Flame acceleration study using obstacles and its application in spark-ignition engines

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FLAME ACCELERATION STUDY USING OBSTACLES
AND ITS
APPLICATION IN SPARK-IGNITION ENGINES

by

VENKATESAN THYAGARAJAN

A Doctoral Thesis
submitted in partial fulfilment of the
requirements for the award of
Doctor of Philosophy
of Loughborough University of Technology

October 1988

Supervisor: Professor J C Dent, PhD, CEng
Department of Mechanical Engineering

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To
Ludwig van Beethoven
Burn rate in homogeneous premixed charges can be enhanced by flame acceleration caused by turbulence generated in the flow field and distortion of the flame front, by placing obstacles in the path of the propagating flame front.

The work presented here makes use of the above phenomenon of using obstacles to enhance burn rate in spark-ignition engines.

From the evaluation of heat loss to work done ratio and from constant volume combustion bomb experiments the obstacled surface has been optimized.

The applicability of the optimized surface has been demonstrated by single cylinder engine tests, the results of which show improvement in engine performance and has been compared with published data of a commercial I.D.I. engine.
ACKNOWLEDGEMENTS

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Science and Engineering Research Council and Loughborough University of Technology for scholarship support to carry out this investigation.

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Mr K Allen for the high speed photography.

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Mr B Ellis for fabricating the piston crowns.

Mr P Norton and the technical staff for their general services.

Mr K W Topley for the photographic services.

All friends who have helped in different ways throughout the project.
NOMENCLATURE

Geometrical parameters:

- \( A_f \): Area of the flame front
- \( b \): Width of obstacles
- \( D \): Height of the channel or bore diameter
- \( e \): Face length of three dimensional roughness
- \( g \): Lateral gap between roughness elements
- \( h \): Height of the ribs
- \( h_s \): Height of the Nikuradse sand grain roughness determined by the sieve used.
- \( L, L_{3r} \): Length of recirculation zone.
- \( p, P \): Pitch between rows of ribs.
- \( R \): Tube radius
- \( r \): Position of the obstacle from ignition source
- \( r_0 \): Position of the first obstacle from ignition source
- \( y \): Distance from the wall
- \( \theta \): Angle

Gas properties:

- \( \rho \): Density
- \( C_p, C_v \): Specific heat at constant pressure and volume
- \( k \): Thermal conductivity
- \( n \): Obstacle count
- \( \gamma \): Ratio of specific heats
- \( \nu \): Kinematic viscosity
- \( \mu \): Dynamic viscosity

Physical properties:

- \( h_c \): Convective heat transfer coefficient
- \( K_{en} \): Kinetic energy of turbulence
- \( P \): Pressure
- \( Q \): Heat loss
- \( q \): Heat flux
- \( \dot{h}_f \): Flame velocity at any position
\( \tilde{R}_{fo} \) Flame velocity at first rib
\( S_l \) Laminar flame speed
\( S_{tu} \) Turbulent flame speed
\( \tau \) Shear stress
\( \delta, \delta_{th} \) Momentum and thermal boundary layer thickness
\( T \) Gas temperature
\( u, U \) Velocity
\( u^*, u' \) Friction velocity, Turbulent velocity
\( V \) Volume
\( V_{gas} \) effective gas velocity
\( W \) Work done
\( V_t \) Turbulent viscosity
\( \alpha \) Thermal diffusivity

Dimensionless groups:
\( A_{hr} \) Slope of the logarithmic temperature profile
\( A_r \) Slope of the logarithmic velocity profile at rough surface
\( A_s \) Slope of the logarithmic velocity profile at smooth surface
\( C_f \) Skin friction coefficient
\( f \) Rough surface friction factor
\( f_0 \) Smooth surface friction factor
\( h^* = h u^*/v \) Roughness Reynolds number based on shear velocity
\( N_u \) Nusselt number
\( P_r \) Prandtl number (~ 0.7 for air)
\( P_{rt} \) Turbulent Prandtl number (~ 1 for air)
\( Re \) Reynolds number
\( R(h^+) \) Hydraulic roughness parameter
\( G(h^+) \) Thermal roughness parameter
\( S_t \) Stanton number
\( t^+, u^+, y^+ \) Dimensionless gas temperature, velocity, distance from the wall
Subscripts:

- $\text{avg}$: average
- $b$: burned
- $c$: combustion
- $m$: motoring
- $\text{max}$: maximum
- $n$: obstacle number
- $o$: Initial
- $r$: rough
- $s$: smooth
- $u$: unburned
- $w$: wall
- $\infty$: free stream

Constants:

- $C$: in eq. 1.6 ($= 0.24$)
- $\alpha$: in eq. 1.2 ($= 0.31$)
### ABBREVIATIONS

<table>
<thead>
<tr>
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<tbody>
<tr>
<td>ABDC</td>
<td>After top dead center</td>
</tr>
<tr>
<td>BTDC</td>
<td>Before top dead center</td>
</tr>
<tr>
<td>BDC</td>
<td>Bottom dead center</td>
</tr>
<tr>
<td>IFP</td>
<td>Initial flame development</td>
</tr>
<tr>
<td>LPP</td>
<td>Location of peak pressure</td>
</tr>
<tr>
<td>MBT</td>
<td>Minimum spark advance for best torque</td>
</tr>
<tr>
<td>MPP</td>
<td>Magnitude of peak pressure</td>
</tr>
<tr>
<td>PCMHNSROU</td>
<td>Prechamber multiple holed nozzle with a rough surface</td>
</tr>
<tr>
<td>PCMHNSMO</td>
<td>Prechamber multiple holed nozzle with a flat surface</td>
</tr>
<tr>
<td>RPM</td>
<td>Revolutions per minute</td>
</tr>
<tr>
<td>TDC</td>
<td>Top dead center</td>
</tr>
<tr>
<td>WWP</td>
<td>Ford world wide point</td>
</tr>
<tr>
<td>FH</td>
<td>Flat head</td>
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<tr>
<td>CC</td>
<td>Conventional chamber</td>
</tr>
<tr>
<td>DC</td>
<td>Divided chamber</td>
</tr>
<tr>
<td>PC</td>
<td>Prechamber</td>
</tr>
<tr>
<td>SSP</td>
<td>Single spark plug</td>
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<td>DSP</td>
<td>Double spark plug</td>
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CHAPTER 1
LITERATURE SURVEY

1.1 Introduction

The combustion process in the gasoline engine is initiated by a spark. Photographic observation and optical diagnostics of the combustion process show that the flame front is not thin and smooth but thick and wrinkled containing islands of unburned mixture within it [Rassweiler & Whithrow /1/, zur Loye & Bracco /2/, Keck /3/ and Smith /4/]. This structure of the flame front can be attributed to turbulence which enhances the process of engulfment of unburned charge into the flame front and also causes wrinkling [Keck /3/, Shelkin /5/, Rothrock & Spencer /6/, Gerstien & Dugger /7/, Mattavi et.al./8/]. During the combustion process undesirable by-products are generated, which are a major source of air pollution through the emission of species such as oxides of nitrogen (NOx), hydrocarbon (HC) and carbon monoxide (CO). In order to meet emission legislations it is necessary to control these pollutants. Previous studies by Fleming & Eccleston /9/, Heywood /10/ and a review by Edwards /11/ have shown that by controlling the engine variables such as the composition of fuel-air mixture, spark timing and turbulence level which yield lower combustion temperature, raise exhaust gas temperature and reduce quenching effects the emissions from the engine can be reduced.

Experimental studies by Quader /12/, Stebar & Benson /13/, Blumberg & Kummer/14/, Muzio et.al /15/ and Wimmer & McReynolds /16/ indicate that emissions of NOx can be controlled by
operating the engine with mixtures yielding low combustion temperatures viz., fuel rich mixtures or mixtures diluted with exhaust gas or excess air. To maintain low levels of CO emissions the latter method of controlling NO\textsubscript{x} is preferred. The chosen path to obtain lower combustion temperatures as well as lower emissions and fuel economy is to dilute the mixture with excess air: i.e., operating the engine with lean mixture. The reasons lie in the fact that lean mixture operation reduces heat loss to the walls, increases specific heat ratio and in addition the pumping or scavenging losses at part load are reduced, due to wider throttle setting, which was first demonstrated with engine tests by Hopkinson /17/ and was also confirmed by Bolt & Holkeboer /18/.

Experimental studies by Bolt & Holkeboer /18/ indicate that as mixtures get leaner combustion duration increases which can be attributed to low flame speeds. Flame speed data on hydrocarbon fuels [Belles & Swett /19/, Glassman /20/, Dugger /21/, Guider /22/, Metghalachi & Keck /23/] indicate that flame speed is strongly dependent on the fuel-oxidant ratio passing from zero at the lean flammability limit through a maximum at about stoichioemtric and back to zero at the rich limit of flammibility. The limits of flammability for methane, propane & iso-octane on the lean side are 5.3, 2.2, 1.11 and on the rich side are 14.0, 9.5, 6.0 respectively based on percent fuel volume in air.

Operating the engine with lean mixture requires advanced spark timing due to longer delay periods evident from the experiments of Bolt & Holkeboer /18/. Increased spark advance results in the ignition of lean mixture at less favourable conditions of temperature and pressure [Belles & Swett /19/]. As a result the ignition and flame propagation deteriorates resulting in incomplete burn and misfires thus disturbing the regularity of successive combustion cycles resulting in cyclic variability [Tanuma et.al. /24/, Quader /25,26/, Hamamoto et.al. /27/, Anderson & Lim /28/]. This causes unstable engine operation,
degradation in fuel economy and higher HC emissions.

To overcome the above problems encountered during lean burn operation it is necessary to reduce the combustion period by promoting fast burn [Tanuma et al. /24/, Quader /25,26/, Hamamoto et al. /27/]. Fast burn can be achieved by increased charge turbulence via an increase in the area of the flame front due to wrinkling and to a lesser extent by the enhancement of transport process as mentioned earlier. Turbulence generation in the past has been achieved by the following methods:

(a) during flow of the intake mixture by use of masked valves such as shrouded valve and vortex valve as shown in Fig 1.1 (a) [Lee /29/, Lancaster /30/, Pischinger & Adams /32/].

(b) specially designed intake ports such as swirl inlet port as shown in Fig 1.1 (b) [Mattavi /33/].

(c) changes in chamber geometry such as May fireball chamber and bowl in piston as shown in Fig 1.1 (c) [May /34/, Belaire et al. /35/].

The above changes generally produce higher in cylinder mean flow velocities and turbulence levels [Lee /29/, Lancaster & coworkers /30,31/, Pischinger & Adams /32/, Belaire et al. /35/]. Higher velocities and turbulence levels are known to adversely affect flame initiation and development from ignition by spark due to quenching effects [Swett /36/, Hattori et al. /37/, de Soete /38/, Douaud et al. /39/]. Increased turbulence while increasing the flame speed narrows the range of mixture composition over which it is effective because the ratio of reaction time to mixing time becomes very large resulting in extinction of the flame front by dilution with unburned mixture [Ohigashi et al. /40/]. Moreover the use of masked valves impose severe breathing penalties resulting in lower volumetric efficiency [Lancaster /30/, Pischinger & Adams /32/].
Another method of increasing turbulence is to allow the propagating flame to generate velocity and shear gradients in the unburnt mixture ahead of it, due to volumetric expansion of burned gases, by placing obstacles along the path of the flame. Non-engine oriented combustion experiments conducted with a view of safety and hazard, posed by the presence of combustible mixtures in coal mines and nuclear reactors indicate that very high turbulent flame speeds can be achieved by placing obstacles along the path of the flame [Mason & Wheeler /41/, Chapman & Wheeler /42/, Robinson & Wheeler /43/, Moen et al. /44/]. Thus the possibility of achieving fast burn rates by using obstacles in quiescent combustion chambers can be realized without the necessity of using masked valves or swirl intake port or changes in chamber geometry which inhibit flame development or impose breathing penalties. Thus the mechanisms involved in the flame acceleration process caused by obstacles placed in the direction of propagation of the flame front and its effects on thermal efficiency due to higher heat loss because of higher turbulence needs further insight.

1.2 Flame propagation over obstacles
During combustion of methane and air in a tube of 30.5 cm diameter & 90 m long open at both ends with obstacles introduced at two places having a blockage ratio of 0.2, Mason and Wheeler /41/, observed the development of a detonation wave or some mode of combustion of similar intensity. They attributed this to the local turbulence generated by the flame as it passed over the obstacles.

Studies of flame propagation were carried out by Chapman and Wheeler /42/ in quiescent mixture of 9.75% methane (by volume) in air along a tube of 5 cm diameter and 240 cm long containing 12 restriction rings placed 5 cm apart having blockage ratios of 0.28, 0.5 & 0.7 to determine the maximum speed at which the flame could travel at atmospheric pressure and temperature. Their observation showed:
(a) the effect of an obstacle was to decrease the mean flame speed ahead and to increase the mean flame speed after the obstacle.

(b) the effect of height of obstacle on flame speed was maximum at a blockage ratio of 0.5 in comparison to blockage ratios of 0.28 and 0.7.

(c) the effect of thickness of the obstacle on flame speed was minimal; rings of thickness 1 mm showing the greatest increase in flame speed in comparison to rings of 10 mm and 20 mm thickness at a blockage ratio of 0.5.

(d) the final flame speed attained continued to increase with increase in the number of restriction rings reaching a maximum speed of 420 m/s with 12 rings placed 5 cm apart at a blockage ratio of 0.5. Further increase in the number of restriction rings produced no change in the final flame speed.

(e) the flame appeared to pass through the restricted zone as a thin tongue spreading axially through the free opening at the restriction.

(f) just beyond each restriction the flame became thicker than normal, thus causing an abnormal increase in the amount of mixture burnt locally.

It was believed that the movement of the unburnt mixture and the convergence of lines of flow in the unburnt mixture, ahead of the flame, increased as the flame advanced. Thus the flame front was moving in a medium which was in motion in the same direction as that of the flame front resulting in its acceleration.

Robinson and Wheeler /43/ extended the above work of Chapman and Wheeler /42/ by carrying out experiments in a tube 32.3 m long and 30.5 cm in diameter, open at both ends, with 11 restriction rings placed 30.5 cm apart having a blockage ratio of 0.66 using mixture of 10% methane (by volume) in air. The first ring was located at 13.4 m from the ignition end of the tube. The rapid increase in the flame speed within the chamber, the region
between successive restrictions [Fig 1.2], was attributed to the release of pressure from the previous chamber which not only induced turbulence in the successive chambers but also folded the flame, thereby increasing the area of the flame surface causing rapid combustion. An analysis of their flame speed data fails to reveal the presence of a detonation wave as noted by Mason and Wheeler /41/ or any discontinuity in combustion. On the contrary the acceleration of the flame speed was smooth and continuous.

Investigation concerning the influence of obstacles and their configuration on the propagation of stoichiometric methane-air flames in a cylindrical chamber [Fig 1.3] of radius 30.5 cm containing obstacles in the form of spirals at various pitch that could be placed on the top or bottom surfaces of the cylinder were carried out at McGill university by Moen et al. /44/. They observed that:

(a) the flame propagated slowly in a chamber without obstacles when compared to the rapidly accelerating flames with repeated obstacles. A flame speed of 6 m/s was observed during experiments in a chamber without obstacles when compared to a flame speed of 34 m/s attained by using spirals placed on bottom surface with a pitch of 1.91 cm having a blockage ratio of 0.25.

(b) for a given obstacle configuration the flame speed nearly doubled when two sets of repeated obstacles were placed on both the upper and lower surfaces than with only one surface having the repeated obstacle. The maximum flame speed observed with spiral obstacles on bottom surface was 73 m/s and a repetition of the same obstacle configuration on both the upper and lower surfaces increased the maximum flame speed attained to 130 m/s.

In another set of experiments performed by Moen et al. /44/ in a 1.22 x 1.22 m chamber containing obstacles in the form of spirals
throughout the chamber, having a blockage ratio of 0.57 and a pitch of 5.43 cm, it was observed that the flame accelerated all the way to the end of the chamber reaching a maximum speed of 57 m/s. For an identical obstacle configuration but with no obstacles beyond a radius of 30 cm the maximum flame speed attained was 25 m/s within the region of the obstacles slowing down to 4.9 m/s at the end of the chamber. This behaviour indicates that the flame adjusts itself to the local flow field unless the turbulence and distortions in the flow field are maintained by having repeated obstacles.

Moen et al. /44/ explained the above phenomenon of rapid acceleration of the flame caused by obstacles, on the basis of a positive feedback mechanism as follows:

In the presence of surfaces and solid boundaries, velocity and shear gradients will be generated in the unburnt gases. If obstacles are also present the flow field will be further distorted; velocity gradients will be formed in the boundary layers on the obstacle surfaces as well as in the wake of the obstacles. As the flame front advances into such a velocity gradient field the flame surface will be stretched and folded [see Fig 1.4]. This distorted flame consumes fresh gas over a larger surface area leading to an increased rate of heat release giving rise to higher effective burning velocity. A higher burning velocity results in larger displacement flow velocity of the unburned mixture ahead of the flame; this in turn gives rise to a more severe gradient field leading to more severe flame stretching - and so on [see Fig 1.5].

Based on the above positive feedback coupling, between the propagating flame and the flow field ahead of the flame, a model for the flame acceleration was proposed by Lee et al. /45/. Assuming the mean flow velocity of the burned gases behind the flame front is zero the flame speed across the channel, \( \dot{R}_f \), is
given by

\[ \hat{R}_f = \frac{p_u}{p_b} S_{tu} \]

where \( \frac{p_u}{p_b} \) is the density ratio of unburned to burned gas ratio.

\( S_{tu} \) is the turbulent burning velocity based on the average heat release per unit area of the flame front.

From the above positive feedback mechanism the increase in flame speed due to increase in turbulent intensity with flame speed is known to play the dominant role in the acceleration mechanism. Therefore denoting the average speeds of the flame front through the regions just before and just after the \( n^{th} \) obstacle by \( \hat{R}_f(n-1) \) and \( \hat{R}_f(n) \) the relationship for the increase in flame speed due to the increase in turbulence \( u' \) is given as

\[ \hat{R}_f(n) = \hat{R}_f(n-1)\left[1 + \frac{du'}{u'} \right] \]  \hspace{1cm} (1.1)

Since it is known that the turbulence \( u' \) generated by the obstacles is dependent on the spacing between obstacles, \( p \), the blockage ratio, \( h/D \), and the recirculation length, \( L \), the term \( du'/u' \) representing the relative increase in turbulence between the \( n-1^{th} \) obstacle and \( n^{th} \) obstacle can be expressed as [Fig 1.6]

\[ \frac{du'}{u'} = C \left( \frac{h}{D} \right)^\alpha \]  \hspace{1cm} (1.2)

by assuming a linear growth of turbulence in the wake of the obstacle and a power dependence on the blockage ratio \( h/D \) where \( \alpha \) and \( C \) are constants.

Substituting for \( du'/u' \) from eq. 1.2 in eq.1.1 we have

\[ \hat{R}_f(n) = \hat{R}_f(n-1)\left[1 + \left( \frac{\beta p}{L} \right)(\frac{h}{D})^\alpha \right] \]  \hspace{1cm} (1.3)
where: $\alpha, \beta$ are constants

- $p$ is the separation between the obstacles
- $h$ is the height of the obstacle
- $D$ is the plate separation, and
- $L$ is the length of the recirculation zone

Since eq. 1.3 does not take into account the initial stages of the flame acceleration process, viz: flame folding & wrinkling, it is necessary to assume that flame fold burning rate dominates the flame acceleration process until the first obstacle $n_o$ has been traversed. Then eq. 1.3 can be written for $n > n_o$ as

$$R_f(n) = R_f(n_o)[1 + C(p/L)(h/D)^{n-n_o}] \quad ...(1.4)$$

For small $p$ the term $C(p/L)(h/D)^{n-n_o} \ll 1$, hence eq. 1.4 reduces to

$$\frac{R_f(n)}{R_f(n_o)} = \exp[C(p/L)(n-n_o)(h/D)^{\alpha}] \quad ...(1.5)$$

To get the above eq. 1.5 into workable form it is necessary to substitute the recirculation length, $L$, and the obstacle count, $n$, in terms of geometrical parameters describing the configuration and reference position of the obstacles respectively. From a study of published literature on reattachment of separated flows, summarized in Table - I below, an average value for recirculation length can be taken as $L = 6h$ and introducing the position of the obstacle from the ignition point

- i.e., $r = p \times n$ and $r_o = p \times n_o$

eq. 1.5 can be rewritten, with $\alpha = 0.31$ from /44/ and /45/ as

$$\frac{R_f(n)}{R_f(n_o)} = \exp[C(p/L)(r/r_o)(h/D)^{0.31}] \quad ...(1.6a)$$

On rearranging the above eq. we have

$$\ln\left(\frac{R_f(n)}{R_f(n_o)}\right) = C(p/L)(r/r_o)(h/D)^{0.31} \quad ...(1.6b)$$
<table>
<thead>
<tr>
<th>Researcher</th>
<th>Method of Investigation</th>
<th>Recirculation length observed</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mueller et.al.</td>
<td>Experimental and theoretical investigation on separation, reattachment and redevelopment of boundary layer in turbulent shear flow</td>
<td>6 to 8 times the obstacle height.</td>
</tr>
<tr>
<td>/46/, 1964</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mantle</td>
<td>Flow visualization of air around roughness placed on a smooth surface</td>
<td>4.5 to 8 times the obstacle height.</td>
</tr>
<tr>
<td>/47/, 1966</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Counihan et.al.</td>
<td>Flow visualization of the boundary layer of a two dimensional block</td>
<td>6 times the obstacle height.</td>
</tr>
<tr>
<td>/48/, 1974</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bergels et.al.</td>
<td>Flow visualization of air past surface mounted obstacles</td>
<td>3 to 11 times the obstacle depending on the width of the obstacle.</td>
</tr>
<tr>
<td>/49/, 1983</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bates et.al.</td>
<td>Velocity measurements in a channel over a rib roughened surface</td>
<td>5 to 6.5 times the obstacle height.</td>
</tr>
<tr>
<td>/50/, 1983</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ghoniem et.al.</td>
<td>Numerical simulation of recirculating flow and comparison with experimental</td>
<td>5 to 7 times the obstacle height.</td>
</tr>
<tr>
<td>/51/, 1985</td>
<td></td>
<td></td>
</tr>
<tr>
<td>/52/, 1986</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
By plotting the flame speed data of /43, 44/ [see Fig 1.7] the constant C in eqs 1.6 a & b evaluated by Dent /53/ is found to be 0.24.

Lee and co-workers /54/ conducted experiments in tubes of 5, 15 & 30 cm in diameter & 14 m in length having blockage ratios varying from 0.25 to 0.6 with different concentrations of H₂ in air ranging from 10 to 50% by volume, and simulation work in rectangular channels having a blockage ratio of 0.6 assuming premixed hydrogen air mixture. From the above study on the mechanisms involved in flame acceleration over obstacles they concluded that:

(a) for lean mixtures turbulence plays an important role because of their sensitivity to flame folding and stretching.

(b) steady state flame speed results from a balance between the positive effects of flame folding via an increase in the burning surface area and the negative effect of flame quenching due to excessive flame stretching. The speed of the flame is said to have reached a steady state when its speed does not change with time and occurred within 10 to 20 tube diameters.

(c) larger diameter tubes appear to increase the flame acceleration rate but their influence on the final steady state flame speed was minimal.

1.3 Application to spark ignition engines

Thus from the above review the possibility of using repeated obstacles to enhance combustion rate via acceleration of the flame by placing obstacles in its path was first explored and its applicability in spark ignition engines demonstrated by Dent /53/.

The presence of the obstacles enhances combustion rate, via distortion of the flame front and higher turbulence as seen from
the above survey on "flame propagation over obstacles" which is caused by the positive feedback coupling mechanism as illustrated in Fig 1.5. This enables the combustion period to be reduced because of faster burn, thereby allowing the possibility of using lean mixture, and the ignition timing to be retarded. Thus the advantages gained are:

(i) by employing lean mixtures lower $NO_x$ & CO emissions and better fuel economy,

(ii) from retarded ignition timing are lower HC & CO emissions as the engine exhaust temperature is raised thereby promoting oxidation of these species,

(iii) by reducing the combustion period results in lower cyclic variability,

(iv) the tendency to knock is reduced because the flame front reaches the end gas before auto ignition occurs,

Though the above fast burn is achieved from higher turbulence it should also be kept in mind that higher turbulence causes:

(a) higher heat loss to the walls because of higher heat transfer coefficient.

(b) increased possibility of flame quenching [Lee et.al. /54/] eventually leading to its extinction [Spalding /55/] especially at lean mixtures.

The end effects of above disadvantages results in lower thermal efficiency, higher HC emissions, increased cyclic variability and poor fuel economy thus taking the problems facing lean mixture combustion to the beginning described in sec 1.1.

Thus a trade-off on the positive and negative effects of using obstacles in Spark-Ignition engines is required. In other words a balance has to be struck between what can be gained effectively, regarding workdone & emissions, and increased heat loss.
Hence a study into the effect of ribs on turbulent friction and heat transfer co-efficient to evaluate heat loss is needed which makes it a necessity to determine velocity and temperature profiles in a fluid over a surface having obstacles i.e., a rib roughened surface.

1.4 Types of rough surfaces
Since the possibility of using rough surfaces has been evaluated above, the question of what type, shape and configuration of rough surface is to be used in an engine combustion chamber remains. As it can be seen there are a large variety of [see Fig 1.8] possible rough surface geometries that can be used to accelerate the flame, it is therefore necessary to develop a global model that can predict the thermohydraulic characteristics: viz., friction and heat transfer co-efficient of rough surface from the parameters, such as transversal and lateral pitch, height and width describing the rough surface geometry.

1.5 Thermohydraulic characteristics of fluid flow over rough surfaces
For the turbulent flow of fluids over a smooth surface, the velocity distribution normal to the smooth wall can be classified into three regimes [see Fig 1.9]:

(a) a predominantly viscous region immediately adjacent to the wall surface, also known as viscous sublayer where the momentum and heat transfer can be largely described by molecular process involving viscous shear and conduction;

(b) a transition region comprising most of the boundary layer where eddy motion is observed, and the momentum & heat transport rates are much greater than in the viscous sublayer;

(c) a region outside the boundary layer where the velocity is constant, and equal to the free stream velocity, and the flow in this region remains unaffected by the wall.
Thus the velocity distribution in a pipe or a channel can be illustrated as shown in Fig 1.10 (a) and as expected the position of the maximum velocity lies at the centerline of the channel. With the introduction of obstacles on one of the walls the resistance offered to the flow [friction] increases, because of drag experienced by the obstacles, resulting in asymmetric distribution of velocity [see Fig 1.10 (b)]. The location of the maximum velocity is now displaced towards the smooth wall. The degree to which the flow is affected depends on the geometrical parameters of the obstacle configuration representing the rough surface and the blockage ratio.

One of the earliest and most important works on friction of rough surfaces was that of Nikuradse /56/ for flow inside pipes having various sand roughnesses. He found the velocity distribution normal to the rough wall is given by

\[ u^+ = 2.5 \ln \left( \frac{y^+}{h^+} \right) + R(h^+) \]  \hspace{1cm} (1.7)

By introducing the friction velocity, \( u^* \), and kinematic viscosity, \( \nu \), in the logarithmic term, eq. 1.7 can be written as

\[ u^+ = 2.5 \ln (y^+) - 2.5 \ln (h^+) + R(h^+) \]  \hspace{1cm} (1.8)

where: \( u^+_r = u_r/U_{\text{max}} \)
\( y^+ = y u^*/\nu \)
\( h^+ = h u^*/\nu \)
\( u^* = \sqrt{T/\rho} \)

On comparison with his earlier work on friction on smooth surfaces, the velocity profiles which can be described by the familiar "law of the wall" in turbulent flow was given by

\[ u^+_s = 2.5 \ln (y^+) + 5.5 \quad \text{for } y^+ \geq 70 \]  \hspace{1cm} (1.9)
Nikuradse found that the fluid velocity in the presence of a rough wall, [eq. 1.8], differs from the velocity distribution on a smooth surface only by an additive factor which becomes prominent near the wall and is influenced by the geometrical parameters defining the rough surface present on the wall. Nikuradse also classified the fluid flow on a rough surface into three regimes:

<table>
<thead>
<tr>
<th>Flow regime description</th>
<th>Range of $h^+$</th>
<th>value of $R(h^+) - 2.5 \ln(h^+)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydraulically smooth regime</td>
<td>$0 \leq h^+ &lt; 5$</td>
<td>5.5</td>
</tr>
<tr>
<td>Transitional regime</td>
<td>$5 \leq h^+ &lt; 70$</td>
<td>variable value</td>
</tr>
<tr>
<td>Completely rough regime</td>
<td>$h^+ \geq 70$</td>
<td>8.5</td>
</tr>
</tbody>
</table>

Integrating eq. 1.7 over $y$ in the cross section of a tube of radius $R$ gives:

$$u_{avg}^+ = u_{max}^+ - 3.75 \quad \ldots(1.10)$$

By definition of friction factor, $f$,

$$f = \frac{2 \tau}{\rho u_{avg}^2} \quad \ldots(1.11)$$

and noting that

$$u_{max}^+ = 2.5 \ln\left(\frac{R}{h}\right) + R(h^+)$$

Nikuradse obtained the similarity law of friction given by:
Schlichting /57/ measured friction factors for various roughnesses in rectangular ducts and to characterize the roughness that was used, he introduced the term 'equivalent sand roughness', i.e., the ratio $h_s/R$ used by Nikuradse. Schlichting /58/ also showed the general applicability of eqs. 1.7 and 1.9 for different geometries such as ducts, flat plates and pipes.

The work of Nikuradse /56/ and Schlichting /57,58/ were essentially carried out to investigate the drag experienced by ships and aeroplanes due to various roughness. Moreover the blockage ratio was very low [0.002 to 0.04].

Earlier experimental work by Nunner /59/ on heat transfer for flow inside artificially roughened tubes with two dimensional roughness was confined to the determination of the Nusselt number, $N_u$. Based on the Reynolds analogy for momentum - heat transfer and by assuming turbulent Prandtl number, $P_{\tau}$, equal to unity he arrived at the following expression for $N_u$

$$N_u = \frac{f}{8} \frac{R_s P_{\tau}}{1 + 5 R_e^{-1/8} P_{\tau}^{1/8} [P_{\tau} (f/f_0) - 1]}$$

... (1.13)

where : $f_o, f$ are friction factors for smooth and rough tubes respectively

$R_e$ is the Reynolds number

$P_{\tau}$ is the Prandtl number

In 1963 Dipprey and Sabersky /60/ published measured friction and heat transfer data for flow of water inside tubes having sand roughness, similar to that used by Nikuradse, for various Reynolds numbers over a Prandtl number range of 1.2 to 5.94. Their method of correlating heat transfer results was very much
similar to that used by Nikuradse to correlate friction data and they assumed the dimensionless temperature distribution normal to the rough wall is given by

$$t^+ = A_h \ln \left( \frac{Y}{h} \right) + G(h^+, P_x) \quad \ldots \text{(1.14)}$$

whereby by definition:

$$t^+ = \frac{(T_w - T) \rho_b C_{pb} u^*}{q'}$$

and $T_w$ is the wall temperature of the rough surface; $T$ is the fluid temperature at a point distance $y$ from the rough wall having a roughness of height $h$; $\rho_b$ and $C_{pb}$ are the averaged fluid density and specific heat at constant pressure; $u^*$ is the friction velocity; $q'_g$ is the heat flux from the wall to the fluid and $A_h$ a constant taken as 2.5. Integrating eq. 1.14 over $y$ in the cross section of a tube of radius $R$ yields:

$$t_{avg}^+ = 2.5 \ln \left( \frac{R}{h} \right) + G(h^+, P_x) - 3.75 \quad \ldots \text{(1.15)}$$

and is analogous to the velocity [ eq. 1.11].

By definition of Stanton number and friction velocity

$$t_{avg}^+ = \frac{\sqrt{El/2}}{S_t} \quad \ldots \text{(1.16)}$$

one obtains from eqs. 1.15 and 1.16

$$\frac{\sqrt{El/2}}{S_t} = 2.5 \ln \left( \frac{R}{h} \right) + G(h^+, P_x) - 3.75 \quad \ldots \text{(1.17)}$$

analogous to the friction similarity law of Nikuradse [eq. 1.12].

On rearranging eqs. 1.12 & 1.17 Dipprey and Sabresky obtained

$$G(h^+, P_x) = R(h^+) + \frac{(f/2S_t) - 1}{\sqrt{El/2}} \quad \ldots \text{(1.18)}$$
Thus the task of determining heat transfer coefficient and friction factor lies in finding the values of $G(h^+, Pr)$ and $R(h^+)$. Based on their experimental data they arrived at a relation for the parameter $G(h^+, Pr)$ in the fully rough flow regime given by

$$G(h^+, Pr) = 5.19 h^+(0.2) Pr^{0.44} \quad \ldots(1.19)$$

Their data correlated quite well with the parameter $G(h^+, Pr)$ and they were also successful in correlating the data of Nunner /59/ for two dimensional roughness with a different $G(h^+, Pr)$ parameter.

An explanation of the above phenomenon of increased heat transfer was put forward by Dipprey and Sabresky /60/, known as the 'cavity vortex hypothesis'; which states: "the rough wall to consist of small cavities containing one or more standing vortices and the heat transmitted through the thin boundary layer formed on the cavity walls is assumed to be convected by the vortices to the cavity openings".

Gomelauri /61/ conducted experiments to study the influence of artificial roughness on friction & heat transfer, using oil & water as fluids, through an annular channel containing ring type two dimensional roughness, having square and circular [Fig 1.12] crosssections, over a blockage ratio ranging from 0.05 to 0.1. His observations showed:

(a) heat transfer rate reaches a maximum at a certain pitch ranging from 12 to 14 times the rib height.

(b) wall temperatures were minimal at the ribs due to thickening of the wall at the ribs and an corresponding increase in heat transfer rate.

Based on his observations he concluded that the laminar sublayer breaks up at the ribs resulting in lower wall temperatures due to reduction in thermal resistance of the sublayer.
Mantle /47/ speculating that a surface roughened by three dimensional roughness might perform better thermally than a sand roughened or two dimensional ribbed surface [Nunner /59/, Dipprey and Sabresky /60/, Gomelauri /61/], selected cubes to be placed in "daimond" orientation, to represent the rough surface, for heat transfer tests [see Fig 1.11]. The cubes chosen for the heat transfer tests were of 0.12 cm high with a spacing of 0.8 cm between them. From the heat transfer tests he found the performance of diamond orientation three dimensional and transverse ribbed two dimensional surfaces to be the same.

Gowen and Smith /62/ assuming the velocity and temperature distribution in rough tubes to follow logarithmic variation [eq.s 1.7 and 1.14] found the constant $A_h$ in eq. 1.14 to be a function of Prandtl number varying from 2.7 at $Pr = 0.7$ to 3.5 at $Pr = 14.3$ in the completely rough regime for flow of fluids in tubes having a sand grain roughness of 0.05.

Webb et. al. /63/ from their experimental observations, on heat transfer to fluids with rib roughness having a blockage ratio ranging from 0.01 to 0.04 at Prandtl numbers of 0.7, 5.1 and 21.7 for a Reynolds number range of $6 \times 10^3$ to $10^5$, concluded that:

(a) the law of the wall similarity could be applied to wider blockage ratios ranging from 0.01 to 0.25.

(b) Reynolds heat - momentum analogy to be applicable to geometrically similar forms of arbitrary roughness.

DalleDonne and Meerwald /65,66/ in their extensive measurements of heat transfer and skin friction coefficients for two dimensional roughnesses having trapezoidal and square crosssections [see Fig 1.8] with blockage ratios varying from 0.01 to 0.04 arrived at a correlation for the roughness parameter $R(h^+)$ [eq. 1.7] as a function of $p/h$ and $h/b$ [Fig 1.8] from which
the heat transfer coefficient was a function of $R(h^+)$ and $G(h^+, Pr)$ [eqs 1.7 & 1.14] which describes the heat transfer in the vicinity of the rough surface.

Based on their results for ribs having trapezoidal /65/ & square /66/ crosssections, the following correlations were suggested for $R(h^+)$ and $G(h^+, Pr)$:

<table>
<thead>
<tr>
<th>Range of $p/h$</th>
<th>$h/b$</th>
<th>Value of $R(h^+)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>any range</td>
<td>&lt; 0.4</td>
<td>6 - 7</td>
</tr>
<tr>
<td>≥ 10</td>
<td>0.4 &lt; $h/b$ &lt; 1.0</td>
<td>$f(p/h)$</td>
</tr>
<tr>
<td>≥ 10</td>
<td>≥ 1.0</td>
<td>$f(p/h, b/h)$</td>
</tr>
</tbody>
</table>

and

$$G(h^+, Pr) = 5.8 h^+ 0.2 Pr 0.44 (T_W/T_B)^{0.2} \text{ for } h^+\geq30 \quad ...(1.20)$$

DalleDonne and Meerwald /66/ also conducted experiments with surfaces having three dimensional roughness and found their performance to be much better than two dimensional ribbed surfaces, contradicting the results of Mantle /47/. The reason being that the spacing between individual roughnesses in Mantle's experiments were too large allowing the flow to reattach to the surface (smooth region) before encountering the next obstacle.

Migai and Bystrov /67/ carried out experiments with air on friction and heat transfer in pipes having ribs in the form of rings and segments of rings to form two and three dimensional rough surfaces respectively. They observed a higher heat transfer rate with semicircular ribs than with circular ribs. It was reasoned that "during the flow over semicircular projections a vortex sheet is formed downstream of the obstacle, characterized by increased turbulence, while the region not occupied by the
projections is less turbulent. This anisotropic turbulence leads to intensive transfer of mass between the low turbulent zone and the increased turbulent zone. As a result a specific portion of the low-turbulent flow is directed along the boundary layer in the trailing zone leading to the formation of secondary flows which contribute towards intensification of heat transfer, which was identified by measuring velocity distributions downstream of the projections.

DalleDonne and Meyer /68/ published experimental data on friction and heat transfer from rough surfaces with two dimensional rectangular ribs for a blockage ratio varying from 0.08 to 0.25. Based on their experimental results for air and earlier data by other investigators /59-63/ it was found that the parameter \( (p-b)/h \) was more significant than \( p/h \) or \( h/b \) for correlating friction data of two dimensional rough surfaces. The reason being that: "for a two dimensional rib the flow reattaches downstream at a distance of about 4 to 6 times the rib height. The region between the rib and the point of reattachment at the wall being occupied by vortices. After reattachment the viscous sublayer starts to grow, therefore the local skin friction & heat transfer coefficient, which have maximum values at the reattachment point, decreases downstream from the point of reattachment. The above phenomenon is mainly dependent on the flow between the ribs and less on the flow pattern at the top face of the rib. For this reason \( (p-b)/h \) is chosen to be a better parameter rather than \( p/h \) or \( h/b \).

From the parameter \( (p-b)/h \) the following correlations for the hydrodynamic and thermal roughness parameters \([R(h^+), G(h^+,P_r)]\) in eq.s 1.7 & 1.14, based on the geometry of the rough surface valid in the range \( 1 \leq (p-b)/h \leq 160 \) and \( 0.086 \leq h/b \leq 5.0 \), were suggested and are given by:

\[
R(h^+) = 2.51\ln(h^+) + 5.5 \quad \text{for} \quad h^+ \leq 30 \quad ...(1.21 \ a)
\]
\[ R_1 = R_{1\phi} + 0.41n\left(\frac{100h}{D}\right) + \frac{5100}{h^{+3}} + \frac{5}{h^{0.5}} \left(\frac{T_w}{T_b} - 1\right)^2 \quad \text{for } h^{+} \geq 30 \]

\[ R_{1\phi} = 9.3\left(\frac{p-b}{h}\right)^{-0.73} - [2 + \frac{7}{(p-b)/h}] \log\left(h\right) \quad \text{for } 1 \leq \frac{p-b}{h} < 6.3 \quad (a) \]

\[ = 1.04\left(\frac{p-b}{h}\right)^{0.46} - [2 + \frac{7}{(p-b)/h}] \log\left(h\right) \quad \text{for } 6.3 < \frac{p-b}{h} < 160 \quad (b) \]

\[ G(h^+) = G_1(h^+)p^{0.44}\left(\frac{T_w}{T_b}\right)^{0.5}\left(\frac{100h}{D}\right)^{-0.053} \quad \text{for } h^{+} \geq 30 \quad (1.23) \]

where \( G_1(h^+) = (3 + 0.3R_{1\phi} h^+(0.32 - 0.017R_{1\phi}) \quad (1.24) \)

The above parameters \( R_{1\phi} \) and \( G_1(h^+) \) shown in Fig 1.13 illustrate the validity of relations given by eq.s 1.22 & 1.24 for two dimensional ribs describing the rough surface.

Based on experimental investigation into heat transfer and friction of air flowing in ducts having rectangular and trapezoidal ribs, Vilemas and Simonis /69/ modified the above relations for \( R(h^+) \) and \( G(h^+) \) parameters [eq.s 1.21 and 1.22] to take into account the boundary layer thickness and the effect of Prandtl number given by

\[ R_1(h^+) = R(h^+) - 0.41n\left(\frac{R}{D}\right) \quad (1.24) \]

and

\[ G_1(h^+) = 8.7 + 0.15 \exp[1.6 \log(h^+)] \quad \text{for } h^{+} \geq 5 \quad (1.25) \]

where (a) for rectangular ribs

\[ R_1(h^+) = R_{1\phi} + 4.3\exp[-0.81\log^2(h^+)] \quad \text{for } h^{+} \geq 5 \quad (1.26) \]
(b) for trapezoidal ribs

\[ R_1(h^+) = R_{1\phi} + 12 \exp[-2.41\log(h^+)] \quad \text{for } h^+ \geq 5 \quad ...(1.27) \]

\( R_{1\phi} \) is calculated from eqs 1.22.

The above equations modified by Vilemas and Simonis /69/ along with the eqs 1.21 to 1.24 proposed by DalleDonne and Meyer give the complete description of thermohydraulic parameters for a two dimensional transverse ribbed surface which are necessary to evaluate friction factors and Stanton numbers from the velocity and temperature profiles.

Meyer et.al. /70-74/ performed extensive measurements of velocity and temperature distribution in rectangular and annular channels having four different types of three dimensional roughnesses [see Fig 1.14] to obtain friction and heat transfer data with air as the fluid for various configuration of non dimensional geometric parameters such as \( h/D \); \( p/h \) and \( g/e \).

From their measurements they observed:

(a) the velocity and temperature profiles [eqs 1.7 and 1.14] do not obey the law of the rough wall with a constant slope unlike two dimensional rib roughened surfaces.

(b) the slope of the velocity at the smooth wall whose value was found to be lower than 2.5 decreased further with increasing Reynolds number.

(c) three dimensional roughness showed a marked improvement in thermal performance in comparison with the best two dimensional repeated roughness. Firth /75/ attributed this to the stronger interaction between the surface roughness and the mainstream fluid caused by a change in the turbulent structure and diffusivity in the mainstream flow.

(d) the thermal performance was found to be very sensitive
to the changes in the surface geometry of the roughness.

(e) From the detailed analysis of their experimental data the following correlations were developed to obtain the slopes for the logarithmic distribution of velocity & temperature profiles at both smooth and rough walls. They are given by:

1) Velocity distribution:

(a) For rough wall

\[ u_r^+ = A_r \ln \left( \frac{y}{h^+} \right) + R(h^+) \]  \hspace{1cm} \text{...(1.28)}

where: 

\[ A_r = 1.5 + \frac{2}{h^+} \]

for \( h^+ \geq 20 \)

\[ = 2.5 \]

for \( h^+ < 20 \) \hspace{1cm} \text{...(1.29)}

and \( R(h^+) = 0.41n \left( \frac{100h}{D} \right) + R_1(h^+) \) \hspace{1cm} \text{...(1.30)}

where \( R_1(h^+) = 2.51n(h^+) + 5.5 \)

for \( h^+ < 5.0 \) \hspace{1cm} \text{(1.31 a)}

\[ = R_1 + 4.3 \exp[-0.81 \log^2(h^+)] \] \hspace{1cm} \text{for } h^+ \geq 5.0 \hspace{1cm} \text{(1.31 b)}

\[ R_1 = p^+ + \left[ \frac{h^+}{2D} \right]^{-0.73} \]

for \( p^+ \leq 4.25 \)

\[ = p^+ \]

for \( 4.25 < p^+ < 5.75 \)

\[ = p^+ - \left[ 2 + \frac{7}{(p^+ - 5)} \right] \log \left( \frac{h^+}{D} \right) \]

for \( p^+ \geq 5.75 \) \hspace{1cm} \text{...(1.32)}

where \( p^+ = \left[ \frac{p-b}{h} \right] (1 + g) \)

(b) for smooth wall

\[ u_s^+ = A_s \ln(y^+) + 5.5 \]  \hspace{1cm} \text{...(1.33)}

where
\[ A_s = 2.44 - \frac{1}{R(h^+)} \log(h^+) \] ...(1.34)

2) Temperature distribution
(a) for rough wall

\[ t_r^+ = A_{hr} \ln(h^+) + G(h^+) \] ...(1.35)

where \[ A_{hr} = 4.4 \quad \text{for } h^+ < 10.0 \] ...(1.36)

\[ = 1.4 + \frac{3.0}{\log(h^+)} \quad \text{for } h^+ \geq 10.0 \]

where

\[ G(h^+,Pr) = G_1(h^+)Pr^{0.44} \left( \frac{T_w}{T_b} \right)^{0.68} \left( \frac{100h}{D} \right)^{0.053} \text{ for } h^+ \geq 30 \] ...(1.37)

\[ G_1(h^+) = 7.8 \quad \text{for } h^+ < 5.0 \]

\[ = 7.8 + 0.15 \exp[1.6 \log(h^+)] \text{ for } h^+ \geq 5.0 \] ...(1.38)

(b) for smooth wall:

\[ t_s^+ = 2.195 \ln(y^+) + 3.715 \] ...(1.39)

1.6 Summary of reviewed literature

From the above survey of literature it can be concluded that rough surfaces intensify the turbulence in the flow via local detachments and reattachments of the boundary layer. As a result of increased turbulence the drag experienced by the surface increases, due to the presence of the obstacles, and also causes the viscous sublayer to be either reduced in thickness or destroyed completely depending on the flow velocity, type, shape and configuration of the obstacle. Thus a higher skin friction and heat transfer coefficient are experienced by using a rough surface due to increased drag and lowering of thermal resistance offered by the viscous sublayer.
From an analysis of the data of Meyer and co-workers /70 - 75/ it can be seen that three-dimensional roughness, create more disturbance to the flow which is a direct consequence of higher friction factor and hence higher thermal performance, in comparison to two-dimensional square cross sectioned transverse ribbed surface [see Figs 1.15]. It can be attributed to the shorter 'dead water' region behind a three-dimensional obstacle compared to those behind two-dimensional ribs. Hence three-dimensional obstacles can be packed closer in the axial direction to produce more drag and hence higher turbulence.

From the relations developed in sec 1.5 for hydraulic and thermal roughness parameters it should be possible to determine the velocity and temperature profile on a rough surface based upon the parameters describing the type and configuration of the rough surface. With the aid of a phenomenological combustion model the configuration of the rough surface can be optimised so as to minimise the ratio of heat loss to work done.
FIG. 1.1  COMMONLY USED METHODS OF GENERATING TURBULENCE

(a) Masked Valves
(b) Swirl inlet port design
(c) Chamber geometry

INLET PORT
SPARK PLUG
SQUISH
SWIRL
PISTON

IN
EX
IN
EX
IN
EX
IN
EX

Small Image:
Swirl
Spark Plug
May fire ball chamber

Bowl in piston
(c) Chamber geometry
Fig 1.2: Schematic Sketch of the Experimental Rig used by Robinson & Wheeler /43/, having a Blockage Ratio of h/R or 2h/D

Fig 1.3: Cylindrical Chamber used by Moen et.al. /44/
Fig. 1.4: Sequence of Schlieren photographs showing the propagation of the flame around the obstacle. [From Ref /45/ ]
Obstacles: (1) Generate shear layers. (2) Stretch flame. Which are proportional to flow velocity.

Above effects lead to increased burning rate

Resulting in higher volumetric expansion and hence higher flow velocities

Fig 1.5: Block diagram illustrating flame acceleration set up by positive feedback loop mechanism.
Fig 1.6: Schematic Representation of Flame Propagation in a Channel with Ribbed Obstacles
Flame Speed Ratio

\[ \frac{\dot{R}_f(n)}{\dot{R}_f(n_0)} \]

Roughness Parameter

\[ [\frac{r - r_0}{6h}]^h \overline{0.31} \]

Fig 1.7: Variation of Flame Speed With Repeated Obstacles; Illustrating the Phenomenon of Positive Feedback Mechanism in Flame Acceleration.
<table>
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Fig 1.15 (b): Comparison of Stanton Numbers for Two-Dimensional and Three-Dimensional Rib Roughened Surface [Ref /74/]
2.1 Introduction

In section 1.4 investigations by several researchers were presented which were mainly concerned with promoting heat transfer from fuel element rods to gases, in gas cooled nuclear reactors. By employing artificially roughened surfaces having regularly spaced ribs or obstacles which act as turbulence promoters the viscous sublayer breaks up near the roughened wall thus reducing the thermal resistance, resulting in higher heat transfer coefficient.

To promote accelerated combustion regularly spaced ribs or obstacles are necessary to promote high turbulence but while increasing the combustion rate the heat loss to the enclosure walls occurs with the additional problem of the ribs acting as extended surfaces. This leads to the necessity for optimization between increased combustion rate and heat loss.

2.2 Modelling the Effect of Roughened Surfaces on Velocity and Temperature Profiles

As seen earlier there are a large variety of roughness elements that can be used as turbulence promoters it is necessary to assume the obstacles to be uniformly spaced so that the velocity and temperature profiles of the fluid as it passes the rough surfaces can be modelled for a given geometry of the obstacle configuration. From the shape of the engine combustion chamber [see Appendix A] it is obvious to choose the flat surfaces on the head and/or piston on which to place the obstacles so that they lie in the path of flame propagation. The location of valves in the engine head limits the applicability of roughened surfaces to piston. This unsymmetric configuration can be treated as the flow of fluid in a channel having one rough surface (piston). As discussed earlier in section 1.4 this causes an assymetric velocity distribution by displacing the position of maximum velocity away from the midplane of the channel towards the
smooth wall (cylinder head). The above displacement causes slope of the velocity profile on the smooth wall to decrease resulting in higher friction factor. Thus the presence of the obstacles increase skin friction and heat transfer coefficient not only at the roughened surface (piston) but also at the smooth wall (cylinder head).

The solution to the above task of determining velocity and temperature profiles lies in determining hydraulic and thermal roughness parameters for a given obstacle configuration on the assumption that the velocity and temperature profiles at both the surfaces (rough and smooth walls) follow the basic logarithmic distribution. Viz.;

i) Velocity
   a) For rough wall:
      \[ u_r^+ = A_r \ln \left( \frac{y}{h} \right) + R(h^+) \]  \hspace{1cm} ...(2.2.1)
   b) For smooth wall:
      \[ u_s^+ = A_s \ln(y^+) + 5.5 \]  \hspace{1cm} ...(2.2.2)

ii) Temperature
   a) for rough wall :
      \[ t_r^+ = A_{hr} \ln \left( \frac{y}{h} \right) + G(h^+) \]  \hspace{1cm} ...(2.2.3)
   b) for smooth wall:
      \[ t_s^+ = 2.195 \ln(y^+) + 3.715 \]  \hspace{1cm} ...(2.2.4)
The effect of the roughness of a surface on the velocity and temperature profiles is thus represented by the degree of influence on the slopes ($A_x$, $A_y$, $A_{hr}$) and roughness parameters.

To obtain hydraulic and thermal roughness parameters for obstacle configuration of different shapes and sizes it becomes a necessity to assume the obstacles to be identical in shape and uniformly spaced for a given geometry describing the roughened surface.

From a review of the literature it is evident that three dimensional roughness elements create more disturbance to the flow and also allows the pitch between subsequent rows of ribs to be smaller due to a smaller dead water region.

This provides an opportunity to have a greater number rows of ribs so that the acceleration of the flame can be maintained by the positive feedback coupling mechanism described earlier in section 1.2.

The following correlations were developed from the data of Meyer et.al. /70-74/ and are adopted to predict the thermohydraulic characteristics of three dimensional roughness elements [see Fig 2.1]:

a) hydraulic roughness parameters:

$$R(h^+) = 0.41\ln\left(\frac{100h}{D}\right) + R_1(h^+)$$

where

$$R_1(h^+) = 2.51\ln(h^+) + 5.5 \quad \text{for } h^+ < 5.0 \quad (1.31 \text{ a})$$

$$= R_1\phi + 4.3\exp[-0.8\log^2(h^+)] \quad \text{for } h^+ \geq 5.0 \quad (1.31 \text{ b})$$

$$R_1\phi = p^+ + [1 + \frac{h}{2b}]^{-0.73} \quad \text{for } p^+ \leq 4.25$$

$$= p^+ \quad \text{for } 4.25 < p^+ < 5.75$$

$$= p^+ - \frac{7}{(p^+ - 5)} \log\left(\frac{h}{b}\right) \quad \text{for } p^+ \geq 5.75$$

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where \( p^+ = \frac{p-b}{h} (1 + \frac{q}{e}) \)

b) Thermal roughness parameter

\[
G(h^+, P_r) = G_1(h^+) P_r^{0.44} \left( \frac{T_w}{T_D} \right)^{0.68} \frac{100h}{D}^{0.053} \text{ for } h^+ \geq 30
\] ...

\[
G_1(h^+) = \begin{cases} 7.8 & \text{for } h^+ < 5.0 \\ = 7.8 + 0.15 \exp[1.6 \log(h^+)] & \text{for } h^+ \geq 5 \end{cases}
\] ...

To account for the variation in the slopes of velocity and temperature profiles due to roughness the following correlations are chosen so that the "logarithmic law of the wall" is applicable at the surface:

i) \( A_r \) in eq 2.2.1 is given by

where: \( A_r = 1.5 + \frac{2}{h^+} \) for \( h^+ \geq 20 \) ...

\[
= 2.5 \quad \text{for } h^+ < 20
\]

ii) \( A_s \) in eq 2.2.3 is given by

\[
A_s = 2.44 - \frac{1}{R(h^+) \log(h^+)}
\] ...

iii) \( A_{hr} \) in eq 2.2.3 is given by

\[
A_{hr} = \begin{cases} 4.4 & \text{for } h^+ < 10.0 \\ = 1.4 + \frac{3.0}{\log(h^+)} & \text{for } h^+ \geq 10.0 \end{cases}
\] ...

The width parameter 'b' for three dimensional roughness elements are evaluated as follows:
The shape of a given three dimensional roughness element is reduced such that the roughness element resembles a rectangular element having the same frontal surface area:

a) for rectangular and parallelogram elements [Figs 2.2a & b]
\[ b_1 = b \]

b) for hexagonal and diamond shaped elements [Figs 2.2b & c]
\[ b_1 \neq b \]

Employing the above correlations [eqs 1.30 & 1.37] for thermohydraulic roughness parameters of three dimensional roughened surfaces the velocity and temperature profiles can be predicted. Since the presence of a rough surface augments heat transfer it is necessary to predict the heat transfer coefficient at the surface either from integrating eq. 2.2.3 and evaluating gas velocity by Woschni's /76/ method or by making use of predicted temperature profile.

i) Integrating eq. 2.2.3 over y across the channel of height H yields

\[ \frac{\sqrt{E/2}}{S_t} = \alpha hr \ln\left(\frac{H}{h}\right) + G(h^+, \Pr) - 3.75 \]

from which Stanton number \( S_t \) can be evaluated.

Heat transfer coefficient \( h_c \) is then given by

\[ h_c = S_t \rho V_{gas} C_p \]

where the gas velocity \( V_{gas} \) is evaluated according to Woschni /76/, described below in section 2.3.3.
2.3 Convective Heat Transfer Modelling

The heat transfer coefficient varies with piston position in the cylinder and is time dependent. To obtain detailed spatial variation of heat transfer coefficient requires solution of the three dimensional unsteady Navier-Stokes equations coupled with energy and combustion species conservation equations, turbulence model and chemical kinetic rate equations. Moreover the modelling of boundary layer requires a very fine grid in comparison to the grid employed for the combustion chamber. These complications lead to a tremendous increase in development and computational costs.

Most of the correlations that have been widely used to predict heat transfer coefficient in engines rely heavily on those developed for steady flow through pipes or over flat plates.

The correlations have the form

\[ N_u = a \, \text{Re}^b \]  \hfill (2.3.1)

where:
- \( N_u \) is the Nusselt number \( = h_c D_1 / k \)
- \( \text{Re} \) is the Reynolds number \( = \rho D_1 V_{gas} / \mu \)
- \( a \) and \( b \) are constants.

- \( h_c \) is the heat transfer coefficient, \( D_1 \) is the characteristic length, \( \rho, \mu, k \) are density, dynamic viscosity & thermal conductivity of the gas.

One such form proposed by Annand /77/ has the following characteristics:
- Characteristic length is chosen as engine bore diameter
- Gas velocity, \( V_{gas} \), is taken to be proportional to the mean piston speed.
- The constants \( a \) & \( b \) are taken as 0.49 and 0.7 respectively
The widely used correlation by Woschni /76/ follows the above general approach [eq 2.3.1] with constants $a = 0.35$ and $b = 0.8$. The characteristic length is taken as the diameter of the engine bore. To represent the gas velocity Woschni also uses the mean piston speed but introduces an additional term to account for gas motion induced by combustion which is assumed proportional to the pressure rise due to combustion. Accordingly

$$V_{\text{gas}} = 2.28 \bar{U}_p + 0.00324 \frac{T_0 V \Delta P_c}{V_0 P_0} \quad \ldots(2.3.2)$$

where: $V_{\text{gas}}$ is the effective gas velocity (m/s)
$\bar{U}_p$ is the mean piston speed (m/s)
$T_0, V_0, P_0$ are temperature, cylinder volume and pressure at intake valve closing.
$V$ is the instantaneous cylinder volume (m$^3$)
$\Delta P_c$ is the instantaneous pressure rise due to combustion which is estimated from the difference between the cylinder pressures in the firing engine and the motored engine at the same crank angle.

The heat transfer coefficient in Annand's correlation is affected through the charge density $\rho$, occurring in Reynolds number evaluation, which is directly proportional to the pressure inside the cylinder. In Woschni's correlation the heat transfer coefficient is affected through the term $\Delta P_c$, occurs during the evaluation of effective gas velocity, which is the difference between the cylinder pressures in the firing and motored engines. The above correlations due to Annand and Woschni, yield spatially averaged heat transfer coefficients as a function of time.

Borgnakke et.al. /78/ proposed a flow based model to highlight the role of turbulence in heat transfer by the solution of the two equation $k - \epsilon$ model of turbulence in conjunction with a two zone combustion model. Subsequent development of the above model by Davis and Borgnakke /79/ led to the use of eq. 2.3.1 in the form
\[ N_u = \frac{h c_{1/2}}{k} = a \left[ \rho \left( \frac{1}{\mu} \right)^{1/2} K_{en} \right]^{0.7} = a \text{Re}^b \]  

\[ \text{...(2.3.3)} \]

where the effective gas velocity is taken to be the square root of turbulent kinetic energy \( (K_{en}) \) and 1 is the length scale of turbulence. The two turbulence equations for \( k \) and \( \epsilon \) are solved as lumped parameters separately for each of the two zones and so the heat transfer coefficients while different from zone to zone are spatially uniform within their respective zones.

Morel and Keribar /80/ proposed a model by assuming the convective heat transfer coefficient to be related to the fluid motion through the Colburn analogy

\[ h_c = - \frac{C_f}{2} \frac{\rho U_{eff} C_p Pr}{\mu} \]  

\[ \text{...(2.3.4)} \]

where: \( U_{eff} \) is the effective gas velocity, \( \rho \) is the density and \( C_f \) is the skin friction coefficient.

In eq. 2.3.4 the effective gas velocity is determined by

\[ U_{eff} = \left( u_x^2 + u_y^2 + 2 K_{en} \right)^{1/2} \]

where \( u_x, u_y \) are the velocity components parallel to the surface under consideration, \( K_{en} \) is the kinetic energy of turbulence and \( C_f \) is obtained from

\[ C_f = 0.0565 \left( \frac{p U_{eff} \delta}{\mu} \right)^{-1/4} \]
based on the correlations obtained for a flat plate boundary layers
and fully developed pipe flows.

The determination of $h_c$ from eq. 2.3.4 requires a knowledge of the
three mean velocity components, $K_{en}$ and the thickness of the boundary
layer, $\delta$. Moreover the effect of boundary layer thickness is weakly
emphasized which is not necessarily true as discussed earlier in
section 2.2 because the boundary layer itself is a major contribution
towards the thermal resistance at the wall.

Thus a model in which the boundary layer thickness plays an important
role and also takes into account the effect of geometry of the
surface is needed.

2.3.1 Heat transfer modelling for a flat plate

The model developed for the convective heat transfer study is based
on the calculation of thermal boundary thickness from the temperature
profile. Initially the model will be developed for a flat plate which
will be modified later to account for the change in surface geometry
by the use of roughness elements.

The heat flux $q_w$ at the wall is given by the fourier flux which can
be expressed as

$$q_w = - \rho C_p a \frac{dt}{dy}$$  ...(2.3.5)

where $a$ is the effective thermal diffusivity
$C_p$ is the specific heat at constant pressure of the fluid
is the gas density
$dt/dy$ is the temperature gradient normal to the wall

Assuming $q_w$ to be one dimensional the integral of eq. 2.3.5 across
the thermal boundary layer of thickness $\delta_{th}$ [defined as the thickness
of the layer at which the temperature in the boundary layer is 99% of
the free stream temperature $T_o$] can be written as

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\[ T_{\infty} - T_w = q_w \int_0^{\delta_{th}} \frac{1}{\rho c_p a} \, dy \]  

...(2.3.6)

where \( T_w \) is the wall temperature.

Eq. 2.3.6 can be written in the form

\[ q_w = K_{\text{eff}} (T_{\infty} - T_w)/\delta_{th} \]  

...(2.3.7)

where: \( K_{\text{eff}} \) is the effective thermal conductivity equal to

\[ K_{\text{eff}} = \frac{\delta_{th}}{\int_0^{\delta_{th}} \frac{1}{\rho c_p a} \, dy} \]  

...(2.3.8)

Now by definition of \( a \) which contains both the laminar and turbulent contributions we have

\[ a = v + v_T = \frac{v}{Pr} + \frac{v_T}{Pr} (1 + \frac{Pr}{Prt} v_T) \]  

...(2.3.9)

where: \( v, v_T \) are the laminar and turbulent viscosity

\( Pr \) is the Prandtl number, and

\( Prt \) is the turbulent Prandtl number

By definition of \( v_T \)

\[ v_T = K_{\text{en}}^{1/2} \]  

...(2.3.10)

and substituting for \( v_T \) in eq. 2.3.9 we have

\[ a = \frac{v}{Pr} [1 + \frac{Pr}{Prt} \frac{(K_{\text{en}})^{1/2} \lambda}{v}] \]  

...(2.3.11)

where: \( K_{\text{en}} \) is the kinetic energy of turbulence and \( \lambda \) is the mixing length in the boundary layer,
Since the "law of the wall" is assumed to be valid it can be said that the flow is one dimensional. In one dimensional flows, according to Prandtl's mixing length hypothesis, the turbulent kinetic energy dissipated at the wall is being supplied by the diffusion process. Assuming a linear variation of the length scale normal to the wall and in addition supposing its validity for a boundary layer in an engine the effective viscosity and hence the thermal diffusivity can be written as:

\[ \alpha = \frac{v}{Pr} \left[ 1 + Pr. \frac{\delta_{th}^{3/2}}{v} \left( \frac{K_\infty}{\delta_{th}} \right)^{1/2} \right] \frac{\delta_{th}^{3/2}}{v} \delta_{th}^{3/2} \]  \quad \ldots(2.3.12)

Therefore the effective thermal conductivity can now be expressed as:

\[ K_{eff} = \frac{K_1}{\int_0^1 (1 + C \delta_{th}^n)^{-1} \, dz} \]  \quad \ldots(2.3.13)

where:

- \( Z = \frac{v}{\delta_{th}} \) and \( n = \frac{3}{2} \)

and

- \( C_1 = \delta_{th}^{3/2} \sqrt{\frac{\rho_{\infty}}{K_\infty}} \cdot \frac{Pr}{v} \)

where \( K_1 \) is the laminar thermal conductivity.

The integral in the denominator of eq. 2.3.13 can be evaluated from numerical integration such as Simpson's one-third rule.

2.3.2 Heat transfer modelling for a roughened surface
The first effect of roughness, as discussed earlier, is to destabilise the viscous sublayer which results in an effectively thinner sublayer. Depending on the flow velocity and for \( h^+ > 30 \), i.e. in the completely rough regime the sublayer disappears entirely which in effect reduces the thermal resistance at the wall resulting in increased heat transfer.
Therefore the expression for effective thermal diffusivity from eq. 2.3.12 can now be written down as:

$$a = \frac{\delta_{th}}{0} \frac{1}{\int_{\infty}^{\infty} y^{3/2}} \cdot \frac{dy}{y^{3/2}} = \sqrt{\frac{\delta_{th}}{K_{\infty}}} \int_{0}^{\infty} \frac{dy}{y^{3/2}}$$

$$= \sqrt{\frac{1}{K_{\infty}}} \frac{2}{\delta_{th}^{1/2}} \quad \text{...(2.3.14)}$$

Thus the expression for effective thermal conductivity is now reduced to:

$$K_{\text{eff}} = \frac{1}{2} \rho C_p \delta_{th}^{3/2} \sqrt{K_{\infty} \delta_{th}} \quad \text{...(2.3.15)}$$

Hence the wall heat flux for any surface can now be evaluated from eq. 2.3.5 provided the value of $K_{\text{eff}}$ is known from eq. 2.3.13 or 2.3.15. Evaluation of $K_{\text{eff}}$ in eqs. 2.3.13 and 2.3.15 requires the variable $\delta_{th}$, to be known, which can be evaluated from the temperature profiles described in Section 2.2 represented by eqs. 2.2.2 and 2.2.4, which reduces to the following equations on rearrangement:

a) For a smooth wall:

$$\delta_{th} = \frac{\delta_{th}}{0.99 T_{\infty}} = \frac{\delta_{th}}{u^*} \exp \left[ \frac{\rho C_p u^* (0.99 T_{\infty} - T_w) \alpha_w - G \beta}{A_{hs}} \right] \quad \text{...(2.3.16a)}$$

b) For a rough wall:

$$\delta_{th} = \frac{\delta_{th}}{0.99 T_{\infty}} = \frac{\delta_{th}}{u^*} \exp \left[ \frac{\rho C_p u^* (0.99 T_{\infty} - T_w) \alpha_w - G(h^+)}{A_{trh}} \right] \quad \text{...(2.3.16b)}$$

Thus for the evaluation of heat flux, $q_w$, an iteration procedure becomes imperative since $q_w$ depends on $\delta_{th}$ which in turn depends on
For fast convergence the algorithm due to Wegstein [81] is employed for the solution of the above equations to yield the value of $q_w$ from an assumed value of wall heat flux $q_w$. The above iteration is carried out until the error limit is less than 2%.

2.4 Thermal Network for Heat Flow

As mentioned earlier, the presence of a rough surface increases heat transfer due to higher heat transfer coefficient and higher surface area. Since the piston is chosen to have the ribbed surface it is subjected to high temperature because of heat flowing through it to the coolants. Hence it becomes necessary to model the flow of heat through the engine head, cylinder wall and the piston in particular from the known distribution of wall heat flux. Evaluation of local temperatures in the engine head and piston requires the need for finite element technique which is prohibitive in terms of development time and costs. So the following analysis of heat flow will be based on the area averaged wall temperature and assuming the transient temperature fluctuation at the surface is negligible, the conduction of heat can be considered to be in steady state.

a) Gas-head coolant [Fig. 2.3]

The engine head can be taken as a slab of constant thermal conductivity, $K_h$, and of thickness $t_h$, and by assuming the slab extends to infinity in other dimensions, the heat flow in the region of consideration becomes one-dimensional. Taking the distribution of heat transfer coefficient on the gas and coolant sides as $h_{ch}$ and $h_w$ respectively [Fig. 2.3] the thermal resistance, $Z_{head}$, can be evaluated by:

$$Z_{head} = \frac{1}{h_{ch}\left(\frac{\pi D^2}{4}\right)} + \frac{t_h}{K_h\left(\frac{\pi D^2}{4}\right)} + \frac{1}{h_w\left(\frac{\pi D^2}{4}\right)} \quad \ldots (2.4.1)$$
b) Gas-sleeve-coolant [Fig. 2.3]
Treating the heat flow through the sleeve by the same assumptions as for the engine head, the thermal resistance $Z_{slc}$ can be evaluated by

$$Z_{slc} = \frac{1}{h_c} \left( \frac{\ln \left( \frac{D/2 + t_c}{D/2} \right)}{\pi D t_{ip}} + \frac{1}{2\pi t_c t_{ip}} + \frac{1}{h_w (2\pi (D/2 + t_c))^2 t_{ip}} \right) \quad \ldots (2.4.2)$$

where $h_c$ is the heat transfer coefficient on the gas side.

c) Gas-piston-coolant
i) Gas-piston [Fig. 2.4]

$$Z_{gp} = \frac{1}{h_{cp}} \left\{ \frac{n_{\text{ribs}}}{R_{\text{rib}}} + \frac{1}{h_{cp} (A_{pist} - A_{\text{rib}})} \right\} t_p \quad \ldots (2.4.3)$$

where:
- $n_{\text{ribs}}$ is the number of ribs on the surface of the piston
- $h_{cp}$ is the heat transfer coefficient at the piston surface
- $K_p$ is the thermal conductivity of piston material
- $A_{pist}$ is the piston cross-sectional area
- $A_{\text{rib}}$ is the area occupied by the ribs
- $R_{\text{rib}}$ is the resistance of the rib given by [Fig. 2.4(d)].

$$R_{\text{rib}} = \frac{\left( R_3 + \frac{R_a R_5}{R_5 + R_a} \right)}{(R_3 + \frac{R_a R_5}{R_a + R_5}) + R_6} \cdot R_6 \quad \ldots (2.4.4)$$

The individual resistances are given by:

$$R_1 = \frac{h/3}{K_p \cdot A_{\text{rib}}} + \frac{1}{h_{cp} (h/3)S}$$
where: \( A_{\text{rib}} \) is cross-sectional area of the rib = \( b \times l \)
\( S \) is the perimeter of the rib = 2 (\( b \times l \))
and \( h \) is the height of the rib.

ii) Piston-Oil

Assuming the heat transfer coefficient on the oil side to be \( h_{\text{oil}} \), underneath the piston crown of area \( A_{\text{oil}} \)

\[ Z_{\text{oil}} = \frac{1}{h_{\text{oil}} A_{\text{oil}}} \]  
\[ \ldots (2.4.5) \]

iii) Heat transfer through the ring: [Fig. 2.5]

Resistance of the oil layer between piston and ring
\[ = \frac{\delta_r/2}{\pi D r K_{\text{oil}}} \]
Resistance of the piston intervals
\[ = \frac{L}{\pi D r K_p} \]
Resistance of the ring
\[ = \frac{1}{\pi D r K_r} \]
Resistance of the oil layer between sleeve and ring
\[ = \frac{\delta_w}{\pi D s K_{\text{oil}}} \]

Therefore
\[ Z_{\text{ring}} = \frac{1}{\pi D} [\frac{\delta_r/2}{2 a b s K_{\text{oil}}} + \frac{1}{K_{\text{ring}}} + \frac{\delta_w}{K_{\text{oil}} b_s}] \]
\[ \ldots (2.4.6) \]

where
\[ a = \frac{1}{[\frac{2L}{\delta_r} K_{\text{oil}} / K_p + 1.0]} \]
iv) Heat transfer through the piston skirt:

The piston skirt can for calculation purposes be regarded as a hollow cylinder without pin bosses and it can be assumed that the temperature changes along the skirt length $L_s$ and not over the individual cross-sections because the piston skirt is thin in comparison to its length. Thus for a quantity of heat $Q_o$ transferred along the top edge of the skirt at a temperature $T_o$ [Fig. 2.6] and by thermal balance at a distance $x$ over a section $dx$, we obtain:

$$Q' - Q'' = dQ$$  \hspace{1cm} ... (2.4.7)

From Fourier law

$$Q' - Q'' = \frac{dQ}{dx} = K_p A_s \frac{dT}{dx^2}$$  \hspace{1cm} ... (2.4.8)

where $A_s = \frac{\pi}{4} [D^2 - (D - 2t_s)^2]$

Assuming a heat transfer coefficient $h_o$ between skirt and sleeve through the oil film, we have

$$dQ = h_o (\pi D dx).T$$  \hspace{1cm} ... (2.4.9)

Hence from the above last two equations 2.4.8 and 2.4.9

$$\frac{d^2T}{dn^2} = \frac{h_o \pi D}{K_p A_s} = m^2 T$$  \hspace{1cm} ... (2.4.10)

and from the following boundary conditions:
i) at \( x = 0 \), \( T - T_0 \)

ii) at \( x = L_s \), \( \frac{dT}{dx} = 0 \) by assuming the quantity of heat transferred at the lower end of the skirt is negligible.

The temperature distribution along the skirt is given by:

\[
T = T_0 - \frac{\frac{m(L_s - x)}{e^{mL_s}} + \frac{m(L_s - x)}{e^{-mL_s}}}{e^{mL_s} + e^{-mL_s}} 
\]

...(2.4.11)

The thermal resistance of the skirt \( R_s \), is then given by:

\[
R_s = \frac{1}{m K_p A_s \left[ e^{mL_s} + e^{-mL_s} \right]} 
\]

...(2.4.12)

Thus the network for the gas piston coolant chain assuming the number of ribs as \( n_{ribs} \) and number of rings as 4 can be represented as shown in Figure 2.7.

The thermal resistance \( R_{Li} \) to \( R_{L4} \) can be evaluated from

\[
R_{Li} = \frac{L_i}{K_p A_i} 
\]

...(2.4.15)

where \( A_i \) is the horizontal cross-section of the \( i \)th ring interval.

Thus the total impedance network for heat flow through the walls of the combustion chamber can be reduced to as shown in Fig. 2.8 and the temperature of the piston can be evaluated once the impedances are known and is given by:

\[
T_p = \frac{T_w/Z_{ps} + T_{oil}/Z_{oil} + T_{gas}/Z_{gp}}{(1 + 1 + 1)} 
\]

\[
= \frac{T_{ps} + T_{oil} + T_{gp}}{Z_{ps} \ Z_{oil} \ Z_{gp}} 
\]

...(2.4.16)
where $T_{\text{gas}}$ is taken as the average temperature of the contents in the cylinder.

Therefore heat lost from the system $Q_{\text{tot}}$ is given by

$$Q_{\text{tot}} = \frac{T_{\text{gas}} - T_{\text{w}}}{Z_{\text{head}}} + \frac{T_{\text{gas}} - T_{\text{p}}}{Z_{\text{gp}}} + \frac{T_{\text{gas}} - T_{\text{w}}}{Z_{\text{sle}}} \quad \ldots (2.4.17)$$

2.5 Modelling of Combustion Process in the Spark-Ignition Engine

The simulation of the combustion process in a Spark-Ignition engine can be approached by two different modelling strategies:

A) zero dimensional
B) multi dimensional

A) Zero dimensional:

The models in this category of simulation are derived by considering the propagating flame front at any instant to subdivide the combustion chamber into two regions - one of burned gas behind the propagating flame front and the other of unburned gas ahead of the flame front which is considered to be thin and of spherical form centered on the spark plug. The equations resulting from the overall balances of mass and energy applied to each region and additional relations for thermophysical properties of the fluids in these regions form a closed set of simultaneous ordinary differential equations whose sole independent variable is time: hence the name zero dimensional model. This modelling approach still requires a method to calculate burn rate in spark ignition engines which are:

(a) Functional relationship

Prescribing the variation of mass fraction burned by a functional relationship where the start of combustion, $\theta_{\text{st}}$, the combustion duration, $\theta_{\text{cd}}$, and the current crank angle $\theta$ are related. Examples of such burning laws are:
i) Cosine burn law

\[
\frac{dx_b}{d\theta} = \frac{\pi}{2\Delta \theta_c} \sin \left[ \pi \left( \frac{\theta - \theta_{st}}{\theta_{cd}} \right) \right] \quad \ldots (2.5.1)
\]

ii) Wiebe function

\[
\frac{dx_b}{d\theta} = a(m+1) \left( \frac{\theta - \theta_{st}}{\theta_{cd}} \right)^m \exp \left[-a \left( \frac{\theta - \theta_{st}}{\theta_{cd}} \right)^{m+1} \right] \quad \ldots (2.5.2)
\]

where \( x_b \) is the mass fraction burned, \( a \) is an efficiency parameter and \( m \) is a slope parameter.

The major drawbacks to predict burn rate using one of the above functions is 'a priori' specification of the burn rate itself which cannot be called predictive. This led to the development of burn rate modelling from more fundamental physical quantities such as: the turbulence intensity \( u' \), the integral length scale and the turbulent microscale.

b) Eddy entrainment concept

One such approach is the 'eddy entrainment concept' suggested by Blizard and Keck /84/. In this concept the flame propagation as first is considered to be a turbulent entrainment of unburned mixture into the front, which is followed by a laminar burn up process with a characteristic length scale. Thus the mass entrained into the front, \( x_e \), is given by

\[
\frac{dx_e}{dt} = \rho_l A_f u_e \quad \ldots (2.5.3)
\]

where \( u_e \) is the entrainment speed assumed to be proportional to the turbulent intensity. The rate at which the mass is burned is given by

\[
\frac{dx_b}{dt} = \frac{x_e - x_b}{t_r} \quad \ldots (2.5.4)
\]

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where $t_r$ is the characteristic reaction time to burn the mass of an eddy of size '1'.

Tabacyznski et al. /85/ extended the above eddy entrainment concept by representing the flame propagation process into a "developing" turbulent flame - the burning of an individual eddy, and a "fully developed" turbulent flame - the entrainment and burning of many eddies. The basic assumptions of this model are:

i) Turbulent eddies have a structure [see Fig 2.9] that is described by the integral scale $L$, microscale $\lambda$, and kolmogorov scale $\eta$.

ii) Combustion on the kolmogorov scale is instantaneous.

iii) Ignition sites in an individual eddy propagate at a rate governed by the sum of the turbulent intensity, $u'$, and laminar flame speed $S_l$.

iv) Laminar burning takes place over the microscale.

v) The integral scale is proportional to the chamber height at the time of spark.

vi) The turbulent intensity at the time of spark is proportional to engine speed.

vii) Turbulent intensity and integral scale change with unburned mixture density, as defined by conservation of angular momentum.

Although this formulation of burn rate prediction has been successfully applied [Tabacyznski et al. /85/] to predict burn rate and cylinder pressure, it incorporates many mechanisms to predict observed heat release behaviour in engines such as the description of the flame structure and the behaviour of the turbulent field which are yet to be experimentally verified.
c) **Turbulent flame speed concept**

Another approach to predict burn rate has been to model the burning process as a flame front of area $A_f$ propagating through the unburned mixture at the turbulent flame speed $S_{tu}$. Thus the rate of mass burning is

$$\frac{dx_d}{dt} = \rho_u A_f S_{tu} \quad \text{...(2.5.5)}$$

where $\rho_u$ is the unburned gas density.

Turbulent flame speed $S_{tu}$ can then be related to laminar flame speed $S_l$ by the expression [see Fig 2.10]

$$S_{tu}/S_l = 1 + 4.01u'/S_l \quad \text{...(2.5.6)}$$

where $u'$ is the turbulent intensity. The ratio $S_{tu}/S_l$ in eq. 2.5.6, also known as the flame speed ratio (FSR), shows that the turbulent flame speed is more important than laminar flame speed in the determination of the burning velocity. This modelling approach for burn rate prediction has been successfully applied to predict burn rate and flame speed in engines by Mattavi et al. /86/ which will be adopted in this study. The above correlation, eq. 2.5.6, has been compared with measured turbulent flame speed by Mattavi et al. /86/.

In spite of the excellent agreement with the experiments one of the shortcomings is that, as the laminar flame speed approaches zero, the formulation [eq. 2.5.6] suggests that the turbulent flame speed, $S_{tu}$, can have a finite value, even at the lean limit of flame propagation, if the turbulent intensity, $u'$, is sufficiently large which is not true. Since the limit of flame propagation is given by the laminar flame speed the influence of turbulence on flame propagation as given by eq. 2.5.6 breaks down at the limit of flame propagation. That is, turbulence never widens the limit of flame propagation but narrows it down as discussed earlier in section 1.1.
B) Multi dimensional models
This class of models to simulate combustion in an engine are usually performed by providing numerical solution of the governing partial differential conservation equations of mass, momentum, energy and individual chemical species. In order to apply a digital computer to the solution of a continuum problem, the continuum is represented by a finite number of discrete elements. The most common method of discretization is the use of finite differences in which the region of interest (combustion chamber) is divided into a number of small zones or cells. These cells form a grid or mesh which serves as a framework for constructing finite-differences approximations to the governing partial differential conservation equations.

The above solution procedure still requires appropriate submodels:
   i) to represent curved and/or moving boundaries.
   ii) explicit or implicit numerical iteration schemes to maintain stability because of confined subsonic flows inside the chamber.
   iii) turbulence model to predict length scales, diffusivities etc.
   iv) flame structure, its propagation rate and resolution in the chamber which not only depend on turbulent diffusivities but also on chemical kinetics.
   v) chemical kinetics with appropriate rate coefficients for chemical reactions.

There are merits and demerits in both the above two types of models:
Zero dimensional models also known as "Thermodynamic" or "phenomenological" class of engine combustion simulations have proven to be the most useful because of their relative simplicity and computational efficiency but they are limited as to the prediction of the details of the fluid flow in the combustion chamber and their influence on flame propagation.
Multidimensional models on the other hand have the capability to provide and predict details of the fluid flow and changes in chamber geometry. But they are very susceptible to the type of mesh that is used to describe the chamber geometry viz. fine mesh or coarse mesh and the chemical rate constants in chemical reactions. In addition computer time and storage constraints together with the uncertainty in various physical input parameters [boundary conditions] and an inadequate level of sophistication in turbulence and reaction rate models are some of the major disadvantages as opposed to phenomenological models.

Hence given the choice the "thermodynamic" or "phenomenological" models are best suited to investigate the effects of varying engine parameters such as engine speed, equivalence ratio and compression ratio. In addition the computational time taken for the above parametric studies are far less than when multidimensional models are used.

2.5.1 Thermodynamic combustion model adopted in this study

The basic premise of the combustion model is that the flame propagates through the unburned mixture at the turbulent flame speed, $S_{tu}$, and the rate of burning as described earlier is defined by

$$\frac{dx_b}{dt} = \delta A_f S_{tu} \quad ...(2.5.5)$$

The flame front area is evaluated from the method described in Appendix B.

Since the flame propagation process is affected due to higher turbulence in the flow caused by the obstacles present in the chamber makes eq. 2.5.5 an ideal choice to model burn rate in engines incorporating obstacles to accelerate the flame. The increase in turbulent intensity can then be related to the obstacle configuration by eq 1.2
from which the turbulent flame speed can be evaluated from eq. 2.5.6 for the evaluation of burn rate from eq. 2.5.5. The laminar flame speed, $S_L$, is evaluated from Van Tiggelen & Deckers /87/ model and is described in Appendix C.

Further to the above concept of predicting burn rate the thermodynamic model proposed by Lavoie et.al. /88/ modified to incorporate time dependent specific heats is adopted to model combustion process based on the following assumptions:

a) Charge prior to spark is assumed to be a homogenous charge of air, fuel and exhaust gas residual. Compression of the charge is assumed to be isentropic with an index of 1.3.

b) During combustion the gases in the cylinder are divided into two zones, separated by a thin flame front, one corresponding to the burnt gas and the other to the unburnt gases. [see Fig 2.11]

c) The two zones are at the same uniform cylinder pressure $p$.

d) Each zone behaves according to the ideal gas law.

e) Each zone is in thermodynamic equilibrium.

f) The gas temperature in their respective zones are uniform.

g) Heat transfer is confined to convective exchange of heat between zone and the walls.

From the above assumptions, by conservation of volume
\[
\frac{V}{m} = x_b v_b + x_u v_u = x_b v_b + (1-x_b)v_u \quad \ldots(2.5.7)
\]

and by conservation of energy
\[
\frac{E}{m} = x_b e_b + x_u e_u = x_b e_b + (1-x_b)e_u \quad \ldots(2.5.8)
\]

where \( V \) is the cylinder volume, \( E \) is the energy of the gases in the cylinder, \( e \) & \( v \) are spatial average internal energy and specific volume, \( x \) is the mass fraction and the subscripts \( b \) and \( u \) refer to the burned and unburned gases respectively.

The thermodynamic state functions for the burnt gas
\[
e_b = e_b(T_b,p) \quad \ldots(2.5.9)
\]
\[
v_b = v_b(T_b,p) \quad \ldots(2.5.10)
\]

are computed from the model of Martin & Heywood /89/ and similarly for the unburnt gas
\[
e_u = e_u(T_u,p) \quad \ldots(2.5.11)
\]
\[
v_u = v_u(T_u,p) \quad \ldots(2.5.12)
\]

are evaluated from Ferguson et.al. /90/. Further to the above assumptions another convenient approximation is to assume that the unburned gas is compressed isentropically. The unburned gas temperature is then a function of pressure alone
\[
T_u = T_u(p) \quad \ldots(2.5.13)
\]

By applying the first law of thermodynamics for the system defined by the cylinder
\[
\dot{E} = -p\dot{V} - \dot{Q} \quad \ldots(2.5.14)
\]

the heat loss \( \dot{Q} \) can be evaluated either from eq. 2.4.17 or from Woschni's formula /76/ assuming a constant wall temperature.
2.5.2 Solution technique

The rate of burning defined by eq. 2.5.5 and the first law of represented by eq. 2.5.14 form a closed set of first order of ordinary differential equations that need to be integrated simultaneously. The derivative evaluations at each time step requires the solution of equations 2.5.7 through 2.5.12 to determine the thermodynamic state of the system which are solved by Newton-Raphson technique. The integration method proposed by Krough /91/ and coded by Ferguson et al. /90/ is used to solve the above set of differential equations.

The above combustion model has been successfully used to predict cylinder pressures for various engines such as Ricardo E6, Ford Standard and May engines by evaluating heat loss from Woschni's correlation /76/ assuming a constant wall temperature of 450 K.

The flow chart of the above combustion model along with the models developed in sections 2.2, 2.3 and 2.4 is shown in Fig 2.12.

2.6 Validation of the Combustion, Heat Transfer and Thermal Network Models

Simulation of the above combustion model along with the convective heat transfer and thermal network models was carried out for single cylinder Ricardo E6 engine for the pancake chamber geometry shown in Fig 2.13. For convective heat transfer calculations a total of 14 stations was assumed - 6 stations each on piston and cylinder head and two stations on the sleeve. The input conditional parameters for the simulation are:
(i) Speed = 1500 RPM  
(ii) Air-Fuel Ratio = 15.9  
(iii) Ignition Timing = 20 deg BTDC  
(iv) Initial pressure at BDC = 0.98 bars  
(v) Compression ratio = 10:1  
(vi) Initial u' = 0.1 m/s at spark

The results of the simulation performed for the above input conditions are shown in Figs 2.14 to 2.15 which indicate:

a) The averaged heat flux of all the 14 stations evaluated by thermal boundary layer is quite close to the heat flux evaluated from Woschni's correlation [Fig 2.14].

b) The average piston temperature [Fig 2.15] evaluated from thermal network by assuming Woschni's correlation and from thermal boundary layer concept are 455 K and 485 K respectively. The transient piston temperature variation evaluated from thermal boundary layer is higher than that evaluated from Woschni's correlation.

In the simulation of the above model the effect of ribs has been switched off.

2.7 Optimization Analysis
As discussed earlier it is necessary to evaluate which type and configuration of the rough surface is to be employed for flame acceleration to enhance burn rate via turbulence generation. But it is also to be noted that attempts to increase turbulence results in undesirable phenomenon of increased heat loss to the cold walls thereby reducing thermal efficiency.

Burn rate enhancement results in increased output (work done) from the engine and increased heat loss can be evaluated from eq. 2.4.17.
Hence the ratio of integrated heat loss to integrated work done (i.e. $\int \frac{dQ}{dW}$) represents the trade-off that can be gained as useful output from the system.

Thus it can be said that the performance $P$ of a rough surface is a function of the parameter $\int \frac{dQ}{dW}$.

$$P(\text{surface}) = f(\int \frac{dQ}{dW})$$

which is to be kept at a minimum in order to obtain effective gain from the system.

This leads to the idea of optimization from which for a given geometry of the rough surface and in conjunction with the above combustion model the parameter $\int \frac{dQ}{dW}$ can be considered to represent the performance of such a surface.

2.7.1 Selection of type of obstacle

As seen earlier there are large variety of obstacle shapes available that could be used to generate turbulence viz., rectangular, diamond, parallelogram, hexagonal, cylindrical, conical, spherical, hemispherical etc., These obstacles are subject to high pressure and temperature inside the combustion chamber. Though they generate turbulence which is necessary to accelerate the flame, they also cause unequal distribution of heat causing thermal stress. Moreover there is a possibility of obstacles acting as hot spots especially with obstacles having thin cross-sectional areas such as diamond, parallelogram conical and hexagonal shapes, which is detrimental to engine performance. Ease of fabrication eliminates the possibility of using hemispherical, spherical or cylindrical shaped obstacles. Thus the only obstacle shape that could be used without much of the above constraints is rectangular shaped obstacle having square cross-section.
2.7.2 Selection of size of obstacle
Geometry of a surface roughened by three dimensional rectangular ribs can be varied by various parameters such as their length, $e$, lateral gap, $g$, width, $b$, and their pitch, $p$, [Fig 2.1].

Since it is known that a repetition of obstacles is needed to maintain flame acceleration and by assuming that at least 2-6 ribs are necessary in the first row to distort the flame constrains us to choose at least one geometrical parameter of the obstacle. By choosing the length of the rib, $e$, as 1 cm: This allows the lateral parameter $g/e$ to be varied from 0.25 for six ribs in the first row to a value greater than 4 for two ribs in the first row located at 2 cm from spark axis [Fig 2.16].

2.7.3 Location of first row
Allowing for ignition delay and for the flame kernel to develop from spark the first row of rib can conveniently be placed at a distance of 2 cm from the spark axis so that by the time the spherical flame hits the first row of ribs the piston is quite close to top dead center thereby accelerating the flame due to higher blockage ratio as reviewed earlier in section 1.2.

2.7.4 Minimum number of rows
From the work of Moen et.al /44/ it is evident that a repetition of obstacles is needed to maintain flame acceleration by the positive feedback mechanism. Thus assuming a minimum of four number of rows of ribs a typical configuration of the rough surface would be as shown in Fig 2.17 with a pitch of 1 cm between consecutive rows of ribs.

2.7.5 Rib arrangements and layout
Based on intuition and fluid flow fundamentals the following rules can be established:

(i) To check whether there is a wall (cylinder sleeve) or free space available ahead of the flame for the flame to accelerate further. If there is availability of free
space the obstacles in that particular row are kept close enough so that the lateral parameter g/e lies between 0.8 and 1.2.

(ii) The obstacles in a particular row do not lie in the dead water region of the obstacles situated in the immediate upstream.

2.7.6 Geometrical parameter variation study

Based on the above considerations a parametric study by modelling was carried out keeping in mind that the configuration of the rough surface used for simulation is a physical reality. The parameter varied in the simulation of the model was the pitch between the rows of ribs and was varied from 10 mm to 5 mm in steps of 1 mm. Thus the surface geometries of the piston for which the simulation was carried out are shown in Fig 2.19. The arrangement and layout of the obstacles are based on the rules given above in section 2.7.5. The conditions of the simulation are:

(i) Speed = 2500 RPM
(ii) Air-Fuel Ratio = 15.6
(iii) Ignition Timing = 30 deg BTDC
(iv) Initial pressure at BDC = 0.9 bars
(v) Compression ratio = 10:1
(vi) Initial u' = 0.15 m/s at spark

The optimization parameter, Q/W, evaluated from the combustion model for the above input parameters in conjunction with the thermal boundary layer and thermal network models for the piston geometries in Fig 2.18 is shown in Fig 2.19.

The information in Fig 2.19 was derived from a consideration of the change in work done (dW) and heat transfer (dQ) from the instant of spark to exhaust valve opening as shown in Fig 2.20. To obtain Fig 2.19 the ratio of \( \int_{\text{spark}}^{\theta} dQ \) to \( \int_{\text{spark}}^{\theta} dW \) is obtained from Fig 2.20. The plots of \( \int_{\text{spark}}^{\theta} dQ \) and \( \int_{\text{spark}}^{\theta} dW \) are shown in Fig 2.21.
2.7.7 Analysis of computational output

Analysis of heat loss to work done ratio [Fig 2.19] indicate the performance of geometrical configurations shown in Fig 2.18 from which the following conclusions are drawn:

a) Geometries N13-1 & N13-2 and N16-1 & N16-2 have the lowest Q/W which indicate that the optimum pitch could be around 7 or 6 mm each having two different types of configurations.

b) Other geometries N10, N12, N18 & N21 appear that their performance to be poor in relative to N13/N16 geometries.

2.8 Critical Evaluation of the Model

The combustion model used for optimization analysis belongs to the "phenomenological" or "thermodynamic" class of models. Since these class of models are unable to provide the intricate details of the fluid flow, the only parameter that has to be used to evaluate the performance of different types and configurations of rough surfaces is through the evaluation of heat loss to work done ratio. Hence the model is unable to provide the actual design, layout and the arrangement of the obstacles of the rough surfaces. This is a major reason as to why there is no noticeable difference in the optimization parameter Q/W for the surfaces N13 and N16 each of which has two different types of configurations [see Figs 2.18]. Based on the rules given in section 2.7.5 the combustion model is still unable to distinguish between the two types of configurations that are possible for the geometries N13 and N16. Hence there is a need for a different mode of diagnostics to identify which type of configuration would give a better performance.
Fig 2.1: Schematic Representation of Flow Past a Three-Dimensional Roughened Surface
Fig 2.2: Evaluation of Width Parameter, $b$, for Three-Dimensional Roughened Surface

(a) $b_1 = b$

(b) $b_1 = b$

(c) $b_1 \neq b$

(d) $b_1 \neq b$
D = bore diameter

t_c, t_h = thickness of cylinder sleeve and head

K_c, K_h = thermal conductivity of material for cylinder sleeve and head

h_w = heat transfer coefficient on coolant side

h_cc, h_ch = heat transfer coefficient on gas side for cylinder sleeve and head

\( \text{l}_{ip} \) = instantaneous position of the piston

Fig 2.3: Heat Flow to Coolants From Gas
Fig 2.4: Heat Flow Through Cylinder Head, Sleeve and Piston - (b) Through Rib (c) Through Ring
Fig 2.5: Heat Transfer Through Piston Ring
Shaded area = Piston area - area occupied by the ribs

\[
\text{Shaded area} = A_{\text{pist}} - A_{\text{rib}} \times n_{\text{ribs}}
\]

- \(h,b,l\) = height, width and length of the rib
- \(D_{\text{oil}}\) = diameter underneath the piston crown cooled by oil
- \(n_{\text{ribs}}\) = number of ribs
- \(D\) = bore diameter
- \(k_p\) = thermal conductivity of piston material
- \(L_i - L_4\) = lengths of the ring intervals
- \(L_s\) = piston skirt length
- \(t_s\) = thickness of piston skirt

Fig 2.6: Heat Transfer Through Piston Skirt
Fig 2.7: Gas - Piston - Coolant Chain
Fig 2.8: Total Impedance Thermal Network
Fig 2.9: Turbulent Eddy Structure Concept
by Tabaczynski et. al. /85/
Fig 2.10: Correlation of Flame Speed Ratio, $S_{tu}/S_1$, to $u'/S_1$ [Ref /86/ and also Symbols]
Fig. 2.11: Schematic Representation of Thermodynamic Combustion Model
Fig 2.12 Flow chart of the Thermodynamic Combustion Model
Fig 2.13 Schematic Diagram of the Pancake Geometry Indicating Stations used for Convective Heat Transfer Modelling
Fig 2.14: Comparison of Heat Flux Evaluated from Woshni's Correlation and from Thermal Boundary Layer Averaged over 14 Stations for the Pancake Geometry Shown in Fig 2.13
Fig 2.15: Piston Temperature Evaluated from Heat Flux by (a) Woschni's Correlation & (b) Thermal Boundary layer - for the Pancake Geometry Shown in Fig 2.13
Fig 2.16: Plan View of Piston Showing The Location of The First Row
Fig 2.17 Typical Configuration of a Roughened Surface for a Piston Diameter of 76.2 mm
**Fig 2.18:** Schematic Sketches of the Possible Piston Surface Geometries Simulated in the Optimization Analysis. \( R_1 \) is the Minimum Distance for the Location of First Row Which is 20 mm Away from the Spark Axis.
Fig 2.18: (contd) A is the Location of the Spark Plug Axis, S From the Wall (11 mm), P is the Pitch Indicated by the Value Given in Each Geometry. The Details of the Dimension of the Obstacle is Also Shown.
Fig 2.19 Optimization Parameter Q/W as a Function of Crank Angle for the Geometries Shown in Fig 2.18
FIGURE 2.20 Variation of dQ and dw as a function of crank angle
FIGURE 2.21 Variation of $\int dq$ and $\int dw$ as a function of crank angle.
CHAPTER 3
CONSTANT VOLUME COMBUSTION EXPERIMENTS

3.1 Introduction

In chapter 2 optimization study was carried out using a phenomenological model. These class of models give a global view of the changes that would occur from a macroscopic point of view, viz., such as the number of ribs and the pitch between rows of ribs for a given surface geometry via the optimization parameter Q/W. The phenomenological model is unable to tell the difference that would occur in engine performance between the two types of configurations that are possible for the geometries N13 and N16. In order to select the best surface, there is a need for constant volume combustion bomb study.

This chapter describes the experimental work and analysis carried out on constant volume combustion bomb experiments in quiescent chambers for the rough surfaces modelled in section 2.7 to obtain pressure data and also photographic work to evaluate flame speed & observe flame development. It is divided into four main sections, the first section dealing with the experiments in a flat bomb and the second section dealing with the experiments in a cylindrical bomb having flat and divided chamber heads. In the third section a novel method of evaluating flame area by a photometric method is given. A closure to the constant volume bomb experiments and a selection of chambers for engine tests are given in the fourth section.

The chamber pressure in the bomb was measured using a Kistler 6121-A1 piezo electric transducer connected to a Kistler 5001 type charge amplifier. The procedure of calibration and the evaluation of the calibration factor is described in Appendix D.
Air fuel ratio was measured by metering the flows of air and fuel continuously to ensure the mixture strength did not change during the course of the experiments. The fuel used was commercial grade propane.

Ignition of the homogeneous mixture was carried out using a transistorized 12 V coil ignition system. Before ignition the valves were closed to ensure that there was no mass flux into or out of the combustion chamber. Between each firing test a time lapse of 3-4 minutes was allowed for the combustion products of a previous test to be flushed by the incoming fresh charge. The setup of the experimental rig is shown schematically in Fig 3.1.

3.2 Combustion experiments in Flat bomb

For ease of access for photographic study the use of a flat combustion chamber which had transparent side walls was preferred. The length of the combustion chamber was chosen to be 9 cm which typically represents the bore diameter of a commercial 2.0 litre gasoline engine. Since the flame propagation across the length of the chamber is the dominant phenomenon, the dimension of the width of the chamber is of secondary importance and was chosen to be 5.0 cm so that at least 4 ribs could be accommodated as per the simulation carried out in section 2.7, viz., surfaces such as N16-1 and N21. The chamber height, D, was chosen to represent the position of the piston at top dead center (TDC) which is of the order of 0.75 cm for a compression ratio of 10:1.

Based on the above considerations the flat bomb [Fig 3.2] was constructed from mild steel having transparent side walls for optical work and an access to the chamber to measure pressure. Since in the simulation of the model it was assumed that the flame front - advancing from the ignition source hits the first row of ribs at the same time, a line source ignition was used to initiate combustion in the flat bomb.
Based on the pitch optimization study carried out in section 2.7, eight types of rough surfaces were fabricated to suit the flat bomb and are shown in Figs 3.3 (a) - (h), including a flat surface for baseline tests. To investigate whether additional rows of ribs are necessary to accelerate the flame, another surface was fabricated [see Fig 3.3 (i)] with a obstacle configuration identical to that of the surface shown in Fig 3.3 (a) but having 4 additional rows of ribs throughout its length. The experiments were carried out at a blockage ratio of 0.4.

3.2.1 Experiments with all the surfaces at stoichiometric Air/Fuel ratio

Any appreciable effect due to change in obstacle configuration should clearly be seen in the burn rate, when combustion occurs with a stoichiometric mixture. Change in chamber pressure due to combustion was stored on storage oscilloscope. Shown in Figs 3.4 (a) - 3.4 (e) are the chamber pressures along with the corresponding surface. Tabulated in Table 3.1 below is the magnitude of peak pressure and the time taken to reach peak pressure for each surface tested. Thus from the results presented in Figs 3.4 and from the data tabulated in Table 3.1 the following conclusions are drawn:

(i) Comparing the pressure changes due to combustion in Figs 3.4 (a) & (b) for N10-ORD & N10-EXT indicate that additional rows of ribs are necessary to maintain flame acceleration without which the flame front conforms to the local flow field rapidly.

(ii) Even though the geometry of the surfaces of N13-1 and N13-2 [Figs 3.4 (d) & (e)], viz., spacing between the obstacles, height & width of the obstacles, are same pressure traces strongly suggests that the configuration of the obstacles play a dominant role in the acceleration the burn rate. The higher peak pressure reached in a shorter time in N13-2 when compared to N13-1 can be attributed to the fact that the initial flame
development is better & more distorted in N13-2, thereby increasing the burn rate resulting in faster flame acceleration due to positive feedback mechanism discussed earlier in section 1.2.

(iii) The above two conclusions that:

(a) extra rows of ribs are necessary to accelerate the flame, and

(b) the configuration plays an important role in the deformation of the initial flame development.

can again be drawn by comparing the pressure traces of N16-1 and N16-2 shown in Figs 3.4 (f) & (g).

<table>
<thead>
<tr>
<th>Surface geometry</th>
<th>Peak Pressure reached (mv)</th>
<th>Time taken to reach peak pressure from triggering pulse (ms)</th>
</tr>
</thead>
<tbody>
<tr>
<td>N10 ORD</td>
<td>951.25</td>
<td>119.0</td>
</tr>
<tr>
<td>N10 EXT</td>
<td>1092.5</td>
<td>72.8</td>
</tr>
<tr>
<td>N12</td>
<td>673.5</td>
<td>111.4</td>
</tr>
<tr>
<td>N13-1</td>
<td>792.5</td>
<td>114.5</td>
</tr>
<tr>
<td>N13-2</td>
<td>903.25</td>
<td>80.9</td>
</tr>
<tr>
<td>N16-1</td>
<td>544.0</td>
<td>211.4</td>
</tr>
<tr>
<td>N16-2</td>
<td>876.0</td>
<td>91.6</td>
</tr>
<tr>
<td>N18</td>
<td>714.0</td>
<td>135.5</td>
</tr>
<tr>
<td>N21</td>
<td>751.0</td>
<td>175.0</td>
</tr>
<tr>
<td>Flat Plate</td>
<td>484.75</td>
<td>242.75</td>
</tr>
</tbody>
</table>
(iv) Comparison of the pressure traces corresponding to the surfaces N18, N21 and flat plate shown in Figs. 3.4 (h), (i) & (j) indicate that as the pitch between the rows of ribs get smaller the roughened surfaces behave like a flat plate. Thus the geometries N18 & N21 can be classified under the skimmed flow behaviour during which the flow does not reattach to the surface and the presence of the obstacles has very little influence on the free stream flow.

Shown in Fig 3.5 are the classification of the above surfaces depending on the flow regimes based on their behaviour.

3.2.2 Selection of surfaces for further tests

Data tabulated in Table 3.1 shows the peak pressure reached in mV and the time taken to reach peak pressure in ms for each of the above surfaces tested. Based on the reasonings given in section 3.2.1 above, and the data tabulated in Table 3.1 the following surfaces were selected for further tests in which the mixture strength was to be varied. These are:

N10-ORD; N10-EXT; N13-2; N16-2 and a flat plate.

3.2.3 Mixture strength study

By changing the flow rates of propane and air a variation in mixture strength was obtained. The five surfaces selected from the above experiments were tested at lean and rich mixtures having equivalence ratios of 0.75 and 1.25 respectively at a blockage ratio of 0.4. The chamber pressure was digitized at a frequency of 10 kHz using the Digital MINC - 23 data acquisition system. The mean chamber pressure was evaluated by averaging five individual firing tests. The burn rate was evaluated from the averaged pressure by the method described in Appendix E. From the averaged pressure and the burn rate shown in Figs 3.6 & 3.7 the following observations are drawn:
(i) At lean and rich mixtures the performance of the surface N13-2 is better than the surfaces N10-ORD, N16-2 and flat plate and the peak pressure reached is nearly equal to that reached by N10-EXT. This behaviour suggests that when mixtures are lean or rich the dominant driving force behind the acceleration of burn rate is the turbulent intensity. Thus it can be concluded that the turbulence generated by the surface N13-2 is higher than those generated by other roughened surfaces.

(ii) Peak pressure is reached earlier with the N13-2 surface when compared with the other rough surfaces, indicating that the burn rate is faster.

(iii) At lean and rich mixtures the burn rate of the N13-2 surface is higher than other rough surfaces.

3.2.4 Photographic observations

To observe flame development and also determine the flame speed shadowgraph or schlieren methods can be used both of which respond to changes in density occurring in the flow field [Fig 3.8]. Schlieren method optical arrangements are similar to those used in the shadowgraph with the exception that an additional lens and a knife edge are necessary to obstruct part of the light refracted from passage through the test section. Because of the obstruction the schlieren method is sensitive to the first derivative of the density, whereas the shadowgraph responds to the second derivative of the density.

In general the schlieren method gives greater contrast than the shadowgraph but the use of an additional focussing lens and a knife edge results in a loss of light intensity. Moreover the density gradients are sufficiently high to give a good contrast with a shadowgraph, which was used in the study here. The
principle of the shadowgraph is that if the density gradient normal to the light beam is non-uniform, adjacent rays will be deflected by different amounts and will converge or diverge on leaving the test section. An image will then be formed directly as a shadow on a diffuser screen placed ahead of the camera.

3.2.5 Experimental setup for high speed photography

The set up of the optical workbench is shown schematically in Fig 3.9. A 10 W copper vapor laser was used as a light source and by suitable arrangement of lenses L1 & L2 and an aperture A1 a collimated beam of sufficient width was obtained to illuminate the combustion chamber [test section] uniformly. The camera used for the photographic work was capable of speeds upto 40000 frames per second [fps]. A close analysis of the pressure traces and a few trails suggested that a speed of 12000 fps was sufficient to capture the shadow of the flame front when rough surfaces were used and for the flat plate a speed of 6000 fps was sufficient. The pulse generated by the camera was used to trigger the laser externally. The first pulse generated from the camera was delayed by a second to initiate combustion by spark thus allowing enough time for the camera to accelerate to the desired framing speed.

The results of the photographic observations for three types of surfaces, N10-ORD, N13-2 and N21 obtained at a blockage ratio of 0.4 for stoichiometric air-fuel ratio are shown in Figs 3.10 to 3.12. Also shown in Fig 3.13 are the traces of the shadowgraphs for a flat plate. Thus the flame speed evaluated from the traces for the three rough surfaces are tabulated in Table 3.2 below.

3.2.6 Interpretation of flame speed data

Thus from the traces of the shadowgraphs and from the data tabulated in Table 3.2 it can be seen that the surface N13-2 maintains a continuous acceleration of the flame. With the surface N10 ORD the flame speed increases and then decreases
<table>
<thead>
<tr>
<th>Row No.</th>
<th>Flame arrival</th>
<th>Time taken between each</th>
<th>Flame speed (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Frame No. t (ms)</td>
<td>rivate t (ms) P / t</td>
<td></td>
</tr>
<tr>
<td>N10 ORD</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>0</td>
<td>-</td>
</tr>
<tr>
<td>3</td>
<td>10</td>
<td>0.83</td>
<td>12.04</td>
</tr>
<tr>
<td>4</td>
<td>30</td>
<td>1.67</td>
<td>4.0</td>
</tr>
<tr>
<td>N13-2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>0.0</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>6</td>
<td>0.5</td>
<td>16</td>
</tr>
<tr>
<td>3</td>
<td>11</td>
<td>0.917</td>
<td>19.2</td>
</tr>
<tr>
<td>4</td>
<td>15</td>
<td>1.25</td>
<td>24.0</td>
</tr>
<tr>
<td>5</td>
<td>18</td>
<td>1.5</td>
<td>32.0</td>
</tr>
<tr>
<td>6</td>
<td>20</td>
<td>1.67</td>
<td>48.0</td>
</tr>
<tr>
<td>N21</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>0.0</td>
<td>-</td>
</tr>
<tr>
<td>3</td>
<td>5</td>
<td>0.417</td>
<td>12.0</td>
</tr>
<tr>
<td>4</td>
<td>10</td>
<td>8.333</td>
<td>12.0</td>
</tr>
<tr>
<td>5</td>
<td>15</td>
<td>1.25</td>
<td>12.0</td>
</tr>
<tr>
<td>6</td>
<td>20</td>
<td>1.666</td>
<td>12.0</td>
</tr>
<tr>
<td>7</td>
<td>25</td>
<td>2.083</td>
<td>12.0</td>
</tr>
</tbody>
</table>
whereas with the surface NZ1 the flame speed remains a constant. This again suggests that the packing of the rows of ribs is too far in the N10 ORD surface which allows the flow to reattach to the surface, whereas with the NZ1 surface the ribs are too close thereby having a skimmed flow configuration. Since there is no turbulence generation in the flat surface the flame takes approximately 25 ms to propagate across the chamber.

3.2.7 Flame speed ratio vs Roughness parameter

Shown in Table 3.3 below is the flame speed ratio for the surface N13-2 evaluated from the data given in Table 3.2. Also indicated in Table 3.3 are the values of the two roughness parameters $R_1$, $R_2$.

<table>
<thead>
<tr>
<th>Row No.</th>
<th>Flame speed ratio based on rib no. 2.</th>
<th>Roughness parameter based on rib no. 2.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>$R_1$</td>
</tr>
<tr>
<td>1</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>1.0</td>
<td>0.0</td>
</tr>
<tr>
<td>3</td>
<td>1.21</td>
<td>0.492</td>
</tr>
<tr>
<td>4</td>
<td>1.5</td>
<td>0.984</td>
</tr>
<tr>
<td>5</td>
<td>2.0</td>
<td>1.476</td>
</tr>
<tr>
<td>6</td>
<td>3.0</td>
<td>1.95</td>
</tr>
</tbody>
</table>

The roughness parameter $R_1$ is given by the RHS of eq. 1.6 (b).

$$R_1 = C_{(\frac{r-h}{h})^2}^{2.5}0.31 \quad ...(3.1 \ a)$$

Since the work of Robinson & Wheeler /43/ and Moen et al. /44/ were performed with two dimensional ribbed surfaces the reattachment length, $L$, was taken as
\[ L = 6h \]

where \( h \) is the obstacle height.

Thus the flame speed ratio when plotted as a function of the roughness parameter, \( R_1 \), for the surface NL3-2 a higher slope can be seen in Fig 3.14 (a).

From the analysis of the traces of the shadowgraphs [Figs 3.10-3.12] the reattachment length for three-dimensional ribs, \( L_{3r} \) can be taken as

\[ L_{3r} = 4.0 \, h \]

In addition to the above necessary correction for reattachment length the roughness parameter has to be modified to account for the higher turbulence generated by three-dimensional ribs. Taking the lateral pitch of three-dimensional ribs into consideration the roughness parameter, \( R_2 \) can now be written as

\[ R_2 = \frac{C(r_r-\varrho)}{4h} \left[1 + n \frac{g}{g+e}\right] \left(\frac{h}{D}\right)^{0.31} \quad ...(3.1 \, b) \]

where \( n \) is the row number
- \( g \) is the lateral gap between the ribs in a row
- \( e \) is the length of the rib, and
- \( C \) is a constant taken as 0.24.

Based on the above modification for the roughness parameter for three-dimensional ribs the plot of 'flame speed ratio' vs 'roughness parameter \( R_2 \)' is shown in Fig 3.14 b.

It can be seen that the above correlation for roughness parameter \( R_2 \) holds good for two-dimensional ribbed obstacles also. The roughness parameter remains unaffected, provided the correction for reattachment length is taken into consideration, because for
two-dimensional ribbed obstacles the lateral gap, \( g \), is zero. Hence the term
\[
\frac{ng}{(g+e)}
\]
in eq. 3.1 (b) reduces to zero.

3.3 Cylindrical bomb experiments

To investigate the effect of blockage ratio caused by the change in the spark timing on the combustion process, two surfaces N10-ORD - adapted for circular geometry and a flat plate were selected. The dimensions of the chamber were chosen so as to replicate the geometry of Ricardo E-6 having a bore diameter of 7.62 cm and a depth of 4.0 cm so that early ignition could be simulated thereby achieving a low blockage ratio [Fig 3.15]. To allow effective utilization of the ribs to accelerate the flame, and based on physical intuition the layout of the ribs for the N10-ORD surface is as shown in Fig 3.16. The results of the experiments performed with stoichiometric air-fuel ratio are shown in Figs 3.17 at blockage ratios varying from 0.2 to 0.5. From the analysis of the pressure diagrams the following observations are made:

(i) At low blockage ratios the peak pressure with a flat surface are higher than the rough surface [Fig 3.17 a&b]

(ii) As the blockage ratio progressively increases the effect of rough surfaces can be seen as the peak pressure occurs earlier in comparison to the smooth surface even though the magnitude of the peak pressure does not change much [Figs 3.17 c & d]. This suggests that the heat loss is quite significant in the rough surface and also the flame is susceptible to quenching phenomenon due to high surface area.

Hence for a rib height of 4 mm and a blockage ratio of 0.5 a compression ratio of 15:1 would be necessary in an engine having a
swept volume of 507 cc. Further to the lowering of mechanical efficiency because of higher cylinder pressures, the problem of flame quenching becomes significant. Thus for a compression ratio of 10:1 or 11:1 other means of attaining higher blockage ratio is required for which the following two methods were considered:

(i) use of divided a chamber so that the flame jet emerging from the prechamber and entering the main chamber hits the ribs so that further acceleration of the burn rate could be maintained in the main chamber.

(ii) by using two spark plugs with a two stepped piston designed to achieve a blockage ratio of 0.5 at a compression ratio of 10:1 to maintain flame acceleration. The total volume of the bowls thus obtained was 21.3 cc [Fig 3.18].

3.3.1 Prechamber design parameters

A review of literature on divided chambers [Gruden et.al. /92/, Weaving & Corkill /93/] suggests that:

(i) to keep combustion harshness to an acceptable level the ratio of throat area, $F_k$, to prechamber volume, $V_{pc}$ has to be maintained at or greater than 0.08 cm$^{-1}$.

(ii) to maintain low HC, NO$_x$ & CO emissions and to reduce the loss of maximum power the ratio of auxiliary to main chamber volume, $V_{pc}/V_{mc}$, was selected to be 10%.

Based on the above considerations the Ricardo Comet Mk-V combustion chamber was considered which is basically an indirect injection diesel engine head designed for the Ricardo E-6 engine. The throat area, $F_k$, of the single holed nozzle was 0.4 cm$^2$ which for a 10:1 compression ratio and for $V_{pc}/V_{mc}$ of 0.1 gives a
chamber volume of 5.6 cc. The ratio of throat area to prechamber volume thus obtained is 0.075 cm^{-1}. Thus the values of $V_{pc}/V_{mc}$ and $F_k/V_{pc}$ are quite close to the requirements for a divided chamber spark igniton engine listed above [see Fig 3.19].

3.3.2 Constant volume experiments with divided chamber and double spark plug chambers

Based on the above design parameters and to evaluate which type of chamber would give the best performance pressure data was obtained for the chambers designated in Table 3.4 below. The experiments were performed at equivalence ratios of 1.0 (stoichiometric) and 0.8 (lean) for total chamber volumes of 36.5 & 46.5 cc.

<table>
<thead>
<tr>
<th>Chamber</th>
<th>Divided chamber type</th>
<th>Conventional chamber type</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(pre chamber head PC)</td>
<td>(Flat head FH)</td>
</tr>
<tr>
<td>Flatt</td>
<td>Rough piston</td>
<td>Single spark plug</td>
</tr>
<tr>
<td></td>
<td>piston (N10)</td>
<td>Two spark plugs</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Flat piston</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Flat piston</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Two stepped piston</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Flat piston</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Piston</td>
</tr>
<tr>
<td>Designation</td>
<td>PCHNSMD</td>
<td>PCHNROU</td>
</tr>
</tbody>
</table>

The pressure data thus obtained for the above chambers are shown in Figs 3.20 to 3.21. The following observations are drawn from the pressure diagrams:

(i) Figs 3.20 and 3.21 indicate that the performance of the double spark plug with a flat piston crown is much
better than the single spark plug with a flat piston crown.

(ii) At stoichiometric air-fuel ratio the performance of the double spark plug two stepped piston is better than other chambers but became worse than double spark plug smooth flat surface at lean mixtures [Figs 3.22 & 3.23]. The reason can be attributed to flame quenching phenomenon because of high surface to volume ratio.

(iii) Figs 3.22 and 3.23 indicate that even though the peak pressure occurs earlier with the divided chamber having a rough piston crown the magnitude of the peak pressure obtained is lower than that obtained with the divided chamber having a flat piston crown. High speed direct photographic observation, at 1500 fps, of the combustion process in the divided chamber showed that the flame jet emerging from the nozzle hits the first rib of the first row and it spreads in a reverse direction to the jet flow and covers most of the main chamber towards the end of the burn [Fig 3.24]. A similar observation in a divided chamber stratified charge was made by Weaving and Corkill /93/. Shown in Fig 3.24 are the traces of the flame fronts at different time intervals. The areas on either side of the nozzle are last to burn and the flame must travel at least one combustion chamber diameter to reach those areas.

As a result the ribs were not being effectively utilized to accelerate the flame. This prompted the development of 3 - holed nozzle [see Fig 3.25] so that the flame jets emerging from the prechamber via the nozzle hits the first row of ribs at the same time. The geometrical parameters were kept the same as with the single holed nozzle viz., throat area of 0.4 cm$^2$ and a prechamber volume of 5.6 cc. The chamber pressures obtained are plotted along with the results for the two surfaces tested earlier with
two spark plugs [Figs 3.26 & 3.27]. The pressure data thus obtained indicate that:

(i) At stoichoimetric mixtures the double spark plug two stepped piston gave the best performance closely followed by double spark plug flat surface and the 3 holed nozzle with a rough surface (PCMNROU).
(ii) At lean mixtures the performance of the double spark plug smooth flat plate and 3 holed nozzle with a rough surface were nearly the same. The poor performance of the double spark plug two stepped piston can be attributed to flame quenching.

3.4 Flame area from light emission by radicals

One of the methods that can be used to determine flame area is by monitoring the light emission from the flame. The continuous light emission signal from the photomultiplier tube (PMT) represents the total area of the flame front.

To monitor the light emitted the use of emission of free radicals such as C2 and CH can be used which are confined at the flame front in hydrocarbon-air mixtures. Thus the intensities of the blue CH (431.5 nm) and green C2 (515.6 nm) bands should be proportional to the volume of the reaction zone, i.e., the area of the flame front. To reduce the effect of background continuum radiation from the burnt gases and isolate other wavelength components in the emission spectra narrow band spectrum filters (5 nm) placed between the collecting optics and the PMT are necessary [Fig 3.28]. In the experiments performed in this study light from the flame was optically filtered to restrict sensitivity to the 431.5 nm wavelength emitted by excited CH radicals in the flame front [Hurle et.al. /94/, Checkel & Thomas /95/].
In order to calibrate the flame area measuring system a high speed video was employed which had a maximum framing rate of 200 frames per second. To get a clearer view of the flame and also record the output of the PMT, the photographic process had to be repeated twice. In one set both the PMT signal and the combustion process were recorded on the video and in another set only the combustion process was recorded.

Shown in Figs 3.29 and 3.30 are selected photographic observations of the combustion process with two spark plugs for a cylindrical chamber having a flat piston crown for the same conditions. The change in the PMT output can be seen clearly as the combustion intensity changes. A clear identification of the flame front could not be obtained since the combustion chamber was being viewed from an angle. However, the flame fronts can be clearly identified in Figs 3.30.

Since the output of the PMT is proportional to the light emitted (light intensity) by the CH radicals the change in voltage level represents the flame area. Thus the plot of the signal level from the PMT as a function of time is shown in Fig 3.31 (a). From the second set of video frames the flame area was evaluated and is shown in Fig 3.31 (b) as a function of time.

Since the combustion process is not in phase with the video framing it is necessary to normalize the data obtained. Thus normalizing the light intensity & the flame area with their peak values and the time by the occurrence of the above peak values, $t_{1\text{max}}$ & $t_{2\text{max}}$, it is seen that the shape of the curves are almost identical [Figs 3.32]. This suggests that there is a one to one correspondence between the flame area and the light intensity emitted.

This process of calibration of flame area has inherent disadvantages. They are:
(i) Since the PMT output is of the order of mV it is susceptible to noises from external sources, such as external light entering the collecting optics unless it is properly shielded.

(ii) The framing speed of the video camera is also low and hence the data has to be taken over a large number of individual combustion cycles.

3.5 Overall view of constant volume combustion experiments

(a) Flat bomb experiments

From the pressure data and from the shadowgraphs of the flame fronts it can be concluded that the surface N13-2 is the optimum configuration in comparison to the other surfaces tested. The flame development with the surface N13-2 in the early stages, compared with the N13-1 configuration is an additional advantage which also accelerates the burn rate.

Thus from the flat bomb experiments the surface N13-2 is chosen for engine based experiments with a single spark plug.

(b) Cylindrical bomb experiments

From the comparison of the pressure data with other chambers it can be concluded that for engine based experiments the following chamber configurations should be chosen:

(i) Flat head double spark plug with a flat piston crown and an discretized two step piston crown to avoid flame quenching at lean mixtures [Fig 3.33].

(ii) Prechamber head having a 3 holed nozzle with flat and rough piston crowns.
Fig 3.1: Schematic sketch of the constant volume combustion rig showing the flowmeters $F_1$, $F_2$ and the data acquisition and storage system. The function of the mixer is to ensure that the charge entering the chamber is homogenous.
Fig. 3.2. FLAT BOMB HAVING TRANSPARENT SIDEWALLS
Fig 3.3: The rough surfaces tested in the flat bomb. 
P is the pitch between rows of ribs and is indicated for each surface. The lateral parameter g/e & A are 0.82 & 11 mm respectively for each geometry.
Figs 3.4: Bomb pressure and the corresponding rough surfaces for the geometries (a) N10 ORD; (b) N10 EXT
Figs 3.4: (contd) for the geometries (d) N13-1 & (e) N13-2. Also shown is the initial flame development (IFD).
Figs 3.4: (contd) for the geometries (f) N16-1 & (g) N16-2. Also shown is the initial flame development (IFD).
Fig 3.4 h
Fig 3.5: Classification of the rough surfaces tested in flat bomb.
Fig 3.6: Chamber pressure (a) and burn rate (b) as a function of time for the five selected surfaces tested in flat bomb at an equivalence ratio of 1.25.
Fig 3.7: Chamber pressure (a) and burn rate (b) as a function of time for the five selected surfaces tested in flat bomb at an equivalence ratio of 0.75.
Fig 3.8: Schematic sketches showing optical arrangements for (a) Schlieren and (b) shadowgraph.

S Light source; L1, L2 lenses; TS test section
KF knife edge; IP image plane; DS diffuser screen
O-A is the optical axis.
Fig 3.9: Optical workbench setup for shadowgraphy.

CC  COMBUSTION CHAMBER
DS  DIFFUSER SCREEN
L₁, L₂  LENSES
A  APPERTURE
CA  CAMERA
DB  DELAY BOX
IS  IGNITION SYSTEM
10W LASER AS A LIGHT SOURCE.
Fig 3.10: Traces of the flame front from shadowgraphs every 10 frames for the N10-ORD geometry.
Fig 3.10: (contd) traces of the flame front from shadowgraphs every 10 frames for the N10-ORD geometry.
Fig 3.11: Traces of the flame front from shadowgraphs every frame for the N13-2 geometry.
Fig 3.11: (contd) traces of the flame front from shadowgraphs every frame for the N13-2 geometry.
Figure 3.11: (contd) traces of the flame front from shadowgraphs every frame for the N13-2 geometry.
Fig 3.12: Traces of the flame front from shadowgraphs every 5 frames for the N21 geometry.
Fig 3.12: (contd) Traces of the flame front from shadowgraphs every 5 frames for the N21 geometry.
Fig 3.13: Traces of the flame front from shadowgraphs every 10 frames for a flat plate.
Fig 3.13: (contd) Traces of the flame front from shadowgraphs every 10 frames for a flat plate.
Fig 3.13: (contd) Traces of the flame front from shadowgraphs every 10 frames for a flat plate.
Fig 3.14: Correlation of flame speed ratio vs roughness parameter: (a) FSR vs $R_1$ - $R_1$ given by eq. 3.1(a) (b) FSR vs $R_2$ - $R_2$ given by eq. 3.1(b)
Exhaust  Spark Plugs  Inlet

Fig. 3.15. CYLINDRICAL BOMB
Fig 3.16: Layout of N10-ORD surface for the piston. 
S - Location of the spark plug axis.
Fig 3.17: Chamber pressure as a function of time for the cylindrical bomb at blockage ratios varying from 0.2 to 0.5.
Fig 3.17: (contd) for the rough surface N10 and a flat surface at stoichioimetric air-fuel ratio.
Fig 3.18: Two stepped piston crown with spark plugs located at $S_1$ and $S_2$ designed to achieve a blockage ratio of 0.5.
Fig 3.19: Sketch of Ricardo Comet Mk - V combustion chamber modified for spark ignition engine.
Fig 3.20: Comparison of chamber pressures for single and double spark plug chambers with a flat piston crown at a total chamber volume of 46.5 cc.
Fig 3.21: Comparison of chamber pressures for single and double spark plug chambers with a flat piston crown at a total chamber volume of 36.5 cc.
Fig 3.22: Comparison of chamber pressures for different chambers (Table 3.4) at a total chamber volume of 46.5 cc.
Fig 3.23: Comparison of chamber pressures for different chambers (Table 3.4) at a total chamber volume of 36.5 cc.
Fig 3.24: Traces of flame development in a single holed nozzle divided chamber bomb at different time intervals \( T_1 < T_2 < T_3 < T_4 < T_5 < T_5 \). Arrows indicate direction of flame propagation. A, B & C are the ribs in the first row of N10 geometry [Fig 3.16].
Fig 3.25: Three-holed nozzle having the same parameters as the single holed nozzle.
Fig 3.26: Chamber pressure comparisons at a total volume 46.5 cc for the three holed nozzle and two spark plug chambers.
Fig 3.27: Chamber pressure comparisons at a total volume 36.5 cc for the three holed nozzle and two spark plug chambers.
Fig 3.28: Schematic sketch to monitor light emission during combustion using Photomultiplier tube.
Fig 3.29: Selected photographs from high speed video showing the combustion process in the bomb and the PMT signal.
Fig 3.30: Selected photographs from high speed video showing the combustion process in the bomb.
Fig 3.31: (a) Light emission signal from the PMT. (b) Flame area under the same conditions.
Fig 3.32: (a) Normalised PMT signal
(b) Normalised Flame area
Fig 3.33: Two stepped discretized piston crown to avoid flame quenching.
CHAPTER 4
ENGINE BASED EXPERIMENTS

4.1 Introduction
The applicability of rough surfaces to enhance burn rate has been demonstrated in the constant volume combustion bomb experiments reported in chapter 3. This chapter extends the application of the rough surface optimized from constant volume combustion bomb experiments to spark ignition engines to obtain fuel consumption, cylinder pressure, and concentrations of nitrogenous oxides ($NO_x$) and unburnt hydrocarbons (HC). The present chapter has been divided into three sections. The first section describes the engine test bed and associated measuring equipment including acquisition & analysis of engine pressure data. The second and third sections deal with the experiments carried out in divided and conventional chambers respectively.

4.2 Test engine and associated combustion diagnostics
4.2.1 Engine test bed and measurements
The tests were conducted on a water cooled single cylinder Ricardo E-6 engine with the specifications listed in Appendix A. Engine power was measured with an electric dynamometer.

The chamber pressure was measured using a Kistler 601 H type transducer in conjunction with a Kistler 5001 type charge amplifier. The calibration procedure and the evaluation of the calibration factor is described in Appendix D. The output of the charge amplifier (pressure signal) was fed into an Analog - Digital (A-D) converter through a signal conditioner. The signal conditioner is a device used to backoff the voltage level of the pressure signal from the charge amplifier. This was necessary because, during the induction and exhaust processes the voltage level of the pressure signal from the charge amplifier was
negative. The A-D converter used had an operating range of 0 to 10 V. Any signal outside this range gets truncated, resulting in loss of data. Hence it was necessary to raise the minimum voltage level of the pressure signal from the charge amplifier to above the zero voltage level using a signal conditioner. The signal was digitized in synchronism with engine crank angle using pulses at two degree crank angle intervals generated by a crankshaft encoder consisting of a slotted disc and an emitting & receiving diode pair. As a result the measured pressure was associated with a specified location in the engine cycle unaffected by variations in engine speed. Since the output voltage of the charge amplifier was a linear function of pressure the data acquisition system provided a linear conversion of the output voltage from the charge amplifier into digitised integer information. The synchronization and the linearity of the digitized information made it possible to process experimental raw data by averaging. The mean cylinder pressure was evaluated by averaging 50 consecutive cycles acquired every 2 degree crank angle for 720 crank angle degrees in the engine cycle. Except in one set of experiments, in a divided chamber engine with a flat piston crown, only 16 consecutive cycles were acquired due to memory problem. This produced an average of 360 data points for each condition. The averaging process was necessary to smoothen the data acquired because of the nature of cyclic variability inherent in spark-ignition engines. The standard deviation of the measured pressure was also computed.

Because a piezo electric transducer measures relative rather than absolute pressures, the absolute value of each integer obtained in a data acquisition process are not related to the absolute cylinder pressure by any constant that could be determined by calibration. Thus the absolute cylinder pressure was obtained by adding a constant amount to the relative cylinder pressure which was determined by assuming that the cylinder pressure at bottom dead center (BDC) of induction process was equal to the mean induction manifold absolute pressure. Shown in Fig 4.1 is a
For exhaust gas sampling the probe was placed in the exhaust manifold. The sampled exhaust gas was passed through a water cooled condenser, to remove water vapour, before feeding to the exhaust gas analysis equipment. Measurements were made of the exhaust gas components, carbon monoxide (CO) & carbon dioxide (CO₂) using Non-Dispersive Infrared (NDIR) analysers and of O₂ using the paramagnetic oxygen analyser. The unburnt hydrocarbon (HC) concentration was measured by a flame ionization detector (FID) and nitrogenous oxide (NOₓ) concentration was measured by a chemiluminescence technique.

Each gas analyser was calibrated with a set of 5 calibration gases except for the O₂ analyser which was calibrated with ambient air. The FID was calibrated with propane. The concentrations of CO, CO₂ & O₂ was measured in percent volume while the concentration of NOₓ was measured in parts per million (ppm) on a volumetric basis. The hydrocarbons in the exhaust are a mixture of unburnt hydrocarbons whose composition changes with engine operating conditions. The FID counts the number of carbon atoms and hence more reliable than the other methods such as NDIR which fails to account for the aromatics in the exhaust. A convenient method to represent hydrocarbon emission is in terms of ppm hexane rather than on a per carbon basis. Thus the value obtained from FID on a per carbon basis is reduced by a factor of six for hexane equivalent.

4.2.2 Method of analysis and interpretation of measured data

(a) Pressure:
From the averaged pressure heat release can be evaluated either from 'first law of thermodynamics' or from the 'method of Rassweiler & Withrow' /1/ which are described below.
(i) First law of thermodynamics

The equation of state for any two thermodynamic states can be written, by assuming ideal gas behaviour, as

\[ P_1 V_1 = mRT_1 \]  \hspace{1cm} \text{(4.1)}
\[ P_2 V_2 = mRT_2 \]  \hspace{1cm} \text{(4.2)}

where \( P, V, T \) & \( R \) are the pressure, volume, temperature and specific gas constant respectively. The subscripts 1 & 2 refer to the two thermodynamic states.

By definition of internal energy, \( U \), the change in internal energy, \( dU \), for the above two thermodynamic states is given by

\[ dU = mC_v(T_2 - T_1) \]  \hspace{1cm} \text{(4.3)}

Substituting for \( C_v \) in eq. 4.3 by the relation, \( C_v = R/(k-1) \), eq. 4.3 can be rewritten with the help of eqs 4.1 and 4.2 as

\[ dU = \frac{P_2 V_2 - P_1 V_1}{k-1} \]  \hspace{1cm} \text{(4.4)}

where \( k \) is the isentropic exponent.

Assuming the two thermodynamic states to be close the pressure curve can be approximated by a straight a line. The workdone, \( dW \), is then given by the term

\[ dW = PdV \]
\[ = \frac{1}{2} (P_1 + P_2) dV \]
\[ = \frac{1}{2} (P_1 + P_2) (V_2 - V_1) \]  \hspace{1cm} \text{(4.5)}

Now applying the first law of thermodynamics for the closed system the heat released, \( dQ \), is given by

\[ dQ = dU + dW \]  \hspace{1cm} \text{(4.6)}
Substituting for $dU$ & $dW$ from eq.s 4.4 & 4.5, $dQ$ in eq. 4.6 can be evaluated and is given by

$$dQ = \frac{P_2V_2 - P_1V_1}{k-1} + \frac{P_1 + P_2}{2} (V_2 - V_1) \tag{4.7}$$

Thus the term $dQ$ in eq. 4.7 is the net heat released which is the difference between the heat released due to combustion and the heat lost to the surroundings.

(ii) Method of Rassweiler and Withrow

Shown in Figs 4.2 is an illustration of the Rassweiler & Withrow's method /1/ to evaluate pressure change due to combustion from measured pressure. The time axis of the measured pressure is divided into intervals of 2 deg crank angle, which is the resolution of the data acquisition system. Each of the pressure increments in Fig 4.2 (a) constitutes the change in pressure due to piston motion and the change in pressure due to combustion. For each of the above intervals, the increment in pressure due to combustion, $P_{c1}$, is determined by subtracting the change in pressure due to piston motion, $P_{m'}$, from the change in measured pressure [Fig 4.2 (b)].

The pressure $P_{c1}$ thus obtained is then corrected for constant volume combustion by the ratio $V_i/V_{ign}$, i.e.,

$$P_c = P_{c1} (V_i/V_{ign})$$

where $V_i$ is the instantaneous chamber volume

$V_{ign}$ is the chamber volume at ignition.

Summation of the above increments in pressure due to combustion for each interval yields the total change in pressure due to combustion only ($P_{tc}$), which reaches a maximum at the end of combustion [curve 2 in Fig 4.2 (a)]. Normalizing the cumulative change in pressure due to combustion at each interval, $P_{cc}$, under
consideration by the total change in pressure due to combustion, \( P_{tc'} \) yields the mass fraction burned.

The above method, due to Rassweiler & Withrow, requires motoring pressure to evaluate the pressure change due to piston motion and can be evaluated from the isentropic relationship,

\[
pv^k = \text{constant} \quad \ldots(4.8)
\]

In Fig 4.3 (b), curves 1 & 2 are the heat release evaluated from 'First law of thermodynamics' and from the 'Method of Rassweiler and Withrow' for the same cylinder pressure shown in Fig 4.3 (a) by assuming the isentropic exponent as 1.35 in eq.s 4.7 & 4.8.

Inaccuracies in the measurement of clearance volume (+ 0.1 cc) and a slight offset (+ 0.5 degree) in the positioning of top dead center (TDC) introduce considerable errors such as:

(i) Low (<1.1) and high (>1.5) values of the isentropic exponent, \( k \), during compression and expansion strokes [Douaud & Eyzat /96/].

(ii) Deviations from a straight line on a log P - log V diagram for a motoring pressure indicate improper phasing of measured data [Lancaster et.al /97/].

Analysis of measured motoring pressure using the data acquisition system described above in section 4.2.1 indicate that the isentropic exponent varies from 1.2 to 1.4, which shows that the phasing of pressure with crank angle and the measurement of clearance volume are within the acceptable experimental error limits.

For data that are properly phased and scaled, however, improper slopes and/or curvature in the compression and expansion lines indicate a wrong selection of the time constant, i.e. a short time constant [Lancaster et.al. /97/].
Hence in order to overcome the above uncertainties the procedure for the evaluation of heat release from averaged pressure adopted in this study is described below and a medium time constant is selected [Douaud & Eyzat /96/, Lancaster et al. /97/, Brown/98/].

The basic procedure adopted for the heat release analysis is the method of Rassweiler and Withrow as described above. For the evaluation of motoring pressure, cylinder pressure measured by motoring the engine is used instead of the isentropic relationship [eq. 4.8]. The advantages of this method are:

(i) it removes any inconsistency involved in the assumption of the value of isentropic exponent, k, which varies during compression and expansion strokes.

(ii) it is also insensitive to the exact location of TDC.

The heat release evaluated by the above method for the same cylinder pressure shown in Fig 4.3 (a) is represented by curve 3 in Fig 4.3 (b). This method of evaluation of heat release from cylinder pressures will be adopted in this study.

Thus the heat release evaluated from averaged pressure data is a good diagnostic of the combustion process itself which indicates whether the burn is fast or slow. A good measure of the nature of cyclic variability, inherent in spark-ignition engines, is the plot of standard deviation of measured pressure and also a plot of the cylinder pressure for each of the 50 cycles in isometric projection indicates the variation in the magnitude of peak pressure from cycle to cycle. Shown in Figs 4.3 - 4.5 are typical diagnostics for any set of pressure data obtained for a given test condition.

(b) Data interpretation:
When the engine had reached a steady state (approximately 15 - 20 minutes) for a given test condition, viz., throttle opening
[given by the position of the butterfly valve on Ricardo E 6 engine], ignition timing, air-fuel ratio, speed and a constant cooling water temperature, the concentrations of the exhaust gas species, namely CO₂, CO, O₂, NOₓ & HC were recorded. The ignition timing was set manually for best torque (MBT) by changing the spark timing in steps of two degree crank angle. The air flow rate was measured using a flow meter and fuel flow was measured by timing the consumption of a fixed volume of fuel. The exhaust gas temperature and the load on the brake dynamometer were also recorded. The engine was judged to be stable if the speed did not vary by ± 10 RPM. For a change in engine speed by ± 50 RPM the engine was considered to be unstable. A common and widely used method for the interpretation of the above data to evaluate engine performance, is to evaluate fuel consumption and pollutants emitted in terms of brake power produced by the engine, i.e. brake specific fuel consumption (BSFC), brake specific nitrogenous oxide (BSNOₓ) and brake specific hydrocarbon (BSHC). The details of the calculation of brake mean effective pressure (BMEP), BSFC, BSNOₓ and BSHC is given in Appendix F.

4.2.3 Error in measurements and precautionary measures
To ensure the repeatability of data obtained it is necessary to evaluate the possible sources of error. They are in the measurement of cylinder pressures, exhaust gas concentrations, and in the estimation of air-fuel ratio which are discussed below.

(a) Pressure:
(i) Precautionary measures are to ensure adequate supply of cooling water so as to prevent thermal shock and drift caused by thermal shock.
(ii) Additional measures are to keep the transducer and the transducer cables free of grease and moisture which also causes drifting of the pressure signal.
(iii) To prevent drifting of the charge amplifier the toggle switch on the charge amplifier is set to 'Reset'.
position, if the cylinder pressure is not being sampled and is changed over to 'operate' position 5 to 6 minutes before the sampling of cylinder pressure.

(b) Exhaust gas concentrations:
Precautionary measures are to ensure that the exhaust gas is completely dried so that the flow lines do not get blocked. In order to achieve this the exhaust gas sample is passed through a water cooled condenser.

(c) Air-fuel ratio estimation:
(i) From the measurements of air and fuel flow rates the air-fuel ratio can be evaluated. Another method to evaluate air-fuel ratio, independently of flow rates, is from exhaust gas analysis as described by Spindt /99/. Comparison of air-fuel ratios estimated from exhaust gas analysis and from flow rates was within $\pm 0.5 - 1.0$ air-fuel ratio [Fig 4.6].

4.2.4 Piston crown fabrication

Since the machining of different rough surfaces on solid pistons is prohibitive in terms of cost, the rough surfaces were machined on Aluminum discs having the same diameter as that of the piston. These detachable piston crowns were secured to the solid piston with the help of screws [Fig 4.7].

4.3 Divided chamber experiments

The Ricardo Mk - V comet combustion chamber modified for use as a divided chamber homogenous charge spark-ignition engine is shown in Fig 4.8. The geometrical parameters of the prechamber used for the experiments had:

(a) a prechamber volume of 5.6 cc.
(b) a throat area of 0.4 cm$^2$.

Thus the ratio of prechamber to main chamber volume for a swept volume of 507 cc at 10:1 compression ratio is 0.1 and the ratio of throat area to prechamber volume is 0.075 cm$^{-1}$, which are
quite close to the requirements for a divided chamber homogenous charge spark-ignition engine discussed earlier in section 3.4. The charge was ignited by a spark plug located in the wall of the prechamber above the orifice. The cylinder pressure was measured via a hole of diameter 2 mm having access to the prechamber. When the cylinder pressure reached its peak an oscillation of the Helmholtz resonator type of frequency 6.3 kHz was observed and was independent of the engine operating conditions. A similar observation of the Helmholtz type oscillation in the cylinder pressure was made by Noguchi et.al. /100/ and the reason was attributed to the location of the spark plug in the prechamber.

To ensure a good homogeneous charge the fuel used was commercial grade propane so that the fresh mixture flows into the prechamber through the orifice during the compression stroke.

From the constant volume cylindrical bomb experiments reported in chapter 3, the following piston crown surfaces are chosen for experiments with the divided chamber engine:

(a) Rough piston crown having N10 type configuration.
(b) Flat piston crown.

4.4 Conventional chamber experiments
The shape of the conventional combustion chamber of Ricardo E-6 engine is of pancake type. The availability of two diametrically opposite holes located at the sides of the combustion chamber between the valves allows the possibility of using the chamber for dual spark plug experiments [Fig 4.9].

4.4.1 Single spark plug experiments
From the rectangular bomb experiments, reported in chapter 3, the engine performance was evaluated for the following piston crown surfaces:

(a) Since it was necessary to compare the performance of
divided chambers with a baseline engine, experiments were performed in a conventional chamber with a single spark plug having a flat piston crown. To ensure the same condition and homogeneity of the mixture the fuel used was propane.

(b) Rough piston crown having a configuration of N13-2. In this set of experiments propane was chosen initially, but due to its low calorific value more propane had to be used for the same power that would be obtained from the same engine fuelled with gasoline. In order to avoid this discrepancy so that the engine performance could be compared with published data the fuel used was changed over to gasoline (commercial grade 4 star petrol).

The cylinder pressure was measured via access no. 2 [Fig 4.9] while the spark plug was situated in access no. 1.

4.4.2 Double spark plug experiments
From the constant volume cylindrical bomb experiments, reported in chapter 3, the performance of dual spark plug chambers was considered to be good. Hence in order to obtain engine performance, data was obtained for the following piston crowns:

(a) Two step discretized piston crown designed to avoid flame quenching.
(b) Flat piston crown.

The fuel used was gasoline (Commercial grade 4 star petrol). To ensure that the two spark plugs discharged spark simultaneously it was necessary to use a different 12 V ignition coil to provide energy for the two spark plugs at the same time. The 12 V ignition coil used for firing the two spark plugs simultaneously is similar to those employed in present day commercial 2 CV cars [Manufactured by Citroen of France]. The chamber pressure was measured via a transducer located in the spark plug [Fig 4.10].
Fig 4.1: Block Diagram of the Data Acquisition System used to Sample Cylinder Pressure in This Study
End of combustion

Crank Angle

Fig 4.2 (a): Evaluation of heat release from measured pressure by evaluating cumulative pressure change due to combustion. Curve 1 is measured pressure. Curve 2 is the summation of pressure changes due to combustion.

Fig 4.2 (b): Sketch illustrating the evaluation of change in pressure due to combustion, $P_c$, by subtracting the change in pressure due to motoring, $P_m$, from the total change in pressure, $P_T$. $V_1$ and $V_2$ are the instantaneous chamber volumes over a two degree crank angle interval.
Fig 4.3: (a) Cylinder Pressure
(b) Heat Release Evaluated by Three Different Methods
For the Same Cylinder Pressure Shown in (a)
Fig 4.4: Plot of Standard Deviation of Measured Pressure - Showing the Nature of Cyclic Variability. Negligible Cyclic Variability Before Start of Combustion and After End of Combustion Can also be Observed.
Fig 4.5: Illustration of Cyclic Variability: Plot of 50 Consecutive Cycles of Measured Pressure in Isometric Projection - Clearly Identifiable Is The Change In The Magnitude of Peak Pressure From Cycle to Cycle
Fig 4.6: Comparison of Air - Fuel Ratios Evaluated by Two Different Methods for Various Operating Conditions. Also Shown is The Line With a Slope of 45; Shows the Reliability of Exhaust gas Measuring Equipment and Measurement of Flow Rates [Symbols represent Different engine operating conditions]
Fig 4.7: Schematic Sketch Illustrating the Solid Piston and the Detachable Piston Crown. The Piston Crowns used for the Experiments were all Detachable.
Spar k Plug

Access to the Pre-chamber for measuring cylinder pressure

Fig. 4. 8:  Ricardo Mk - V Comet combustion chamber modified for use as divided chamber spark ignition engine. Adjacent to the Pre-chamber is the three holed nozzle insert.
Fig 4.9: Schematic Sketch of (a) Cylinder Head and Arrangements For (b) Single Spark Plug and (c) Two Spark Plugs Experiments
Fig. 4. 10: Spark plug modified to measure cylinder pressure in dual spark plug experiments
5.1 Introduction

In this chapter the performance of the single cylinder engine experiments performed in chapter 4 are evaluated and is classified into five sections. The first section gives a summary of the conditions under which the engine performance data were obtained. The second section deals with the comparison of engine performance between the divided chambers and the conventional pancake chamber having a flat piston crown. The third section deals with the comparison of engine performance parameters between the optimized piston crown N13-2 and the dual spark plug experiments. The fourth section evaluates the performance of the optimized surface N13-2 at low loads and a comparison is made with a commercial indirect injection diesel engine at different speeds. The fifth section gives an overview of the single cylinder engine tests.

5.2 Summary of experimental test conditions

For rough surfaces to be applicable to high compression commercial spark-ignition engines the experiments were performed at a compression ratio of 10:1. higher compression ratios result in a loss of mechanical efficiency due to higher cylinder pressures and increased frictional losses. To investigate the behaviour of the engine and evaluate its performance at part throttle (25%) and at wide open throttle (100%) conditions, diagnostic data was obtained at these throttle openings at a speed of 1500 RPM for lean (17.6), stoichoimetric (15.6) and rich (14.4) mixtures. The throttle opening is given by the position of
the butterfly valve fitted to the Ricardo E-6 engine. A more representative method for throttle opening is the measurement of manifold depression in the induction, which was 4 cm & 0.5 cm of Hg for part (25%) & wide open (100%) throttle openings respectively at an engine speed of 1500 RPM. The ignition timing was set manually for best torque (MBT). To investigate the susceptibility of the chamber to knock the ignition timing was advanced in steps of 2 degree crank angle from MBT. From the distinctive metallic audible sounds heard (as if the cylinder was struck with an hammer) and from the observation of the pressure traces on the oscilloscope, the chamber was judged to be prone to knock or having antiknock qualities.

In order to compare the performance of different engines and chambers it is necessary to evaluate them on a reference scale. A common and widely used reference scale to represent engine performance is in terms of specific power produced by that particular engine such as the mass of fuel consumed or mass of pollutants emitted per unit brake power produced by the engine.

5.3 Divided chamber experiments

Shown in Figs 5.1 are the engine performance maps for the divided chambers and the conventional pancake chamber having a flat piston crown. For convenience the different combustion chambers will be designated as given in Table 5.1 below.
Table 5.1

<table>
<thead>
<tr>
<th>Designation of chamber</th>
<th>Engine type &amp; description</th>
</tr>
</thead>
<tbody>
<tr>
<td>DC - R</td>
<td>Divided Chamber engine with a Rough piston crown having N10 type configuration</td>
</tr>
<tr>
<td>DC - F</td>
<td>Divided Chamber engine with a Flat piston crown.</td>
</tr>
<tr>
<td>CC - F</td>
<td>Conventional Pancake shape Chamber with a Flat piston crown and a single spark plug.</td>
</tr>
</tbody>
</table>

The performance maps of specific fuel consumption and emissions are shown in Figs 5.1 and the raw emissions, along with the measured exhaust gas temperature ($T_{exh}$) and MBT spark timings are given in Table 5.2 below. The complete engine test data for the above three chambers in Table 5.1 is given in Table G-1 in Appendix G. The diagnostic data indicate:

(a) Fuel economy [Fig 5.1 (a)]

(i) the fuel economy is better in engine CC-F in comparison to engines DC-R & DC-F. The reason can be attributed to the heat loss being higher in engines DC-R & DC-F due to higher gas velocities and turbulence levels caused by the jet action of the flames emerging from the orifices [Noguchi et.al. /100/, Dwyer & Sanders /101/]. Similar observations of higher specific fuel consumption with stratified charge divided chamber engines were made by
### Table 5.2

<table>
<thead>
<tr>
<th>Conditions</th>
<th>Speed = 1500</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>DC - R</td>
</tr>
<tr>
<td>Wide open</td>
<td></td>
</tr>
<tr>
<td>Throttle;</td>
<td></td>
</tr>
<tr>
<td>AFR = 15.6</td>
<td></td>
</tr>
<tr>
<td>$\text{NO}_x$ (ppm)</td>
<td>2800</td>
</tr>
<tr>
<td>HC (ppm - per carbon basis)</td>
<td>1680</td>
</tr>
<tr>
<td>$T_{\text{exh}}$ (deg K)</td>
<td>555</td>
</tr>
<tr>
<td>$\text{MET}$ (deg BTDC)</td>
<td>8</td>
</tr>
<tr>
<td>Wide open</td>
<td></td>
</tr>
<tr>
<td>Throttle;</td>
<td></td>
</tr>
<tr>
<td>AFR = 17.6</td>
<td></td>
</tr>
<tr>
<td>$\text{NO}_x$</td>
<td>1720</td>
</tr>
<tr>
<td>HC</td>
<td>1830</td>
</tr>
<tr>
<td>$T_{\text{exh}}$</td>
<td>516</td>
</tr>
<tr>
<td>$\text{MET}$</td>
<td>12</td>
</tr>
<tr>
<td>Part (25%) NOx</td>
<td></td>
</tr>
<tr>
<td>Throttle;</td>
<td></td>
</tr>
<tr>
<td>AFR = 15.6</td>
<td></td>
</tr>
<tr>
<td>$T_{\text{exh}}$</td>
<td>559</td>
</tr>
<tr>
<td>$\text{MET}$</td>
<td>10</td>
</tr>
</tbody>
</table>

Gruden et al. /102/ and Oblander et al. /103/. Noguchi et al. /100/ while comparing the specific fuel consumption of open & divided chambers homogenous charge engines found better fuel economy with the open chamber upto an air-fuel ratio of 19:1 and beyond it the divided chamber indicated better fuel economy. Lower exhaust gas temperatures in divided chambers [Table 5.2] also indicate higher heat loss except at part throttle operation.

During the course of the experiments on divided chambers both the engines, DC-R & DC-F, had to be stopped between test conditions because of the inadequacy in the supply of cooling water to cool oil. Vapours of burnt oil
emerging from the crank case breather was observed which indicates high oil temperatures due to high heat loss from the piston.

(ii) engine DC-R does not show any improvement over engine DC-F because the ribs on the piston crown act as extended surfaces thereby increasing the heat loss & hence higher fuel consumption. The same observation of lower exhaust gas temperature in engine DC-R in comparison to engine DC-F can be made from Table 5.2. Shown in Figs 5.2 (a) & (b) is the rough piston crown used in engine DC-R before and after engine operation. The thermal damage caused by the high heat loss to the rough piston crown can be noticed from the melting of the edges of the obstacle and is shown in Fig 5.2 (b).

(b) Oxides of Nitrogen emission [Fig 5.1 (b)]

(i) At wide open throttle and at stoichoimetric mixture operation there is no significant change in the emission level of nitrogenous oxides between the engines DC-R, DC-F and CC-F. But at lean mixtures the emission level in engines DC-R & DC-F are lower by order of 1-2 gm/kW-hr. The reduction in raw NOx concentration in divided chambers at wide throttle operation can be attributed to the following reasons:

(a) Since the rate of heat release is increased due to the jet flame emerging from the prechamber the ignition timing can be retarded compared with that of the conventional engine. Therefore the portion of the gas that is burnt first is less compressed by the piston motion resulting in lower maximum temperature [see Figs 5.4 & 5.5 and Table 5.2].

(b) Because the jet flame emerging from the prechamber
induces a highly turbulent combustion in the main chamber a number of ignition sources are distributed in the unburned mixture [breaking up of the flame into small individual fragments]. As a result the burned & the unburned gases in the main chamber are mixed thereby reducing high temperature gradients in the main chamber. This type of combustion can be termed as "avalanche" type of combustion [Gussak & Turkish /104/].

(c) Higher heat loss due to higher gas velocities caused by the jet action of the flames which also lowers burnt gas temperatures.

(ii) At part throttle there is no significant change in the emission level of oxides of nitrogen. Higher NO\textsubscript{x} concentration at part load operation can be explained from the observation of heat release shown in Fig 5.3. At TDC nearly 75% & 50% of the charge is burnt in engines DC-R & DC-F respectively in a relatively small chamber volume thereby causing rapid rise in pressure and burnt gas temperature which are favourable for the formation of NO\textsubscript{x} and hence higher NO\textsubscript{x} concentration.

(c) Emission of unburnt hydrocarbons [Fig 5.1 (c)]
There is no significant change in the specific unburnt hydrocarbon emission (1 - 1.5 gm/kW-hr) level between the engines DC-R, DC-F and CC-F, of which engine DC-F shows the lowest emission level. But lower concentration [Table 5.2] in the divided chambers at both wide and part throttle operation can be attributed to a reduction in flame quenching because the area of the flame front in contact with the cold walls is smaller since it maintains its coherent jet structure during the burn. A similar observation of lower hydrocarbon concentration in a divided chamber stratified charge engine was
made by Krieger & Davies /105/. They attributed the above behaviour to the change in the structure of the quenching mechanism as follows: "the jet emerging from the orifice would entrain some of the quench layer. Once away from the cooling effect of the wall the entrained HC would quickly oxidize and hence lower HC emissions".

However higher HC concentration in engine DC-R in comparison with engine DC-F can be attributed to the presence of the obstacles which have dead water regions between the rows of ribs. Visible in Fig 5.2 (b) are the dark and light patches around the obstacles which suggests that the obstacles are too close and hence a possibility of skimmed flow configuration which have dead water regions consisting of pockets of unburned.

The burning rate shown in Figs. 5.3 - 5.5 are obtained by normalising the burn rate at any crank angle by the peak burn rate. This definition of burning rate holds good for Figs 5.12 - 5.17.

chambers (engines DC-R & DC-F), the cylinder pressures are low in comparison with the conventional chamber (engine CC-F) because of higher heat loss, as discussed above, due to higher gas velocities caused by the jet action of the flames and increased pumping losses. This in turn decreases the amount of work done thereby reducing power output from the engine and hence higher specific fuel consumption.

In spite of the main disadvantage of higher specific fuel consumption in divided chambers there are certain advantages of divided chambers also which are described below.

(i) Even though the cyclic variability[Figs 5.6 - 5.8] in divided chamber engines (DC-R & DC-F) are reduced the
average cylinder pressures are lower than the conventional chamber engine (CC-F) for the same operating conditions and hence less work done resulting in a corresponding increase in specific fuel consumption. A similar observation of reduction in cyclic variability with divided chamber engine were made by Gruden et.al. /102/. Noguchi et.al. /100/ also observed lower torque fluctuation and higher specific fuel consumption with divided chamber in comparison with conventional chamber at air-fuel ratios less than 19:1 beyond which the divided chamber showed better performance.

(ii) During the course of the experiments the conventional chamber (CC-F) was judged to be unstable at an air-fuel ratio of 17.6:1 and was not possible to operate at very lean mixture (air-fuel ratio of 19:1), where as the divided chambers (DC-R & DC-F) were stable even at lean mixture operation at an air-fuel ratio of 19:1 [Fig 5.9].

(iii) Extension of the lean limit operation.

(iv) Insensitivity to knock which allows either the compression ratio to be increased without changing the quality of the fuel or the use of lower octane fuels, whereas the conventional chamber (engine CC-F) was very sensitive to knock [Fig 5.10].

The reduction in cyclic variability and extension of the lean limit operation in divided chambers can be explained as follows: "Fresh mixture flows into the prechamber through the orifice during the compression stroke resulting in strong eddies in the mixture within the prechamber. Thus the high turbulence level within the prechamber causes rapid combustion, initiated by a spark, in the prechamber. As a result the jet plume emerges into
the main chamber which increases the burn rate of the mixture in the main chamber resulting in improved combustion. Thus the role of the prechamber is to generate turbulence in the main chamber caused by the jet action of the flame which in turn extends the lean limit operation".

The anti knock qualities of the divided chamber can be attributed to higher heat transfer because of higher gas velocities and turbulence levels caused by the jet action of the flames. Thus a cooling effect is provided for the end gas which inhibits auto ignition. Similar observations i.e., low octane requirement and attainability of higher compression ratios without changing the quality of the fuel was also made by Gruden et.al. /102/ whose experiments were conducted in a stratified charge divided chamber engine.

From the above results it can be concluded that even though the cyclic variability is less in divided chambers and have anti knock qualities the performance is judged to be poor in comparison with the conventional chamber within the air-fuel ratio range varying from 14.5:1 - 17.6:1. At mixtures leaner than 19:1 air-fuel ratio the divided chambers show better performance than the conventional chamber [The pancake chamber with a flat piston piston crown in this study is designated as the conventional chamber: Engine CC-F in Table 5.1].

5.4 Conventional chamber experiments

Shown in Figs 5.11 are the engine performance maps for the conventional chamber with the optimized piston crown, surface N13-2, and dual spark plug studies with flat and two-step discretized piston crowns. For convenience the engines will be designated as described in Table 5.3 below.

The performance maps of specific fuel consumption and emissions are shown in Figs 5.11 and the raw emissions, along with the
measured exhaust gas temperature ($T_{\text{exh}}$) & MBT spark timings are
given in Table 5.4. The complete engine test data for the three
chambers in Table 5.3 is given in Table G-2 in Appendix G. In
order to compare the performance of engines SSP-O, TSP-F & TSP-D
with a commercial indirect injection diesel engine on a common
basis the performance maps with have been plotted as a function
of percentage of maximum BMEP produced by the engine. The
performance and diagnostic data show:

<table>
<thead>
<tr>
<th>Designation of engine</th>
<th>Engine type &amp; description</th>
</tr>
</thead>
<tbody>
<tr>
<td>SSP-O</td>
<td>Conventional Pancake shape chamber with the Optimized piston crown, NL3-2 and a Single Spark Plug.</td>
</tr>
<tr>
<td>TSP-F</td>
<td>Conventional Pancake shape chamber with Two Spark Plugs and a Flat piston crown.</td>
</tr>
<tr>
<td>TSP-D</td>
<td>Conventional Pancake shape chamber with Two Spark Plugs and a two stepped Discretized piston crown.</td>
</tr>
</tbody>
</table>

(a) **Fuel economy** [Fig 5.11 (a)]

(i) The specific fuel consumption of engine SSP-O shows a significant improvement over the specific fuel consumption of engines TSP-F & TSP-D at both wide and part throttle operations.

(ii) At wide open throttle operating conditions engine TSP-D shows a marginal improvement in fuel economy over engine TSP-F.
<table>
<thead>
<tr>
<th>Conditions</th>
<th>NO\textsubscript{x} (ppm)</th>
<th>HC (ppm - per carbon basis)</th>
<th>T\textsubscript{exh} (deg K)</th>
<th>MBT (deg BTDC)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Part (25%)</td>
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<td>4170</td>
<td>632</td>
<td>22</td>
</tr>
<tr>
<td>Throttle; AFR = 14.5</td>
<td>2400</td>
<td>3070</td>
<td>611</td>
<td>18</td>
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<tr>
<td></td>
<td>1560</td>
<td>5060</td>
<td>622</td>
<td>16</td>
</tr>
<tr>
<td>Part (25%)</td>
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<td>3370</td>
<td>600</td>
<td>32</td>
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<tr>
<td>Throttle; AFR = 15.6</td>
<td>2000</td>
<td>2880</td>
<td>576</td>
<td>24</td>
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<tr>
<td></td>
<td>2160</td>
<td>4400</td>
<td>605</td>
<td>20</td>
</tr>
<tr>
<td>Part (25%)</td>
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<td>3250</td>
<td>560</td>
<td>44</td>
</tr>
<tr>
<td>Throttle; AFR = 17.6</td>
<td>560</td>
<td>2800</td>
<td>555</td>
<td>30</td>
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<tr>
<td></td>
<td>1500</td>
<td>4600</td>
<td>569</td>
<td>24</td>
</tr>
<tr>
<td>Wide Open</td>
<td>2000</td>
<td>2650</td>
<td>661</td>
<td>16</td>
</tr>
<tr>
<td>Throttle; AFR = 14.5</td>
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<td>2800</td>
<td>656</td>
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<td></td>
<td>1960</td>
<td>4400</td>
<td>647</td>
<td>10</td>
</tr>
<tr>
<td>Wide Open</td>
<td>1575</td>
<td>1630</td>
<td>595</td>
<td>16</td>
</tr>
<tr>
<td>Throttle; AFR = 17.6</td>
<td>1850</td>
<td>2100</td>
<td>574</td>
<td>22</td>
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<tr>
<td></td>
<td>1940</td>
<td>2520</td>
<td>563</td>
<td>22</td>
</tr>
<tr>
<td>Wide Open</td>
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<td>1600</td>
<td>640</td>
<td>20</td>
</tr>
<tr>
<td>Throttle; AFR = 15.6</td>
<td>2050</td>
<td>1470</td>
<td>626</td>
<td>12</td>
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<tr>
<td></td>
<td>2275</td>
<td>2230</td>
<td>601</td>
<td>16</td>
</tr>
</tbody>
</table>
(iii) At part throttle operating conditions engine TSP-F shows a marginal improvement over engine TSP-D at rich & stoichiometric mixtures, while at lean mixture operation there is a marked increase in fuel consumption. This behaviour can be attributed to the phenomenon of flame quenching and is indicated by the relative increase in hydrocarbon emission level. Because of the flame quenching phenomenon, flame speeds are low and hence slow burn rate resulting in lower burnt gas temperatures and hence low NOx emissions.

(b) Oxides of Nitrogen emission [Fig 5.11 (b)]

(i) At wide open throttle operation the change in the specific emission level is only marginal, with engine SSP-O showing the lowest (1 - 2 gm/kW-hr).

(ii) At part throttle operation there is no significant change in the specific emission level at rich and stoichiometric mixtures, whereas at lean mixtures engine TSP-F shows the lowest because of low cylinder pressure [see Fig 5.12].

(c) Emission of unburnt hydrocarbons [Fig 5.11 (c)]

(i) At wide throttle operation the change in the specific unburnt hydrocarbon emission level is only marginal, with engine SSP-O showing the lowest (1 - 1.5 gm/kW-hr).

(ii) At part throttle operation there is no significant change in the specific emission level at rich and stoichiometric mixtures between engines SSP-O & TSP-E, whereas at lean mixture the emission level is higher in engine TSP-F, due to flame quenching as explained earlier.
(d) Cylinder pressure and heat release [Figs 5.12 - 5.17]

(i) Even though the burn duration is short in dual spark plug chambers (TSP-F & TSP-D) at wide open throttle operation in comparison with engine SSP-O, the peak pressures are higher in engine SSP-O. The reason can be attributed to the fact that heat losses are higher in engines TSP-F & TSP-D due to high surface to volume ratio. Shown in Fig 5.18 are the simulated flame front areas as a function of its radius for a frozen piston position of 1.3 cm from the engine head which represents the ignition of the mixture at 10 deg BTDC. The spark plugs are assumed to be located at the same position as in the conventional engine shown earlier in Fig 4.8 in chapter 4. They indicate:

(1) larger flame front area in a chamber with two spark plugs rather than a single spark plug. A direct consequence of larger flame front area is that more charge is burnt for a given time interval in comparison with a single spark plug for the same conditions & hence shorter combustion duration.

(2) As a result of two advancing flame fronts the burnt gas wetted area also increases accordingly. Larger burnt gas wetted area results in higher heat loss and hence less contribution of the energy released towards rise in cylinder pressure.

Thus from the above simulation carried out for a simple pancake geometry of the Ricardo E-6 engine suggests that the ratio of flame front area to burnt gas wetted area is an important parameter. The plots shown in Fig 5.18 indicate that the above ratio is invariant whether a single spark plug or a double spark plug is used.
Though the above simulations are for a constant volume combustion the observations made can be related to the combustion in an engine because the change in volume from 10 deg BTDC to 10 deg ATDC can be considered to be negligible.

(ii) At part throttle operation the peak pressures are higher in engines TSP-F & TSP-D, but in the expansion phase there is a rapid decrease in cylinder pressure in engines TSP-F & TSP-D in comparison with engine SSP-O. Due to longer burn duration the cylinder pressure in the expansion phase is higher in engine SSP-O than in engines TSP-F & TSP-D. Moreover in the dual spark plug chambers there is higher heat loss because of high burnt gas wetted area to volume ratio. Contribution of this phase, to the work done on the piston via the term $PdV$ is larger for engine SSP-O resulting in higher output from the engine and hence lower specific fuel consumption.

5.4.1 Interpreting cyclic variability in combustion

Shown in Fig 5.19 is a plot of the peak cylinder pressure as a function of its location in the cycle for 50 consecutive cycles for the optimized and dual spark plug chambers (SSP-O, TSP-F and TSP-D). The operating conditions are:

1. Speed = 2000 RPM
2. Throttle opening = 25%
3. Air -Fuel Ratio = 17.6
4. Spark timing - Set at MBT for each chamber configuration.

Indicated for reference are the lines representing fast burn and slow burn lines which represent the phasing of the burn. It also indicates the nature of cyclic variability with engines TSP-F &
TSP-D showing the worst - even incomplete burn and misfires because the peak cylinder pressure in the two spark plug chambers occur at about 180 deg ABDC [TDC] for some of the cycles. The following explanation to the above behaviour was forwarded by Matekunas /106/ who also observed the same phenomenon: "If the initiation period (ignition delay) in a cycle is long but the duration of combustion is unaffected, the peak pressure will move to later in the cycle but remain on the same burn line. But if the initiation period remains a constant and the combustion duration fluctuates between cycles, late peak pressures fall on a slow burn line and early peak pressure on a fast burn line". Thus from Fig 5.19 and in conjunction with the above explanation of Matekunas /106/ the following observations can be drawn:

1. The bandwidth ($\lambda_0$) of location of peak pressure (LPP) for individual cycles for the optimized piston geometry (SSP-O) is very narrow ($\pm 5$ deg Crank Angle). Whereas for the dual spark plug chambers (TSP-F & TSP-D) it is quite large - varying from 180 deg ABDC upto 215 deg ABDC.

2. Even though some of the cycles for the dual spark plug chambers (TSP-F & TSP-D) lie in the zone of fast burn their peak cylinder pressures are low which also indicates higher heat loss, whereas for the optimized chamber (SSP-O) the burn is neither fast or slow but occurs at an optimum time in phase with the rate of expansion.

Thus from the above evaluation of heat release rates and from the analysis of location of peak pressure (LPP) vs magnitude of peak pressure (MPP), for the different chambers designated in Table 5.3, it can be said that for efficient operation the following factors are important:

1. The timing or phasing of the burn.
(ii) The rate of heat release relative to the rate of expansion.

In addition to the above advantages of the optimized surface NL3-2 (SSP-O), the combustion chamber was insensitive to knock at stoichiometric and lean mixtures even when the spark was advanced by 4 to 8 degrees of crank angle relative to MBT timing, while at rich mixtures trace knock was observed for the same spark advance. Whereas the dual spark plug chambers (engines TSP-F & TSP-D) were sensitive to knock even for spark advances of 2 to 4 degrees crank angle relative to MBT timing. Shown in Fig 5.20 is the plot of the MBT timing as a function of engine speed. Also shown is the knock limit for the different chambers given in Table 5.3.

The reason for the antiknock qualities of the chamber with the optimized piston surface NL3-2 (SSP-O) can be attributed to the following reasons:

(i) The flame acceleration set up by the presence of the obstacles due to higher turbulence and flame folding allows the flame to reach the end gas before the end gas auto-ignites. A similar observation on the suppression of knock by generation of turbulence was demonstrated with engine tests by Heron and Felt /107/ using a shrouded valve and a bowl in piston. The shrouded valve and the bowl in piston increase turbulence in the combustion chamber by creating swirl and squish both of which increase the burn rate and the heat loss from the chamber.

(ii) Obstacles act as extended surfaces thereby providing a cooling effect for the end gas which inhibits auto-ignition. Heron & Felt /107/ also observed the effect of coolant temperature on knock and from their investigations they found that low coolant temperatures
inhibit knock because the end gas temperatures are low due to heat loss.

During the course of the experiments the chamber with the optimum geometry (engine SSP-0) was judged to be stable at lean mixture (AFR of 17.6:1) at both wide open and part throttle (25%) operating conditions. It was not possible to operate the engine at very lean mixture (AFR 19:1) because of the frequent occurrence of misfires.

5.5 Further experiments and comparison

5.5.1 Engine stability at low load

From the above experiments it is clear that the performance of the engine with the optimized surface N13-2 (SSP-0) showed a significant improvement in fuel economy over other chambers.

Since the engine with the optimized surface, N13-2, was stable at both wide & part throttle (25%) operation it was decided to test the engine at low load. A common and widely used test point to evaluate the performance of Ford engines at low load is known as the Ford World Wide point. The characteristic requirements of this test point are:

(i) BMEP of the engine should be 2.8 bar.

(ii) Spark timing is set at MBT with the air fuel ratio and the throttle opening kept as variables.

Thus the operating conditions satisfying the above requirements of the world wide point obtained with the optimized surface N13-2 was:
(a) Throttle opening of 15% (manifold depression - 8.3 cm of Hg).
(b) Air to Fuel ratio of 17.2:1.
(c) MBT timing was 40 deg BTDC.

This particular test point is also indicated in Fig 5.11.

During the course of this experimental test point the engine was judged to be stable. The standard deviation of the measured pressure and the isometric plot of the pressure for 50 consecutive cycles acquired is shown in Figs 5.21 along with the heat release. It was difficult to operate the dual spark plug chambers (TSP-F & TSP-D) at Ford World Wide point.

5.5.2 Comparison with an IDI engine

The fuel economy of the engine with the optimized piston crown N13-2 at 1500 RPM evaluated above in section 5.3 has been compared with the fuel economy of a commercial indirect injection diesel (IDI) engine [Hofbauer & Sator /108/, Wiedmann & Hofbauer /109/] and is shown in Fig 5.11 along with the dual spark plug chambers. The IDI engine was chosen because of:

(a) The fuel economy of present day commercial IDI engines lies in between the spark-ignition engines and direct injection (DI) diesel engines, with DI engines showing the lowest.

(b) In addition the IDI data of Hofbauer et.al. /108, 109/ was selected because it was the only set of data published giving engine performance maps at various conditions.

The chamber with the optimized piston surface, N13-2, shows a significant improvement in fuel economy at wide open throttle operation over the IDI engine but as the throttle opening is decreased the IDI shows improved performance.
5.5.3 Engine performance at 2000 and 2500 RPM

Encouraging results of fuel economy at 1500 RPM prompted further tests to evaluate engine performance at engine speeds of 2000 & 2500 RPM. Shown in Fig 5.22 is the engine performance of specific fuel consumption and and exhaust gas emissions for the chambers indicated in Table 5.3 along with the IDI engine. In Fig 5.23 are the specific fuel consumption maps of the optimized surface and the IDI engine at speeds of 2000 RPM and 2500 RPM. The fuel economy of the IDI engine shows a gradual improvement over the optimized piston geometry N13-2 (SSP-0) as the engine speed is increased from 1500 RPM to 2500 RPM and was attributed to the following reasons:

(1) decrease in mechanical efficiency which is a direct consequence of increased frictional losses as the engine speed increases \([\text{which is high for single cylinder engines (Taylor & Taylor /111/, Ricardo /112/)}]\). Evaluation of the mechanical efficiency from the ratio of BMEP to IMEP showed a decrease from 85% at 1500 RPM to 75% at 2000 RPM [Fig 5.24]. The method of evaluation of FMEP is described in Appendix H.

(2) increased heat loss due to higher gas velocities as the engine speed increases, not only through the piston but also through the cylinder head. This is because the rough surface (piston crown) increases heat loss at the smooth surface (cylinder head) due to assymetric distribution of velocity and temeprature, as discussed earlier in section 2.2.
5.6 Overview of the single cylinder experiments

From the experimental results presented above the following observations are made:

(a) The performance of the engine with the surface N13-2 showed a significant improvement over other chambers within the air-fuel ratio operation limit ranging from 14.5 to 17.6 and the specific fuel consumption is comparable with an IDI engine.

(b) Even though the cyclic variability was reduced in divided chambers the performance can be considered to be poor. However, extension of the lean limit operation is a considerable advantage over conventional chambers.

(c) The double spark plug chambers did not show any improvement over single spark plug chambers, except that the cyclic variability was reduced considerably at rich and stoichiometric mixtures for both wide & part throttle operation. But at lean mixture part throttle operation the performance is considered to be poor.
Fig 5.1: Influence of Combustion Process on Fuel Economy (a) and Exhaust Gas Emissions (b & c) for the chambers designated in Table 5.2. Curve 1 is at Wide open Throttle. Curve 2 is at Part Throttle. Symbols represent data at different Air-Fuel Ratios.
## Summary of Engine test Conditions

1) DC - R : Divided chamber with a rough (N10) piston crown
2) DC - F : Divided chamber with a flat piston crown.
3) CC - F : Conventional chamber with a flat piston crown.

<table>
<thead>
<tr>
<th>Conditions</th>
<th>Speed = 1500</th>
</tr>
</thead>
<tbody>
<tr>
<td>DC - R</td>
<td>DC - F</td>
</tr>
<tr>
<td>Wide open Throttle AFR = 15.6</td>
<td>MBT (deg BTDC)</td>
</tr>
<tr>
<td>Wide open Throttle AFR = 17.6</td>
<td>MBT</td>
</tr>
<tr>
<td>Part (25%) Throttle AFR = 15.6</td>
<td>MBT</td>
</tr>
</tbody>
</table>
Fig. 5. 2. The Rough Piston Crown used in Divided Chamber experiments. (a) Before engine operation.
Fig. 5. 2(b). The Rough Piston Crown after engine operation. Noticable are the Thermal Damage caused and the Dark & Light Patches.
The burning rate shown in Figs. 5.3 - 5.5 are obtained by normalising the burn rate at any crank angle by the peak burn rate. This definition of burning rate holds good for Figs 5.12 - 5.17.

**Fig 5.3:** Averaged Pressure, Burning Rate and Mass Fraction Burned at 1500 RPM, Part Throttle (25%), Stoichioimetric Mixture Operation at MBT timings.
Fig 5.4: Averaged Pressure, Burning Rate and Mass Fraction Burned at 1500 RPM, Wide Open Throttle (100%), Stoichiometric Mixture Operation at MBT timings.
Fig 5.5: Averaged Pressure, Burning Rate and Mass Fraction Burned at 1500 RPM, Wide Open Throttle (100%), Lean Mixture Operation at MBT timings.
Fig 5.6: Cyclic Variability Showing Minimum & Maximum Pressure Cycles at 1500 RPM Part Throttle (25%) Operation.
Fig 5.7: Maximum and Minimum Cylinder Pressure cycles at 1500 RPM, Wide Open Throttle Stoichioimetric Mixture Operation set at MBT Timings.
Fig 5.8: Maximum and Minimum Pressure Cycles at 1500 RPM Wide Open Throttle Lean Mixture Operation.
Fig 5.9: Cyclic Variability of Divided Chambers at Very Lean Mixture Operation.
Fig 5.10: Illustrating Knock Sensitivity of Divided Chambers and Conventional Chamber given in Table 5.1.
The specific fuel consumption data of IDI is pertinent to the discussion in section 5.5.2.

Fig 5.11: Influence of Combustion process on fuel economy (a) Exhaust gas emissions (b & c), for the chambers in Table 5.3 at the operating conditions of Table 5.4 Curves 1 & 2 represent wide open & part throttle data
Summary of experimental conditions

1) SSP - O : Single spark plug with optimized piston crown.
2) TSP - F : Two spark plugs with a flat piston crown.
3) TSP - D : Two spark plugs with a dicretized piston crown

<p>| Conditions     | Speed = 1500 |</p>
<table>
<thead>
<tr>
<th></th>
<th>SSP-O</th>
<th>TSP-F</th>
<th>TSP-D</th>
</tr>
</thead>
<tbody>
<tr>
<td>Part (25%)</td>
<td></td>
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<tr>
<td>Throttle</td>
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<td>AFR = 14.5</td>
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<td></td>
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<tr>
<td>Part (25%)</td>
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<tr>
<td>Throttle</td>
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<td>AFR = 15.6</td>
<td>MBT</td>
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Fig 5.12: Averaged Cylinder Pressure, Burning Rate and Mass Fraction Burned at 1500 RPM, Part Throttle (25%), Lean Mixture Operation and MBT timings.
Fig 5.13: Averaged Pressure, Burning Rate and Mass Fraction Burned at 1500 RPM, Part Throttle (25%), Stoichiometric Mixture Operation and MBT timings.
Fig 5.14: Averaged Pressure, Burning Rate and Mass Fraction Burned at 1500 RPM, Part Throttle (25%), Rich Mixture Operation and MBT timings.
Fig 5.15: Averaged Pressure, Burning Rate and Mass Fraction Burned at 1500 RPM, Wide Open Throttle (100%), Lean Mixture Operation and MBT timings.
Fig 5.16: Averaged Pressure, Burning Rate and Mass Fraction Burned at 1500 RPM, Wide Open Throttle (100%), Stoichiometric Mixture Operation and MBT timings.
Fig 5.17: Averaged Pressure, Burning Rate and Mass Fraction Burned at 1500 RPM, Wide Open Throttle (100%), Rich Mixture Operation at MBT timings.
Fig 5.18: Geometrical Simulation to Evaluate Flame Front Areas, Burned Gas Volume and Burnt Gas Wetted Area for a Pancake Geometry.
Fig 5.19: Location of Peak Pressure vs Magnitude of Peak Pressure for 50 Consecutive Cycles for the Three Chambers Designated in Table 5.3.
Fig 5.20: Plot of MBT Timing vs Engine Speed for the Optimized Piston Geometry and Dual Spark Plug Chambers. Also shown are the Knock Limited Spark Advances; [NK - No knock; TK - Trace Knock; HK - Heavy Knock]
Fig 5.21: Ford World Wide Point Operation with the Optimized Piston Geometry: (a) & (b) Illustrate the Cyclic Variability and (c) Represents the Burning Rate and the Mass Fraction Burned Evaluated from Average Cylinder Pressure.
Fig 5.22: Engine performance at 2000 RPM (a) fuel economy and (b & c) Exhaust gas emissions; for chambers in Table 5.3 Curves 1 & 2 represent wide open & part throttle data Symbols represent data at different air fuel ratios.
Fig 5.23: Fuel consumption maps at engine speeds of 2000 & 2500 RPM for the optimized piston geometry and IDI engines.
Fig 5.24: Mechanical efficiency and frictional mean effective pressure (FMEP) as a function of engine speed.
CHAPTER 6

CONCLUSIONS, FURTHER WORK
AND COMMERCIAL APPLICATIONS

6.1 Conclusion
The applicability of the optimized surface N13-2 has been demonstrated by single cylinder engine experiments and from the results presented the following conclusions are drawn:

(1) Low specific fuel consumption and the fuel economy can be compared with a commercial IDI engine.

(2) Less susceptible to knock, and

(3) The engine is judged to be stable at low loads which is a direct consequence of low cyclic variability.

6.2 Commercial applications and Further work

(1) Commercial applications

The results presented indicate a significant improvement in engine performance in a single cylinder engine. However the present and future commercial applications demand the applicability of the optimized surface N13-2 to multicylinder engines. Such a demand requires the evaluation of the optimized surface N13-2 in multicylinder engines to obtain engine performance maps of fuel consumption and emissions.

The viability of such an application in multicylinder engines would remain in its competitiveness with the present art and technology involved in automotive engines. However, one major
advantage is that it can be easily adapted in the production line without any major modifications. Moreover it is definitely cheaper than the present day fuel injected stratified charge engines which require sophisticated control systems to inject a precise quantity of fuel at the right time, as a conventional carburettor would suffice its requirements.

(II) Variations for changes in chamber geometries and spark plug locations

Though the above design of the optimized surface has been designed for the Ricardo E-6 engine which has an ignition source near the wall, variations on the basic design would be necessary for changes in the location of the ignition source viz., central ignition, ignition slightly offset from the center of the chamber etc.

(III) Octane requirements

The anti knock characteristics of the surface N13-2 suggests that higher compression ratios could be attained without changing the quality of the fuel or a commercial grade 2 star petrol could be used. However the interference of the valves with the obstacles at higher compression ratios need to be taken into account to prevent any damage being caused.

In chapter 5 the anti knock behaviour of the combustion chamber with the optimized surface N13-2 was attributed to the following reasons:

(1) Flame front reaches the end gas before auto ignition occurs.

(2) The ribs act as extended surfaces thereby providing a cooling effect for the end gas which also inhibits knock.
The above reasonings still need to be proven either from:

(a) Experimental diagnostics using optical or ionization probes to detect the arrival of the flame front.

(b) Multidimensional modelling to investigate the cooling of the end gas due to extended surfaces.

(IV) Convective heat transfer

The decrease in the performance of the engine with the surface N13-2 at higher speeds was attributed to lowering of mechanical efficiency and increased heat loss.

The convective heat transfer model developed in this study relies on the velocity and temperature profiles which are based on global parameters such as the pitch between rows of ribs and number of ribs. But it does not take into account the actual layout of the obstacles on the piston which might contribute towards the cooling of the end gas. Hence a model that takes into account the layout of the obstacle needs to be developed.

(V) Design and layout

Even though the phenomenological model developed and used in this work is able to provide the designer with the difference in engine performance between the macroscopic geometrical parameters such as the pitch between the ribs and the number of ribs from the optimization parameter Q/W, it is unable to provide the effect of the microscopic parameters such as the configuration & layout of the obstacles and also the effect of the sharp corners of the obstacle on the combustion process. The layout of the obstacles in this study has been based mainly on physical intuition and fluid flow fundamentals.

A more rigorous approach for the design and the layout of the
obstacles would however need multidimensional models to provide the guidelines and also predict the effect of microscopic geometries on engine performance.

It is admittedly debatable whether a combination of the phenomenological model with constant volume combustion experiments is not an easier solution and possibly cheaper than the multidimensional approach. Moreover, multidimensional models are hindered by drawbacks such as storage constraints, inadequate level of sophistication in the turbulence and chemical models as stated earlier in section 2.5.

However, multidimensional models pose a challenging task for the modeller, to model the flame propagation process across the chamber which would help in understanding the physical process involved to a greater depth. The details of the fluid flow provided by these models could be very useful in the design and the layout of the obstacles.
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### List of Ricardo E-6 Engine Specifications and Variables

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value(s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore, cm</td>
<td>7.62</td>
</tr>
<tr>
<td>Stroke, cm</td>
<td>11.11</td>
</tr>
<tr>
<td>Displacement, cc</td>
<td>507.0</td>
</tr>
<tr>
<td>Connecting Rod Length, cm</td>
<td>24.0</td>
</tr>
<tr>
<td>Maximum Valve lift, cm</td>
<td>0.8</td>
</tr>
<tr>
<td>Fuel Used</td>
<td>Propane, Iso-Octane</td>
</tr>
<tr>
<td>Cooling Water Temperature, deg K</td>
<td>343 - 345</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>10:1</td>
</tr>
<tr>
<td>Inlet Valve Opening, deg BTDC</td>
<td>9.0</td>
</tr>
<tr>
<td>Inlet Valve Closing, deg ABDC</td>
<td>37.0</td>
</tr>
<tr>
<td>Exhaust Valve Opening, deg BBDC</td>
<td>41.0</td>
</tr>
<tr>
<td>Exhaust Valve Closing, deg ATDC</td>
<td>10.0</td>
</tr>
<tr>
<td>Spark Timing</td>
<td>Electronically controlled transistorized coil ignition system; Ignition timing set manually.</td>
</tr>
<tr>
<td>Equivalence Ratio</td>
<td>Variable</td>
</tr>
<tr>
<td>Speed, rpm</td>
<td>1500, 2000</td>
</tr>
<tr>
<td>Inlet Pressure, atm</td>
<td>Variable - evaluated from inlet manifold depression.</td>
</tr>
</tbody>
</table>

![Fig Al: Sketch of Ricardo E-6 Pancake Combustion Chamber](image-url)
**Appendix B**

**Method of calculating flame area**

The burn rate prediction by the turbulent flame speed concept requires a known value of the flame area. The basic assumption of the phenomenological model is that the flame front is thin which separates the two zones viz., burned and unburned gases, and will be adopted in this study to evaluate flame area.

Further to the above assumption, flame front can be considered to be spherical in nature from ignition till it hits the first row of ribs. After the flame front has hit the ribs it can be assumed that the flame front is parabolic in shape [see Figs B1 & B2]. Hence the parabola can be described by the equation,

\[ y^2 = a * x \]

The length of any curve is given by

\[ \int \sqrt{1 + (dy/dx)^2} \, dx \]

... (B.1)

where \(x_1\) and \(x_2\) are the limits of integration.

Therefore the flame length (curve A-B-C), \(F_1\), is given by

\[ F_1 = 2.0 \left\{ bx_1^2 + a/4.0 + a/4 \ln(\sqrt{ax_1 + \sqrt{ax_1 + a^2/4.0}}) - \sqrt{a/4 + a/4 \ln(a/2)} \right\} \]

Hence the flame area, \(F_a\), is given by

\[ F_a = F_1 * \{2.0 \theta_{rad}\} \]
where $\theta_{rad}$ is the angle subtended by the flame at the spark plug and is evaluated by assuming a mean flame radius, $R_{\text{mean}}$, from the spark. The mean flame radius is evaluated by an iteration procedure such that the volume of the burned gases before and after entering the flame geometry subroutine remains the same.

The flame area before the flame hits the ribs is evaluated from the algorithm of Lakshminarayan & Dent /110/. 

Fig B1

Fig B2
For the evaluation of laminar flame speed, \( S_L \), the model of Van-Tiggelen & Deckers /87/ model is adopted and is given by the correlation,

\[
S_L = K C_m \left[ Y_F^a Y_{O_2}^b \exp\left(-\frac{E}{RT_m}\right)\right]^{1/2}
\]

where

- \( C_m = \frac{8 R T_m}{\pi M_R} \) is the mean molecular speed of chain carriers whose mean molecular weight is \( M_R \).
- \( Y_F \) = mole fraction of fuel molecules in unburned mixture.
- \( Y_{O_2} \) = mole fraction of oxygen molecules in unburned mixture.
- \( a \) = reaction order with respect to fuel.
- \( b \) = reaction order with respect to oxygen.
- \( E \) = activation energy of fuel (Kcal/mole)
- \( T_m = T_u + 0.74 (T_b - T_u) \) is the mean temperature of the reaction zone.

To take into account the dependence of \( S_L \) on pressure Tabacyznski et al. /85/ modified the dimensionless parameter, \( K \), and is given by

\[
K = \frac{2T_u}{3 \pi T_m^a} p^a
\]

\( T_u \) and \( T_b \) are the unburned and adiabatic flame temperatures respectively.

Thus for a given fuel the constants to be determined for the evaluation of flame speed are \( E, M_R, a, b \& a \) and are given in Table B1 below.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>( E )</th>
<th>( M_R )</th>
<th>( a )</th>
<th>( b )</th>
<th>( a )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Iso-octane</td>
<td>38.6</td>
<td>67.7</td>
<td>1.706</td>
<td>-0.706</td>
<td>-0.22</td>
</tr>
<tr>
<td>Propane</td>
<td>37.3</td>
<td>31.0</td>
<td>1.460</td>
<td>-0.460</td>
<td>-0.06</td>
</tr>
</tbody>
</table>
APPENDIX D

Calibration of piezoelectric transducers

Chamber pressure in bombs and engine was measured with Kistler 6121-Al and 601 H type transducers respectively in conjunction with Kistler 5001 type charge amplifier to amplify the transducer output for input to the data acquisition system. Calibration of the transducer and charge amplifier was done with reference pressures supplied with a dead weight tester, and the charge amplifier output was monitored with a digital voltmeter. For calibration purposes the toggle switch is set to long time constant but was switched over to medium time constant for dynamic pressure measurements such as those occurring in bombs and engines. Thus the calibration of the whole pressure measuring installation was carried out by loading a dead weight tester from 10 bar to 100 bar for 601 H type transducer (for engine pressure measurements) and from 5 bar to 20 bar for 6121-Al type transducer (bomb pressure measurements). The correlation between the applied pressure on the transducer and the output voltage of the charge amplifier was found to be linear within the range considered. The slope of the line fitted to the above calibrated data points gives the calibration factor [Figs. D1 & D2].
Calibration Factor = 2.35  
6121 - A1  

Calibration Factor = 10.55  
601 H  

Legend  
□ Best fitted curve  
▲ Data points  

Fig D1  
Fig D2  

Pressure (bars)  

Voltage (Volts)
Appendix E

Heat release calculations from pressure data for constant volume combustion

From first law of thermodynamics

\[ dQ = dU + dW \]

\[ = dU + PdV + VdP \]  \hspace{1cm} \ldots E.1

where \( dQ \) is the net heat released = Heat released due to chemical reactions - Heat losses

Since \( dV = 0 \) because of constant volume combustion eq. E.1 reduces to

\[ dQ = dU + VdP \]  \hspace{1cm} \ldots E.2

By definition of internal energy, \( U \),

\[ mC_v = dU/dT \]  \hspace{1cm} \ldots E.3

Substituting for \( C_v \) by \( R/(\gamma - 1) \), since heat is assumed to be released by burnt gas only, and on rearranging eq. E.3 one obtains

\[ dU = mRdT/(\gamma - 1) \]  \hspace{1cm} \ldots E.4

From the ideal gas law it can be shown that

\[ mRdT = VdP \]  \hspace{1cm} \ldots E.5

Making use of the relationship in equations E.4 & E.5, the rate
of heat release over a time interval, $dt$, can be expressed from eq. E.2 by

$$\frac{d\Omega}{dt} = \frac{Y_b}{Y_b - 1} \frac{V}{dt}$$

The ratio of specific heats, $b$, for the burnt gases can be evaluated from the relationship [Blizard & Keck /84/]

$$Y_b = \frac{1 + Y_u}{1 + E}$$

where

$$Y_u = 1.0 + 0.34 \{1 - 0.06(\phi - 1)\}$$

and

$$E = -1. + 1.70 \{1 + 0.23(\phi - 1)\} \quad \text{if} \quad \phi \leq 1.0$$

$$= -1. + 1.70 \{1 + 0.04(\phi - 1)\} \quad \text{if} \quad \phi \geq 1.0$$

is the equivalence ratio of the fresh charge.
Appendix F

Method of evaluation of BSFC, BSNOx and BSHC

Let $m_f$ be the mass flow rate of fuel in gm/hr, then the brake specific fuel consumption is given by

$$BSFC = \frac{m_f}{P_b} \text{ gm of fuel/kW-hr}$$

where $P_b$ is the brake power produced by the engine in kW, given by

$$P_b = \frac{W \times N}{3 \times 60} \text{ bar}$$

Brake mean effective pressure (BMEP) is the work done per unit swept volume; Thus for a four stroke engine

$$\text{BMEP} = \frac{2P_b}{100V_s(N/60)} \text{ bar}$$

where $V_s$ is the swept volume in m$^3$, and $N$ is the engine speed in RPM.

The total flow rate of the charge, $m_t$, can be evaluated by summing up the mass flow rates of fuel and air. From the measurements of exhaust gas concentrations the cumulative percentage of the species measured in dry exhaust gas can be evaluated. By assuming the fuel to be pure the ratio of H/(H+C) by weight for Propane and Isooctane are 0.182 and 0.16 respectively which will be denoted by HCR.

Therefore the rate of water formation will be

$$m_w = 9\times\text{HCR} \times m_f$$

Therefore the mass of dry exhaust gas $= m_{\text{exh}} = m_t - m_w$

Since the percentage of hydrocarbon and oxides of nitrogen are known in dry exhaust gas by exhaust gas analysis the specific
fuel hydrocarbon and nitric oxide emissions can be evaluated as follows:

\[
BSHC = \frac{x_{hc}}{x_\alpha} \times (\frac{m_{exh}}{P_b})
\]

\[
BSNO_x = \frac{x_{no}}{x_\alpha} \times (\frac{m_{exh}}{P_b})
\]

The units of BSHC & BSNO\textsubscript{x} are gm/kW-hr (hexane) and gm/kW-hr respectively.
Appendix G

Summary of Engine test data

(a) In Table G - 1 data obtained for the divided chambers having a rough piston crown (DC-R) & a flat piston crown (DC-F) along with the baseline engine (Conventional chamber having a flat piston crown - CC-F) is given.

Table G - 1

<table>
<thead>
<tr>
<th>Conditions</th>
<th>DC - R</th>
<th>DC - F</th>
<th>CC - F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wide open</td>
<td>( \text{NO}_x ) (ppm)</td>
<td>2800</td>
<td>2850</td>
</tr>
<tr>
<td>Throttle;</td>
<td>HC (ppm - per carbon basis)</td>
<td>1680</td>
<td>1270</td>
</tr>
<tr>
<td>AFR = 15.6</td>
<td>( T_{\text{exh}} ) (deg K)</td>
<td>555</td>
<td>571</td>
</tr>
<tr>
<td></td>
<td>MBT (deg BTDC)</td>
<td>8</td>
<td>10</td>
</tr>
<tr>
<td></td>
<td>Brake Load (lbs)</td>
<td>13.2</td>
<td>13.6</td>
</tr>
<tr>
<td></td>
<td>Fuel flow rate for 1 cu.ft (s) ( \dot{m}_f )</td>
<td>134.4</td>
<td>130.4</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Conditions</th>
<th>DC - R</th>
<th>DC - F</th>
<th>CC - F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wide open</td>
<td>( \text{NO}_x )</td>
<td>1720</td>
<td>2000</td>
</tr>
<tr>
<td>Throttle;</td>
<td>HC</td>
<td>1830</td>
<td>1490</td>
</tr>
<tr>
<td>AFR = 17.6</td>
<td>( T_{\text{exh}} )</td>
<td>516</td>
<td>524</td>
</tr>
<tr>
<td></td>
<td>MBT</td>
<td>12</td>
<td>14</td>
</tr>
<tr>
<td></td>
<td>Brake Load (lbs)</td>
<td>11.9</td>
<td>12.2</td>
</tr>
<tr>
<td></td>
<td>( \dot{m}_f )</td>
<td>151.9</td>
<td>148.8</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Conditions</th>
<th>DC - R</th>
<th>DC - F</th>
<th>CC - F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Part (25%)</td>
<td>( \text{NO}_x )</td>
<td>2500</td>
<td>2500</td>
</tr>
<tr>
<td>Throttle;</td>
<td>HC</td>
<td>2470</td>
<td>1950</td>
</tr>
<tr>
<td>AFR = 15.6</td>
<td>( T_{\text{exh}} )</td>
<td>559</td>
<td>550</td>
</tr>
<tr>
<td></td>
<td>MBT</td>
<td>10</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>Brake Load (lbs)</td>
<td>10.2</td>
<td>9.9</td>
</tr>
<tr>
<td></td>
<td>( \dot{m}_f )</td>
<td>156.0</td>
<td>161.6</td>
</tr>
</tbody>
</table>
Table G-1 continued

<table>
<thead>
<tr>
<th>Wide open</th>
<th>DC - R</th>
<th>DC - F</th>
<th>CC - F</th>
</tr>
</thead>
<tbody>
<tr>
<td>NO&lt;sub&gt;x&lt;/sub&gt;</td>
<td>900</td>
<td>1050</td>
<td>2750</td>
</tr>
<tr>
<td>HC</td>
<td>2100</td>
<td>2000</td>
<td>2040</td>
</tr>
<tr>
<td>T&lt;sub&gt;exh&lt;/sub&gt;</td>
<td>520</td>
<td>495</td>
<td>625</td>
</tr>
<tr>
<td>Brake load (lbs)</td>
<td>11</td>
<td>11.2</td>
<td>16.4</td>
</tr>
<tr>
<td>m&lt;sub&gt;f&lt;/sub&gt;</td>
<td>152</td>
<td>160</td>
<td>124</td>
</tr>
<tr>
<td>AFR</td>
<td>19.3</td>
<td>18.8</td>
<td>14.5</td>
</tr>
<tr>
<td>MBT</td>
<td>14</td>
<td>18</td>
<td>12</td>
</tr>
</tbody>
</table>
(b) In this Table the engine test data obtained for the conventional chamber is given for the following chambers:

(i) SSP - O : Single Spark Plug with the optimized piston geometry.

(ii) TSP - F : Two Spark Plugs with a Flat piston crown.

(iii) TSP - D : Two Spark Plugs with a two step Discretized piston crown.

The fuel used was gasoline (Commercial grade four star petrol).

Table G - 2

<table>
<thead>
<tr>
<th>Conditions</th>
<th>Speed = 1500</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>SSP-O</td>
</tr>
<tr>
<td>Part (25%)</td>
<td>NO\textsubscript{x} (ppm)</td>
</tr>
<tr>
<td>Throttle:</td>
<td>HC (ppm - per carbon basis)</td>
</tr>
<tr>
<td>AFR = 14.5</td>
<td>T\textsubscript{exh} (deg K)</td>
</tr>
<tr>
<td></td>
<td>MBT (deg BTDC)</td>
</tr>
<tr>
<td></td>
<td>Brake load (lbs)</td>
</tr>
<tr>
<td></td>
<td>Fuel flow rate for 50 cc (s)</td>
</tr>
</tbody>
</table>
Table G - 2 (contd)

<table>
<thead>
<tr>
<th>Wide Open</th>
<th>NO$_x$</th>
<th>2000</th>
<th>2400</th>
<th>1960</th>
</tr>
</thead>
<tbody>
<tr>
<td>Throttle;</td>
<td>HC</td>
<td>2650</td>
<td>2800</td>
<td>4400</td>
</tr>
<tr>
<td>AFR = 14.5</td>
<td>T$_{exh}$</td>
<td>661</td>
<td>656</td>
<td>647</td>
</tr>
<tr>
<td></td>
<td>MBT</td>
<td>16</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td></td>
<td>Brake load (lbs)</td>
<td>19.6</td>
<td>16.5</td>
<td>16.9</td>
</tr>
<tr>
<td></td>
<td>$m_f$</td>
<td>78.98</td>
<td>81.40</td>
<td>80.65</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Wide Open</th>
<th>NO$_x$</th>
<th>1575</th>
<th>1850</th>
<th>1940</th>
</tr>
</thead>
<tbody>
<tr>
<td>Throttle;</td>
<td>HC</td>
<td>1630</td>
<td>2100</td>
<td>2520</td>
</tr>
<tr>
<td>AFR = 17.6</td>
<td>T$_{exh}$</td>
<td>595</td>
<td>574</td>
<td>563</td>
</tr>
<tr>
<td></td>
<td>MBT</td>
<td>16</td>
<td>22</td>
<td>22</td>
</tr>
<tr>
<td></td>
<td>Brake load (lbs)</td>
<td>17.0</td>
<td>14.1</td>
<td>14.4</td>
</tr>
<tr>
<td></td>
<td>$m_f$</td>
<td>94.80</td>
<td>98.70</td>
<td>96.52</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Wide Open</th>
<th>NO$_x$</th>
<th>2425</th>
<th>2050</th>
<th>2275</th>
</tr>
</thead>
<tbody>
<tr>
<td>Throttle;</td>
<td>HC</td>
<td>1600</td>
<td>1470</td>
<td>2230</td>
</tr>
<tr>
<td>AFR = 15.6</td>
<td>T$_{exh}$</td>
<td>640</td>
<td>626</td>
<td>601</td>
</tr>
<tr>
<td></td>
<td>MBT</td>
<td>20</td>
<td>12</td>
<td>16</td>
</tr>
<tr>
<td></td>
<td>Brake load (lbs)</td>
<td>20.2</td>
<td>15.0</td>
<td>15.5</td>
</tr>
<tr>
<td></td>
<td>$m_f$</td>
<td>83.7</td>
<td>88.37</td>
<td>88.1</td>
</tr>
</tbody>
</table>
Appendix H

Evaluation of FMEP

From the measured cylinder pressure (by using 601 H type transducer) the work done by the piston was evaluated from inlet valve closing (IVC) till exhaust valve opening (EVO). Since the 601 H transducer was not sensitive to low cylinder pressures it was necessary to evaluate pumping work by using a transducer that was sensitive at low pressures (701 H). Thus from the pumping loops obtained at different speeds for wide open throttle the pumping work was evaluated from EVO to IVC. The difference between the work done by the piston evaluated, by using the 601 H transducer, and the pumping work evaluated by using the 701 A transducer gives the indicated work. The ratio of indicated work to swept volume gives the indicated mean effective pressure (IMEP) and the from the brake load measurements, the brake mean effective pressure (BMEP) can be evaluated.

Thus the difference between the IMEP and the BMEP gives the frictional mean effective pressure (FMEP) and the ratio of BMEP to IMEP gives the mechanical efficiency.