Investigations of turbulent flow in the combustion chamber of a spark-ignition engine

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INVESTIGATIONS OF TURBULENT FLOW IN THE COMBUSTION CHAMBER OF A SPARK IGNITION ENGINE

By

E. H. JAMES

THESIS Submitted in fulfillment of the requirements for the award of Doctor of Philosophy of Loughborough University of Technology

SUPERVISOR: DR. G. G. LUCAS

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SUMMARY

The thesis describes an investigation into the turbulent flow existing in the two limiting designs of spark ignition engine combustion chamber i.e. 'squish' and cylindrical disc designs. The analysis is concentrated on the compression stroke and the early part of the expansion stroke.

The application of hot wire anemometry to such work is described especially with regard to anemometer adjustment for optimum frequency response and hot wire probe calibration. The latter was achieved by utilizing an analytical procedure in which a heat balance of the wire was generated. The varying effects of temperature, pressure and flow velocity on the wire's convective heat loss characteristics were catered for by the Nusselt Number - Reynolds Number relationship of Collis and Williams\(^4\). Excellent calibrations were achieved.

The development of the probe traverse mechanism facilitating probe entry and positioning in the combustion chambers is detailed together with a description of the instrumentation set-up developed. The output voltage signals of the anemometer were processed so that quantities proportional to the turbulent flow fluctuations within the following frequency bands were obtained:

- 0-200 Hz;
- 90-180 Hz;
- 180-360 Hz;
- 360-700 Hz;
- 700-1500 Hz;
- 1500-2800 Hz;
- 2800-5800 Hz.

The effect of engine speed is particularly investigated as also is the effect of compression ratio for the 'squish' chamber design. It is shown that most of the turbulent energy in the combustion chambers studied is confined within the frequency region below 1000 Hz. Also, the main influence of a change in combustion chamber design is noted to be an alteration in the magnitude of the low frequency flow fluctuations.
The measured flow velocities in the 'squish' chambers were utilized in the development of a theoretical turbulent burning velocity expression to predict rates of turbulent flame propagation. Modifications to an existing computer program (1), simulating the combustion process in a spark ignition engine, were made to achieve this. It was found that a good correlation with experimental flame travel time measurements could be achieved by using the Karlovitz (98) turbulent burning velocity expression. This includes a term for the effects of flame generated turbulence which were shown to be extremely significant in boosting up the burning velocities in spark ignition engine combustion chambers.
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Finally, the encouragement of Professor F. P. Rebello of the Department of Mechanical Engineering, I.T.A. São José dos Campos, São Paulo, Brazil in the thesis write-up is most appreciated.
**NOTATION**

\( a \) ............ Overheating Ratio \( \frac{R_w-R_C}{R_o} \)

\( A \) ............ Amplifier Gain: Wire Cross-Sectional Area

\( A^* \) ............ modified value of contact area between hot wire and prong tip

\( A_s \) ............ Surface Area of Hot Wire

\( C \) ............ modified heat capacity of hot wire

\( C_{pw} \) ............ specific heat of wire material

\( d \) ............ wire diameter

\( d_p \) ............ prong tip diameter

\( db \) ............ decibel

\( e, \bar{e}, e' \) ....... instantaneous, mean and fluctuating Bridge balance voltages

\( e_1, \bar{e}_1, e'_1 \) ....... instantaneous, mean and fluctuating square wave bias inputs to amplifier

\( E, \bar{E}, E' \) ....... instantaneous, mean and fluctuating Bridge Output Voltages

\( f \) ............ frequency (Hz)

\( f_o \) ............ open-loop frequency response (Hz)

\( f_c \) ............ closed-loop anemometer probe frequency response (Hz)
\( F \) ............ capacitance (farads)

\( G \) ............ amplifier transconductance

\( h \) ............ 'squish' height i.e. distance above piston at TDC to 'squish' area of cylinder head; convective heat transfer coefficient

\( h' \) ............ instantaneous distance from piston to 'squish' area of cylinder head

\( H_w, \overline{H_w}, H_w' \) .. instantaneous, mean and fluctuating values of overall heat transfer coefficient

\( I, I, I' \) ..... instantaneous, mean and fluctuating values of current through hot wire probe (amperes)

\( l \) ............ hot wire length

\( l_d \) ............ size of viscous dissipation eddies

\( L \) ............ inductance (henries): scale of turbulent eddies

\( m \) ............ metre

\( Ma \) ............ Mach Number

\( n \) ............ engine speed

\( Nu \) ............ Nusselt Number

\( P \) ............ Pressure (atm)

\( Pr \) ............ Prandtl Number

\( Q_c \) ............ heat loss by conduction along the hot wire
\( Q_e \) .......... heat supplied to hot wire by anemometer
\( Q_h \) .......... convective heat transfer to normal air flow
\( Q_r \) .......... heat loss by radiation from hot wire surface
\( r \) .......... radius of flame front from spark plug
\( R_c \) .......... total resistance in the leads to the hot wire of the probe – excluding wire resistance itself (ohms)
\( R_w, \bar{R}_w, R'_w \) .......... instantaneous, mean and fluctuating values of hot wire operating resistance (ohms)
\( R_0 \) .......... hot wire resistance at ambient temperature (ohms)
\( R_1 \) .......... resistance in Bridge arm of anemometer in series with probe arm (ohms)
\( Re \) .......... Reynolds Number
\( s \) .......... second
\( t \) .......... time (seconds)
\( T_f \) .......... film temperature (\( = \frac{T_w + T_g}{2} \))
\( T_g \) .......... cylinder gas temperature
\( T_H \) .......... prong tip temperature
\( T_o \) .......... ambient temperature
\( T_s \) .......... temperature of surroundings
$T_w$ ........... mean hot wire operating temperature

$u$ ........... turbulent velocity fluctuations of
frequency less than 200 Hz at an
instantaneous point during an engine cycle

$\bar{u}$ ........... average value of $u$ over many engine cycles

$u'$ ........... turbulent velocity fluctuations within
discrete frequency bandpass ranges at an
instantaneous point during an engine cycle

$\bar{u}'$ ........... average value of $u'$ over many engine cycles

$U$ ........... steady mean flow velocity

$U_L$ ........... laminar burning velocity

$U_T$ ........... turbulent burning velocity

$U_{FG}$ ........... flame generated turbulence

$v$ ........... anemometer output voltage fluctuations
below 200 Hz

$v'$ ........... anemometer output voltage fluctuations in
discrete frequency bandpass ranges

$V_B$ ........... anemometer output voltage in steady flow

$V_w$ ........... voltage across hot wire only
\( \alpha \) ........... temperature coefficient of resistance of wire material (per \( ^\circ C \))

\( \Omega \) ........... ohms

\( \tau \) ........... Time Constant of hot wire only (seconds)

\( \tau_c \) ........... Time Constant of hot wire probe and anemometer circuitry (seconds)

\( \lambda_o \) ........... air thermal conductivity at ambient temperature

\( \lambda_g \) ........... air thermal conductivity at gas temperature, \( T_g \)

\( \lambda_w \) ........... air thermal conductivity at hot wire temperature, \( T_w \)

\( \lambda_H \) ........... thermal conductivity of wire material at prong tip temperature, \( T_H \)

\( \lambda_m \) ........... thermal conductivity of wire material at temperature \( T_w \)

\( \rho \) ........... density of air (gm/cm\(^3\))

\( \rho_w \) ........... density of wire material (gm/cm\(^3\))

\( \rho_r \) ........... resistivity of wire material (micro-ohm cm)

\( \rho_u \) ........... density of unburnt charge in combustion chamber

\( \rho_b \) ........... density of burnt charge in combustion chamber

\( \delta \) ........... width of laminar combustion zone
ϕ ............ angle of probe rotation from normal position

ω₀ ............ natural frequency of anemometer feedback circuitry

η ............ damping ratio of anemometer feedback circuitry

ε_I ............ surface emissivity of wire

κ ............ Stefan Boltzman Constant

μ₀ ............ air dynamic viscosity at ambient conditions

μ_g ............ air dynamic viscosity at gas temperature, T_g

ATDC ........ After Top Dead Centre

BTDC ........ Before Top Dead Centre

TDC ............ Top Dead Centre

D.I.S.A. ... Dansk Industri Syndikat A/S

Pt ............ Platinum

Rh ............ Rhodium

Ir ............ Iridium
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CHAPTER 1

INTRODUCTION
CHAPTER 1
INTRODUCTION

Very little fundamental research has been conducted into the nature of the turbulent flow which exists in spark ignition engine combustion chambers. This is somewhat surprising in view of the known great importance which the turbulent flow has on the rate of flame propagation, cyclic dispersion and also heat transfer. It may be explained partly by the great difficulties encountered practically in such investigations and partly because the significance has not been fully realized of the interaction between the combustion zone and the turbulent flow field in determining rates of turbulent burning. The requirement at present seems to be that only a condition of general and indiscriminate turbulence is required in the combustion space to speed up the flame sufficiently to complete combustion within the short time available. The type of turbulent flow is still largely of little interest in engine design.

The work reported here arose from a previous study\(^{(1)}\) which attempted to simulate the combustion process in a spark ignition engine on a digital computer. Although this was in many ways successful, it was considered that the inability to accurately describe the rate of turbulent burning detracted from the generation of a realistic model.

Consequently, a study was envisaged which had as its main goals

1) the measurement of the fluctuating flow velocities in different types of spark ignition engine combustion chambers over the frequency range in which the fluctuations were thought to be of importance to flame propagation (up to 6000 Hz).
ii) the development of an expression for the turbulent burning velocity based on the measured flow velocities. The accuracy of this expression, when used in a computer simulated model of the combustion process, was to be determined by comparison of flame travel times across the combustion chamber from the results of the computer model with some flame speed measurements in a firing engine.

The first part of the work is concerned with the application of hot wire anemometry techniques for the flow velocity measurements and with the development of the engine rig and instrumentation system to facilitate such measurements. Results are later presented for all the forms of combustion chamber studied together with an analysis of the rates of turbulent flame propagation in these chambers.

The scope of the work is such that an insight is provided into the physical processes underlying the phenomenon of flame propagation in engine cylinders. Additionally, the type of turbulent flow normally present in such cylinders is defined. It is only when knowledge such as this becomes available that improvements can subsequently be effected in engine design.
CHAPTER 2

FLOW VELOCITY MEASUREMENT IN INTERNAL COMBUSTION ENGINE CYLINDERS
Various techniques are available to measure the unsteady, dynamic fluctuating flows which exist in engine combustion chambers. Enormous practical difficulties are encountered, however, because the flow velocities are generally completely random in both magnitude and direction, not only from one instant to another within the same cycle but also from cycle to cycle. Additionally, measurements made at one location in the combustion chamber may be unrepresentative of those at any other.

A great strain is thus imposed on the measuring equipment used in order to attain accurate results.

The measurement of unsteady flows can be approached in two ways:

i) by the use of optical methods.

ii) by the use of a detector within the flow.

2.1 Optical Methods of Measurement

These include:

a) methods involving the use of a tracer, or other indicator, introduced into the fluid to make the flow pattern visible or observable by a suitable detecting apparatus outside the field of flow. Particle-track photography is one example in this category. This has been used by Levy and Weinberg(5) who used 4 micron Betonite particles. In calculating the gas flow velocity, corrections have to be applied to allow for the downwards velocity due to free fall of the particles. The corrections decrease as the gas temperature, and hence viscosity, increase.
b) Schlieren and Shadowgraph techniques - these are based on the influence of changes in the physical properties of a fluid on the passage of a light beam through the fluid. If the light beam passes through a density gradient in a gas (and, therefore, through a gradient in refractive index), it is deflected as though by a lens. In many cases the density gradients are sufficiently large to make such phenomena observable by optical methods. The Schlieren method measures the density gradient and is normally only used to give a simple visual picture of the flow field together with a rough picture of the density variations in the flow. The Shadowgraph method measures the second gradient of density and, therefore, only indicates those parts of the flow where the density gradients change rapidly. Its main application is, consequently, in the study of shock waves.

c) Interferometry - this is more expensive and complicated to use than a schlieren or shadowgraph set-up but it enables more quantitative results to be obtained. It enables the refractive index to be determined which, as has already been stated, depends upon the density. Subsequent calculations can reveal the velocity in a flowing gas.

d) Laser Anemometry - the laser is proving a most valuable tool in flow measurement work especially in combustion studies. The principles of operation of such anemometers have received detailed attention in, for example, References 7 and 8. A typical system consists of a collimated beam of high
intensity, monochromatic light together with an arrangement of mirrors, beam splitters and lenses to permit operation in any chosen optical mode e.g. fringe, Doppler or reference beam mode. A light collecting system allows the signal from a particular measuring control volume to be observed by the cathode of a photomultiplier and the resulting electrical signal can be analysed. It is essential that the fluid, in which the measurements are to be made, is seeded with particles of sufficient size to allow the intensity of scattered light to be detected while still representing the instantaneous velocity of the fluid. Along with other optical techniques, laser anemometry has the advantage that the flow is not disturbed at the point of measurement. Additionally, measurements are not affected by the medium density, pressure and temperature and no calibration is required. At the time this work was started, no commercial system was available and so this technique was not used. However, D.I.S.A. have since started marketing a unit based on the laser doppler technique. Depending on application and number of particles in the measurement system, four different modes of operation are available. The flow velocity, \( U \), is calculated from

\[
U = \frac{f_d \lambda}{2 \sin(\theta/2)}
\]

where

- \( f_d \) is the Doppler frequency.
- \( \lambda \) is the wavelength of the laser light.
- \( \theta \) is the scattering angle.
Such a system should be extremely adaptable to flow measurements in engine cylinders. As well as the advantages listed above, it is easy to align, robust and not very susceptible to vibration. However, it has a poor signal/noise ratio.

A big disadvantage associated with optical techniques is that the glass windows used to observe the flow tend to refract the beams transmitted through them. They ought, therefore, to be perfectly flat and of constant thickness. In laser anemometry work in combustion, the refractive index in hot flames also becomes important.

Many examples exist in the literature of optical techniques, and high speed photography, being used to study the combustion process in both spark ignition and compression ignition engines (10)(11)(12)(13). However, the author is unaware of any that have attempted to calculate the flow velocities by such methods, except that of Lee (14) in 1938. Air flow motion was made visible by mixing feathers with the induction air in a single cylinder, 4-stroke engine. These were photographed through a glass cylinder wall while the piston moved through its cycle. Different types of swirl velocities were produced by the use of shrouded intake valves. The use of optical methods in studying the air flow into the cylinder during induction is widespread and will not be discussed here.

2.2 Detector Methods of Measurement

Certain requirements are essential of a detector used to measure flow velocities. These include:

a) miniaturization of the sensing element so that minimum disturbance of the flow pattern is created and so that the velocity can be assumed constant in the region occupied by the element.
b) low inertia of the detecting element in order to record high frequency flow oscillations and variations.
c) high mechanical strength and rigidity to withstand vibrations, oil film contamination, pressure variations etc.
d) adequate sensitivity to small changes in flow velocity.
e) minimal drift in the calibration.
f) a good directional sensitivity.

Pitot-static tubes and vanes are, thus, of little use when measured against the above ideal criteria. Measurements of air flow velocities can be accomplished using inductive pressure pick-ups to indicate the dynamic pressure of the flowing air. However, their relatively large dimensions are a disadvantage.

The only instruments that satisfy most of the detector requirements listed above are the electric discharge anemometer and the hot wire and hot film anemometers.

Electric discharge anemometry is based on the premise that atmospheric air is not a perfect insulator as it permits the flow of a more or less feeble current in a region of an electric field between two electrodes. This current depends upon several variables among which are field strength, the electrode material, spacing and configuration, the production of ions by any outside influence, the pressure and the air velocity. If all but the last of these are held constant, the dependence of the current on velocity suggests the use of an electric discharge as a device for anemometry. This method gives local and instantaneous values of the velocity rather than values averaged over space and time. Three types of discharge are used. When the tips of two wires have a gap separation of a few thousandths of an inch, an applied
voltage of 6 to 12 kilovolts will cause a current of $10^{-12}$ to $10^{-9}$ amperes between them. This current depends on a continuous supply of ions, produced, for example, by X-rays or ultraviolet illumination. It would cease if all such outside ionization was stopped and it is known as a "Townsend discharge". If the voltage is increased, the current rises to $10^{-7}$ or $10^{-6}$ amperes and the discharge gradually becomes luminous and is now known as a "corona discharge". It is now no longer dependent upon external ionization. Further increase in the current to a few milliamperes causes the discharge to become a "glow discharge" which is also self-sustaining.

The effect of pressure upon electric discharges reduces somewhat their anemometric usefulness. The static pressure must, in principle, be known before the velocity can be deduced.

Hinze(15) describes in detail the basic development research of electric discharge anemometry and the work of Fucks(16) and Lindvall(17) in this connection. Great difficulties can be visualized with this technique in the measurement of very high frequency flow fluctuations.

Both hot wire and hot film anemometry are now established techniques in measuring both mean flow velocities and turbulent flow fluctuations. Almost all the information available on such flows to date has been obtained by the use of such methods. The essential principle involved is as follows. If a small body is placed in a moving medium and heated to a temperature higher than that of the medium, a heat exchange occurs between the body and the medium. The rate of heat loss depends on the geometrical and physical properties of the body, the physical characteristics of the medium and the characteristics of the flow conditions. Assuming that only one of the parameters affecting the rate of heat loss varies, the heat loss can be interpreted as a direct measure
of the quantity in question e.g. the velocity of a fluid of constant composition, temperature and pressure. Therefore, a flow velocity calibration can be effected. The electronic section of the anemometer serves the purpose of operating the probe and enhancing its performance with respect to frequency response. Two typical modes of operation are normally conceived of:

a) constant current operation (see Fig. 1a).

b) constant temperature (or resistance) operation (see Fig. 1b).

In either mode, it is normal practice to operate the sensor in one arm of a Wheatstone Bridge circuit. In the constant current mode, the bridge circuit is operated from a source of current in series with a resistance which is high compared with the bridge resistance. The current through the hot wire can thus be kept constant. The resistance changes, or DC bridge unbalance voltages, are then a measure of the mean flow velocity. The AC component provides a picture of the fluctuations in the flow. This is only valid, however, at low frequency fluctuations because the temperature of the sensor varies with the flow conditions and its thermal inertia, although small, becomes important. To compensate for this, the output signals are amplified by a compensating amplifier which is so adjusted that its harmonic response corrects the error due to the thermal inertia (Fig. 1a) - that is, the time constant of the compensating network must equal the time constant of the wire. The latter is a function of the mean flow velocity and the compensating network should, therefore, be readjusted when the mean velocity changes. In general, the constant current anemometer is unsuitable for measurements when large velocity fluctuations are present due to the high risk of burning out the wire. It is, therefore,
largely unsuited to use in piston engine work. A lot of fundamental experimental research on turbulent flow, however, has been performed with constant current anemometers but generally at low flows where its sensitivity is greatest. Hot film probes are not truly compatible with the constant current mode of anemometer operation because their frequency response characteristics are quite complex. This is due to the influence of the substrate (usually quartz) on which the film is deposited. Consequently, the design of a compensating amplifier, having the inverse dynamic characteristics of the probe, is also very difficult.

The idea of a constant temperature anemometry system had been conceived many years before it became a functional reality. This was because it was difficult to obtain a satisfactory servo-amplifier (see Fig. 1b). Developments in transistorized devices corrected this deficiency. The basic technique involved is the minimization of the effect of the probe thermal inertia by maintaining the sensor at a constant temperature (resistance). The heating current then becomes a measure of the heat transfer and, hence, also of flow velocity. In practice, although nominally called a constant temperature or resistance anemometer, very small changes in these parameters do occur as they are needed to provide a signal to drive the servo-amplifier (Fig. 1b). The operation of the system is as follows. When the bridge circuit is in balance, a voltage is present in the vertical bridge diagonal. This is supplied by the servo-amplifier. If the heat transfer from the sensor changes slightly, a small voltage is generated across the horizontal diagonal of the bridge. This is fed to the servo-amplifier where it is greatly amplified before appearing across the vertical bridge diagonal. In this manner, the temperature and resistance variations of the sensor are kept extremely small.
The constant temperature system is, therefore, operationally more simple and accurate than its constant current counterpart and is well suited to measurements of large velocity fluctuations as exist, for example, in engine cylinders. Another advantage is the attainment of greater flow sensitivity at very high sensor temperatures with no risk of probe burnout if the velocity decreased suddenly.

As stated above, the sensor used with anemometers is either a thin wire suspended between two prongs or a thin metal film fused to a substrate. An analytical calibration procedure is very much simpler for the hot wire probe and its frequency response characteristics are also much simpler. The mechanical characteristics of the hot film probe are superior, however, so that it finds more ready application in 'hostile' environments. The mechanical strength of wire probes has been shown to be very adequate for many applications nevertheless.

Many examples appear in the literature of flow detectors, as have been described above, being utilized for flow velocity measurement in piston engine combustion chambers. For example, Urlaub\(^{(18)(19)}\) has constructed and used a probe incorporating an inductive pressure pick-up which operates on the principle of dynamic-head measurement of the flow. A diaphragm is deflected under the incidence of the flow and the deflection is measured by the inductive transducer. Any variation in deflection caused by the gas pressure is avoided by suitable design of the probe. Its low sensitivity to the flow fluctuations which exist in spark ignition combustion chambers and large dimensions prohibits against its use in such a situation. Rödig and Zalud\(^{(20)}\) have also experimented with such devices. In place of the inductive transducer, however, strain gauges have been used. Evaluation of in-cylinder flow velocity measurements had to take into account the varying effects of
temperature and pressure on the standard calibration.

Electric discharge anemometry has also been applied to measurements of flow velocities in piston engine combustion chambers. A modified form of this technique has been successfully reported by Nakajima et al (21), who made measurements of swirl velocities in various types of swirl chambers of the Ricardo type. High frequency (up to 20000 Hz) and high voltage (up to 120 kv) pulses were supplied to a pair of electrodes. The discharges created plasmas along their paths, the locations of which changed with the movement of the air. By supplying several electric pulses to the electrodes for a definite time and filming the resulting flashes of light through a transparent window in the combustion chamber, striped patterns were obtained on the film. These show the movement of the air in equal time intervals.

Ohigashi et al (22) published some work on a similar device based on the phenomenon that the path of an electric discharge moves downstream under the influence of a gas stream. Instead of photography being used to trace the path of the discharge however, a probe was placed at a fixed distance downstream from the electrodes to detect when the discharge path arrived. The stream velocity was then a function of the time interval between the beginning of the spark and the rising of the probe voltage. Results were obtained using this technique on a two-stroke, spark ignition engine.

Many examples can be found in the literature of hot wire anemometry being used to study the air flow into a piston engine cylinder during the induction stroke e.g. Tindal (23), Lobo (24). Additionally, many steady-flow models have been used for this purpose. Such work is not of direct importance here and so will not be reviewed.
Wenger\textsuperscript{(25)} seems to have been the first to use a hot wire probe in the combustion space of a 'motored' single cylinder engine. Constant current anemometry was used and the hot wire was 5 mm long, 15 micron in diameter and made of tungsten. Mean flow velocities were recorded for the entire engine cycle.

Hassan\textsuperscript{(26)} used a constant temperature anemometry system, to measure swirl velocities in a single cylinder, 'motored' engine. The varying effects of temperature, pressure and flow velocity on the heat loss from the wire were catered for by an analytical calibration procedure which used a correlation developed by Davies and Fisher\textsuperscript{(27)}. This expressed the Nusselt Number as a function of the skin friction coefficient and is described in Chapter 4. Hassan, eventually used his measurements to derive a heat transfer expression.

Horvatiln\textsuperscript{28} and Hussmann\textsuperscript{28} measured flow velocities in the cylinder of a 'motored' diesel engine equipped with a MAN-swirl producing cylinder head and a MAN-M piston. A constant temperature anemometer system was used and the hot wire was made of Pt-10\%Rh and was 15 micron in diameter and 3 mm long. An analytical calibration of the hot wire was effected by means of the formula of Grigull\textsuperscript{(29)}. As a corollary to this work, Horvatiln\textsuperscript{(30)} evaluated, in considerable detail, the characteristics of hot wires immersed in a fluid of varying pressures and temperatures.

Molchanov\textsuperscript{(2)}, in 1955, appears to have been the first person to use a hot wire probe in a 'motored' spark ignition combustion chamber where no large scale swirl was present. Flow velocities and fluctuations of the flow velocities up to 2000 Hz were recorded in a cylindrical disc chamber both during the inlet and compression strokes. The detector was a tungsten wire of diameter 19 microns and the length was 4.5 mm. The effect of compression ratio was investigated at one engine speed (900 rev/min) only. A detailed
account of Molchanov's work has been described in Ref. 1.

In 1958, Semenov(3) published results of an excellent study into the flow in a cylindrical disc chamber in a 'motored' spark ignition engine. Ref. 1 again summarizes these results. Both the induction and compression strokes were studied and many engine operating parameters were varied e.g. engine speed, degree of throttling and compression ratio. The probe was also traversed across the combustion chamber between the valves. Semenov maintained that the conditions during the flame propagation period represented practically constant turbulence and he showed that the flow fluctuations, during this period, were the result of the velocity gradients, arising from the jet flow of gas through the inlet valve, during induction. The jet flow was found to have disappeared during the compression stroke so that only an eddying gas motion remained. Most of the turbulent energy at this time was noted to be concentrated in the frequency band below 1000 Hz. Semenov utilized a complex instrumentation set up(31) to automatically process the output signals of a constant temperature anemometer and a resistance thermometer. The latter was used to measure cylinder gas temperatures. His set-up also enabled a statistical treatment to be made of a 'mean flow' velocity (involving frequency fluctuations below 300 Hz) and the fluctuating flow velocities (in the range 300 to 6200 Hz) over many engine cycles.

The calibration procedure he used is described in Chapter 4. No experimental comparison was given. Only one combustion chamber shape (i.e. cylindrical disc) was studied and no attempt was made to relate the measured flow velocities to flame speeds in the chamber. Additionally, the flow velocity measurements must be treated with some suspicion because his calibration procedure did not include end conduction losses from the hot wire to the prongs.
This can significantly affect the calibration especially at the low flow velocities he recorded. An 11 micron diameter sensing element was used and it was made of tungsten. This material is known to oxidize at about 350°C. Consequently, the difference between his wire temperature and the gas temperature cannot have been very great at the end of the compression stroke. This will reduce the accuracy of this results. Relatively wide bandpass ranges were used to plot a turbulent energy spectrum of the flow fluctuations and he indicates the need for smaller ranges in the low frequency region of the flow.

A hot wire probe has also been used by Patterson (32) in a spark ignition engine to substantiate an hypothesis that the cyclic fluctuations in gas flow near the spark plug during ignition were responsible for the cyclical variations in flame speeds and, hence, pressure development during combustion. No attempt was made to calibrate the probe.

Barton et al (33) used a constant temperature hot film probe to also determine the low frequency velocity fluctuations from cycle to cycle in a "motored" engine in the vicinity of the spark plug. The objective was to develop an empirical model for correlating cycle-by-cycle gas motion and combustion variations in a spark ignition engine. A 50 micron diameter quartz rod substrate was used with a 0.1 micron film of platinum on the surface. The platinum sensor was coated with a 1 micron thick protective film of quartz. The sensor's heat dissipation, air properties, temperature and velocity were correlated by King's Law (34).

In conclusion, it should also be noted that flame speed measuring techniques have been used to indicate how flow velocities in spark ignition combustion chambers are influenced by large changes in engine operating conditions. No exact measurements are possible however.
The literature survey above indicates that a constant temperature, hot wire anemometry system would be the most suitable for the investigation of the turbulent flow in various types of spark ignition engine combustion chambers. At least, this was true at the time the work was started since it was the most developed system available. A problem that had to be faced at the outset was that of calibration. In particular, calibration procedures for such systems are strictly only valid for a mean flow and may not be applicable to the completely dynamic, random conditions, with no mean flow in many cases, which exist in spark ignition combustion chambers. However, because of the lack of any other suitable detector, having the same frequency response and other desirable characteristics required of a detector, it was decided to proceed with hot wire anemometry. Estimations of the errors introduced from the above source were, in fact, made from the results obtained. Additionally the most detailed and comprehensive calibration techniques were used to eliminate as many inaccuracies as possible.
FIG. 1a — CONSTANT CURRENT ANEMOMETER

FIG. 1b — CONSTANT TEMPERATURE ANEMOMETER
CHAPTER 3

THE CONSTANT TEMPERATURE ANEMOMETRY SYSTEM AND ITS CHARACTERISTICS
CHAPTER 3

THE CONSTANT TEMPERATURE ANEMOMETRY SYSTEM AND ITS CHARACTERISTICS

The development of the constant temperature anemometer has taken place over a period of about 30 years. Consequently, many excellent reviews have been published related to this notably by Corrsin (35) and Kovaszny (36). These will not be repeated here.

The operation of the device has been described briefly in Chapter 2. An understanding of hot wire operation, however, requires a much more detailed knowledge of its behaviour in both steady and unsteady flows and, also, a study of the characteristics of the feedback circuitry under steady and unsteady conditions. Knowing this, the adjustments that have to be made for the anemometer to operate at optimum conditions for a given situation become apparent.

The characteristics of the hot wire in steady flows are discussed in detail in Chapter 4 in connection with the calibration procedures used. At this stage, for the purposes of illustrating certain effects, it is simply necessary to state the general form of the heat loss, $Q$, from a heated wire as a function of the steady flow velocity, $U$. This is

$$Q = A + BU^m$$

in which $A$, $B$ and $m$ are constants.

3.1 The Hot Wire Anemometer in Unsteady Flow

The objective is to determine the response characteristics of the hot wire in a fluctuating flow.

First of all, an open-loop system will be considered in which the current through the sensor is kept constant resulting in a response which is mainly determined by the probe. There are two factors which can
limit the frequency response of the wire under such conditions:

i) the resistance, and, therefore, temperature, fluctuations which inevitably arise in order to generate changes in the heating current by means of the feedback loop (see Fig. 2).

ii) the fluctuations in local temperatures within the hot wire when the flow conditions change.

The first limitation arises because the energy absorbed in or lost from the wire, as the resistance and mean temperature change, introduces a first-order time constant into the system. Its value can be calculated after initially stating the basic equation for the thermal equilibrium of the wire under steady conditions.

This is

\[ I^2 R_w = (R_w - R_0) H_w \]  

(3-2)

where \( H_w \) is an overall heat transfer coefficient for the complete wire, \( I \) is the current in the wire and \( R_w \) and \( R_0 \) the wire resistances during operation and at ambient conditions respectively.

In an unsteady flow, a term has to be included in the heat balance equation (3-2) to express the wire's thermal inertia as a result of the heat stored in the wire. Thus,

\[ C \frac{dR_w}{dt} + (R_w - R_0) H_w = I^2 R_w \]  

(3-3)

It will be noted that the temperatures are expressed in terms of the wire resistances in this analysis according to the relationship

\[ R_w = R_0 (1 + \alpha (T_w - T_0)) \]  

(3-4)

where

- \( T_w \) is the wire operating temperature
- \( T_0 \) is the ambient temperature
- \( \alpha \) is the temperature coefficient of resistance.
Therefore, in Equation (3-3), \( C \) is the modified heat capacity of the wire defined by

\[
C = \rho_w \frac{C_{pw}}{d} \frac{\pi d^2}{4} \frac{L}{R_0} \alpha \tag{3-5}
\]

in which

- \( \rho_w \) = density of wire material
- \( C_{pw} \) = specific heat of wire material
- \( d \) = diameter of hot wire
- \( L \) = length of hot wire.

The quantities \( R_w \), \( H_w \) and \( I \) in Equation 3-3 can be considered as the sum of a time-averaged value (denoted by superscript bar) and a fluctuating or time-dependent part as follows

\[
R_w = \bar{R}_w + R'_w \\
H_w = \bar{H}_w + H'_w \\
I = \bar{I} + I'
\]  
(3-6)

Substituting these into Equation (3-3) and subtracting out the time-average part using Equation (3-2), the following equation is obtained for the resistance variations after neglecting second order fluctuating quantities,

\[
C \frac{d}{dt} \left( R'_w - C \bar{R}_w \right) + R'_w \bar{H}_w = - (\bar{R}_w - R_0) \cdot H_w' + 2 \bar{I}' \bar{R}_w + I^2 R'_w \]  
(3-7)

Dividing throughout by \( R_0 H_w \) and rearranging gives

\[
\left( \tau \frac{d}{dt} + 1 \right) \frac{R'_w}{R_0} = 2 \left( \frac{\bar{R}_w}{R_0} - 1 \right) \frac{I'}{I} - \left( \frac{\bar{R}_w}{R_0} - 1 \right) \frac{H'_w}{H_w} \]  
(3-8)

where \( \tau = \frac{C \bar{R}_w}{\bar{H}_w R_0} \) and is the wire thermal time constant.
Since $H_w$ can be approximated by an expression similar to that in Equation (3-1), $t$ is noted to decrease as the flow velocity, $U$, increases and to be proportional to the heat capacity and resistance ratio of the wire.

The above analysis is necessarily simplified since it assumes that the time-averaged components $\bar{R}_w, \bar{H}_w$ and $\bar{I}$ are all independent of time. This is not strictly true under all transient conditions since it is known (Fig. 3) that the temperature distribution differs for each steady flow. This temperature distribution must take a finite time to change even when the average temperature and resistance are nearly constant. The question, therefore, arises as to precisely what influence the fluctuating local temperatures have on the dynamic response under such conditions. This is the second limitation which was thought to limit the response of the wire in unsteady flow.

A consideration of this problem requires an assumption that the probe is part of a closed-loop anemometer system (see Fig. 2) in which perfect feedback characteristics pertain (i.e. the inductances are neglected). Davis and Davies (37) have studied the steady state voltages around this circuit and have derived the following equation relating resistance and voltage fluctuations:

$$\frac{R'_w}{E'} = \frac{(R_1 + R_w) \left( \frac{1}{G} - \frac{e}{E} \right)}{\overline{E} \left( \frac{R_1}{R_1 + R_w} \right) + E'(\frac{R_2}{R_2 + R_3} - \frac{1}{G})}$$  (3-9)

Analysis of this expression in terms of wire temperature redistribution when the flow changes its speed or direction was not found to affect the heating current response at all since this takes place more slowly than the closed-loop response of the feedback system.
For unsteady cooling above about 1000 Hz, the distribution in temperature does not have time to find a new equilibrium and its mean value is retained. Below 1000 Hz, the temperature distribution is continually changing due to the fluctuating cooling but, because there is no coupling between the redistribution and the feedback system, no limit is placed on the anemometer frequency response from this source.

The time constant of a hot wire probe in an open-loop system is, thus, determined solely by the magnitude of the mean wire resistance and temperature changes. In practice with such a system, fluctuating flow can only be measured up to a frequency of around 1000 Hz depending, in fact, on the parameters influencing the value of time constant $\tau$.

As already stated however, constant temperature anemometry employs a feedback technique to raise the frequency response by a factor of several hundred, dependent on the anemometer's gain, bandwidth and bridge configuration. To estimate the performance of the anemometer in its closed-loop form, the amplifier and the rest of the bridge must be included in the analysis. Andersen (38) has performed such an analysis and has shown that the anemometer's upper frequency limit is increased by a factor, $g$, of

$$g = 2 a R_w G$$

(3-10)

where

$$a = (R_w - R_o) / R_o$$

the overheating ratio.

and $G = \text{amplifier transconductance}$. 

Davies et al. (39) have also evaluated this improvement factor, $g$. From Fig. 2, they determined that the current in the hot wire, $I$, is given by

$$I = -2 R_1 I_v / \left| (1 + A) R_w + (1 - A) R_1 \right|$$

(3-11)
where

\[ I_v \] = current when bridge is perfectly balanced at a standard flow condition.

\[ I \] = current at any other flow condition.

and \[ A \] = gain of amplifier.

Using the relevant relationships of Equation (3-6) in Equation (3-11) and ignoring the second order fluctuating quantities, the fluctuating part of Equation (3-11) becomes

\[
\frac{R_w'}{R_w} = - \frac{1}{G} \left( \frac{\bar{R}_w - R_1}{\bar{R}_w} \right) \frac{I'}{I} \tag{3-12}
\]

where

\[ G = \frac{A}{2} \left( R_1 + \bar{R}_w \right) \] i.e. the transconductance of the bridge.

When Equations (3-12) and (3-8) were combined, the relation

\[
\frac{I'}{I} = \frac{K}{1 + 2K} \frac{R_w'}{R_w} \tag{3-13}
\]

was obtained, in which

\[ K = \frac{\bar{R}_w - R_o}{R_o} \frac{\bar{R}_w G}{1 + G(\bar{R}_w - R_1)} \]

\[ = a. \bar{R}_w G \left( = \frac{g}{2} \right) \text{ from Equation (3-10)} \]

when \[ \bar{R}_w = R_1 \]

Therefore, the hot wire time constant in an open-loop system has been reduced by the factor \((1+2K)\) due to the feedback characteristics of the closed-loop arrangement. This is of the same order as was calculated by Andersen \((38)\) - see Equation (3-10).
Improving the frequency response by increasing the amplifier transconductance is difficult, however, due to the appearance of high frequency oscillations in the amplifier bridge system. Instability of this type is well-known in all feedback systems. The art of constant temperature anemometry is, therefore, based on the design of a stable servo system having high closed-loop gain. The high frequency oscillations have been traced by Davies and Davis\(^{(40)}\) and Freymuth\(^{(41)}\) to the reactances associated with the probes and their leads. The reactances are generated because the current is varying due to the unsteady flow. Noise in the circuit is also important.

Davies and Davis\(^{(40)}\) have considered, in particular, the effect of a series inductance, \(L\), on the dynamic performance of the feedback system (see Fig. 2). The voltage which is fed to the amplifier is now shown to be

\[
e = I(R_w - R_l) + L \frac{dI}{dt}
\]  

(3-14)

The hot wire current equation (3-11) is thus modified to

\[
I - I_v + G I R_w + G L \frac{dI}{dt} - G I R_l = 0
\]  

(3-15)

Again utilizing Equation (3-6) for \(I\) and \(R_w\) and removing the time-averaged parts, we obtain

\[
\frac{R_w'}{R_w} = - \frac{1}{G R_w} + \frac{\bar{R}_w - R_l}{R_w} + \frac{L}{R_w} \frac{d}{dt} I'
\]  

(3-16)

Putting this expression in Equation (3-8) gives the transient system response equation

\[
\frac{d^2 I'}{dt^2} + \frac{1}{L G} \frac{\bar{R}_w - R_l}{R_w} + \frac{1}{L} \frac{d}{dt} \frac{1}{L T G} \frac{(\bar{R}_w - R_l) + 2\bar{R}_w (\bar{R}_w - R_o)}{L \tau R_o} I' + \frac{1}{R_l \tau R_o} \frac{H'_w}{R_w} = 0
\]  

(3-17)
This is a second-order linear, differential equation describing the dynamic behaviour of a system with a natural frequency, $\omega_0$, and damping ratio, $\eta$, defined by

$$\begin{align*}
2\omega_0^2 &= \frac{R_W}{L_T} \left| \frac{1}{G R_W} + \frac{R_W - R_1}{R_W} + \frac{2(R_W - R_0)}{R_0} \right| \\
\eta &= \frac{1}{2\omega_0} \left| \frac{1}{GL} + \frac{(R_W - R_1)}{L} + \frac{1}{\tau} \right|
\end{align*}$$

(3-18) (3-19)

At high gain, the transconductance $G$ is large and since the difference $(R_W - R_1)$ is normally small, the characteristic frequency becomes

$$\omega_0 = \sqrt{\frac{2(R_W - R_0)}{L_T R_0}}$$

(3-20)

For best performance therefore, either the wire temperature must be high or the inductance small. Equation (3-20) also shows that the natural frequency rises with the flow velocity i.e. as $\tau$ is reduced.

The useful bandwidth of the anemometer (i.e. before the system bursts into oscillation and becomes unstable) depends, therefore, on the damping $\eta$. Obviously, when $\eta$ approaches zero, the system becomes unstable. The useful bandwidth has been found to attain its optimum value when $\eta$ is around $0.7$. Equation (3.19) shows that $\eta$ is increased when $R_W$ is increased and is inversely proportional to the amplifier gain. This provides a criterion for fixing the optimum value of the gain. A conflict arises, therefore, at increasing flow speeds (when $\omega_0$ increases) since the anemometer response will be insufficiently damped if the amplifier gain is too high.

In this type of situation, there are two ways in which the anemometer performance can be improved:
i) by reducing the inductance of the probe and leads.

ii) by inserting an adjustable compensating inductor $L_1$ in series with the resistor $R_1$ (in Fig. 2).

The latter method is available on the D.I.S.A 55D01 anemometer used in this work. This increases the bandwidth of the system until the response again becomes limited by higher order effects when two natural frequencies now become apparent. A greater understanding of these higher order effects is necessary before further improvements in the dynamic response of constant temperature anemometers become available.

The relevance of the above discussion to the particular problem of balancing the bridge and setting the amplifier characteristics on the D.I.S.A. 55D01 anemometer for its application to the measurement of flow velocities in engine cylinders, is described in Sections 3.3 and 3.4.

3.2 The Hot Wire Probe

In addition to the basic requirements of a sensor for measuring fluctuating flow velocities over a wide frequency range in an internal combustion engine cylinder which were mentioned in Chapter 2, certain other factors become important when a hot wire probe is specifically used for this purpose. These include:

i) a high temperature coefficient of resistance, $\alpha$, of the wire material being used. This ensures good sensitivity as a small change in temperature produces a large change in resistance.

ii) the lowest possible values of wire diameter, $d$, and length, $l$, to facilitate the best frequency response characteristics (see Equation 3-8 for definition of wire
time constant $\tau$.

iii) short wires to permit measurements being made at a point in a given flow.

iv) good ductility of the wire material to attain low values of wire diameter, $d$.

v) high aspect ratio wires so that end conduction losses are minimized. In connection with this, the supporting prongs of the wire should be made of material of low conductivity.

(N.B. aspect ratio = wire length/wire diameter).

vi) operation of the hot wire at very high wire temperatures to achieve a large difference, relative to the gas temperature, around top dead centre in the engine cylinder. A wire material must, therefore, be chosen which does not oxidize under such conditions. In this connection also, a wire material should be used which maintains its property values constant over relatively long periods at high temperatures.

vii) high tensile strength of the wire.

viii) minimization of inductances of wire, prongs and leads. The significance of this is that the frequency response of the system is not limited by oscillations and instability in the feedback circuit (see Equation 3-20).

ix) no difficulty in fitting a new wire to the prongs.

x) the wire supports or prongs must be rigid so that they do not oscillate in the dynamic flow in the combustion chamber and, in so doing, distort the frequency spectrum results. Additionally, high support strength and rigidity will avoid large stresses being imposed on the wire due to support deflection or vibration.
xi) the measurement of prong tip temperatures. This is especially important at low flow velocities where conduction losses to the supports form a significant part of the total heat loss from the wire.

xii) the properties of the wire material must be available in the literature. These are required for the analytical calibration procedure which has to be used (see Chapter 4).

Such ideal requirements must, of necessity, result in a compromise in the probe design and manufacture. The resulting probe configuration developed is shown in Fig. 4 and was manufactured to specification by D.I.S.A.

Many wire materials commonly used on hot wire probes at ambient temperature and pressure could be immediately discarded. Thus, tungsten, although having excellent mechanical properties, oxidizes at 350°C so that an inadequately low wire temperature could only be used. As a consequence, the choice was immediately reduced to either platinum or one of its many alloys e.g. Pt-10%Rh or Pt-10%Ir. Table 1 compares many of the properties of these materials in addition to those of tungsten. Pure platinum has the best ageing resistance but its alloys have a superior mechanical strength. This was a more important consideration as the wire had to operate in a 'hostile' environment where air impurities and oil contamination of the wire was likely to be high. The Pt-10%Ir alloy has a tendency to oxidize at high wire temperatures and so Pt-10%Rh was used.

The wire diameter of this material had to be a compromise between the required mechanical strength (indicating large d) and the desirability of limiting the thermal inertia to as low a value as possible so as to have a wide frequency response. A value of 10 microns was eventually chosen. The hot wire length was nominally 2.2mm giving an aspect ratio of around 220.
<table>
<thead>
<tr>
<th>MATERIAL PROPERTY</th>
<th>Pt</th>
<th>Pt-10%Rh</th>
<th>Pt-10%Ir</th>
<th>Tungsten</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \alpha ) (%/°C)</td>
<td>0.0039(42)</td>
<td>0.0017(42)</td>
<td>0.00138(43)</td>
<td>0.0045(50)</td>
</tr>
<tr>
<td>Resistivity (micro-ohm cm)</td>
<td>10.5(42)</td>
<td>21.1(42)</td>
<td>24.39(43)</td>
<td>5.5(50)</td>
</tr>
<tr>
<td>( C_p ) (cal/gm °C)</td>
<td>0.0321(45)</td>
<td>0.0358(42)</td>
<td>0.031(43)</td>
<td>0.032(50)</td>
</tr>
<tr>
<td>U,T.S. (tons/in²)</td>
<td>10.2(42)</td>
<td>21.5(42)</td>
<td>36.2(42)</td>
<td>76.5(50)</td>
</tr>
</tbody>
</table>

Maximum Operating Temperature (°C):
- 1000°C
- 800°C
- 750°C
- 350°C

Density (gm/cm³) | 21.5(42) | 19.97(42) | 21.5(43) | 19.3(50) |

Thermal Conductivity (watt/cm°C) | 0.7(45) | 0.302(44) | 0.314(44) | 0.4(50) |

The properties above are those pertaining at ambient conditions (25°C).
A drawback to the use of Pt-10% Rh was noted by Spangenberg (47), who observed that the tensile strength of the material decreased considerably at very high wire temperatures. This, however, only becomes important at flow velocities in the supersonic region.

The prongs (see Fig. 4) are made from nickel-chromium and their finish is very smooth so as not to cause flow interference at the hot wire. At their innermost ends (away from the hot wire and within the probe body), they are soldered to copper leads and these pass through a tube in the probe body to be ultimately connected to the anemometer. The tube also houses thermocouple wires which are also attached to prongs situated close to the hot wire prongs (see Fig. 4). They are sufficiently far away, however, so that the flow near the hot wire is not disturbed by them. 20 micron diameter nickel - nickel chromium wires are attached to these and form a thermocouple bead at one prong tip. It is assumed that the temperature variation along the wire is symmetrical, at least for calibration purposes, so that temperature measurement at one prong tip is sufficient.

The probe was connected via a 5 metre probe cable to the D.I.S.A. 55D01 anemometer. The total resistance of the probe body, prongs and leads (excluding the hot wire) was measured to be 1.50Ω whilst that of the 5 metre probe cable was 0.27Ω.

The greatest care was taken at all times with the probe and, when not in use, the hot wire and prong part was covered by a sheath for protection (Fig. 4).

In hot wire work, it is essential to be able to repair probes easily after breakage. Otherwise, the scope of the work is severely limited due to time delays etc. The availability of suitable spot-welding equipment for this was greatly appreciated (see Acknowledgments). It consisted of a Micromanipulator, comprising a probe holder, a coil holder and wire guide
for the coil of replacement Pt-10% Rh wire, a wire manipulator and welding electrodes. A microscope of enlargement factor 25 was also available. The current pulses for the spot welder were supplied by a generator, which consisted of a series of capacitors that could be charged with a suitable current and then discharged through the spot welding circuit.

The method used in the fitting of a new wire was as follows. The wire and prong tip were held together in the correct position and a current passed through the point of contact. A large temperature rise in this localised region occurred which, if of the right value, melted the metal on either side of the contact surface to a certain depth but not right through. A trial and error method was required to determine the correct current discharge. If this is too low, no welding takes place and, if too high, the prong tip disappears due to melting. Great care is, therefore, needed. A good weld was normally indicated by a puff of smoke accompanying the current discharge. The wire was then stretched over the second prong tip at a suitable tension and the procedure repeated.

An annealing procedure followed the fitting of a new wire. This was necessary because the wire "cold" resistance, $R_o$, was noted to increase after being heated to its operating temperature. A method similar to that used by Spangenberg (47) to stabilize the hot wire was adopted and involved the heating of the hot wire probe in the anemometer circuit over a period of 5 minutes at gradually increasing wire temperatures to the highest at which the probe would be used. Failure to do this resulted in unstable wires and a permanent increase in $R_o$ each time the wire temperature was raised above its previous maximum value.

A photograph of the probe is shown in Fig. 5 installed in part of the probe traverse mechanism.
3.3 Adjustment of D.I.S.A. 55D01 Anemometer for Flow Velocity Measurement in Engine Cylinders

A block diagram showing the bridge configuration and main components of the D.I.S.A. 55D01 constant temperature anemometer is shown in Fig. 6. A Bridge Ratio of 1:20 was used with a resistance $R_1$ of 50$\Omega$ in the bridge arm in series with the probe arm. This meant that the probe resistance decades, which appear on the front panel of the unit, could be used for adjustment of the probe operating resistance. Additionally, bridge balance could be facilitated by the cable compensating inductance and capacitance ($L$ and $Q$) controls (see Fig. 6). This technique is described later. A 1:1 Bridge Ratio was also available.

The wire 'cold' resistance was obtained with little trouble by using a dummy probe in place of the real one to balance out the leads and probe body resistances etc. (using the ZERO OHMS control) before inserting the real probe for its wire resistance measurement. The passage of a 5mA current, by depressing switch TR, facilitates this. With the ZERO OHMS control set in this manner, the resistance decades could be immediately set to the desired wire operating resistance $R_w$.

The anemometer output voltages were measured at the Bridge Top terminal. The Amplifier Output Terminal could also have been used. This is less accurate due to effects of contact resistances in the Loop Control switch. The latter is normally at STD. BY and is set to INT to heat up the probe.

The voltage, $V_B$, at the Bridge Top is directly proportional to the current, $I$, which flows through the probe because the wire resistance $R_w$ is constant. Therefore

$$V_B = I(R_w + R_1 + R_c) \quad (3-21)$$
where
\[ R_1 = 50 \Omega \text{ (the resistance in the bridge arm in series with the probe arm)} \]
and
\[ R_c = \text{total conductor resistance in the leads and probe body (measured to be 1.77\Omega - see Section 3.2).} \]

As a result, the voltage across the hot wire only, \( V_w \), is
\[ V_w = I \cdot R_w \]

This quantity \( V_w \) has to be evaluated for the analytical calibration procedure in Chapter 4.

The gain of the amplifier can be varied in eleven steps from 150 to about 4700.

The analysis in Section 3.1, which took account of the dynamic response of the closed-loop anemometer system, inclusive of inductive effects, showed the importance of balancing the bridge and selecting the amplifier characteristics to attain the optimum dynamic performance. This is especially important in the application of the anemometer to the measurement of flow velocity fluctuations over a wide frequency range (up to 5800 Hz) in engine combustion chambers. It was shown that the natural frequency of the system, \( \omega_0 \), can be greatly increased by inserting a compensating inductor in the bridge circuit. This facility is available on the anemometer used in this work in the form of the Land Q cable compensation controls (see Fig. 6).

To achieve bridge balance, a controlled sudden change in the flow velocity over the hot wire should be used using, for example, a shock tube. Freymuth(41) and Davies and Fisher(27) have indicated that approximately the same effect can be achieved by feeding square-wave electrical disturbances, \( e^i_1 \), into
the amplifier (see Fig. 2) so that small variations in the operating resistance of the wire are produced. These small, sudden changes require a transient impulse of energy to be supplied to the wire to maintain its equilibrium. The general form of the impulse response under such conditions is shown in Fig. 7.

Therefore, small square-wave voltages were injected across the bridge in this manner through Test Signal In Terminal (Fig. 6) and the bridge output monitored on a Tektronix oscilloscope. Since the bridge circuit may have been completely unbalanced initially, the probe was heated up at the lowest amplifier gain setting. Whilst stepwise increasing the amplifier gain and bandwidth to the highest setting, a smooth impulse response function was monitored on the Tektronix 'scope without superimposed damped oscillation which would evidence a peak in the anemometer amplitude characteristic at the frequency of oscillation. This was achieved by adjustment of the Cable Compensation Land Q controls (see Fig. 6).

The type of oscillation shown in Fig. 8a was noted (and removed by Land Q control adjustment) during this procedure. When using inductive compensation techniques to improve the dynamic performance of the system, another type of oscillation becomes apparent (see discussion at end of Section 3.1). This has the form shown in Fig. 8b and occurs at very high flow velocities when the hot wire time constant, \( \tau \), is very low. It is attributed to inadequate amplifier bandwidth. Consequently, the bridge was balanced, in the manner described above, at a high flow velocity of 80 m/s, with the probe inserted in the D.I.S.A. Calibration Wind Tunnel (see Fig. 9). No such oscillation as shown in Fig. 8b was noted, most probably because the probe time constant is inherently of a relatively high value. A check was then made that oscillation of the type in Fig. 8a did not occur at a low flow velocity of 4 m/s. None was found.
Therefore, the bridge was in perfect balance.

3.4 Frequency Response Characteristics

Since the hot wire anemometer is to be used for fluctuating flow velocity measurements up to 5800 Hz, a knowledge of its frequency response is essential before measurements made with it may be interpreted. This should be known not only at ambient temperature and pressure but also at those conditions existing in the "motored" engine combustion chamber during the period at which flame propagation occurs. A detailed analysis was, therefore, undertaken based on both theoretical calculations and experimental measurements.

It was shown in Section 3.1 that the time constant of the hot wire alone is given by

\[ \tau = \frac{C R_w}{H_w R_w} \]

where C is defined by Equation (3-5). Also, Equation (3-2) shows that

\[ H_w = \frac{t^2 R_w}{R_w - R_o} = \frac{V_w^2}{R_w (R_w - R_o)} \]

As an initial exercise to indicate trends, the time constant, \( \tau \), was evaluated at ambient conditions using the expressions above and utilising relevant data from calibration runs made in a D.I.S.A. Calibration Wind Tunnel. These will be described in more detail in Chapter 4. Values of \( V_w \) were calculated from the Bridge Output volts, \( V_B \), (see Fig. 6) as described in Equations (3-21) and (3-22). Under the following conditions:

\[ R_o = 5.76 \Omega \]
\[ d = 0.001 \text{ cm} \]
\[ \ell = 0.2375 \text{ cm} \]
\[ \alpha = 0.001495 \text{ (evaluated in Chap. 4)} \]
the results in Fig. 10 were obtained at the varying wire operating resistances and temperatures shown. The corresponding frequencies at the -3db point (when phase shift is -45°) are also plotted and were calculated from

$$f_o = \frac{1}{2\pi \tau}$$

where $f_o$ is the open-loop frequency response.

The increases in $\tau$ with increasing wire temperature are generally noted in the literature. Such trends are reversed when the loop is closed and feedback is employed, as will be shown below.

Andersen (38) and Davies et al (39) showed that the upper frequency limit, $f_c$, of the probe is increased by a factor of approximately (see Section 3.1).

$$g = 2a R_w G$$

under these conditions. For the circuit used in this work (see Fig. 6),

$$G = \frac{A}{2 (50 + R_w)}$$

The calculated closed-loop time constants, $\tau_c$, and upper frequency limits, $f_c$, at an amplifier gain, A, of 760 are now shown in Fig. 11. This shows the importance of a high wire temperature for good closed-loop response characteristics.

Before pursuing this theoretical analysis any further - especially to firm conclusions about anemometer adjustments etc. for its desired application - the accuracy of the method was gauged by comparison with some experimental measurements. As stated in Section 3.3, step flow velocity changes produced in a shock tube should ideally be used for
This accounts for local temperature variations within the wire as well as for mean temperature changes of the wire. However, approximately the same effects can be achieved by injecting square wave signals into the bridge circuit as already described. Davies and Fisher (27) noted the upper frequency limit to be underestimated using this technique compared with shock tube measurements. An important pre-requisite of the method, to avoid obtaining misleading values of frequency response, is that the bridge has to be in balance.

The Square Wave signals were applied to the Test Signal Input Terminal (see Fig. 6). The resulting output response (Fig. 7) can be analysed in terms of the amplifier bandwidth and system bandwidth. The rise time at the front edge is determined by the amplifier bandwidth whilst the decay time, $\tau_c$, at the rear edge is a function of the system bandwidth. If the decay time, $\tau_c$, is high compared with the rise time, as happens, for example, at relatively low amplifier gain, the system frequency response has amplitude and phase characteristics resembling those of the probe alone but shifted to higher frequencies. The upper limit, $f_c$, occurs at the -3db point as before ($f_c = \frac{1}{2\pi \tau}$) and the response falls off at 6 db per octave above $f_c$.

The responses obtained to the Square Wave Signals were monitored on a Type 555 Dual Beam, Tektronix Oscilloscope where they were photographed for later analysis. Rolleicord film was used with a polaroid camera. Fig. 12 shows a typical oscillogram obtained under the stated conditions. The probe was placed in known airflows for these measurements and various wire operating resistances and amplifier gain settings were used.

The results from this experimental analysis are shown in Table 2 for Amplifier Gains of 390, 760 and 1650. The theoretical evaluations of the time constants,
<table>
<thead>
<tr>
<th>WIRE OPERATING RESISTANCE, $R_w$</th>
<th>CLOSED-LOOP TIME CONSTANTS, $t_c$ (μsec) AT FLOW VELOCITY OF</th>
<th>AMPLIFIER GAIN</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>0 = 2.5 m/s EXP. THEOR.</td>
<td>0 = 2.5 m/s EXP. THEOR.</td>
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<td>24.5 28.7</td>
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<tr>
<th>WIRE OPERATING RESISTANCE, $R_w$</th>
<th>CLOSED-LOOP FREQUENCY RESPONSES, $f_c$ (Hz) AT FLOW VELOCITY OF</th>
<th>AMPLIFIER GAIN</th>
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</table>
\( T_c \) and frequency responses, \( f_c \), are also shown. These were calculated by the method outlined previously under the same operating conditions etc. as were used for the experimental analysis.

It is seen that the calculated frequency response values are underestimated by an average factor of 20%. Thus, in setting the lowest frequency response value to be greater than 5800 Hz (the highest flow velocity fluctuation frequency to be recorded in the engine cylinder) by the theoretical method and, also, at a very low flow velocity — since \( f_c \) increases with this parameter (Fig. 11) — an adequate anemometer response should be achieved.

In practice, however, it is desirable that \( f_c \) be much greater than 5800 Hz because the above results are only valid for flow normal to the wire. At this time, the temperature distribution in the wire is symmetrical and it has been stated in Section 3.1 that temperature distribution effects under such conditions are relatively unimportant. When the flow is yawed at a large angle to the wire, however, as must happen in the engine combustion chamber where eddy flow directions are unknown, the amount of redistribution that occurs within the wire can be considerable. Indeed, Davis and Davies\(^{51}\) calculated the wire temperature distributions shown in Fig. 13 with the probe yawed at angles \( \theta \) to the air flow direction.

To attain the desired frequency response, the lowest amplifier gain possible should be used in order to forestall any possible influence of bridge unbalance and noise (see Section 3.5). Also, the highest wire operating resistance, \( R_w \), should be used. This is compatible with increasing the sensitivity of the low flow velocity measurements in the combustion chamber.

To evaluate the trend of the frequency response
characteristics of the probe-anemometer system at similar temperatures and pressures to those existing in engine cylinders, the analytical calibration of the hot wire probe (described in Chapter 4) was used to provide values of \( V_w \), flow velocity etc. for use in a theoretical analysis. At the following gas temperatures and pressures:

\[
i) \quad T_g = 323 \, ^\circ\text{K} \quad P = 6.8 \, \text{ATM} \\
ii) \quad T_g = 373 \, ^\circ\text{K} \quad P = 10.2 \, \text{ATM} \\
iii) \quad T_g = 423 \, ^\circ\text{K} \quad P = 13.6 \, \text{ATM}
\]

corresponding values of \( \tau \), and \( \tau_c \) were determined.

Figs. 14 and 15 show the results obtained. It is seen that both the probe and anemometer closed-loop system frequency responses are greatly increased compared with ambient conditions.

Consequently, the setting of the anemometer at an Amplifier Gain of 165 and with \( R \) at least 100 is more than adequate to measure flow frequency fluctuations up to 5800 Hz (see Table 2).

3.5 Noise and Signal/Noise Ratios of Anemometer

It is well-known that the one-dimensional energy spectrum distribution of longitudinal velocity fluctuations in a turbulent flow decreases with increasing frequency \( (15) \). Consequently, it was thought that the signal/noise ratios may also decrease severely and obscure the turbulence measurements in the high frequency region of the flow to be investigated in engine combustion chambers.

Freymuth \( (49) \) has analysed anemometer noise in detail from a theoretical viewpoint. He assumes that the bridge and amplifier noise can be represented by an equivalent resistance emitting white noise. His equation is
\[ \Delta U_N = (1 + \frac{1}{m}) (\frac{2}{1+m}) (\frac{R_o}{R_w - R_o} + \frac{2}{1+m}) (1 + \frac{\omega^2 \cdot c^2}{a^2 \cdot r^2}) e_t \]

\[ \ldots \quad (3-23) \]

where \( \Delta U_N \) = anemometer output noise
\( r = f(U, R_w, R_o, m) \)
\( m = \) ratio between resistances in the bridge
\( \omega = 2\pi f \) (rad/sec).

The quantity \( e_t \) is the equivalent noise of the bridge and amplifier at the amplifier input. For the D.I.S.A. 55D01 anemometer, it is specified as

\[ e_t = N_w \left( 1 + \frac{500}{f} \right)^{1/2} \]

where the 'white' noise spectrum, \( N_w \), is given by

\[ N_w = 15 \text{ nV/Hz} \]

It is much simpler and accurate, however, to measure anemometer noise. In this sense, this can be defined as that part of the anemometer output signal that is not related to the turbulence of the fluid flow. An analysis was conducted, therefore, with two aims in view:

i) the experimental measurement of anemometer noise at flow velocities, wire operating temperatures and amplifier gains similar to those at which the combustion chamber results are to be obtained.

ii) the determination of signal/noise ratios under such conditions to ensure that these are adequate.

The first aim was achieved with the probe placed in a laminar flow rig (described in Chap. 4) so that the
noise measurements were not obscured by any turbulence in the flow. A Muirhead K-134-A Wave Analyser was connected to the Bridge Top output of the anemometer (see Fig. 6) and the noise spectrum over the frequency range up to 31000 Hz was determined. The results obtained are shown in Figs. 16 and 17 under the stated conditions. The scale is in millivolts to indicate the low order of noise existing compared with the anemometer output signal. It is seen that the D.I.S.A. 55D01 anemometer noise is characterized by two distinct frequency regions:

a) one, below about 500 Hz, where the noise is composed almost entirely of harmonics of the mains frequency.

b) the other, above 500 Hz, where the noise rises continually with frequency. This is significant because flow fluctuations are to be measured in engine cylinders up to a frequency of 5800 Hz.

The effect of high Amplifier gain is immediately apparent. Also, in the frequency range up to about 10000 Hz, it is desirable to use the highest wire temperatures for low noise. A more detailed observation of Figs. 16 and 17—obtained by cross-plotting—showed that anemometer noise is reduced as the flow velocity increases up to 7500 Hz, above which frequency the reverse trend is noted.

The technique used to evaluate signal/noise ratios was based on considerations of the effect on the voltage at the hot wire, \( V_w \), of a small flow velocity fluctuation \( \Delta U \) at the hot wire probe. From the thermal equilibrium equation for the hot wire, Equation (3-2), it can be shown that

\[
\Delta V_w = \frac{(R_w - R_0) R_w}{2V_w} \cdot \frac{dH_w}{dU} \cdot \Delta U
\]

(3-24)
Calculations of $\Delta V_w$ (and hence of $\Delta V_B$) were performed at conditions under which the noise measurements were made—in particular, when the flow velocity was 4.8 m/s. At a low flow fluctuation value of 5 cm/s and using the noise measurements of Fig. 17, the signal/noise ratios in Table 3 were obtained. These decrease with increasing frequency and amplifier gain and increase with $R_w$. Overall, however, in the frequency range of interest in this work, they are of high enough values.

Such noise considerations are, therefore, of secondary importance in setting the anemometer compared with those of frequency response.

3.6 Anemometer Drift

Drift in the anemometer output voltage assumes importance in this work since it can greatly upset the calibration of the probe at the low flow velocities envisaged in the combustion chamber. Two causes of drift are normally conceived of due to

a) particulate matter on the wire causing changes in the wire "cold" resistance

b) temperature changes within the instrument causing varying contact resistances in switches etc.

The first source does not apply here and is dealt with in Chapter 4.

Drift due to b) above was minimized by letting the anemometer warm-up for at least 2 hrs. before any measurements were taken.

A further cause of drift can be traced to the bridge circuit and is due to the dissimilar heating of the resistances there. Little can be done about this.
### TABLE 3
#### SIGNAL/NOISE RATIOS

<table>
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<tr>
<th>FREQUENCY (Hz)</th>
<th>( R_w = 7.44 \Omega )</th>
<th>( R_w = 8.71 \Omega )</th>
<th>( R_w = 9.98 \Omega )</th>
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<td>10000</td>
<td>48</td>
<td>26</td>
<td>72</td>
<td>42</td>
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</table>

A.G. = Amplifier Gain
FIG. 2 — CONSTANT RESISTANCE ANEMOMETER FEEDBACK CIRCUIT
Tungsten wire
l=0.22cm
d=0.0005cm

Wire in vacuum

Air velocity
150 m/s
80 m/s
30 m/s
7 m/s

Figure 3—Temperature distribution on a hot wire (Ref. 40)
FIG. 4 — HOT WIRE PROBE WITH THERMOCOUPLE
FIG. 5 — HOT WIRE PROBE IN PART OF PROBE TRAVERSE MECHANISM
FIG. 6 — D.I.S.A. 55D01 ANEMOMETER BRIDGE & FEEDBACK CIRCUITRY
1. step change in direct applied power

2. system impulse response to maintain wire equilibrium

3. resultant output signal

FIG. 7 — ANEMOMETER RESPONSE TO SQUARE WAVE TEST SIGNAL

FIG. 8 — OSCILLATIONS DURING BRIDGE BALANCING
FIG. 9 - HOT WIRE PROBE IN D.I.S.A. WIND TUNNEL

$R_W = 10.5 \text{ Ohms}$

Zero Flow Velocity

Amplifier Gain = 390

FIG. 12 - PROBE FREQUENCY RESPONSE DETERMINATIONS
FIG.10 — HOT WIRE FREQUENCY RESPONSE CHARACTERISTICS

d = 0.001 cm
l = 0.238 cm
Amplifier Gain = 760
\(d = 0.001\text{cm}\)
\(l = 0.238\text{cm}\)

\(T_w = 960^\circ\text{K}\)
\(T_w = 843^\circ\text{K}\)
\(T_w = 437^\circ\text{K}\)
\(T_w = 611^\circ\text{K}\)

Flow Velocity (m/s)

FIG.11—HOT WIRE PROBE FREQUENCY RESPONSE CHARACTERISTICS
d = 0.0005 cm
l = 0.17 cm
Flow Velocity = 5.2 m/s

$\tau = \frac{I - 54}{237}$

$\theta = \text{Yaw Angle}$

**FIG. 13 -- WIRE TEMPERATURE DISTRIBUTION IN YAWED FLOW (REF. 37)**
FIG. 14—EFFECT OF AMBIENT CONDITIONS ON HOT WIRE RESPONSE

- $d = 0.001\, \text{cm}$
- $l = 0.238\, \text{cm}$

Wire Temperature = $843^\circ\text{K}$

$T_g = 293^\circ\text{K}$ : $P = 1\, \text{Atm.}$

$T_g = 323^\circ\text{K}$ : $P = 6.8\, \text{Atm.}$

$T_g = 373^\circ\text{K}$ : $P = 10.2\, \text{Atm.}$

$T_g = 423^\circ\text{K}$ : $P = 13.6\, \text{Atm}$
**FIG. 15 — EFFECT OF AMBIENT CONDITIONS ON HOT WIRE PROBE RESPONSE**

- Wire Temperature = 843°K
- Amplifier Gain = 760
- $d = 0.001 \text{ cm}$
- $l = 0.238 \text{ cm}$
- $T_g = 293°K : P = 1 \text{ Atm.}$
- $T_g = 323°K : P = 6.8 \text{ Atm.}$
- $T_g = 373°K : P = 10.2 \text{ Atm.}$
- $T_g = 423°K : P = 13.6 \text{ Atm.}$
FIG. 16 — ANEMOMETER NOISE MEASUREMENTS

l = 2.034 mm
d = 0.001 cm

flow velocity = 1.90 m/s

- $R_w = 7.44 \, \Omega$ amplifier gain = 760
- $R_w = 7.44 \, \Omega$ amplifier gain = 1650
- $R_w = 8.71 \, \Omega$ amplifier gain = 760
- $R_w = 8.71 \, \Omega$ amplifier gain = 1650
l = 2.034 mm
d = 0.001 cm

Flow velocity = 4.8 m/s

- $R_W = 7.44 \, \Omega$ amplifier gain = 760
- $R_W = 7.44 \, \Omega$ amplifier gain = 1650
- $R_W = 8.71 \, \Omega$ amplifier gain = 760
- $R_W = 8.71 \, \Omega$ amplifier gain = 1650
- $R_W = 9.98 \, \Omega$ amplifier gain = 760
- $R_W = 9.98 \, \Omega$ amplifier gain = 1650

**FIG. 17 — ANEMOMETER NOISE MEASUREMENTS**
CHAPTER 4

HOT WIRE PROBE CALIBRATION
CHAPTER 4
HOT WIRE PROBE CALIBRATION

Flow velocity measurements in "motored" engine cylinders are complicated by the fact that not only does the velocity change but also the pressures and temperatures. Consequently, standard hot wire calibrations at ambient temperature and pressure are of no use.

Semenov\(^{(3)}\) overcame this problem by reducing the current flowing through the anemometer bridge circuit, \(I\), at temperature and pressure conditions different from ambient, to the bridge current that flows at ambient conditions, \(I_0\). The flow velocity, \(U\), was then evaluated from a calibration of \(I_0\) against \(U\) at ambient temperature and pressure. The equation

\[ I_0 = K(\rho, T) \cdot I \]

was used, where \(K(\rho, T)\) is the coefficient of reduction to initial conditions and is a function of density, \(\rho\), and temperature, \(T\). It was determined from the following simple heat balance on the wire

\[ I^2 R_w = A_s h(T_w - T_g) \]

with the heat transfer coefficient, \(h\), evaluated from a relationship of the form

\[ Nu = b \cdot Re^m \]

In the above expressions, \(A_s\) is the wire surface area, \(T_g\) the gas temperature and \(b\) and \(m\) are unspecified constants. Expressing the density in terms of a filling coefficient, \(\eta\), \(K(\rho, T)\) was eventually shown to be
\[ K(p,T) = \left| \frac{1}{\eta} \cdot \left( \frac{T}{T_0} \right)^{n-1} \right|^{m/2} \cdot \left( \frac{\lambda_o}{\lambda_g} \right)^{1/2} \cdot \left( \frac{\mu}{\mu_o} \right)^{m/2} \]

where

- \( n \) is the polytropic index of compression
- \( \lambda \) is the fluid thermal conductivity
- \( \mu \) is the fluid viscosity

and the subscripts \( o \) and \( g \) refer to conditions at ambient temperature and the higher gas temperature respectively.

Hassan\(^{(26)}\) used a technique based on the work of Davies and Fisher\(^{(27)}\). A comprehensive heat balance was generated for the hot wire with the convective heat transfer coefficient given by

\[ h = \frac{C_f \rho U C_v \lambda_w}{\pi \lambda_g} \]  
\[(4-1)\]

in which

- \( \rho \) = air density at the free-stream temperature
- \( C_v \) = specific heat at constant volume of the fluid approaching the wire
- \( \lambda_w, \lambda_g \) = air thermal conductivity at the hot wire temperature, \( T_w \), and at the freestream temperature respectively

and

\[ C_f = \text{skin friction coefficient defined by} \]
\[ C_f = 1.4 \text{ Re}^{-0.5} \text{ when } 40 < \text{ Re } < 1000 \text{ and } \]
\[ C_f = 2.6 \text{ Re}^{-0.666} \text{ when } 0 < \text{ Re } < 50 \]

Equation (4-1) was obtained\(^{(27)}\) from an analogy of the transfer of heat and the transfer of momentum at the hot wire. A knowledge of the gas properties and the physical and electrical properties of the sensing material is needed for this type of approach.

Horvatin and Hussmann\(^{(28)}\) used a similar method but expressed the convective heat transfer by means of
the correlation of Grigull (29) viz

\[ \text{Nu} = 0.43 + 0.48 \text{Re}^{0.5} \]  \hspace{1cm} (4-2)

with the air property values calculated at the freestream temperature.

Another technique that could be used requires two exactly similar hot-wires operated at two different overheating ratios, \( \frac{T_w - T_o}{T_o} \). This will remove the effects of the fluctuating temperatures and pressures so that the hot wire will respond to the velocity fluctuations only.

Finally, since the probe is in a compressible flow in the combustion chamber, a compressible flow equation could be used e.g.

\[ \text{Nu} = a_1 f_1(\text{Ma}) + a_2 f_2(\text{Ma}) \cdot \text{Re}^m \]

where \( f_1(\text{Ma}) \) and \( f_2(\text{Ma}) \) are functions of the mach number \( \text{Ma} \). Hassan (26) has shown, however, that two different mach numbers can be obtained for a given Nusselt Number, depending on whether variations in velocity or density are considered. In this connection, Spangenberg (47) has studied the relationships between \( \text{Nu} \) and \( \text{Re} \) at varying \( \text{Ma} \) but only in transonic and supersonic flows.

The basic approach used in this work to effect a calibration was, therefore, to set up a heat balance on the hot wire to include convective heat losses, end conduction losses to the prongs and radiation losses and to consider the effects of the varying gas pressure, as well as the gas temperature, on the air property values.

4.1 Heat Balance of the Hot Wire

The model considered is shown in Fig. 18. Initially, a small element of the wire was studied in
terms of its heat generation and heat losses in a steady air flow. After then making certain assumptions, the analysis was extended to the complete wire.

Referring to Fig. 18, a wire element of length \( dx \) is seen to receive heat, \( Q_e \), electrically from the anemometer system. There are two sources of heat generation in the element:

i) due to the resistance, \( R_L \), offered by the element to the current flow. This has the value \( I^2 R_L \).

ii) as a result of the current flow with and against the temperature gradients. Such gradients exist in hot wires because the aspect ratios are of relatively small magnitude thereby facilitating a heat loss by conduction to the prongs. Small positive and negative components exist because of this of value \( \psi I (dT_L/dx) \). This mode of heat generation is termed the Thomson effect and \( \psi \) is known as the Thomson coefficient. The net thermal output from this source is zero, however, when the two ends of the wire are at the same temperature, as they are assumed to be in this steady flow calibration. \( T_L \) is the element temperature.

The element of wire, \( dx \), in Fig. 18 loses heat by the following processes:

i) by convective heat transfer to the normal air flow, \( Q_h \).

ii) by conduction along the wire, \( Q_c \).

iii) by radiation to the surroundings, \( Q_r \).

In setting up the heat balance for the wire element, the following assumptions were made:
a) the radial temperature variation in the wire is small
b) the wire element surface temperature is constant
c) the element resistance, $R_L$, and, hence, temperature, $T_L$, is constant along its length

The balance can then be expressed by

$$I^2 R_L = A_x h (T_L - T_g) + A_x \frac{d}{dx} (\lambda_L \cdot T_L) + A_x C_T (T_L^4 - T_s^4) \sigma$$

.... (4-3)

where

$A_x = \pi d \cdot dx$ (element surface area)
$A = \pi d^2 / 4$ (wire cross-sectional area)
$\lambda_L =$ thermal conductivity of wire material at temperature $T_L$
$\sigma =$ Stefan-Boltzmann constant
$(1.36 \times 10^{-12} \text{ cal/cm}^2 \text{ sec} \cdot \text{°K}^4)$
$C_T =$ surface emissivity of wire
$T_s =$ temperature of the surroundings

In expanding Equation (4-3) to include the complete hot wire of length $l$, certain further assumptions were made:

i) the average wire operating resistance, $R_w$, and temperature, $T_w$, are constant along the wire length. This is only approximately true (see Fig. 3) but the calculations are greatly simplified if it is assumed so.

ii) the thermal conductivity of the wire material is constant along its length

iii) heat conduction losses to the 2 prongs occur at both ends of the wire and are of equal magnitude.
The heat balance for the entire hot wire then becomes

\[ I^2R_w = A_s \cdot h(T_w - T_g) + 2\lambda_H A\left(\frac{dT_w}{dx}\right)_{x=0,t} + A_s \cdot \varepsilon_T (T_w^4 - T_s^4) \sigma \]  

(4-4)

where

- \( A_s = \pi dt \)
- \( \lambda_H \) = thermal conductivity of the wire material at temperature of the prong tip, \( T_H \).

4.2 Convective Heat Transfer from a hot wire in steady flow

The initial work on heat transfer from electrically heated wires was conducted by King\(^{(34)}\). In fact, he was first to use such a device as an anemometer in 1914. He formulated the relationship for the heat loss from the wire

\[ Q = \lambda_o 8 \left[ 1 + \left(2\pi \rho C_p \frac{dU}{\lambda_o}\right)^{1/2} \right] (T_w - T_o) \]  

(4-5)

from measurements on large aspect ratio wires in an incompressible potential flow. Recent work, however, has revealed flaws in its accuracy and improved expressions have been developed. Its basic form is still largely maintained nevertheless.

Many other more recent empirical relations are now available for the variation of Nusselt Number with Reynolds Number and in some instances, Prandtl Number, \( \text{Pr} \). Some of these are listed below:

\[
\begin{align*}
\text{Nu} & = .42 \Pr^{.2} + .57 \Pr^{.33} \Re^{.5} & \text{Kramers} \quad (51) \\
\text{Nu} & = 0.75 \Pr^{.2} + .67 \Pr^{0.9} \Re^{.5} & \text{Bourne et al} \quad (52) \\
\text{Nu} & = c \left| \text{Re} \left(\frac{T_w}{T_g}\right)^{0.25} \right| \text{m} & \text{Hilpert} \quad (53) \\
\text{Nu} & = (a + b \Re^{m}) (\frac{T_f}{T_g})^{0.17} & \text{Collis and Williams} \quad (4)
\end{align*}
\]
Also, Equations (4-1) and (4-2), due to Davies and Fisher (27) and Grigull (29), are available.

Initially, the Davies and Fisher expression was tried in this work. The flow velocity calibration obtained in the low flow range of interest (0 - 60 m/s) was not at all accurate however. This was probably due to the two following reasons:

a) the skin friction coefficient expression (see Equation (4-1)) for Re.Nos between 0 and 50 is in fact only valid above Reynolds Numbers "not much less than 10" (27). In this work, it is extremely important to maintain accuracy down to a Re.No of 0.15. This corresponds to an 80 cm/sec flow at ambient conditions

b) the inadequacy of the expression at very high wire temperature operation without the use of a correction factor. This drawback has been shown by Hassan and Dent. (54)

An alternative correlation was sought and the Collis and Williams (4) relationship ultimately used. This is of the form

\[ T_f - 0.17 \quad \text{Nu}. \left( \frac{T_f}{T_g} \right) = a + b \cdot \text{Re}^m \]  

(4-6)

where the constants a, b and m are functions of the Reynolds Number (based on wire diameter, d) as follows

\[
\begin{align*}
0.02 < \text{Re} < 44 & \quad 44 < \text{Re} < 140 \\
m & 0.45 & 0.51 \\
a & 0.24 & - \\
b & 0.56 & 0.48
\end{align*}
\]

In this work, only the lower Re.No. range is of interest. A feature of the relationship is that all the fluid properties are evaluated at a mean film temperature, \(T_f\), given by
There is some debate as to the validity of this since many authors of papers in this field insist that all such properties should be evaluated at the free-stream temperature, $T_g$. Excellent calibrations were obtained, however, with the Collis and Williams formulation. The over-riding consideration which ensured that it was used was that it was derived specifically after experiments aimed at establishing "precise heat transfer laws for forced convection at low Reynolds Numbers". Additionally, end conduction losses were minimized during the experiments by the use of very high aspect ratio wires. Indeed, at flow velocities less than 2m/s (where end conduction losses become extremely important in hot wire work), a wire specimen of aspect ratio 5370 was used. The expression is also directly applicable to high wire operating temperatures through the temperature loading term $(T_f/T_g)^{-0.17}$.

Although very low flow velocities are being considered in the calibration (down to about 80cm/sec), this is still above the criterion \(^{(4)(27)}\) where free convection and buoyancy effects become important, i.e., at $Re < 0.1$ (= 50 cm/sec) at ambient temperature and pressure.

When using Equation (4-6), it is vital that the same expression for the air thermal conductivity, $\lambda_g$, be used as was used by Collis and Williams\(^{(4)}\) in their derivation. This is because different reference sources give differing $\lambda_g$ values. Fig. 19, for example, compares the expression used by Collis and Williams viz.

\[ \lambda_g = 2.41 \times 10^6 (1 + .00317t - .0000021t^2) \text{cal/cmsec}^\circ K \]

(from Kannuluik and Carman\(^{(55)}\))

\[ (4-6a) \]
where
\[ t = (T_f - 273.) \]

with that of Sutherland \(^{(56)}\) viz.

\[ \lambda_g = \frac{6.28}{(T_f + 170.)} \cdot \left( \frac{T_f}{273.} \right)^{1.5} \text{ cal/cm sec } \circ K \]

Although the differences are numerically small, they were found to have a pronounced effect on the calibration since

\[ U \alpha \left( \frac{1}{\lambda} \right) g \]

If the Kannuluik and Carman \(^{(55)}\) expression is not used, Collis and Williams \(^{(4)}\) indicate that the temperature loading term must be modified accordingly

### 4.3 Conductive Heat Losses

An accurate evaluation of the end conductive losses is imperative at low flow velocities (see Fig. 30). Equation (4-4) shows that this requires a knowledge of the temperature gradient at the ends of the hot wire. It is known that this varies with flow velocity in the manner shown in Fig. 3. Ideally, therefore, this must be taken into account.

A method suggested by Horvat\(^{1}\)n and Hussmann \(^{(28)}\) makes the assumption that the temperature distribution over the hot wire length is parabolic in shape at all times with the integral, over the wire length, equal to \( k(T_w - T_H) \), where \( T_H \) is the prong tip temperature. The temperature gradient according to this method is

\[ \frac{dT}{dx} = \frac{6}{k} (T_w - T_H) \]

\( (4-7) \)

At a constant mean wire temperature, this only changes when \( T_H \) changes. Measurements made during the
experimental calibration runs (described later) showed that $T_H$ varied with flow velocity in the manner shown in Fig. 20. From this, $dT/dx$ is seen to decrease as the flow velocity increases. This is contrary to noted observations (see Fig. 3). Consequently, Equation (4-7) was of no use in accurately evaluating temperature gradients.

In this respect, the approach of Davies and Fisher (27) was much more promising. The temperature gradient by their method is given by

$$\frac{dT}{dx} \bigg|_{x=0, l} = \frac{K_2}{\sqrt{K_1}} \cdot \tanh \left( \frac{\sqrt{|K_1|}}{2} \right)$$  \hspace{1cm} (4-8)

where

$$K_2 = \frac{I_R^2}{\lambda_m A L}$$  \hspace{1cm} (4-9)

and $\lambda_m$ = thermal conductivity of wire material at $T_w$.

Equation (4-8) was obtained by differentiation of a unique solution for the temperature distribution in a hot wire which, in turn, was evaluated from the thermal equilibrium equation for the hot wire in differential form.

The quantity $K_1$ could not be calculated directly since it is a function of $Nu/Nu_T$, i.e., of the ratio between the Nusselt No. for the convective heat loss alone, $Nu$, and the Nusselt No. measured directly from the total electric power supplied to the wire, $Nu_T$. This ratio is given by (27).

$$\frac{Nu}{Nu_T} = Nu_r = 1 - \frac{R}{R_w} \cdot \frac{\lambda_H}{\lambda_m} \cdot \tanh \left( \frac{\sqrt{|K_1|}}{2} \right)$$  \hspace{1cm} (4-10)
where

\[ \lambda_H = \text{thermal conductivity of the wire material} \]
\[ \text{at the prong tip temperature, } T_H. \]

However, \(|K_1|\) is also given by

\[
|K_1| = \frac{4 \lambda_w}{\lambda_m d^2} \cdot \nu_T \left| \nu_r - \frac{R_w - R_o}{R_w} \right|
\]  

(4-11)

Knowing that

\[
\nu_T = \frac{I^2 R_w}{\pi \lambda_w T_w^3}
\]

Equations (4-10) and (4-11) can be solved by successive approximation to obtain a unique value of \(|K_1|\). This value of \(|K_1|\) can then be substituted in Equation (4-8) to give the temperature gradient at the wire ends.

With this approach, \(dT/dx\) was found to increase with flow velocity and was considered to accurately depict conditions at the wire ends. It was, therefore, used in the calibration techniques in this work.

A problem arose, however, in the calculation of the conductive heat losses since no information could be found, after an extensive literature search, on the relationship between the thermal conductivity of Pt-10% Rh wire and wire temperature over a wide temperature range. The only values found are listed below from Ref. 44.

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>(\lambda_m) (watts/cm °C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>17</td>
<td>.302</td>
</tr>
<tr>
<td>100</td>
<td>.306</td>
</tr>
</tbody>
</table>

An analysis was performed in which \(\lambda_m\) values were varied over a wide range at high wire operating temperatures \(T_w\). The low flow velocity region was particularly studied. However, the calibration remained relatively unaffected. Medium values of \(\lambda_m\) were, thereafter, used between a value of .306 and a linear
increase in $\lambda_m$ with temperature based on the above values.

The requirement that the thermal conductivity of the wire material be evaluated at the temperature of the prong tip, $\lambda_H$, was facilitated by the presence of the thermocouple bead at this point (see Section 3.2). The thermocouple was calibrated in an oven after having been positioned in a special draught-free container. Oven temperature was measured by the oven thermometer and also, as a check, by a separate thermometer. The prong thermocouple temperature was recorded on a Comark pyrometer (described in Chapter 5) which was self-compensating over the ambient temperature range 0 - 40°C. The results of the calibration are shown in Fig. 21.

A further point arose in considerations of the conductive heat losses. Equation (4-4) indicates that the cross-sectional area, $A$, of the wire material should be used in this connection. However, examinations of the welding joint between the wire and the prong tip under a high-powered microscope indicated that the contact area was greater than $A$. Its exact value, $A^*$, was difficult to determine but estimations were made for it as a function of prong tip diameter, $d_p$, each time a new wire was fitted to the probe. This shows the necessity for individual calibration of hot wire probes. The incorporation of this corrected value for $A$ in the calibration calculations was henceforth used in Equation (4-4).

4.4 Radiation Losses

Because the hot wire probe was going to be used to measure quite low flow velocities in engine combustion chambers and was also to be operated at very high wire temperatures, the radiation loss term in Equation (4-4) was maintained. Normally, it is in order to ignore this quantity as has been done in all
such previous analytical calibrations(26)(27)(28).

In a strict analysis of radiation considerations under these circumstances, allowances should be made for

i) radiation loss between the hot wire and the surrounding solid surfaces

ii) a heat exchange between the hot wire and the surrounding gaseous medium

iii) reflection effects between an individual section of the wire and the rest of the wire

iv) decaying reflection effects.

However, the first factor (i) above) is normally accountable for by far the greatest part of the total radiation loss and was only studied here. Its magnitude can be evaluated from the last term in Equation (4-4).

A literature survey failed to reveal the emissivity, $\varepsilon_T$, of Pt-10% Rh wire in the temperature range of interest, i.e., up to 1000 °C. However, such values were found for pure platinum and pure rhodium(45) and, also, for Pt-13% Rh in the temperature range 1450-2050 °K(61). An average value of $\varepsilon_T = 0.1$ was determined as being probable for Pt-10% Rh. Little error would at any rate be involved in the flow velocity calculations even if this value of $\varepsilon_T$ was greatly in error.

By substitution of

i) the heat transfer coefficient from Equation (4-6)

ii) the temperature gradient from Equation (4-8) and other considerations used in the determination of conductive heat losses (Section 4.3)

iii) radiation loss factors (Section 4.4)

the heat balance equation (4.4) was rearranged to give
the flow velocity $U$ directly, i.e.,

$$
U = \frac{V^2 - 2\lambda \frac{dT}{dx} R - 0.24\pi \frac{\lambda}{\rho \sigma} \left( \frac{R}{\rho} \right) \left( \frac{T_f}{T} \right) - A^2 \sigma \epsilon (T_f - T) R}{2.222}
$$

$$
0.56\pi \frac{\lambda}{\rho \sigma} R \left( \frac{T_f}{T} \right) \left( \frac{T_f}{T} \right) 0.17
$$

$x = \frac{u}{\rho d}$

(4-12)

4.5 Physical Properties and Dimensions of the Wire Material

Equation (4-12) shows that it is extremely important to know the wire length, $l$, and wire diameter, $d$, accurately.

The nominal wire diameter of the Pt-10% Rh wire used was 10 microns. However, it is known that tolerances of approximately $\pm 2\%$ are allowable for such thin drawn wires. Even if attempts are made to measure the wire diameter directly using, for example, an electron microscope, errors of the order of $\pm 2\%$ are still present. Additionally, it is highly likely that slight variations occur in the wire diameter along its length. Consequently, it appeared a good approximation to accept the manufacturer's wire nominal diameter of 10 microns.

The wire length, $l$, and prong tip diameter, $d_p$, were gauged with excellent accuracy, to tenths of a micron, using a Universal Measuring Machine (Type Mu-214B) manufactured by the Société Genevoise d'Instruments de Physique of Switzerland. Typical dimensions of the hot wire and prong tips, along with a plan view of the welds at the wire ends, are shown in Fig. 22. This indicates that the effective wire length involved in forced convection heat loss, without interference effects appearing from the prong tips, is slightly less than the true overall wire length by a factor of approximately 5%. However, since forced
Convection heat loss also takes place from the ends of the wire which lay across the prong tips, an accurate evaluation of this effect is extremely difficult. At any rate, excellent calibrations were obtained by assuming that the entire wire length, to the ends of the welds, is involved in the convective heat loss deduced from Equation (4-6).

The wire temperature, $T_w$, was determined from a knowledge of $R_w$, $R_0$ and the temperature coefficient of resistance, $\alpha$. A literature search revealed that there were wide discrepancies among the various sources on the value of $\alpha$ for Pt-10% Rh wire. Table 4 shows the variations found. Consequently, $\alpha$ was established for the particular wire used in this work. This was done by heating the probe wire in an oven over a period of 4 hours and the technique described in Section 3.3 used to measure the wire resistance at various temperature intervals up to 800 °C. Recordings were also made during cooling. The results obtained could be correlated by the relation

$$R_w = R_0 (1 + \alpha(T_w - T_0)) \quad (4-13)$$

A higher order form of this equation was found to be unnecessary. The temperature coefficient of resistance, $\alpha$, was thus obtained by plotting

$$\left(\frac{R_w}{R_0} - 1\right) \quad \text{against} \quad (T_w - T_o)$$

Fig. 23 illustrates this and the slope of the straight line obtained is $\alpha$ which equals 0.001495.

For the measurements above, the hot wire was placed in a container to eliminate any effects of draughts and convection currents. No permanent increase in wire resistance was noted after cooling as was observed by Spangenberg (47) since the wire had already been annealed before being placed in the oven.
<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>Temp. Coeff. of Resistance, α, (°C)</th>
<th>Resistivity (micro-ohm cm)</th>
<th>Reference Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>0 - 100</td>
<td>.0017</td>
<td>21.1</td>
<td>(42)</td>
</tr>
<tr>
<td>20</td>
<td>.00135</td>
<td>22.45</td>
<td>(58)</td>
</tr>
<tr>
<td>0 - 100</td>
<td>.00147</td>
<td>21.853</td>
<td>(57)</td>
</tr>
<tr>
<td>0</td>
<td>.00143</td>
<td>21.14</td>
<td>(60)</td>
</tr>
<tr>
<td>16.15</td>
<td>.00143</td>
<td>21.635</td>
<td>(60)</td>
</tr>
<tr>
<td>92.25</td>
<td>.00143</td>
<td>23.943</td>
<td>(60)</td>
</tr>
</tbody>
</table>
At all times, extreme care was taken in the calculation of the probe "cold" resistance $R_0$ at ambient temperature, $T_0$. This was because of the decisive influence even slight variations in $R_0$, (caused for example by anemometer drift (Section 3.6), ageing of the wire, minute increases in wire length at high flow velocities, particulate and oil contamination of the wire) have on the wire mean temperature, $T_w$. The error introduced from this source was found to increase as $T_w$ decreased.

It should be noted that wire resistance measurements normally also include the contact resistance of the welds attaching the wire to the prongs. To allow for these, one could accurately measure $l$ and the fluctuations in $d$ along the wire and then calculate the wire resistance, $R$, using the formula

$$ R = \frac{\rho \times l}{A} \quad (4-14) $$

However, Table 4 also shows the wide discrepancies found in the value of $\rho_r$, the resistivity of Pt-10%Rh. Therefore, allowances for the welds' contact resistances are likely to be completely unreliable. They were neglected in this analysis as a result since, at any rate, they are generally considered to be negligible.

Because of the noted variations in $\rho_r$ (see Table 4), this quantity was at all times evaluated from Equation (4-14). A typical value calculated by such a method for the wire used is 19.83 micro-ohm-cm at 23.6 °C.

In Equation (4-12), the voltage across the wire $V_w$ was evaluated as shown in Section 3.3.
4.6 Physical Properties of Air

Both the effects of varying temperature and pressure were considered on the relevant physical properties of air in Equation (4-12).

As a function of temperature, the air dynamic viscosity, $\mu$, was expressed by

$$\mu = (0.43868 + 5.13195y - 1.31065y^2 - 0.668597y^3 + 0.922798y^4 - 0.342237y^5 + 0.042674y^6) / 10^4 \text{ poises}$$

where

$$y = 0.001 \times T_g$$

This was obtained from a polynomial curve fit to viscosity values in Ref. 62.

The air thermal conductivity expression, $\lambda_g$, used is that stated in Equation (4-6a) as a function of temperature.

Spiers (63) states that "at pressures in the neighbourhood of atmospheric, $\mu$, for gases is independent of pressure". At higher gas pressures, the dynamic viscosity is noted to increase as a varying function of gas temperature but the effect is no greater than a 1.5% increase at 15 atm. No interdependent effects of temperature and pressure on $\mu$ are apparent, however, in the ranges of interest in this work. The slight dependence of $\mu$ on pressure was nevertheless incorporated in the calibration.

The effect of pressure on $\lambda_g$ has been found negligible up to 20 atm in both Refs. 63 and 44. Spiers (63) has observed a slight increase at high pressures and a decrease at low pressures below about 1mmHg however.

The Collis and Williams (4) relationship (Equation (4-6)) used in the calibration was derived at ambient temperature and pressure where the Prandtl Number, $Pr$, is constant. To evaluate if any marked changes in $Pr$ occur
over the pressure and temperature conditions encountered in the engine combustion chambers in this work, which would upset the application of Equation (4-6) to the measurement of flow velocities in such situations, the following analysis was conducted. This evaluated

$$\text{Pr} = \frac{\mu C_p}{\lambda_g}$$

over the temperature range 273-673 ⁰K and the pressure range up to 20 atm. $C_p$ values for air at the varying temperatures and pressures were obtained mainly from Ref. 65 but Ref. 64 was also consulted. The results obtained are shown in Table 5.

Five different compression ratios were used in the experimental work. The maximum gas temperatures and pressures measured in these are listed below.

<table>
<thead>
<tr>
<th>Compression Ratio</th>
<th>Max. Gas Temperature(⁰K)</th>
<th>Max. Gas Pressure(ATM)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.9</td>
<td>439</td>
<td>6.32</td>
</tr>
<tr>
<td>5.21</td>
<td>480</td>
<td>8.86</td>
</tr>
<tr>
<td>5.29</td>
<td>484</td>
<td>9.45</td>
</tr>
<tr>
<td>6.49</td>
<td>520</td>
<td>11.0</td>
</tr>
<tr>
<td>8.88</td>
<td>563</td>
<td>15.4</td>
</tr>
</tbody>
</table>

From a comparison of these with the corresponding Prandtl Numbers in Table 5, it is seen that Equation 4-6 can be used with confidence up to a maximum compression ratio of 6.49. Its use at the 8.88 compression ratio may not be valid around T.D.C. on the compression stroke.

4.7 Theoretical and Experimental Calibration Comparisons

With all the considerations in the Sections above included, Equation (4-12) was programmed for use on a computer. A listing of this is shown in Appendix I as part of the program used to calculate flow velocities
<table>
<thead>
<tr>
<th>Temperature (°K)</th>
<th>Pressure (ATM)</th>
<th>( C_p ) (Cal/gm °K)</th>
<th>( \mu(\times 10^6) ) (gm/cm sec)</th>
<th>( \lambda_s(\times 10^6) ) (cal/cm sec °K)</th>
<th>Pr</th>
</tr>
</thead>
<tbody>
<tr>
<td>299</td>
<td>1</td>
<td>.24</td>
<td>184</td>
<td>62.4</td>
<td>.708</td>
</tr>
<tr>
<td>323</td>
<td>1.52</td>
<td>.241</td>
<td>195</td>
<td>66.2</td>
<td>.711</td>
</tr>
<tr>
<td>350</td>
<td>2.25</td>
<td>.2415</td>
<td>207</td>
<td>71.0</td>
<td>.706</td>
</tr>
<tr>
<td>407</td>
<td>4.77</td>
<td>.243</td>
<td>232</td>
<td>80</td>
<td>.706</td>
</tr>
<tr>
<td>452</td>
<td>7.48</td>
<td>.244</td>
<td>250</td>
<td>86.6</td>
<td>.706</td>
</tr>
<tr>
<td>472</td>
<td>9.0</td>
<td>.246</td>
<td>258</td>
<td>89.2</td>
<td>.711</td>
</tr>
<tr>
<td>573</td>
<td>13.0</td>
<td>.251</td>
<td>295</td>
<td>101.5</td>
<td>.732</td>
</tr>
<tr>
<td>673</td>
<td>20.0</td>
<td>.255</td>
<td>328</td>
<td>111</td>
<td>.751</td>
</tr>
<tr>
<td>520</td>
<td>11.0</td>
<td>.248</td>
<td>276</td>
<td>95</td>
<td>.719</td>
</tr>
</tbody>
</table>
in the engine cylinders.

Various techniques were used to check the accuracy of this with experimental measurements. The main approach was to insert the probe into a D.I.S.A. Calibration Wind Tunnel (see Acknowledgements) and to vary the flow velocity from 0 to 60 m/s whilst measuring Bridge Output Volts (see Fig. 6) at various wire temperatures. Fig. 9 shows the probe positioned in this manner and Fig. 24 the overall view of the set-up required, including Betts manometers and the D.I.S.A. anemometer system.

A description of the D.I.S.A. Calibration Wind Tunnel is given in Ref. 66. The flow velocity in the measuring section is evaluated from

\[
U^2 = \frac{2 \gamma}{\gamma - 1} \cdot RT_0 \left| 1 - \left(1 - \frac{\Delta P}{P_o}\right)^{\frac{\gamma-1}{\gamma}} \right|
\]

where

\[\gamma = 1.4 \text{ for air}\]
\[R = \text{Gas constant}\]
\[P_o = \text{ambient pressure measured at the air inlet point (on the left in Fig. 9)}\]

and \(\Delta P = \text{pressure difference across the nozzle}\).

The quantities \(P_o\) and \(\Delta P\) were recorded by the Betts manometers (see Fig. 24). Because of space limitations, the probe could only be inserted into the measuring section of the wind tunnel at right angles after an adaptor had been constructed from perspex to facilitate this. This can be seen in Fig. 9.

Unfortunately, differences in probe calibration become apparent when a probe is inserted into a mean flow with its axis at right angles instead of horizontally to the flow \((67), (65), (69)\). This is thought to be due to the prongs introducing a potential flow at the wire ends and such factors as
prong length, wire length and prong tip diameter are important in this connection.

To evaluate the magnitude of this effect, experiments were conducted in an open-section wind tunnel capable of attaining flow velocities of 25 m/s. Measurements were made at four wire temperatures with the probe both horizontal and vertical to the mean flow direction. The results are shown in Fig. 26. No unique effect of wire temperature was obtained but a velocity dependence is certainly apparent. This is of the same form as was noted by Jorgensen (67). Additionally, the magnitude of the overall effect, in terms of velocity change, agrees with the measurements of Dahm and Rasmussen (70). To determine the $T_w$ influence more accurately, a more sophisticated arrangement than the one used here is required.

Correction factors, based on this analysis, were there-after applied to the measurements made at right angles to the flow in the D.I.S.A. Wind Tunnel.

Fig. 27 compares the results of the theoretical, analytical calibration with those obtained experimentally using the D.I.S.A. Wind Tunnel at ambient temperature and pressure. Excellent comparisons are observed. The prong tip temperature, $T_H$, was also measured under these conditions and its variation with flow velocity and $T_w$ is shown in Fig. 20.

However, due to the relative insensitivity of the Betts manometers in accurately recording pressure differentes at the very low flow range below 4m/s, a separate rig was constructed to investigate this flow region with great accuracy. The essentials of this are shown in diagrammatic form in Fig. 28. A large metal container was built and supplied with air from a compressor at a controllable rate. A capillary tube, of 0.317 cm diameter and 137.7 cm length, was attached to this air reservoir so that the pressure difference across
the tube generated a laminar, Poiseulle flow in the tube. The flow velocity at the capillary tube output end, as a function of the pressure difference, tube length and diameter etc., is derived in Appendix 2. The probe was positioned horizontally to the flow at a distance of one tube diameter from the output end during the calibrations.

With the reservoir pressure adjusted to achieve varying flow velocities, the Bridge Output volts on the anemometer were recorded and the resulting plots in Fig. 29 were derived. Once more, excellent theoretical, analytical calibrations were achieved. This is certainly true above 40-50 cm/sec. Below this flow velocity, free convection, Grashof Number effects should be included in the analytical calibration. This was not done here since only flows above 50 cm/sec are of interest. For obvious reasons, the low flow velocity measurements above were conducted in a draught-free environment.

To illustrate the great significance of heat losses by end conduction to the prongs at low flow velocities, Fig. 30 has been included. Results on this plot are taken from the D.I.S.A. Wind Tunnel measurements mentioned above. Fig. 30 also indicates the magnitude of the heat losses by radiation which were evaluated under the same conditions.

All the calibrations so far discussed have been at ambient temperature and pressure. To check the analytical calibration at high gas temperatures, a further rig was designed and constructed. This involved passing air, by means of a Roots blower, over an engine exhaust pipe on an engine test bed (see Fig. 31). The temperature and flow velocity of the air was controlled by adjusting the speed of the Roots blower in conjunction with the manipulation of a valve just upstream of the air filter. Flow velocities were calculated from pressure measurements made with a Betts manometer attached to a Pitot-static tube. Table 6 shows the very good agreement obtained using the analytical calibration at varying gas and hot wire temperatures and at various
<table>
<thead>
<tr>
<th>Wire Temp. (°K)</th>
<th>Gas Temp. (°K)</th>
<th>( V_m ) (Volts)</th>
<th>Measured Flow Velocity (m/s)</th>
<th>Calculated Flow Velocity (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>613</td>
<td>335</td>
<td>0.742</td>
<td>411</td>
<td>439</td>
</tr>
<tr>
<td>613</td>
<td>357</td>
<td>0.782</td>
<td>781</td>
<td>810</td>
</tr>
<tr>
<td>613</td>
<td>374</td>
<td>0.862</td>
<td>1699</td>
<td>1797</td>
</tr>
<tr>
<td>613</td>
<td>362</td>
<td>0.899</td>
<td>1921</td>
<td>1976</td>
</tr>
<tr>
<td>815</td>
<td>345</td>
<td>1.069</td>
<td>403</td>
<td>419</td>
</tr>
<tr>
<td>815</td>
<td>355</td>
<td>1.113</td>
<td>568</td>
<td>597</td>
</tr>
<tr>
<td>815</td>
<td>370</td>
<td>1.142</td>
<td>688</td>
<td>797</td>
</tr>
<tr>
<td>815</td>
<td>393</td>
<td>1.185</td>
<td>1291</td>
<td>1213</td>
</tr>
<tr>
<td>815</td>
<td>390</td>
<td>1.26</td>
<td>1810</td>
<td>1714</td>
</tr>
<tr>
<td>957</td>
<td>349</td>
<td>1.300</td>
<td>426</td>
<td>431</td>
</tr>
<tr>
<td>957</td>
<td>361</td>
<td>1.391</td>
<td>689</td>
<td>748</td>
</tr>
<tr>
<td>957</td>
<td>379</td>
<td>1.467</td>
<td>1171</td>
<td>1182</td>
</tr>
<tr>
<td>957</td>
<td>395</td>
<td>1.603</td>
<td>2450</td>
<td>2230</td>
</tr>
</tbody>
</table>
flow velocities. The general tendency is for the calculated results to be higher than the measured ones. This could be due to the turbulence, superimposed on the mean flow, not having decayed at the point of measurement (see Fig. 31).

To check the analytical hot wire calibration at both high temperature and pressure is extremely difficult due to the problems associated with the design of such a rig. Hassan (26), however, attempted this and claimed reasonable accuracy between his experimental and computer predicted results. It should be noted that some equipment has recently become available commercially which is suitable for hot wire calibrations under such conditions (71).

In all the experimental results obtained, the Bridge Output Volts (see Fig. 6) was measured on a D.I.S.A. Digital Voltmeter to 3 significant figures.

4.8 Calibration in a Yawed Flow

The calibration procedures used above are based on a steady, mean flow normal to the hot wire. When the flow is yawed or pitched at an angle to the normal flow direction, the calibration is upset. Many expressions have been developed to express the effective cooling velocity, \( U_{\text{eff}} \), acting on the sensor under such conditions. One of the most detailed analysis of this has been conducted by Jorgensen (67) who studied the effect of the cooling velocity in the directions shown in Fig. 32. The effective cooling velocity is expressed by

\[
U_{\text{eff}}^2 = U_x^2 + k_1^2 U_y^2 + k_2^2 U_z^2
\]

(4-15)

where

\[
U_{\text{eff}}^2(\alpha) = U(0)^2(\cos^2\alpha + k_1^2 \sin^2\alpha)
\]

(4-16)

for \( \theta = 0 \)
\[ u_{\text{eff}}^2(\theta) = U(0)^2 \left( \cos^2 \theta + k_2 \sin^2 \theta \right) \quad (4-17) \]

for \( \alpha = 0 \)

In the above expressions, \( k_1 \) is termed a yaw factor and \( k_2 \) a pitch factor. They are expressed by

\[
k_1 = \frac{1}{\sin\alpha} \left( \frac{V(\alpha)^2 - a^2}{V(o)^2 - a} \cos^2 \alpha \right)^{1/2} \quad (4-18)
\]

and

\[
k_2 = \frac{1}{\sin\theta} \left( \frac{V(\theta)^2 - a^2}{V(o)^2 - a} \cos^2 \theta \right)^{1/2} \quad (4-19)
\]

where \( V(o) \), \( V(\alpha) \) and \( V(\theta) \) are the anemometer output voltages at angles of zero, \( \alpha \) and \( \theta \) respectively (see Fig. 32). The term "a" is defined in Equation (4-6).

Equation (4-16) is identical with Hinze's proposal for the directional sensitivity also (15).

It is impractical to consider such effects as these on the flow measurements in the engine combustion chambers used in this work since the eddy fluctuation directions are unknown. Nevertheless it is important to realize that directional sensitivity is important in obtaining accurate flow velocity values.

From the discussion in this Chapter, it is clear that, to evaluate the flow velocities existing in engine combustion chambers from the theoretical, analytical calibration procedure, four separate measurements are needed:

i) anemometer output voltage (hence \( V_w \))
ii) gas temperature, \( T_g \)
iii) gas pressure, \( P \)
iv) prong tip temperature, \( T_H \).
flow direction normal to wire

FIG. 18 — HOT WIRE IN A STEADY AIR FLOW
FIG. 19 — AIR THERMAL CONDUCTIVITY

FIG. 21 — PRONG THERMOCOUPLE CALIBRATION
l = 2.32 mm
d = 0.001 cm

FIG. 20 — PRONG TIP TEMPERATURE MEASUREMENTS
FIG. 22 — TYPICAL DIMENSIONS OF PROBE
FIG. 23 — TEMPERATURE COEFFICIENT OF RESISTANCE OF WIRE

FIG. 26 — EFFECT OF PROBE ORIENTATION
FIG. 27 — HOT WIRE PROBE CALIBRATION

Voltage across Hot Wire, $V_w$

Flow Velocity (m/s)

$l = 0.208$ cm
$d = 0.001$ cm

$T_w = 954^\circ K$

$T_w = 811^\circ K$

$T_w = 610^\circ K$

$T_w = 419^\circ K$
FIG. 28 - LOW AIR FLOW VELOCITY CALIBRATION RIG
Experimental

l = 0.203 cm

d = 0.001 cm

Tw = 892 °K

Tw = 744 °K

Tw = 554 °K

Tw = 374 °K

FIG. 29 - LOW FLOW VELOCITY CALIBRATION
FIG. 30 — END CONDUCTION AND RADIATION LOSSES

$T_W = 811^\circ K$

$l = 0.208 \text{ cm}$

$d = 0.001 \text{ cm}$
FIG. 31—HIGH AIR TEMPERATURE CALIBRATION RIG
FIG. 32 — HOT WIRE VELOCITY VECTOR RESOLUTION

FIG. 24 — HOT WIRE PROBE CALIBRATION SET-UP
CHAPTER 5

ENGINE RIG AND INSTRUMENTATION
In order to accurately calculate the flow velocities in the engine combustion chambers, the probe and its prong thermocouple attachment has to be positioned in the combustion chamber along with suitable transducers for measuring the gas temperature and pressure. Additionally, a crankangle degree marker is required to indicate at what points in the engine cycle certain events are occurring.

The output from the anemometer-probe configuration and the other transducers has then to be processed into the required form and suitably recorded for later analysis. This Chapter describes how this was attained in this work.

5.1 The "Motored" Engine Configuration

The basic engine used was a single cylinder, Petter W1 Laboratory Engine. This was modified from its normal side-valve form to accommodate a cut-away section of an overhead valve, Vaux hall Wyvern Cylinder Head, external views of which can be seen in Figs. 33 and 34. The dimensions of the combustion space on this particular cylinder head were found to coincide with the bore diameter of the basic Petter engine whilst, at the same time, providing a large "squish area" for the generation of fluctuating flow velocities during the compression stroke (see Fig. 74 in Chap. 7). The following data is relevant to this modified engine:
Bore diameter - 85.1 mm
Stroke - 82.5 mm
Connecting Rod Length - 165 mm
Swept Volume - 468 cm³

Valve Timing:
Inlet Valve Opens - 16° B.T.D.C.
Inlet Valve Closes - 120° B.T.D.C.
Exhaust Valve Opens - 121° A.T.D.C.
Exhaust Valve Closes - 19° A.T.D.C.

Various compression ratios were obtained by inserting spacers between the cylinder head and the cylinder block. These gave compression ratios of nominally 3.9, 5.29, 6.49 and 8.88. The actual values varied slightly depending on the probe position in the combustion space. After the engine tests involving this "squish" combustion chamber, the "squish" part was drilled out to facilitate the provision of a cylindrical disc chamber of compression ratio 5.21. This is shown in Fig. 35.

The engine was fitted with a new cylinder liner and piston and was "run in" by motoring for a period of 30 hrs. It was then completely dismantled and washed scrupulously in Trichloroethylene to remove all trace of particulate matter and oil. This process was essential to prevent dirt particles entering the combustion space and breaking the delicate hot wire of the probe. After re-assembly, the thickest possible oil was used for lubrication to inhibit the formation of an oil mist in the combustion space. Additionally, the oil holes in the connecting rod were blanked off and the rocker shaft lubrication supply disconnected to prevent oil seepage into the combustion chamber. Because the engine was to be used in the "motored" condition, the coolant passages were also blocked off to forestall the presence of loosened rust particles. Such extreme measures eventually proved worthwhile since all the engine
tests were conducted without breakage of the hot wire. It is extremely important to emphasize the requirement of extreme cleanliness both in and around the engine in order for such work to be performed.

A simple, intake manifold system was designed consisting of a pipe in which a butterfly valve was incorporated. A micronic air filter was attached to the outermost end. This can be seen very well in Figs. 33, 34 and 35. The filter ensured the prevention of any dust particles of damaging size entering the inlet manifold and combustion space. Downstream of the butterfly valve, a small tube was inserted in the manifold piping to record manifold pressure.

The engine was motored by a 10 h.p., 6000 rev/min, DC electric motor made by Lancashire Dynamo Electronic Products Ltd. and driven by a Ward Leonard Set. It is suitable coupled to the engine as shown in Figs. 36 and 37. These latter Figures also show complete views of the "motored" engine configuration used.

5.2 Probe Traverse Mechanism

Several basic requirements are demanded of the design arrangement which enables the probe to enter the combustion chamber. These include:

i) a central location of the hot wire in the chamber

ii) the ability to traverse the probe right across the chamber

iii) the ability to rotate the probe about its own axis

iv) relatively easy insertion and withdrawal - to be effected without removal of the cylinder head

v) complete sealing

vi) a "stop" to prevent the hot wire hitting the far wall of the combustion chamber.
Additionally, a gas temperature measuring transducer should be fitted as close as possible to the hot wire of the probe at all probe positions in the combustion chambers so that it records the same gas temperature as the hot wire operates in. It should not interfere with the flow near the wire nor be influenced itself by the heat transfer by convection from the hot wire. A fast-response thermocouple was used for this purpose — this is described later.

The above requirements are largely met by the probe traverse mechanism shown in diagrammatic form in Fig. 38. This was designed and built for use in this work. It fits into the cylinder head in the position shown in Fig. 34 and the probe enters the combustion space centrally and between the valves, as shown in Fig. 35.

Referring to Fig. 38, it can be seen that the probe fits into a probe holder and is located at its inner end by a brass sleeve held in position by a countersunk screw. At its outer end, the probe is fixed to its holder in the manner illustrated in Fig. 38. For clarity, this particular section has been re-drawn to a larger scale in Fig. 39. The thermocouple wires pass through a groove cut in the sleeve and a brass olive bears on the circumference of the sleeve as shown. A brass hexagonal nut is screwed tight against the olive to achieve adequate location and a tight fit relative to the probe holder. The thermocouple wires pass out of the probe holder through a small groove cut in the inner surface of the brass hexagonal nut (see Fig. 39) next to the probe itself. Silkaset was liberally used at this point to ensure no contact between the thermocouple wires and the probe or brass hexagonal nut. The washer in this region (see Fig. 38) is of sufficient width to prevent the hot wire hitting the far end of the combustion chamber wall. A photograph of the probe fitted into the probe holder is shown in Fig. 5.
A fixing groove was machined on the outside of the probe holder as can be seen in Fig. 38. This acts as a locating track for the fixing screw (see Fig. 38) of the probe holder tube to position itself into when the probe is being inserted or withdrawn from the combustion space. It is necessary because the thermocouple leads will only pass through the combustion chamber sleeve (see Fig. 38) at one point where a narrow groove has been cut. The sleeve itself is a tight fit relative to both the cylinder head and the probe to effect good sealing.

A series of probe positioning slots were undercut around the circumference of the probe holder at 9 mm intervals (Fig. 38). These enable the probe to be positioned in the combustion chamber at different known points. The exact positioning point can be detected by the spring-supported ball bearing falling into one of the undercut slots. When such a position has been attained, the fixing screw is tightened down on to the probe holder.

The probe and probe holder, as one complete unit, can be withdrawn without removal of the probe holder tube (see Fig. 38). Before its insertion into the probe holder tube however, the latter was always centralized relative to the hole in the combustion chamber in which the combustion chamber sleeve fits. Very great care was taken during the insertion procedure for obvious reasons. To this end, a guiding track (not shown) was constructed and clamped to the probe holder tube at the outermost end. This provided central location for the probe and probe holder unit relative to the axis of the probe holder tube before the probe actually entered this latter tube. A small amount of grease on the surface of the probe holder was found to ease the insertion since its fit in the probe holder tube was a tight push fit.
To facilitate probe rotation in the combustion chamber, the combustion chamber sleeve had several notches countersunk in its circumference for the holding screw to locate in (see Fig. 38).

Fig. 41 is a close-up view of the probe and thermocouple positioning in the cylindrical disc combustion chamber. It will be noted that the thermocouple leads were strapped to the circumference of the probe body by cellotape.

The design of the probe traverse system described above had the added advantage that the probe and probe holder unit could be fitted into the D.I.S.A. Wind Tunnel for calibration as one unit without the need for dismantling (see Fig. 9). The perspex attachment piece, used for this purpose, fitted comfortably on to the probe body at the hot wire end.

To investigate the effects of probe vibration, under "motored" engine conditions resulting from the particular probe insertion and traverse configurations used in this work (Fig. 38), a theoretical analysis was conducted. This is described in Appendix 3.

5.3 Signal Recording System

Preliminary investigations of the flow in the spark ignition engine combustion chambers (described in Section 5.4) indicated the desirability of recording simultaneously two flow fluctuating quantities:

i) the anemometer output signals corresponding to the flow in the frequency range below 200 Hz

ii) anemometer output fluctuations corresponding to the flow in other discrete frequency bandpass ranges.

Additionally, the following quantities have to be recorded in order to correctly compute the turbulent
flow fluctuation velocities corresponding to the anemometer output signals in i) and ii) above:

iii) cylinder pressure
iv) cylinder temperature
v) prong thermocouple temperature
vi) crankangle degree marker

It was initially hoped to achieve the recording of these six separate signals by photography off an oscilloscope screen. However, due to the non-availability of a drum camera, to facilitate the desired aim of recording successive cycles, and the congestion on the oscilloscope screen when 6 separate traces are present, this approach was discarded.

An alternative that was considered was to store the transducers' output signals on tape for later analysis. The lack of a suitable 6-channel system, however, prevented this technique being used also.

Resort was thus made to an Ultra-Violet Recording System. Several basic requirements were demanded of this in its application to this work. These were:

a) a fast, stable paper speed in order to record the transducer signals with excellent definition at the highest engine speeds to be used (approximately 1500 rev/min)
b) at least 6 data channels available
c) the ability to record frequencies up to 5800 Hz
d) excellent acceleration to the desired paper speed because it was intended to only heat up the hot wire for a short period during measurements, since it has already been noted that the mechanical strength of Pt-10% Rh decreases at high temperatures
e) adequate deflection of the ultra-violet beam for a good level of accuracy
f) the availability of suitable galvanometers.
The above criteria were found to be present in the series M 1250, Direct Recording Ultra-Violet Oscillograph manufactured by Southern Instruments. This had a maximum paper speed of 254 cm/sec, an acceleration time of less than 0.2 sec to the selected recording speed, 6 data channels and a maximum beam deflection of 15 cm. Additionally, it could record frequencies up to 10000 Hz with a suitable galvanometer.

Only six galvanometers were available for fitting to the U.V. recorder. Their characteristics are given below.

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>Sensitivity (mA/cm)</th>
<th>Sensitivity (mV/cm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>.0022</td>
<td>.132</td>
</tr>
<tr>
<td>200</td>
<td>.01</td>
<td>.80</td>
</tr>
<tr>
<td>300</td>
<td>.02</td>
<td>2.30</td>
</tr>
<tr>
<td>500</td>
<td>.05</td>
<td>6.0</td>
</tr>
<tr>
<td>1000</td>
<td>.5</td>
<td>16.0</td>
</tr>
<tr>
<td>5000</td>
<td>10</td>
<td>320</td>
</tr>
</tbody>
</table>

An optimization series of experiments was therefore conducted to connect each transducer in the engine, after suitable signal processing, to a suitable galvanometer in order that the UV recorder may accurately record its output with no lack of frequency response and sensitivity.

The 5000 Hz galvanometer was immediately designated to the recording of the flow fluctuations in the bandpass ranges and the 200 Hz galvanometer to the anemometer output signals in the frequency range below 200 Hz. Cylinder gas pressure was recorded on the 1000 Hz galvo, gas temperature on the 500 Hz galvo, prong temperature on the 100 Hz galvo and the crankangle degree marker was attached to the 300 Hz galvanometer.
5.4 Anemometer Output Voltage Processing

Some initial experiments were conducted in which the raw anemometer output signal, corresponding to the flow in the engine, combustion chambers, was displayed on an oscilloscope screen. This indicated that, during induction, the hot wire probe was recording a mean flow of gas with high frequency flow fluctuations superimposed on it. Towards the end of induction, this was noted to have largely disappeared so that, during compression, the flow could be considered as comprising low frequency eddies but still with higher frequency fluctuations superimposed. Such an observation was also noted by Semenov(3).

Consequently, a signal processing system was devised to treat the raw anemometer output signal so as to simultaneously record:

i) flow fluctuations in the frequency band below 200 Hz

ii) flow fluctuations off this low frequency flow in discrete bandpass ranges.

After much experimentation and troubleshooting, the system shown in block diagram form in Fig. 42 was eventually used. No attempt was made to measure absolute intensities of turbulence over the bandpass ranges because the time constants involved would occupy a long period of crankshaft rotation especially at the low frequencies. Such values could not be considered as occurring at definite points in the engine cycle and, hence, could not be used with confidence to predict flame propagation rates in the combustion chambers. The technique used, therefore, was to measure instantaneous flow fluctuations and to average these, at each point in the engine cycle, over many engine cycles. This is described in greater detail, however, in Chapter 6.
Referring again to Fig. 42, the functions of the various components comprising the system will first be discussed.

**Connection and Switching Unit**

This serves the purpose of receiving the raw anemometer output signal and distributing it for processing and eventual recording in two out of three possible ways:

i) to Auxiliary Unit 1, the Attenuator Unit and finally the UV recorder for the measurement of the low frequency fluctuations below 200 Hz. This is always recorded

ii) When set to IN position (see Fig. 42), for the transmission of the signal to the Dawe Bandpass Filter Unit and, hence, through Auxiliary Unit 2, Full Wave Rectifier etc. for the eventual recording of the turbulent flow fluctuations in certain bandpass ranges.

iii) When set to OUT position, the Dawe Filter Unit is bypassed. It was initially hoped to record the turbulent fluctuations in the complete frequency range from 200-5000 Hz with this circuit by setting the filters on Auxiliary Unit 2 accordingly. However, the signal analysis on the UV paper proved too difficult. Consequently, this signal processing circuit was never used thereafter.

**Auxiliary Units**

These are units, marketed by D.I.S.A., that contain a variable gain amplifier (of 3:1 maximum gain), a series of adjustable filters of gentle (3db/octave) roll-off, a square-wave generating circuit, a control for setting zero volts on the Digital Voltmeter (see Fig. 42) and a signal inversion and non-inversion capability. At all
times, this was set so as to be non-inverting in this work. Its input and output impedances are 16kΩ and 160Ω respectively.

With the Dawe Filter Unit in circuit, the filters on Auxiliary Unit 2 are always by-passed. Alternatively the low pass filter on Auxiliary Unit 1 is always set to pass frequencies below 200 Hz. The characteristics of this filter in terms of attenuation, amplification and phase change over the frequency range up to 200 Hz are shown in Appendix 4.

**Dawe Bandpass Filter Unit**

This was used because it was especially desired to study the low frequency range of the turbulent energy spectrum in the engine combustion chambers. This is the region where most of the energy was noted by Semenov\(^{(3)}\) to occur. The bandpass ranges of the unit used in this work are

<table>
<thead>
<tr>
<th>Bandpass Ranges</th>
<th>Frequencies</th>
</tr>
</thead>
<tbody>
<tr>
<td>90 - 180 Hz</td>
<td></td>
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<tr>
<td>180 - 360 Hz</td>
<td></td>
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<tr>
<td>360 - 700 Hz</td>
<td></td>
</tr>
<tr>
<td>700 - 1500 Hz</td>
<td></td>
</tr>
<tr>
<td>1500 - 2800 Hz</td>
<td></td>
</tr>
<tr>
<td>2800 - 5800 Hz</td>
<td></td>
</tr>
</tbody>
</table>

and so it is particularly applicable to the study of low frequency fluctuations. Its characteristics in terms of attenuation, amplification and phase change were determined with it in the position shown in the circuit in Fig. 42 so that its actual operating conditions were simulated. These characteristics are given in Appendix 5 for each particular frequency band.

Its input and output impedances are both 600Ω.
**Full Wave Rectifier**

The purpose of this unit was to simplify the analysis of the anemometer output voltage fluctuations on the UV traces. Various circuits were tried initially but in all, relatively large voltages of around 150 mV were required to start the diodes working. Since the high frequency turbulent fluctuations in the combustion chambers can give anemometer output signals of this order, an alternative unit was sought. The circuit shown in Fig. 43 was eventually used. Its characteristics were investigated over the complete frequency and voltage range of interest and were found to be acceptable when the Power Supply Unit supplied more than $+10$ V (see Fig. 42). Below this figure, considerable distortion and oscillations on the rectified wave forms were apparent. Consequently, the Power Supply Unit was always set to $+12$ V.

The diodes were found to start working at about 30 mV on this unit and were operational at this voltage right through the frequency range required. A certain small amount of distortion of the rectified wave forms did occur at the highest frequencies used but this was negligible and unlikely to upset the accuracy of the results.

**40μF Electrolytic Condenser**

This was inserted in the circuit because the Full Wave Rectifier was noted to be extremely sensitive to any DC standing voltage greater than 15 mV (measured on Digital Voltmeter) at its input. Under such conditions, it failed to rectify evenly. Examination of the Dawe Filter and Auxiliary Unit circuits showed no series capacitance in line to ensure complete blockage of D.C. and so the above condenser was used. Its influence in the circuit with regard to attenuation and phase change was checked and found to be negligible. The 40μF condenser used has a 25 volt loading capability.
Power Amplifier

The power amplifier was required in the circuitry because the 5000 Hz galvanometer in the UV recorder needs a much higher current and voltage (10 mA/cm and 320 mV/cm respectively) to achieve a certain deflection than do the other galvanometers used. It was designed for this work with its input and output in phase. The circuit diagram on which the amplifier is based is shown in Fig. 44. An input potentiometer is included enabling the input voltage to be adjusted as required so that the amplifier does not become overloaded (indicated by reference sine wave signals, for example, distorting) and so that the required output level can be set.

In the circuit (see Fig. 44), the operational amplifier is signified TA20 and the booster amplifier TA50. The latter is capable of delivering an output current of 100 mA.

The gain of the amplifier was investigated by adjusting resistor RF. It did not matter what value was used, however, since amplifier overloading occurred at the same output voltage at all times. A value of 3.3KΩ was eventually used giving a gain of 3.95.

The output potentiometer in the circuit enabled the current supplied to the 5000 Hz galvanometer to be set independently of the input current. The input impedance of the Power Amplifier is always greater than 100KΩ (to a maximum of 1MΩ ) and its output impedance is always less than 250Ω.

Digital Voltmeter (D.V.M.)

This was used especially during the development of the circuit (Fig. 42) to indicate any "loading" effects of certain units and to detect any standing voltages at both Auxiliary Unit Outputs. During the ensuing measurements of flow velocities in the engine combustion chambers, such standing voltages were continually checked for and, if present, eliminated using the zero
adjust control on the Auxiliary units in conjunction with the meter reading on the D.V.M.

The D.V.M. used was made by D.I.S.A. and could provide an accurate digital readout from 1 mV up to 100 volts. Its input impedance was 1 MΩ.

**Attenuator Unit**

The function of this unit was to adjust the degree of attenuation of the signals being supplied to the UV recorder in order to achieve maximum sensitivity at all times on the UV traces. The circuit diagram of the unit developed for this work is shown in Fig. 45. It has 6 channels and was also used for the above same purpose with the gas pressure, temperature, prong thermocouple temperature and crankangle degree marker transducer outputs. A coarse and fine attenuator adjustment is incorporated on each channel. Various degrees of damping were included also. When used in the circuit in Fig. 42, its input impedance was sufficiently high at all times so that no "loading" of Auxiliary Unit 1 occurred. Once an attenuator position was set, it remained unaltered during a series of tests.

**1 Volt D.C. Calibration Unit**

This was used to provide a calibration voltage for the circuit which records the low frequency flow fluctuations below 200 Hz. Its circuit diagram is shown in Fig. 46.

**Oscillator**

The oscillator was included (see Fig. 42) to set 1:1 Gains across both circuits used and to provide a calibration signal for the bandpass circuit.

The input and output impedances of each unit in the circuitry in Fig. 42 were noted and arranged so that
no mis-match occurred in terms of "loading". The development of the system involved considerable time and troubleshooting to achieve this end.

Additionally, when each unit was installed in the circuit, phase checks were made across it to confirm that these were correct. This was done by providing a random signal input to the unit and monitoring both input and output on a Tektronix Type 555 Dual Beam Oscilloscope. The random signal input was conveniently supplied by "motoring" the engine and operating the probe-anemometer system. Fig. 47 shows the sort of results obtained under the conditions stated. A Rolleicord camera using Polaroid Black and White 400 Speed Type 32 film was used.

Final confirmation that the measurements at the UV recorder corresponded exactly to conditions at the anemometer output were then made in the above manner. Fig. 48 is an example of the results obtained for the low frequency flow (< 200 Hz) circuit. The top trace is the anemometer output and the bottom trace is the attenuator output in which the fluctuations above 2000 Hz are filtered.

Calibrations

These were performed with the anemometer disconnected since both the 1 volt D.C. Calibration Unit and the Oscillator were found to "load" the anemometer.

For the low frequency (< 200 Hz) flow circuit, the oscillator was first used to provide a 100 Hz input and a 1:1 Gain was set across the circuit (see Fig. 42) by suitable adjustment of the amplifier on Auxiliary Unit 1. The 1 volt D.C. Calibration Unit was then used to provide a calibration on the UV paper.

In the bandpass circuit, calibrations were effected
by feeding a $\frac{1}{2}$ volt peak-to-peak signal from the oscillator at 125 Hz (being the midpoint of lowest bandpass range used) into the Dawe filter input. To compensate for the various attenuations in the circuit, the amplifier on Auxiliary Unit 2 was adjusted to maintain a $\frac{1}{4}$ volt peak rectified signal at the output of the Full-Wave rectifier. For the other bandpass ranges on the Dawe Filter Unit, different levels of attenuations were known to exist (see Appendix 5). Corrective values were applied to the standard 125 Hz calibration described above throughout the other bandpass ranges to avoid calibration at each mid-frequency setting. This was done except at the 2800-5800 Hz bandpass range on the Dawe Filter Unit where, because the Full-Wave Rectifier effectively doubles the frequency and the galvanometer in this circuit was limited to 5000 Hz, a calibration was independently obtained. For this, the Electrolytic Condenser, Full-Wave Rectifier and Power Supply Unit were removed from the circuit and a 4300 Hz, $\frac{1}{2}$ volt peak-to-peak signal was used to achieve the calibration as before.

Such procedures meant that the attenuator unit and power amplifier could be adjusted independently to give maximum sensitivity on the UV traces without the calibrations being upset.

Figs. 49, 50 and 51 show views of the instrumentation described in this Section in relation to the complete engine-instrumentation arrangement.

5.5 Pressure Measurement

Originally, a Kistler piezo-electric transducer connected to a Charge Amplifier etc. was used as the pressure-measuring device. However, an enormous amount of trouble was experienced due to drift. This
was so bad that no repeatable calibrations could be achieved. Therefore, the Kistler was discarded in favour of an inductance transducer feeding into an FM system.

The particular transducer used was a Southern Instruments Type G301 of range 37 atmospheres and inductance 15.08 microhenries. These types of pressure transducer were known (73) to suffer negligible zero drift, maintain their calibration over long periods and be ideal for use in "motored" engine applications where temperature effects are minimal.

The transducer formed part of a tuning circuit in the usual manner and its output was fed to an oscillator (type M.785) and, hence, to a pre-amplifier (type MR.513) incorporating a discriminator, which produced a DC output voltage proportional to the pressure being measured. All equipment in the system was manufactured by Southern Instruments. The carrier frequency was nominally 2 MHz.

Calibration of the transducer was effected with its output fed into the same equipment as was used during the engine tests (Fig. 52) so that no error from this source was involved. The transducer was installed in a pressurized air-bottle with the pressure being indicated on a gauge. The DC output voltages were obtained from a digital voltmeter. Excellent repeatability over separate calibrations was achieved as is shown in Fig. 53.

The transducer was then installed in the cylinder head of the engine in the position normally occupied by the spark plug (on the right in Fig. 33). Both during calibration and installation in the engine, the transducer was tightened on to its seat with a 172 kg.cm torque as recommended by Southern Instruments.
Because the 1000 Hz galvanometer, used for recording the pressure trace on the UV recorder, required a relatively large current of 0.5 mA per cm of UV beam deflection, it was found that, to achieve large traces on the UV paper, the Attenuator Unit had to be set with no attenuation on the relevant channel. This meant that the Attenuator Unit was loading the FM system since the latter had an output impedance of 1500Ω. To overcome this, so as to maintain the Attenuator Unit for adjustment of maximum sensitivity on the UV traces, some means of providing a high input impedance to the FM equipment whilst, at the same time, showing a low output impedance to the Attenuator Unit was required. A low drift, Fylde DC amplifier (type FE-151-BD) was ultimately used for this. Its input and output impedances are 1MΩ and 1Ω respectively. The phase relationship on this amplifier was made non-inverting and a suitable gain was set across it.

The complete pressure measuring system used in this work is thus shown in Fig. 52.

Both during calibration and subsequent engine testing, the same transducer cable was maintained. This was because it was an intrinsic part of the oscillatory circuit.

5.6 Gas Temperature Measurement

A resistance thermometry system was initially considered but the availability of exceedingly fast response, micro-miniature thermocouples dictated that this latter method would be used.

The thermocouples mentioned were produced by BLH Electronics in the U.S.A. and the particular type used was an exposed junction, chromel-alumel thermocouple. This was suitable for use in an
oxidizing atmosphere at 1000°C for short periods and at 500°C for long periods. Fig. 41 shows the thermocouple installed in the cylinder head next to the hot wire probe.

The probe of the thermocouple consists of a 0.0355 cm metallic tube through which two fused ceramic tubes pass. These measure 0.0076 cm in outside diameter. Inside these two insulating tubes are the 0.00254 cm diameter thermocouple wires. The hot junction is formed by electrically fusing the two wires together at the end of the insulation. At the end away from the hot junction, the very fine thermocouple wires are connected to 0.0254 cm leads. This is done inside a ceramic connector (see Fig. 41).

The small size of the unit minimizes radiation and conduction errors and, also, disturbances to the air flow pattern. Its response to a step change in temperature from ambient to 180°C at an air flow velocity of 14 m/s has been measured to be 13 msec. This fast response was confirmed by some "motored" engine tests, the results of which were displayed on a Tektronix Storage Oscilloscope. Fig. 54 shows a typical oscillogram obtained. During the compression stroke, the response was certainly fast enough as the peak gas temperature corresponded exactly with the T.D.C. mark. However, during expansion, the heat capacity of the thermocouple bead obscured actual temperature measurements (Fig. 54). This did not matter too much since the gas temperatures at this time could be easily evaluated from the polytropic law, knowing the cylinder pressures and the gas temperature during compression.

Calibrations of the thermocouple were achieved by its being placed in a small special container inside an oven. The purpose of the container was to reduce convection current effects. The thermocouple
temperature was measured on a Comark electronic thermometer (described in Appendix 6) and the oven temperature was recorded both by a temperature gauge on the oven and by a separate thermometer. As for the pressure transducer calibration, the thermocouple calibration was performed with it in the gas temperature measuring system used during the engine tests (Fig. 55) to simulate its actual operating conditions. Fig. 56 shows the results obtained from these measurements together with the thermocouple output in millivolts, obtained using a Portable Pye Potentiometer.

Fig. 55 shows the gas temperature measuring system used during the engine tests. A suitable gain was set on the Fylde DC Amplifier in order to obtain adequate deflections on the UV recorder. Its high input impedance and low output impedance ensured that it was ideally matched in the circuit.

5.7 Prong Temperature Measurement

Details of the thermocouple, positioned on the tip of one prong of the hot wire probe, have already been described in Chapter 4 along with the calibration of the thermocouple. Exactly the same processing system was used before the prong thermocouple output was fed into the 100 Hz galvanometer on the UV recorder as was used for the gas temperature measuring thermocouple.

5.8 Crankangle Degree Marker

Because of the limitation in the number of high frequency galvanometers available for use in the UV recorder, it was not possible to obtain a crankangle degree mark at short intervals e.g. every 2° of crankangle rotation. As stated previously, a
300 Hz galvanometer was designated to the recording of this particular parameter. This greatly restricted the number of marks per engine cycle that could be recorded on the UV traces. Only six were finally used and, to achieve these, six holes were drilled at 60° intervals in the circumference of the coupling between the DC electric motor and the engine flywheel (see Fig. 36). An inductive, magnetic pick-up was placed adjacent to this coupling so that a series of voltage blips, similar to those shown in Fig. 57a, were generated.

At the highest engine speeds used (about 1500 rev/min), the galvanometer response was not fast enough to eliminate interference between the negative-going pulse from one degree marker with the positive-going pulse from the next degree marker 60 crankangle degrees away. To overcome this, the pulses in Fig. 57a were passed through the diode circuit shown in Fig. 57b to eliminate the negative-going pulse. Fig. 57c indicates the final pulse-shape obtained.

The exact points of crankangle indication required are arrowed in Fig. 57c because these are the points at which the inductive pick-up first "sees" the holes in the circumference of the coupling. The rest of the pulse, which can occupy quite a large number of crankangle degrees, depending on engine speed, is due to the lack of frequency response of the 300 Hz galvanometer.

A deeper hole than the others was drilled to indicate T.D.C.. This is because the magnitude of the voltage pulse generated in the inductive pick-up is given by

\[
e.m.f. = K \frac{d\ell'}{dt}
\]
where \( l' \) = depth of the hole.

The output from the inductive pick-up and diode circuit was then fed to the Attenuator Unit and, hence, to the UV recorder.

5.9 Engine Speed Measurement

Two techniques were used in this work

i) a Smiths' Tachometer

ii) a Racal Counter.

The Tachometer was mainly used since this gave an accurate, continuous and quick indication of engine speed. This was essential because the time involved in adjusting the engine speed to a predetermined value had to be kept to a minimum out of consideration for the hot wire probe installed in the combustion chamber.

However, the Racal Counter system was also utilized especially during the development of the engine-instrumentation set-up. An inductive, magnetic pick-up was positioned adjacent to the engine flywheel in which 6 holes had been drilled at 60° crankangle intervals (Fig. 36). Voltage blips were generated as in Fig. 57a but the magnitude of each pulse varied from pulse to pulse due to vibrations of the flywheel. When the output from the inductive, magnetic pick-up was fed to the Racal Counter, the speed indication given by the Counter was greatly in error and varied wildly at a constant engine speed. This was because the Racal was triggering off both the positive and negative-going pulses.

To counteract this, the circuit shown in Fig. 58 was developed in which the pulsed wave forms at each stage are shown. The two diodes OA 202 in series
act as a Zener diode i.e. they provide a constant voltage output no matter what the variations in input voltage may be. When the final, pulsed wave forms were fed to the Racal Counter, excellent agreement was obtained when compared with the Tachometer measurements.

Views of the complete engine-instrumentation arrangement described in the above Sections can be seen in Figs. 50 and 51.
FIG. 33 — PART OF ENGINE RIG

FIG. 34 — PART OF ENGINE RIG
FIG. 35—VIEW OF PROBE IN COMBUSTION CHAMBER

FIG. 36—'MOTORED' ENGINE CONFIGURATION
FIG. 37—'MOTORED' ENGINE CONFIGURATION

FIG. 41—PROBE IN CYLINDRICAL DISC CHAMBER
FIG. 38 — PROBE TRAVERSE MECHANISM
FIG. 39—PART OF PROBE TRAVERSE MECHANISM
FIG. 42 — BLOCK DIAGRAM OF ANEMOMETER OUTPUT VOLTAGE PROCESSING
FIG. 43 — FULL WAVE RECTIFIER CIRCUIT
FIG. 44 — POWER AMPLIFIER CIRCUITRY
FIG. 45 — ONE CHANNEL OF ATTENUATOR UNIT
FIG. 46 — 1 VOLT DC CALIBRATION UNIT

Input to Full Wave Rectifier — Bottom Trace
Output from Full Wave Rectifier — Top Trace

FIG. 47 — TYPICAL PHASE CHECK OSCILLOGRAM
FIG. 48 - TYPICAL OSCILLOGRAM USED TO CHECK CORRESPONDENCE BETWEEN SIGNALS AT ANEMOMETER & U.V. RECORDER

FIG. 49 - PART OF INSTRUMENTATION SET-UP
FIG. 50—COMPLETE INSTRUMENTATION ARRANGEMENT

G.51—COMPLETE ENGINE-INSTRUMENTATION SET-UP
CALIBRATION

FIG. 52 — PRESSURE RECORDING SYSTEM
FIG. 53 — PRESSURE CALIBRATION CURVE
FIG. 58 — CIRCUIT BEFORE RACAL COUNTER

FIG. 54 — OSCILLOGRAM OF CYLINDER GAS TEMPERATURE RECORDED BY GAS THERMOCOUPLE
FIG. 55 — GAS TEMPERATURE RECORDING SYSTEM
FIG. 56 — GAS THERMOCOUPLE CALIBRATION
FIG. 57 — CRANKANGLE DEGREE MARKING
CHAPTER 6

RESULTS AND DISCUSSION
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RESULTS AND DISCUSSION

Observations of the raw anemometer output signal displayed on an oscilloscope indicated the presence of a mean flow with high frequency fluctuations superimposed during the engine's induction stroke (see Fig. 48). After this period, during the compression stroke, this mean flow had largely disappeared and appeared to have been converted into a predominately low frequency, eddying gas motion. Again, a certain amount of high frequency flow fluctuation seemed to be present and superimposed on the low frequency eddies.

Consequently, it was determined to investigate the low frequency eddying motion and to conduct a spectrum analysis into the higher frequency fluctuations. Chapter 5 describes the techniques used to process the raw anemometer output signal in order to facilitate this.

Typical results obtained on the UV paper during the engine tests are shown in Fig. 59. This includes traces of:

i) the anemometer output fluctuations below 200 Hz. This is designated \( v \).

ii) the rectified anemometer output fluctuations in a certain, discrete frequency band, \( v' \)

iii) gas pressure, \( P \)

iv) gas temperature, \( T_g \)

v) prong tip temperature, \( T_H \)

vi) crankangle degree marker

The analysis procedure was extremely laborious since each trace had to be dealt with individually. It was concentrated on the period of crankangle rotation from inlet valve closure on the compression stroke to about 40° A.T.D.C. Instantaneous measurements of all the required parameters above were made during this
period at 10° crank angle intervals.

The flow velocities in the combustion chamber corresponding to v (i.e. the anemometer output fluctuations below 200 Hz) were obtained by feeding the measured values of v, p, T\textsubscript{g} and T\textsubscript{H} into the calibration computer program derived in Chapter 4. Appendix I contains a listing of this program. Additionally, the hot wire probe operating characteristics were fed in e.g. R\textsubscript{w}, R\textsubscript{0}, etc. The resulting flow velocities are given the symbol u.

However, values of u obtained for one engine cycle are likely to be untypical of the values over many cycles. Consequently, ensemble averages of u were made, at the same crank angle rotation points, over 22 consecutive cycles. This average value is given the symbol  \bar{u}. Ideally, more than 22 cycles ought to be considered but the time factor involved in the laborious procedure that had to be used to analyse the UV traces prohibited against this.

Because u is a non-linear function of v (see Equation 4-12), average values of v over the 22 consecutive cycles could not be used in the calibration computer program. The flow velocities, u, in each cycle had to be obtained individually and the resulting values averaged over the 22 cycles to give  \bar{u}.

The flow fluctuations, u', corresponding to the anemometer output fluctuations, v', in the bandpass ranges investigated in this work, were obtained by assuming that the fluctuations v' were deviations off v at each point in the engine cycle. This was determined from a study on an oscilloscope screen of the basic anemometer output signal. Consequently, values of \(v + v'\) were used in the computer program (Appendix I) to evaluate the corresponding flow velocity (u + u'). The flow fluctuation velocity, u', was then obtained by subtracting u from this latter quantity. Average values of u' were derived in this way over 12
consecutive engine cycles to give the ensemble average, \( \bar{u}' \). This technique was also used by Semenov\(^{(3)}\) to study the flow fluctuations in certain bandpass ranges. However, he averaged \( u' \) and \( u \) over a crankangle interval of 24° instead of at instantaneous points, as has been done in this work.

In the computer program used to evaluate the flow velocities in the engine combustion chamber (Appendix I), the gas temperatures during both compression and expansion were evaluated from the measured gas temperature at inlet valve closure and the measured gas pressures during these phases using the polytropic equation. This was because it was noticed that the heat capacity of the thermocouple, used to record gas temperature, gave misleading results on the expansion stroke (see Fig. 54 in Chapter 5). Comparisons were made of the measured and calculated gas temperatures during the compression stroke and these were found to coincide exactly. This procedure meant a small saving in UV trace analysis time.

A polynomial curve fit was made of the pressure calibration curve in Fig. 53 for use in the computer program also.

It was extremely fortunate that all the engine tests were conducted without breakage of the hot wire. Its dimensions were

\[
\begin{align*}
 d & = 0.001 \text{ cm} \\
 l & = 0.208 \text{ cm}
\end{align*}
\]

during the engine tests.

Before and after each test, the wire "cold" resistance \( R_0 \) was accurately determined. Normally, the two values coincided exactly but, sometimes, very slight increases in \( R_0 \) were measured after a test. An average value was then calculated. The mean wire operating temperature, \( T_W \), was evaluated from Equation 4-13. At all times, this was set to be very high to achieve the required frequency response and sensitivity.
A typical value used was 850°K.

Results are presented in terms of the individual combustion chambers studied. For the "squish" chamber at the point marked 3.

For all the experiments, the probe was orientated as shown in Fig. 41 so that the axis of the hot wire was perpendicular to the direction of piston movement. The axis of the probe itself was precisely between the valves at a distance of 8 mm from the top of the chamber and 6.5 mm from the cylinder head surface in contact with the cylinder head gasket. The axial positioning of the probe plane is defined by the line drawn through the probe positioning points 1, 2, 3, 4 and 5 in Fig. 60. This latter Figure is drawn to scale.

ratio under the conditions stated. This cyclic variation can be considered a fundamental cause of the cyclic dispersion in flame travel times and pressure development commonly observed on firing engines.

The variation in $\bar{u}$ from inlet valve closure to 40° A.T.D.C. at engine speeds of 700, 950, 1200 and 1450 rev/min is shown in Fig. 62. There is a general tendency for $\bar{u}$ to decrease continuously throughout the compression stroke. The noted decreases in $\bar{u}$ mean that energy is being transferred out of the low frequency fluctuation band (< 200 Hz) into the higher bandpass ranges, where it is eventually dissipated into heat by the action of viscous friction. This transfer of energy appears to be occurring in a more pronounced manner at the highest engine speed due probably to the higher flow velocities at this time augmenting the mixing process.

The "squish" effect as the piston approaches T.D.C. seems to be virtually non-existent except possibly at the highest engine speed condition where a
velocity component seems to be introduced into the flow. This is due to the "squish" height, h, (Fig. 60) being too large at this compression ratio to generate any noticeable velocities. Confirmation of this was obtained from a theoretical study of the velocity with which the charge in the clearance space (i.e. between the piston and the "squish" part of the cylinder head) is ejected as the piston approaches T.D.C. The analysis is described in Appendix 7. It can only give an indication of how the "squish" velocities are influenced by piston movement and engine speed however.

Its results, when applied to the particular operating conditions used at this compression ratio, are shown in Fig. 63a. The hot wire probe does not detect the very small "squish" velocities at this time in any obvious manner in terms of u probably because the velocities break down into higher frequency fluctuations and because the wire orientation is not ideal for the recording of such low velocities.

During the flame propagation period (see Fig. 62), \( \bar{u} \) increases almost linearly with engine speed.

At 30 - 40° A.T.D.C., marked increases in \( \bar{u} \) occur due to velocities generated as the piston retracts on the expansion stroke.

Measurements of \( \bar{u}' \) in the relevant frequency bandpass ranges are given in Figs. 64a, b, c and d. In most cases, smooth curves could not be drawn through the measured points. Two factors are responsible for this:

a) only 12 engine cycles were analysed to achieve each point

b) the eddies in the combustion chamber could never attain exactly the same position relative to the hot wire probe from cycle to cycle. Thus, on one engine cycle, the hot wire may have been located within an eddy at some point whilst, on the next cycle, it may have been
positioned either at a different point within an eddy or on the edge of an eddy.

Nevertheless, the overall trends are quite clear from Figs. 64a, b, c and d. The $u'$ fluctuations increase with engine speed as expected and most of the turbulent energy is contained in the low frequency region below 700 Hz. Around T.D.C., the distribution of the energy in the three lowest bandpass ranges studied is not very clear except possibly at the lowest engine speed (Fig. 64b). At this speed, the relatively low flow velocities during the compression stroke could result in energy not being dissipated into higher frequency fluctuations at a very great rate.

At 1450 rev/min close to T.D.C. on the compression stroke, there does seem to be an indication of a "squish" velocity effect in the 180 - 360 Hz and 360 - 700 Hz bandpass ranges to correspond to that noted in Fig. 62.

During the early part of the compression stroke, the trend is for the flow fluctuation velocities to increase as the frequency bandpass range decreases.

In the three highest frequency bandpass ranges (Fig. 64c and d), the energy distribution is very clear in that $u'$ decreases with the increasing frequency band. Comparison of Figs. 64c and d with Figs. 64a and b also indicates a clear distinction between the flow velocities in the 700 - 1500 Hz range with those in the three lowest ranges.

The overall trend in all bandpass ranges is for $u'$ to decrease from inlet valve closure onwards. However, at about 30° A.T.D.C., the $u'$ fluctuations increase sharply in correspondence with $u$ increasing at this time (see Fig. 62).

The low values of $u'$ in the highest frequency bandpass range (2800 - 5800 Hz) led to a belief that the scale of the dissipation eddies may be much larger in a high temperature and pressure flow than in a
similar flow at ambient conditions. To investigate this, the following analysis was conducted.

The size of the viscous dissipation eddies, $l_d$, is given by (15)

$$l_d = \left( \frac{\nu^3}{\varepsilon} \right)^{1/4}$$

where $\nu$ is the kinematic viscosity and $\varepsilon$ is the turbulent energy dissipation rate given by (15)

$$\varepsilon \propto \frac{u'^3}{l_e}$$

In this latter expression, $l_e$ is the size of the energy containing eddies in the turbulent energy spectrum. This is considered to be independent of temperature and pressure. Thus,

$$l_d \propto \nu^{3/4}$$

$$\propto \left( \frac{u}{\rho} \right)^{3/4}$$

To a good approximation, $\mu \propto T_g^{0.7}$ and $\rho \propto (1/T_g)$. Also, $\mu$ is independent of pressure whilst $\rho \propto P$. Therefore,

$$l_d \propto \frac{T_g^{1.7}}{P^{3/4}}$$

In a "motored", spark ignition engine, the absolute temperature and pressure are increased by factors of 2 and 10 approximately during the compression stroke. Substitution in the above equation indicates that the scale of the dissipation eddies is, in fact, approximately the same as at ambient conditions.

The conclusion must be, therefore, that the transfer rate is not great enough during the compression stroke to pass much energy to the higher frequency region.
Therefore, most of the energy remains in the low frequency bands. The energy that is transferred seems to be offset by an equally high dissipation rate.

As the engine speed increases, the flow velocities also increase. The mixing process is, thus, augmented and $\tilde{u}'$, in the high frequency bands, increase.

In common with the $\tilde{u}$ results in Fig. 62, the $\tilde{u}'$ fluctuations at T.D.C. in the three highest frequency bands increase in an approximately linear manner with engine speed (see Fig. 64c and d). The overall effect in the three lowest bandpass ranges (Fig. 64a and b) is similar but the individual frequency band increases do not follow this law because of the effects noted earlier in this Section.

6.2 "Squish" Combustion Chamber - Compression Ratio 3.9:1

The attainment of this compression ratio required a spacer between the cylinder head and cylinder block so that height, $h$, was 1.89 cm in Fig. 60.

In terms of the low frequency flow fluctuations, $\tilde{u}$, very similar results (see Fig. 62) were obtained as for the 5.29:1 compression ratio condition. However, the initial flow velocities at inlet valve closure are lower in all cases because of the greater volume into which the induction air expands. No "squish" velocity effect can be detected at all as the piston approaches T.D.C. on the compression stroke. As for the previous compression ratio condition (Section 6.1), this is due to the "squish" height, $h$, (see Fig. 60) being too great and is again confirmed by the results of the theoretical analysis described in Appendix 7. These results are plotted in Fig. 63a.

Comparing the plots in Fig. 62 for the 3.9 and 5.29:1 compression ratios, there seems to be a general trend, immediately after inlet valve closure, for the
higher compression ratio \( u \) values to decrease more rapidly. This indicates a greater degree of mixing due to the higher flow velocities and a greater transfer of energy to the higher frequency fluctuations. Confirmation of this was found in the \( u' \) fluctuations which were about 10\% greater at T.D.C. at the higher compression ratio.

At all engine speeds, the \( u \) values are lower for the 3.9:1 compression ratio combustion chamber at T.D.C. This occurs even though they are of comparable order, and greater in many instances, than for the 5.29:1 compression ratio engine during most of the compression stroke. The indications are, therefore, that the hot wire probe is after all detecting the higher "squish" velocities at the 5.29:1 compression ratio.

A linear increase in \( u \) at T.D.C. with engine speed is again apparent, as also is the increase in flow velocities at 30 to 40° into the expansion stroke.

Exactly similar trends were obtained for the \( u' \) fluctuations throughout the period of the engine cycle studied as were noted in Figs. 64a, b, c and d for the 5.29 compression ratio. Consequently, these results will not be presented except to show the manner in which the \( u' \) fluctuations, at T.D.C., vary as a function of frequency for the four engine speeds investigated (see Fig. 65). The magnitude of the fluctuations are lower at the 3.9 compression ratio.

6.3 "Squish" Combustion Chamber - Compression Ratio 6.49:1

The 6.49:1 compression ratio condition was achieved by arranging for height, \( h \), in Fig. 60 to be 0.6 cm.

The results for the variation in \( u \) throughout the compression stroke, at engine speeds of 750, 950, 1150 and 1400 rev/min, are given in Fig. 66.
Compared with the measurements at the 3.9 and 5.29:1 compression ratios (Fig. 62), the "squish" effect is now becoming apparent in a distinctive way as the piston approaches T.D.C.. Such results are borne out by the increasing "squish" velocities predicted by the theoretical method described in Appendix 7 and presented in Fig. 63b.

After inlet valve closure, the $\bar{u}$ values at this compression ratio decrease more rapidly with piston movement than is noted from the measurements of the two previous compression ratios studied (see Fig. 62). This indicates a greater degree of mixing, due to the higher initial flow velocities, so that energy is transferred out of the low frequency range into higher bands. It occurs on such a scale that the $\bar{u}$ values are comparable with and in many instances less than, the values presented in Fig. 62 for the two lower compression ratios at this stage in the engine cycle. The "squish" effect counterbalances this, however, so that the magnitude of $\bar{u}$ around T.D.C. is always greater at the 6.49:1 compression ratio. An almost linear increase in $\bar{u}$ with engine speed is again noted at T.D.C.. The non-linear increase in $\bar{u}$ with engine speed in the period immediately following inlet valve closure is attributed to wire orientation relative to the eddies.

The recordings of the flow fluctuations, $\bar{u}'$, in the relevant bandpass ranges at this compression ratio are plotted in Figs. 67a, b, c and d throughout the compression stroke. As was noted in Section 6.1 for the 5.29:1 compression ratio, there appears little chance of indicating which is the predominant frequency bandpass range at each of the engine speeds at the three lowest bandpass ranges investigated (Figs. 67a and b). Again, this is attributed to wire orientation effects and the relatively low number of engine cycles analysed. It is clear, nevertheless, that most of the energy is again contained below 700 Hz. The overall magnitude of the $\bar{u}'$ fluctuations is greater,
especially in the lowest bandpass ranges studied, at this compression ratio than for the previous two discussed.

Many of the comments made in Section 6.1 for the 5.29:1 compression ratio also apply here. The results themselves are fairly self-explanatory from the graphs.

6.4 "Squish" Combustion Chamber - Compression Ratio 8.88:1

No spacer was fitted between the cylinder head and the cylinder block at this compression ratio so that height, h, in Fig. 60 corresponds only to the width of the gasket i.e. 0.15 cm.

A large "squish" effect, corresponding to the indications of the theoretical analysis (see Fig. 63b) for this compression ratio, is now obtained (see Fig. 66). The magnitude of the "squish" velocities increase with engine speed as shown. Only three engine speeds have been analysed at this compression ratio because of an inaccurate calibration setting at the 1200 rev/min test.

An approximately linear increase in \( \bar{u} \) with engine speed at T.D.C. is once more obtained. Many of the same statements can be made about the variation in \( \bar{u} \) throughout the compression stroke as have already been made in previous Sections in this Chapter. These will not, therefore, be repeated since they are apparent from the relevant plots.

A most noticeable effect at this compression ratio, however, is the large increase in \( \bar{u} \) at about 30 - 40° A.T.D.C. caused by the suction effect generated as the piston retracts on the expansion stroke.

The \( \bar{u}' \)fluctuations at this compression ratio setting are of a similar form to those shown in Figs.
64 and 67 for the 5.29 and 6.49 compression ratios except that the magnitude of the fluctuations is greater around T.D.C.. Additionally, the "squish" effect is more apparent especially on the lowest bandpass ranges at the highest engine speed. These results are presented here in the form of a plot of \( \bar{u}' \) at T.D.C. as a function of frequency for the three engine speeds investigated (see Fig. 68).

### 6.5 Cylindrical Disc Combustion Chamber - Compression Ratio 5.21:1

After all the "squish" chamber tests had been completed, the "squish" part of the cylinder head was drilled out and the resulting gaps filled with araldite to facilitate the provision of a cylindrical disc combustion chamber. In this, no displacers were present which could generate "secondary" flows during the compression stroke. The combustion chamber is shown in Fig. 41.

Similar tests were carried out to those conducted on the "squish" chambers. Thus, flow velocities, \( \bar{u} \), and fluctuating flow velocities, \( \bar{u}' \), were measured through an engine speed range of 350 to 1500 rev/min.

Fig. 68 plots the variation in \( \bar{u} \) that was recorded throughout the compression stroke. Comparison with the \( \bar{u} \) results for the various "squish" combustion chambers shows that the flow velocities at inlet valve closure are lower for this combustion chamber form. This is due to the air flow through the inlet valve during induction expanding immediately into the large combustion chamber volume because the valves are now relatively small in relation to the cross-sectional area of the chamber (Fig. 41). In the "squish" chambers, this does not happen to such a large extent due to the constraining effect of the "squish" parts of the chamber (see Fig. 74).

A low frequency motion is generated in the
cylindrical disc combustion chamber at 40 - 50° before T.D.C. which is not apparent in the "squish" chambers. Semenov(3) also recorded this. It could be due to a velocity component being introduced into the flow at the mid-stroke position, when the piston speed is greatest, which is recorded at the hot wire with a phase lag determined by the distance between the wire and the piston and the velocity of the induced flow. Such an effect was not noticed in the "squish" chambers probably because it was obscured by the chamber form.

The manner in which \( \bar{u} \) varies throughout the compression stroke indicates that less energy is transferred into the higher frequency fluctuation regions compared with the previous chamber forms studied. Again, this is due to the lower flow velocities in the cylindrical disc chamber just after inlet valve closure.

A linear increase in \( \bar{u} \) with engine speed is again noted at T.D.C. (Fig. 69).

In terms of the \( \bar{u}' \) fluctuations, the measurements shown in Figs. 70a and b were obtained. Compared with the corresponding measurements for the "squish" chambers (see Figs. 64 and 67), the \( \bar{u}' \) values are much lower during the compression stroke, which lends support to the statement above that less energy is transferred into the higher frequency ranges. Thus, in the region of T.D.C., the \( \bar{u}' \) values are correspondingly less. The distribution of energy within the three lowest bandpass ranges is again not clear (see Fig. 70a) and is attributed to effects of wire orientation, as previously described, and not enough engine cycles being analysed. However, it is nevertheless apparent from Figs. 70a and b that, in the region of T.D.C., the energy predominance is concentrated in the frequency region below 700 Hz.
6.6 **General Comments on the effect of Combustion Chamber Shape on Flow Velocities**

It is apparent from the results presented so far that the main influence of a change in combustion chamber shape is an alteration in the magnitude of the flow velocities in the frequency range below about 1000 Hz. The higher frequency fluctuations remain, to a large extent, relatively unaffected. In this connection, it should be noted that the main source of the turbulent fluctuations occurring during the compression stroke is the velocity gradients in the jet flow of gas into the cylinder during the intake stroke. This has been conclusively shown by Semenov\(^{(3)}\). These turbulent fluctuations, which have been shown to be predominantly low frequency in nature, then transfer energy into the higher frequency ranges during the compression stroke at a rate primarily dependent upon the rate of mixing and, hence, upon their flow velocities.

The increases in \( \bar{u} \), at T.D.C., with engine speed, \( n \), for the four "squish" combustion chambers studied are shown in Fig. 71. These are of a linear form as already noted and, if extrapolated back to zero engine speed, pass through the origin. The equations of the lines are:

- Compression Ratio 3.9:1 \( \bar{u} = 0.227 \, n \)
- Compression Ratio 5.29:1 \( \bar{u} = 0.273 \, n \)
- Compression Ratio 6.49:1 \( \bar{u} = 0.327 \, n \)
- Compression Ratio 8.88:1 \( \bar{u} = 0.379 \, n \)

Fig. 71 also plots the \( \bar{u} \) values, at T.D.C., for the Cylindrical Disc Combustion Chamber which is again of a linear form passing through the origin. The following equation applies to this combustion chamber over the speed range studied:

\[ \bar{u} = 0.24 \, n \]
This latter relationship does not agree with Semenov's measurements\(^{(3)}\). These indicate a relationship of the form

\[ \bar{u} = n^{2.1} \]

However, Semenov neglected end conduction losses in the hot wire calibration technique he used. These have a non-linear effect with varying flow velocity as shown in Fig. 30 and, if they had been included in Semenov's analysis, a more linear relationship would have been obtained.

6.7 Effect of Hot Wire Probe Position in Combustion Chamber

For this series of tests, the 8.88:1 "squish" Combustion Chamber was used. The engine was "motored" at full throttle and at a constant engine speed of 600 rev/min. The probe was traversed right across the combustion chamber and the results shown in Fig. 72, in terms of the \( \bar{u} \) velocities at T.D.C., obtained.

It is apparent that the above-mentioned flow velocities remain fairly constant across the chamber cross-section but increase greatly at the point nearest the "squish" part of the cylinder head. This is thought to be due to the "squish" velocities generating a large vortex at this point as they become ejected from the space between the piston and the cylinder head. The velocity of the vortex will decrease as the height, \( h \), in Fig. 60 increases.

6.8 Effect of Hot Wire Probe Rotation in Combustion Chamber

These tests were conducted to determine if any large departures from isotropic, turbulent flow exist in spark ignition engine combustion chambers. In principle, a hot wire probe with only a single wire
cannot be used to measure the turbulent fluctuations in three dimensions. However, rotation of the probe about its own axis can indicate the degree of isotropism without actually measuring the three components of the fluctuations.

Such measurements were made with the probe rotated at angles, $\phi$, of $-45^\circ$, $+45^\circ$, and $+90^\circ$ from its normal operating position in the combustion chamber. They were conducted in the 3.9:1 compression ratio, "squish" chamber at an engine speed of 700 rev/min and at full throttle. This particular combustion chamber was used because Semenov (3) found that the greatest departures from isotropic flow occurred at low compression ratios. Due to the time consuming analysis procedure used to process the results off the UV paper, only the worst condition was investigated in this work.

The results are presented in Fig. 73 in terms of the ratio \( \bar{u}_\phi/\bar{u} \). Very little effect is noted. Although the tests conducted are much too limited to draw firm conclusions, it appears a reasonable approximation to consider the flow to be more or less isotropic.

6.9 Analysis of Errors

It is clear from the results presented that, although the hot wire anemometer is not the ideal instrument for such work in terms of absolute accuracy, it can be used to illustrate trends very clearly. The variability in the direction of the flow velocities relative to the hot wire probe, and the positioning of the probe at different points within and on the edges of eddies from cycle to cycle, makes it extremely difficult to be categorical about the magnitudes of the errors involved in the measurements of $\bar{u}$ and $\bar{u}'$. It appears fairly clear, however, that the results obtained represent a tendency towards the lower limits for these quantities. This is because the anemometer output voltage attains its maximum value when the flow
is normal to the hot wire (see Section 4.8) and it is unlikely that this condition is frequently achieved with the probe at a constant position in the chamber. Fig. 61 confirms this to a certain extent since, on some engine cycles, the $u$ velocities recorded are much higher than on others. Whilst much of this may be due to the cyclical variation in $u$, a large part can also be attributed to the hot wire probe positioning, relative to the eddies in the combustion chamber, being completely different from cycle to cycle.

Semenov (3) has stated that $\bar{u}$ values may be up to 15 - 20% lower than those which actually exist in the combustion chambers. However, no indication is given of how this factor was achieved. Based on such results as those shown in Fig. 61, the $\bar{u}$ values may, in fact, be too low by a factor of 50% at the end of the compression stroke.

Because, also, it is highly probable that eddy sizes exist in the combustion chambers which are smaller than the wire length, a correction factor has to be included to correct for this on the $\bar{u}'$ values obtained. Semenov (3) used a factor developed by Dryden et al (75). This is

$$K' = \frac{c}{(2 (e^{-c} - 1 + c))^{1/2}}$$

where

$c = \text{wire length/scale of the eddies.}$

Calculations show that $K'$ is about 2.5 at a frequency of 1000 Hz and increases to 5 at 5000 Hz. In these calculations, the scale of the eddies, $L$, was arbitrarily determined from

$$L = \frac{1}{2\pi} \frac{\bar{u}}{k}$$
where $f$ is the frequency corresponding to the value $\bar{u}$. The factors, $K'$, are the maximum correction factors possible.

Besides the effects noted above, other sources of error could arise in the velocity measurements due to inaccuracies in the measurements of cylinder temperature and pressure and, also, in the analytical calibration procedure used. However, extreme lengths were resorted to in order to minimize these, as has been explained in previous Chapters.

In the recordings of $\bar{u}$ in the "squish" combustion chambers, the generally low levels of "squish" velocities obtained could, in part, be explained by leakage of the cylinder gas through the piston rings in the period close to T.D.C.. A large reduction in "squish" velocity measurements due to this has been observed by Shimamoto and Akiyama (77).

The small differences in flow velocities between combustion chambers that have been noted in this Chapter required a tremendous amount of care and patience during the engine tests and in the processing of the results. If this had not been taken, errors would have completely obscured the differences between combustion chambers.
FIG. 59 — SIGNALS ON U.V. PAPER
FIG. 60 — 'SQUISH' COMBUSTION CHAMBER FORM
'SQUISH' COMBUSTION CHAMBER
Compression Ratio 5.29:1
700 Rev/Min

FIG. 61 — CYCICAL VARIATION IN FLOW VELOCITIES
FIG. 62 — FLOW VELOCITIES IN 'SQUISH' COMBUSTION CHAMBERS
FIG. 63 — PREDICTED 'SQUISH' VELOCITIES
FIG. 64c

'Squish' Combustion Chamber
Compression Ratio 5.29:1
- - 1450 Rev/Min
- - 1200 Rev/Min

Fluctuating Flow Velocities, \( \bar{u} \) (cm/sec)

Piston Position (Crankangle Degrees)

700-1500 Hz
1500-2800 Hz
2800-5800 Hz

FIG. 64d

'Squish' Combustion Chamber: Compression Ratio 5.29:1
- 950 Rev/Min
- 700 Rev/Min

Fluctuating Flow Velocities, \( \bar{u} \) (cm/sec)

Piston Position (Crankangle Degrees)

700-1500 Hz
1500-2800 Hz
2800-5800 Hz

FIG. 64 — FLUCTUATING FLOW VELOCITIES IN 'SQUISH' COMBUSTION CHAMBER
FIG. 65—FLOW FLUCTUATION VELOCITIES IN 'SQUISH' CHAMBER

FIG. 68—FLOW FLUCTUATION VELOCITIES IN 'SQUISH' CHAMBER
FIG. 66 — FLOW VELOCITIES IN 'SQUISH' COMBUSTION CHAMBERS.
FIG.67a

- Squish Combustion Chamber
- Compression Ratio 6.49:1
- 1400 Rev/Min
- 1150 Rev/Min

Fluctuating Flow Velocities, \( \bar{u} \) (cm/sec)

Piston Position (Crankangle Degrees)

\[ \times 90-180 \text{Hz} \]
\[ + 180-360 \text{Hz} \]
\[ \circ 360-700 \text{Hz} \]

FIG.67b

- Squish Combustion Chamber
- Compression Ratio 6.49:1
- 950 Rev/Min
- 750 Rev/Min

Fluctuating Flow Velocities, \( \bar{u} \) (cm/sec)

Piston Position (Crankangle Degrees)

\[ \times 90-180 \text{Hz} \]
\[ + 180-360 \text{Hz} \]
\[ \circ 360-700 \text{Hz} \]
FIG. 67c

FIG. 67d

FIG. 67 — FLUCTUATING FLOW VELOCITIES IN 'SQUISH' COMBUSTION CHAMBER
FIG. 69 — FLOW VELOCITIES IN CYLINDRICAL DISC COMBUSTION CHAMBER

FIG. 71 — FLOW VELOCITY VARIATION WITH ENGINE SPEED
Compression Ratio 5.21:1

- 1450 Rev/Min
- 1150 Rev/Min
- 700 Rev/Min

[Graph showing fluctuating flow velocities vs. piston position with frequency bands indicated]

FIG. 70a

[Graph showing fluctuating flow velocities vs. piston position with frequency bands indicated]

FIG. 70b

FIG. 70—FLUCTUATING FLOW VELOCITIES IN CYLINDRICAL DISC CHAMBER
FIG. 73 — ISOTROPIC NATURE OF TURBULENCE

FIG. 72 — PROBE TRAVERSE ACROSS 'SQUISH' COMBUSTION CHAMBER
CHAPTER 7

RATE OF TURBULENT FLAME PROPAGATION IN COMBUSTION CHAMBERS
CHAPTER 7

RATE OF TURBULENT FLAME PROPAGATION
IN COMBUSTION CHAMBERS

The second aim of this work (see Chapter 1) is the utilization of the measured fluctuating flow velocities to predict and describe more accurately than henceforth possible, the rate of turbulent burning in the engine combustion chambers. To achieve this, a computer model of the combustion process in a spark ignition engine must be derived. Such a model was developed by the author in a previous work (1).

However, this was based on the combustion and flame propagation process in a Renault, I.F.P. Variable Compression Ratio Research engine and must be suitably modified for application to a different engine configuration. The modifications performed are described later in this Chapter. A major problem that exists in this connection is the development of a theoretical turbulent burning velocity expression for use in the computer simulated model. This again is detailed in a later Section.

As a check on the burning velocity predictions of the computer model, some means of measuring the flame travel times in the combustion chamber must be available so that a direct comparison can be effected under identical operating conditions. A technique to do this, based on flame speed measurements from the spark plug to an ionization probe, has been previously used with good results (1). It was, therefore, repeated here.

The work described in this Chapter is concentrated entirely on the "squish" combustion chamber form. This was because another Petter engine was available on which a Vauxhall Wyvern cylinder head could be installed. An exactly similar engine configuration to that used for the "motored" anemometry tests was, therefore, available for the flame speed tests so that the anemometry results are directly
applicable. The work was not extended to the cylindrical disc combustion chamber because araldite was used as a "filler" in this chamber. This would not have been able to withstand the high temperatures and pressures during the firing mode of engine operation.

7.1 Flame Speed Measurements

The objective was to measure the time interval involved in the passage of the flame from the spark plug to a point diametrically opposite the spark plug. An ionization probe was fitted at this point to detect flame passage. The principle of the technique stems from the fact that active particle concentrations in the flame front are greater than both those in the already burnt combustion products and, obviously, those in the unburnt part of the charge. When the flame passes an ionization gap therefore, a pulsed electrical potential is generated and a pulsed current starts to flow. Consequently, flame arrival times at a probe can be recorded on suitably available equipment.

The fitting of the ionization probe in the combustion chamber presented quite a few problems. It was desired to have it positioned as shown in Fig. 74, i.e. directly opposite the spark plug. This meant, however, that some water coolant passages had to be negotiated. The problem was overcome by the design shown in Fig. 75. An outer sleeve was screwed through the "squish" part of the combustion chamber in exactly the same position as the hot wire probe entered the chamber (see Fig. 38) during the anemometry tests. A hole had been drilled in this sleeve to accommodate the rest of the ionization probe assembly. At the innermost end of the sleeve, this hole diameter was reduced to provide a surface for the ceramic piece to bear on and, by so doing, to effect good sealing. This was especially important since it was required to achieve complete sealing without having to resort to cement or araldite to make a permanent fixture of the
A 0.077 cm diameter wire passed through the hole in the centre of the ceramic piece as shown (Fig. 75) to form the ionization gap with the combustion chamber surface at the innermost end. The wire was increased in diameter at its outer end and covered with some insulating material to prevent contact with the metal surroundings. The ceramic piece is held in position by an inner tube. This is, in turn, supported by a brass plug and nut. The latter screws on to the outer end of the outer sleeve so that the whole assembly locks up.

Initially, much trouble was experienced during firing engine operation due to poor sealing. Two thin copper washers placed as shown in Fig. 75 eventually solved the problem.

The entire unit, fitted in the combustion chamber, is illustrated in Fig. 74. It can be seen that the ionization gap is formed well away from the "squish" end wall to prevent any possibility of the flame being quenched before it actually reaches the gap.

The lead from the ionization gap was fed out to a Combustion Interval Meter (see Acknowledgments), the principle of operation of which is described by Harrow. This indicates directly on a scale the time interval, in crankangle degrees, between the spark firing and the flame arriving at the ionization probe. Consequently, an input was also required to the Meter to indicate the time at which the spark fires. This was obtained by forming a coil of many turns of wire around the high tension lead to the spark plug. A special adaptor was constructed to facilitate this. This is shown in Fig. 76.

The Combustion Interval Meter was used because flame travel time intervals were measured over a restricted engine speed range of 600 to 1500 rev/min and the differences between these time intervals are relatively small over this range. Originally, a storage
oscilloscope was used as the recording device but an enormously extensive sampling procedure was found to be required. The Combustion Interval Meter proved extremely useful as it integrates the flame travel times over a period of time thereby minimizing the randomness due to the cyclic dispersion. Extremely steady readings were almost always obtained.

The engine fitted on the test bed and the associated instrumentation set-up is shown in Fig. 77. Engine speed was measured by a tachometer and ignition timing by a stroboscope bearing on a pointer above the engine fly wheel. A magneto, fixed ignition timing was used. Air flow rates into the engine were obtained from a 3" diameter Viscous Flowmeter and fuel flow rates by measuring on a stop-watch the time for a certain quantity of fuel to be consumed by the engine. A Solex carburettor was used which incorporated a modification to allow the size of the main jet to be altered to achieve the required air/fuel ratios. The fuel was premium grade gasoline and all tests were conducted at a constant air/fuel ratio of 12.8:1 since it is known that cyclic dispersion is a minimum around this air/fuel ratio (79).

Flame travel time measurements were made throughout the engine speed range mentioned above on all four "squish" chambers described in Chapter 6. The results of the experiments for the three lowest compression ratios (i.e. 6.49, 5.29 and 3.9:1) are shown in Figs. 78, 79 and 80. Measurements were also made for the highest 8.88:1 compression ratio configuration but, because the engine was "knocking" appreciably, these are relatively meaningless and, as such, are not presented.

A range of ignition timings was used at each "squish" chamber configuration to achieve good results over the complete engine speed range. This was because the measurements tended to be influenced by the flame speeds suddenly increasing (indicated by flame travel
times, in crankangle degrees, reducing) as the engine speed was increased at a constant ignition timing. At the 6.49:1 compression ratio configuration (see Fig. 78), this is clearly observed and is thought to be due to the flow velocities in the combustion chamber increasing greatly on the expansion stroke due to the suction effect generated as the piston descends. This hypothesis is compatible with the greatly increased values of $\bar{u}$ measured in the hot wire anemometry tests at this stage in the engine cycle (see Fig. 66). It is further confirmed by the fact that it occurs earlier when the ignition timing is retarded (see Fig. 78). Consequently, the flame can be visualized as being pulled at great speed towards the ionization gap and into the area between the piston and the "squish" part of the cylinder head as the piston descends.

In the computer simulation work, it is desired to predict flame propagation rates with no such mean flow present since the problem would be greatly complicated. Consequently, only those flame travel time measurements are utilized for comparative purposes that increase in an almost linear manner with engine speed (see Figs. 78, 79 and 80).

The flame travel time measurements indicate that the least scatter is obtained on the 6.49:1 compression ratio plot (see Fig. 78). Consequently, this was the combustion chamber configuration that was specifically used to effect the initial direct comparison between the measured flame travel time values and those from the computer simulated combustion model. Excellent repeatability was obtained at all times at this compression ratio when measuring flame travel time values. This was also good for the 5.29:1 compression ratio condition (Fig. 79), but the scatter and general poor repeatability is fairly obvious from Fig. 80 for the 3.9:1 compression ratio combustion chamber.
7.2 Development of the Computer Simulated Combustion Model

An exhaustive account of the development of the computer program to simulate the flame propagation and combustion processes in the Renault I.F.P. Engine is given by James (1). The model takes into account finite rates of flame propagation, heat transfer and dissociation according to chemical equilibrium considerations. A realistic flame pattern development across the combustion chamber was included from some experimental observations using ionization gaps.

The Semenov (80) laminar flame propagation theory is the basic factor in the turbulent burning velocity expression used and this was multiplied by a factor, based on some flame speed measurements in the I.F.P.

Detailed information on the Semenov laminar flame propagation theory used is given in Appendix 9.

where

\[ U_T \] is the turbulent burning velocity
\[ U_L \] is the laminar burning velocity
and

\[ n \] is the engine speed (rev/min)

Mixing between the newly burnt parts of the charge and those previously burnt was assumed to take place instantaneously. Consequently, no temperature gradients exist in the combustion chamber. This is known to be inaccurate (81) but is, nevertheless, a good approximation when predicting turbulent burning velocities.

Dissociation in the burnt gases was catered for by the techniques developed by Brinkey (82) and the calculations were performed from first principles. Heat transfer between the burnt charge and its surroundings
7.2 Development of the Computer Simulated Combustion Model

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The Semenov (80) laminar flame propagation theory is the basic factor in the turbulent burning velocity expression used and this was multiplied by a factor, derived from some flame speed measurements in the I.F.P. engine, to cater for the effects of turbulence. The factor, as a function of engine speed only, is

\[
\frac{U_T}{U_L} = 1 + 0.00197n
\]  

(7-1)

where

- \(U_T\) is the turbulent burning velocity
- \(U_L\) is the laminar burning velocity

and

- \(n\) is the engine speed (rev/min)

Mixing between the newly burnt parts of the charge and those previously burnt was assumed to take place instantaneously. Consequently, no temperature gradients exist in the combustion chamber. This is known to be inaccurate (81) but is, nevertheless, a good approximation when predicting turbulent burning velocities.

Dissociation in the burnt gases was catered for by the techniques developed by Brinkley (82) and the calculations were performed from first principles. Heat transfer between the burnt charge and its surroundings
was included by using the Annand (83) formulation and instantaneous surface areas, involved in the heat transfer, were evaluated as the flame progressed across the combustion chamber.

The simulation started at inlet valve closure on the compression stroke and ended at exhaust valve opening on the expansion stroke. Many parameters could be adjusted including air/fuel ratio, ignition timing, compression ratio, charge temperature and pressure at inlet valve closure, engine speed, exhaust residual concentrations and the presence of injected water (if required).

The intention was to apply this basic computer program to simulate the flame propagation and combustion process in the 6.49:1 compression ratio, "squish" chamber. Several modifications were required before this could be achieved, the main ones of which were:

i) modifications to the part of the program which calculated the burnt volumes, and hence masses, of charge in the Renault I.F.P. combustion chamber at any piston position and flame front distance from the spark plug. This had to be altered in order to perform the same calculations in the "squish" combustion chamber.

ii) modifications to the instantaneous surface areas of the cylinder head, piston and cylinder walls involved in the heat transfer during the flame propagation across the combustion chamber. Also, changes were required in the overall surface areas of the cylinder head, piston and cylinder walls for evaluations of heat losses during the compression and expansion phases.

iii) determinations of the turbulent burning velocity at any point in the flame travel across the combustion chamber. This is now
required to be based on the measured values of the turbulent flow fluctuation velocities presented in Chapter 6. This, however, is dealt with in more detail in Section 7.3.

Additionally, many smaller changes were required too numerous to mention.

As stated above, one of the main problems was the development of a suitable subroutine for use in the computer program which would accurately calculate the burnt volumes of charge at any piston and flame front position during the flame propagation process. This was particularly difficult since the combustion chamber form is relatively complex with "squish" areas present and with the spark plug situated in a retraction in the smallest "squish" area (see Fig. 74).

To achieve this aim, however, a spherical flame pattern development was assumed at all times for the flame propagating outward from the spark plug. This is shown in two views in Fig. 81. The assumption of a spherical flame pattern development is known to be inaccurate especially just after ignition. The reasons for this are given in Ref. 1. However, it is nevertheless a reasonably good approximation and has been successfully used by Patterson (84) and Phillipps and Orman (85).

The manner in which the burnt volumes were evaluated is now described.

The combustion space volume (above line BB in Fig. 81) was treated separately from the cylinder volume (below line BB). In the plan view, circles were drawn at small, constant intervals with their centre at the position of the spark plug. In the side view, the combustion space was divided into 8 strips of width 2 mm and the cylinder volume into many strips of thickness 0.1 mm.

For the combustion space, when \( r < a \), (see Fig.
the volume is given simply by the relevant equation for a sphere. As \( r \) increases to a value

\[ a < r < b \]

the volume occupied by the burnt gases in the retracted part of the small squish area has to be calculated. This was done by measuring the surface areas in the plan view at radii between \( r = a \) and \( b \) at each of the planes \( s(0) \) to \( s(8) \). Corrected radii, \( r' \), at each of the planes, relative to the line AA (see Fig. 81), were calculated using Pythagoras' theorem. As shown in the plan view, consideration was thus taken, at each plane, of the retraction in the combustion chamber in which the spark plug fits. The volume was then evaluated by integrating the measured surface areas over all planes with the limits determined by the flame radius, \( r \).

When the radius, \( r \), is greater than \( b \), the surface areas at the plane \( s(3) \), at all the radii shown in the plan view within the combustion space limitations, were measured with a planimeter. A polynomial curve fit was then made of this. This enabled calculations to be made of the surface areas at each of the remaining planes when a corrected radius, \( r' \), determined from Pythagoras' theorem, was substituted for the true flame radius. The volume occupied by the burnt gases within the combustion space (above line BB) at any flame radius could, therefore, be obtained by the integration method described above.

When the flame radius, \( r \), is greater than \( c \) (see Fig. 81), the volume occupied by the burnt gases in the cylinder volume (below line BB) has to be calculated. The piston position has also now to be considered. As for the combustion space, the cylinder volume was divided into strips of very small thickness (0.1 mm) to achieve good accuracy. The surface areas at each of the radii drawn in the plan view were again determined by planimeter but now the limits to these measurements were
set by the cylinder volume walls. A polynomial curve fit was again made of these results. As before, this enabled surface area calculations to be made at any plane in the cylinder volume by modifying the radius, measured from line AA, accordingly. Burnt volumes could, thus, be obtained in the cylinder volume by integrating over the number of planes in the cylinder to which the burnt gas extends. In some cases, this is determined by the piston position.

The routines described above were programmed as a subroutine (SUBROUTINE BURNTVOL) for use in the computer simulated combustion program. A listing of this is given in Appendix 8.

Excellent accuracy was achieved with this method when comparisons were made of the measured known total volumes in the combustion chamber with the predictions of the program. Thus, the overall volume of the combustion space (above line BB in Fig. 81) was measured to be 50 cc exactly, whereas the program predicted a value of 50.22 cc. Some results from the program are plotted in Fig. 82 and include burnt gas volumes for

a) the combustion space alone at various radii, r, from the spark plug

b) the combustion space and cylinder volume at various radii, r, and distances, h', from cylinder head to piston position

The second major modification to the computer simulated combustion program, developed in Ref. 1, was the requirement to calculate the surface areas of the cylinder head, cylinder volume walls and piston in contact with the burnt mass of charge at various radii, r, of the flame front from the spark plug.

For the cylinder head surfaces, account was taken of the surface area of the retracted zone in which the spark plug is positioned. Measurements were also made of the vertical surfaces in the combustion space in contact
with the burnt charge at various radii and of the surface areas of the "squish" parts of the combustion chamber. The latter were determined by planimeter measurements off the plan view in Fig. 81.

Cylinder volume surface areas were computed by first measuring the circumferential lengths of the cylinder wall in contact with the burnt gases at the radii, r, in the s(3) plane (see Fig. 81). A polynomial curve fit was made of these so that such lengths could also be determined at any plane in the cylinder volume by modifying the radius, measured from line AA, accordingly. Cylinder volume surface areas were finally obtained by integrating these lengths over the number of planes involved. As for the burnt volume calculations, the number of planes is limited in some cases by the piston position.

The piston surface area in contact with the burnt gases is easily calculated from the surface area measured at the s(3) plane (see Fig. 81) but using a corrected radius corresponding to that in the plane in which the piston is situated and measured from the AA line.

The surface area measurements described above have been programmed as a routine (SUBROUTINE HTRAN) for use in the computer simulated combustion program (see Appendix 8).

Excellent agreement was again obtained between the measured total combustion chamber surface areas and those predicted by the program. Fig. 83 plots the surface areas of the cylinder head, cylinder walls and piston as a function of flame radius, r, at two heights, h', from the cylinder head to the piston.

No measurements were made of the combustion chamber surfaces in contact with the unburnt gas because this is assumed to behave isentropically.
7.3 Turbulent Burning in the "Squish" Combustion Chamber

In attempting to simulate the turbulent flame propagation process in a spark ignition engine combustion chamber, the main problem encountered is the lack of any suitable theoretical expression to predict rates of turbulent burning as a function of the parameters of the flow which are known to influence this process. The main requirement is that the expression should cater for the effects of

i) the fluctuating flow velocities, in many instances in a zero mean flow

ii) the scale of the eddies in the turbulent flow

iii) spectral distribution of turbulence

Additionally, the expression should be dependent upon air/fuel ratio, temperatures of the burnt and unburnt portions of the charge and pressure.

Such ideal requirements have not been combined into a useful expression as yet however. Indeed, the present situation regarding turbulent flame structure and propagation is somewhat inconclusive and, in many instances, contradictory. Thus, many turbulent burning expressions insist that the laminar burning velocity should form the basis of the expression, whilst others maintain that turbulent burning is completely independent of laminar burning and should be a function of the turbulence intensity and burnt gas temperature. This situation has arisen because turbulent flame propagation has been compared in completely different turbulent flows. Such comparisons should only be made when the parameters of the turbulent flow are similar e.g. intensity, scale of the eddies and spectral distribution.

The turbulent burning velocity expressions currently available have been documented by the author in a previous work (1). Broadly speaking, they are divisible into two main groups:
a) the Surface or Wrinkled Flame Front theories - in which the behaviour of a laminar flame front in large scale, relatively low intensity turbulence is considered. Under the influence of this, the laminar flame front is thought to undergo fluctuating deformations so that it becomes wrinkled. The surface area of the turbulent flame is, therefore, considered to be greater than that of the laminar flame to an extent which satisfies the greater part of the increase of turbulent flame speed over laminar flame speed. Among the theories in this group are those of Damköhler (86), Shchelkin (87), Karlovitz (88), Leason (89) and Scurlock and Grover (90).

b) the Volumetric or Three-dimensional theories - in these, the eddy diffusion in turbulent flow is thought to increase the transport properties of the two basic mechanisms of flame propagation viz. the thermal and diffusional mechanisms. In some of these theories (e.g. Smmerfield (91)), the turbulent flame front is considered to be a unified and continuous wave, just as in the laminar case, but made thicker by the action of the increased heat and active particle transportation due to the turbulent diffusion. In others (e.g. Sokolik (92) and Shchetinkov (93)), the increased diffusion is assumed to generate microvolumes within the turbulent flame front comprising mixtures of partial and total combustion products and fresh gases.

A turbulent burning velocity expression has also recently been proposed by Sanematsu (94). This is of the form

$$\frac{U_T}{U_L} = c' + d' \left( \frac{u'}{U_L} \right)^{L_y} b' \left( \frac{L_y}{L_b} \right)$$

(7-2)
where \( c' \) and \( d' \) are constants
\( u' \) is the r.m.s. fluctuating velocity
\( L_y \) is the integral scale of turbulence
and \( b' \) is a characteristic length of a grid used to generate the turbulence.

This expression, however, gives results that are incompatible with experimental findings and the use of
the parameter \( b' \) has no significance to the flow in spark ignition combustion chambers.

In view of the evident confusion that exists in discussions on turbulent flame propagation theory
therefore, it appears that little else can be achieved initially, in relation to the mechanism of
burning in the "squish" spark ignition combustion chambers, other than to analyse the flow in the chamber,
utilising the measured flow fluctuation velocities, and to try and indicate which mechanism of turbulent
burning is likely to be prevalent. In this connection, it should be noted that no measurements of the scales
of turbulence were made. This is beyond the capacity of hot wire anemometry in such a flow. However,
observations of flame photographs (95) indicate that the scale is very large compared with the thickness of a
laminar flame front.

To facilitate the above aim, the work of Shchelkin and Troshin (96) will be utilized. These authors considered in fundamental terms the turbulent burning process in a premixed fuel-air mixture of
varying scales and intensities. They determined that when the following criterion is satisfied

\[
\frac{L}{u'} > \frac{\delta}{U_L}
\] (7-3)

any tendency towards the volumetric mechanism of combustion, in which microvolumes are generated within
the turbulent flame front and mixing occurs between the burnt and unburnt parts of the charge within these
microvolumes, is completely absent. Therefore, the Surface Area or Wrinkled Flame Front mechanism predominates. In equation (7-3),

\[ L \] is the scale of turbulence
\[ u' \] is the fluctuating flow velocity
and \[ \delta \] is the width of the laminar flame front.

The criterion in Equation (7-3) is obeyed, consequently, when, during the time of rotation of an eddy, the flame has time to propagate at its laminar burning velocity, \( U_L \), a distance comparable to the width of the laminar combustion zone, \( \delta \). As can be seen from Equation (7-3), this occurs especially in a large scale, low intensity flow.

Alternatively, Shchelkin and Troshin considered that volumetric combustion takes place only when the mixing time between the burnt and unburnt gas, within a particular microvolume, is less than the chemical reaction time. That is, when

\[ \frac{L}{u'} < \frac{\delta}{U_L} \]  

(7-4)

This mechanism is thus seen to be satisfied when the scales of the eddies are small and the turbulence intensity is high. It is apparent from the parameters in the Shchelkin and Troshin criteria that only the turbulent flow conditions can alter the mechanism of turbulent burning. This is because \( \delta \) and \( U_L \) are virtual constants for a given fuel, air/fuel ratio etc..

Although the scale of the turbulence, \( L \), was not measured during the anemometry tests in the "squish" combustion chambers, the above criteria can still be applied to indicate which mechanism of turbulent burning exists. Typical values of the laminar burning velocity, \( U_L \), were obtained from previous computer runs (1) of the computer simulated combustion program. With iso-octane as the fuel and over a wide range of
air/fuel ratios, $U_L$ values were found to vary between 70 and 260 cm/sec at the conditions in the combustion chamber during the flame propagation period. Also, $\delta$ has been determined\(^{(97)}\) to be of the order of 0.02 cm.

Values of the fluctuating flow velocity, $u'$, have been measured in this work (see Chapter 6). Concentrating, initially, on the conditions most likely to generate the volumetric combustion mode (using the criterion in Equation 7-4), it is apparent that this has a tendency to occur when $u'$ is very high. Using, therefore, the highest $u'$ values from Fig. 67 for the 6.49:1 compression ratio, "squish" chamber, which occur at the highest engine speed, a typical value of $u'$ of 110 cm/sec is obtained. In Section 6.9, it was stated that a maximum error of 250% may be apparent in this $u'$ value. Consequently, the $u'$ magnitude in Equation 7-4 never exceeds 275 cm/sec over the engine speed range used in the anemometry tests in this work. Substituting the relevant values above in Equation (7-4) which are most likely to ensure that the Volumetric turbulent combustion mechanism exists and at the largest eddy size, $L$, we have

$$\frac{L}{275} < \frac{0.02}{70}$$

It is, therefore, apparent that the scale of the eddies, $L$, in the flow in the "squish" combustion chamber must be less than 0.078 cm for this condition to prevail. From observations of the flame propagation process in engine combustion chambers\(^{(95)}\), it is clear that the sizes of the predominant eddies, which control the propagation, are well in excess of this value. The conclusion must be, therefore, that the Surface Area or Wrinkled flame front mechanism prevails, at least over the engine speed range considered in this work.

Consequently, resort was made to a theory within this group to depict the rate of turbulent burning within the "squish" combustion chambers. As a first attempt, the expression of Shchelkin\(^{(87)}\) was used.
This is of the form

\[
\frac{U_T}{U_L} = \left(1 + \left(\frac{2}{U_L} \sqrt{u'_{\text{rms}}^2} \right)^2 \right)^{1/2}
\]  

(7-5)

and was developed by assuming that the breaking up of the laminar flame front under large scale turbulence resulted in the formation of a series of regular flame cones. When an expression such as this is applied to predict turbulent burning velocity values in engine combustion chambers, the problem arises as to what value of \( \sqrt{u'_{\text{rms}}^2} \) (equivalent to \( u' \) in the anemometry measurements in Chapter 6) to use, since it has been found to change with the frequency of the eddies in the flow (see Fig. 67). The following analysis was conducted to resolve this situation.

The flame travel time measurements made on the 6.49:1 compression ratio, "squish" chamber indicate that the number of crankangle degrees involved is about 30 (see Fig. 78). As a function of engine speed and frequency, Table 7 indicates the number of revolutions made by one eddy during a crankangle interval of 30°. Obviously, only those eddies of a certain frequency value that rotate at least twice during the period of flame travel are going to influence the flame propagation process appreciably. The lower frequency fluctuations probably cause some minor deformations at the flame front which are unlikely to be very significant. Based on the above criterion therefore, the maximum values of \( u' \), from Figs. 67a, b, c and d, which satisfied it were used in the Shchelkin expression (Equation (7-5)).

This expression was programmed into the computer simulated combustion model, and runs made over the engine speed range to determine how accurately it predicted the flame travel times when compared with the measured values in Fig. 78. In all cases, however, the
<table>
<thead>
<tr>
<th>Engine Speed (rev/min)</th>
<th>Number of Eddy Rotations During a Crankangle Interval of 30° at a Fluctuating Frequency of</th>
<th>125 Hz</th>
<th>250 Hz</th>
<th>500 Hz</th>
<th>1000 Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>1450</td>
<td></td>
<td>0.43</td>
<td>0.86</td>
<td>1.7</td>
<td>3.5</td>
</tr>
<tr>
<td>1200</td>
<td></td>
<td>0.52</td>
<td>1.0</td>
<td>2.0</td>
<td>4.0</td>
</tr>
<tr>
<td>950</td>
<td></td>
<td>0.66</td>
<td>1.32</td>
<td>2.64</td>
<td>5.3</td>
</tr>
<tr>
<td>750</td>
<td></td>
<td>0.84</td>
<td>1.68</td>
<td>3.35</td>
<td>6.7</td>
</tr>
</tbody>
</table>
predictions were far too high (i.e. flame speed values were too low). Consequently, the $u'$ values used in the Shchelkin expression were multiplied by the factor 2.5 (see Section 6.9) so that the maximum error correction was utilized. The flame travel time predictions were still much too high however. Thus, the Shchelkin expression was of no use in accurate calculations of turbulent burning velocity.

It should be noted that the computer simulated combustion runs were made under identical operating conditions to those under which the flame travel time measurements were made. The initial conditions at inlet valve closure, in the computer simulation runs, were set to

- pressure = 1.2 atm
- temperature = 400 °K
- mass fraction of exhaust = 5%
- residuals

The Karlovitz expression (88) was next used since this was noted to take into consideration the effects of flame generated turbulence. Much debate has been conducted over the years as to whether, in fact, this source of turbulence actually exists. Shchelkin and Troshin (96) have reviewed the position completely however and have indicated that it certainly does exist.

The Karlovitz theory of turbulent burning is based on instantaneous schlieren photographs of turbulent flames. The irregularities in the flame front were seen as acute angles pointing towards the products of combustion and rounded surfaces towards the unburned gas. The expression was developed especially for large scale, low intensity flow similar to that which is considered to exist in the "squish" combustion chamber. It is of the form

$$U_T = (2 \sqrt{\frac{-u'^2}{2}} U_L(1 - \frac{U_L}{\sqrt{u'^2}} (1 - \exp(- \frac{\sqrt{u'^2}}{U_L}))))^{1/2} + U_L$$

(7-6)
A clear and concise account of the manner in which additional turbulence can be generated by a turbulent flame in a large scale, low intensity flow is given by Karlovitz in Ref. 98. It is shown that the total intensity of the flame generated turbulence is given by

\[ \frac{\tilde{u}_{FG}^2 + \tilde{v}_{FG}^2 + \tilde{w}_{FG}^2}{\rho_b} = - \left( \frac{\rho_u - \rho_b}{\rho_b} \right)^2 \cdot \left( \frac{\rho_u - \rho_b}{\rho_b} \right) \cdot U_L \cdot \frac{U_T}{U_L}^2 \]  

(7-7)

where \( \tilde{u}_{FG}^2, \tilde{v}_{FG}^2 \) and \( \tilde{w}_{FG}^2 \) are the mean square values of the flame generated turbulent velocity components. The flow in "squish" combustion chambers has been shown to be more or less isotropic (see Section 6.8) and it is known that the quantity \( U_T/U_L \), in the second term on the right hand side of Equation (7-7), can be approximated by an expression similar to that in Equation (7-1). Indeed, the flame speed measurements of Hodgetts (99) for a "squish" combustion chamber, can be rearranged into the form

\[ \frac{U_T}{U_L} = 1 + 0.0017 n \]  

(7-8)

Consequently, Equation (7-8) was substituted into Equation (7-7). No such expression as Equation (7-8) could be derived from the flame speed measurements made in this work (see Section 7.1) because a second ionization probe is required to facilitate this in the close vicinity of the spark plug.

With the above considerations included, Equation (7-7) is reduced to

\[ \tilde{U}_{FG} = \frac{1}{\sqrt{3}} \left( \frac{\rho_u - \rho_b}{\rho_b} \cdot U_L \right)^2 - \left( \frac{\rho_u - \rho_b}{\rho_b} \cdot \frac{U_L}{1+0.0017n} \right)^2 \right]^{1/2} \]

(7-9)
and this was the expression used to express the effects of flame generated turbulence, $U_{FG}$.

In his original work (88), Karlovitz does not state how in fact $U_{FG}$ should be incorporated in his basic turbulent burning velocity expression (see Equation (7-6)). Shchelkin and Troshin (96) believe that it should simply be added on to this expression. However, in a later work (98), Karlovitz considers that the additional turbulence produced by the flame diffuses against the flow to the front of the flame reaction zone so that "the energies of the two random motions, that of the turbulence of the approach flow and that of the flame generated turbulence, are additive". Therefore, the $\sqrt{U'^2}$ term in Equation (7-6) comprises the turbulence of the flow and also that produced by the flame.

These concepts were programmed into the computer simulated combustion model to predict turbulent burning velocities.

Much better agreement was immediately obtained between the computer predicted flame travel time measurements and those obtained experimentally. However, deviations from the measured rate of increase of flame travel time with engine speed (see Fig. 78) were still apparent. Trial and error computer runs were then made using different values of $\bar{u}'$ over various frequency bands (see Fig. 67) to try and obtain the best agreement possible. It was found that, in order to obtain the same rate of increase in flame travel time with engine speed as was experimentally obtained, correction factors had to be incorporated in the $\bar{u}'$ values used. These were not based on the technique used by Semenov (3), described in Section 6.9, but were instead obtained from a re-analysis of the UV traces. This time, however, only the maximum values of $u'$, at each crank angle point over the 12 engine cycles analysed, were utilized. It is considered that these values are more accurate than the
average $u'$ values as it is probable that the hot wire is recording the eddy fluctuations normal to the hot wire and not at an angle to the normal flow direction. Generally, these maximum values were of the order of twice the average values, $\bar{u}'$.

Limitations were imposed on the number of trial and error computer runs conducted due to computer execution time restrictions but the indications were that the maximum flow fluctuation velocities, $u'$, calculated as described above and within the frequency range 500 to 1500 Hz, were primarily involved in the flame propagation process. Higher frequency fluctuations, over the engine speed range investigated, were found to be too low in magnitude to be significant whilst fluctuations of a lower frequency order have been shown to have little effect (see Table 7).

To be more specific than this is impractical due to errors that are known to exist in the measured values of the flow fluctuating velocities. In view of the results above, therefore, it can be stated that, providing the Karlovitz expression is reasonably accurate, then flame generated turbulence effects are extremely important in boosting up the turbulent burning velocities in spark ignition engine combustion chambers. Additionally, flow fluctuations of frequency between 500 and 1500 Hz, seem to be predominant in the flame propagation process over the engine speed range investigated.

The effects of the flame generated turbulence throughout a typical period of flame travel is shown, in a non-dimensionalized form with laminar burning velocity $U_L$, in Fig. 84. Also shown are the corresponding values of $U_r/U_L$. These results comply with Karlovitz's statement that "even at moderate turbulence intensity of the approach flow, the flame generated turbulence is more important than the turbulence of the
approach flow although some turbulence in the approach flow is necessary for the flame to generate turbulence".

The conclusions drawn from the 6.49:1 compression ratio "squish" chamber were then applied to the two lower compression ratios. The measured flame travel time intervals for these are given in Figs. 79 and 80. Again, the results on the UV traces were re-analysed in the manner described previously.

Utilizing suitably modified initial conditions at inlet valve closure in the computer simulation runs, the flow fluctuations within the frequency range 500 to 1500 Hz were once more found to be primarily involved in the flame propagation process.

These results are compatible with the findings of the anemometry results, where it is shown that only flow fluctuations within this frequency range and lower are significantly altered with change in engine speed. Since flame speeds increase with engine speed and are known to be a function of the velocity fluctuations in the turbulent flow, this general conclusion must be accurate. Additionally, a degree of trust can be placed in the Karlovitz expression (Equation (7-6)) for the rate of turbulent burning.
FIG. 74 — IONIZATION PROBE IN 'SQUISH' CHAMBER

FIG. 76 — COIL AROUND HIGH TENSION LEAD
FIG. 75 — IONIZATION PROBE ASSEMBLY
FIG. 77 — ENGINE RIG FOR FLAME SPEED MEASUREMENTS
FIG. 79—FLAME TRAVEL TIME MEASUREMENTS

'Squish' Combustion Chamber
Compression Ratio 5.29:1
Air/Fuel Ratio 12.8:1

Ignition Timing:
○ TDC
× 7° BTDC
+ 13° BTDC
△ 21° BTDC

Flame Travel Time (Crankangle Degrees)

Engine Speed (rev/min)
'Squish' Combustion Chamber
Compression Ratio 3.90:1
Air/Fuel Ratio 12.8:1

FIG. 80—FLAME TRAVEL TIME MEASUREMENTS
F.I.G. 81 — BURNT GAS VOLUME AND HEAT TRANSFER AREA EVALUATIONS
Combustion Space and Cylinder Volume (h = 0.526 cm)

Combustion Space and Cylinder Volume (h = 0.175 cm)

Combustion Space only

'Squish' Combustion Chamber

**FIG. 82 — BURNT GAS VOLUME VARIATION WITH FLAME RADIUS**
FIG. 83 - HEAT TRANSFER SURFACE AREA VARIATION WITH FLAME RADIUS
FIG. 84 — TURBULENT BURNING VELOCITY AND FLAME GENERATED TURBULENCE
CHAPTER 8

CONCLUSIONS
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CONCLUSIONS

The work undertaken has shown that a hot wire anemometry system can be used most effectively to indicate the spectral distribution of the turbulent flow fluctuations in spark ignition engine combustion chambers under "motoring" conditions. A necessary prerequisite, however, is a complete and thorough understanding of the behaviour of the anemometer in both steady and unsteady flows. Only when this is achieved can the anemometer be adjusted to give its optimum performance for its desired application.

The investigation was primarily concerned with the effect of combustion chamber design on the type of turbulent flow which exists in commonly used chamber forms. To this end, the two extreme cases have been considered i.e. a "squish" design and a cylindrical disc design. The period of the engine cycle during which flame propagation takes place was particularly studied.

Flow fluctuations up to a frequency of 5800 Hz have been recorded but the analysis was mainly concentrated in the low frequency region below 1000 Hz where most of the turbulent energy was known to be contained.

It has been shown that the main effect of a change in combustion chamber shape is an alteration in the magnitude of the flow velocities in the frequency range below 1000 Hz. The higher frequency fluctuations remain, to a large extent, relatively unaffected. This indicates that the energy which is transferred from the low frequency eddies to the higher frequency fluctuations during the compression stroke is offset by an equally high dissipation rate.
In all chamber forms studied, the low frequency fluctuations below 200 Hz increased in a linear manner with engine speed over the speed range investigated. Additionally, in the "squish" combustion chambers, the flow velocity fluctuations increased as the amount of "squish", instilled into the flow during the compression stroke, increased. This was entirely as expected and confirmed the results of a theoretical analysis.

A large amount of scatter was obtained on the results of the flow fluctuations, $\bar{u}'$, within the bandpass ranges studied. This was due to an insufficient number of engine cycles analysed and to the effects of the hot wire orientation relative to the eddies, the velocities of which were being measured. Because of this, it is not possible to be definitive about the frequency bandpass range in which most of the turbulent energy occurs nor to the rate of increase of $\bar{u}'$ with engine speed for the lower bandpass ranges studied. All that can be stated is that most of the turbulent energy is confined within the frequency range below 700 Hz.

The flow fluctuations, $\bar{u}'$, in the cylindrical disc combustion chamber were noted to be much lower than those in the "squish" chambers which incorporated the greatest amounts of "squish". This confirms the well-known fact that the rate of burning is much lower in such chamber designs compared with that in highly turbulent chambers.

From the limited tests conducted, the flow appears to be more or less isotropic in nature at the end of the compression stroke. Much more work will be required, however, using double wired probes for example, to confirm this.

Errors in the ensemble average values of $\bar{u}$ and $\bar{u}'$ over many engine cycles are inevitable for the reasons mentioned above. An estimate has been made of these which has indicated that $\bar{u}$ is probably too low by a maximum factor of 50% whilst the $\bar{u}'$ measurements are too
low by a factor of 2.5 at the most.

An analysis, based on the work of Shchelkin and Troshin (96), showed that the Surface Area mechanism of turbulent flame propagation exists in all the combustion chamber forms investigated over the engine speed range up to 1500 rev/min. A computer simulation of the flame propagation process in the "squish" combustion chamber design has indicated that excellent flame travel time predictions, compared with experimental measurements, can be obtained by using the turbulent burning velocity expression of Karlovitz (88). This incorporates a factor allowing for the concept of flame generated turbulence which was shown to be extremely significant in boosting up the turbulent burning velocities in spark ignition combustion chambers. In fact, it was found to contribute over half of the total turbulent burning velocity. This section of the work also showed that eddy fluctuations within the frequency range 500 to 1500 Hz were primarily involved in the turbulent flame propagation process.

The anemometry part of the work undertaken required the very greatest care at all times not only in relation to the probe handling but also in the signal processing system utilized. The latter took a considerable amount of time to perfect before a reliable system was achieved. It was imperative that accurate values of gas temperature, pressure and the anemometer output voltages be recorded for use in the computer program used to evaluate the flow velocities. The diligence with which this aim was pursued was a major contributory factor to the differences between chamber designs that have been recorded. The fact that the same hot wire probe was used for all the engine tests conducted was also extremely important in this connection.

The great drawback to this work was the time occupied in the analysis of the UV traces. Ideally, an automatic data recording system should have been used
incorporating an analogue-to-digital converter with the signal outputs eventually appearing on, for example, paper tape. This could then be fed into a computer, as data, for evaluations of the flow velocities utilising a program similar to that listed in Appendix 1. Some work was conducted along these lines when a Modular One analogue-to-digital system became available late in the time schedule for the research program. However, due to time considerations and the lack of certain essential equipment, this was not pursued to completion.
CHAPTER 9

SUGGESTIONS FOR FUTURE WORK
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SUGGESTIONS FOR FUTURE WORK

The extension of the work conducted here to much higher engine speeds is an obvious area for future study. However, it is suggested that an automatic data recording system be utilized to simplify the analysis that such work entails due to the reasons mentioned in the previous Chapter.

Another most fruitful area for future deliberations might be the generation of different turbulent flow conditions in engine combustion chambers during the flame propagation period than those that have been shown to exist at present. In particular, the attainment of a very high intensity, low scale turbulent flow may be conducive to engine operation at very weak mixtures. Some success along these lines has already been achieved by Stivender (100). The application of hot wire anemometry to such work could form a useful development aid. The basic concept involved would be to try and generate the volumetric mechanism of turbulent flame propagation (see Section 7.3). In this connection, the Shchelkin and Troshin (96) criterion in Equation (7-4) indicates that, if the turbulent fluctuating velocities can be greatly increased over a certain frequency range, then a fairly large eddy size may in fact ensure that this mode of flame propagation exists. The development of a turbulent burning velocity expression for such a flow would, however, be extremely difficult.

The recent appearance of commercially available laser anemometry systems (see Chapter 2) will find a ready application in the study of the flow in engine combustion chambers. Indeed, it may be possible to achieve accurate results in a firing engine.
In relation to computer modelling of the combustion process, the Karlovitz\textsuperscript{(88)} turbulent burning velocity expression has been shown to give excellent results under the flow conditions in the combustion chambers studied in this work. This may be because it was specifically developed for a large scale flow similar to that considered to exist in the engine combustion chambers. Its application to turbulent burning velocity predictions at different flow conditions, e.g. at higher engine speeds, remains to be determined.

A computer simulation of the combustion process should also include the effects of temperature gradients in the burnt gases and, also, chemical kinetics in order to predict exhaust emission concentrations more accurately. These are further areas for future work.
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COMPUTER PROGRAM FOR FLOW VELOCITY EVALUATION
I FOR
10C S(CARD, 1132PRINTER, DISK)
DIMENSION CN(30), THS(30), PRE(30), OL(20, 20), RHH(30),
1FLCO(7), RMSC(7), VC(30)
READ(2, 1) DL, WL
1 FORMAT(F7.5, F9.6)
ALPHA=0.001495
READ(2, 2) TM, RO, TO, TG, CR, SP
2 FORMAT(F2.9, 6*F6.2, *F7.4, F6.1)
READ(2, 3) PCAL, FCAL, RMSC, RMSC(6), JN
3 FORMAT(F7.4, 3*F6.4, 13)
READ(2, 16) KNJ
16 FORMAT(12)
TGX=1G
RTM=RTM-1.5+0.0185
RO=RO-1.5+0.0185
A=3.1412*D**2/4.
XIVC=240.
EVO=480.
XIVO=715.
EVC=10.
DO 5163 I=1, 5
5163 RMSC(1)=RMSC
RMSC(7)=1.0
EN=1.3
PRTIP=0.012
RES=RO/RAYL
RESO=RESD
R=50.
RC=1.77
RT=RTM+RG+RC
TM=((RTM/RO)-1.)/ALPHA+TO
RESS=RES*1.1+ALPHA*(373.-TO)
WRITE(3, 20)D
WRITE(3, 21)WL
WRITE(3, 22)TM
WRITE(3, 23)RO
WRITE(3, 24)TO
WRITE(3, 25)TM
WRITE(3, 390)SP
WRITE(3, 391)CR
20 FORMAT(19H WIRE DIAMETER(CM)=*F10.7/)
21 FORMAT(17H WIRE LENGTH(CM)=*F8.6/)
22 FORMAT(33H WIRE OPERATING RESISTANCE(OHMS)=*F8.5/)
23 FORMAT(28H WIRE COLD RESISTANCE(OHMS)=*F8.5/)
24 FORMAT(28H AMBIENT TEMPERATURE(DEG K)=*F6.1/)
25 FORMAT(35H WIRE OPERATING TEMPERATURE(DEG K)=*F8.1/)
390 FORMAT(23H ENGINE SPEED(REV/MIN)=*F6.1/)
391 FORMAT(19H COMPRESSION RATIO=*F6.2/)
DO 6217 I=1, 19
6217 READ(2, 6216) CN(1), THS(1)
6216 FORMAT(F5.1, *F6.1)
DO 5763 K=1, 7
5763 READ(2, 5764) FLCO(K)
5764 FORMAT(F6.4)
DO 2930 I=1, 19
2930 READ(2, 2931) PRE(I)
2931 FORMAT(F7.4)
DO 18 J=1, KNJ
536 FORMAT(1H END CONDUCTION TOTAL WATTAGE END CONDUCTION
1 TOTAL VOLTAGE/69H LOSS(WATT) SUPPLY(WATT) LOSS(WATT)
2VOLT(S)

537 FORMAT(F16.6,F19.6,F17.5/F)

853 FORMAT(2F19.6/F)

642 FORMAT(2F12.5/////)

PX=PR
TX=TG
V1=V2

5215 CONTINUE
TG=TX

18 CONTINUE
KN=KN-1

IF (JN) 5916,3916,5925

5916 CONTINUE
ADD=0.

DO 431 I=1,19
DO 432 J=1,KN

432 ADD=ADD+UV(1,J)

433 FORMAT(F10.2)
ADD=0.

431 CONTINUE

5925 CONTINUE

CALL EXIT

END
APPENDIX 2

LOW AIR FLOW VELOCITY CALCULATIONS

The equipment used to generate the low flow velocities (below 4 m/s) has been described in Section 4.7. It is shown in diagrammatic form in Fig. 28.

For fully developed, laminar flow in a circular pipe, the volume flow rate, \( F \), is given by

\[
F = \frac{\pi d_c^4}{128 \mu L} (P_1 - P_0)
\]  

(2-1)

where

- \( d_c \) = pipe internal diameter
- \( L \) = pipe length
- \( P_1, P_0 \) = pressures at either end of the pipe. In Fig. 28, \( P_1 \) is the reservoir pressure and \( P_0 \) is the ambient pressure.

Also

\[
\frac{dP}{dx} = -\frac{32 \mu \bar{u}}{d_c^2}
\]  

(2-2)

where \( \bar{u} \) is the mean flow velocity in the pipe given by

\[
\bar{u} = \frac{F}{\pi d_c^2}
\]  

(2-3)

and

\[
u = -\left(\frac{r_c^2 - r^2}{4\mu}\right) \cdot \frac{dF}{dx}
\]  

(2-4)
In Equation (2-4),

\[ u = \text{flow velocity at any point across tube} \]
\[ \text{cross-section (see Fig. 28)} \]
\[ r_c = \text{radius of pipe (}= \frac{d_c}{2}) \]

and \[ r = \text{distance from centre of pipe}. \]

Substituting Equation (2-1) in Equation (2-3),

\[ \bar{u} = \frac{d_c^2}{32 \mu L} (P_1 - P_0) \] (2-5)

and then putting Equation (2-5) in Equation (2-2),

\[ \frac{dP}{dx} = - \frac{(P_1 - P_0)}{L} \] (2-6)

Equation (2-4) then becomes

\[ u = \frac{(r_c^2 - r^2) \mu}{4 \mu} \cdot \frac{(P_1 - P_0)}{L} \] (2-7)

This has a maximum value at the centre of the pipe where \( r = 0 \) and is zero when \( r = r_c \).

When the hot wire of length, \( L \), is placed in this flow (see Fig. 28), the flow over the wire has a parabolic form. Its average value can be shown to be

\[ U = \frac{(P_1 - P_0)}{40 \mu L} \frac{d_c^2}{4} \cdot \frac{L^2}{12} \text{ cm/sec} \] (2-8)

in which

\[ (P_1 - P_0) = \text{pressure drop across tube (mm. H}_2\text{O)} \]
\[ \mu = \text{gravity (981 cm/sec}^2) \]
\[ L = \text{hot wire length (cm)} \]

Equation (2-8) was, therefore, used to evaluate the flow velocity, \( U \), emanating from the output end of the capillary tube.
APPENDIX 3
APPENDIX 3

THEORETICAL STUDY OF HOT WIRE PROBE VIBRATION

This was conducted to determine the probe's natural frequencies so that note may be made of any such frequency occurring within a particular bandpass range which might lead to misleading measurements of air flow fluctuations in that range. Vibrations of the probe, when fitted into the engine combustion chamber in the manner shown in Fig. 38, occur as a result of the piston and crankshaft motions transmitting periodic disturbing forces to the cylinder head which are then conveyed to the probe.

Fig. 38 shows that a cantilevered structure can be effectively considered since the probe is pinned down by the brass sleeve and countersunk screw. The following equation applies (74)

\[ \frac{d^4 y}{dx^4} - s^4 y = 0 \]  

(3-1)

where

- \( y \) is the deflection of the probe from its normal position at any distance \( x \) from the pinned end
- \( s^4 = \frac{m \cdot \omega^2}{EI} \)  

(3-2)

in which

- \( m \) is the mass per unit length of the cantilevered section of the probe (0.17 gm/mm)
- \( \omega \) is the frequency
- \( E \) is the Modulus of Elasticity
- \( I \) is the moment of inertia of the probe evaluated from

\[ I = \frac{\pi}{64} \left| d_o^4 - d_i^4 \right| \]
In the latter expression, \( d_0 \) is the external diameter of the probe body tube (5 mm) and \( d_1 \) the internal diameter (4 mm).

The solution to Equation (3-1) is

\[
y = A_1 \cos sx + A_2 \sin sx + A_3 \cosh sx + A_4 \sinh sx
\]  

(3-3)

where \( A_1, A_2, A_3 \) and \( A_4 \) are constants.

Applying the boundary conditions,

at \( x = 0 \), \( y = 0 \)
at \( x = 0 \), \( \frac{dy}{dx} = 0 \)
at \( x = l \), \( \frac{d^2y}{dx^2} = 0 \)
at \( x = l \), \( y_{\text{max}} = \frac{m_k^4}{8EI} \) and \( \frac{dy}{dx} = \frac{m_k^3}{6EI} \)

where \( l \) is the cantilevered length (7.6 cm), it can be shown that

\[
\cosh s_l \cos s_l = -1
\]  

(3-4)

The values of \( s_l \) which satisfied this equation were found by consulting tables of hyperbolic and trigonometric functions. The first five were

\[
\begin{align*}
s_1 &= 1.875 \\
s_2 &= 4.694 \\
s_3 &= 7.855 \\
s_4 &= 10.996 \\
s_5 &= 14.137
\end{align*}
\]

Substituting these in Equation (3-2), only the two lowest values of \( s_l \) above gave natural frequencies within the bandpass ranges of the flow velocity fluctuations being investigated in the engine combustion chambers. These calculated natural frequency values were 451 Hz and 2820 Hz.
APPENDIX 4
The filter referred to is that in Auxiliary Unit 1 in Fig. 42. Its characteristics were determined with it in the circuit so that its actual operating conditions were simulated. A reference 100 Hz sine wave from the oscillator was fed to the Auxiliary Unit and the amplifier gain adjusted on this unit so that the input and output levels were the same at this mid-frequency of the 200 Hz range.

The results obtained in terms of attenuation and phasing characteristics are shown in Fig. A-1.
FIG. A-1 — AUXILIARY UNIT FILTER CHARACTERISTICS
APPENDIX 5

CHARACTERISTICS OF DAWE BANDPASS FILTER UNIT

The bandpass ranges of the Dawe Filter Unit used in this work are given in Section 5.4. A reference 125 Hz sine wave signal (the mid-frequency of lowest range) was fed from the oscillator to the Dawe Filter Unit in the circuit shown in Fig. 42 and the amplifier gain on Auxiliary Unit 2 adjusted so that the output from this unit equalled the input to the Dawe Unit. The filters on Auxiliary Unit 2 were by-passed during these tests.

With the amplifier gain setting on Auxiliary Unit 2 thereafter remaining unaltered, the bandpass filter characteristics in all the ranges were determined relative to the 125 Hz reference signal. These are plotted in Fig. A-2. The small attenuation differences between the ranges were noted and used to adjust the standard calibration at 125 Hz during the engine tests (see Section 5.4).
FIG. A-2—DAWE BANDPASS FILTER CHARACTERISTICS
DETAILS OF COMARK ELECTRONIC THERMOMETER

Type 1601 - suitable for use with Cr/AI thermocouples

Cold Junction - 0 to 40 °C Ambient
Correlation Range

Temperature Ranges:
- 87 °C to 25 °C
- 0 to 100 °C
- 0 to 300 °C
- 0 to 1000 °C

Temperature Coefficient: < 0.05% per °C

DC output: 1 volt for full-scale deflection.
2 mA maximum current
APPENDIX 7
APPENDIX 7

THEORETICAL ANALYSIS OF "SQUISH" VELOCITIES IN COMBUSTION CHAMBERS

The analysis is due to Lichty \(^{(76)}\) and, as described here, is based on the combustion chamber form shown in Fig. 60.

Incompressible flow is assumed and the object is to determine the velocity with which the gas between the piston and the "squish" part of the combustion chamber is ejected as a function of the relevant parameters \(h\), \(A\), \(S\), \(b\) and \(b'\) (see Fig. 60).

The instantaneous charge density, \(p\), as a function of piston position \(S\) from top dead centre is given by

\[
p = \frac{\rho_1}{1} \frac{|S_1/(CR - 1)|CR}{|S_1/(CR - 1)| + S}
\]

\[
= \frac{\rho_1^B}{C + S} \quad (7-1)
\]

where

- \(B\) and \(C\) are constants for a given engine stroke and compression ratio
- \(CR\) = compression ratio
- subscript 1 indicates conditions at the start of compression

The mass of mixture, \(m'\), in the volume between the "squish" part of the cylinder head and the piston is

\[
m' = \rho A(S + h) \quad (7-2)
\]

Substituting for \(p\) from Equation (7-1) gives

\[
m' = \rho_1^AB \frac{(S + h)}{(C + S)} \quad (7-3)
\]
This mass is ejected from the "squish" part of the combustion chamber at a rate determinable from the differentiation of Equation (7-3) with respect to crankangle $\theta$. Thus

$$\frac{dm'}{d\theta} = AB\rho_1 \left| \frac{(C+S) \frac{dS}{d\theta} - (S+h) \frac{dS}{d\theta}}{(C+S)^2} \right|$$

and this is ejected through the area $b'(S+h)$ — see Fig. 60.

Also,

$$\frac{dm'}{d\theta} = \rho b'(S+h) V'$$

Consequently, with $\rho$ given by Equation (7-1), and Equation (7-5) substituted in Equation (7-4), the following expression results for the velocity of mixture ejection, $V'$.

$$V' = \frac{A}{b'(S+h)} \left( \frac{dS}{d\theta} - \frac{S+h}{C+S} \frac{dS}{d\theta} \right) \text{ cm/rad}$$

Multiplying this by $2\pi n$ for an engine speed $n$(rev/sec) gives $V'$ in units of cm/sec.

Lichty(76) indicates that $S$ can be approximated by

$$S = r'(1 - \cos \theta) + \left( r'^2 / 4l' \right) (1 - \cos 2\theta)$$

where $r'$ and $l'$ are the radii of the crank circle and connecting rod respectively.

Consequently,

$$\frac{dS}{d\theta} = r'\sin \theta + \left( r'^2 / 2l' \right) \sin 2\theta$$
APPENDIX 8

COMPUTER PROGRAM SIMULATING THE COMBUSTION PROCESS IN VAUXHALL WYVERN ENGINES.
FUNCTION CVINIT()
U=0.29815
CV0=0.02853*U+0.08886*U**2+1.16526*U**3-1.77671*U**4+1.2422*U**5-0
1+20161*U+6*U+0.5739*U**7
U=U+1
CV1=CV0+0.5739*U**7
RETURN
END

FUNCTION CVINIT()
U=0.29815
CV0=0.25416*U-0.061495*U**2+0.116066*U**3-0.07853*U**4+0.021734*U**5-16*0.05034*U**6+6*0.0037066*U**7
U=U+1
CV1=CV0+0.0037066*U**7
RETURN
END

FUNCTION CVINIT()
U=0.29815
CV0=0.45596-U-1.52545*U**2+1.70833*U**3-0.58783*U**4+7.09294*U**5-13*1.9322*U**6+6*0.560262*U**7
U=U+1
CV1=CV0+0.560262*U**7
RETURN
END

FUNCTION CV01()
U=0.29815
CV0=0.25416*U-0.061495*U**2+0.116066*U**3-0.07853*U**4+0.021734*U**5-16*0.05034*U**6+6*0.0037066*U**7
U=U+1
CV1=CV0+0.0037066*U**7
RETURN
END

FUNCTION CV01()
U=0.29815
CV0=0.25416*U-0.061495*U**2+0.116066*U**3-0.07853*U**4+0.021734*U**5-16*0.05034*U**6+6*0.0037066*U**7
U=U+1
CV1=CV0+0.0037066*U**7
RETURN
END

FUNCTION CPPINIT()
U=0.29815
CV0=0.45596*U-1.52545*U**2+1.70833*U**3-0.58783*U**4+7.09294*U**5-13*1.9322*U**6+6*0.560262*U**7
U=U+1
CV1=CV0+0.560262*U**7
RETURN
END

FUNCTION CPPINIT()
U=0.29815
CV0=0.25416*U-0.061495*U**2+0.116066*U**3-0.07853*U**4+0.021734*U**5-16*0.05034*U**6+6*0.0037066*U**7
U=U+1
CV1=CV0+0.0037066*U**7
RETURN
END

FUNCTION CPP01()
U=0.29815
PROCP0=0.45596-3.0509*U+17.125*U**2-34.3513*U**3+35.1547*3393*U**4+3.92978*U**6
RETURN
END

FUNCTION CP0100()
U=0.29815

FUNCTION BENC (1)
U=U0*1
SENC=0.02873*U+0.49579*U**2+7.1084*U**3+6.21113*U**4-2.5
RETURN
END

FUNCTION AHCP (1)
U=U0*1
AHCP=0.25416-0.12699*U+0.348198*U**2-0.31412*U**3+0.13867*U**4-0
RETURN
END

FUNCTION CPMEAN(u)
X=U0*1
CV0=0.25416*X-0.05495*X**2+0.116066*X**3-0.07853*X**4+0.027734*X**5
RETURN
END

FUNCTION SENC
U=U0*1
SENC=0.02873*U+0.49579*U**2+7.1084*U**3+6.21113*U**4-2.5
RETURN
END

FUNCTION AHCP
U=U0*1
AHCP=0.25416-0.12699*U+0.348198*U**2-0.31412*U**3+0.13867*U**4-0
RETURN
END

FUNCTION CPMEAN(u)
X=U0*1
CV0=0.25416*X-0.05495*X**2+0.116066*X**3-0.07853*X**4+0.027734*X**5
RETURN
END

RETURN
END
APPENDIX 9

THE SEMENOV LAMINAR FLAME PROPAGATION THEORY

The Semenov laminar flame propagation theory is more comprehensive than many other such theories since it attempts to cater for both the thermal and diffusional mechanisms involved in the propagation of a flame. However, in its final, simplified form, it includes the diffusion of fuel molecules but not free atoms and radicals. As a result, it tends to emphasize the thermal mechanism. The original, full derivation of this theory is given in Ref. 80 whilst Ref. 1 reviews this derivation.

A one-dimensional, steady state combustion model is assumed and the reaction is considered to be second-order and bi-molecular. The resulting equation for the laminar flame propagation velocity, $U_L$, is

$$U_L = \frac{n_u}{n_b} \cdot \beta \left[ \frac{2}{\lambda_B} \frac{W Z C_{f_u} (1 - \frac{1}{\phi}) \exp(-E/RT_B)}{\sqrt{u^3 C_{p_B} \phi \frac{C_{f_u}}{C_{O2_u}}}} \right]^{\frac{1}{2}}$$

(9-1)

where

- $\frac{n_u}{n_b}$ is the ratio of moles of reactants to products in the stoichiometric equation
- $\beta = \frac{R T_B^2}{E (T_B - T_u)}$

in which

- $R$ is the Universal Gas Constant
- $T_B$ is the equilibrium flame temperature.
- $T_u$ is the unburnt gas temperature.
- $E$ is the Activation Energy.
- $C_{p_B}$ is the specific heat at constant pressure evaluated at temperature $T_B$.
- $D_B$ is the diffusion coefficient evaluated at $T_B$. 
\( \lambda_B \)

is the thermal conductivity at temperature \( T_B \).

\( C_{fu} \)

is the fuel concentration in the unburnt gas.

\( W \)

is a steric or probability factor.

\( \phi \)

is the equivalence ratio.

\( \rho_u \)

is the unburnt gas density.

\( \frac{C_{fu}}{C_{O_2 u}} \)st

is the stoichiometric fuel-oxygen ratio

and

\[
Z = \frac{\left( \frac{d_{col. f}}{2} + \frac{d_{col. o_2}}{2} \right)}{\left( \frac{8 \pi RT}{A} \right) \left( \frac{M_f + M_{O_2}}{M_f M_{O_2}} \right)^{\frac{1}{2}}}
\]

in which

\( d_{col. f} \)

is the effective collision diameter of the fuel.

\( d_{col. o_2} \)

is the effective collision diameter of the oxygen.

\( A \)

is Avogadro's number.

and

\( M_f, M_{O_2} \)

are the molecular weights of the fuel and oxygen respectively.

For lean mixtures, the term \((1 - \frac{1-\rho}{\phi})\) is replaced by \((1 - \phi(1-\rho))\) and, for stoichiometric mixtures, it becomes simply \(\rho\).

The detailed application of equation (9-1) to the computer simulated combustion model is described in Ref. 1. The main problem encountered was the evaluation of the steric factor, \( W \). To achieve this, experimentally determined, absolute values of the laminar burning velocity were required. No such values could be found, after an extensive literature search, for a combustible mixture of premium grade gasoline and air.

However, experimental measurements, conducted according to the techniques outlined in Chapter 7 (Section 7.1), indicated no difference between the flame travel time measurements when using iso-octane and premium grade gasoline.
Consequently, the laminar burning velocity values of iso-octane-air mixtures must be of virtually the same order as premium grade gasoline-air mixtures and were thus used to evaluate the steric factor, W. These $U_L$ values were obtained from the following literature source:


A slight drawback to the use of equation (9-1) in predicting $U_L$ values is that it dictates that pressure has no effect whereas the literature tends to report that there might be a small effect in some instances. The exact pressure dependence is difficult to gauge accurately since it depends so much on the experimental equipment used. Additionally, the fuel type appears to have great significance with some combustible mixtures showing a positive, some a negative and others no effect of pressure on $U_L$. For some hydrocarbon-air flames, there does appear to be a trend of a decrease in laminar burning velocity with increasing pressure at very low pressures (2 atm). However, as far as can be ascertained, there is no definitive law (that is agreed between a number of investigators over a wide range of pressures up the 50 atm) which can be affixed to this behaviour. In this connection, even the hydrocarbon type seems important. Thus, an acetylene-air mixture seems insensitive to pressure whilst a propane-air mixture has a negative dependence of pressure on burning velocity (97).

Clearly, in computer simulations of the flame propagation process in spark ignition engines, there is a real need to determine the unique dependence of this pressure effect for all fuels commonly used in these engines and over a wide range of air/fuel ratios since this variable could also be significant.