION **

COMMON/BLOCK 3/SPTHE, SPTHX, CV, CSV, CIV, CR, HEX, HIN, NI
COMMON/BLOCK 6/KDI
PI=3.142
VBV, SAV, FAV=0.0
IF(H.LE.0.0) RETURN
IF(RF.LT.(X-R)) RETURN
IF(RF.GT.(X+R)) GO TO 10

CALCULATE VALVE RECESS VOLUME AREAS FOR SINGLE IGNITION

w=(RF**2+X**2-RF**2)/2.0
2=X-w
Y=SQRT(RF**2-7**2)
TH=ASIN(Y/RF)
BF=ACOS(W/R)
VBV=(RF*2*TH-Y*7+R*7-W*Y)*H*KDI
SAV=2.0*BE*RF*H*KDI
FAV=2.0*BF*RF*H*KDI
IF(KDI.EQ.1) RETURN
RMIN=SQRT(SV**2*(C-R)**2)
RMAX=SQRT(SV**2*(C+R)**2)
IF(RF.LE.RMIN) RETURN
IF(RF.GE.RMAX) GO TO 10

CALCULATE VALVE RECESS VOLUME AREAS FOR DUAL IGNITION

TH=(TH*ACOS(SV/X)-ACOS(SV/RF))/2.0
BE=(BE+ACOS(C/X))/2.0
P=C-SQRT(RF**2-SV**2)
VBV=(RF*2*(TH-SIN(TH)*COS(TH)))*R*2*BE-R*P*COS(2.0*BE-PI/2.0)/2.0
FAV=4.0*BE*RF*H
IF(RF.GE.RMIN) RETURN
IF(RF.LE.RMAX) RETURN
VAV=PI*R*2*H
SAV=2.0*PI*R*H
FAV=0.0
RETURN
END
FUNCTION FLARAD(FR,X,VF)

** FLARAD CALCULATES THE FLAME RADIUS CORRESPONDING TO A
** GIVEN BURNT VOLUME, PISTON POSITION AND CHAMBER DESIGN
**

COMMON/BLOCK 1/RR,PP,SR,SC,RC,RCE,TOCC,PERP,PPERD,PMAM,PMADD,B,DI
COMMON/BLOCK 5/VB,RC,RS,RSB,RC,PH,SH,UNF(12),TTHF
RF=FR
VF=VF1
RFM=SGRT((RS+RC)**2+(SH+BC+X-PH)**2)
IF(JTOCC.EQ.7.OR.JTOCC.EQ.9)RFM=SGRT(SGRT(RS**2+RC**2)+(SH+BC+X-PH)**2)
IF(JTOCC.EQ.8)GO TO 6
RFK=SGRT(SGRT(RS+RC)**2+(SH+X+PH)**2)
GO TO 4
6 RFM=SGRT(SGRT(RS**2+RC**2)+(BC+X-PH)**2)
RFK=SGRT(SGRT(RSB+RNB)**2+(SH+BC-PH)**2)
IF(RFK.GT.RFM)RFM=RFK

** INITIAL RADIUS INCREMENT OF 1.0 MM
**

3 RI=1.0
J=0
CALL SAVOL(RF,X)
5 J=J+1
IF(VF-VF1).GT.50,20

** VOLUME FOR ESTIMATED FLAME RADIUS TOO SMALL, INCREASE RAD BY INCREMENT
**

10 RF=RF+RI
IF(RF.GT.RFM)GO TO 40
VF=VF1
CALL SAVOL(RF,X)
IF(VF-VF1).LT.50,0

** RADIUS WITHIN INCREMENT, LINEAR INTERP TO OBTAIN IMPROVED VALUE
**

15 IF(J.GT.1)GO TO 50
CALL SAVOL(RF,X)

** REDUCE INCREMENT TO 0.1 MM AND REPEAT PROCESS
**

30 RI=.1
GO TO 5

** VOLUME FOR ESTIMATED FLAME RADIUS TOO LARGE, DECREASE RAD BY INCREMENT
**

20 RF=RF-RI
VF=VF1
IF(RF.LE.0.0)GO TO 30
CALL SAVOL(RF,X)
GO TO 35

30 RI=RF+RI
VB=0.0
35 IF (VB-VF) 0.50, 20
C ********** RADIUS WITHIN INCREMENT, LINEAR INTERP TO OBTAIN IMPROVED VALUE
C ********** RF = RF - ((VF-VBF)/(VB-VBF)-1.0)*RI
G0 TO 15
40 IF (J.GT.1) GO TO 45
RF = RF - 1.0
RI = 0.1
G0 TO 5
45 WRITE(2, 900)
RF = RFM
50 F I A R A D = RF
900 FORMAT (1H , '*** COMPUTED FLAME RADIUS EXCEEDS MAX RADIUS ***')
RETURN
END
**SUBROUTINE RESIDUL(RES,WF)**

**RESIDUL CALCULATES THE COMPOSITION OF THE CHARGE PRIOR TO IGN**

**COMMON/BLOCK 4/PN(11),PM(11),RN(12),WMOL(12),Q(4),PHI,JCH**

<table>
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<th>2</th>
<th>3</th>
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<td>OH</td>
<td>H</td>
<td>O</td>
<td>N</td>
<td>O2</td>
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**AFR=15.1/PHI**

**WUB=1.0**

**MASS FRACTIONS FOR ONE UNIT OF TOTAL CYLINDER MASS**

| WAIR=WUB*(1.0-RES)/(1.0+WF+(1.0/AFR)) |
| WACT=WAIR/AFR |
| WMOST=WAIR+WF |
| WOX=WAIR*0.233 |
| WNTT=WAIR*0.767 |

**CO2 Rn(3)=12.5/PHI**

**CONSIDER ONE MOL OF FUEL WITH N2, N2 AND MOISTURE ONLY**

| RN(1), RN(2), RN(6), RN(7), RN(8), RN(9), RN(10), RN(11) = 0.0 |
| RN(3) = 12.5 + 79.00/21.0/PHI |
| RN(4) = 12.5*WF/0.233 + 32.0/18.0/PHI |
| RN(5) = 12.5/PHI |
| RN(12) = 0.0 |

**CALCULATE MOLAR FRACTIONS FOR ZERO DISSOCIATION... I.E Q(1) TO Q(4)**

**IF(PHI=1.0)30,0,0**

| Q(1) = RN(1) + 2*RN(5) - RN(6) - RN(7) + (RN(8) - RN(9))/2.0 - RN(10) - 17*RN(12) |
| Q(2) = RN(2) - 2*RN(5) + RN(6) - RN(7) - (RN(8) - RN(9))/2.0 - RN(10) + 25*RN(12) |
| Q(3) = RN(3) + RN(7)/2.0 + RN(11)/2.0 |
| Q(4) = RN(4) + RN(6) + RN(8)/2.0 + RN(9)/2.0 + 9*RN(12) |

**IF((O(2),LT,.0,2.0,.AND,.J,.EQ.,.1),0(2),=0.2**

**SWAP SPECIE (2) AND SPECIE (5) FOR THIS CALCULATION**

**IF(Q(1),.LT,.0,.0,.AND,.J,.EQ.,.1),O(1),=0.2**

**GO TO 40**

**IF(Q(1),.LT,.0,.0,.AND,.J,.EQ.,.1),O(1),=0.2**

**DO 40 1=1,4**

**IF(Q(1),.LT,.0,.0,.AND,.J,.EQ.,.1),O(1),=0.2**

**DO 45 1=1,4**

**DO 30 I=1,4**

**45 O(1)=O(1)/QT**

**END**
50  P(N(I)) = Q(I)
    P(N(5)) = P(N(7))
    CALL E-BURN(1500.0, 1.0)
    DO 55 I = 1, 11
      55  R(M(I)) = P(M(I))

C  Mass/Molar fractions for reactants
C

*** MASS/MOLAR FRACTIONS FOR REACTANTS ***

DO 65 I = 1, 11
   GO TO (0, 0, 65, 65, 65), I
   R(M(I)) = R(M(I)) * RES
65  CONTINUE
   R(M(3)) = R(M(3)) * RES + W(HIT
   R(M(4)) = R(M(4)) * RES + W(HOIST
   R(M(5)) = R(M(5)) * RES + W(OXY
   R(M(12)) = W(OCT
   RNT = 0.0
   DO 58 I = 1, 12
      R(N(I)) = R(M(I)) / W(MOL(I))
58  RNT = RNT * R(N(I))
   DO 59 I = 1, 12
59  R(N(I)) = R(N(I)) / RNT
52  RETURN
   END
SUBROUTINE COMP(T1, P1, VOL1, X1, UP1, JCEX)

**COMP CALCULATES THE STATE PROPERTIES AT END OF COMP/EXP PROCESS**

```
COMMON/BLOCK 1/RR, SQA, SOR, BA, ITOCC, PERP, PERPD, RMAM, RMAMD, B, PI
COMMON/BLOCK 2/GMIN, C, RFX, RIN, XEX, XIN, ST, RSP, SAH, SAM, SAP, SAS, SAF
COMMON/BLOCK 3/SPTHE, SPTHX, SV, CEV, CIV, CR, HEX, HIN, NI
COMMON/BLOCK 4/PN(11), PM(11), RM(12), WMOL(12), Q(4), PH1, JCM
COMMON/BLOCK 5/VB, RC, RS, RSB, AC, PH, SH, UOF(12), TTHE
COMMON/BLOCK 8/CLVOL, HSA, HTF, THE, QLOSS, TW, WT, WMOLR, ER, RU, CEFF
COMMON/BLOCK 9/PAT, GAMMAR, WDI

JCEX LE 1 FOR COMPRESSION, GT 1 FOR EXPANSION

If(JCEX.LE.1)JL=0
Rn=THE/S7.296
X=ST/2.0*(1.0-COS(RH)+ER-SQRT(P1**2-SIN(PH)**2))
VOL4=BA*X4+CLVOL
SA=HSA+PI*RC*X4
P4=P1*(VOL1/VOL4)**1.25
T4=P1*(VOL4/VOL1)**1.0000
DO 40 JL=1,JCM
20 TF4=T4
PAV=(P1+P4)/2.0
TAV=(T1+T4)/2.0
HTC=HTF*SQRT(PAV*TAV)
QLOSS=HTC*SA*(TAV-TW)*TTHE
If(JCEX.LE.1)Go To 30
If(JL.EQ.1)Go To 30
If(TAV.LT.1600.0)JL=1
CALL EDURN(TAV,PAV)
UP4=UP4+UOF(1)/WMOL
UP4=UP4*CEFF
P4=WT*RP*T4/VOL4*10000.0
Go To 40
```

30 CONTINUE(T4, T1, JCEX)
CNM=CPMEAN(T4, T1, JCEX)
CPM=CPM-8.3143
GAMMAR=CPM/CVM
If(JCEX.GT.1)Go To 38
T4=T1*(VOL1/VOL4)**(GAMMAR-1.0)-QLOSS/TW/CVM*WMOLR
P4=WT*RP*T4/VOL4*10000.0
Go To 40
38 T4=T1*(VOL1/VOL4)**(GAMMAR-1.0)-(QLOSS+(UP4-UP1)*WT)/WT/CVM*WMOLR
P4=WT*RP*T4/VOL4*10000.0
Go To 40
40 If(ABS(T4-TE4).LT.0.2)Go To 50
WRITE(2, 810)
```
50 WRITE(2, 800)P4, T4, VOL4, QLOSS, THE
WDI=(P4*VOL4-P1*VOL1)/(1.0-ALOG(P4/P1)/ALOG(VOL1/VOL4))
T1=T4
P1=P4
X1=X4
VOL1=VOL4
UP1=UP4
```
800 FORMAT(1H,5F15.3)
810 FORMAT(1H,'*** COMP OUTSIDE SPECIFIED RANGE ***')
RETURN
END
SUBROUTINE EBURN(T,PB)
C
** FOR THE GIVEN TEMPERATURE, PRESSURE AND MIXTURE STRENGTH **
C ** EBURN CALCULATES THE EQUILIBRIUM COMPOSITION FOR THE PRODUCTS **
C
REAL LX
DIMENSION LX(12), PE(12), EK(12)
COMMON/BLOCK 4/PN(11), PM(11), RN(12), RM(12), WMOL(12), Q(4), PHI, JCM
JCOUNT=0
P=P8/1.01325
JCT=JCM+10
TB=T
IF(T0.LT.1600.0)TB=1600.0
IF(PN(2).LT.0.001)PN(2)=0.001
PNT=PN(1)+PN(2)+PN(3)+PN(4)
DO 22 I=1,4
22 PF(I)=PN(I)
IF(PHI.GE.1.0)GO TO 79
C
EQUILIBRIUM CONSTANTS FOR WEAK MIXTURES
C
LY(5)=4.389-1452/TB
LY(6)=3.06-13170/TB
LY(7)=0.6615-4729/TB
LY(8)=2.2465-8666/TB
LY(9)=4.66-18413/TB
LY(10)=3.4855-13934/TB
LY(11)=3.484-19065/TB
DO 60 I=5,11
60 EK(I)=10.0*LY(I)
DO 65 I=1,4
65 PN(I)=PE(I)*0.9+PN(I)*0.1
66 PPT=P/PNT
C
CALCULATE FRACTION OF 7 DERIVED CONSTITUENTS
C
PN(5)=EK(5)*PN(1)/SQRT(PPT*PN(2))
PN(6)=EK(6)*PN(4)/SQRT(PPT*PN(2))
PN(7)=EK(7)*SQRT(PN(2)*PN(3))
PN(8)=EK(8)*PPT**0.25*PN(2)**0.25*PN(4)**0.5
PN(9)=EK(9)*PPT**0.75*PN(4)**0.5*PN(2)**0.25
PN(10)=EK(10)*SQRT(PN(2)/PPT)
PN(11)=EK(11)*SQRT(PN(5)/PPT)
C
CALCULATE FRACTION OF 4 INDEPENDENT COMPONENTS
C
PN(1)=Q(1)-PN(5)
PN(2)=Q(2)*PN(5)+PN(6)-PN(7)-PN(10))/2.0+(PN(9)-PN(8))/4.0
PN(3)=Q(3)-(PN(7)+PN(11))/2.0
PN(4)=Q(4)-PN(4)-(PN(8)+PN(9))/2.0
DO 67 I=1,4
67 IF(PN(I), LT, 0.001)PN(I)=0.001
PNT=PN(1)+PN(2)+PN(3)+PN(4)+PN(5)+PN(6)+PN(7)+PN(8)+PN(9)+PN(10)+
1N(11)
**EQUILIBRIUM CONSTANTS FOR RICH MIXTURES**

79  
L(N) = 8.778 - 29.055 / T8  
L(K) = ALOG10 (EXP (21.77 - 3.15X + ALOG(T8)) + 0.102)  
L(K) = 5.05 - 19.256 / T8  
L(K) = 4.441 - 15.930 / T8  
L(K) = 2.166 - 10.799 / T8  
L(K) = 7.875 - 27.761 / T8  
L(K) = 3.486 - 10.065 / T8  
D0 80 I = 1, 11  

80  
E(K) = 10.0 * L(K)  
G0 TO 110  

82  
JCOUNT = JCOUNT + 1  
D0 85 I = 1, 4  

85  
P(I) = PE(I) * 0.9 + PN(I) * 0.1  
R8  
PF(I) = P(I)  
R8  
PPNT = P / PNT  
C

**CALCULATE FRACTION OF 7 DERIVED CONSTITUENTS**

76  
P(5) = E(K) / PPNT + PN(1) / 2 / PN(2) / 2  
P(6) = E(K) / PN(1) + PN(2) / PN(4)  
P(7) = E(K) / SORT(PPNT) * PN(1) / PN(2) / SORT(PN(3))  
P(8) = E(K) / SORT(PN(1) / PN(4) / PN(2))  
P(9) = E(K) / SORT(PN(2) / PN(4) / PN(4) / PN(1))  
P(10) = E(K) / PPNT * PN(1) / PN(2)  
P(11) = E(K) / SORT(PN(3) / PPNT)  
C

**CALCULATE FRACTION OF 4 INDEPENDENT COMPONENTS**

87  
P(1) = (1) = 2 * PN(5) + PN(4) - PN(7) - PN(10) + PN(9) - PN(8) / 2, 0  
P(2) = (2) = 2 * PN(5) + PN(4) + PN(7) + PN(10) + PN(8) - PN(9) / 2, 0  
P(3) = (3) = (PN(7) + PN(8)) / 2, n  
P(4) = (4) = (PN(8) + PN(9)) / 2, 0 - PN(6)  
D0 90 I = 1, 4  

90  
IF (PN(I) < 0.001) PN(I) = 0.001  
PNT = PN(1) + PN(2) + PN(3) + PN(4) + PN(5) + PN(6) + PN(7) + PN(8) + PN(9) + PN(10) + 1N(11)  
D0 88 I = 1, 11  

88  
P(I) = PN(I) / PNT  
IF (JCOUNT, EQ, JCT) GO TO 110  
D0 90 I = 1, 4  

90  
IF (ABS(P(E(I) - PN(I)) < 0.005) GO TO 82  
JCOUNT, EQ, JCT WRITE (2, 90A)  
PMT = 0, 0  
D0 120 I = 1, 11  
P(I) = PN(I) - WML(1)  
D0 120 I = 1, 11  

120  
P(I) = PN(I) / PNT  
D0 130 I = 1, 11  

130  
RETURN

END
FUNCTION CPMEAN(TL, TU, JJ)

** CPMEAN CALCULATES THE MEAN ISOBARIC SPECIFIC HEAT FOR **
** MIXTURE FOR GIVEN TEMPERATURE RANGE AND COMPOSITION **

** DIMENSION C(12,2), AN(12), CI0(12), CDAT(7,12,2) **
COMMON/BLOCK 4/PN(11), PM(11), RN(12), RM(12), WMOL(12), Q(4), PHI, JCM
COMMON/BLOCK10/ADAT(7,12), BDAT(7,12), CIDAT(12) 

** J I , LT,1 FOR REACTANTS , GT,1 FOR PRODUCTS **

CP=0.0
DO 5 J=1,12
  IF(J,JG,LT.1,AND.J,FG,12)GO TO 90
  A=0.001*TL
  DO 6 L=1,2
  M=1
  IF(A,GT,2,0)M=2
  C(J,L)=0.0
  DO 7 K=1,7
  IF(M.EQ,1)CDAT(K,J,L)=ADAT(K,J)
  IF(M.EQ.2)CDAT(K,J,L)=BDAT(K,J)
  7 C(J,L)=CDAT(K,J,L)+A*K/K+C(J,1)
  IF(J,GT.1)AN(J)=PN(J)
  IF(J,JG,1,AND.J,FQ,'2)GO TO 140
  A=0.001*TU
  IF(J,JG,GT.9)AN(J)=PN(J)
  CPJ=CPJ+CP
  5 CONTINUE

** MEAN CP VALUES (JOULES/GRAM MOL DEG K) OVER TEMP RANGE TU TO TL **

90 CPMEAN=CP*4,1BAR/(TU-TL)*1000
RETURN
END

** INDIVIDUAL CP POLYNOMIALS FOR TEMPERATURES BELOW 2000 DEG K **

A 4.324933, 20.80895,-22.94591, 16.8448,-7.95567, 2.12167,-0.24087
B 7.812249,-6.668293, 17.78296,-17.2871, 8.86013,-2.31482, 0.24478
C 7.709929,-5.508677, 13.12136,-11.6796, 5.23040,-1.17317, 0.10388
D 7.988860,-1.506271, 6.661376,-4.65597, 1.69646,-0.37062, 0.03992
E 7.361141,-5.369589, 20.54179,-25.8653, 15.9457,-4.85889, 0.58615
F 6.185043, 4.710457,-10.92136, 17.5409,-7.16276, 1.92340,-0.20841
G R,462334,-10.40669, 27.54506,-30.2812, 17.1851,-4.95725, 0.57553
H 7.615100,-1.936000, 0.977000, 2.61530,-2.69090, 0.17789,-0.12695
I 4.968800, 0.000000, 0.000000, 0.000000, 0.000000, 0.000000, 0.000000
J 5.976134,-4.241883, 7.931254,-7.94423, 4.40336,-1.27134, 0.14914
K 4.966526, 0.011505,-0.033335, 0.04617,-0.03246,-0.01095, 0.01337
L 2.547740, 153.6559,-13.38270,-93.7770, 70.9791,-16.4959, 0.00000

** INDIVIDUAL CP POLYNOMIALS FOR TEMPERATURES ABOVE 2000 DEG K **

C
**DATA DDAT/**
M 8.153021, 8.411419, -4.795179, 1.54313, -0.28312, 0.02766, -0.00111
N 5.966461, 3.288911, -1.660467, 0.47645, -0.07854, 0.00696, -0.00026
O 5.649167, 3.579035, -1.794312, 0.51213, -0.08419, 0.00744, -0.00027
P 3.401967, 4.633046, 0.067415, 1.04985, -0.16202, 0.01377, -0.00049
Q 8.439106, -0.376523, 0.621716, -0.19235, 0.02900, -0.00236, 0.00009
R 4.103273, 3.981784, -1.426509, 0.26927, -0.01866, -0.00081, 0.00012
S 6.590193, 2.604241, 1.291210, 0.36511, -0.05937, 0.00519, -0.00019
T 4.946400, 3.264500, -1.202690, 0.25849, -0.03184, 0.00207, -0.00005
U 4.968000, 0.000000, 0.000000, 0.00000, 0.00000, 0.00000, 0.00000
V 4.743426, 0.480906, -0.364666, 0.12739, -0.02117, 0.00171, -0.00005
W 4.845706, 0.080796, 0.087980, -0.10425, 0.03658, -0.00487, 0.00023
X 0.000000, 0.000000, 0.000000, 0.00000, 0.00000, 0.00000, 0.00000

**DATA CIDAT/**
Y 1.080780, -0.161340, 0.311350, -1.61502, 0.93325, -1.31135, -0.0030
Z 0.979550, 0.000000, -0.317400, 0.15448, 0.00000

**C**

**CONSTANT OF INTEGRATION DIFFERENCES BETWEEN UPPER AND LOWER EQUATION**

**DATA CIDAT/**
Y 1.080780, -0.161340, 0.311350, -1.61502, 0.93325, -1.31135, -0.0030
Z 0.979550, 0.000000, -0.317400, 0.15448, 0.00000

**FINISH**
**MASTER PRESS**

**PRESS CALCULATES THE CYLINDER PRESSURE AND**

**BURNT MASS FRACTION FROM EXPERIMENTAL DATA**

**INTEGRAL**

REAL LPL, LPSL, LPVU, LPU, IPU, IPUU, LVSL, LVSU, LVU, LVL, JMAX, NTHE 
DIMENSION LVU(90), LVL(90), LPVU(90), LPV(90), PRMAX(100), JMAX(100) 
DIMENSION VCL(256), PVG(256), WR(256), WRR(256), PRR(256) 
DIMENSION PFUN(256), IDAT(256), PRES(256), NTHE(256), RH(256), X(256) 
PI=3.14159 
READ(1,102) BORE, ST, CR, CRL, TRCAL, PTCAL 

**CALCULATE CONSTANT PARAMETERS FOR ENGINE**

ER=CRL/ST**2.0 
BA=PI*BORE**2/4.0 
READ(2,100) E, BC 
E=E/64 
CONST=1.0*BCE10.0**E/32767.0**2 
NN=0 
CIVOL=BA*ST/(CR-1.0) 
DO 13 J=1,256 
NTHE(J)=200+J*2 
RH(J)=NTH(E(J))/57.296 
X(J)=(ST/2.0)*(1.0-COS(RH(J)))+ER-SQRT(ER*ER-SIN(RH(J))**2)) 
VOL(J)=RA*X(J)+CIVOL 
13 VOL(J)=VOL(J)/1000.0 

**CALCULATE VOLUME VALUES FOR LEAST SQUARES CALCULATIONS**

SLVL,LVSL=0.0 
DO 6 J=25,34 
LVU(J)=ALOG(VOL(J)) 
LVSL=LVU(J)**2+LVSL 
6 SLVL=SLVL+LVU(J) 

**START CALCULATIONS FOR NEXT ENGINE CYCLE**

900 NN=NN+1 
DO 15 J=1,256 
15 PFUN(J)=0.0 
WRITE(4,120) NN 
READ(9,118) JS 
JS=(160-JS)/2 
SLVU,LVSU=0.0 
JSB=JS-10 
JST=JS+1 
DO 42 J=JSB,JST 
LVU(J)=ALOG(VOL(J)) 
LVSU=LVU(J)**2+LVSU 
42 SLVU=SLVU+LVU(J) 

**READ PRESSURE DATA FROM MAGNETIC TAPE FILE**

**CALCULATE MAX AND MEAN RELATIVE PRESSURES AND MAX PRESSURE ANGLE**

DO 25 K=1,99 
READ(2,101) (IDAT(J), J=1,256) 
DO 30 J=1,256 
30 PFUN(J)=PFUN(J)+IDAT(J)
M = JS + 15
DO 22 J = M, 120
    IF (IDAT(J) - IDAT(J - 1)) 24, 0, 0
22 CONTINUE
24 PRMAX(K) = IDAT(J - 1)
    JTMAX(K) = 200 + 2*(J - 1)
25 CONTINUE
DO 35 J = 1, 256
35 PRMAX(J) = PFUN(J) * CONST * PTCAL * TRLCAL / 99.0
    PERM = (5.0 - PFUN(250) - PFUN(251) - PFUN(252) - PFUN(253) - PFUN(254)) / 5.0
    JCN = 0
    PERD = 0.08

C ABSOLUTE PRESSURE CORRECTION CALCULATIONS

36 JCN = JCN + 1
    DO 40 J = 1, 256
40 PRES(J) = PFUN(J) + PRMAX(J) + LPSL, SLPL, LPLV = 0.0
    DO 43 J = 25, 34
        LPL(J) = ALOG(PRES(J))
        LPSL = LPL(J)**2 + LPSL
        LPLV = LPL(J) * LVL(J) + LPLV
        SLPL = LPL(J) + SLPL

C LEAST SQUARES METHOD FOR CALCULATING COMPRESSION INDEX

C CIL = (LPLV - SLPL) * SLVL / 10.0) / (LVSL - SLVL)**2 / 10.0)
    IF (CIL.GT.1.38) GO TO 481
    IF (CIL.LT.1.24) GO TO 510
    LPSU, SLPU, LPVU = 0.0
    DO 44 J = JSB, JST
        LPU(J) = ALOG(PRES(J))
        LPSU = LPU(J)**2 + LPSU
        LPVU = LPU(J) * LVL(J) + LPVU
        SLPU = LPU(J) + SLPU

44 SLPU = LPU(J) + SLPU
    CIU = (LPLV - SLPU) * SLVU / 10.0) / (LVSL - SLVU)**2 / 10.0)
    IF (JCN.GT.20) GO TO 484
    IF (CIU.GT.1.38) GO TO 481
    IF (CIU.LT.1.24) GO TO 510
    IF (ABS(CIL - CIU).LT.0.005) GO TO 484
    IF (CIL - CIU) 0, 484, 481
510 CONTINUE

C DECREASE PRESSURE

C IF (JCN.EQ.1.0 OR JCM.EQ.1) GO TO 511
    PERD = PERD/2.0
511 PRFR = PFR + PERD
    JCM = 1
    GO TO 36

C INCREASE PRESSURE

481 IF (JCN.EQ.1.0 OR JCM.EQ.0) GO TO 512
    PERD = PERD/2.0
512 PRFR = PFR + PERD
    JCM = 0
    GO TO 36
484 CONTINUE
DO 37 K=1,100
37 PRMAX(K)=PRMAX(K)*CONST*PTCAL+TRCAL+PERR

C CALCULATE PERCENTAGE PRESSURE STANDARD DEVIATION

PTOT=0.0
DO 72 K=1,99
72 PTOT=PTOT+PRMAX(K)
PAV=PTOT/99
SSQP=0.0
DO 73 K=1,99
73 SSQP=SSQP+(PRMAX(K)-PAV)**2
SP=SQRT(SSQP/98)/PAV*100

C CALCULATE PERCENTAGE MASS OF CHARGE BURNT

JT=JS+15
PVG(JT-1)=0.0
DO 54 J=JT,200
54 IF(PVG(J).LT.PVGeJ-1)GO TO 56
PVG(J)=PRES(J)*VOL(J)**(CIU-0.05)
PVM=PRES(J)*VOL(J)**CIU
WRITE(4,200)CIU
WRITE(4,205)
WRITE(4,206)
WR(4)=0.0
DO 60 J=5,200
60 PVG(J)=PRES(J)*VOL(J)**CIU
WR(J)=(PVG(J)-PVG(J-1))/PVM-PVG(J)*100.0
PRR(J)=(RES(J)-RES(J-1))/2.0
WRITE(4,214)PAV
IF(NN.LT.8)GO TO 900
100 FORMAT(10X,10X,10F10.0)
101 FORMAT(10X,10F12.3)
102 FORMAT(10X,10F12.0)
110 FORMAT(1H,10F12.3)
111 FORMAT(1H,10F12.0)
118 FORMAT(1H)
120 FORMAT(1H1,20X,'ENGINE RUN NUMBER = ',I7)
140 FORMAT(1H0,30X,'MAX PRESSURE FOR 99 CONSECUTIVE CYCLES (BAR)'
145 FORMAT(1H0,30X,'MAX PRESS ANGLE FOR 99 CONSECUTIVE CYCLES (DEG)'
200 FORMAT(1H0,5X,'COMPRESSPION INDEX = ',F6.3)
205 FORMAT(1H0,3X,'CRANK ANGLE',5X,'VOLUME',9X,'PRESSURE',3X,'PRESS R
1E RATE',4X,'MASS BURNT',2X,'MASS BURN RATE',9X,'DEG',9X,'PERCE
11.5X,'PERCENT/DEG//)
206 FORMAT(1H,5X,'DEGREE',10X,'CC',13X,'BAR',9X,'BAR/DEG',9X,'PERCE
11.5X,'PERCENT/DEG//)
210 FORMAT(1H,1F12.0,6F15.2)
212 FORMAT(1H0,'PERCENTAGE STANDARD DEVIATION = ',F7.3)
214 FORMAT(1H,'MEAN MAX PRESSURE = ',F7.2)
STOP
END
FINISH
****
### COMPUTED SIMULATION OF COMBUSTION IN A SPARK IGNITION ENGINE

<table>
<thead>
<tr>
<th>TEST CONDITIONS ARE</th>
<th>ENGINE DETAILS ARE</th>
</tr>
</thead>
<tbody>
<tr>
<td>ENGINE SPEED (RPM) = 2000.0</td>
<td>NUM (MM) = 10.0</td>
</tr>
<tr>
<td>IGNITION TIMING (DEG BTDC) = 30.0</td>
<td>SIMHAT (MM) = 10.0</td>
</tr>
<tr>
<td>VOLUMETRIC EFFICIENCY (%) = 75.0</td>
<td>CUM MUD LENGTH (MM) = 120.0</td>
</tr>
<tr>
<td>LENS-LALLINE RATU = 1.10</td>
<td>COMMISSION RATIO = 0.0</td>
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