Air motion in an indirect injection diesel engine

This item was submitted to Loughborough University's Institutional Repository by the/an author.

Additional Information:


Metadata Record: [https://dspace.lboro.ac.uk/2134/32459](https://dspace.lboro.ac.uk/2134/32459)

Publisher: © Bamidele Ayoade Ajakaiye

Rights: This work is made available according to the conditions of the Creative Commons Attribution-NonCommercial-NoDerivatives 4.0 International (CC BY-NC-ND 4.0) licence. Full details of this licence are available at: [https://creativecommons.org/licenses/by-nc-nd/4.0/](https://creativecommons.org/licenses/by-nc-nd/4.0/)

Please cite the published version.
AIR MOTION IN AN INDIRECT INJECTION DIESEL ENGINE

by

B.A. AJAKAIYE, M.Sc, D.I.C.

A Doctoral Thesis
Submitted in Partial Fulfilment of the Requirements
for the Award of
Doctor of Philosophy of Loughborough University of Technology

April 1976

Supervisor: J.C. Dent, Ph.D, C.Eng, M.I.Mech.E.

© by Bamidele Ayoade Ajakaiye
SUMMARY

The present investigation of the air motion in the prechamber of an indirect injection diesel engine is in two main parts. In the first part a mathematical model was developed to describe the air motion in the prechamber. In the second part experimental measurements were made of mean air velocities in two different cylindrical prechambers. The prechamber tests were carried out with the engine under motoring conditions. The computations of the mathematical model were compared with the present experimental results and also with previously published data by other authors. A discussion of these comparisons was made.

The mathematical model was based on the assumption of solid body rotation and the conservation of angular momentum in the prechamber. The effect of friction at the prechamber wall was included. Subsequent computations show that neglecting the friction effect leads to a small increase in the prechamber gas velocity of the order of 8%.

A three-wire probe was manufactured and used to measure the gas mean velocity vectors in the two cylindrical prechambers. Radial velocity profiles were measured at engine speeds between 600 and 1100 rpm. An error analysis based on data obtained in a wind-tunnel predicted a measurement error of the order of 20% in the magnitude of the velocity vector and 25 degrees in the vector angles.

The agreement between the mathematical model computations and the experimental results was poor. This was due, in part, to the fact that it was not possible to obtain values of throat discharge coefficients to be used in the model. The measured vector angles suggest that the assumption of uniform solid body rotation for the
model may be an over-simplification of the true air motion in the prechamber. It was thus decided to investigate a correlation between the swirl component of the prechamber gas velocity and the engine speed and this was successfully achieved. In addition a factor was defined in a general way such that the mathematical model could be matched to the experimental measurements. The results of the correlation and the correction factor analyses show enough promise to encourage further investigations on these lines,
ACKNOWLEDGEMENTS

The author wishes to express his sincere thanks to:

Dr. J.C. Dent (Reader) for his supervision of the project.

Messrs. G. Manship, S. Taylor and A. Slade of the I.C. Engines Laboratory for their help.

Mr. P. Clayton for his painstaking manufacture of the hot-wire probes and the prechamber assembly.

Messrs. K. Topley and G. Halls for producing the photographic plates appearing in the thesis.

Mrs. J. Smith and Mrs. M. Pugsley for typing the thesis.

The University of Ife, Nigeria, for providing the personal maintenance scholarship throughout the project.

Above all, the author is grateful to his wife, Adefunke, for her understanding and tolerance throughout the period of study.
NOMENCLATURE

a  i) Prechamber throat cross-sectional area (m$^2$)

ii) Constant in prechamber velocity correlation law (Eq.4.3)

A  Main cylinder bore cross-sectional area (m$^2$)

b  Constant in equation (4.3)

b$_l$  Thickness of cylindrical prechamber (m)

C$_f$  Skin friction coefficient

C$_p$  Specific heat at constant pressure

C$_v$  Specific heat at constant volume

CR  Compression ratio

d  Prechamber throat diameter (m)

d$_w$  Diameter of hot wire (m)

E  Bridge voltage (volts)

E$_s$  Sum of kinetic energy (Nm)

f( )  Denotes 'function of'

FM  Correction factor (Equation 4.11)

g  Gravitational constant ($\frac{\text{kg} \cdot \text{m}^2}{\text{N} \cdot \text{sec}^2}$)

h  i) Specific enthalpy (kJ/kg)

ii) Heat transfer coefficient ($\frac{\text{kJ}}{\text{m}^2 \cdot \text{°C}}$)

HA  Velocity vector horizontal angle (degrees)

I  i) Hot wire probe current (A)

ii) Moment of inertia (kg m$^2$)

k  Thermal conductivity ($\frac{\text{kJ}}{\text{m} \cdot \text{s} \cdot \text{°C}}$)
i) Con-rod length (m)

ii) Length of hot wire (m)

m Mass of gas (kg)

M Mach number

N Engine Speed (rpm)

P Absolute pressure (N/m^2)

PR Ratio of prechamber and main cylinder pressures

Q Heat transfer quantity (kJ)

R Radius

R_l Distance between throat axis and centre of prechamber (m)

R_i Radius of gyration (m^2)

R_e Reynold's number

R Gas constant (\frac{kJ}{kg \cdot ^\circ K})

S Piston stroke (m)

SR Swirl ratio

t Time (sec)

T Temperature (^\circ K)

T_1, T_2, T_3, T_4 Friction torques at prechamber wall (Nm)

u Internal energy (kJ/kg)

U Velocity (m/s)

U_swirl Swirl component of velocity (m/s)

V Volume (m^3)

VA Velocity vector vertical angle (degrees)
VCOMP \hspace{1cm} \text{Total volume in engine at TDC} \hspace{0.5cm} (m^3)

V_h \hspace{1cm} \text{Piston swept volume} \hspace{0.5cm} (m^3)

x,y,z \hspace{1cm} \text{General length coordinates}
Greek Symbols

\( \alpha \)    temperature coefficient of resistance \((^\circ\text{C})\)
\( \beta \)    wire material resistivity \((\text{Ohm} \ \text{m})\)
\( \gamma \)   polytropic index
\( \rho \)    density \((\text{kg/m}^3)\)

\( \mu \)
  i) flow coefficient of throat (Eq. 1.5)
  ii) dynamic viscosity \((\text{kg/ms})\)

\( \tau \)    wall friction shear stress \((\text{N/m}^2)\)
\( \pi \)    ratio of circle circumference to diameter
\( \sigma \)   ratio of prechamber volume to compression volume
\( \omega \)   angular velocity \((\text{rad/sec})\)
\( \theta \)   crank angle displacement \((\text{degrees})\)

Subscripts

\( e \)    exit conditions
\( g \)    gas property
\( i \)    inlet conditions
\( m \)    main cylinder property
\( p \)    prechamber property
\( s \)    wire support property
\( \text{th} \)   prechamber throat property
  i) wall property
  ii) hot wire property
Abbreviations

BDC  bottom dead centre
TDC  top dead centre
BTDC before top dead centre
ATDC after top dead centre
IVC  inlet valve closure
DC   direct current
ESBRV equivalent solid body rotational velocity
ADC  analog to digital converter
INDEX OF CONTENTS

SUMMARY i - ii
ACKNOWLEDGEMENTS iii
NOMENCIATURE iv

1.0 INITIAL DISCUSSION AND SURVEY OF PUBLISHED LITERATURE 1 - 36

Initial Discussion 1
Summary of Published Literature 5

1.1 Mathematical Analyses of Air Motion in the Prechamber of a Diesel Engine 5

1.2 Summary of Methods of Velocity Measurement in Internal Combustion Engines 11

1.2.1 Direct methods 11
1.2.2 Optical methods 12
1.2.3 Flow sensors 12
1.2.4 Other methods for measuring air velocities in engines 12

1.3 Gas Velocity in a Pre-Chamber Engine. Summary of Published Experimental Results 19

1.4 Exhaust Emission from Swirl-type Diesel Engines 26

1.5 Conclusions to the Literature Survey 27

Figures 1.1 - 1.19 28

2.0 A MATHEMATICAL MODEL FOR THE PREDICTION OF AIR MOTION IN THE PRECHAMBER OF AN ENGINE 37 - 64

Introduction to the Mathematical Model 37

2.1 Analysis of Air Motion in the Prechamber During the Compression Period 39

2.1.1 Gas velocity in the connecting passage. Mass transfer based on the instantaneous ratio of prechamber volume to total volume 39

2.1.2 Throat velocity obtained from the energy equation 40
2.2. Prechamber Velocity Analysis - Conservation of Angular Momentum 44

2.3 Friction Effects at the Prechamber Wall 45
   2.3.1 Cylindrical prechamber
   2.3.2 Spherical prechamber

2.4 Angular Momentum Equations for Air Motion in the Prechamber 48
   2.4.1 Compression period
   2.4.2 Expansion stroke

2.5 Discharge Coefficient for the Prechamber Throat - a General Discussion 49

2.6 Comparison of the Energy and the Proportional Mass Transfer Solutions for the Throat Velocity 51

2.7 Effects on the Throat Velocity of Varying Some Parameters in the Theoretical Solutions 53

2.8 Conclusions to the Mathematical Model 54

Figures 2.1 - 2.19 56

3.0 EXPERIMENTAL TECHNIQUE AND ERROR ASSESSMENT 65 - 111

Introduction 65

3.1 The Theory of Three-Wire Operation 66
   3.1.1 Magnitude of the velocity vector in a three-wire measuring system
   3.1.2 Direction of the velocity vector

3.2 The Design and Manufacture of a Three-Wire Probe and Traversing Mechanism 69

3.3 Three-Wire Probe Calibration in Wind Tunnel 71
   3.3.1 Hot-Wire anemometer bridge(s) used to operate three-wire probe:
       i) Heat transfer theory of a hot-wire.
       ii) Hot-wire anemometer circuit.
   3.3.2 Physical and thermal properties of sensing wire
   3.3.3 Sensing wire resistance at ambient temperature
3.3.4 Bridge voltage output versus velocity: calibration procedure for three-wire probe

3.3.5 Temperature factor for velocity measurement in an engine

3.4 Velocity Measurement Error Analysis for the Three-Wire Probe

3.4.1 Verification of the 'Sine Law'

3.4.2 Error in the velocity vector

3.4.3 Error in the velocity magnitude: wind tunnel velocity prediction error - rotation of three wire probe through 360 degrees

3.4.4 Error in the direction of the velocity vector

3.4.5 Velocity detected by a wire lying along the direction of flow.

3.4.6 Discussion of the error analysis

3.5 Engine Data Acquisition

3.5.1 Timing mark

3.5.2 Recording of data on magnetic tape in analog form
   i) Velocity components record
   ii) Gas pressure and temperature record
   iii) Calibration of the tape recorder channels

3.6 Processing of Engine Data

3.6.1 Digitisation of the velocity and pressure signals

3.6.2 Tape recorder calibration factors to be used with digitised data.

3.6.3 Computer programs for calculating the engine velocity

Figures 3.1 - 3.25
4.0 A COMPARISON OF THE MATHEMATICAL MODEL WITH EXPERIMENTAL MEASUREMENTS AND DISCUSSION OF THE RESULTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page Nos.</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.1 Introduction</td>
<td>112</td>
</tr>
<tr>
<td>4.2 Experimental Program</td>
<td>112</td>
</tr>
<tr>
<td>4.3 Absolute Velocity Vectors in the One-Inch and the Two-Inch Diameter Cylindrical Prechambers</td>
<td>114</td>
</tr>
<tr>
<td>4.3.1 Comparison of absolute velocity vectors measured with the probe set at three different orientations</td>
<td>112</td>
</tr>
<tr>
<td>4.3.2 Effect of measuring position around cylindrical prechamber circumference on the measured velocity vectors</td>
<td>114</td>
</tr>
<tr>
<td>4.3.3 Radial variation in the absolute mean velocity vector</td>
<td>112</td>
</tr>
<tr>
<td>4.4 Swirl Components of Velocity in the One-Inch and Two-Inch Cylindrical Prechambers</td>
<td>120</td>
</tr>
<tr>
<td>4.4.1 Comparison of the swirl velocities at the measuring positions A, B and C</td>
<td>120</td>
</tr>
<tr>
<td>4.4.2 Radial profiles of swirl velocity in the one-inch and the two-inch cylindrical prechambers</td>
<td>120</td>
</tr>
<tr>
<td>4.5 Comparison of the Mathematical Model Computations with the Experimental Results of the Present Study</td>
<td>122</td>
</tr>
<tr>
<td>4.6 Comparison of the Mathematical Model Computations with Other Published Experimental Measurements of Mean Velocity in the Prechamber of Indirect Injection Diesel Engines</td>
<td>124</td>
</tr>
<tr>
<td>4.7 Some Comments Arising from the Comparison of the Mathematical Model with Present Experimental Measurements and with Previous Published Experimental Data by Other Authors</td>
<td>126</td>
</tr>
<tr>
<td>4.8 A Correlation Between Mean Air Velocity in the Prechamber and Known Engine Dimensions and Speed</td>
<td>128</td>
</tr>
<tr>
<td>4.9 An Empirical Factor Enabling the Matching of the Mathematical Model with the Experimental Measurements</td>
<td>131</td>
</tr>
</tbody>
</table>

Figures 4.1 - 4.40 | 135
5.0 CONCLUSIONS TO THE PRESENT INVESTIGATION AND SUGGESTIONS FOR FURTHER WORK  

<table>
<thead>
<tr>
<th>Section</th>
<th>Page Nos.</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.1 Conclusions to the Present Investigation</td>
<td>170</td>
</tr>
<tr>
<td>5.2 The Practical Value of the Present Study</td>
<td>173</td>
</tr>
<tr>
<td>5.3 Suggestions for Further Work</td>
<td>174</td>
</tr>
</tbody>
</table>

Appendix 2A Computer Program 'THRVEL' of the Mathematical Model  

Appendix 3A Error in Gas Temperature Measurement Using a Wire of Small Aspect Ratio  

Appendix 3B Computer Program 'ENGDATA' for Calculating Measured Prechamber Mean Velocity  

Appendix 3C Computer Program 'ESBRV' for Calculating the 'Equivalent Solid Body Velocity'  

REFERENCES  

Page Nos.: 170 - 175

<table>
<thead>
<tr>
<th>Page Nos.</th>
</tr>
</thead>
<tbody>
<tr>
<td>176 - 195</td>
</tr>
<tr>
<td>196 - 199</td>
</tr>
<tr>
<td>200 - 215</td>
</tr>
<tr>
<td>216 - 223</td>
</tr>
<tr>
<td>224 - 227</td>
</tr>
</tbody>
</table>
CHAPTER 1

INITIAL DISCUSSION AND SURVEY
OF PUBLISHED LITERATURE
INITIAL DISCUSSION

Many experimental and theoretical investigations have been carried out in the attempt to fully understand the combustion process in a diesel engine. It has been possible to isolate the most important physical and chemical factors affecting the combustion of fuel in internal combustion engines. In this respect the mechanism of the burning of a fuel droplet, the concept of physical and chemical delays and the effects of combustion chamber shape have received some attention. The sequence of combustion has been photographed by different techniques and an overall qualitative picture of the occurrences during combustion presented.

However, it is difficult to make detailed investigations of particular aspects of combustion in diesel engines. There are great difficulties in obtaining measurements in an engine under firing conditions, and the complexities of the mixing of fuel and air and of combustion products make an accurate mathematical description even more difficult.

Because of the continuing desire for more economical engines and reduction in environmental pollution, there is active research into combustion in diesel engines. Despite the many investigations that have taken place, the details of combustion in diesel engines are not yet clearly understood. Among these is the air-motion in the pre-chamber-type of diesel engine. In this context the pre-chamber engine is taken as the type in which the swirl in the separate combustion chamber (pre-chamber) is the main effector of the mixing of air and fuel. All the fuel is injected into the pre-chamber and most of the combustion takes place in the pre-chamber. A narrow throat links the main cylinder and pre-chamber, see Fig. 1.1.
The importance of being able to control the air motion in both direct injection and divided chamber types of engine has been recognised for many years. It is widely accepted that, in a pre-chamber diesel engine, the shape, relative volume, throat area and other design factors in the pre-chamber system exert a greater influence on combustion than small variations in the rate of fuel injection or the beginning of fuel injection (1,2,3). Swirl or rotational motion of the air charge within the combustion chamber has long been used in diesel engines as a means of effecting fuel distribution particularly in the small high-speed engine. Air movement is needed both in petrol and in diesel engines but there is an important difference. In the ready-mixed charge of the petrol engine, the air-flow is required to spread the combustion flame, whilst in the diesel engine, the temperature in the combustion chamber is high enough to effect ignition throughout the chamber. In this case the air motion is required to achieve the mixing of fuel and air such that burnt gas is swept away from the bigger unburnt fuel droplets. We see that it is desirable to be able to generate, control and utilize air swirl to obtain optimum combustion efficiency and performance from diesel engines.

There are two ways of producing air swirl in a diesel engine:

(a) Induction Swirl is produced by the oblique air inflow to the cylinder during the induction stroke. The obliquity is generally produced by an inlet valve shroud or by oblique inlet ports in the cylinder head.

(b) Compression Swirl is produced during the compression stroke by forcing the air charge through a restricted passage.
In a pre-chamber type of diesel engine the main form of air motion in the pre-chamber is the swirl due to compression. In the main cylinder the Induction Swirl may be significant and, after the initiation of combustion, there is some combustion induced swirl as a result of the contents of the pre-chamber discharging through the throat.

It is suggested that, while some degree of swirl is desirable for efficient combustion, there is an optimum Swirl Ratio. In this study, Swirl Ratio is defined as the ratio of the swirl r.p.m. of contents of chamber to the engine speed. If the swirl is too weak, combustion is incomplete, if too strong, there is over-mixing also leading to inefficient combustion and also excessive heat loss which may cause difficult starting from cold (3).

In the last few years the problem of environmental pollution by engine exhaust gases has commanded increasing attention. It is considered that the air motion, in the way it influences the evaporation and mixing of fuel, can have considerable influence on the combustion process and hence on the composition of the exhaust gases. It has also been shown that the quenching in the main chamber of the hot gases discharging from the pre-chamber may be a very important factor controlling the formation of oxides of nitrogen (4,5).

We note that the two main areas where a thorough knowledge of air motion in a diesel engine is necessary are in (1) improving engine performance by improving combustion efficiency, (2) controlling the pollutants in the engine exhaust gases. Most of the investigations carried out up to date have provided a qualitative assessment of combustion in the diesel engine. The difficulties of measurement have made reliable quantitative results hard to obtain and there has not been much correlation of theoretical and experimental investigations.
It would be beneficial to have a mathematical model available to the design engineer to predict air motion and how it affects the performance of the diesel engine. There needs to be many more theoretical and experimental investigations before a full understanding of the complex air motion in a pre-chamber diesel engine is obtained.
SUMMARY OF PUBLISHED LITERATURE

As was pointed out above, although there have been many investigations into the various combustion phenomena, there are only a limited number of investigations into the particular aspect of gas velocity distributions in both the main cylinder and the pre-chamber.

1.1 Mathematical Analyses of Air Motion in the Pre-Chamber of a Diesel Engine

The earliest available published work on air swirl in a pre-chamber type of diesel engine is by Alcock in 1934 (1). His paper was concerned with the production and utilization of air swirl in diesel engines. He gave an analysis of compression-produced swirl in a spherical pre-chamber on the following basis:

(a) During the compression stroke the total trapped mass in the engine remains constant.
(b) The pressure and temperature in the engine is uniform at each crank angle.
(c) There is no friction effect in the main cylinder, throat and pre-chamber.
(d) The throat is tangential to the pre-chamber.
(e) Momentum is conserved in the pre-chamber.

Alcock considered a compression-swirl chamber as shown in Fig. 1.1.

As the piston travels up from BDC gas is transferred into the pre-chamber. The mass of gas in each of the two chambers (pre-chamber and main cylinder) at any instant is directly proportional to the relative volume, the gas properties (pressure, temperature and density)
being the same in both spaces. It is thus possible to calculate the instantaneous throat velocity. The resulting increase in the angular momentum in the pre-chamber leads to an increase in the swirl velocity of the gas mass. At the end of the compression stroke the angular velocity of the gas in the pre-chamber, $\omega_p$, is given by:

$$\omega_p = \frac{V_p N}{a} \frac{R_p}{R_I^2} \frac{1}{\int_0^\pi \frac{f_1^2(\theta)}{f_2^2(\theta)} d\theta}$$  \hspace{1cm} \ldots \ldots 1.1$$

where

$K = \text{constant}$

$V_p = \text{pre-chamber volume}$

$N = \text{engine r.p.m.}$

$a = \text{throat cross-sectional area}$

$R_p = \text{pre-chamber radius}$

$R_I = \text{radius of gyration of the air mass in the pre-chamber}$

$f_1(\theta) = \text{function of piston travel}$

$f_2(\theta) = \text{function of piston velocity}$

$\theta = \text{crank angle from BDC (radians)}$

The swirl ratio, SR, is given by

$$SR = \frac{\omega_p}{2 \pi N}$$  \hspace{1cm} \ldots \ldots 1.2$$

The swirl ratio is dimensionless and could be used in comparing air motion in different types of engines.

Knight (6) in 1964 published a paper part of which was a mathematical analysis of air motion in a pre-chamber diesel engine. Knight's paper was a report of his investigations carried out in order to obtain heat transfer estimates in predicting diesel engine cycle
efficiencies, compression temperatures for starting, noise and nozzle cooling. The relevant part here is the effect of gas velocities on the heat transfer coefficients. It was necessary to obtain the mean velocities in the main cylinder and the pre-chamber. The net kinetic energy in each chamber was obtained by adding the kinetic energy of inflowing gas and subtracting the kinetic energy of the outflowing gas mass. The gas main stream velocity is calculated from the mean kinetic energy per unit mass of gas in each chamber.

Knight made the same assumptions as Alcock regarding uniform pressure and temperature at each crank angle, and the total trapped mass remains constant during the compression stroke. However, Knight allowed for the loss of energy by wall friction by using pipe friction coefficients. A simplified form of operation of the engine valves is supposed in calculating the gas velocities. The gas in the cylinder is retained on the firing stroke until BDC. From there to the next TDC the gas is supposed to be at atmospheric pressure and at the exhaust temperature. From TDC induction to the BDC of the induction stroke the gas is taken to be air at atmospheric pressure. The gas velocity calculation is started at the beginning of the induction stroke. Total kinetic energy retained at this time is ignored because it is small in relation to the maximum kinetic energy during the cycle. The gas velocities in the main chamber, the throat and the pre-chamber are calculated on the basis outlined above. The kinetic energy quantities are calculated under the following headings:

(i) Induction induced air motion.
(ii) Fuel-injection induced air motion.
(iii) Piston induced air motion (squish energy).
(iv) Energy induced by transferring gas to and from the pre-chamber.
(v) Kinetic energy loss from pre-chamber due to outflow of gas during expansion.
(vi) Energy loss due to surface friction at the walls.
The mean kinetic energy in each chamber at any crank angle is obtained by summing the increments (i)-(vi) above so that

\[ E_s = \sum_i^\Theta (\int_0^\Delta E) \quad \ldots \quad 1.3 \]

and the mean velocity, \( U_s \), in the chamber given as

\[ U_s = \frac{2 E_s}{m_s} \quad \ldots \quad 1.4 \]

where

\[ m_s = \text{instantaneous mass in the chamber.} \]

Jankov (7) published a mathematical analysis of air motion in a pre-chamber engine. He set up a system of differential equations for the compression stroke. His equations took into account the heat transfer across walls. Jankov considered that it is not of much benefit to solve his equations exactly because of the uncertainties of wall and gas temperatures and pressure at the beginning of compression. He thus proceeded to solve his equations by making simplifying assumptions:

(a) the pressures and temperatures in the main and swirl chambers are equal at the beginning of compression.

(b) the change of state of the gases in the cylinder, pre-chamber and throat proceeds adiabatically with equal and constant specific heat.

(c) the 'trapped mass' remains constant during compression.

(d) there is no friction and the laws of ideal gases apply.

Jankov solved the energy equations to obtain, among other values, the pressures in main cylinder and pre-chamber and the gas velocity in the connecting passage.
The effects of changes in the following parameters were investigated:

(i) compression ratio,
(ii) the crank angle at which compression begins,
(iii) the ratio of crank radius to connecting-rod length,
(iv) the ratio of pre-chamber to total combustion volume at TDC,
(v) the adiabatic exponent, and
(vi) the characteristic quantity \[ C = \frac{V_h N}{30\mu a} \sqrt{\frac{288}{T_0}} \]  \[ \ldots \ldots \ldots 1.5 \]

where

- \( V_h \) is swept volume,
- \( N \) is engine r.p.m.,
- \( \mu \) is flow coefficient of passage,
- \( a \) is throat cross-sectional area,
- \( T_0 \) is air temperature at the beginning of compression.

Jankov presented his many results diagrammatically. His conclusions can be briefly summarized as below:

(1) the adiabatic exponent has little effect on the trend and maximum values of the flow quantities (except for the pressures).

(2) the effect of moderate variation in the beginning of compression and of \( \lambda \) are negligible.

(3) the volume of air flowing through the throat increases more than proportionally with increasing \( \sigma \), the effect being greater at smaller compression ratios.

(4) calculations performed for an engine with a spherical combustion chamber show that the fuel spray tip is deflected from the jet axis linearly with pre-chamber air swirl and that with this type of combustion chamber the throat size influences mixture formation.
The analyses of Alcock, Knight, and Jankov represent the two main approaches to the problem of providing a mathematical description of air motion in the pre-chamber of a diesel engine, viz: (1) Momentum Conservation and (2) Energy Conservation.

At first sight it may appear that the energy analysis will be the more complete but, on close inspection, it will be found that an exact solution to the energy balance is near impossible. Ideally, the energy balance will include the transfer of kinetic energies due to induction and gas transfer to the pre-chamber during compression and from the pre-chamber during the expansion stroke. It may also be necessary to account for the squish energy due to piston motion during compression. The above quantities are small compared with the two major energy quantities of heat transfer across chamber walls. An attempt to convert the kinetic energies into heat energies and thus determine the air velocity in the chamber produces unrealistic results. If the kinetic energies are neglected, there remains the problem of relating wall heat transfer coefficients to the velocity pattern in the combustion chamber. Knight's (6) consideration of only the kinetic energy quantities appears to be an over-simplification of the problem and thus not acceptable. It is therefore noted that, although Jankov set out energy equations to solve for the throat velocity, he did not extend them to obtain gas velocities in the pre-chamber. His solution for the throat velocity is made less attractive by the introduction of a throat flow coefficient which has not been exactly defined and so cannot be used in a general way.
1.2 Summary of Methods of Velocity Measurement in Internal Combustion Engines

The flow inside an internal combustion engine is non-steady because of the periodic motion of the piston. This fluctuation of flow is also accompanied by large changes in the density and temperature of the working gases which result in complications to the problem of gas velocity measurement inside the engine combustion chamber.

The methods for measuring these non-steady velocities may be divided into three main categories:

(a) "Direct" methods,
(b) Optical methods,
(c) Flow sensors.

1.2.1 (a) "Direct" Methods

These are the earliest methods of swirl velocity measurements. There are two types. In the first one a spindle carries a freely rotating vane. The spindle is connected to a counting mechanism which indicates the vane speed. This method of the freely rotating vane has been used by Fitzgeorge et al. (8) and Alcock (1). The method cannot give an instant measure of gas rotational speed. The indicated vane speed is a mean taken over a length of time greater than one engine cycle. However, the rotating vane method may be used to compare the swirl velocities in different engines with similar combustion arrangement.

In the second "direct" method the force on a fixed vane is used to actuate a pencil working on an indicator drum (9). This fixed-vane method has two important advantages over the rotating vane method, (1) it can be used in cases where the swirl axis is occupied by an obstruction,
it gives a continuous record thus indicating the swirl variation during a cycle. However, it is difficult to allow for the swirl-reducing effect of the fixed vane and the inertia of the moving parts would affect the swirl records at high engine speeds. The measurement of velocity by the above two "direct" methods is not accurate but they give the order of magnitude and can still be useful tools in the preliminary estimation of swirl ratios.

1.2.2 (b) Optical Methods

There are two main phenomena employed in the optical methods for measuring air velocity in engines.

(i) Tracer Technique

An indicator is introduced into the flow to make the flow pattern visible or observable by a suitable detecting apparatus outside the field of flow. This method includes the use of smoke for flow visualization. The results obtained by this tracer method are generally qualitative.

(ii) Methods making use of changes in the density of the gas

The methods which come under this heading detect changes in gas density whilst passing a light beam through the gas. These methods may depend on one of the following physical phenomena:

(1) the speed of light is related to the refractive index of the medium through which it passes. The refractive index of the gases in an engine can be related to the density, therefore this phenomenon of detecting changes in density can be used to determine velocity values.

(2) light passing through a density gradient in a gas is deflected as though by a prism. The interferometer is based on phenomenon (1). It measures the density changes directly and is suited to quantitative determination of the density field from which
the velocity pattern can be derived. The Schlieren technique is based on the phenomenon (2) and measures the changes in the density gradient in the gas. It is theoretically possible to interpret the results quantitatively but its main use is to provide a qualitative picture of the flow pattern.

(3) the Shadowgraph method measures the second gradient of density. It is therefore suitable in a situation where large variations of the density gradient occurs.

Of these three optical methods the interferometer gives the most useful results but it is expensive to manufacture and difficult to operate. In the case of engine measurements, this method requires expensive quartz windows.

The optical methods discussed have been widely used in the study of the combustion process in internal combustion engines - Lyn and Valdmanis (10); Allan and Khin (11); Nagao and Kakimoto (12). Most of the results obtained provide an insight to the air motion. The three dimensional nature of gas motions in engines make it difficult to ascribe magnitudes to the results obtained by such optical methods.

1.2.3 (c) Flow Sensors

In this category are Pitot Tubes, Hot-Wire Anemometry and the technique of utilizing the movement of ion particles introduced into the flow by a capacitive discharge.

Ower (13) has given various designs of Pitot tubes for flow measurements. The Pitot-Static differential pressure is fed to opposite sides of a sensing diaphragm of an electronic transducer. This necessitates having a relatively long length of narrow tube and a comparatively large chamber at the diaphragm. This can lead to
significant errors. Moreover the response of the pitot tube is poor and this makes it unsuitable for measuring transient flow as occurs in an engine.

In general the basic requirements of a flow sensor, if it is to be suitable for gas motion measurements in an engine, are:

(i) the sensor should cause minimum possible disturbance to the flow by its presence,
(ii) it should be sufficiently rugged to withstand the arduous conditions of operation inside the combustion chamber,
(iii) it must have a high spacial resolution,
(iv) it must have a very rapid response so as to keep in step with the rapidly changing state of the gases,
(v) it must have a high sensitivity,
(vi) it must give repeatable results.

The hot-wire anemometer, in comparison with the pitot tube, meets most of the requirements listed above. The main drawback is the fragile nature of the wire.

There are two main methods of operation of the hot wire anemometer - Constant Current and Constant Temperature. In the constant-current type, the changes in resistance provide a measure of gas velocity while, in the constant-temperature type, the changes in current, necessary to maintain the wire at a set temperature (or resistance), is a measure of the flow velocity. The constant-current hot-wire anemometer is unsuitable for an application where large fluctuations in the mass-flow-rate are likely to occur. Therefore in the case of gas velocity measurements inside the engine combustion chamber, where such fluctuations are present, the constant-temperature hot-wire anemometer is more suitable.
Hassan (14) has given a comprehensive discussion of the constant-temperature hot-wire anemometer. His discussion includes methods of compensating for gas temperature fluctuations. He also highlights the requirements of manufacture and calibration of such a hot-wire system.

Many investigators have used the constant-temperature hot-wire in measurements of air velocities in internal combustion engines or in models – Pearson (15), Lobo (16), Horvatin (17), Dent & Derham (18), Heubner & MacDonald (19), and Tindal et.al. (20). These and other investigators have shown that the hot-wire is an acceptable method for measuring both the magnitude and angular vectors of air velocities in internal combustion engines.

1.2.4 (d) Other Methods for Measuring Air Velocities in Engines

There are some less widely used methods that have been employed to measure air velocities in engines. Nakajima et.al. (21) applied high frequency high voltage electric pulses to the spark gap in the combustion chamber. The swirl motion of air was photographed by using the plasma so created as a light source.

The principle of measurement is shown in Fig. (1.2). The high frequency high voltage pulses creates a plasma of high conductivity across the electrodes. Once this plasma has been created, the succeeding discharges across the electrodes will show the movement of the plasma which is the movement of the air. Several electric pulses are supplied to the electrodes for a definite time and a photographic film is exposed at the same time to flashes of light caused by the sparks. The striped patterns obtained show the movement of air in equal time intervals, Fig. (1.2), and from these striped figures, the air velocity can be calculated.
Another method that has been used recently is based on the phenomenon that the path of the inductive discharge after a capacitive discharge moves downstream under the influence of the gas stream. This technique has been described and used by Ohigashi et al. (22) to make air velocity measurements in an engine under both motoring and firing conditions. An electric discharge is initiated at an electrode. Ion clouds are created by this discharge providing a path for succeeding discharges. A probe downstream of the electrode indicates a rapidly increasing voltage when the discharge current reaches it. By measuring the time between the initiation of a discharge and when the probe, which is at a fired distance downstream, senses the ion current, the gas stream velocity can be obtained from the previous calibration of the probe. Ohigashi et al. claim that the main advantage of this technique is that it can be used during a firing run in a compression ignition engine.

We notice that the two methods of Nakajima et al. (21) and Ohigashi et al. (22) are similar except that, in one case, the final determination of velocity is based on photographic records of ion/spark movements whereas, in the second case, a sensor probe is used. In comparison with the hot-wire anemometer, the two obvious limitations common to these two methods are: (1) they are less flexible and less suitable to determine the angular vectors of velocity, (2) the requirements of high voltages may result in the probes being large so that they cause significant disturbance to the flow to be measured in an engine.

One recent method of air velocity measurement in an engine utilizes the Karman Vortex Shedding Frequency as a velocity sensor. Chen et al. (23) describe their application of this in an internal combustion engine cylinder. They stress that the quality of data
obtained depends on the relative frequency distribution of the free-stream turbulence and of the vortex shedding induced by the vortex generator. The difficulty of separating the shedding signal from turbulence noise is a major limitation for this method. Its main advantage is that the measured velocity is independent of gas property variation.

Another recent method of flow measurement that may be used in an engine is the "Laser Photon Correlation Anemometer". This method has been shown suitable for the measurement of turbulent flows (24). The principle of operation depends on the detection of particles in the flow as the particles cross an interference pattern set at the measuring point by the intersection of two laser beams. The output of the photon detector is digitally correlated (multiplying the pulse train with many different time delayed versions of itself). The output of the correlator is in the form of a cosine wave. The time between peaks (which is the time the particles take to travel between fringes) can be accurately measured against a crystal clock in the correlator. The velocity of flow can thus be determined from a knowledge of the fringe spacing and the correlated time.

Fig. 1.4 illustrates the operation of this method. It is said to have the capability of measuring velocities of up to 500 m/s to an accuracy better than 1% (25). It also has the advantage of not causing any disturbance to the flow. However, one main drawback is the inability to provide a continuous record of transient flow as in the pre-chamber of an engine.

This brief survey has shown the different methods that have been used to measure air velocity in engines. Fig. 1.5 is a summary of these various methods. Each of these methods has its advantages and disadvantages which must be weighed together.
In the present study, it is considered that the vane-type methods are inaccurate and inadequate in their spatial resolution. The optical methods are most useful in cases where all that is required are qualitative details of gas motion. It is very difficult and unreliable to obtain velocity vectors from the optical records. They are also expensive. The less common methods of spark discharge and vortex shedding suffer from the difficulty of interpretation and the main fact that they do not readily give angular vectors of velocity.

As was mentioned above many investigators have demonstrated the suitability of the hot-wire probe for the measurement of air velocities in the combustion chamber of internal combustion engines under motoring conditions. It meets most of the requirements in terms of accuracy, repeatability, spacial resolution and flexibility. However, it requires consistent calibration and compensation for variation in gas temperature. Hot wire anemometry has been used for several years here at the Mechanical Engineering Department of Loughborough University.

The experience gained during this time confirms that the hot wire probe is suitable for the type of measurements to be made in the present investigation. We have decided to adopt the hot-wire technique for our velocity measurements in the present study.
1.3 Gas Velocity in a Pre-Chamber Engine.

Summary of Published Experimental Results

There have not been a large number of published reports of velocity measurements in the pre-chamber of pre-chamber-types of engine. This has been mainly due to the difficulties of measurement.

Alcock (1), using a vane-type swirl meter, had suggested that the air mass in a pre-chamber rotates as a free vortex. This has been disputed by other investigators (2,10,21) who suggest either a forced vortex type of rotation or a combination of forced and free vortices.

Bryan (2) reported a photographic study of combustion in a single-cylinder diesel engine. The photographs were taken through windows set in the pre-chamber walls. He observed the winnowing effect of the air swirl on the fuel spray. Air motion is mainly solid body rotation.

Lyn and Valdmanis (10) used high speed Schlieren photography to study the combustion in a diesel engine. The emphasis was on technique rather than on the results. They gave descriptions of the air movement in the pre-chamber and of the mixing and evaporation process. There is very little movement of air in the pre-chamber during the suction and exhaust strokes except for a short time during valve overlaps. Air starts to flow into the pre-chamber towards the end of the suction stroke and the flow increases gradually during the compression stroke. However, significant movement of air in the pre-chamber really begins at about 60 degrees before top-dead-centre. At this stage, air turbulence patterns (eddies) begin to form within the air stream as it enters the chamber and the area covered by such eddies grows rapidly until at about 20 degrees B.T.D.C., it uniformly covers the
whole field. The size of the eddies is of the order of 1 mm and their appearance is similar to those produced in the wake behind a shock wave. The boundary layer as far as can be judged from the cinematograph is of the order of 2 mm. Swirl ratios are estimated and plotted against crank angle. Variation of air swirl speed with radius of pre-chamber is also estimated (Figs. 1.6, 1.7). It appears that the air mass rotates as a forced vortex, but with a large scale mixing motion superimposed on the mean motion, rather than as a free vortex as had been suggested by Alcock (1). The mechanism of mixing and evaporation observed from the pictures indicate that air mixes with the fuel droplets at the boundary of the jet. The swirling air stream carries away very thin layers of fuel from the boundary of the fuel jet. These thin layers then evaporate and mix with the air mass. Lyn and Valdmanis calculated that, in their engine, pressures at the end of compression were above the critical point values of most hydrocarbons so that the transition of the fuel spray from liquid to gas was therefore gradual and difficult to observe. Although the technique of Lyn and Valdmanis cannot alone give conclusive quantitative results, it nevertheless gives a good picture of the type of air motion which occurs in a pre-chamber-type diesel engine. This could be useful in formulating a mathematical model for such engines.

Nagao and Kakimoto (12) also used the technique of high-speed photography to investigate swirl and combustion in divided combustion chamber types of diesel engine. Measurements of pressure variations and photographs of fuel spray and flame patterns were obtained from specially designed combustion chambers. An attempt was made to determine an optimum direction of fuel injection. It was concluded that fuel injection against the direction of swirl results in slow combustion and heavy soot formation. The maximum power output, specific fuel consumption and exhaust smoke are markedly improved with fuel
injection along the swirl direction, see Fig. 1.8. The effect of the swirl chamber shape was investigated. A chamber having a flat bottom was compared to a spherical chamber. The influence of the fuel injection direction on performance is small in the flat-bottomed chamber and the optimum position of fuel injection is central. (Fig. 1.9). Evidence from the photographs suggests that it is preferable to have an over-rich mixture around the periphery of the pre-chamber rather than at the centre since an over-rich mixture at the centre may lead to inferior fuel consumption and black exhaust. For this reason, excessive swirl in the pre-chamber should be avoided.

The study of Nagao and Kakimoto has concentrated on performance tests and does not include air velocity measurements. However, it has raised the important problem of understanding the mechanism of fuel evaporation and mixing and their influence on the combustion process.

Nagao et al. (3) carried out a study of the air-swirl and its effects on mixture formation and the subsequent combustion. Swirl characteristics were studied in performance tests, the results of which were evaluated by means of a water model technique together with high-speed photography.

For the performance tests, the engine used was a four-stroke single-cylinder diesel engine. (Bore = 95 mm, Stroke = 115 mm, Compression Ratio = 16:1, Engine Speed = 1500 r.p.m., Throat Area/Piston Area = 1.0%, Volume of Pre-chamber/Compression Volume = 51%).

The effects of variations in the inclination of the throat at entry into the pre-chamber was investigated. The different throat directions investigated are shown in Fig. 1.10. The tests show that optimum performance with respect to smoke levels and specific fuel
consumption is obtained when the throat direction is away from the pre-chamber wall towards the centre of the chamber although performance deteriorates when the direction is directly towards the centre.

The effect of varying the direction of fuel injection is shown in Fig. 1.12. It shows that the lowest smoke level and specific fuel consumption is achieved for the case in which the direction of fuel injection is downstream of the solid swirl rotation. However, it should be appreciated that, if the direction of the fuel injection is too near the pre-chamber wall, the jet may become attached to the wall (Coanda effect) thus leading to poor evaporation and mixing and consequently high smoke levels and s.f.c.

It was found that a projection of suitable dimensions placed upstream of the fuel injector led to improved specific fuel consumption. There was no corresponding improvement when the projection was placed downstream of the fuel injector.

Nagao et.al. also developed a water model in an attempt to analyse the air motion in a pre-chamber engine. The basis of simulation is the equality of Mach No. and Froude No. Also, the surface area ratio of pre-chamber to total surface area at maximum compression was made equal to the volume ratio in the real engine. The water flow patterns were rendered visible by using aluminium powder as tracer. Results, estimated from photographs, of the flow patterns for different conditions of the swirl chamber are shown in Fig. 1.13. The length of each curved line is a measure of the mean velocity for the period 20° to 15° B.T.D.C. An examination of the flow patterns shows that off-setting the throat position from the tangential position results in a decrease in swirl velocity. Projections on the pre-chamber wall both upstream and downstream reduces swirl velocity and causes secondary swirls. It appears that, by a
simultaneous adjustment of throat area and inclination, of projection on the pre-chamber wall, and of the size and shape of the pre-chamber, it is possible to control swirl such that combustion is complete and at the same time that effective outflow of gas from the pre-chamber is achieved. However, it should be noted that the above variations will have interacting effects which may be difficult to isolate. For example, a small throat cross-sectional area might give a suitable intensity of swirl in the pre-chamber but may be unfavourable as regards the outflow of gas from the pre-chamber. Also, there is no direct way of determining the optimum dimension of a projection on the wall of the pre-chamber. This has been one of the reasons why engine development has relied on the experimental procedure of "trial and error".

Nakajima et.al. (21) reported an experimental study of air swirl motion in a swirl type combustion chamber. As was mentioned above, in the discussion of methods of velocity measurement, a method was used in which high frequency, high voltage electric pulses generated plasma as a light source for the high speed photographs. The test chamber was cylindrical. One side of it was made of transparent heat resisting glass. The compression ratio for the test engine was 7:1 and 18:1 for the practical engine. Different shapes - spherical, flat-bottomed, bell-type - of pre-chamber were investigated.

From an inspection of the photographs, Nakajima et.al. gave descriptions of air motions in the different types of pre-chambers. For a spherical pre-chamber, Fig. 1.14 is a sample of air motion patterns at particular crank angles. The length of the curved lines correspond to velocity values.
Compared with the spherical type, air velocities in the flat-bottomed and bell-type pre-chambers are lower being lowest in the bell-type. Fig. 1.15a shows the radial distribution of air velocity in various types of pre-chambers near the end of the compression stroke. Fig. 1.15b shows the effect of throat position in a spherical pre-chamber. Performance tests on the practical engine shows that both mean effective pressure and specific fuel consumption have been improved with the throat directed towards the centre of the chamber. The performance of the bell-type is a little better than for the spherical chamber but not as good as for the flat-bottomed chamber. Nakajima et.al. concluded that, as far as engine performance is concerned, a rather weaker strength of swirl is suitable and moderate mixture formation in the swirl chamber and a prompt discharge of unburned mixture into the main chamber is necessary.

Allan and Khin (11) carried out an investigation of combustion phenomena in the swirl chamber of a compression-ignition engine using Schlieren techniques. The Schlieren photographs show in some detail the distinct phases involved in the injection, evaporation, and mixing of the fuel with air. The extent of fuel penetration in relation to local swirl was well demonstrated. Allan and Khin reached the obvious conclusion that the combustion processes in a compression-ignition engine are very complicated. It is easy to visualise that the flow field inside the swirl chamber is indeed complex. The penetration of the fuel jet into the swirling air mass is dependent on the degree of swirl, the fuel being injected directly across the swirl. At a particular injection advance fuel penetration was more at low swirls. When the degree of swirl is low, the fuel jet travels far into the swirl chamber and, due to high centrifugal forces, fuel droplets may be deposited on the pre-chamber wall. This may result in a flame being attached to the chamber wall. With high swirls, the jet is
immediately deflected upon its introduction. As a consequence only one side of the jet was exposed to the swirling air and the jet's break-up and mixing was adversely affected. An over-rich mixing region occupied an annular area close to the swirl chamber wall. The observation of Allan and Khin (11) concerning the penetration of the fuel jet is in broad agreement with Nagao and Kakimoto (12), depending on whether the fuel injection is against or along the direction of swirl. Rife and Heywood (27) have also shown that the deflection of the fuel jet increases with the intensity of the air swirl.

Horvatin (17) used hot-wires to make velocity measurements in the pre-chamber of a diesel engine. He found that for a uniform throat tangential to the pre-chamber, the air motion is a solid vortex in the sixty degrees B.T.D.C. up to T.D.C. He also found that swirl velocity increases with engine speed though the increase is less than linear. For the case of a convergent throat, he found that the high swirl occurs only near the wall and decreases rapidly towards the centre of the pre-chamber. Horvatin's evidence supports the findings of other investigators that air motion in the pre-chamber resembles a forced vortex around T.D.C. of the compression stroke, for the case where the throat is tangential to the pre-chamber periphery. Horvatin's results are shown in Figs. 1.16-1.19.
1.4 Exhaust Emission from Swirl-Type Diesel Engines

In the last ten years, the problem of air pollution by engine exhausts has received increasing attention. Up till the mid-sixties many investigators had considered the diesel engine as a minor contributor to air pollution. This might have been due to the fact that, on the basis of total motor fuel consumed, the diesel engine contributed a small percentage of all air pollutants. However, the increasing number of diesel units operating on the roads and especially in cities has caused more awareness of the pollution risk from diesel engine exhausts. The growing popularity of the pre-chamber type of diesel engine coupled with the expressed desire to limit atmospheric pollution has resulted in more attention being given to the exhaust emission characteristics of these types of engine. It is necessary to ascertain how the control of the combustion processes in a diesel engine can lead to a minimization of pollutants in exhaust products. It is in this respect that it will be useful to understand the influence of air motion on the combustion processes (4, 5, 28).
1.5 Conclusions to the Literature Survey

The following conclusions have been drawn as a result of the Literature Survey:

(1) Although there have been many investigations into the general phenomenon of combustion in diesel engines, there have been only a few reported studies into the particular problem of measuring air velocities in the pre-chamber of the pre-chamber-type of diesel engine.

(2) There are two main approaches to obtaining a mathematical model to predict air velocities in pre-chamber engines - conservation of angular momentum and conservation of energy. Simplifying assumptions have to be made and idealized shapes of pre-chamber assumed. Applying the principle of conservation of angular momentum to the air in the pre-chamber appears to be the more promising of the two approaches because of its sensitivity to the geometry of the pre-chamber and throat.

(3) There are many methods of velocity measurement in engines each with its limitations. The hot-wire anemometer is as reliable as any other method. We have elected to employ the hot-wire technique in the present study, because of accumulated experience with these devices here in the Mechanical Engineering Department of Loughborough University.

(4) A thorough knowledge of the air motion in the pre-chamber of a diesel engine will help considerably in understanding the processes of fuel evaporation and mixing and how this affects engine performance and composition of the exhaust gases.
Fig. (1.1)  Illustrative Sketch of an Indirect Injection Diesel Engine
Fig. (1.3) Velocity Measurement using Electric Plasma as a Tracer (21)

Fig. (1.3) Velocity Measurement using Ions as tracer (22)
RF307 OPTICAL UNIT
Function: to split laser beam into two and to recombine to form crossover at point of intersection.

RF323 DIGITAL PHOTON DETECTOR - Observing Flow

MALVERN K 7023 DIGITAL CORRELATOR

DISTANCE BETWEEN FRINGES
\[ D = \frac{\lambda}{2 \sin \theta} \]

PARTICLES

Magnified illustration of the beam intersection point showing optical fringe pattern caused by interaction between the two beams. Particles contained in the flow cause flashes of light as they pass through the optical fringes.

Flashes of light caused by particles in the flow result in pulse train bursts from the output of the Photon Detector.

In order to extract the periodicity of the pulse train it is necessary to multiply the pulse train with many different time delayed versions of itself, (Digital Correlation)

Form of output from the Correlator where \( T \) defines the time to a particle to pass through one fringe. \( T \) is measured with crystal clock accuracy. Spacing between fringes is defined by geometry of optics.

Hence velocity of flow = \( \frac{D}{T} \)

Fig. (1.4) Velocity Measurement using Laser Photon Correlation Technique
Fig. (1.6) Variation of air swirl speed with radius. (10)

Fig. (1.7) Variation of air swirl speed with crankangle degree. (10)
Methods capable of being used in firing and motoring runs.

- High Speed Photography (2, 12)
- Schlieren and Shadowgraph Method (10, 11)
- Electric Discharge Method (22)
- Laser Photon Spectroscopy (24, 25)
- Rotating or Fixed Vane (1, 9)
- Tracer Technique (21, 26)
- Pitot Tube (13)
- Hot Wire Anemometer (14, 17, 18, 29)
- Vortex Shedding Technique (23)

Methods used in motoring runs

Fig. (1.5) Experimental Methods of Investigation of Gas Motion in Internal Combustion Engines

Fig. (1.8) Comparative performance curves with different injection directions: test engine: Yanmar ST-95 (modified), engine speed: 1500 rpm, area ratio of the connecting passage: 1% of piston area, injection: DNS51. (12)

Fig. (1.9) Influence of fuel injection direction with different types of injector and of swirl chamber; engine speed: 1500 rpm. (12)
Fig. (1.11) $p_1$: brake mean effective pressure
$\eta$: index of exhaust dilution of Bouch eductometer
$\eta_1$: fuel quantity of every firing stroke per unit swept volume
$s$: specific fuel consumption and $c_2$ air-utilization factor
Comparative performance curves with different directions of air-influx into the swirl chamber

Fig. (1.13) Flow patterns in the swirl chamber
Fig. (1.14) Velocity Distribution in a Cylindrical Prechamber (21)

Fig. (1.5) Effects of Prechamber Shape and Throat Location at a Particular Crank Angle (21)
Fig. (1.16) Air Velocity in a Cylindrical Prechamber with a Convergent Throat (17)

Fig. (1.17) Radial Profile of Velocity in a Cylindrical Prechamber with a Convergent Throat (17)
Fig. (1.18) Air Velocity in a Cylindrical Prechamber with a Uniform Tangential Throat (17)

Fig. (1.19) Radial Profile of Velocity in a Cylindrical Prechamber with a Uniform Tangential Throat (17)
A MATHEMATICAL MODEL FOR THE PREDICTION
OF AIR MOTION IN THE PRECHAMBER OF AN ENGINE
CHAPTER 2

INTRODUCTION TO THE MATHEMATICAL MODEL

As a result of the literature survey it was seen that a mathematical model capable of predicting the mean velocity of the gas in the prechamber of an engine would be of value. Previous studies (10, 11) had indicated the complexity of the gas motion in the prechamber of engines so it was obvious from an early stage that simplifying assumptions would have to be made and idealised shapes of prechamber investigated in order to be able to formulate the gas motion in equations amenable to solution. This, of course, places a limitation on such a mathematical model especially in its comparison with experimental measurements, and in its application to non-idealised shapes of prechamber. However, it is considered that a simple mathematical model could still be a useful tool in a parametric investigation of the gas motion in the prechamber of an indirect injection diesel engine.

Previous experimental investigations have suggested that the air motion in a uniformly spherical or cylindrical prechamber tends towards solid body rotation near the TDC of the compression stroke (10, 11, 21). The solid swirl does not become well established until about 60 degrees BTDC and the contents of the prechamber are scavenged by about 40-50 degrees ATDC. In a diesel engine fuel injection usually starts at about 20 degrees BTDC and combustion is nearly complete by about 50 degrees ATDC, in the indirect injection engine. Therefore the parts of the engine cycle that are of most interest as regards significant air motion will be between 90 degrees BTDC to 90 degrees ATDC. In the following analysis the gas motion for the whole of the compression and the expansion strokes will be computed for completeness.
It had to be decided whether to base the analysis of air motion in the prechamber on the principle of conservation of angular momentum or of energy. The main difficulty with an analysis based on the conservation of energy is in relating the air velocity in the prechamber to the heat transfer across the prechamber wall. In addition, the energy solution will be totally insensitive to the inclination of the throat relative to the prechamber curved surface and to the throat position relative to a reference diameter.

Figs. 2.1a and 2.1b illustrate these two points which must have a major influence on the magnitude and direction of the mean velocity in the prechamber. The principle of conservation of angular momentum does not have the disadvantages referred to above and it appears realistic to apply it especially to the case where the throat is tangential to the prechamber periphery.

A prerequisite for any model is the air velocity in the connecting passage linking the prechamber and the main cylinder. Two methods of solution were considered. The first method was outlined by Alcock (1). It was assumed that the separate masses of gas in the prechamber and the main cylinder at any instant during the compression and expansion strokes are proportional to the ratio of each volume to the total volume. The second method was described by Jankov (7). This method considers energy conservation, the piston work and the heat transfer across the cylinder and prechamber walls. The cylinder and prechamber were considered as separate non-steady flow systems, and the unsteady flow energy equation was applied.

Both of these methods were used to obtain the velocity in the prechamber throat during the compression and expansion periods in order to enable comparison of the two methods when applied to the particular prechambers studied in the present investigation.
2.1 Analysis of Air Motion in the Prechamber During the Compression Period

Assumptions

1. The pressure and temperature of the gas in both the main cylinder and the prechamber are uniform, at inlet valve closure.

2. There is no overlap of the inlet and exhaust valves.

3. The total trapped mass remains constant throughout the compression and expansion strokes.

4. The trapped air mass is considered to behave like an ideal gas. The compression and expansion are taken to be polytropic processes with an index of 1.33 assumed.

2.1.1 Gas Velocity in the Connecting Passage

Mass Transfer Based on the Instantaneous Ratio of Prechamber Volume to Total Volume

The equations of flow are set out below using the notation in Fig. 2.2.

Piston travel from BDC, $Z$, is given by:

\[
Z = (S/2 + x) - \ell
\]

\[
x = \sqrt{(l^2 - S^2 \sin^2 \theta)} - \frac{S}{2} \cos \theta
\]

\[
Z = \frac{S}{2} (1 - \cos \theta) + \sqrt{(l^2 - \frac{S^2}{4} \sin^2 \theta)} - \ell \quad \cdots (2.1)
\]

Piston velocity at any crank angle from BDC is given by:

\[
U_\theta = \frac{dZ}{dt} = \frac{\pi}{180} 6 N \frac{dZ}{d\theta} \quad \cdots (2.2)
\]
where \( N \) is engine rpm

\( \theta \) is in radians.

\[
U_\theta = \frac{\pi}{30} N \left[\frac{S}{2} \sin \theta - \frac{S^2 \sin \theta \cos \theta}{4\sqrt{\frac{L}{S} - \frac{S^2}{4} \sin^2 \theta}}\right] \quad \ldots \quad (2.3)
\]

\[
U_\theta = K_1 N S f_1 (\theta) \quad \ldots \quad (2.4)
\]

where \( K_1 = \frac{\pi}{30} \)

The instantaneous total volume above the piston is the sum of the instantaneous main cylinder volume and the compression volume.

\[
V_\theta = A * (S - Z) + V_{COMP} \quad \ldots \quad (2.5)
\]

where \( V_{COMP} \) is the total volume at TDC.

\[
V_\theta = A S \left[\left(\frac{1 + \cos \theta}{2}\right) - \sqrt{\frac{L}{S}} - \frac{S^2}{4} \sin^2 \theta + \frac{L}{S} + \frac{V_{COMP}}{A S}\right] \quad \ldots \quad (2.6)
\]

The rate of gas flow into the prechamber is proportional to the rate of change of total volume above the piston.

\[
\frac{dm}{dt} = \rho_\theta \frac{V_p}{V_\theta} \frac{d V_\theta}{dt} \quad \ldots \quad (2.7)
\]

where \( \rho_\theta \) is the instantaneous gas density.

\[
\frac{dm}{dt} = K_1 N \rho_0 \frac{V_\theta}{A S} \frac{f_1 (\theta)}{f_2 (\theta)} \quad \ldots \quad (2.8)
\]
The mass flow rate through the throat is given by:

\[ \frac{dm_{\text{th}}}{dt} = a U_{\text{th}} \frac{\rho}{\theta} \]  \hspace{1cm} \text{(2.9)}

where \( a \) is throat X-sectional area, but the mass flow rate through the throat must equal the rate of change of mass in the prechamber.

\[ \frac{dm_{\text{th}}}{dt} = \frac{dm}{dt} \]

\[ U_{\text{th}} = K_1 \frac{N V p}{a} \frac{f_1 (\theta)}{f_2 (\theta)} \]  \hspace{1cm} \text{(2.10)}

### 2.1.2 Throat Velocity Obtained from the Energy Equation

The general energy equation for a non-steady flow applied to the main cylinder yields, in differential form,

\[ -dQ_m - P_m \frac{dV_m}{dt} = d(mu)_m + dm_e h_e \]  \hspace{1cm} \text{(2.11)}

where

\[ dQ_m = \sum_{i=1}^{3} h_i A_i (T_m - T_{wi}) \]  \hspace{1cm} \text{(2.12)}

is the sum of the heat transfer across the three bounding surfaces of the cylinder. It may be noted that for solution during the compression period \( \frac{dV_m}{dt} \) is negative in sign and \( dm_e = -dm_m \). For the expansion period \( \frac{dV_m}{dt} \) is positive and the equivalent expression for Eq. 2.11 is:

\[ -dQ_m - P_m \frac{dV_m}{dt} = d(mu)_m - dm_i h_i \]  \hspace{1cm} \text{(2.13)}

where in this case \( dm_i = dm_m \).
For the prechamber during compression:

\[-dQ_p = d(mu)_p - dm_i h_i \quad \text{..... (2.14)}\]

where \(dm_i = dm_p\)

For the prechamber during expansion

\[-dQ_p = d(mu)_p + dm_e h_e \quad \text{..... (2.15)}\]

where \(dm_e = - dm_p\)

The compressible flow equation for gas flow through an orifice when applied to flow in the prechamber throat gives the throat velocity, \(U_{th}\), as

\[U_{th} = \sqrt{\frac{2Y}{(\gamma - 1)} R T \sqrt{1 - (PR) \frac{\gamma - 1}{\gamma}}} \quad \text{.....(2.16)}\]

where \(PR\) is \((P/P_m)\) for compression,

and \((P_m/P_p)\) for expansion

\(\gamma\) is the polytropic index.

The mass flow through the throat in an interval \(dt\) is given by the continuity equation as:

\[dm = (U_{th} a \rho_{th}) dt \quad \text{..... (2.17)}\]

Since it has been assumed that leakage loss of gas is negligible

\[|dm| = |dm_i| = |dm_e| \quad \text{..... (2.18)}\]

although it must be remembered to use the appropriate signs in equations 2.14 and 2.15.
The density of the gas jet from the throat is given by:

\[ \rho_{th} = \rho \ast (PR)^{\gamma} \]

...... (2.19)

The gas law

\[ T = \frac{PV}{mR} \]

...... (2.20)

is taken to apply separately to both the main cylinder and the prechamber.

Apart from the geometric dimensions of the cylinder and prechamber the following data are required for the solution of the equations 2.11 to 2.20:

2. The average wall temperatures.
3. The gas conductivity, viscosity and specific heat as a function of temperature.
4. Expressions for heat transfer coefficient.

Equations 2.11 to 2.20 were solved iteratively to give the instantaneous gas properties - pressure, temperature and mass distribution - in the cylinder and the prechamber and to give the throat velocity \( U_{th} \) to be used later for the conservation of angular momentum analysis in the prechamber.

The heat transfer coefficient relationships of Woschni, Annand and for forced convective heat transfer to a flat plate were used separately and a comparison was made of the resulting throat velocities.
2.2 Prechamber Velocity Analysis - Conservation of Angular Momentum

A general statement of the principle of Conservation of Angular Momentum is:

"The resultant external torque on the matter momentarily occupying a fixed volume equals the rate of change of angular momentum of the matter inside the volume plus the net rate of flow of angular momentum through the control surface". (30).

In this case the fixed volume is the prechamber. The resultant external torque is the torque due to friction. The net flow of angular momentum is due to the flow through the throat. Therefore the principle of conservation of angular momentum yields:

\[
\frac{d}{dt} (I\omega)_p - \frac{dm}{dt} U \text{ th } R_1 = - F_T
\]  

\[ ...... (2.21) \]

where

- \( I_p \) is the moment of inertia of the air mass in the prechamber.
- \( F_T \) is the friction torque.

It is noted that \( R_1 \), the perpendicular distance between the axis of the prechamber and the axis of the throat, is less than the radius of the prechamber. The throat is tangential to the curved surface of the prechamber where they meet.

It is recalled that the air mass in the prechamber rotates as a solid body. The expression for the moment of inertia, \( I_p \), is for a cylindrical prechamber:

\[
I_p = \frac{1}{2} m_p R_p^2
\]  

\[ ...... (2.22) \]

and for a spherical prechamber.
\[ \mathbf{I}_p = \frac{2}{5} \mathbf{m}_p R_p^2 \]  

\[ \ldots (2.23) \]

where \( R_p \) is the radius of the prechamber  
\( m_p \) is the instantaneous mass of air in the prechamber.

2.3 Friction Effects at the Prechamber Wall

2.3.1 Cylindrical prechamber

The friction torque in the cylindrical prechamber is made up of (1) the friction torque at the curved surface, and (2) the friction torque at the flat disc surfaces of the side walls.

For the curved surface:

\[ T_1 = \tau_1 * A_1 * \frac{D_p}{2} \]  

\[ \ldots (2.24) \]

where \( D_p \) is the prechamber diameter

and \( \tau_1 \) the shear stress at the wall - is given by:

\[ \tau_1 = C_f \frac{\rho U_p^2}{2} \]  

\[ \ldots (2.25) \]

where \( U_p \) is the air velocity in the free stream at the wall  
\( C_f \) is the wall friction coefficient.

For a flat plate and turbulent flow:

\[ C_f = 0.074 \Re^{-0.2} \]  

\[ \ldots (2.26) \]

The characteristic length used in the evaluation of the Reynolds number, \( \Re_1 \), is the circumference of the prechamber.

\[ \Re_1 = \frac{\rho * \omega_p D_p * \pi_p}{\mu} \]  

\[ \ldots (2.27) \]
where \( \omega_p \) is the angular velocity

\( \mu \) is the kinematic viscosity of air.

The curved surface area:

\[
A_1 = \pi \frac{D_p}{p} b_1 
\]

(2.28)

where \( b_1 \) is the thickness of the cylindrical prechamber.

Substituting equations 2.25 to 2.28 in 2.24 gives the expression for the friction torque on the curved surface as:

\[
T_1 = \frac{0.074}{16} \left( \frac{\pi}{2} \right)^{-0.2} \left( \frac{\omega_p^2 D_p^2}{\mu} \right)^{-0.2} \left( \frac{\rho^2 D_p^2}{2} \right) \left( \frac{2}{2} \right) \left( \frac{\pi D_p b_1}{p} \right) \left( \frac{D_p}{2} \right) \]

(2.29)

The friction effects on the two side walls is equivalent to that on two flat discs, with the resultant friction force acting at a mean radius of \( \frac{3D_p}{4} \). So, the friction torque on the two side walls, \( T_2 \), is given by:

\[
T_2 = 2 * \tau_2 * \frac{\pi D_p^2}{4} * \frac{D_p}{4} 
\]

(2.30)

\[
\tau_2 = 0.074 \ Re_2^{-0.2} \ * \ \frac{\rho (\omega_p D_p / 4)^2}{2} 
\]

(2.31)

\[
Re_2 = \frac{\rho * (\omega_p D_p / 4) * (\pi D_p / 2)}{\mu} 
\]

(2.32)

\[
\therefore T_2 = \frac{0.074}{256} \ * \ \left( \frac{\pi}{8} \right)^{-0.2} \left( \frac{\omega_p^2 D_p^2}{\mu} \right)^{-0.2} \left( \frac{\rho^2 D_p^2}{2} \right) \left( \frac{2}{2} \right) \left( \frac{\pi D_p^2}{2} \right) \left( \frac{D_p}{2} \right) 
\]

(2.33)

Therefore for a cylindrical prechamber the total friction torque \( T_3 \) is given by:

\[
T_3 = T_1 + T_2 
\]

(2.34)
where \( T_1 \) is given in equation (2.29) \\
\( T_2 \) in equation (2.33)

2.3.2 Friction Effect at the Wall of a Spherical Prechamber

The assumption of a solid body swirl is made, such that the angular velocity about the axis of rotation is equal over the whole wall surface. However, the perpendicular distance from the axis to points on the wall varies because of the spherical curvature of the wall. Hence the friction torque must be integrated over the wall surface area. Fig. 2.3 is a sketch showing an elemental strip of the spherical surface. The total friction torque, \( T_4 \), is given by:

\[
T_4 = 2 \int_{x=0}^{R} (\tau_4 \, dA_4 \times y) \\
\text{d}A_4 = 2 \pi y \text{d}x \\
\tau_4 = C_f \frac{1}{2} \rho U^2 \\
C_f = 0.074 \, \text{Re}_4^{-0.2} \\
\text{Re}_4 = \frac{\rho (\omega_p y) (\pi y)}{\mu} \\
T_4 = 2 \int_0^R \left[ 0.074 \left( \frac{\pi \rho \omega_p}{\mu} \right)^{-0.2} \times \left( \frac{\rho \omega_p^2}{\mu} \right) \times y^{3.6} \right] \text{d}x \\
\text{but} \quad y^2 = R^2 - x^2 \\
T_4 = 2 \times 0.074 \times \pi^{0.8} \rho^{0.8} \omega_p^{1.8} \mu^{0.2} \times \int_0^R \frac{1.8}{(R^2 - x^2)^{1.8}} \text{d}x \quad (2.41)
\]
2.4 Angular Momentum Equations for Air Motion in the Prechamber

2.4.1 Compression Period

We recall equation 2.21 from above:

\[
\frac{d}{dt} (I_\omega)_p - \frac{dm}{dt} U_{\text{th}} R_1 = - F_T \\
\]

(2.43)

The moment of inertia \( I_p \) is given in equation 2.22 for the cylindrical prechamber and 2.23 for the spherical prechamber.

The rate of mass transfer into the prechamber is given in equation 2.17.

The throat velocity, \( U_{\text{th}} \), is given in equation 2.10 or 2.16.

The perpendicular distance from the prechamber axis to the throat axis \( R_1 \) is given by:

\[
R_1 = \frac{D_p}{2} - \frac{D_{\text{th}}}{2} \\
\]

(2.44)

where \( D_{\text{th}} \) is the throat diameter.

The friction torque, \( F_T \), is equal to \( T_3 \) as given in equation 2.34 for a cylindrical prechamber.

It can now be seen that the angular velocity \( \omega_p \) is the only unknown in equation 2.43, and can thus be obtained at any instant during the compression period.

2.4.2 Angular Momentum Equation for Air Motion in the Prechamber During the Expansion Stroke

As for the compression stroke we apply the principle of angular momentum making the same assumptions. Hence:

\[
\frac{d}{dt} (I_\omega)_p + \frac{dm}{dt} \omega_p R_1^2 = - F_T \\
\]

(2.45)
In this case the second term on the left hand side is the net flow of angular momentum from the prechamber. The friction torque term is the same as for the compression stroke.

It should be noted that the rate of change of moment of inertia, \( \frac{dI}{dt} \), is positive in sign for the compression stroke but negative for the expansion stroke.

2.5 Discharge Coefficient for the Prechamber Throat - A General Discussion

Sitkei (37) and Jankov (7) had indicated that it might be necessary to assign a discharge coefficient to the flow through the throat leading into the prechamber.

Therefore this point will be discussed.

A research of published literature demonstrated the absence of any existing relationship for throat discharge coefficient, that could be applied to the particular case of gas flow in an indirect injection engine.

The throat linking the prechamber to the main cylinder acts like a restriction and is likely to lead to a pressure loss between the two volumes. The total pressure loss will be made up of three major parts due to the following:

(1) Sudden contraction between the main cylinder and throat.
(2) Friction in the throat.
(3) Sudden expansion at the intersection of the throat and the prechamber.

It is usual to express pressure loss in fluid flow as a function of dynamic head in the form:
\[ \frac{\Delta P_i}{\rho} = K_i \times \frac{U_i^2}{2g} \] .... (2.46)

where \( \Delta P_i \) is the pressure loss
\( \rho \) is the fluid density
\( K_i \) is the loss coefficient
\( U_i \) is the velocity of flow

In the present case the total pressure loss will be a summation of the individual losses under the headings (1) - (3) above. See Fig. 2.4 also. Equation 2.46 does not give a particular value of throat discharge coefficient. The loss coefficients given in publications are usually for flow in pipes and ducts. These coefficients obtained experimentally can vary widely for different flow configurations. On the other hand it is very difficult to measure accurately the small pressure differences between the main cylinder and prechamber. In this case the high mean pressures in the chambers make the task impossible using available equipment. It may thus be noted that equation 2.46 cannot be used to obtain an estimate of the loss coefficient.

The possibility of determining a throat coefficient by making measurements of throat velocities and comparing with values obtained from a theoretical solution was investigated. However, the hot wire was consistently broken when placed at the throat cross-section at its entry into the prechamber and measurements were only possible when the side plates of the prechamber were removed and the throat was thus discharging to the atmosphere. This condition does not correspond to the actual motoring conditions of the engine especially since it cannot take into account the effect of the relative volumes of the prechamber and main cylinder. For this same reason the discharge coefficient data of Litchtarowicz et al (40) for long orifices cannot be used in the present case.
Therefore, because of the lack of published data on discharge coefficients for the flow in the throat of the prechamber of an indirect injection engine and because of the unsuccessful attempt to find a theoretical or a direct empirical basis for specifying a discharge coefficient for the throat it was decided to investigate the experimental values of mean velocity in the five cylindrical prechambers examined in the present study with a view to identifying a possible correlation between the experimental data and the theoretical computation of mean velocity. It is thus hoped to propose a general relationship for a factor matching the theoretical prechamber velocities and the experimental data for different prechambers. This is discussed in Section 4.9.

2.6 Comparison of the Energy and the Proportional Mass Transfer Solutions for the Throat Velocity

The throat velocity was obtained by the two methods described in Section 2.1. The results for the one-inch cylindrical prechamber (Engine C.R. = 12:1) are shown in Fig. 2.6. The results for the cylindrical prechamber of Horvatin (Engine C.R. = 15.5:1) (17) are shown in Fig. 2.8.

The one-inch diameter cylindrical prechamber and Horvatin's prechamber have the smallest and highest ratios of prechamber volume to compression volume among the five cylindrical prechambers discussed in the present study (see Fig. 4.1).

Figs. 2.6 and 2.8 show the comparison of the throat velocities obtained by the Energy Equation Solution and the Mass Transfer Solution. The latter does not take into account the pressure difference between the prechamber and main cylinder. It can be seen that the velocities agree to within eight per cent of each other at any time during the
compression and expansion periods. The Mass Transfer Solution is symmetrical about TDC compression following from the fact that the transference of mass between main cylinder and prechamber was directly related to the periodic motion of the piston. For the Energy Solution, the peak velocity in the throat during the expansion stroke is higher than the peak during the compression stroke by 5% for the one-inch diameter prechamber and 15% for Horvatin's prechamber. This is explained by the fact that the ratio of prechamber volume to compression volume is 25% for the one-inch diameter cylindrical prechamber and 76% for Horvatin's cylindrical prechamber. After TDC the relatively larger mass of gas in the prechamber used by Horvatin results in a relatively higher value of the ratio of prechamber to main cylinder pressure. For both the Energy and Mass Transfer Solutions, the peak throat velocity occurs at about 30 degrees before and after TDC. The non-symmetry of the Energy Solution is not considered to be important in the present case since solution for the prechamber velocity during expansion does not involve the throat velocity.

The variation of pressure and temperature with crank angle are shown in Figs. 2.7 and 2.9. For the Mass Transfer Solution the pressure and temperature in the main cylinder and prechamber are equal at a particular crank angle. For the Energy Solution the pressure in the main cylinder during compression is always slightly greater than the prechamber pressure. The converse is true during the expansion period. It may be noted that the flow in the throat is reversed during expansion. The differences in pressure are so small that they are not apparent on the scale used in Fig. 2.7. The Energy Solution gives slightly higher gas temperatures (10%) than the Mass Transfer solution. The Energy Solution shows a
difference in gas temperature in the prechamber and main cylinder. For the one-inch diameter prechamber engine the peak main cylinder temperature is greater than the prechamber peak temperature by only 4% and occurs at TDC. For Horvatin's engine the peak prechamber temperature is greater than the peak main cylinder temperature by about 10%. The peak prechamber temperature, as given by the Energy Solution, becomes relatively greater than the main cylinder temperature the greater the ratio of prechamber volume to compression volume at TDC. For the case of the one-inch diameter prechamber and Horvatin's prechamber, this may be explained by the fact that the surface area to volume ratio of Horvatin's prechamber has a value about half that of the one-inch prechamber and the heat transfer across the prechamber wall is thus relatively more significant in the one-inch diameter prechamber.

2.7 Effects on the Throat Velocity of Varying Some Parameters in the Theoretical Solutions

Figs. 2.10 and 2.11 show the effect on throat velocity of using different combinations of heat transfer coefficients for the wall heat transfer terms for the main cylinder and prechamber. It is seen that the effect of using either Woschni's or the 'flat plate' expressions of heat transfer coefficients is very small and may be taken as negligible. The same is true of neglecting heat transfer across the wall of the main cylinder. The only noticeable and obvious effect is that the lower the value of heat transfer coefficient the higher the gas temperature at TDC.

The engine speed has a linear effect on the throat velocity.

The trapped mass was increased by increasing the stroke of the engine. This resulted in a corresponding increase in compression
ratio since the compression volume had been held constant.

Figs. 2.12 to 2.15 show the variation of throat velocity with crank angle for three compression ratios. For the same compression volume the throat velocity increases less than proportionally with compression ratio. This is true for both the Energy and Mass Transfer Solutions. The crank angle at which peak velocity occurs is shifted nearer to TDC by about 5 degrees for the compression ratio of 20:1 compared to 12:1.

The ratio of prechamber volume to compression volume, \( \sigma \), was varied by varying the compression volume. This also results in variation of the compression ratio. The compression volume was varied to give the same compression ratios as in Figs. 2.12 to 2.15. It can be seen from Figs. 2.16 to 2.19 that throat velocity increases with \( \sigma \) when the only variable is the compression volume. Comparison of the curves for the same compression ratios in Figs. 2.12 to 2.15; and in Figs. 2.16 to 2.19, shows that the increase in throat velocity achieved by varying the trapped mass is slightly more than that achieved by varying the compression volume.

2.8 Conclusion to the Mathematical Model

The attempt made to find a theoretical basis for determining a flow coefficient for the prechamber throat or of measuring direct empirical values proved unsuccessful. The "pressure loss" approach proved impractical because it was not possible to determine the pressure drop between the main cylinder and prechamber accurately.

It was also not possible to make measurements of throat velocities under actual running conditions and hence to determine empirical values of throat flow coefficients. This was due to the
inadequacy of the hot wire in being constantly broken by the force of the throat jet. The correlation of Litchtarowicz et al (40) for long orifices cannot be applied in the present study because of the non-similarity of the flow configurations.

In the solution for the throat velocity the Energy and Mass Transfer methods gave values of throat velocity which agree to within ±8% in all cases. The Mass Transfer Solution is symmetrical about TDC of compression period, but the Energy Solution gives higher values (5% - 15%) during the expansion period. This is not considered to be of critical importance in the present study. Changes in engine parameters such as engine speed, compression ratio and ratio of prechamber volume to compression volume result in the same relative changes (throat velocity increases with N, CR and σ) for both the Energy and Mass Transfer Solutions.

It is recommended that the Mass Transfer Solution be adopted in preference to the Energy Solution because it is more direct and simple to use than the Energy Solution.
FIG. 2.2 Schematic Diagram for the Analysis of Gas Flow in a Prechamber Engine

FIG. 2.1a Throat Inclination to Prechamber Curved Surface

FIG. 2.1b Throat Position Relative to Centre of Prechamber
FIG. 2.3 Elemental Surface Area in Spherical Prechamber

FIG. 2.4 Pressure Losses in Flow Between Main Cylinder and Prechamber

FIG. 2.5 Variation of Throat Discharge Coefficient with Engine Speed when Throat is Discharging to Atmosphere
FIG. 2.6 Velocity and Mass Flow Rate in Prechamber Throat. 1\textquoteleft\textquoteleft-Diameter Cylindrical Prechamber

FIG. 2.7 Pressure and Temperature in Prechamber and Main Cylinder. 1\textquoteleft\textquoteleft-Diameter Cylindrical Prechamber
FIG. 2.8  Velocity and Gas Density in Prechamber Throat. Horvatin's Data, 1000 rpm.

FIG. 2.9  Pressure and Temperature in Prechamber and Main Cylinder. Horvatin's data, 1000 rpm.
HC, HP = Heat Transfer Coefficients in Cylinder and Prechamber

W = Woschni

FP = Flat Plate

FIG. 2.10 Effect of Heat Transfer Coefficient on Throat Velocity.
1” Diameter Cylindrical Prechamber, 1100 rpm.

FIG. 2.11 Effect of Heat Transfer Coefficient on Throat Velocity.
Horvatin’s Data, 1000 rpm
FIGS 2.12, 2.13  Effect of Compression Ratio on Throat Velocity.
(Energy Analysis)
Effect of Compression Ratio on Throat Velocity. (Momentum Analysis)

FIGS. 2.14, 2.15
FIGS. 2.16, 2.17  Effect of Ratio of Prechamber Volume to Compression Volume on Throat Velocity. (Energy Analysis)

1" Prechamber
1100 rpm

\[ \sigma = \frac{\text{Prechamber volume}}{\text{Total volume at TDC}} \]

\( \sigma = 0.25 \)
\( \sigma = 0.40 \)

1000 rpm

Horvat (17)

\( \sigma = 0.58 \)
\( \sigma = 0.76 \)
FIG. 2.18

Effect of Ratio of Prechamber Volume to Compression Volume on Throat Velocity (Momentum Analysis)

- Prechamber 1"
- 1100 rpm

- $\sigma = 0.25$
- $\sigma = 0.40$

FIG. 2.19

Effect of Ratio of Prechamber Volume to Compression Volume on Throat Velocity (Momentum Analysis)

- Prechamber 1"
- 1000 rpm

- $\sigma = 0.58$
- $\sigma = 0.76$
CHAPTER 3

EXPERIMENTAL TECHNIQUE AND
ERROR ASSESSMENT
It had been decided, in conclusion to the literature survey, that a hot-wire anemometry system would be adopted for the measurement of air velocities in the experimental engine. Hassan (14), Derham (41), Lobo (16), Horvatin (17) and Tindal et al (20) have demonstrated the suitability of the hot-wire for making velocity measurements in engines running under motored conditions. The experience that had been built up over the past eight years in the use of hot-wire anemometers in the Mechanical Engineering Department of Loughborough University provided a sound basis from which to work.

The single-wire probe is suitable for velocity measurement for the case in which one-dimensional flow is established as is possible in a wind tunnel. For two or three-dimensional flows which occur in engines a multi-wire probe must be employed or several orientations of a single wire probe used if it is intended to obtain the velocity vector in both magnitude and direction. Hassan (14), and Horvatin (17) have assumed that in their cylindrical prechamber type of engine the flow pattern would be a forced vortex and essentially one-dimensional. They thus used single-wire probes to measure the swirl velocity. It could be considered a very optimistic assumption, that the flow field will be a uniform solid swirl.

Derham (41) and Salama (29) have used multi-wire probes to make velocity measurements in the cylinder of internal combustion engines. Derham arranged his three wires on a pitch diameter of 6.3 mm. Such a large diameter probe will be unsuitable for measurements in a prechamber because of the excessive disturbance it would cause to the flow.
Lobo (16) developed a "five-wire" theory which should leave no ambiguities about the direction and sense of the velocity vector. However, such a five-wire system is elaborate and complicated in its operation, and in the reduction of data, especially in cases where a large number of measurements are required.

A three-wire probe will give both the magnitude and direction of the velocity vector. There may be some ambiguity about the sense of the vector, but this can be effectively resolved by examination of the geometry of the combustion chamber. For example, consider the cylindrical prechamber as shown in the schematic Fig. 3.1. It is considered reasonable and more likely that the sense of the velocity vector in the position C will be downwards and inwards rather than as shown by the dotted vector.

For the above reasons it was decided to construct a three-wire probe for the prechamber velocity measurements.

3.1 The Theory of Three-Wire Operation

When a hot-wire is operated in a wind tunnel with a onedimensional axial flow, the actual flow velocity is that recorded by the wire when it is perpendicular to the flow direction. When the wire is inclined to the flow at an angle $\theta$ it is possible to express the velocity measured by the wire in terms of the incident maximum velocity $U$ and the angle $\theta$. Various investigators have suggested relationships between $U$ and $\theta$. These have been tabulated by Derham (41). The most generally used of these relations is:

$$U_\theta = U \sin \theta \quad 10^\circ < \theta \leq 90^\circ \quad \ldots \quad 3.1$$

where $U_\theta$ is the velocity measured by the wire.
The reliability of equation 3.1 has been verified by Lobo (16), Pearson (15) and Derham (41). This sine relationship will be adopted for the present study.

It may be noted that equation 3.1 is obtained on the assumption that the wire is totally insensitive to the component of flow along its length. If a hot-wire is placed in a three-dimensional flow field the most general case is for the flow vector to be inclined to the wire in more than one plane. It is always possible to resolve the velocity vector into three components, two of which will be perpendicular to the wire and the third along the wire length as illustrated in Fig. 3.2. Following the above assumption only the components $U_a$ and $U_b$ perpendicular to the wire will be detected by the wire.

### 3.1.1 Magnitude of the Velocity Vector in a Three-Wire Measuring System

Let $U_1$, $U_2$ and $U_3$ be the velocity readings obtained from the wires whose lengths lie along the $X$, $Y$ and $Z$ axes respectively. Let $U_x$, $U_y$ and $U_z$ be the resolved components of the velocity vector along the specified $X$, $Y$ and $Z$ axes as in Fig. 3.3. Each wire will measure the resultant of the velocity components perpendicular to it.

The wire readings can thus be written as:

$$U_1^2 = U_x^2 + U_z^2 \quad \ldots \quad 3.2$$

$$U_2^2 = U_x^2 + U_z^2 \quad \ldots \quad 3.3$$

$$U_3^2 = U_x^2 + U_y^2 \quad \ldots \quad 3.4$$
Equations 3.2, 3.3 and 3.4 can be rearranged to give:

\[ \begin{align*}
U_x^2 &= \frac{1}{2} \left( -U_1^2 + U_2^2 + U_3^2 \right) \quad \ldots \quad 3.5 \\
U_y^2 &= \frac{1}{2} \left( U_1^2 - U_2^2 + U_3^2 \right) \quad \ldots \quad 3.6 \\
U_z^2 &= \frac{1}{2} \left( U_1^2 + U_2^2 - U_3^2 \right) \quad \ldots \quad 3.7
\end{align*} \]

Hence the magnitude of the velocity vector is given by:

\[ U^2 = U_x^2 + U_y^2 + U_z^2 \quad \ldots \quad 3.8 \]

\[ = \frac{1}{2} \left( U_1^2 + U_2^2 + U_3^2 \right) \]

From equation 3.8 we see that the velocity readings given by three wires set mutually perpendicular to each other directly gives the magnitude of the velocity vector.

### 3.1.2 Direction of the Velocity Vector

Let \( \alpha \), \( \beta \) and \( \gamma \) be the angles between the velocity vector and the \( X \), \( Y \) and \( Z \) axes respectively as shown in Fig. 3.4.

\[ \cos \alpha = \frac{U_x}{U} \quad \ldots \quad 3.9 \]

\[ \cos \beta = \frac{U_y}{U} \quad \ldots \quad 3.10 \]

\[ \cos \gamma = \frac{U_z}{U} \quad \ldots \quad 3.11 \]

It is preferable to describe the direction of the velocity vector in terms of its inclination to the mutually perpendicular planes.
formed by the wires. The advantage of this is that the velocity vector direction is directly linked to the geometrical setting of the wires and also these planes can be made to coincide with the engine geometry. This makes it easy to visualise the experimental measurements. It is possible to deduce from Fig. 3.4 that the horizontal angle $HA$ is given by:

$$\cos HA = \frac{\cos \gamma}{(\cos^2 \gamma + \cos^2 \alpha)^{\frac{1}{2}}}$$

$$= \frac{U_z}{(U_z^2 + U_x^2)^{\frac{1}{2}}} \quad \ldots \ldots 3.12$$

and the vertical angle $VA$ is given by:

$$\cos VA = (\cos^2 \gamma + \cos^2 \alpha)^{\frac{1}{2}}$$

$$= \frac{U_z}{U \cos HA} \quad \ldots \ldots 3.13$$

By obtaining the velocity measurements given by three mutually perpendicular wires equations 3.8, 3.12 and 3.13 can be used to describe both the magnitude and direction of the velocity vector of air motion inside the combustion chamber of a prechamber type of engine.

3.2 The Design and Manufacture of a Three-Wire Probe and Traversing Mechanism

The design requirements of a hot-wire for use in engine measurements have been outlined in Chapter 1, Section 1.2.4. In addition a three-wire probe (probe body + wires) must still satisfy the operational requirements of strength, ability to withstand high
gas temperatures in the engine, little disturbance to the flow being measured and good spatial resolution. In the present study the need to be able to traverse the probe along its axis and to rotate it on this axis, is also considered an important factor.

Fig. 3.5 is an illustrative sketch of one of the probes used in this study. In the interests of clarity only one set of pin supports and connections are shown.

The wire supports are steel needles cut and shaped as required. The supports are held rigidly to the probe body by means of Pyrotenax glazing flux, which has a melting point of 350°C, or high temperature strain gauge cement (Trade name - "Brimor" Type U527). The supports are joined to conductors within the probe body using silver solder which has a melting point of 320°C.

The conducting material within the probe body was magnesium oxide insulated, twin-conductor, nichrome sheathed cabling available commercially as 'Pyrotenax'. The actual conducting wires are of copper. In the three-wire probe the three conducting lengths of pyrotenax were tied, then bonded together with araldite. The space between the pyrotenax and the stainless steel probe body was also filled with araldite. The exposed parts of the copper wires especially at the soldered joints were coated with electrically insulating paste and each lead kept separate in order to prevent any short circuits within the probe.

The above arrangement provided a very rigid probe and served to keep the temperature of the connecting wires well below the mean gas temperature in the prechamber.

The connecting plug at the rear of the probe was manufactured from individual gold plated pins and sockets separately fixed in
insulating material. The output leads were fitted with BNC type of sockets. These gave low and consistent contact resistance.

The three pairs of pin supports were set mutually perpendicular using an ISOMAR Optical Projector, Type M119, which gave a magnification of 50. With this equipment it was possible to set the supports for perpendicularity to within 2 degrees.

The fine sensing wires of 10 μm diameter were welded to the pin supports using a DISA 55All micro manipulator and a capacitor discharge unit in conjunction with a binocular microscope. Figs. 3.6 and 3.7 shows a typical three-wire probe.

Facilities of probe rotation about its own axis was required in order to be able to determine more definitely the magnitude, sense and direction of the velocity vector. A traverse movement of the probe along its axis would enable the velocity profile across the prechamber to be investigated. In addition to these facilities it is necessary to have adequate sealing at the points where the probe is introduced into the prechamber. It is also very important to be able to insert and withdraw the probe without damaging the wires. This was achieved by arranging to assemble the probe and traverse mechanism as one unit with the prechamber sealing flanges which are interchangeable. Fig. 3.8.

3.3 Three-Wire Probe Calibration in Wind Tunnel

3.3.1 Hot-Wire Anemometer Bridge(s) Used to Operate Probe

Hassan (4) had built and used a hot-wire anemometer bridge similar to that described by Davies and Fisher in reference (42). Derham (41) had subsequently used this bridge in his velocity measurements in a direct injection type of engine. The same velocity measuring system has been adopted for the present study.
3.3.1(i) Heat Transfer Theory of a Hot-Wire

Davies and Fisher (42) showed that the theoretical equation describing the thermal equilibrium of an electrically-heated wire placed in a stream of cooler gas may be written as:

\[ kA_r \frac{d^2 T_w}{dx^2} + \left( \frac{I^2 \beta_g \alpha - \pi h d^{2}}{A_r} \right) (T_w - T_g) + \frac{I^2 \beta_g}{A_r} = 0 \quad ... \ 3.14 \]

where
- \( k \) is the thermal conductivity of the gas
- \( A_r \) is the cross-sectional area of the wire
- \( I \) is the current flowing in the wire
- \( \beta_g \) is the electrical resistivity of the wire at the gas temperature
- \( \alpha \) is the temperature coefficient of the wire material
- \( h \) is the heat transfer coefficient
- \( d \) is the wire diameter
- \( T_w \) is the wire temperature
- \( T_g \) is the gas temperature

If the following boundary conditions are taken to apply:

(i) maximum temperature occurs at the middle of the wire length such that \( dT/dx = 0 \) at that point,
(ii) the wire supports are at the instantaneous gas temperature,

equation (3.14) with boundary conditions (i) and (ii) can be solved to give:

\[
\frac{I^2}{A_r k_t} \frac{R_w}{l(T_w - T_g)} \left[ 1.0 - \frac{2}{l} \frac{k_t}{k} \frac{R_g}{R_w} \frac{\tan h \left( \frac{R_w}{2K_1} \sqrt{|K_1|} \right)}{K_1} - \left( \frac{R_w - R_g}{R_w} \right) \right]
- \left| K_1 \right| = 0 \quad ... \ 3.15
where \( k_1 = \frac{1^2 \beta_g \alpha}{k_w A_r^2} - \frac{\pi h d}{k_w A_r} \) .... 3.16

\( k_g, k_t \) are the thermal conductivities of the wire material at the gas and wire temperatures respectively.

\( k_w \) is the thermal conductivity of the gas at the wire temperature

\( R_g, R_w \) are the resistances of the wire at the gas and wire temperatures respectively.

Equation 3.15 is implicit in \( K_1 \) which can be obtained iteratively. The heat transfer coefficient, \( h \) can thus be obtained from equation 3.16.

Davies and Fisher (42) showed that the wire heat transfer coefficient can be related to the flow velocity over the wire by:

\[ h = C_f \rho \frac{C_v k_w}{\pi k_a} U \] .... 3.17

where \( \rho \) is the fluid density

\( C_v \) is the constant volume specific heat

\( k_a \) is the thermal conductivity of the gas at the gas temperature

\( U \) is the fluid velocity

\( C_f \) the skin friction coefficient is given by:

\[ C_f = 2.6 \ Re^{-2/3} \quad (0 < \ Re < 40) \] .... 3.18

\[ C_f = 1.4 \ Re^{-1/2} \quad (40 < \ Re < 1000) \] .... 3.19

where \( \Re \), the Reynolds number, is based on the wire diameter.
After obtaining the heat transfer coefficient from equation 3.16 above, the flow velocity, $U$, over the wire can be obtained from equation 3.17 in conjunction with either equation 3.18 or 3.19.

3.3.1(ii) Hot-Wire Anemometer Circuit

In the solution of the theoretical equations above the one other quantity required apart from the wire and gas properties is the electric current flowing through the wire. This current can be obtained experimentally by operating the wire either as a constant current or constant temperature anemometer.

The constant temperature hot-wire anemometer circuit of Davies and Fisher (42) is shown in Fig. 3.9. It may be noted that this bridge is symmetrical and balance is obtained when the current flowing through the wire makes its resistance equal to the set operating resistance. If the voltage drop across the resistances terminals A and B is given as $E$ then the current flowing through the wire is given by:

$$I = \frac{E}{2(R_w + R_L)} \quad \text{..... 3.20}$$

where $R_w$ is the operating resistance.

and $R_L$ is the resistance of the connecting leads.

3.3.2 Physical and Thermal Properties of Sensing Wire

The sensing wire used in this study is of platinum/20% Iridium alloy. The wire is produced by a precision drawing process and it is considered that variation in its diameter will be negligible for wire
lengths of 2 mm. For this reason the diameter quoted by the manufacturers have been used. For all the tests in this investigation a 10 µm diameter wire has been used.

The wire length was measured using a microscope with a magnification of 20. It is estimated that the error in wire length measurement is ± 0.05 mm.

The thermal conductivity and the resistivity of the wire material are taken from published data (43) (44).

The temperature coefficient was determined by the standard method of the oven test. The iron-constant thermocouple ends were silver-soldered in view of the high temperature to which the oven was heated.

In the course of the experimental work many individual wires were used. It was impracticable to determine the temperature coefficient, α, for each wire so it was decided to determine α for each batch of wire reel from which the wires were taken. Fig. 3.10 is a typical example of the graphs from which α was obtained. It was discovered that in a particular reel of wire there was little variation in the value of α - for different samples of wire.

3.3.3 Sensing Wire Resistance at Ambient Temperature

It had been recognised (41), (42) that an error in the determination of the wire resistance at room temperature can lead to errors in the computation of flow velocity. It is considered that such error in resistance measurement is mainly due to the Peltier effect generated by the need to join dissimilar metals.
Davies and Fisher (42) tried to overcome this source of error by calculating the 'cold' resistance from the measured wire length and published values of wire material resistivity. However, it should be pointed out that since there could be an error in determining the wire length the 'cold' resistance value may still be suspect. Also, the theoretical calculation assumes that there is perfect adhesion between the wire and its supports at the welded joints. In the present case the wires are spot-welded at the supports and this makes it likely that there will be some contact resistance at the wire supports.

Derham (41) corrected for the uncertainty in cold resistance by matching the theoretical curve given by Davies and Fisher equation with the experimental points obtained in the wind tunnel calibration tests. This matching was achieved by varying the 'cold' resistance and the wire operating temperature until the theoretical curve was a least square fit of the experimental calibration points. It is considered that this is a more consistent and effective method of obtaining the 'cold' resistance than that of Davies and Fisher. It has the advantage that the value of measured 'cold' resistance is not critical provided it had been determined with the due care associated with such measurements and provided the wire remains stable.

In the present study the 'cold' resistance of the sensing wire was determined using a method developed by the hot wire anemometry group in the Mechanical Engineering Department of Loughborough University. It is based on the fact that a hot-wire ceases to operate as an anemometer when the operating resistance becomes equal to the 'cold' resistance of the wire. When the operating resistance is made less than the 'cold' resistance the wire output voltage becomes asymptotic as shown in Fig. 3.11a.
If the wire is operated in free air with no fluid flow over it, and in flows with known velocities, then, a subtraction of the zero flow bridge voltage values from the values at each of the known velocities would eliminate any free convection and Peltier effects and yield the 'cold' resistance at the point where the difference in voltages equal zero. With the Davies and Fisher bridge the difference between the velocity and zero curves do not tend to zero in a definite manner. This is due to the fact that the bridge is never in total balance and there is always a small current flowing even when the operating resistance becomes equal to the wire 'cold' resistance.

An examination of Fig. 3.11b shows that the differences between two different velocity curves and the zero flow curve are linear in the region where the operating resistance is greater than the nominal 'cold' resistance. Extrapolation of this linear part for both curves shows intersection at the same point on the 'cold' resistance axis. This was taken as the wire cold resistance, and is then used in the calibration-matching computer program of Derham referred to above. The iterated value of cold resistance was usually within 2% of the measured cold resistance.

3.3.4 Bridge Voltage Output Versus Velocity: Calibration Procedure for Three-Wire Probe

The wind-tunnel used in all the probe calibration tests is the DISA Type 55 D41 unit, a sketch of which is shown in Fig. 3.12. An adaptor was made to fit the test section such that a probe can be mounted in a vertical position.
Each wire was calibrated separately. The probe was installed such that the wire length lay perpendicular to the axis of the wind-tunnel. A wire operating resistance was selected on the basis of knowledge of the 'cold' resistance and a desired operating temperature of between $600^\circ$ and $700^\circ$. It was found that a ratio of operating resistance to cold resistance of about 1.5 gave an adequate operating temperature. A suitable manometer was used to measure the pressure differences between the wind-tunnel inlet and test section, and the flow velocities determined from these pressure difference measurements. Data of air velocity and corresponding bridge output voltage, together with the wire physical and thermal properties, the air ambient temperature and pressure, are fed into the calibration computer program referred to in Section 3.3.3. The wire was calibrated at different operating temperatures and the matching done for each operating condition. The iterated values of cold resistance and wire operating temperature then ensure coincidence between the theoretical curve and the experimental calibration over the range of temperatures used in the engine measurements. Fig. 3.13 shows a set of experimental calibration points together with the theoretical curve. They show good agreement. It is now very convenient to use the theoretical calibration equation in reducing the engine data.

3.3.5 Temperature Correction Factor for Velocity Measurement in an Engine

The calibration procedure described in Section 3.3.4 holds for the case in which the flowing air is at ambient temperature and pressure. The flow conditions are different in an engine. Here the gas pressure and temperature continuously varies during the cycle. Hassan and Dent (46), and Dent and Derham (18) have shown that it is necessary to apply a
correction term to the velocity as calculated by the Davies and Fisher equation when this is applied to velocity measurement in an engine.

Hassan and Dent (46) proposed that this correction factor should be \( (T_w/T_g)^{0.3} \), where \( T_w \) is the wire operating temperature and \( T_g \) is the instantaneous gas temperature. Dent and Derham (18) described a method of checking the consistency of applying this correction factor.

In brief, the method involves operating the wire at two different temperatures in the engine. Since there is little cyclic variation in the mean velocity in the cylindrical prechamber under test the velocity records for the two separate runs should be nearly identical. The output voltage at one operating temperature can thus be used together with the heat balance equation for the wire to predict the current in the wire at a different operating temperature. This then gives the corresponding bridge voltages at this new operating temperature. A comparison can now be made with the measured output voltage.

It was decided to check this temperature correction factor. The wire was operated at 605°C and then at 418°C. Maximum gas temperature was about 320°C. The measured voltage at 605°C was used to predict the voltage at 418°C. Fig. 3.14 shows the predicted voltage compared with the measured voltage record. The good agreement shown confirms that the temperature correction factor as established by Hassan and Dent (46) is accurate and consistent.
3.4 Velocity Measurement Error Analysis for the Three-Wire Probe

3.4.1 Verification of the "Sine Law"

As was pointed out in Section 3.1 the use of the three-wire probe in the present study has been based on the theory that the "Sine Law" applies when the incident flow is not normal to the wire length. It was decided to verify the "Sine Law" for the probes used in this study.

The probe was mounted in an adaptor on the wind tunnel. The adaptor permits rotation of the probe on its axis through 360 degrees. The sensing wire is set in a horizontal position and perpendicular to the wind tunnel axis. In this position the angle, $\theta$, between the tunnel axis and the wire length is 90 degrees. The wind tunnel was set to run at a selected velocity. The probe was then rotated at intervals of 10 degrees and the corresponding wire voltage output recorded.

The above procedure was done at three speeds in the range 0-100 m/s. Fig. 3.15 shows a typical plot of the results obtained. It can be concluded that the "Sine Law" is justified for $10^\circ < \theta < 90^\circ$ as assumed in Section 3.1. However it was noted that if the sensing wire is not completely straight between its supports the range of validity becomes $20^\circ < \theta < 90^\circ$. For this reason great care was taken when welding the wire to the supports.

3.4.2 Error in the Velocity Vector

The wires of the three-wire probe were calibrated separately as described in Section 3.3.4 with the air flow perpendicular to the wire length.
When these wires are used as a unit for velocity measurements in the prechamber of an engine the 'character' of the three-wire unit is different from the individual wires. This is due to the fact that in the three-dimensional flow in the prechamber the flow velocity will in general be incident on the wires at an angle other than normal to them. It was pointed out in Section 3.4.1 that the "Sine law" may become invalid for small angles of incidence. This will lead to error in the calculation of absolute velocity from the individual wire measurements. Also, it is difficult to keep the three wires mutually perpendicular when they are welded unto their supports. Although great care was taken to set the support pins mutually perpendicular (Section 3.2) it was inevitable that slight bowing in the wires during the welding process occurred.

These two sources of error are in addition to the errors incurred in the determination of the physical dimensions and the thermal properties of the wires. For these reasons it was considered essential to estimate the possible error margin in velocity values as measured by the three-wire probe taking into consideration the combined effect of the sources of error outlined above.

It is possible to attempt to determine the error margin by means of a theoretical analysis as was done by Derham (41). A result, $R$, is expressed in terms of selected variables $X$, $Y$ and $Z$ likely to cause an error in the determined value $R$.

$$ R = f (X, Y, Z) \quad \ldots \ldots \text{3.21} $$

If the error in the derived result is $r$ and the errors in $X$, $Y$ and $Z$ are $x$, $y$ and $z$ respectively, then

$$ R_{c} + r = f (X_c + x, Y_c + y, Z_c + z) \quad \ldots \ldots \text{3.22} $$
where the subscript \( c \) denotes the correct values. Equation 3.22 can be expanded in a Taylor's Series and by re-arranging obtain:

\[
r^2 = \left( \frac{R}{X} \right)^2 x^2 + \left( \frac{R}{Y} \right)^2 y^2 + \left( \frac{R}{Z} \right)^2 z^2 \quad \ldots \quad 3.23
\]

Derham (41) chose the wire diameter, the 'cold' resistance and the operating resistance as the three error variables. He quoted an error of \( \pm 8.56\% \) in the velocity magnitude as measured by his three-wire probe, as a result of his analysis.

In the present study the flow situation in the prechamber is less predictable than in the cylinder of Derham's direct injection engine. Derham was able to assume that air flow in the cylinder was two-dimensional and set his wires accordingly. In the present case it appears reasonable to take that air flow in the prechamber may be three-dimensional and that the velocity vectors may be incident on the wires at angles where the "Sine law" is not valid.

We recall the fact that slight curvatures inevitably occurred in the wires when they were spot-welded.

In addition to the above points most of the wires used in the present study had aspect ratios of 200 or less. It is known that end conduction losses may become significant for wires operated with aspect ratios less than 200.

We consider that any theoretical error analysis which attempts to include the three points above together with the other sources of error will lead to the analysis becoming very complicated. It is desired to have an error estimation procedure which while taking into account all the major likely sources of error remains basically simple. Such a procedure is described below.
3.4.3 Error in the Velocity Magnitude

Wind Tunnel Velocity Prediction Error - Rotation of Three-Wire Probe Through 360 Degrees

After each wire of the three-wire probe had been separately calibrated, the three-wire unit was used to predict set wind tunnel velocities. The probe was mounted at a reference position. A particular tunnel velocity was selected and maintained constant. The bridges of the three wires were operated simultaneously and output voltages recorded. The composite probe assembly was rotated on its axis at 10 degree intervals and the corresponding bridge voltages recorded. The tunnel velocity was kept constant while the probe was rotated through 360 degrees. The tunnel velocity was calculated from each set of bridge voltages. These values of velocity should be equal for all the probe rotational positions and should also be equal to the set wind tunnel velocity. Deviations between the set velocity and the measured values indicate the error margin.

Fig. 3.16 shows the comparison between the set wind tunnel and the three-wire probe measurements at the different rotational positions for two selected velocities.

The error in the measurement of the velocity magnitude lies between -5.0% and +15.0%.

3.4.4 Error in the Direction of the Velocity Vector

The direction of the velocity vector is specified by the horizontal angle, HA and the vertical angle VA. See Section 3.1.2. It was stated in Section 3.4.3 that the probe was initially located in a reference position. In this reference position the three wires were made to coincide with a set of reference axes. In this position the horizontal and vertical angles are equal to zero. In fact one of the wires was always vertical and normal to the direction of flow.
along the tunnel axis. For this reason the vertical angle should always be equal to zero. The horizontal angle should vary from zero, in 10 degree intervals, to 90 degrees in the four segments covered when the probe is rotated through 360 degrees.

Fig. 3.17 shows the comparison between the expected and measured horizontal and vertical angles.

An examination of Fig. 3.17 shows that the error in the determination of the horizontal angle is about ±10 degrees. The error in the vertical angle is between 5-25 degrees.

3.4.5 Velocity Detected by a Wire Lying Along the Direction of Flow

It had been assumed that when a hot wire is lying along the direction of flow that it will measure zero velocity. However, it was found that the wire did measure some apparent velocity when it lay along the direction of flow.

A wire was aligned along the tunnel axis. The tunnel flow velocity was gradually increased and the wire output voltages were recorded. The reduced velocities measured are plotted in the form of Fig. 3.18, and this shows that the proportion of axial flow detected increases with the axial velocity and reaches an upper value of about 20.0% at the higher velocities. It can be shown that this leads to an error of 2.0% at a tunnel velocity of 100 m/s for measurements made with a three-wire probe.

3.4.6 Discussion of the Error Analysis

The reliability of the error estimation procedure presented above depends on the accuracy of the calibration of the sensing wires.
It has been shown that an accurate calibration procedure was followed. See Section 3.3.

Once the accuracy of the wire calibration procedure is established the three-wire probe error assessment as described above, is a simple, direct and effective way of accounting for all the major sources of error. A theoretical error analysis cannot achieve this in a direct manner. The lower error margin of -5.0% will occur in the particular case when the wire supports may mask one or more of the wires. The upper limit of +15% is high compared with the value of ±8.56% quoted by Derham as a result of his theoretical error analysis. However it is considered that the present empirical approach is realistic in the present situation.

The error of 5-25 degrees in the vertical angle is for the case when the reference vertical angle is zero. This is the worst possible case. Derham (41) has shown that for the case when the reference vertical angle is greater or equal to 20 degrees the error is about ±12.0%. It seems reasonable to expect that the average error in the vertical angle will be of the same order (±10 degrees) as for the horizontal angle.

3.5 Engine Data Acquisition

The data required to be measured in the prechamber of the test engine were:

1. Gas mean velocities.
2. Instantaneous gas pressures.
3. Instantaneous gas temperatures.
3.5.1 Timing Mark

It was necessary to establish a fixed mark on the engine cycle for two reasons:

i) The analysis of the gas mean velocities required the capability to identify the same corresponding point on each cycle. In this case the TDC of the induction stroke was selected.

ii) The Analog to Digital Converter (ADC) used in the data processing procedure (Section 3.6) required a pulse to start its operation.

It was therefore arranged to generate a triggering pulse at 'TDC Induction' by closing a reed switch circuit by a magnet mounted on an aluminium rotating disc running at half the crankshaft speed. It was necessary to set the pulse to coincide accurately with the required TDC. This was achieved by using the pulse to trigger a stroboscope, the flashing light of which was then used to view the degree markings on the rotating flywheel of the engine. The reed circuit was adjusted until the TDC marking on the flywheel coincided with the fixed TDC pointer on the engine casing.

3.5.2 Recording of Data on Magnetic Tape in Analog Form

In the absence of on-line computer processing facilities it was decided to record and store the velocity data on magnetic tape.

The tape recorder used was the RACAL Store 4. It had a recording speed range of 15/16 to 60 in/sec with corresponding frequency response of 313 to 20,000 Hz. The input limit was ± 20 volts and output ± 3 volts for full deviation.
3.5.2(i) Velocity Components Record

The three components of velocity measured by the three-wire probe must be recorded simultaneously. The RACAL 4 with its four recording channels satisfied this requirement. The timing mark was recorded on the fourth channel.

The output voltages from the three bridges were always less than 5.0 volts. The output from the bridges at zero flow was about 1.5 volts. Thus it was possible to record the velocity signals directly without such intermediate operation as biasing off the zero-level. A recording speed of 15 m/s was chosen. At this tape speed the frequency response of the recording head has a bandwidth of between DC and 5 kHz. This range is suitable for the engine speeds investigated in this study. It may be noted that only the mean gas velocities are investigated.

The tape recorder was set to give an output level of ±2.0 volts.

3.5.2(ii) Gas Pressure and Temperature Record

The gas pressure was measured with a KISTLER capacitive pressure transducer (Type 601H) used in conjunction with a charge amplifier (Type 5001 SN). The charge amplifier had a frequency range of 0-180 kHz.

Fig. 3.19 shows six consecutive cycles of the gas pressure record. It was observed from an examination of several records that there were little or no cyclic variation in the mean gas pressure.

Fig. 3.20 shows a comparison between measured pressure and pressure calculated from the theoretical compression ratio using a
polytropic index of 1.33.

It will be seen that as from about 90 degrees before and after 'TDC compression' the calculated pressure is greater than the measured values by about 8%. It is thought that this may be due to a small over-estimation of the compression ratio resulting from some gas leakage that was not accounted for.

The mean gas temperature was measured using the standard method of resistance thermometry. Hassan (14) showed that for a suitable aspect ratio the resistance thermometer can measure gas temperature with an error of 6%. In this study the aspect ratio of the wire used was about 170. A theoretical error analysis (Appendix 3A) shows that a wire with such a low aspect ratio may give a measurement error of -20%. In fact in the present study the measured gas temperature is 12% less than the calculated gas temperature at TDC. Fig. 3.21 shows the comparison between the measured gas temperature, the calculated temperature from theoretical compression ratio and the gas temperature calculated from the measured gas pressure. This latter gas temperature has a peak value 9% greater than the measured temperature. It is also noted that the measured temperature is higher than the calculated values in the first 60 degrees of the compression stroke. It was decided to use the gas temperature values calculated from the measured gas pressures to evaluate gas velocities.

3.5.2(iii) Calibration of the Tape Recorder Channels

It was necessary to calibrate the tape recorder channels in order to determine the attenuation ratio (Input/Output) for each channel, and to account for the standing DC level present on each channel even when the input is zero.
For these reasons each of the channels was calibrated after every test. The tape recorder settings were maintained as at recording. Known DC levels were then recorded on each channel in the range 0-5.0 volts which covered the minimum and maximum values from the engine data. Later, in the data processing procedure (Section 3.6) these recorded DC levels are used to provide the calibration factors to be used with the digitised engine data.

3.6 Processing of Engine Data

3.6.1 Digitisation of the Velocity and Pressure Signals

The pressure and velocity records were examined for evidence of cyclic variation. It was obvious that for the pressure there was no cyclic variation. Small variations were noticed in the velocity record. These small variations occur in the induction and exhaust periods and in the first 90 degrees of the compression stroke and the last 90 degrees of the expansion stroke. These are the regions of least interest in the present study. The velocities are very low and have little effect on the period 60 degrees of either side of the compression TDC. At and around TDC the cyclic peaks of the velocity traces remained nearly constant. After some examination it was decided that by taking an average of six consecutive cycles the ripples on the mean velocity traces were smoothed out. Fig. 3.22 shows a comparison between a mean trace averaged over six cycles and a typical single cycle record. Fig. 3.23 shows a plot of six consecutive cycles.

The recorded signals were digitised on the 'Analog to Digital Converter' (ADC) of a Hewlett Packard Fourier Analyser (Model 5451A). The tape recorder was replayed at the recording speed (15 in/sec). The ADC had fixed sampling rates varying from 10 \( \mu \)sec/sample to 2m sec/sample. The selection of a suitable sampling rate was based
on two considerations:

i) the necessity of reproducing the original signal accurately,

and

ii) the practical point of keeping the volume of generated data within manageable proportions.

It was found that the velocity profiles were such that no details were lost by arranging to have sample widths of between 5 and 7 degrees for the engine speeds investigated.

The digitised data output from the ADC are obtained on paper tape. The output format of the ADC is not directly compatible with the input format to the central ICL computer. The transformation to ICL format was achieved by a simple FORTRAN programme using the scale factors generated by the ADC. These intermediate data were stored on cards to permit flexibility of use in other data analysis programs.

3.6.2 Tape Recorder Calibration Factors to be Used with the Digitised Data

The tape recorder (RACAL Store 4) has stepped input sensitivities from ± 0.1 volts to ± 20 volts. However, the reproduce level is limited to ± 3.0 volts for full deviation. This means that an input signal is attenuated on recording and it is necessary to determine the attenuation factor each channel. It is for this reason that the calibration voltage levels were recorded as referred to in Section 3.5.2.(iii).

The known recorded DC voltages were digitised and a graph plotted of known DC voltages against the corresponding digitised values, the calibration graph for each channel of the tape recorder is linear. The slope of the line is the attenuation factor. Equation 3.24 describes the conversion of the digitised recorded velocity and
pressure signals to the actual signals input at the tape recorder.

\[ BV_{\text{actual}} = AF \times (BV_{\text{digitised}} - BV_I) \] .... 3.24

where

- \( BV_{\text{actual}} \) is the actual output from either the anemometer bridge or the pressure charge amplifier
- \( AF \) is the attenuation factor
- \( BV_{\text{digitised}} \) is the digitised values of the recorded signal
- \( BV_I \) is the intercept on the digitised value axis.

It was discovered that when a DC of zero was input at the tape recorder and recorded the actual value as given by equation 3.24 was not zero. This was due to the fact that there was always a fixed DC level output by the ADC due to some fault in the system. This voltage, although small, would be significant if it was not accounted for in the velocity computations. The voltage remained constant during a particular run of data digitisation but varied slightly between runs carried out on different occasions. This problem was resolved by using the pressure signal as the basis for finding \( BV_I \) as in equation 3.24. The attenuation factor for each tape recorder channel remained constant at all times. The intercept \( BV_I \) was chosen such that the values of gas pressure during induction up to inlet valve closure was nearly equal to the ambient value. This formed a consistent basis for applying equation 3.24 to the digitised data.

3.6.3 Computer Programs for Calculating the Engine Velocity Data

The gas velocity vectors in the prechamber were calculated at the particular crank angles decided during the digitising procedure. The vectors were calculated as outlined in Section 3.1 from the
components measured by each of the three hot wires. A program called ENGDATA was written in FORTRAN IV language and a listing is given in Appendix 3B. ENGDATA utilises the wind tunnel calibration data plus the "temperature correction factor" as discussed in Section 3.3. The measured gas pressure and the gas temperature calculated from this pressure were used to obtain the instantaneous gas and wire properties (Section 3.5.2(ii)).

It was found that the radial velocity profiles measured in the prechamber were not always uniformly of a forced vortex profile. This is discussed in detail in Chapter 4. The mathematical model (Chapter 2) had been developed on the basis of assuming a forced vortex profile in the prechamber. It was decided to calculate an equivalent radial forced vortex profile similar to that outlined by Tindal et al (20). It is assumed that velocity varies linearly between the radial measuring points. In the present study the measured velocity at the centre of the prechamber, although small, is not zero. A typical profile is shown in Fig. 3.25. The momentum represented by the area under this measured profile was equated to that under a forced vortex line such that it was possible to determine an equivalent solid body velocity. It is this equivalent solid body velocity that is compared with the mathematical model (Chapter 2). The computer program called "ESBRV" used is listed in Appendix 3C.
Fig. 3.1 Sense of velocity vector in prechamber

Fig. 3.2 Velocity components detected by hot wire
Fig. 3.3 Three-dimensional coordinate system

Fig. 3.4 Velocity vector in a three-dimensional coordinate system
Fig. 3.5  Illustrative sketch of probe body

- 2mm approx. 

- wire supports 

- glazing fluz (350 deg. C m.p.) 

- h.m.p. solder (320 deg. C) 

- copper twin conductor mineral insulated cable 

- 6mm dia. stainless steel tube 

- epoxy resin 

- gold plated contact pins and sockets 

- locking thimble 

- co-axial cable
Fig. 3.6 Three-wire probe body and lead assembly

Fig. 3.7 Three-wire probe tip
Fig. 3.8 Installation of hot-wire probe in the 1" diameter cylindrical prechamber
Fig. 3.9 Davies and Fisher hot wire anemometer circuit
\[ \alpha = \frac{R - R_0}{R_0} \times (T - T_0) \]

Platinum - 20% Iridium

Fig. 3.10 Experimental determination of temperature coefficient of wire material
Figs 3.11a, b   Determination of sensing wire resistance at ambient temperature

- Pt - 20% Iridium wire
- \(1.52 \text{ mm} \times 10 \text{ m}\)

- \(U = 64.0 \text{ m/s}\)
- \(U = 28.8 \text{ m/s}\)
- \(U = 0 \text{ m/s}\)
Sectional view

Type 5SD41 Calibration Unit with Nozzle, Diffuser and Airshunt

Fig 3.12 DISA CALIBRATION WIND TUNNEL

PRESSURE DIFFERENCE $\Delta p$ VERSUS
FLOW VELOCITY $v$ FOR AIR ($k=1.4$).
RESERVOIR CONDITIONS: $P_0$ mm Hg, $T_0$ °K

Calibration Curve for the Wind Tunnel
Fig. 3.13  Typical hot wire calibration curves

Wire length = 1.80 mm
Wire diameter = 10 μm
Cold resistance = 6.20 ohm
Lead resistance = 0.40 ohm
wire material Pt - 20% Ir.
Fig. 3.14 Experimental verification of consistency of temperature correction factor.
Fig. 3.15 Verification of 'Sine law' - $U = U_{\text{max}} \sin \theta$
Fig. 3.16 Error estimation in velocity magnitude measured with 3-wire probe system in wind tunnel
$U_{\text{ref}} = 14.6 \text{ m/s}$

$U_{\text{ref}} = 87.0 \text{ m/s}$

expected values

Fig. 3.17 Error estimation in horizontal and vertical angles
Fig. 3.18  Velocity detected by hot wire when flow is along the wire length

Fig. 3.19  Consecutive cycles of prechamber pressure
Comparison of calculated and measured pressure in the prechamber

Fig. 3.20  Comparison of calculated and measured pressure in the prechamber
Fig. 3.21 Gas temperature in the prechamber

1" cylinder
N = 900 rpm
CR = 12.00

Gas temperature (deg. C)

Crank angle (deg)
Fig. 3.22 Comparison of single cycle velocity trace and average of six consecutive cycles

Fig. 3.23 Six consecutive cycles of gas velocity trace taken from prechamber experimental data
Fig. 3.25  Equivalent solid body velocity configuration
CHAPTER 4

A COMPARISON OF THE MATHEMATICAL MODEL WITH EXPERIMENTAL MEASUREMENTS AND DISCUSSION OF THE RESULTS
4.1 Introduction

This Chapter is a report of the experimental measurement of mean velocity profiles in two cylindrical prechambers by means of the three-wire probe described in Section 3.2. The measured absolute velocity vectors were resolved to give the swirl components at each of the three measuring positions spaced around the prechamber circumference. An equivalent solid-body rotation profile was obtained from the average of the three positional swirl components as outlined by Tindal et al (20).

The results of the mathematical model described in Chapter 2 were compared with the equivalent solid-body velocities. The mathematical model was also applied to previously published experimental data obtained on engines (14) (17) (21) and comparisons made. The discrepancies between the experimental and the mathematical model results suggested that it would be beneficial to investigate the possibility of correlating air swirl velocity in the prechamber with engine dimensions and speed. This investigation led to the proposal for a factor enabling the theoretical and experimental results to be matched in the period 45 degrees BTDC to TDC. This factor, being defined in a general way, can thus be determined for any particular set of engine data and used in the mathematical model.

4.2 Experimental Program

The test engine was a Ruston and Hornby WB four-stroke, air-cooled, single cylinder engine with a side valve arrangement and a flat-topped piston. The engine geometric specifications are given in Table 4.1.
The cylinder head was specially fabricated in order to achieve the idealised cylindrical shape of the prechamber and to make the circular throat tangential to the curved periphery of the prechamber at their intersection. Fig. 4.2 is a sketch showing the main features of the prechamber arrangement. The side valve arrangement results in recesses both around the valves, and in the cylinder head to accommodate the valve lift. It was therefore inevitable that there was a large clearance volume and for this reason the maximum compression ratio that could be investigated was 12:1, for a realistic prechamber volume of 8.0 cm\(^3\). The throat and the prechamber were machined in separate pieces of material in order to enable the interchange of the one-inch and the two-inch diameter cylindrical prechambers in the same cylinder head. The throat was inclined at 45 degrees to the cylinder block.

There were three probe positions spaced around the prechamber circumference where radial profiles of mean velocity were measured. The position labelled A was along the horizontal diameter immediately downstream of the throat opening into the prechamber, while position B was on the vertical diameter and position C was diametrically opposite to A. When measurements were being made in one position the other two positions were blanked off by plugs.

The assembly of the three-wire probe was described in Section 3.2. A micrometer screw arrangement was used to enable the traversing of the probe across the prechamber. Provision was made such that the probe could be rotated on its axis. By measuring the mean velocity with the probe set at different orientations it was hoped that the effect of masking of the wires by the pin supports and the disturbance of the flow by the probe would be minimised if the average of the measurements were taken.
Radial profiles of velocity were measured in both the one inch and the two inch diameter cylindrical prechambers. There were four radial locations in the one inch and seven locations for velocity measurement in the two inch prechamber. These radial locations are shown in Fig. 4.3. The existing motoring arrangement for the test engine was such that the upper limit of engine speed was 1100 rpm. This is low and was thought to be a disadvantage at the beginning of the experimental measurements. Subsequently, a lot of breakage of the hot-wire was experienced in the engine and it became apparent that it would have been very difficult to make velocity measurements at engine speeds much greater than 1100 rpm. The three speeds investigated for the one inch prechamber were 600, 900 and 1100 rpm. For the two inch prechamber a complete set of measurements were made at 600 rpm, but it was not possible to make measurements at the higher speeds because the wires broke persistently. This was due to the combination of engine vibration and high swirl velocities in the prechamber. The only velocity measurements at 1100 rpm obtained in the two inch prechamber was at the probe position B at locations 1 and 2.

Fig. 4.4 is a photograph of the experimental equipment used in the tests and Fig. 4.5 is a photograph of the prechamber assembly.

4.3 Absolute Velocity Vectors in the One Inch and the Two Inch Diameter Cylindrical Prechambers

It is necessary, for the understanding of the following discussion, to recall the representation of a velocity vector in terms of its magnitude and the two vector angles referred to above as the horizontal and vertical angles (see Section 3.1.2). For the
case of the cylindrical prechamber in the present study the chosen reference planes are the two vertical planes V-V and H-H, as shown in Fig. 4.6. The vertical angle, VA, is the angle that the velocity vector makes with plane V-V. The horizontal angle, HA, is the angle which the normal projection of the velocity vector on plane V-V makes with the vertical on plane H-H. This angle is not equal to the normal inclination of the velocity vector to plane H-H.

If the air motion in the prechamber is truly a solid-body rotation, as was assumed in the mathematical model, then, at the measuring position A both the vertical and the horizontal angles will be zero when the probe is set at the reference orientation (Orientation 1). As was mentioned above, velocity measurements were made with the probe set at three different orientations. Orientation 2 was 90 degrees clockwise and Orientation 3 was 135 degrees clockwise from the reference Orientation 1. The velocity vectors measured with the probe at Orientations 2 and 3 were referred to the reference planes V-V and H-H. So, at the measuring position A the reference horizontal and vertical angles are always zero for solid swirl. At position B the reference horizontal and vertical angles are both 90 degrees for the ideal case of solid-body swirl. At position C the reference angles are both zero as for A except that the sense is opposite.

4.3.1 Comparison of Absolute Velocity Vectors Measured with the Probe set at Three Different Orientations

Figs. 4.7a, b, c and 4.8 a, b, c show the mean velocity vectors in the one-inch diameter cylindrical prechamber as measured by the
three-wire probe at the three different orientations. The results shown are at the probe position C for which the reference horizontal and vertical angles should be zero. It is obvious that there is good agreement of the magnitudes of the velocity vector. The error analysis (Section 3.4) predicted a measurement error in the velocity magnitude of between -5% and +15%. For the location 1 data the velocity magnitudes agree within the above error margin. The agreement is not as good for the location 3 data, the discrepancy in peak velocities being as much as 50%. It is thought that the main reason for this discrepancy was due to the masking of the probe wires at Orientation 3.

An examination of Figs. 4.7a, b, c and 4.8a, b, c shows that the vertical angles in most cases, in comparison to the reference value of zero, agree within the predicted error of 25 degrees especially during the period 40 degrees BTDC to 40 degrees ATDC. All the horizontal angles should also be zero, but the predicted error is ±10 degrees. It can be seen that the measured horizontal angles do not always fall within the limits of ±10 degrees.

The data plotted in Figs. 4.7a, b, c and 4.8a, b, c are typical of the rest of the absolute velocity data. It may therefore be concluded that, in general, there is fairly good agreement in the magnitude of the velocity vector and in the horizontal and vertical angles when comparison is made between data obtained at three different probe orientations. The reasons for discrepancy is thought to be due to the masking of the wires by the probe supports, the velocity vector being incident on the wires at angles where the 'Sine law' does not apply, and the unpredictable interference caused to the flow by the probe body and the wire supports.
4.3.2 Effect of Measuring Position Around Cylindrical Prechamber Circumference on the Measured Velocity Vectors

The three measuring positions A, B and C are as shown in Fig. 4.3 and described in Section 4.2. Figs. 4.9a, b, c shows the comparison of the velocity vectors for the one-inch prechamber at an engine speed of 1100 rpm and radial location 1 (Fig. 4.3). It is apparent that there is good agreement in the magnitudes of the mean velocity. The set of data for radial location 3 are shown in Figs. 4.10a, b, c.

The measure of attainment of solid-body rotation are the horizontal and vertical angles. These lie between 5 and 30 degrees for data obtained at positions A and C. In comparison with the expected angles of zero for a full solid-body rotation, these show a departure from solid swirl. For position B the reference angles for solid swirl should be 90 degrees. An examination of Figs. 4.9b and 4.10b again shows some departure from solid swirl although it should be remembered that the expected error in the horizontal angle is ±10 degrees. The observations above hold in general for all three engine speeds of 600, 900 and 1100 rpm.

The absolute velocity vector data for the two-inch diameter cylindrical prechamber are shown in Figs. 4.11a, b, c to 4.13a, b, c for the radial locations 1, 3 and 5. Again it is apparent that there is good agreement in the magnitude of mean velocity for the three positions A, B and C. The peak velocities agree within the expected measurement error band of -5% to +15%. The horizontal and vertical angles show that the departure from solid-body rotation is more in the two-inch prechamber in comparison to the one-inch prechamber.
It is thought that this is due to the fact that in the two-inch prechamber the radius of rotation is large. Therefore at low engine speeds the jet entering the prechamber from the throat is not strong enough to maintain full solid swirl. This effect is similar to that observed by Horvatin (17) for the data obtained with the smaller diameter throat.

4.3.3 Radial Variation in the Absolute Mean Velocity Vector

An examination of Figs. 4.9a, b, c and 4.10a, b, c shows that for the one-inch prechamber the mean velocity decreases significantly between location 1 and location 3. If the flow was entirely solid swirl the decrease in velocity between locations 1 and 4 would be proportional to the radius of location. For the one-inch prechamber the decrease in velocity between locations 1 and 2 is more than linear, but becomes nearly linear between locations 2 to 4. It is thought that this may be due to the throat jet having a greater effect near the prechamber wall than near the centre.

For the two-inch prechamber an examination of Figs. 4.11a, b, c to 4.13a, b, c shows that the magnitude of the mean velocity at location 3 is still nearly equal to that of location 1. This contrasts with the sharp decrease between locations 1 and 2 for the one-inch prechamber. It is noted that in the two-inch prechamber the throat jet and prechamber swirl have longer paths to travel within the prechamber. So, whereas in the one-inch prechamber the effect of jet may carry to positions A and B, the jet would have been totally diffused in the two-inch prechamber before it reaches position A. This observation is supported by evidence of the behaviour of film cooling jets. Sturgess (50) has given some data for the case of application of film cooling to aircraft gas turbine chambers.
He defined a 'potential core' which is representative of the mixing region between the main stream flow and the cooling film injected through a slot (Fig. 4.38). In the present case the flow boundary is the prechamber wall and the slot is the throat. There is some difference in the geometric configuration because of the curvature of the prechamber wall and the circular cross-section of the throat. However, if it is assumed that the film jet behaviour will be similar, then an estimate can be made of the mixing region for the present case of the one inch and the two inch diameter cylindrical prechambers.

The data of Sturgess (50) is shown in Fig. 4.39. In the present case the velocity ratio \( \frac{U_{\text{free stream}}}{U_{\text{jet}}} \) is the ratio of the gas velocity in the prechamber to the velocity of the gas jet from the throat. From the mathematical model computations this velocity ratio is about 1.5 for the prechambers at about 15 degrees BTDC and 15 degrees ATDC. The slot height, \( s \), is taken to be the throat diameter. From Fig. 4.39 the data for the two slots suggest that the potential core length ratio \( \frac{X_p'}{s} \) is approximately proportional to the square of the inverse of the slot height. On this basis it can be estimated that for a velocity ratio of 1.5 and a slot height of 0.30" (throat diameter) the corresponding potential core length ratio is approximately equal to 4. This gives a potential core length of 1.2". The circumferential distance from the throat opening to the measuring position B is about \( \frac{\pi D_p}{2} \). So for the one inch prechamber this distance is 1.50" and 3.0" for the two inch prechamber. Comparison of these distances with the potential core length of 1.2" indicates why the effect of the throat jet may be detected at positions A and B in the one inch prechamber, but not in the two inch prechamber.
A major objection to the direct comparison of the magnitudes of the absolute mean velocity vectors is the fact that the horizontal and vertical angles vary for different positions and locations. Direct comparison of the magnitudes of vectors can only be valid if corresponding vector angles are equal. It can be seen from Figs. 4.9a, b, c to 4.13a, b, c that although the vector angles lie, in the main, within the same range the corresponding instantaneous values are not equal. For this reason it was decided that it would be preferable to resolve the absolute velocity vectors in the direction of solid-body rotation. If all vectors are thus resolved then direct comparison of the resolved components can be made. This course of action was also justified in view of the fact that the mathematical model was based on the assumption of solid-body rotation of the gas mass in the prechamber.

4.4 Swirl Components of Velocity in the One Inch and Two Inch Cylindrical Prechambers

4.4.1 Comparison of the Swirl Velocities at the Measuring Positions A, B and C

As was mentioned in Section 4.3.2 the absolute velocity vectors were resolved to give the solid swirl components. It is further considered that the average of swirl velocity for the three probe orientations would be the best measurement of velocity at each of the measuring positions A, B and C. These averaged results are shown in Figs. 4.14 to 4.16 for the one inch prechamber.

It can be seen that in most cases the swirl velocity at position A is larger by up to 30% than the swirl velocities at positions B and C. The same effect of the throat jet noted above for the absolute velocity still holds for the swirl velocities. The differences in
swirl velocities are more apparent at locations 1 and 2. If a full solid-body rotation exists in the prechamber then the velocities at location 4 (centre of prechamber) should be equal to zero in all cases. Evidence from Figs. 4.14 to 4.16 shows that small velocities were measured at the centre of the prechamber. It is not surprising that the velocities at location 4 are not equal to zero in view of the fact that the three-wire arrangement occupied a cube of side 3 mm. The probe is therefore unable to make exact point measurements. This point must be remembered when examining the experimental results in this study.

The swirl velocities in the two inch prechamber are shown in Fig. 4.17. There is better agreement between the velocities at positions A, B and C. The throat jet effect detected in the one inch prechamber is absent here. Another difference between the two sets of data is that peak velocity occurs at about 20 degrees BTDC in the two inch prechamber, but at or very close to TDC in the one inch prechamber. This confirms that the swirl in the one inch prechamber is dissipated less quickly than in the two inch prechamber as the piston approaches TDC.

4.4.2 Radial Profiles of Swirl Velocity in the One Inch and Two Inch Cylindrical Prechambers

Following from Section 4.4.1 and Figs. 4.14 to 4.17 a selection of radial velocity profiles at particular crank angles are shown in Figs. 4.18 to 4.21.

For the one inch prechamber, it can be seen from Fig. 4.18 that at the four selected crank angles, (30 degrees BTDC, 15 degrees BTDC, TDC and 15 degrees ATDC), the velocity profiles for an engine speed of 600 rpm are nearly linear although they do not pass through
the origin (centre of prechamber) as would be the case for solid swirl. An 'equivalent solid body rotational velocity' (ESBRV), was calculated as outlined in Section 3.6.3.

At the higher engine speeds of 900 and 1100 rpm the radial velocity profiles still show a consistent decrease in velocity from the prechamber wall towards a minimum at the centre. However, the radial profiles tend to be concave near TDC. This suggests that the throat jet which is tangential to the prechamber wall does not mix uniformly inside the prechamber and has a relatively greater effect at the location nearer to the prechamber wall. This effect is retained for between 30 to 50 degrees ATDC.

For the two inch prechamber the radial profiles of velocity also show a linear trend (Fig. 4.21). However, the decrease in swirl velocity from location 1 towards location 7 at the centre of the prechamber is less well defined as in the one inch prechamber. The concave profile noted above for the one inch prechamber is also absent.

4.5 Comparison of the Mathematical Model Computations with the Experimental Results of the Present Study

It will be recalled that the mathematical model employed the principle of conservation of angular momentum to compute the mean velocity in the prechamber during the compression and expansion periods for the case when the test engine was motored (Chapter 2). The experimental measurements of absolute mean velocity made with the three-wire probe had shown some departure from the theoretically assumed solid swirl. For this reason it was decided to calculate an equivalent solid body rotational velocity, ESBRV, as outlined in Section 3.6.3.
The comparisons of the theoretically computed mean velocities in the prechamber with the 'ESBRV' of measured data are shown in Figs. 4.22 to 4.29.

It is seen from Figs. 4.22 to 4.24 that the agreement between the model and the experimental results is poor in the case of the one inch prechamber. For the three engine speeds of 600, 900 and 1100 rpm the experimental and the mathematical model results are nearly equal in the earlier part of the compression stroke and the latter part of the expansion stroke. But these parts are not very important. The important part of the engine cycle as regards air motion, lies between 60 degrees BTDC and 60 degrees ATDC. In this region the experimental results are greater than the theoretical model computations by up to 100%. It is also noted that the peak of the theoretical results occur at about 15 degrees BTDC whereas the peak of the experimental results occurs at about TDC.

For the two inch prechamber the comparison of experimental measurements and the model is shown in Fig. 4.25 for an engine speed of 600 rpm. The experimental results are lower than the theoretical model results by about 30% in the region 60 degrees BTDC to 60 degrees ATDC. This agreement is much better than in the case of the one inch prechamber. However the discrepancies are in opposite directions. The peak velocity occurs at about 15 degrees BTDC. This is also in contrast to the one inch prechamber. There is good agreement between theory and experiment for the 1100 rpm data of the two inch prechamber, the discrepancy in peak velocities being about 15%. However, although this agreement is good the experimental results for the two inch prechamber for an engine speed of 1100 rpm is not considered conclusive.
It was not possible to obtain the full set of data in all probe positions and locations. The experimental velocity curve shown in Fig. 4.25 is for a single measurement at location 1 in the measuring position B. It was thus not possible to calculate an 'ESBRV' for the two inch prechamber at 1100 rpm engine speed.

The discrepancies between the experimental results and the theoretical model computations cannot be entirely explained by reference to the expected measurement error of between -5% to 15% (Section 3.4). It would appear that the mathematical model is an over-simplification of the actual air motion in the prechambers. As was pointed out in Section 2.8 it was not possible to obtain a value of discharge coefficient representative of the flow between the main cylinder and the prechamber. The discussion of the discrepancy between the theoretical model and the experimental results will be resumed after the comparison of the model computations with the published data of other authors.

4.6 Comparison of the Mathematical Model Computations with other Published Experimental Measurements of Mean Velocity in the Prechamber of Indirect Injection Diesel Engines

The survey of published literature (Chapter 1) had shown that there are only a few publications of experimental measurement of mean velocity in the prechamber of indirect injection diesel engines. Hassan (14) used a single-wire probe to measure velocity in a two inch diameter cylindrical prechamber at (using the notation in the present study) probe position C location 1. Horvatin (17) also used a single-wire probe to measure gas velocity in a 54 mm diameter cylindrical prechamber at two radial locations and two engine speeds. Nakajima (21) used a spark discharge method to measure prechamber
radial velocity profiles at four particular crank angle periods. These three sets of data are not as detailed as in the present study and it was not possible to calculate 'ESBRV' for them.

Fig. 4.27 shows the comparison of Hassan's experimental data with the mathematical model computation. The experimental results are greater than the theoretical computations by up to 100%. This is similar to the one inch prechamber data of the present study. Hassan's prechamber was used with the same single cylinder test engine used in the present study. It might have seemed that Hassan's data would be more similar to the two inch prechamber data of the present study in comparison with the mathematical model. The major difference between the two prechambers is that the thickness of Hassan's cylindrical prechamber was exactly equal to the throat diameter of 9.5 mm, whereas the thickness of the two inch prechamber in the present study was 15 mm and the throat diameter was 7.62 mm. The prechamber thickness in the present study was made so large because it was considered important to have a large clearance between the probe body and the side walls of the prechamber. This would minimise the disturbance to the air motion in the prechamber. It would appear that the effect of the ratio of prechamber thickness to throat diameter has not been entirely accounted for by the mathematical model.

The effect of the thickness of the prechamber can be illustrated as outlined below. In the case of Hassan's prechamber in which the prechamber thickness was exactly equal to the throat diameter, the jet from the throat is constrained and cannot expand as it discharges into the prechamber. In the case of the two inch diameter prechamber in the present study the thickness of the prechamber is greater than the throat diameter. Hence the jet from the throat can expand as it
discharges into the prechamber and this may also lead to the formation of secondary flows (Fig. 4.40).

Fig. 4.28 shows the comparison of Horvatin's experimental results with the computations of the mathematical model. In this case the experimental results are again higher than the theoretical results by up to 100% between 70 degrees BTDC and 15 degrees BTDC. The agreement becomes much better from 15 degrees BTDC to TDC and the discrepancy at TDC is between 7% and 22%. It is also noticed that the peak velocity occurs at about 30 degrees BTDC for the experimental measurement. This is opposed to peak velocity occurring at 15 degrees BTDC for the mathematical model, 20 degrees BTDC for the two inch prechamber and about TDC for the one inch and Hassan's prechambers.

Fig. 4.29 shows the comparison of Nakajima's data with the mathematical model computations. It may be noted that a complete record of prechamber mean velocity versus engine crank angle has not been given by Nakajima et al (21). In addition the given velocity profiles are averaged over 4 degree periods. The experimental results are much lower than the theoretical results - by up to 40% at 1500 rpm and 90% at 3000 rpm.

4.7 Some Comments Arising from the Comparison of the Mathematical Model with Present Experimental Measurements and with Previous Published Experimental Data by Other Authors

The discrepancies between the experimental data and the theoretical model computations do not follow any consistent pattern. For the experimental data in the present study the discrepancies are greater, in most cases, than the estimated velocity measurement error of between -5% and 15%. There are no values quoted for the measurement errors in the data of Hassan (14), Horvatin (17) and Nakajima et al (21).
However, it is unlikely that such error values would explain the large discrepancies observed.

For the mathematical model the effect of engine speed on the mean velocity of gas in the prechamber is linear. In the present study the experimental measurements show that the engine speed effect is less than linear for the one inch prechamber, but more than linear for the two inch prechamber although it should be remembered that the two inch prechamber data at 1100 rpm is not conclusive (Section 4.5). The engine speed effect is slightly less than linear for both Horvatin's and Nakajima's data.

By neglecting the friction term in the analysis of the air velocity in the prechamber there is an increase in prechamber velocity of between 5 and 10% for the five prechambers already discussed. This effect of neglecting wall friction does not alter the discussion above in any significant way.

Another consideration that may be significant is the type of flow in the prechamber, due to the mixing of the throat jet with the gas already in the prechamber. The attainment of a high degree of swirl may be enhanced by the smaller radius of rotation in the one inch prechamber, but hindered by the large radius of rotation in the two inch prechamber. It also seems likely that swirl (two dimensional flow) will be more pronounced if the prechamber thickness is small as in Hassan's prechamber, but swirl will be dispersed and three-dimensional effects would prevail with larger prechamber thickness as in the two inch and Nakajima's prechambers. Examination of all the experimental results in conjunction with the engine's geometric data tabulated in Fig. 4.1 indicates that the ratio of prechamber volume to the main cylinder swept volume may also be a very important factor.
The observations above encouraged the investigation of a correlation between given engine's data and the measured air velocities in the prechambers discussed in the present study.

4.8 A Correlation Between Mean Air Velocity in the Prechamber and Known Engine Dimensions and Speed

The correlation of experimental data is a well tried technique commonly employed in engineering. However this technique has not been extended to the investigation of the gas motion in the pre-chamber of an indirect injection engine. Because of the absence of any previous investigation on this line a method of 'trial and error' was adopted. It was attempted to find a relationship between a prechamber Reynolds's number based on swirl velocity and a main cylinder Reynolds's number based on the engine speed. The various engine geometric parameters were examined. For the main cylinder the two dimensions of interest are the bore diameter, \( D \), and the piston stroke, \( S \). It is noted that the mean piston speed is proportional to the product of engine speed and piston stroke. If the mean piston speed is chosen as the characteristic velocity in the main cylinder Reynolds's number then the obvious dimension to be taken as the characteristic length is the bore diameter. For the cylindrical prechamber the important dimensions are the diameter \( D_p \) and the thickness \( b_l \). If the characteristic velocity chosen for the prechamber Reynolds number is the swirl velocity, \( U_{\text{swirl}} \), at the periphery of the prechamber, then it may be argued that the effect of the prechamber diameter is included in \( U_{\text{swirl}} \). The thickness of the prechamber, \( b_l \), could then be taken as the characteristic length.

It was recognised that the prechamber and the main cylinder properties must be related by the size of the throat connecting them.
The theoretical model had shown that a reduction in the ratio of throat cross-sectional area to piston area \((a/A)\), leads to an increase in the throat velocity and hence to an increase in the mean velocity of the gas in the prechamber. It had also been shown that an increase in the ratio of prechamber volume to the total volume at TDC, \(\sigma^*\), results in a proportional increase in the prechamber gas velocity. The above examination of the engine geometric parameters formed the basis of the derivation of the correlation outlined below.

A Reynolds number, \(R_{em}\), was defined for the main cylinder as:

\[
R_{em} = \frac{\rho_i \cdot \left( \frac{2\pi NS}{60} \right) \cdot D}{\mu_i} \tag{4.1}
\]

where
- \(N\) = engine speed (rpm)
- \(S\) = piston stroke (m)
- \(D\) = cylinder bore (m)
- \(\rho_i\) = instantaneous gas density (kg/m\(^3\))
- \(\mu_i\) = instantaneous dynamic viscosity (kg/m.s)

It is noted that the term in brackets is the mean piston speed and that the characteristic length is the cylinder bore diameter.

A 'Reynolds Number', \(R_{ep}\), was defined for the prechamber as:

\[
R_{ep} = \frac{\rho_i \cdot U_{swirl} \cdot b_l}{\mu_i} \cdot \left( \frac{a}{A} \right) \cdot \left( \frac{1}{\sigma^*} \right) \tag{4.2}
\]

where
- \(U_{swirl}\) = 'equivalent solid body rotational velocity' (ESBRV)
- \(b_l\) = prechamber thickness
- \(a\) = throat cross-sectional area
- \(A\) = cylinder bore cross-sectional area
- \(\sigma^*\) = ratio of prechamber volume to total volume at TDC.
Here it will be noted that the term \((\frac{a}{A})\), the ratio of throat cross-sectional area to the cylinder cross-section, represents the sudden contraction in the flow between the cylinder and the prechamber. The term containing \(\sigma\) takes into account the proportion of total trapped mass present in the prechamber as the piston comes up to TDC. The characteristic length, \(b_1\), has been chosen because in the cylindrical prechamber the throat is situated symmetrically on the thickness of the prechamber. It had been pointed out above that the prechamber diameter is indirectly involved in the swirl velocity \(U_{\text{swirl}}\). Instantaneous values of \(U_{\text{swirl}}\) were selected in the period 45 degrees BTDC to TDC. This period was selected because it is the most relevant part of the cycle with regard to the mixing of injected fuel and air. The data for the five cylindrical prechambers previously discussed were included.

Fig. 4.30 shows a log-log plot of the prechamber "Reynolds number", \(Re_p\), versus the main cylinder Reynolds number, \(Re_m\). A curve was fitted to the data by means of a least squares polynomial procedure which searched continuously until the highest degree of polynomial with a least squares fit was found. The resulting correlation was a straight line of the form:

\[
Re_p = a + b Re_m
\]  

(4.3)

where \(a\) and \(b\) are constants.

In the present case:

\(a = -1881\)
\(b = 0.556\)
There are two major points of criticism to note in Fig. 4.30 - (i) the scatter in the data and (ii) the limited ranges of $Re_m$ and $Re_p$. The problem of the limited range can only be solved if there were more published studies of the measurement of air velocity in various prechambers. Because of the scatter in the data this correlation in the present form of equation 4.3 can only be used within the ranges of Reynolds numbers for which it has been obtained.

The above correlation will need to be tested with other experimental data when they become available. Its usage will depend on its consistency. At present it can only be regarded as a first step.

4.9 An Empirical Factor Enabling the Matching of the Mathematical Model with the Experimental Measurements

Following from the experimental correlation described in Section 4.8 it was considered possible to determine a factor that may be used to match the mathematical model to the experimental measurements. This factor would be defined in a general way and could be determined empirically for any given prechamber engine data and used in the mathematical model.

After considering various possibilities it was decided to adopt a relationship for the factor, $FM$, of the form:

$$FM = f \left( \frac{V_P}{V_h}, \frac{Re}{Re_n} \right) \quad (4.4)$$

where $V_P =$ prechamber volume

$V_h =$ cylinder swept volume
\[ Re_n = \frac{\rho_o \left( \frac{2\pi N_s}{60} \right) R_p}{\mu_o} * \left( \frac{A}{a} \right) * \left( \frac{b}{d} \right)^2 \] (4.5)

where \( R_p \) = prechamber radius
\( d \) = throat diameter

It may be noticed that the Reynolds number \( Re_n \) is a partial combination of \( Re_p \) and \( Re_m \) in the correlation equation 4.3.

Since the intention was to match the mathematical model to the experimental results the correction factor, \( FM \), was defined by:

\[ FM = \frac{U_{swirl}}{U_{theory}} \] (4.6)

and the 'corrected' velocity, \( U_{match} \) will be given by:

\[ U_{match} = U_{theory} \times FM \] (4.7)

Equation 4.4 can be rewritten as:

\[ FM = \left( \frac{V_p}{V_h} \right)^j \left( Re_n \right)^k \] (4.8)

This equation contains two unknown constants \( j \) and \( k \), values of which may be obtained by introducing two known conditions. In the present study we have available the experimental measurements of velocity in both the one inch and the two inch diameter cylindrical prechambers.

For the same reason as given in Section 4.8 in discussing the prechamber velocity correlation it was decided to apply the correction factor only in the period 45 degrees BTDC to TDC. By inspecting Figs. 4.22 and 4.25 average values of \( FM \) were chosen for the one inch and two inch prechamber data at 600 rpm. These average values were taken to apply over the 45 degrees for which the matching is
being done. On the above basis the value of FM selected for the one inch 600 rpm data was 2.0 and for the two inch 600 rpm data FM was 0.67. Equation 4.8 can now be solved to give j and k by substituting the two values of FM and the corresponding values of \( \frac{V_p}{V_h} \) and Re:

\[
2.00 = 0.021^j \times 1,632,701^k
\]

\[
0.67 = 0.082^j \times 3,340,256^k
\]

Solving equations 4.9 and 4.10 gives:

\[ j = -0.721 \]
\[ k = -0.147 \]

Therefore equation 4.8 can now be written as:

\[
FM = \left( \frac{V_p}{V_h} \right)^{-0.721} (Re_n)^{-0.147}
\]

This general relationship for the correction factor can now be used to obtain \( U_{\text{match}} \), using equation 4.7, for any other engine data.

Figs. 4.31 to 4.37 show the comparison of the 'corrected' theoretical computations with the experimental measurements. The agreement between 'corrected' theory and experiment is fairly good in most cases although there is still some significant discrepancy especially for Hassan's data and the two inch prechamber 1100 rpm data. It was recognised that it would be better to define the correction factor such that it varies instantaneously with crank angle. Some initial investigation was carried out, but it was found that there was no obvious direct way of matching the instantaneous
values of velocity. However, the fairly good agreement shown in most cases between the 'corrected' theoretical model computations and the experimental measurements indicates that a factor determined in the way outlined above could make the mathematical model a realistic representation of the air motion in a prechamber.
<table>
<thead>
<tr>
<th></th>
<th>PRESENT CYLINDRICAL PRECHAMBERS</th>
<th>HASSAN (14)</th>
<th>HORVATIN (17)</th>
<th>NAKAJIMA (21)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1&quot; Dia 2&quot; Dia</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bore, D mm</td>
<td>76.2</td>
<td>76.2</td>
<td>76.2</td>
<td>115.0</td>
</tr>
<tr>
<td>Stroke, S mm</td>
<td>82.5</td>
<td>82.5</td>
<td>82.5</td>
<td>140.0</td>
</tr>
<tr>
<td>Con-Rod Length Crank Radius</td>
<td>4.0</td>
<td>4.0</td>
<td>4.0</td>
<td>5.0</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>12:1</td>
<td>7.8:1</td>
<td>8:1</td>
<td>15.5:1</td>
</tr>
<tr>
<td>Prechamber Volume Compression Volume</td>
<td>0.25</td>
<td>0.53</td>
<td>0.39</td>
<td>0.76</td>
</tr>
<tr>
<td>Prechamber Diameter mm</td>
<td>25.5</td>
<td>51.0</td>
<td>51.0</td>
<td>54.0</td>
</tr>
<tr>
<td>Prechamber Thickness mm</td>
<td>15.0</td>
<td>15.0</td>
<td>9.5</td>
<td>33.2</td>
</tr>
<tr>
<td>Throat Area Bore Area %</td>
<td>1.00</td>
<td>1.00</td>
<td>1.56</td>
<td>2.50</td>
</tr>
</tbody>
</table>

Table 4.1 Geometric Specifications of Prechamber Engines Discussed in Present Study
Fig. 4.2  Illustrative Sketch of Prechamber (approx. \( \frac{1}{2} \) scale). See Figs. 4.1 and 4.3.
Fig. 4.3a Measuring Locations in Two-inch Diameter Cylindrical Prechamber

Fig. 4.3b Measuring Locations in One-inch Diameter Cylindrical Prechamber
Fig. 4.4  Experimental equipment used for the measurement of mean velocity in the cylindrical prechamber
Fig. 4.5  The cylindrical prechamber assembly mounted on the test engine
Reference Values of Vertical and Horizontal Angles if Swirl is Solid Body Rotation

<table>
<thead>
<tr>
<th>Probe Position</th>
<th>( \text{HA}_{\text{ref}} ) (deg)</th>
<th>( \text{VA}_{\text{ref}} ) (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>B</td>
<td>90</td>
<td>90</td>
</tr>
<tr>
<td>C</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Fig. 4.6 Reference Planes in Cylindrical Prechamber
Absolute Velocity Vector in One-inch Diameter Cylindrical Prechamber
(600 rpm, Probe Position C, Location 1)
Fig. 4.7c Absolute Velocity Vector in One-inch Diameter Cylindrical Prechamber.
(600 rpm, Probe Position C, Location 1)
Figs. 4.8a, b, c. Absolute Velocity Vector in One-inch Diameter Cylindrical Prechamber.
(600 rpm, Probe Position C, Location 3)
Figs. 4.3a, b, c
Probe Position A

Reference Angles
if Forced Vortex

Probe Position B

Absolutve Velocity
--- Horizontal Angle
--- Vertical Angle

Absolute Velocity (m/s)

Crank Angle (deg)

(a)

(b)
Fig. 4.9c Absolute Velocity Vector in One-inch Diameter Cylindrical Prechamber.

(1100 rpm, Orientation 1, Location 1).
Figs. 4.10a, b, c  Absolute Velocity Vector in One-inch Diameter Cylindrical Prechamber (1100 rpm, Orientation 1, Location 3)
Figs. 4.11a, b, c Absolute Velocity Vector in Two-Inch Diameter Cylindrical Prechamber. (600 rpm, Orientation 2, Location 1)
Figs. 4.12a, b, c. Absolute Velocity Vector in Two-Inch Diameter Cylindrical Prechamber.
(600 rpm, Orientation 2, Location 3)
Figs. 4.13a, b, c. Absolute Velocity Vector in Two-inch Diameter Cylindrical Prechamber.
(600 rpm, Orientation 2, Location 5)
Fig. 4.14 Comparison of Swirl Velocity at Probe Positions A, B and C - 600 rpm Diameter Cylindrical Prechamber, 600 rpm

Velocity

Location 1

ESBRV

Crank Angle (deg)

Location 2

ESBRV

Prechamber Swirl Velocity (m/s)

Location 3

ESBRV

Prechamber Swirl Velocity (m/s)

Location 4
Comparison of swirl velocities at three probe positions A, B, and C. Location 1, 2, 3, and 4 show the varying velocities with crank angles. The plots indicate the effect of different probe positions on the measured velocities.
Fig. 4.17 Comparison of swirl velocities at probe positions A, B, and C -

2" Diameter Cylindrical Prechamber, 600 rpm

Location 1

Location 2

Location 3

Location 4

Location 5

Location 6

Location 7

Crank Angle (deg)

Probe Position A
Probe Position B
Probe Position C

Jet Stream Velocity (m/s)
ESBRV = Equivalent Solid Body Rotational Velocity (m/s)

\[
(\frac{U_A + U_B + U_C}{3})
\]

25 mm = 50 m/s

30° BTDC

15° BTDC

Fig. 4.18 Radial Profiles of Swirl Velocity - 1" Diameter Cylindrical Prechamber, 600 rpm
Fig. 4.20  Radial Profiles of Swirl Velocity - 1" Diameter Cylindrical Prechamber, 900 rpm
Fig. 4.19  Radial Profiles of Swirl Velocity - 1" Diameter Cylindrical Prechamber, 1100 rpm
Fig. 4.21: Radial Profiles of Swirl Velocity - 2" Diameter Cylindrical Prechamber, 600 rpm.
Comparison of Equivalent Solid Body Velocity with Mathematical Model.
1" Diameter Cylindrical Prechamber, 600 rpm
Fig. 4.23 Comparison of Equivalent Solid Body Velocity with Mathematical Model
1" Diameter Cylindrical Prechamber, 900 rpm
Fig. 4.24  Comparison of Equivalent Solid Body Velocity with Mathematical Model
1" Diameter Cylindrical Prechamber, 1100 rpm
Fig. 4.25 Comparison of Equivalent Solid Body Velocity with Mathematical Model.
2" Diameter Cylindrical Prechamber, 600 rpm
Fig. 4.26 Comparison of Swirl Component of Velocity with Mathematical Model
2" Diameter Cylindrical Prechamber (Position B, Location 1 data only), 1100 rpm.
Fig. 4.27 Comparison of Hassan's Data (14) with Present Mathematical Model
Fig. 4.28 Comparison of Horvatin's Data (17) with Present Mathematical Model
Fig. 4.29 Comparison of Nakajima's Data (21) with Present Mathematical Model
Fig. 4.30 Correlation of Prechamber Swirl Velocity with Engine Speed

\[ \text{Re}_p = -1881 + 0.556 \text{Re}_m \]

- 1" Diameter Prechamber
- 2" Diameter Prechamber
- Hassan (14)
- Horvatin (17)
- Nakajima (21)
Comparison of 'Corrected' Theoretical Model with Experimental Results in the Period 45 degrees BTDC to TDC
Figs. 4.35 to 4.37 Comparison of 'corrected' theoretical model with experimental results in the period 45 degrees BTDC to TDC.
Transition to Turbulence

Fig. 4.38 Model of Film Cooling Arrangement (50)

Prechamber Thickness
Exactly Equal to Throat Diameter

Throat
Prechamber Thickness

Prechamber Thickness
Bigger than Throat Diameter

Fig. 4.40 Effect of Prechamber Thickness not Accounted for in the Mathematical Model

Potential Core Length $X_p/s$

- $s = 0.125$ in
- $s = 0.160$ in

Fig. 4.39 Lengths of Potential Core Region (50)
CHAPTER 5

CONCLUSIONS TO THE PRESENT INVESTIGATION AND SUGGESTIONS FOR FURTHER WORK
5.1 Conclusions to the Present Investigation

The present investigation into the air motion in the prechamber of indirect injection diesel engines has been in two main parts:

i) the development of a mathematical model, and

ii) the measurement of mean velocity in two different cylindrical prechambers, under motoring conditions.

For the mathematical model it is recommended that a direct 'mass transfer' analysis be used in preference to an energy solution for the air velocity in the prechamber throat. The mass transfer and the energy solutions give approximately the same (maximum discrepancy = 8%) values of throat velocity but the mass transfer solution is simpler to use.

The principle of conservation of angular momentum is the most realistic method to apply to the air motion in a prechamber which has the ideal shape of a cylinder especially if the prechamber throat is tangential to the chamber surface.

The mathematical model computations show that the mean swirl velocity increases linearly with engine speed. Neglecting friction at the wall of the prechamber leads to an increase of 5-10% in the prechamber velocity. The peak prechamber velocity occurs at about 15 degrees BTDC and the reduction in velocity between this point and TDC is about 8%.

A three-wire probe with the wires set mutually perpendicular was manufactured and used to measure the mean velocity vector profiles in the two cylindrical prechambers. It is important to have a consistent and accurate calibration of the three-wire probe.
A major limitation of the three-wire probe used in the present study is the fact that the spatial resolution is poor since the three-wire arrangement occupies a cube of side 3 mm.

An error analysis carried out on data obtained from wind tunnel tests shows that the expected error in the measurement of the velocity magnitude by the three-wire probe is between -5% to 15%. The expected error in the horizontal angle was ± 10 degrees and in the vertical angle 5 to 25 degrees (Section 3.6.3).

Absolute velocity vectors were measured using the three-wire probe in one inch and two inch diameter cylindrical prechambers. The engine speed range investigated was 600-1100 rpm the upper limit being restricted by the available motoring arrangement. The highest compression ratio in the one inch prechamber was 12:1. The measured absolute vectors show that there is some departure from the solid swirl assumed in the mathematical model, this being true for the two prechambers. The departure from a full solid swirl as shown by the horizontal and vertical angles (Figs. 4.7 to 4.13) led to the decision to obtain the solid swirl components.

Comparison of the swirl velocities at the three measuring positions A, B and C (Figs. 4.14 to 4.16) show that for the one inch diameter prechamber the velocity at position A is in general about 30% greater than at positions B and C. This is thought to be due to the throat jet velocity being detected at position A. This effect was not present in the two inch diameter prechamber. It appears that at the low engine speed of 600 rpm swirl in the two inch prechamber was suppressed.

An equivalent solid body rotational velocity, ESBRV, was calculated from the velocity record at each engine speed, in a way similar to that of Tindal et al (20). The comparison of the ESBRV
with the mathematical model computations show poor agreement, there being discrepancies of more than 100% in some cases (Figs. 4.22 to 4.29). For this reason and also because it was not possible to determine theoretically values of discharge coefficient for the throat, it was decided to investigate a correlation for the experimental prechamber air velocity.

The correlation equation was found to be a straight line of the form:

\[ \frac{\text{Re}_p}{\text{Re}_m} = a + b \text{Re}_m \]

This correlation is described in Section 4.8.

Following from the above correlation it was decided to propose a relationship, in a general form, for a 'correction' factor to match the mathematical model computations to the corresponding experimental data in the period 45 degrees BTDC to TDC. This period is where solid swirl is best realised and it is also of major interest in a practical engine. The relationship arrived at for the 'correction' factor, \( FM \), is:

\[ FM = \left( \frac{V_p}{V_h} \right)^{-0.721} \times (\text{Re}_n)^{-0.147} \]

where \( V_p \) = prechamber volume
\( V_h \) = cylinder swept volume
\( \text{Re}_n \) = Reynolds number (Equation 4.5)

Application of the 'correction' factor in the mathematical model for each of five cylindrical prechambers and comparison of the 'corrected' velocities with the experimental velocities showed fairly good agreement only in some and not all the prechambers investigated.
5.2 The Practical Value of the Present Study

The literature survey (Chapter 1) had shown that a thorough knowledge of the air motion in the prechamber of an indirect injection diesel engine would assist the complete understanding of the combustion process and could also help in the control of the level of pollutants in the diesel engine exhaust gases. In particular, the knowledge of the velocity pattern in the prechamber would be useful in the modelling of fuel evaporation and mixing in the prechamber. Another area in which knowledge of the air motion in the prechamber may be useful is in the prediction of heat transfer across the prechamber wall.

The long term objective of a study such as the present one is to enable the mathematical model to be used in initial design considerations. It would then be possible to evaluate the effect of varying some engine parameters without undertaking extensive experimental work. The present study has shown that there are yet a number of points to be resolved before the mathematical model can be used effectively. There is insufficient data for the discharge coefficient of the prechamber throat. Little is known about the mechanism of mixing of the throat jet with the air in the prechamber. It may be necessary to revise the assumption of a uniform solid swirl in the prechamber in formulating the mathematical model.

The present study is the first, as far as is known, that has made a direct comparison of a mathematical model and experimental measurement of air motion in the prechamber of a test engine. It is hoped that, at least, the areas that may need further investigation have been identified. There is room for considerable improvement especially with regard to the correlation equation 4.3 and the matching factor, FM, discussed in Section 4.9.
5.3 Suggestions for Further Work

1) It is desirable to resolve the question of the discharge coefficient of the prechamber throat. This may be approached by investigating the flow in the connecting passage linking two spaces. A flow rig could be built to have a reciprocating piston motion in a cylinder such that gas in the cylinder is transferred into a small volume (prechamber) during the compression stroke. Velocity measurements across the jet from the throat could be made and compared with the theoretical solution for the throat velocity. The effects of the variation of the following parameters should be investigated:
   (a) ratio of prechamber volume to cylinder swept volume,
   (b) ratio of throat cross-sectional area to piston cross-section,
   (c) ratio of throat diameter to prechamber thickness for a cylindrical prechamber or to some suitable representative length for other shapes of prechamber. It is important to determine the effect of the sudden expansion at the intersection of throat and prechamber,
   (d) engine speed.

2) The use of the hot-wire for making velocity measurement in the prechamber has been justified. However, it should be pointed out that new laser techniques are available. For example the Laser Photon Correlator (25) and the Pulsed Laser Technique (49) have the potential of enabling more reliable measurements of the velocity vector in the prechamber. The main advantages are the ability to make point measurements and the fact that no disturbance is caused to the flow as is the case when using a hot-wire probe.
iii) It is suggested that further investigation of the correlation of prechamber velocity with engine speed and main cylinder dimensions (as described in Section 4.8) should be undertaken. In this connection investigation of the correction factor may also be included. The attraction of the correction factor is that in addition to accounting for a throat discharge coefficient it will include the effect of the mixing of the throat jet in the prechamber.

iv) Finally the modelling of the air motion in the prechambers of commercially operating diesel engines should be attempted. It is possible that difficulties will be encountered in formulating the mathematical model if the prechamber has an irregular shape.

It is not likely that the above suggestions can be taken up in one investigation. The work involved is extensive.
THRVEL

Program to compute Mean Gas Velocity in Cylindrical Prechamber.

Master THRVEL

JOB Control Cards

Read Initial_consts, Open Graph Plotter

Input Engine Dimensions

Start Calculation Procedure at IVC I = 1

Calculate
- Piston Travel from BDC
- Instant Total Volume
- Rate of Change of Cylinder Volume

Calculate Inst. Gas Properties
Select Wall Heat Transfer Coeffs.
Obtain Heat Transfer Quantities in Interval Δt

'Direct Mass Transfer' Solution
(Section 2.1.1)

Start Energy Solution for the Throat Velocity. (Equations 2.11 to 2.20)

Assume Initial ΔTm

Obtain Δm, Tp, Pp, Pm, ρθ, & PRA

Check for Flow Reversal
Check for Choked Flow
Calculate Throat Velocity from Compressible Flow Equation

\[ U_{th} = \sqrt{\frac{2\gamma R T}{\gamma - 1}} \left( P_{in} - PR \right)^{\frac{\gamma - 1}{\gamma}} \]

\[ \Delta m = U_{th} \Delta \rho \Delta t \]

Obtain New \( T_m \)

Wegstein's Iteration Statement

\[ \Delta T_m = T_m^{\text{Initial}} - T_m^{\text{New}} \]

Yes

No

\[ T_m^{\text{Initial}} = T_m^{\text{New}} \]

Use \( U_{th} \) & \( \Delta m \) in Conservation of Momentum in Prechamber

- Include Wall Friction Effects

Calculate \( \Delta \omega_p \) in Interval \( \Delta t \)

Update all Instant Values

End of Expansion Period?

Yes

No

\[ t = t + \Delta t \]

\[ I = I + 1 \]

Output Instant Values

Call Graph Plotter

END
MASTER THROAT
C
AIRMOTION NEW
C
IF NUEF = 2 DO NOT INCLUDE EXPERIMENTAL DATA CARDS
C
FOR NO FRICTION E=0.0
C
CALCULATION OF SWIRL RATIO ACCORDING TO ALCOCK ANALYSIS
D1
M1
D1
M1
D1
M1
C
SET INITIAL VALUES
C
S = PISTON STROKE, T ANGULAR CRANK MOVEMENT FROM BDC, CL CONROD
C
LENGTH, D = PISTON DIA, D1 THROAT DIA AT START OF THROAT, D2 THROAT
C
DIA AT ENTRY INTO PRECHAMBER, D3 PRECHAMBER DIA, VCL CLEARANCE
C
VOLUME, VP = PRECHAMBER VOLUME, RH0 = AIR DENSITY AT END OF INDUCTION,
C
VD TOTAL VOLUME AT END OF INDUCTION, CR = COMPRESSION RATIO,
C
XL THROAT LENGTH, DX STEP LENGTH ALONG THROAT, DA1 INCREASE IN
C
THROAT AREA BETWEEN SECTIONS DX APART, DA2 SURFACE AREA OF
C
THROAT LENGTH DX, RE = REYNOLDS NUMBER, F = FRICTION FACTOR FOR
C
FLAT PLATE, CP = SPECIFIC HEAT OF AIR AT 20 DEG CENT AND 1 BAR,
C
P = POLYVOTOPIC INDEX, RH0 = DENSITY AT ANY CRANK ANGLE, RHOT
C
DENSITY ALONG THROAT, TI AVERAGE TEMP. TO TEMP ALONG THROAT,
C
R = GAS CONSTANT PER UNIT MASS OF AIR, VIS = VISCOSITY OF AIR AT
C
STANDARD ATMOS
C
NP1 = NO OF CALCULATED POINTS, NPE = NO OF EXPERIMENTAL POINTS
C
NADJ = NO OF POINTS TO MATCH FORMAT FOR READING NPE
C
M1 = 1,3000) (VC(1), 1 = 1.9), (CPAC(1), 1 = 1.9)
C
3000 FORMAT ( SE20.12 )
C
READ (1,3010) TGO, TPO, PCO, PPO, CP00, CPPO
C
3010 FORMAT ( 6F8.0 )
C
READ(1,101) NGO, MLIN, NP, NPE, NADJ
C
WRITE(2,102) NGO, MLIN, NP, NPE, NADJ
C
101 FORMAT ( 5I0 )
C
102 FORMAT ( 1H, 5I10 )
C
OPEN GRAPH PLOTTER
C
IF( NGO .EQ. 1 ) CALL UTPOP
C
DO 100 K = 1, NPE
C
DEGF(K) = ( K-1 ) * 10
C
100 CONTINUE
C
PI = 3.1416
C FOR LIG THROAT
\[ d_1 = \text{SORT}(0.025) \times D \]
\[ CL = 0.350 \]
C ENGINE SPEED = N RPM
\[ n = 500 \]
\[ d_3 = 0.054 \]
\[ VCOMP = 0.000100 \]
\[ VP = 0.74 \times VCOMP \]
\[ WOK = 320 \]
\[ CR1 = 15.5 \]
\[ CR2 = 15.5 \]
\[ PCL = (22.0/10000.0) \times (254/10000.0) \]
\[ VS = VCOMP - VP \]
C THROAT INCLINATION AND POSITION ON CYLINDER HEAD
\[ THRINC = PI / 4.0 \]
\[ RF = 1.0 \]
\[ RC = RF \times (D/2.0) \]
C PRECHAIRER IS CYLINDRICAL IN SHAPE
\[ TH = (4.0 \times VP) / (P1 \times d3 \times 0.0) \]
\[ XL = 0.04 \]
\[ hX = 0.001 \]
\[ G = 1.0 \]
\[ P = 0.287 \]
\[ CP = 1.0 \]
\[ CV = CP / 1.4 \]
\[ VISO = 1.02 / 100000. \]
\[ VISCON = 0.00256 \]
\[ TO1 = 2.3.0 \]
\[ TO = TO1 \]
\[ p00 = 100000. \]
\[ RH00 = 1.10 \]
\[ TC60 = 2.6 / 100000. \]
\[ PH = 1.53 \]
\[ ANEF = 1.0 \]
\[ JJ = 1 \]
C DRAW AXES FOR GRAPH
IF ( MLIK .NE. 1 ) GO TO 1

REAL ( 1 , 1050 ) DMIN , DMAX , HMIN , HMAX , DINS , HINS
WRITE ( 2 , 1057 ) DMIN , DMAX , HMIN , HMAX , DINS , HINS

1050 FORMAT ( A00 , 0 )
1052 FORMAT ( 1H , 6FR.2 )
CALL HTPA ( DMIN , DMAX , HMIN , HMAX , DINS , HINS , 19H CRANK ANGLE DEGREE ,
13 , 14H VELOCITY m/s , 2 )

C DRAW GRID ON GRAPH
CALL GRID ( DMIN , DMAX , HMIN , HMAX , DINS , HINS )

1 CONTINUE
NPT = NPF + NADJ
NUPE = 2
IF ( NUPE .NE. 1 ) GO TO 37
READ ( 1 , 1053 ) ( UPE ( K ) , K = 1 , NPT )
WRITE ( 2 , 1054 ) ( UPE ( K ) , K = 1 , NPT )

1053 FORMAT ( 12F0.0 )
1054 FORMAT ( 1H , 12F6.1 )

37 CONTINUE
J = 1
1H = 1
NH = 3
NH1 = 3
UCO = 0.1
UPO = 2.0

C ASUME VALUES OF GAS VELOCITIES IN CYLINDER AND IN SWIRL-CHAMBER
C AT START OF COMPRESSION
UC = UCO
30 UP = UP0
UPX = UP0
UPX = UPX + 2.0 / D3

C CALCULATE IFAN PISTON VELOCITY
UPH = 2.0 * S * N / 60.0
WRITE ( 2 , 1101 ) UPH

1101 FORMAT ( 1H , 22H IFAN PISTON VELOCITY = .FR.2 )

C AT INLET VALVE CLOSURE
T = 41.0 * P1 / 180.0
THETA = ( T * 180.0 / PI ) + 180.0
BOO = PI / 180.
VOLEFF = 1.0
RHO = 1.16 * VOLEFF
SET INITIAL PRESSURE SUCH THAT INITIAL TEMP OBTAINED FROM \( T = \frac{P}{\rho \cdot R} \)

\[
\begin{align*}
\text{PC}(1) &= \text{PC0} \\
\text{PP}(t) &= \text{PP0} \\
\text{TC}(1) &= \text{TC0} \\
\text{TP}(t) &= \text{TP0} \\
\text{CPC} &= \text{CPC0} \\
\text{CVC} &= \text{CVC0} \\
\text{CPP} &= \text{CPP0} \\
\text{CVP} &= \text{CVP0} \\
\text{TPC} &= \text{TPC0} \\
\text{THMAX} &= 400 \\
\text{DTN} &= 0.5 \\
\text{DTP} &= 0.5 \\
\text{VISCF} &= \text{VIS0} \\
\text{VIS1} &= \text{VIS0} \\
A1 &= P1 \cdot D1 \cdot D1 / 4.0 \\
M &= 36 \\
A &= P1 \cdot D2 \cdot D2 / 4.0 \\
P1 &= CR2_1 \\
C &= CR / (CR1 - 1) \\
\text{DA1} &= 0.0 \\
\text{VCOMP} &= (VCL + VP + VTH) \quad \text{THE TOTAL COMPRESSION VOLUME} \\
\text{VTH} &= \text{VOLUME OF THE THROAT SECTION} \\
\text{VCDP1} &= A \cdot S / (CR1 - 1) \\
\text{VCDP2} &= A \cdot S / (CR2 - 1) \\
\text{VCP} &= \text{VCOMP} \\
\text{VP1} &= P1 \cdot D1 \cdot D1 \cdot TH / 4.0 \\
\text{VP2} &= \text{VCOMP} - \text{VCOMP1} + \text{VP1} \\
\text{VP} &= \text{VP2} \\
\text{R} &= \text{SORT} (\text{C1}/S)^{+}2 - (\text{SIN}(T)^{+}\text{SIN}(T))/4.0 \\
\text{ZIVC} &= 5/2.0 \cdot (1.0 - \text{COS}(T)) + S \cdot B - CL \\
\text{VCD} &= A \cdot (S - 2 \cdot \text{ZIVC}) + \text{VCOMP} \\
\text{VIVC} &= A \cdot (S - \text{ZIVC}) + \text{VCOMP} \\
\text{TH} &= \text{RH00} \cdot \text{VIVC} \\
\text{SIVC} &= \text{RH00} \cdot \text{VIVC} \\
\text{SIVC} &= \text{RH00} \cdot (\text{VCD} - \text{VP})
\[ \text{SNP} = R \times 100 \times VP \]
\[ \text{SNP} = \text{SNP} \]
\[ \text{SNP} = 0.0 \]
\[ \text{SNP} = 0.0 \]
\[ \text{DPX} = \text{PHO} \]
\[ \text{TTX} = \text{TIN} \]
\[ \text{DAU} = \text{P1} \times \text{M1} \times \text{DX} \]
\[ \text{N} = \text{D1} \]

C VELocities IN MAIN CYLINDER AND SWIRL-CHAMBER AT IVC

\[ \text{UC} = \text{UC} \times 2.0 / \text{NP} \]
\[ \text{WP} = \text{UP} \times 2.0 / \text{DP} \]

C CALCULATE PISTON TRAVEL FROM BDC

\[ \text{WR} = (2.0)^{1/2} \]
\[ \text{Z} = S/2 \times (1.0 - \cos(T)) + S \times R = \text{CL} \]
\[ \text{Z1} = Z \]

C PISTON VELOCITY AT ANGLE THETA FROM BDC

\[ \text{C1} = \text{PI} / \text{CT} \]
\[ \text{F1} = \sin(T) / 2.0 - (\sin(T) \times \cos(T)) / (4.0 \times \text{B}) \]
\[ T = T + \text{M0} \]
\[ \text{R} = \text{SOR} \times (\text{CL} / \text{S} \times 2.0 - (\sin(T) \times \sin(T)) / 4.0 \times \text{B}) \]
\[ \text{Z1P1} = Z \]
\[ \text{F2} = (\cos(T) - 1.0) / 2.0 = R + \text{CL} / \text{S} + \text{C} \]
\[ \text{VOIP1} = A \times S \times \text{F2} \]
\[ \text{PHOP1} = \text{T1} \times \text{VOIP1} \]
\[ T = T - \text{M0} \]
\[ \text{NO} = C1 \times \text{N} \times S \times \text{F1} \]

C VOLUME ABOVE PISTON INCLUDING PRECHAMBER AT THETA FROM TDC

\[ \text{R} = \text{SOR} \times (\text{CL} / \text{S} \times 2.0 - (\sin(T) \times \sin(T)) / 4.0 \times \text{B}) \]
\[ \text{F2} = (\cos(T) - 1.0) / 2.0 = B + \text{CL} / \text{S} + \text{C} \]
\[ \text{VO} = A \times S \times \text{F2} \]

C RATE OF CHANGE OF PISTON VOLUME DVO/DNO

\[ \text{DVO} = A \times S \times \text{F1} \]
\[ \text{DV} = A \times (\text{Z1} - \text{Z1P1}) \]

C START LEGENDS OF ITERATION PROCEDURE

\[ \text{NH} = 2 \]
\[ \text{TCA} = \text{TC}(1) \times \text{DT} \]
\[ \text{TCR} = \text{TCA} \]
\[ \text{TC}(1+1) = \text{TC}(1) + \text{DT} \]
CALCULATE INSTANT VALUES OF GAS SPECIFIC HEATS AND VISCOSITY

TC(I) = TC(I) - 273.0
TP(I) = TP(I) - 273.0
C1 = CNTC / 1000.0
C2 = CNTP / 1000.0
VIS = V(TV, 0.00001717, 0.0, TC(I), 0.8395)
VISP = V(TV, 0.00001717, 0.0, TP(I), 0.8395)
CPC = CNT(CPAC, TC(I), 9)
CPP = CNT(CPAC, TP(I), 9)
TC(I) = TC(I) + 273.0
TP(I) = TP(I) + 273.0
CVC = CPC/1.40
CVP = CPP/1.40
C NH=1 FOR WOSCHNI'S H NH=2 FOR ANNAND'S NH=3 FOR HP=FLAT PLATE
RN1 = RHOP * UP1 * D / VIS
IF ( NH .NE. 3 ) GO TO 55
C DEFINE HC EITHER = ANNAND OR WOSCHNI AND HP = FLAT PLATE
XNU1 = CNTP / ( PI * D3 )
XNU2 = CNTP / ( PI * D3/2.0 )
PRNO = ( VISP * ( CPP/1000.0 ) / CNTP ) ** (1.0/3.0)
RENP1 = ( RHOP * UP * PI * D3 / VISP ) ** 0.8
RENP2 = ( RHOP * (UP/2.0) * PI * (D3/2.0) / VISP ) ** 0.8
C HC IS AS FOR WOSCHNI
HC = (110.0 * PC(I)/D0870)**0.8*(2.28*UPH)**0.8*(4.187)/(D**0.2)
(*T(C(I)**0.5))
C HP IS AS FOR TURBULENT FLOW ON FLAT PLATE
HP1 = 0.047 * XNU1 * PRNO * RENP1 * 3600
HP2 = 0.037 * XNU2 * PRNO * RENP2 * 3600
HP3 = HP2
GO TO 54
55 CONTINUE
IF ( NH .NE. 2 ) GO TO 53
C DEFINE HEAT TRANSFER COEFF. AS FOR ANNAND
REM = RHOH * UPH * D / VISC
HCA = 0.25 * (CMT/C/6.24) * REM**0.7 / (D/0.3048)
HPA = 0.80 * (CMT/C/6.24) * REM**0.7 / (D/0.3048)
HC = HCA * 20.5 * 3600
HP = HPA * 20.5 * 3600
HP1 = HP
HP2 = HP
HP3 = HP
GO TO 54
53 CONTINUE
IF (NH .LE. 1) GO TO 54
C DEFINE HEAT TRANSFER COEFFICIENTS AS OF WOSCHNI
HC = (110.0 * PPI/C/6.24) ** 0.8 * (2.22 * UPH) ** 0.8 * 4.187 / (D**0.2)
* TPI(1) ** 0.53
HP1 = 1.0 * UPH
HP = (110.0 * PPI/C/6.24) ** 0.8 * (2.22 * UPH1) ** 0.8 * 4.187 / (D**0.2)
* TPI(1) ** 0.53
HP1 = HP
HP2 = HP
HP3 = HP
54 CONTINUE
XAC = PI**D*(S-Z) + PI*D*D/4.0 + PI*D*D/4.0
XAP1 = PI**H3 + TH
XAP2 = PI**H3 * D3 / 4.0
XAP3 = XAP2
XAP = PI**D3**TH + PI*D3*D3/4.0 + PI*D3*D3/4.0
DTIME = D00 * 30.0 / (PI*N*3600.0)
C OBTAIN HEAT TRANSFER QUANTITIES IN 'KJ':
TC1 = (TC1 + TCR) / 2.0
TP1 = (TPI + TPI+1) / 2.0
DUC = HC * XAC * (TC1 - TWC) * DTIME
DUP = (HP1*XAP1) + (HP2*XAP2) + (HP3*XAP3) * (TPI - TWP) * DTIME
IF (NH .LE. 4) DGC = 0.0
IF (NH .LE. 5) DGP = 0.0
IF (NH .LE. 5) DOP = 0.0
C OBTAIN SHAFT WORK OF PISTON MOTION
DWC = PCC + DV / 1000.0
C INCLUDE HEAT TRANSFER QUANTITIES IN THE LAST 90 OR 60 DEGREES BTDC
IF (I .LT. 0) GO TO 52
52 CONTINUE
C OBTAIN MASS TRANSFER INTO PRECHAMBER IN INTERVAL DTIME
IF ( I .GT. 140 ) GO TO 62
nPh = ( DM*CPP - (CMC*CVC/1000.0*DTM)) / (TC(I) * (CPP-CVC)/1000.0)
GO TO 63
62 CONTINUE
nPh = ( DM*CPP - (CMC*CVC/1000.0*DTM)) / (TC(I) * (CPP-CVC)/1000.0)
63 CONTINUE
nTA = DTM
TA(I+1) = TC(I) + nTA
VC1 = ( A*(S-ZI) + VCOMP - VP )
VCIP1 = ( A*(S-ZI) + VCOMP - VP )
VTI = VC1 + VP
VTIP1 = VCIP1 + VP
RHOM = TM / VTIP1
nMX = ( VP/VTI ) * RHOM * dV
nMX1 = ARS(DMX)
RHOM1 = RHOM * dV
nMX1 = ARS(DMX1) * ( VP/VTI )
nMX2 = 2.0 * DMX
nMX = DMX
TCXI = TCO * ( VOO/VTI )**PN
TCXIP1 = TCO * ( VOO/VTIP1 )**PN
nTX = ( TCXI - TCXI )
PCT1 = PClO * ( VOO/VTI )**PN
PCTIP1 = PClO * ( VOO/VTIP1 )**PN
nPX = ( PCT1 - PCT1 )
IF ( I .GT. 140 ) GO TO 64
nTP = ( nDPP * DM * (CPP-CVP)/1000.0 * TP(I) ) / ( SMP*CVP/1000.0 )
nPP = PP(I) - ( nT/P/TP(I) + nM/SMP )
GO TO 65
64 CONTINUE
nTP = ( nDPP * DM * (CPP-CVP)/1000.0 * TP(I) ) / ( SMP*CVP/1000.0 )
nPP = PP(I) - ( nT/P/TP(I) + nM/SMP )
65 CONTINUE
nTX = 2.0 * DTX
TP(I+1) = TP(I) + nTP
nPX = 2.0 * DPX
PP(I+1) = PP(I) + DPP
RHON = SMC / VCI
RHOP = SMP / VP
IF ( I .GT. 140 ) GO TO 66
SIPC1P1 = SMC = DM
SIP1P1 = SMP + DM
SIPX2 = SIPX1 + DMX
DPM = PC(I) * ( DMA/TC(I) = DM/SMC = DV/VCI )
GO TO 67
66 CONTINUE
SIPC1P1 = SMC + DM
SIP1P1 = SMP + DM
SIPX2 = SIPX1 + DMX
DPM = PC(I) * ( DMA/TC(I) = DM/SMC = DV/VCI )
67 CONTINUE
RHONP1 = SMCPI1 / VC1P1
RHOPP1 = SMPP1 / VP
PC(I+1) = PC(I) + DPM
IF ( I .GT. 140 ) GO TO 68
RHON = RHONP1 * ( PP(I+1)/PC(I+1) ) ** (1.0/PN)
GO TO 68
68 CONTINUE
RHON = RHONP1 * ( PC(I+1)/PP(I+1) ) ** (1.0/PN)
69 CONTINUE
IF ( I .GT. 140 ) GO TO 74
PRA = ( PP(I+1)+PP(I) ) / ( PC(I+1)+PC(I) )
GO TO 75
74 CONTINUE
PRA = ( PC(I+1) + PC(I) ) / ( PP(I+1) + PP(I) )
75 CONTINUE
C CHECK FOR FLOW REVERSAL
NY = 0
IF ( PRA .GT. 1.0 ) NY = 1
C CHECK FOR CHECKED FLOW IN THE THROAT
PRCH = ( 2.0/(PN+1.0) )**((PN/(PN-1.0))
IF ( PRA .LT. PRCH ) PRA = PRCH
C CHECK FOR REVERSED FLOW AND INVERT 'PRA' IF Y = 1.0
IF ( NY .EQ. 1 ) PRA = 1.0/PRA
IF ( PRA .LT. PRCH ) PRA = PRCH
THI = ( TC(I+1) + TC(I) ) / 2.0
C CALCULATE THROAT VELOCITY FROM COMPRESSIBLE FLOW EQUATION

U1 = 2.0 * R * 1000.0 * TTHIII * G * (1.0 - PT) / DIX

IF ( UY .EQ. 1 ) U1 = - U1

U2 = U1

U3 = ABS(U1)

DM = 1.06

D1 = U1 * A1 * RH01 * (DTIME*3600.0) * CD1

CD = DM / ( U1 * A1 * RH01 * DTIME * 3600 )

IF ( I .GT. 140 ) GO TO 72

DTM = TC(I) * ( DPM/PC(I) + DV/VC1 + DM/SMC )

GO TO 73

72 CONTINUE

DTM = TC(I) * ( DPM/PC(I) + DV/VC1 + DM/SMC )

73 CONTINUE

TC(I+1) = TC(I) + DTM

TX(I+1) = TC(I+1)

IF ( NH .NE. 2 ) GO TO 70

TCB = TX(2)

NB = NH+1

GO TO 60

70 CONTINUE

C WEGSTEINS ITERATION STATEMENT

TCF = TX(3)

TCF = TX(2)

TCG = TCR

TCH = TCA

NITRATE = NH

TCD = TX(3) - ((TX(3) - TX(2)) *(TX(3) - TCR)) / (TX(3) - TX(2) - TCB + TCA)

C CHECK WEGSTEIN ITERATION AND ALSO WHETHER TC(I+1) = TA(I+1)

IF ( ABS(TCD - TCB) / (TX(3) - TX(2)) .LE. 0.0001 .AND. ABS(TCD - (TC(I) + DTA)) .LE

1.0 .01 ) GO TO 5

NB = NH + 1

TCA = TCR

TCB = TCD

TX(?) = TX(3)

GO TO 60

80 CONTINUE
81 CONTINUE
C CHECK WHETHER TC(I+1) = TA(I+1)
   IF ( ABS(TC(I+1) - (TC(I) * DTA)) .LE. 0.01 ) GO TO 95
   IF ( NE .EQ. 1 ) TCB = TCB + 0.01
   IF ( NE .EQ. 2 ) TCB = TCB - 0.01
   IF ( NE .EQ. 3 ) TCB = TCB + 0.01
   GO TO 60
95 CONTINUE
RHOX = TM / VTIP1
SMX1 = SMX1 + DI
SMXX = SMXX + DNX
PPX = PPX + DPX
TTX = TTX + DTX
DX = DNX / ( RHOX * A1 * DTIME * 3600.0 )
C CALCULATE SWIRL VELOCITY
C RADIUS OF ENTRY INTO PRECHAMBER
C OF FLUID BALL IN PRECHAMBER
22
   P4 = 0.454 * D3
   P5 = SORT( 0.1 ) * D3
   RP = D3/2.
   DNP = DNP
   TM1 = ( SHC + SHCIP1 ) / 2.0
   SHC = SHCIP1
C MASS OF FLUID IN PRECHAMBER AT ANY CRANK ANGLE = TMP
   TMP = ( SHP + SHCIP1 ) / 2.0
   TMPX = SMPX2
   SMX1 = SMX2
   SHP = SHCIP1
C CALCULATE INSTANT VALUES OF THROAT DISCHARGE COEFFICIENT
RHOAV = ( RHOM + RHIOM ) / 2.0
CDTH(1) = CD
WRITE(2,2500) HC, HP1, DQC, DPQ, PWC, SHC, SHP, CPC
2500 FORMAT(W, REAL 7)
WRITE(2,2300) DM, DNP, DNX, SMX1, SMXX, TTX, SMPX2, RH01, RH0X
3001 FORMAT(W, E13.6 )
THETA = THETA
WRITE(2,4000) ITHETA, ZI, TC(I), TP(I), P(1), P(1), PPX, PRA, UI, U1, UX
4000 FORMAT(W, 14, E12.5, 2F10.2, 4E12.5, 2F6.1 )
C INSTANTANEOUS COMPRESSION RATIO = CRI
CRI = V00 / V0
15 SUN = 0.
20 SUN? = 0.1
25 SUN? = 0.
30 J = 1
35 C CALCULATE SWIRL VELOCITY AND SWIRL RATIO AT START OF COMPRESSION
40 SUM1 = SUM1/SUM2
45 VI = (C1 * VP * N * R4 * SUN1) / (A1 * R5 * R5)
50 SRI = VI / (2. * PI) / 60. * N
55 C ASSUME VALUE OF ANGULAR VELOCITY AT START OF COMPRESSION
60 WP = VI
65 SK = SRI
70 UP = WP * RP
75 WPN = ( WP * 60.) / (2.0 * PI)
80 IF (J.EQ.0.1) GO TO 23
90 CONTINUE
95 IF (UP.LE.0.) GO TO 26
100 GO TO 27
105 IF (UP.LE.0.) GO TO 26
110 UP = 0.1
115 RP = RP * RP * PI * D3 / VISP
120 RPX = RP * RPX * PI * D3 / VISP
125 FO = 0.0740 / RFPX**0.2
130 IF (FO.LE.0.0) GO TO 34
135 IF (FO.LE.0.0) GO TO 34
140 GO TO 35
145 UC = 0.1
150 UC = UC * 2.0 / D
155 CONTINUE
160 REC = RHOP * UC * D * PI * D / (2.0 * VISC)
165 FOC = 0.0740 / REC**0.2
170 FF = FOC
175 C CALCULATE ANGULAR VELOCITY INCREMENT FOR EACH CRANK POSITION
180 IF (UC.LE.0.1) UC = 2.0
185 IF (UP.LE.0.1) UP = 2.0
190 VIS = VISC
195 PI.C = (0.074) * (8.0**1.2) * PI * (D * 2.6) * (RHOP**0.8) * (VISC**0.2) * 
200 LC = 0.015 * (8.0**1.2) * PI * (D * 2.6) * (RHOP**0.8) * (VISC**0.2) * 
210 UC = (PI**0.2) * (256.0 * C1 * N) / (256.0 * C1 * N) * 
215 RH2C = (2.0 * PI * FF * RHOP * UC * UC * 2 * D00) / (TM1*C1*N)
C IF NO FRICTION IN PRECHAMBER RR=0.0
C FOR A CYLINDRICAL PRECHAMBER
C R6 IS THE RADIUS OF CURVATURE OF THE FLOW PATH OF FLUID
C FOR A CYLINDRICAL PRECHAMBER
R6 = R5
C SOLID ROTATION STARTS AT 90DEG BEFORE TDC
VIS = VISP
M1 = (0.074*(8.0**1.2)*PI*(n3**2.6)*((RHOP**0.8)*(VIS**0.2)*
     (1**0.8)*(n100)) / ( (PI**0.2)*256.0*n1*n1MP )
M2 = (0.074*(8.0**1.2)*PI*(n3**2.6)*((RH0X**0.8)*(VIS**0.2)*
     (1**0.8)*(n100)) / ( (PI**0.2)*256.0*n1*n1MPx )
FX = FOX
FF = FD
RE2 = ( 2.0*PI*FX*RHOP*UP*UP*TH*n100 ) / ( TMP*C1*N )
REXP = ( 2.0*PI*FX*RH0X*UPX*UPX*TH*n100 ) / ( TMPX*C1*N )
RE = RI1 + RB2
RR = ( UC * n1 / TH1 )
RD = DI * UC * RC * RC * 8.0 / ( TH1 * n * n )
RC = UP * DMP / TMP
RCX = UPX * DNX / TMPX
RA = ( 2.0 * U3 * RA * DNX ) / ( TMPX * RP * RP )
RA = RA * A11FF
RD = DNP*RC*RC*S1N(THRING)*UC * 8.0 / ( TMP*D3*D3 )
ND = RA * A11FF
RE = DMP * UP * 2.0 / TIP
RF = 0.0
WF = DMP*RC*RC*COS(THRING)*UC*D3*8.0 / (2.0*RC*TMP*D3*D3)
BF = WF * A11FF
RF = 0.0
C INDUCTION PFFIND
IF ( I.GT. 0 ) GO TO 160
NUC = 16.0 / 90.0
MUC = DUC * 2.0 / D
MUP = - RD
GO TO 300
160 CONTINUE
IF ( I.GT. 0 ) GO TO 200
N1 = DUC * 2.0 / D
NUP = RB
GO TO 300

200 CONTINUE
C COMPRESSION PERIOD
IF ( I .GT. 140 ) GO TO 210
DUC = BCC - BBC - BCD
DWP = GA - BR - RC
DUX = BAX - RBX - BGX
GO TO 300

210 CONTINUE
C EXPANSION PERIOD
IF ( I .GT. 220 ) GO TO 220
DUE = DH
RCA = DUE + H2 * d * 2.0 / ( TH1 + h * h )
DUC = RCA - BBC - BCC
DWP = -BR - RC
DUX = -BRX - BGX
GO TO 300

220 CONTINUE
C EXHAUST PERIOD
DUC = - BBC
DWP = - RC

300 CONTINUE
WU = UC + DUC
UC = UC * (N/2.0)
UCCL(1) = UC
DEG(1) = 1
WUX = UPX + DUX
WUX = WUX * RF
WP = UP + DUP
UP = UP * Rp
UPPX(I) = UPX
UPB(I) = UP
SH = UP / ( (2.0 * PI)/10.0, * N )
UPN = ( UP / 60.0 ) / ( 2.0 * PI )

23 WRITE(2,4001) UP, UPX
4001 FORMAT(1H+, 6F6.2, 2E12.5)
1001 FORMAT(1H+, F12.5, 5X,E12.5,5X,E12.5)
1002 FORMAT(1X,6H,F12.5,8X,F12.5,5X,E12.5)
1003 FORMAT(1H,7HCRANK,12HPISTON,7H15HAIR VELOCITY AT,2X,15HAIR
  1 VELOCITY AT,2X,20HAIR STREAM VELOCITY,2X,16HAIR ANGULAR VELOCITY,4X,
  25HS:1PL/4H,4X5SH:4X4TRAVEL,7X4HSTART OF THROAT,3X,13HEN
  5D OF THROAT,4X,4H IN PRECHAMBER,6X,4H IN PRECHAMBER,6X/5HRATIO//
  14X,4H5HTHETA=11X,4HZ M, 6X,9HU1 M/SEC,8X,9HU2 M/SEC,8X,9HU
  8D / SEC,9X11HWPN REV/MIN,11X,2HSW/)  
1004 FORMAT(//)  
1005 FORMAT(1H,3HD3=.F6.2,5X,5HSUM1=.F8.2,5X,5HSUM2=.F8.2,5X,5HSUM3=,lE12.5)
1006 FORMAT(1H+,51X,12.5)  
1007 FORMAT(1H+,3H1I=.E14.7,10X,4H5HI=.E3.)  
1008 FORMAT(1H,3HDI=.F6.2,5X,3HD3=.F6.2,5X,2HF=.E10.3,5X,3HFF=.E10.3)
     IF (1.EQ.320) GO TO 21  
        T = T + 1  
     THENA = ( T * 180.0 / PI ) + 180.0  
     GO TO 14  
21 CONTINUE  
C GRAPH OUTPUT  
1F(MIN,HF=1) GO TO 1060  
C ARRANGE TO PLOT BETWEEN CRANK ANGLE 180 - 540 ONLY
     J0 = J0 + 1, JK  
     CAAL(I) = DEC(I) * 220  
     UPA(I) = UPP(I)  
     UPP(I) = UPPX(I)  
400 CONTINUE  
     CALL UTPDE(CCAA,UPA,JK,7)  
     CALL UTPDE(CCAA,UPB,JK,2)  
1060 CONTINUE  
     IF (IH.EQ.1) GO TO 41  
     IF (IH.EQ.3) GO TO 41  
     IF (IH.EQ.1) PN = 1.20  
     IF (IH.EQ.2) PN = 1.40  
     IH = IH + 1  
     GO TO 30  
41 CONTINUE  
     IH = 1  
     NH1 = 3
IF ( J .EQ. 1 ) N = 1000
J = J + 1
GO TO 30
31 CONTINUE
IF ( MLIN .NE. 1 ) GO TO 1070
IF ( NUPE .NE. 1 ) GO TO 1070
CALL UTPL ( DEFE, UPE, NPE, 2 )
1070 CONTINUE
C CLOSE GRAPH PLOTTER
IF ( NGO .EQ. 1 ) CALL UTPCL
STOP
END
FUNCTION VT(CF,VO,TO,TT,TC)
C CALCULATES THE DYNAMIC VISCOSITY OF A GAS AT TEMPERATURE TT.
C INPUT PARAMETERS:
C CF - POLYNOMIAL COEFFICIENTS FOR VISCOSITY FUNCTION VALUES.
C VO - VISCOSITY AT REFERENCE TEMPERATURE TO.
C TO - REFERENCE TEMPERATURE.
C TT - GAS TEMPERATURE.
C TC - TEMPERATURE OF GAS AT ITS CRITICAL POINT.
C FOR REF. SEE KEND AND SHERWOOD, PROPERTIES OF GASES AND LIQUIDS.
C CHAPTER 6, PUBLISHERS - McGRAW-HILL
DIMENSION CF(9),X(2),F(2)
TH1 = (TO+273)/(TC+273)
TH2 = (TT+273)/(TC+273)
X(1) = 1.33*TH1
X(2) = 1.33*TH2
DO 1 I=1,2
DO 1 J=1,9
F(1) = 0.0
DO 1 J=1,9
1 F(I) = F(I)+CF(J)*X(I)**(J-1)
CONTINUE
VT = VO*F(2)/F(1)
RETURN
END

FUNCTION CT(CNO,TO,P,TT)
C INPUT PARAMETERS:
C CNO - THERMAL CONDUCTIVITY AT REFERENCE TEMPERATURE.
C TO - REFERENCE TEMPERATURE.
C P - PRESSURE FOR PARTICULAR GAS UNDER CONSIDERATION.
C FINDS THE THERMAL CONDUCTIVITY OF A SINGLE GAS AT TEMPERATURE TT.
C FOR REF. SEE TSFEDERBERG, THERMAL CONDUCTIVITY OF GASES AND LIQUIDS, CHAPTER 3, PUBLISHERS ARNOLD.
C CT = CNO*(((TT+273)/(TO+273))**P)
RETURN
END
FUNCTION CPY(AC,T,N)
C
FINDS THE SPECIFIC HEAT OF A SINGLE GAS AT TEMP T
C
BY POLYNOMIAL FIT TO PUBLISHED DATA
C
AC- 1-D ARRAY OF LENGTH N CONTAINING THE COEFFICIENTS.
C
C
T-- GAS TEMPERATURE
C
N-- ORDER OF POLYNOMIAL + 1
C
DIMENSION AC(N)
C
CPY = 0.0
C
AC(0) = T
C
CPY = CPY + AC(0)*T*(T-1)
1 CONTINUE
C
RETURN
C
END

SUBROUTINE GRID(XMIN,XMAX,YMIN,YMAX,XINS,YINS)
C
DEALS WITH A GRID ON A SET OF AXES PROVIDED BY UTP4A IN THE MASTER.
C
DIMENSION X(2),Y(2)
C
NX = 1/FIX(XINS)
C
NY = 1/FIX(YINS)
X(1) = XMIN
Y(1) = YMIN
X(2) = XMAX
NX = (XMAX-XMIN)/XINS
DY = (YMAX-YMIN)/YINS
D0 1 =1,NX
X(1) = X(1)+DX
X(2) = X(1)
1 CALL UTP4A(X,Y,2,3)
C
X(1) = XMIN
Y(1) = YMIN
NX = (XMAX-XMIN)/XINS
DY = (YMAX-YMIN)/YINS
D0 2 =1,NY
Y(1) = Y(1)+DV
Y(2) = Y(1)
2 CALL UTP4A(X,Y,2,3)
RETURN
END
FINISH

****
DOCUMENT DATA
APPENDIX 3A

ERROR IN GAS TEMPERATURE MEASUREMENT USING A WIRE OF SMALL ASPECT RATIO

Consider a wire of length $2l$ and diameter $d$ welded to two pin supports.

The thermal inertia of the pin supports are large so for this reason the pins will not respond to the instantaneous variation of the gas temperature, $T_g$. The supports will attain some mean temperature less than the peak gas temperature. This will result in heat transfer from wire to support when the wire temperature is greater than the support temperature. The converse is also true. The instantaneous temperature distribution in the wire is as shown in Fig.

The mechanism of heat transfer from gas to wire may be treated as for an extended surface. The general solution to such a problem is a 2nd order differential equation which results in:

$$ (T_g - T_{w_0}) = Ae^{mx} + Be^{-mx} $$

..... 3A.1
where $T_g$ is gas temperature

$T_w_x$ is wire temperature

$x$ is distance along wire from the centre

$m = \left(\frac{4h}{kd_w}\right)^{\frac{1}{2}}$

d$_w$ is the wire diameter

$k$ is the wire conductivity

$h$ is heat transfer coefficient

$A$ and $B$ are constants

Equation 3A.1 can be solved to obtain the constants $A$ and $B$ by using the two boundary conditions:

1. At $x = 0$, $\frac{dT_w}{dx} = 0$ \hspace{1cm} ...... 3A.2

2. At $x = l$, $T_w = \bar{T}_s$ \hspace{1cm} ...... 3A.3

Equation 3A.2 assumes that the wire temperature distribution is symmetrical about the centre.

Substituting 3A.2 and 3A.3 in equation 3A.1 gives:

$$A = -B = \frac{(T_g - \bar{T}_s)}{2 \sinh m \frac{l}{2}}$$ \hspace{1cm} ...... 3A.4

Therefore equation 3A.1 can be written as:

$$(T_g - T_w_x) = (T_g - \bar{T}_s) \left( \frac{\sinh m x}{\sinh m \frac{l}{2}} \right)$$ \hspace{1cm} ...... 3A.5

The mean wire temperature, $\bar{T}_w$, is given by:

$$\bar{T}_w = \frac{1}{l} \int_0^l T_w_x \, dx$$ \hspace{1cm} ...... 3A.6

Equations 3A.5 and 3A.6 combine to give:
\[ \bar{T}_w = T_g - \frac{(T_g - \bar{T}_s)}{m_{\ell} \sinh m_{\ell} (\cosh m_{\ell} - 1)} \quad \text{(3A.7)} \]

It is reasonable to assume that the mean support temperature during the compression period is equal to the mean gas temperature.

If we assume a sinusoidal variation of \( T_g \) with crank angle such that

\[ (T_g - 273) = (T_{g\text{peak}} - 273) \sin \frac{\theta}{2} \quad \text{(3A.8)} \]

\[ \therefore \bar{T}_s = \frac{1}{\pi} \int_0^{\pi} (T_{g\text{peak}} - 273) \sin \frac{\theta}{2} d\theta + 273 \quad \text{(3A.9)} \]

\[ \therefore \bar{T}_s = 0.637 (T_{g\text{peak}} - 273) + 273 \quad \text{(3A.10)} \]

For a nominal compression ratio of 12:1 and using a polytropic index of 1.33 and an initial temperature of 300\(^\circ\)K, the peak gas temperature is 681\(^\circ\)K. This gives the mean support temperature \( \bar{T}_s \) of 533\(^\circ\)K.
It is now possible to solve for the mean wire temperature \( \bar{T}_w \) in equation 3A.7.

Substituting typical values at TDC:

\[
\begin{align*}
    h &= 1200 \text{ kJ/m}^2 \text{ hr} \, ^\circ\text{K} \\
    k &= 700 \text{ kJ/m} \text{ hr} \, ^\circ\text{K} \\
    d_w &= 10 \times 10^{-6} \text{ m} \\
    &\quad = \frac{1.5}{2} \times 10^{-3} \text{ m}
\end{align*}
\]

in equation 3A.7 gives

\[
\bar{T}_w = 610^\circ\text{K}
\]

This mean wire temperature is the measured gas temperature and represents an error of about -20% compared with the theoretical TDC gas temperature at a compression ratio of 12:1, for a 10 \( \mu \text{m} \) diameter wire of length 1.5 mm.
ENGDATA

Program to calculate absolute Velocity Magnitude and Vector Angles from 3-wire Probe Measured Data.

- JOB Control Cards
- Read Initial Constants
- Read Gas Pressures
  Calculate Gas Temperatures
- Read Velocity Bridge Voltages
  Read Wire Properties
- Fill Current Array
- Calculate Velocity for Each Wire of 3-Wire Probe,
  CALL Subroutine DF
- Calculate Absolute Velocity Magnitude and Vector Angles,
  CALL Subroutine VELVEC
- Output Absolute Velocity and Vector Angles
- END
MASTER ENG DATA
C (1) PROGRAM TO VERIFY 3-WIRE MEASURING TECHNIQUE FROM WIND-TUNNEL
C DATA AND/OR TO (2) PROCESS ENGINE 3-WIRE DATA OR 1-WIRE DATA
C
C AS REQUIRED
REAL K1
DIMENSION VUT(3,37), C(3,150), VP(3,150), PRP(150), BV(150),
C(150), TG(150), RHOG(150), HAREF(37)
EXTERNAL EK1
COMMON/PROP/VC(c), CPAC(i)/Fl.7/HDF,DA,CNUT, XA, XR, RG, ALPHAA, XTW, TOA
C
C NT=1 = WIND TUNNEL DATA, NT=2 = ENGINE DATA, NA=NO OF ANGLES =
C KORD, NV=NO OF VELOCITIES, NL=NO OF LOCATIONS, NW=NO OF WIRES
C N=LOCATION COUNTER, UH, UC, UD ARE REF. VELOCITIES
DIMENSION UH(140), UC(140), UD(140), UX(140), UY(140), UZ(140),
C(140), NA(140), VA(140)
DIMENSION CAA(100), GPA(100), UCA(100), HAA(100), VAA(100)
READ (1,3000) (VC(i), i=1,9)
READ (1,3010) NV, NGO
C
OPEN GRAPH PLOTTER
IF (NGO,FG,1) CALL UTPLOT
READ (1,3020) NT, NA, NV, NG, NL, KORD, NADJ
NPT = KORD + NADJ
C
NGP IS TO BE USED IN GRAPH PLOT
NGP = NA
IF (NG .NE. 1) GO TO 20
C BEFORE EACH RUN DECIDE PARTICULAR DATA TO BE PROCESSED AND INPUT
C APPROPRIATE AXES VALUES
READ (1,3070) CHIN, CMAX, VIN, VMAX, CINS, VINS
20 CONTINUE
C DECIDE WHETHER DATA IS FOR WIND-TUNNEL OR FOR ENGINE
N = 1
10 IF (NT .NE. 1) GO TO 11
IF (NV .EQ. NV) GO TO 250
READ (1,3025) UHFF
READ (1,3026) UH, UC, UD
READ (1,3030) (VUT(J,1), J=1,3), I=1,NA
DO 20 J = 1, UH
READ (1,3040) D, Z, ALPHA, BETO, P, T0, RL, RCM, RCI, RWI, W1
DO 30 I = 1, NA
HAREF(I) = (I-1) * 10.0
C CALCULATE GAS VELOCITY USING DAVIES AND FISHER EQUATION
CALL DF( TO, TWI.C(J, I) - RWI, NCI, KETO, 7, D, ALPHA, P, V, H, M )
VP(J, I) = V
80 CONTINUE
90 CONTINUE
91 CONTINUE
C ENGINE DATA SECTION
IF ( NT .NE. 2 ) GO TO 190
C INPUT PRESSURE AND VELOCITY CALIBRATION CONSTANTS
IF ( N .GT. 1 ) GO TO 145
C CHECK NUMBER OF PRESSURE CARDS
READ (1,3066) PII(I), I = 1, NPT
READ (1,3045) PCs, CPI, PCF, PAML, PAD, TMIN, PGMIN, PLI
C FILL CRANK ANGLE ARRAY
CAI = 720.0 / ( KORD - 1 )
DO 105 II = 1, KORD
C FOR 1st CYLINDER 600 RPM DATA
NI = 0
I = II - NI
CAI(JI) = ( II - 1 ) * CAI
PRP(JI) = ( PCs * ( PRP(I) - CPI ) ) * PCF + PAML + PAD
IF ( PRP(JI) .LE. 0.0 ) PRP(JI) = ABS(PRPI(JI)) + 0.01
T(JI) = ( TMIN * (PRP(I)/PGMIN)**((PLI-1.0)/PLI) = 273.0 )
105 CONTINUE
115 CONTINUE
NPT = KORD + NADJ
C LOCATION CHECK
IF ( ( N-1 ) .EQ. NL ) GO TO 250
121 J = 1, NI
READ (1,3066) VCS, CVI, RL, RWI
106 CONTINUE
READ (1,3066) ( RVP(I), I = 1, NPT )
C FILL CURRENT ARRAY
C CHECK WHETHER NI IS SAME FOR VELOCITIES AS FOR PRESSURES
DO 120 II = 1, KORD
I = II - NI
RVP(JI) = VCS * ( BVP(I) - CVI )
C(J, II) = RVP(JI) / ( 2.0 * ( RL + RWI ) )
120 CONTINUE
121 CONTINUE
C SET INSTANTANEOUS VALUES BEFORE ENTRY INTO SUBROUTINE DF
    NAME = 'DF'
    DO 130 J = 1, NUM
    READ (1,30) D,J,ALPHA,RETO,P,TO,RL,RCH,RCI,RWI,TWI
    RC = RCI
    DO 120 I = 1, KORD
    TO = T(I)
    PE = PRP(I) * 6894.0
    RETA = RETO * ( 1.0 + ALPHA * ( TO - TAMB ) )
    RC = RC * ( 1.0 + ALPHA * ( TO - TAMB ) )
    CI = C(I,J)
C CALCULATE GAS VELOCITY USING DAVIES AND FISHER EQUATION
C CALL DF(TO,TWI, CI, RII,RCI,RETA,Z,D,ALPHA,P,V,H,M)
    VP(I,J) = V
130 CONTINUE
131 CONTINUE
C NGP IS TO BE USED IN GRAPH PLOT
    NGP = KORD
140 CONTINUE
C CALCULATE ABSOLUTE VELOCITY AND HORIZONTAL AND VERTICAL ANGLES
C ABOUT REFERENCE AXES
C REFERENCE AXES ARE: U1=UX=UYELLOW, U2=UY=UWHITE, U3=UZ=UBLACK
C REMEMBER TO DESCRIBE EXACTLY THE PHYSICAL POSITION OF PROBE WIRES
C EITHER IN WIND-TUNNEL OR IN THE ENGINE
C FOR THE CYLINDRICAL PRECHAMBERS HA IS REF. TO THE HORIZONTAL PLANE
C AND VA TO THE VERTICAL PLANE IN THE PLAN VIEW OF THE PRECHAMBER
C CHECK NO OF WIRES
    DO 210 I = 1, NGP
    U1(I) = VP(I,1)
    U2(I) = VP(I,2)
    U3(I) = VP(I,3)
210 CONTINUE
    IF ( NW .EQ. 1 ) GO TO 220
    CALL VELVEC ( U1,U2,U3,NGP,UX,UY,UZ,VA,HU,HA,VA )
    GO TO 230
220 CONTINUE
    DO 290 I = 1, NGP
    HA(I) = VP(I,1)
230 CONTINUE
WRITE (2,4000) D,Z,RCM,RCI,RI,TUI
WRITE (2,4010) D,Z,RCM,RCI,RI,TUI
C SELECT APPROPRIATE WRITE STATEMENTS
C FOR TUNNEL DATA NT=1
IF (NT.EQ.1 .AND. NW .EQ. 3) WRITE (2,4020) UREF
IF (NT.EQ.1 .AND. NW .EQ. 1) WRITE (2,4021) UB, UC, UD
IF (NT.EQ.1 .AND. NW .EQ. 3) WRITE (2,4025) (UA(I),HAREF(I),HA(I),VA(I),
1 U1(I),U2(I),U3(I),UX(I),UY(I),UZ(I), I = 1, NGP)
IF (NT.EQ.1 .AND. NW .EQ. 1) WRITE (2,4027) (HAREF(I),U1(I),
1 U2(I),U5(I), I = 1, NGP)
C FOR ENGINE DATA NT=2 AND NUMBER OF OPERATING WIRE = NW
IF (NT.EQ.2 .AND. NW .EQ. 3) WRITE (2,4030) ( CA(I),PRP(I),
1 TG(I),UA(I),HA(I),VA(I),U1(I),U2(I),U3(I), I = 1, NGP)
IF (NT.EQ.2 .AND. NW .EQ. 1) WRITE (2,4040) ( CA(I),PRP(I),
1 TG(I),U1(I),U2(I),U5(I), I = 1, NGP)
C DECIDE WHETHER TO CALL GRAPH PLOTTER
IF (NG.IF.1) GO TO 240
C ALSO WHETHER TO OUTPUT UA(I), HA(I), VA(I) ON CARDS
JI = KORD/10 + 1
DO 30 J = 1, JI
11 = (J-1)*10
WRITE (2,4060) (UA(M1+I), I = 1,10), J
30 CONTINUE
DO 40 J = 1, JI
11 = (J-1)*10
WRITE (2,4060) (HA(M1+I), I = 1,10), J
40 CONTINUE
DO 50 J = 1, JI
11 = (J-1)*10
WRITE (2,4060) (VA(M1+I), I = 1,10), J
50 CONTINUE
DRAW AXES AND GRID FOR VELOCITY CURVES
CALL UTP4A (CHN,CHAX,VMIN,VMAX,CINS,VINS,'CRANK ANGLE':2,'VELO
1CITY',H/S,2)
CALL GRID (CHN,CHAX,VMIN,VMAX,CINS,VINS)
C ARRANGE TO PLOT BETWEEN CRANK ANGLE 180 = 540 ONLY
JK = KORD/2 + 1
JJ = KORD/4
DO 400 J = 1, JK
...
C AAA(I) = CA (I + JJ)
C UPA(I) = UA (I + JJ)
C MAA(I) = HA (I + JJ)
C WAA(I) = VA (I + JJ)

C FOR ORIENTATION (REFERENCE) DATA
C VERTICAL ANGLES SHOULD REMAIN CONSTANT SINCE ORIENTATION OF PROBE
C IS IN THE Y-Z PLANE
C 400 CONTINUE

C CALL GRAPH PLOTTER
CALL UTPAR (CAA, UPA, JK, ?)
CALL UTPLA (CMIN, CHAX, VMIN, VMAX, CIH, VINS, 'CRANK ANGLE', 2, 'H&R
1 ANGLES DEG', 2)
CALL UTPRF (CAA, HAA, JK, ?)
CALL UTPAR (CAA, VAA, JK, ?)

C 240 CONTINUE
N = N + 1
GO TO 10

C CLOSE GRAPH PLOTTER
C IF ( HGO .EQ 0 ) CALL UTPC1

C INPUT FORMATS
3000 FORMAT (3E20.12)
3010 FORMAT (2I3)
3020 FORMAT (7I0)
3025 FORMAT (F6.1)
3026 FORMAT (3F0.0)
3030 FORMAT (3F0.0)
3040 FORMAT (1F50.0)
3045 FORMAT (RF0.0)
3046 FORMAT (RF0.0)
3050 FORMAT (10F0.0)
3060 FORMAT (10F0.0)
3070 FORMAT (6F0.0)

C OUTPUT FORMATS
4000 FORMAT (2X, '3-WIRE DATA PROCESSING; TO GIVE ABSOLUTE VELOCITY
1 AND HORIZONTAL AND VERTICAL ANGLES: ///
4010 FORMAT (2X, 'WIRE DIA = ', E12.4, ' WIRE LENGTH = ', E12.4, 'MEASURED
1 COLD RESISTANCE = ',F7.2, 'ITERATED COLD RESISTANCE = ',F7.2,///
2 IN-PAID OPERATING RESISTANCE = ',F7.2, 'OPERATING TEMPERATURE =
3 ',F7.2, ///)
4021 FORMAT (1H = 3F20.1)
4025 FORMAT (1H = 10F10.2)
4027 FORMAT (1H = 4F20.1)
4030 FORMAT (1H = 9F10.1)
4040 FORMAT (1H, F10.1, F10.1, F10.1, 10X, F10.1, 10X, F10.1, 10X, F10.1)
4050 FORMAT (111)
4060 FORMAT (2X. 10F7.1, 18)
STOP
END
SUBROUTINE DF(T0, TW, CUR, RU, RC, ETO, Z, D, ALPHA, R, V, H, M)
C CALLS GAS VELOCITY FOR A HOT WIRE ANEMOMETER IN AIR USING
C DAVIES AND FISCHER EQUATION FOR V = F(H)
C
PARAMETERS
T0 = AMBIENT TEMPERATURE
TW = WIRE OPERATING TEMPERATURE.
CUR = WIRE CURRENT.
RU = WIRE OPERATING RESISTANCE.
RC = WIRE COLD RESISTANCE AT TEMPERATURE TO.
ETO = WIRE RESISTIVITY AT TEMPERATURE TO
Z = WIRE TOTAL LENGTH.
D = WIRE DIAMETER.
ALPHA = WIRE MATERIAL 1ST TEMPERATURE COEF. OF RESISTANCE.
P = AMBIENT PRESSURE.
V = CALCULATED VELOCITY.
H = HEAT TRANSFER COEFFICIENT.
M = NUMBER OF ITERATION LOOPS COMPLETED IN SUBROUTINE CJMIT.

REAL K1
COMMON/PROP/VC(0), CPAC(0)
PI = 3.141592654
VTO = VT(VC, 0.00001717, 0.0, T0, -139.5)
CNT0 = CNT(T0, 0.02435, 0.0 + 0.82, T0)
CPT0 = CPT(CPAC, T0, 0.9)
CVT0 = CINT0/1.403
RHO = P*2*A**3/(814.3*(T0+273.))
CINT1 = CINT(T0, 0.02435, 0.0 + 0.82, TW)
CALL HTRANS(H, K1, CUR, RU, RC, TW, TO, ETO, Z, D, ALPHA, H)
IF (HGT.09600696.0) GOTO 1
G1 = H*PI*CINT0/(2.6*VTO*CINT1)
G2 = D/VTO
FC = (TW+273.)/(T0+273.)**0.3
V = (G1**3.)*(G2*2.)*FC/RHO
RETURN
1 WRITF(2,2) K1
2 FOR I AT(20H ITERATION FAILURE IN DF )FRR = .E12.4)
RETURN
END
SOLVES GENERAL NON-LINEAR EQUATIONS OF THE FORM \( f_n(x, a, b, c, d) = 0 \)

PARAMETER `fn` CALLS AN EXTERNAL FUNCTION SUPPLIED BY THE USER

DESCRIPTION OF PARAMETERS

\( x \) - RESULTANT ROOT OF EQUATION \( f_n(x, a, b, c, d) = 0 \)

\( v_1 \) - \( v_4 \) VALUES OF CONSTANTS \( a, b, c, d \)

\( fn \) - NAME OF EXTERNAL FUNCTION USED

\( x_{li} \) - INITIAL LEFT BOUND OF THE ROOT \( x \)

\( x_{ri} \) - INITIAL RIGHT BOUND OF THE ROOT \( x \)

\( eps \) - UPPER BOUND OF THE ERROR OF RESULT \( x \)

\( iend \) - MAX NUMBER OF ITERATION STEPS SPECIFIED

\( ier \) - RESULTANT ERROR PARAMETER

\( ifr=1 \) NO CONVERGENCE AFTER \( iend \) ITERATIONS FOLLOWED

BY \( iend \) SUCCESSIVE STEPS OF BISECTION.

\( ifr=2 \) BASIC ASSUMPTION \( f_n(x_{li}) \times f_n(x_{ri}) \) LESS THAN ZERO

IS NOT SATISFIED.

\( ifr=0 \) NO ERROR

SOLUTION OF EQUATION \( f_n(x, a, b, c, d) = 0 \) IS ACHIEVED BY MEANS OF

MUELLER'S ITERATION METHOD OF SUCCESSIVE BISECTION AND INVERSE

PARABOLIC INTERPOLATION, WHICH STARTS AT THE INITIAL BOUNDS \( x_{li} \)

AND \( x_{ri} \). CONVERGENCE IS QUADRATIC IF THE DERIVATIVE OF \( f_n \) AT THE

ROOT \( x \) IS NOT EQUAL TO ZERO. ONE ITERATION REQUIRES TWO

EVALUATIONS OF \( f_n(x, a, b, c, d) \).

FOR REFERENCE SEE G. K. KRISTIANSEN, ZERO OF ARBITRARY FUNCTIONS, BIT,

VOL. 7 (1963), PP 205-206.

COMMON/E12/HDF,DAA,CNUT,xa,XRw,rag,ALPHaa,XTW,TOA,KR,DGA,DKA

PREPARE ITERATION

\( ier=0 \)

\( xl=x_{li} \)

\( xr=x_{ri} \)

\( xl=x_{li} \)

\( tol=x \)

\( f=fn( tol, v_1, v_2, v_3, v_4 ) \)

IF \( (f) \) \( 1,16,1 \)

\( ifle=1 \)

\( x=xl \)

\( tol=x \)

\( f=fn( tol, v_1, v_2, v_3, v_4 ) \)
IF(F) 2,16,2
FR=1

IF(SIGN(1.00,FL)*SIGN(1.00,FR)) 25,3,25
C BASIC ASSUMPTION FL<FR LESS THAN 0 SATISFIED.
C GENERATE TOLERANCE FOR FUNCTION VALUES.

3 I=0
TOLF=100.0*EPS
C START ITERATION LOOP

4 I=1

5 I=I+I
M=I
C START BISECTION LOOP
DO 13 K=1,IFND
X=0.5*(XL+XR)
TOL=X
F = EH (TOL, V1, V2, V3, V4)
IF(F) 5,16,5

IF(SIGN(1.00,F)*SIGN(1.00,FR)) 7,6,7
C INTERCHANGE XL AND XR IN ORDER TO GET SAME SIGN IN FL AND FR

6 TOL=XL
XL=XR
XR=XL
TOL=FL
FL=FR
FR=TOL

7 TOL=F+F
A=E*TOL
A=A*A
IF(A=FR*(FR-FL)) 8,0,9

8 IF(I=IFND) 17,17,9

9 XR=X
FR=FR

C TEST ON SATISFACTORY ACCURACY IN BISECTION LOOP
A = ABS(XR)
TOL = EPS
IF (A < 1.00 ) 11, 11, 10

10 TOL=TOL*A

11 IF(ABS(XR-X1)=TOL) 12,12,13

12 IF(ABS(FR-F1)=TOL) 14,14,13

CONTINUE

C END BISECTION LOOP
NO CONVERGENCE AFTER 1 END ITERATIONS AND BISECTIONS

SET IER=1 ERROR RETURN.
	IER= 1

14 IF(A(ARS(FR)-ARS(FL))) 16,16,15
15 X=XL
	F=FL
16 RETURN

COMPUTATION OF ITERATED X-VALUE BY INVERSE PARABOLIC INTERPOLATION

17 A=FR-F
18 D=\((X-XL)\times FL-1.00+(A-TOL)/(A-(FR-FL)))#/TOL
19 X=XL+D
20 F=FR
21 TOL=TOL

TEST ON SUFFICIENT ACCURACY IN ITERATION LOOP

18 TOL=TOL
	A=ABS(X)
19 IF(A=1.00) 20,20,19
20 TOL=TOL+1
21 IF(A(F)-TOL) 21,21,22
22 IF(A(F)-TOL) 16,16,22

PREPARATION OF NEXT BISECTION LOOP

22 IF(SIGN(1.00,F)=SIGN(1.00,FL)) 24,23,24
23 XREF

34 X=XL
25 XL=X
	FL=F
26 X=XM
27 F=FM
28 IF(F) 16,16,15

ERROR RETURN IN CASE OF WRONG INPUT DATA.

35 IER= 2
36 RETURN

END
SUBROUTINE UTRANS(H,K1,CUR,RW,RO,TW,TO,RETO,Z,D,ELP,M)
C
CALCULATES THE HEAT TRANSFER COEFFICIENT FOR A FINE WIRE IN A
CROSS FLOW OF GAS.  H--HEAT TRANSFER COEFF.,  K1--FUNCTION VARIABLE
C
CUR--WIRE CURRENT.
C
RW--WIRE OPERATING RESISTANCE.
C
RO--WIRE RESISTANCE AT TEMPERATURE TO
C
TW--WIRE OPERATING TEMPERATURE.
C
TO--REFERENCE AMBIENT TEMPERATURE.
C
RETO--RESISTIVITY OF WIRE MATERIAL AT TO.
C
Z--TOTAL WIRE LENGTH.
C
D--WIRE DIAMETER.
C
ELP--FIRST TEMPERATURE COEFFICIENT OF RESISTANCE FOR WIRE MATERIAL.
C
M--NO OF ITERATIONS EXECUTED IN CJM17.
C
ADDITIONAL SUBROUTINES REQUIRED-- CJM17,FK1.
C
REAL K1
DIMENSION FN(20)
EXTERNAL FK1
PI = 3.141592654
B= HT*D*n/4
CNWS= 2.25*(TO+273)/(10.**B)*BETO)
CNWT= 2.25*(TW+273)/(10.**BETO)*(1+ELP*(TW-TO))
G1 = CUR*CUR*RW/(A*CNWT*Z*(TW-TO))
G2 = CNWS*RO/(Z*CNWT*RU)
G3 = (RW*RO)/RW
G4 = 7/2
XL = 0.01
XR = 10.**(-7)
EPS = 10.**(-7)
IEND = 160
CALL CJM17(K1,G1,G2,G3,G4,FK1,XL,XR, EPS, IEND, IER, M)
IF(IER.LT.1) GOTO 1
H= 1000000000.0  GOTO 1
H = F
K1 = IER
RETURN
1 CONTINUE
K1 = ABS(K1)
H= (K1*(CNWT*Z*CUR*CUR*ELP*RO/Z))/(PI*pi)
RETURN
END
SUBROUTINE VELVEC(U1, U2, U3, UX, UY, UZ, HA, VA)

DIMENSION U1(NGP), U2(NGP), U3(NGP), UX(NGP), UY(NGP), UZ(NGP),
           UA(NGP), HA(NGP), VA(NGP)

C REFERENCE AXES ARE: U1=UX=UYELLOW; U2=UY=UWHITE; U3=UZ=UBLACK

C FOR THE WIND-TUNNEL: AXIS Z IS HORIZONTAL AND ALONG TUNNEL AXIS;
C AXIS Y IS VERTICAL; AXIS X IS HORIZONTAL AND PERPENDICULAR TO Z
C FOR THE ENGINE: MAKE SURE OF THE PHYSICAL PLACING OF THE WIRES RELA-
C TIVE TO THE ENGINE GEOMETRY
C FOR THE CYLINDRICAL PRECHAMBERS HA IS REF. TO THE HORIZONTAL PLANE
C AND VA TO THE VERTICAL PLANE IN THE PLAN VIEW OF THE PRECHAMBER

DO 500 I = 1, NGP

   UX(I) = (- (U1(I)**2) + U2(I)**2 + U3(I)**2) / 2.0
   IF (UX(I) LT 0.0) UX(I) = - UX(I)
   UX(I) = SQRT(UX(I))

   UY(I) = ( (U1(I)**2) - (U2(I)**2) + U3(I)**2) / 2.0
   IF (UY(I) LT 0.0) UY(I) = - UY(I)
   UY(I) = SQRT(UY(I))

   UZ(I) = ( (U1(I)**2) + U2(I)**2 - (U3(I)**2) ) / 2.0
   IF (UZ(I) LT 0.0) UZ(I) = - UZ(I)
   UZ(I) = SQRT(UZ(I))

   HA(I) = ABS(U1(I)) / SQRT(U2(I)**2 + U3(I)**2)
   HA(I) = ACOS(HA(I))
   HA(I) = HA(I) * 180.0 / 3.142
   RA = ABS(UY(I)) / (ABS(HA(I)) * AA)
   VA(I) = ACOS(RA)
   VA(I) = VA(I) * 180.0 / 3.142

500 CONTINUE

RETURN
END
C DEAL'S A GRID ON A SET OF AXES PROVIDED BY UTP4A IN THE MASTER.
DIMENSION X(2),Y(2)
NX = IFIX(XINS)
NY = IFIX(YINS)
X(1) = XMIN
Y(1) = YMIN
X(2) = XMAX
Y(2) = YMAX
DX = (XMAX-XMIN)/XINS
DY = (YMAX-YMIN)/YINS
DO 1 I = 1,NX
   DO 2 J = 1,NY
      Y(J) = Y(J) + DY
      X(I) = X(I) + DX
1 CALL UTP4A(X,Y,P,3)
   X(1) = XMIN
   Y(1) = YMIN
2 CALL UTP4A(X,Y,P,3)
RETURN
END

FUNCTION CNT(CH0,TO,P,TT)
C INPUT PARAMETERS.
C CH0- THERMAL CONDUCTIVITY AT REFERENCE TEMPERATURE.
C TO-REFERENCE TEMPERATURE.
C PPOWER FOR PARTICULAR GAS UNDER CONSIDERATION.
C FINDS THE THERMAL COND OF A SINGLE GAS AT TEMPERATURE TT
C FOR REFERENCE SEE TSEDERBERG,THERMAL CONDUCTIVITY OF GASES AND
C LIQUIDS,CHAPTER 3,PUBLISHERS ARNOLD.
C CNT= CH0*(((TT+273)/(TO+273))**P)
RETURN
END
FUNCTION FK1(K1, G1, G2, G3, G4)
C REAL K1, KA, KSA
C COMMON/E12/HDF, DA, CNWT, XA, XRW, RG, ALPHAA, XTW, TOA, KR, DGA, DKA
C EXTERNAL FUNCTION TO CJM02.
C DESCRIBES THE HEAT TRANSFER FROM A HEALED WIRE IN A CROSS FLOW OF
C GAS.
C INPUT PARAMETERS.
C K1 = ITERATION VARIABLE.
C
C G1 = EQUATION CONSTANT. (2*CNWS*RO/(Z*CNWT*RW))
C G2 = EQUATION CONSTANT. (2*CNWS*RO/(Z*CNWT*RW))
C G3 = EQUATION CONSTANT. ((RW-RO)/RW)
C KA = ABS(K1)
C DGA = KA
C KSA = SQRT(KA)
C
C CHECK FOR VELOCITY CYCLE OR FOR SECOND TEMP VOLTAGE CHECK
K = 2
IF ( KR .EQ. 1 ) GO TO 80
FK1 = G1*(1-G2*TANH(KSA*G4)/KSA-G3)*KA
GO TO 81
80 CONTINUE
C SECOND TEMP ENGINE VOLTAGE PREDICTION
DGA = 3.142 * HDF * DA / ( CNWT * XA )
DBG = XRW/RG
DGC = ALPHAA * (XTW-TOA)
G5 = (DGA-KA)*DBG/DGC
G1 = 1.0 * G5
FK1 = G1* (1.0 - G2*TANH(KSA*G4)/KSA-G3) - KA
81 CONTINUE
RETURN
END

FUNCTION VT(CF, VO, TO, TT, TC)
C CALCULATES THE DYNAMIC VISCOSITY OF A GAS AT TEMPERATURE TT.
C INPUT PARAMETERS.
C CF = POLYNOMIAL COEFFICIENTS FOR VISCOSITY FUNCTION VALUES.
C VO = VISCOSITY AT REFERENCE TEMPERATURE TO.
C TO = REFERENCE TEMPERATURE.
**FUNCTION CPT(AC,T,N)**

*Finds the specific heat of a single gas at Temp T* 

**By polynomial fit to published data**

**Input Parameters:**

- **AC** = 1-D array of length N containing the coefficients.
- **T** = Gas Temperature
- **N** = Order of polynomial + 1
- **DIMENSION AC(5)**

**CPT = 0.0**

**DO I=1,N**

**CPT = CPT + AC(I)*(T**(I-1))**

`1 CONTINUE`

**RETURN**

**END**

****

**DOCUMENT DATA**
ESBRV

Program to calculate 'Equivalent Solid Body Velocity' from the Swirl Components of Gas Velocity in the Prechamber.

1. JOB Control Cards
2. Read Constants
   Set Initial Values
3. Read Absolute Velocities
   and Vector Angles
4. Refer all Vector Angles to
   the same Reference Planes
5. Obtain Swirl Components of
   Gas Velocity and Output Results
6. Obtain Position Averaged Values
   of Swirl Velocities
7. Calculate 'ESBRV' using Simpson's
   Rule to Obtain Area under Radial Velocity
   Profile at Each Crank Angle
8. Output Results
   - Write and Plot
9. END
POSITIONS FOR EACH LOCATION; THEN TO DETERMINE THE EQUIVALENT SOLID
BODY ROTATIONAL VELOCITY FOR THE PARTICULAR PRECHAMBER

D. H. (140), CA (140), UC (140), UO (140), UO2 (140),
U03 (140), UO4 (140), UA1 (140), UA2 (140), UA3 (140), UA4 (140),
UA5 (140), UAA (140), UAA1 (140), UA1 (140), UA4 (140),
UA6 (140), UA7 (140), UAR (140)

NPOS = NO OF PROFILE POSITIONS, N = NO OF LOCATIONS DURING TRAVERSE

NOC = NO OF PROFILE ORIENTATIONS, NO = INPUT VELOCITIES

UO1, UO2, UO3, UO4 = ABSOLUTE VELOCITIES AT DIFFERENT ORIENTATIONS AT
PARTICULAR PROFILE POSITION AND LOCATION

UA = (UO1 + UO2 + UO3 + UO4) / NOC, UA1, UA2, UA3, UA4 = POSITION-AVERAGED
VELOCITY FOR EACH OF NL LOCATIONS = SUM(UA) / NPOS

UH = EQUIVALENT SOLID BODY VELOCITY AT PRECHAMBER PERIPHERY TO BE
COMPARSED WITH THE THEORY MATHEMATICAL MODEL

NG0 = 1 = GRAPH PLOT REQUIRED, OPEN AND CLOSE GRAPH PLOTTER
NG = 1 = READ AXES VALUES AND PLOT RESULTS

NG0 = 1
NG = 1
IF (NG0 .EQ. 1 ) CALL UTOP
READ (1, 1020) XMIN, XMAX, YMIN, YMAX, XINS, YINS
PI = 3.142
N = 600
READ (1, 1000) KORD, NADJ, NPOS, NL, NOC
CAI = 720.0 / (KORD - 1)
NPT = NADJ + KORD
NO = 1, KORD
UA(1) = 0.0
UA2(1) = 0.0
UA3(1) = 0.0
UA4(1) = 0.0
UA5(1) = 0.0
UA6(1) = 0.0
UA7(1) = 0.0
CA(1) = CAI * (I - 1)

CONTINUE
NO 50 JP = 1, NPOS
NO 40 JL = 1, NL
NO 5 1 = 1, KORD
IIA(J) = 0.0
3 CONTINUE
DO 20 NOR = 1, NOC
C READ ABSOLUTE VELOCITY AND HORIZONTAL AND VERTICAL ANGLES
READ (1,1010) (UO(I), I = 1, NPT)
READ (1,1010) (VA1(I), I = 1, NPT)
READ (1,1010) (HA1(I), I = 1, NPT)
DO 10 I = 1, KORN
C OBTAIN SWIRL COMPONENT OF VELOCITY
C THE ABSOLUTE VELOCITY MUST BE RESOLVED TO GIVE THE SWIRL COMPONENT
C REFER ALL VECTOR ANGLES TO ORIENTATION 1 REFERENCE AXES
C FOR ORIENTATION III = ORIENTATION NUMBER 2
  IF (JP .EQ. 1 .AND. NOR .EQ. 2) HA1(I) = 90.0 = HA1(I)
  IF (JP .EQ. 2 .AND. NOR .EQ. 2) VA1(I) = 90.0 = VA1(I)
  IF (JP .EQ. 3 .AND. NOR .EQ. 2) HA1(I) = 90.0 = HA1(I)
C FOR ORIENTATION IV = ORIENTATION NUMBER 3
  IF (JP .EQ. 1 .AND. NOR .EQ. 3) HA1(I) = ABS(45 = HA1(I))
  IF (JP .EQ. 2 .AND. NOR .EQ. 3) VA1(I) = ABS(45 = VA1(I))
  IF (JP .EQ. 3 .AND. NOR .EQ. 3) HA1(I) = ABS(45 = HA1(I))
HA1(I) = HA1(I) * PI / 180.0
VA1(I) = VA1(I) * PI / 180.0
C OBTAIN THE SWIRL COMPONENTS
IF(NOR .EQ. 1 .AND. JP .EQ. 1) UO1(I) = UO(I) * COS(HA1(I)) * COS(VA1(I))
IF(NOR .EQ. 1 .AND. JP .EQ. 1) UO2(I) = UO(I) * COS(P1/2.0 - VA1(I))
IF(NOR .EQ. 1 .AND. JP .EQ. 3) UO2(I) = UO(I) * COS(HA1(I)) * COS(VA1(I))
IF(NOR .EQ. 2 .AND. JP .EQ. 2) UO2(I) = UO(I) * COS(P1/2.0 - VA1(I))
IF(NOR .EQ. 2 .AND. JP .EQ. 3) UO2(I) = UO(I) * COS(HA1(I)) * COS(VA1(I))
IF(NOR .EQ. 3 .AND. JP .EQ. 1) UO3(I) = UO(I) * COS(HA1(I)) * COS(VA1(I))
IF(NOR .EQ. 3 .AND. JP .EQ. 1) UO3(I) = UO(I) * COS(P1/2.0 - VA1(I))
IF(NOR .EQ. 3 .AND. JP .EQ. 3) UO3(I) = UO(I) * COS(HA1(I)) * COS(VA1(I))
IF(NOR .EQ. 1) UA(I) = VA1(I) + UO1(I)
IF(NOR .EQ. 2) UA(I) = VA1(I) + UO2(I)
IF(NOR .EQ. 3) UA(I) = VA1(I) + UO3(I)
10 CONTINUE
C ARRANGE TO PLOT ABSOLUTE VELOCITY AND VECTOR ANGLES BET. 180 - 540 DEGREES
JK = KORN/2 + 1
JJ = KORN / 4
DO 500 JI = 1, JK
CAA(JI) = CA(JI+JJ)
VAA(JI) = VAI(JI+JJ) * 180.0/PI

501 CONTINUE
IF ( JP < EQ. 1 .AND. NON < EQ. 1 ) GO TO 660
IF ( JP < EQ. 2 .AND. NON < EQ. 1 ) GO TO 660
IF ( JP < EQ. 3 .AND. NON < EQ. 1 ) GO TO 660
IF ( JP < EQ. 3 .AND. NON < EQ. 2 ) GO TO 660
IF ( JP < EQ. 3 .AND. NON < EQ. 3 ) GO TO 660
GO TO 601

601 CONTINUE
CALL UTPLA(XMIN,XMAX,YMIN,YMAX,XINS,VINS, 'CRANK ANGLE', 2,
'ABSOLUTE VELOCITY M/S', 3)
CALL UTPLA(CAA, UPA, JK, 2)
CALL UTPLA(CAA, HAA, JK, 2)
CALL UTPLA(CAA, VAA, JK, 2)
CALL GRIN(XMIN,XMAX,YMIN,YMAX,XINS,VINS)
GO TO 602

C ARRANGE TO PLOT SWIRL VELOCITY COMPONENT BETWEEN 180 - 540 DEGREES
502 CONTINUE
UPA(JI) = U01(JI+JJ)
503 CONTINUE
IF ( NON < NF. 2 ) GO TO 505
504 CONTINUE
UPA(JI) = U02(JI+JJ)
505 CONTINUE
IF ( NON < NF. 3 ) GO TO 507
506 CONTINUE
UPA(JI) = U03(JI+JJ)
507 CONTINUE
CALL UTPLA(XMIN,XMAX,YMIN,YMAX,XINS,VINS, 'CRANK ANGLE', 2,
'SWIRL COMPONENT M/S', 3)
CALL UTPLA(CAA, UPA, JK, 2)
CALL GRIN(XMIN,XMAX,YMIN,YMAX,XINS,VINS)
601 CONTINUE
602 CONTINUE
20 CONTINUE
DO 30 I = 1, KORD
   UA(I) = UA(I) / NPOS
   IF ( JL .EQ. 1 ) UA1(I) = UA1(I) + UA(I)
   IF ( JL .EQ. 2 ) UA2(I) = UA2(I) + UA(I)
   IF ( JL .EQ. 3 ) UA3(I) = UA3(I) + UA(I)
   IF ( JL .EQ. 4 ) UA4(I) = UA4(I) + UA(I)
30 CONTINUE
C FOR THE 1st CYLINDER
   WRITE (2,2000) ( (CA(I),UO1(I),UO2(I),UO3(I),UO4(I),UA(I) ), I = 1,1, KORD )
   WRITE (2,2001)
40 CONTINUE
GO TO 500
C ARRANGE TO PLOT ORIENTATION AVERAGED SWIRL COMPONENT BET. 180-540 DEGREES
DO 500 J1 = 1, JK
   UPA(J1) = UA(J1+J)
500 CONTINUE
CALL UTP4A(XMIN,XMAX,YMIN,YMAX,XINS,YINS, 'CRANK ANGLE', 2, 'ORIENT AV. SWIRL VS', 3)
   CALL UTP4D( CA, UPA, JK, 2)
   CALL GRID( XMIN,XMAX,YMIN,YMAX,XINS,YINS)
500 CONTINUE
WRITE (2,2011)
50 CONTINUE
DO 60 I = 1, KORD
   UA1(I) = UA1(I) / NPOS
   UA2(I) = UA2(I) / NPOS
   UA3(I) = UA3(I) / NPOS
   UA4(I) = UA4(I) / NPOS
60 CONTINUE
C FOR 1st CYLINDER ONLY
   WRITE (2,2012) (CA(I),UA1(I),UA2(I),UA3(I),UA4(I), I = 1, KORD)
2012 FORMAT (1H , 5F15.1 )
C START THE INTEGRATION PROCEDURE TO OBTAIN EQUIVALENT SOLID BODY VELOCITY
C DECIDE WHETHER PRECHAMBER IS SPHERE OR CYLINDER AND USE APPLICABLE
C FUNCTION FOR INTEGRATION
C SPECIFY RADIAL LOCATIONS RADII FROM CENTRE OF PRECHAMBER
C FOR CYLINDRICAL PRECHAMBER
   R0 = 0.0125
   R1 = 0.0115
C USE TRAPEZOIDAL RULE TO OBTAIN AREA UNDER RADIAL VELOCITY PROFILE
C NI = # OF STRIPS BETWEEN TWO RADII, H = WIDTH OF STRIP, NS = COUNTER
C OF 4 SEPARATE INTEGRATION SECTIONS CORRESPONDING TO 4 RADIAL LOCATIONS

NI = 20
DO 100 R0 = 1, KORD
NS = 1
ATOT = 0.0
X1 = R1
J = 1
H = ( R0 - R1 ) / NI
100 Y2 = X1 + H
X3 = X2 + H
ATOT = ATOT + ( H/3.0 ) * ( FC(UA1(I),X1,R1,R0,0,0) + 4.0*FC(UA1(I)
1,Y2,R1,R0,0,0) + FC(UA1(I),X3,R1,R0,0,0) )
IF ( J .EQ. (NI-1) ) GO TO 110
J = J + 2
X1 = X3
GO TO 100

110 CONTINUE
NS = 2
X1 = R2
J = 1
H = ( R1 - R2 ) / NI
120 Y2 = X1 + H
X3 = X2 + H
ATOT = ATOT + ( H/3.0 ) * ( FC(UA2(I),X1,R2,R1,UA1(I)) + 4.0*FC(UA
2(I),X2,R2,R1,UA1(I)) + FC(UA2(I),X3,R2,R1,UA1(I)) )
IF ( J .EQ. (NI-1) ) GO TO 130
J = J + 2
X1 = X3
GO TO 120

130 CONTINUE
NS = 3
X1 = R3
J = 1
H = ( R2 - R3 ) / NI
140 Y2 = X1 + H
\[ x_3 = x_2 + h \]
\[ ATOT = ATOT + (H/3,0) \cdot (FC(UA3(I), x_1, r_3, r_2, UA2(I)) + 4,0 \cdot FC(UA3(I), x_2, r_3, r_2, UA2(I))) \]

IF \((J.FQ. (NI-1)) \) GO TO 150
\[ j = j + 2 \]
\[ y_1 = x_3 \]
GO TO 140

150 CONTINUE
\[ NS = 4 \]
\[ y_1 = r_4 \]
\[ j = 1 \]
\[ H = \frac{(R_3 - R_4)}{NI} \]

160 \[ y_2 = x_1 + H \]
\[ x_3 = x_2 + H \]
\[ ATOT = ATOT + (H/3,0) \cdot (FC(UA4(I), x_1, r_3, r_3, UA3(I)) + 4,0 \cdot FC(UA4(I), x_2, r_3, r_3, UA3(I))) \]

IF \((J.FQ. (NI-1)) \) GO TO 161
\[ j = j + 2 \]
\[ y_1 = x_3 \]
GO TO 160

161 CONTINUE

170 CONTINUE
C EVALUATE SOLID BODY MEAN VELOCITY AT WALL RADIUS, RS = SOLID BODY RADIUS
C FOR CYLINDRICAL PRECHAMBER
\[ RS = R_0 - 0.0005 \]
\[ U_{M1}(I) = (4,0 / RS**3,0) \ast ATOT \]

180 CONTINUE

181 CONTINUE
C WRITE OUT 'EQUIVALENT SOLID BODY VELOCITY' FOR ALL CRANK ANGLES
C IF \((NG.NE.1)) \) GO TO 190
C ARRANGE TO PLOT BETWEEN 180 = 360 CRANK ANGLE
C FOR KORD = EVEN \(JK = (KORD/2)\), FOR KORD = ODD \(JK = (KORD/2) + 1\)
\[ JK = (KORD/2) \]
\[ JJ = (KORD/4) \]
\[ NO 360 I = 1, JK \]
C USE KORD/4 ON THE RIGHT HAND SIDE OF THE FOLLOWING EXPRESSIONS
\[ CA(I) = CA(I+JJ) \]
\[ UA1(I) = UA1(I+JJ) \]
C  FOR THE INTEGRATION BETWEEN TWO MEASURING POINTS (LOCATIONS)
C  REFERENCE POINT IS CENTRE OF PRECHAMBER
C  A = FIRST VELOCITY,  B = INSTANTANEOUS RADIUS,  C = FIRST RADIUS
C  D = SECOND RADIUS,  E = SECOND VELOCITY
C  FOR A CYLINDRICAL PRECHAMBER
FC = ( A + ( (B+C)/(D-C) ) * (E-A) ) * (R**2.0)
RETURN
REFERENCES


28. WALDER, C.J. 'Reduction of Emissions from Diesel Engines' SAE Paper 730214


