Some Diagnostic Methods Applied to Air-Fuel Mixing Processes in Internal Combustion Engines, and Their Phenomenological Modelling

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ABSTRACT

This paper presents the application of the principles of hot wire anemometry to the determination of mean air motion and turbulence in spark ignition and diesel engines, and the extension of these principles to the determination of concentration in binary (gas-air) mixtures. Application of this technique to the study of air-fuel mixing in a simulated diesel engine combustion chamber is described, and the relevant scaling parameters evaluated.

To study fuel concentration during evaporation in the ignition delay and post delay period in a quiescent chamber diesel engine, the application of double exposure holographic interferometry is outlined, and a suitable method of scaling engine conditions to those in a combustion bomb are described. Finally, the application of holographic interferometry to the preliminary study of transient gas injection into a narrow cavity to produce an asymmetric flow and concentration field is discussed, and preliminary concentration profiles presented.

1. INTRODUCTION

The influence of in-cylinder charge turbulence and fuel distribution on the combustion process in spark ignition and diesel engines is now well recognised and exploited in "lean burn" spark ignition engines(1) and direct injection diesel engines, utilising a high level of "squish" induced turbulence(2) to achieve good fuel economy and exhaust emissions.

The problem of measuring charge velocity, turbulence, and fuel concentration in an engine cylinder, has been the subject of active research over the past decade, and has been approached with a range of techniques. The objective of the present paper is twofold. Firstly, to indicate those flow parameters which appear to be of significance in spark ignition and diesel engine combustion processes, and to show, after a short review of the application of hot wire anemometry to engine flow velocity and turbulence measurements, how this technique may be used in conjunction with associated electronics to measure engine turbulence parameters of interest in real time, thereby providing a tool to facilitate test bench development of an engine. The second objective of the paper is to indicate how hot wire anemometry and holographic interferometry may be applied to determine in-cylinder fuel distribution in diesel engines, and hence obtain empirical information on fuel spray development and air entrainment.

2. TURBULENT FLUID MOTION AND ITS CHARACTERISATION

In the turbulent flow of a fluid, random fluctuations of the fluid velocity are superimposed on the mean fluid motion. An Eulerian description of the flow field is usually adopted in characterising the turbulence, which is convenient for measurement purposes.

The spatial characteristics of the turbulence are described through the macro or integral scale (L), the micro or Taylor scale (\lambda) and the dissipation or Kolmogorov scale (n), which represent the average size (L) of the turbulent eddies in the flow, the smallest measurable eddy size (using hot wire techniques) (\lambda) into which the larger eddies break up before eventual dissipation in the very fine eddy structure (n) by viscous effects in the fluid. The Kolmogorov scale (n), though not directly measurable, is inferred from the Taylor scale (\lambda) and the kinematic viscosity of the fluid.

The random turbulent fluctuations in fluid velocity is characterised by the intensity \( u'^2 \), which is the r.m.s. value of the velocity fluctuation \( u \).

Turbulence can be considered to be made up of many superimposed periodic motions of the fluid, therefore, a further characterising parameter is the distribution of the kinetic energy of the turbulence \( u'^2 \) with frequency \( f \) throughout the flow field. The kinetic energy at each particular frequency is usually normalised with the intensity squared \( u'^2 \) to yield the spectral density function \( F(f) \).

Because of the random nature of turbulent motion, the quantitative description of its characteristics in terms of the parameters mentioned above, can only be made in statistical terms, and to assist this it is usually assumed that the random turbulence is stationary, thereby
ensuring the time invariance of the statistical properties of the turbulence. This can be approximated in practice if the characteristic time scale of the turbulence ($\lambda_t$) is smaller than the interval ($\Delta t$) over which the turbulence is studied, which in turn must be smaller than the period ($t_p$) of any slow variation in the mean flow field (3) so that

$$\lambda_t < \Delta t < t_p$$  \hspace{1cm} (1)

The macro or integral scale of turbulence ($L$) is defined as the integral of the correlation coefficient $\rho$ of the fluctuating velocity at two adjacent points in the flow, with respect to a variable distance between them. For the direction $(X)$ parallel to the mean flow ($\overline{U}$), $L_X$ would be

$$L_X = \int_0^L R_X dX = \int_0^\infty \overline{u_0^*-u_X^*} dX$$  \hspace{1cm} (2)

where $u_0^*$ and $u_X^*$ is the time average of the product of the turbulent velocity fluctuation $u$ measured at points 0 and $X$, and $\overline{u_0^*-u_X^*}$ is the time average of $u_0^*$ and $u_X^*$. The micro or Taylor scale ($\lambda_X$) is defined in terms of the correlation coefficient curve at its origin, and is

$$\lambda_X^2 = -2/\left(\partial^2 R_X / \partial X^2\right)_{X=0}$$  \hspace{1cm} (3)

Similarly for the direction $(Y)$ perpendicular to the mean flow $\overline{U}$, length scales $L_Y$ and $\lambda_Y$ may be defined. If, in addition to assuming that the turbulent field is stationary, it is assumed to be homogenous and isotropic, then the length scales $L_X$, $L_Y$, and $\lambda_X$, $\lambda_Y$ are simply related (3) so that

$$L_X = 2L_Y$$

$$\lambda_X = \sqrt{2}\lambda_Y$$ \hspace{1cm} (4)

Assuming that the turbulence is stationary, homogeneous and isotropic enables further relationships to be established through the use of macro or integral ($L$) and micro ($\lambda$) time scales, based on the auto-correlation coefficient ($R_T$). Hence

$$L_X = \overline{\dot{U}} L_t$$

$$\lambda_X = \overline{\dot{U}} \lambda_t$$  \hspace{1cm} (5)

where

$$L_t = \int_0^T R_T dt = \int_0^T \overline{u_t^*-u^*} dt$$

$$\lambda_t = \int_0^T R_T dt = \int_0^T \overline{u_t^*-u^*} dt$$ \hspace{1cm} (6)

where $\overline{u_t^*-u^*}$ is the time average of the product of the turbulent velocity fluctuation $u_t$ measured at a fixed point at an instant $t$, and a delayed version of itself $u^*$ where $T$ is a variable time delay. $\overline{u_t^*}$ is the time average of $u_t$. The form of the auto-correlation curve is similar to that of $R_T$, $\lambda_t$ is related to the curvature of the auto-correlation curve in a manner similar to $\lambda_X$ and $R_X$ in equation (3).

In physical terms $L_t$ can be considered to be the lifetime of a turbulent eddy, and $\lambda_t$ as being inversely proportional to the highest measurable frequencies in the turbulent velocity field. The inverse of the Kolmogorov time scale ($t$) which is directly related to $u_0^*$ and $\lambda_X$ can be considered a measure of the mixing rate at the molecular level, at which chemical reaction occurs.

From the foregoing discussion, it can be seen that the measurement of the turbulent velocity fluctuation at a single location can, with suitable signal processing, yield the parameters which characterise the turbulence at that point.

3. APPLICATION OF HOT WIRE ANEMOMETRY TO TURBULENCE MEASUREMENT

The good frequency response characteristics and spatial resolution of the constant temperature hot wire anemometer make it suitable for turbulent flow measurements in a wide range of applications. Additionally, low cost and ease of use make it advantageous as a development tool in the internal combustion engine context.

In principle, the hot wire anemometer employs the relation between convective loss from a heated wire to the velocity of the gas flow over it. The instrument consists of a Wheatstone Bridge for sensing the out-of-balance voltage which results from the cooling of a fine heated wire (10 $\mu$m dia x 2 mm length) of high temperature coefficient of resistance, which forms one arm of the bridge circuit. The out-of-balance signal from the bridge is amplified, and a suitable feedback current passed to the wire to maintain its temperature (resistance) constant.

The anemometer is calibrated in a wind tunnel having a low level of turbulence, to obtain the bridge output voltage as a function of tunnel flow velocity. This calibration (which is non-linear) is utilised to obtain the magnitude of tunnel flow fluctuations. This procedure is permissible (3) provided the turbulent intensity does not exceed the mean flow velocity by about 30%. If this limit is exceeded, corrections have to be applied to the measured voltage levels. Similarly, corrections have to be applied for the averaging effects of finite hot wire length on the measured intensity and scales of turbulence. Corrections to the wind tunnel calibration of the hot wire for variable gas temperature and pressure encountered in a motored engine cylinder need to be undertaken. These techniques are discussed in detail elsewhere(4),(5), where typical hot wire calibration data are presented in terms of non-dimensional parameters (Nusselt, Reynolds and Prandtl Numbers) for forced convective heat transfer from heated cylinders in air with a high surface-to-air temperature ratio.

While calibration of the hot wire is essential for the determination of mean velocity $\overline{U}$, turbulence intensity and micro scale $\lambda$, it is possible to obtain information on the mixing rate $1/\lambda_t$ without calibration of the hot wire, or information concerning its spatial orientation to the flow vector. This is an important point in the context of engine development applications and will be dealt with later.
4. GENERATION OF TURBULENCE IN ENGINES AND ITS EFFECT ON COMBUSTION

Turbulence in the internal combustion engine is generated by jet mixing produced during the induction process, and the so-called squash generated air motion due to piston movement close to top dead centre (t.d.c.) on the compression stroke, forcing the gas charge into a small combustion space in the cylinder head or piston crown.

In the large low speed diesel engine, turbulence created during the induction process has essentially decayed in the time the piston has reached the vicinity of top dead centre on the compression stroke. Because of the low squash produced in this type of engine — open bowl combustion chamber — the sole mechanism for producing turbulence is the fuel injection system, hence the requirements for high injection pressure and rates.

In the spark ignition engine the induction jet and squash mechanisms produce turbulence prior to ignition. However, the propagating flame front creates a flow field ahead of it in the unburned charge, and the turbulence created in this field can be exploited to accelerate combustion(7).

Some of these points are briefly illustrated with reference to the results of Dent and Salama (6) shown in Fig. 1, which presents typical hot wire anemometer signals representative of mean motion (\( \bar{U} \)) and turbulence (\( u' \)) obtained on two engines having differing combustion chamber geometries. In both cases the anemometer probe was located at the spark plug. The strong jet mixing effects during induction are apparent on \( \bar{U} \), in (a) the decay of \( \bar{U} \) and \( u' \) are apparent during the compression period exhibiting little effect of combustion chamber geometry which is of the low squash type. The wedge chamber geometry of Fig. 1(b) shows the strong effect of squash in generating mean motion \( \bar{U} \) and turbulence \( u' \).

It is appropriate at this point to consider those parameters of the turbulent in-cylinder charge motion which have a direct bearing on the combustion process.

In the spark ignition engine the combustion process is of the pre-mixed type in which combustion initiated by a spark propagates through the air-fuel mixture. From Lancaster et al.(8) it is seen that the turbulent flame propagation speed increases directly with increase in \( u' \), where increase in \( u' \) is achieved through higher volumetric flow rate of charge through the engine cylinder, or through the use of inlet valve masking. Alternatively, the generation of turbulence (\( u' \)) by squash induced charge motion, will also cause an increase in flame propagation speed(9). The modelling studies of McCulloch et al.(9), Tabaczynski et al.(10) and Hires et al.(11) indicate the importance of turbulent scales in addition to intensity \( u' \) in characterising the burning time of a turbulent eddy (\( t_1 \)). Hires et al.(11) expressed this as

\[
\tau_b = \text{constant} \left( \frac{u'}{\nu} \right)^{\frac{1}{3}} \left( \frac{\nu}{S_L} \right)^{\frac{1}{6}}
\]

(7)

For isotropic turbulence, the Taylor and Integral scales are related by

\[
\lambda/L = (15)^{\frac{1}{3}} \left( \frac{u'/\nu}{L} \right)^{\frac{1}{2}}
\]

(8)

Hence

\[
\tau_b = \text{constant} \left( \frac{\lambda}{\nu S_L^2} \right)^{\frac{1}{2}}
\]

(9)

where \( \nu \) is the kinematic viscosity and \( S_L \) the laminar flame speed.

The extensive results of Mattavil et al.(13) for open (low \( \lambda \)) and wedge (high \( \lambda \)) chambered engines lend strong support for the dependence of \( \tau_b \) on \( \lambda \) expressed in equation (9).

The study of turbulence structure — intensity \( u' \) and Taylor scale \( \lambda \) — in a motored direct injection diesel engine(14) was undertaken with the object of showing the influence of intake port geometry on the turbulence level, and hence on the combustion and emissions from the engine under normal operating conditions. The data of reference (14), which covers a range of engine speed, inlet port geometry and injection timing, are shown in Fig. 2(a) as plots of intensity \( u' \) and micro scale \( \lambda \) against exhaust smoke level. The studies of Magnusson and co-workers(15)(16) consider soot oxidation rate to be characterised by turbulent dissipation at the Kolmogorov level. The author(17) has used the turbulent energy dissipation rate (\( E \)) and the Kolmogorov time scale (\( t \)) which can be expressed(12) as

\[
E = 15 \nu (u'/\lambda)^2
\]

(10)

\[
\tau = (\nu/E)^{\frac{1}{2}}
\]

(11)

from which a dissipative mixing rate (\( 1/\tau \)) can be obtained so that

\[
1/\tau = 3.87 (u'/\lambda)
\]

(12)

The data from Fig. 2(a) may now be replotted as exhaust smoke level (Bosch) versus (1/\( \tau \)), this is shown in Fig. 2(b).

Employing the fact that \( E \) can be expressed in physical terms as \( E = \text{constant} \left( \frac{\nu}{\lambda} \right)^{\frac{1}{2}} \left( \frac{N/\theta_p}{Q/\pi \rho \nu^3} \right)^{\frac{1}{2}} \left( \frac{\theta_p}{\nu \rho d} \right)^{\frac{1}{2}} \)

(13)

where \( N \) is the engine speed, \( \theta_p \) the injection period in degrees crank angle, \( Q \) the volume of fuel injected per injection stroke, \( n \) the number of holes in the injector nozzle, and \( d \) their diameter. Data for exhaust smoke emission and mixing rate for a quiescent chamber diesel engine has been shown, for a given fuel density, to be(17)

\[
1/\tau = \text{constant} \left( \frac{1}{\nu} \right)^{\frac{1}{2}} \left( \frac{N/\theta_p}{Q/\pi \rho \nu^3} \right)^{\frac{1}{2}} \left( \frac{\theta_p}{\nu \rho d} \right)^{\frac{1}{2}}
\]

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From the foregoing discussion it can be seen that any proposed scheme for measuring in-cylinder turbulence in an engine, should have the capability for determining the intensity $u'$ and the micro scale $\lambda_u$.

5. HOT WIRE ANEMOMETER TURBULENCE MEASUREMENTS IN ENGINES

Application of hot wire anemometry to in-cylinder measurements of engine turbulence has, in more recent times, made extensive use of data acquisition and processing facilities, though individual investigators have differed in the detail of their approach.

The local velocity at any instant will comprise low frequency periodic components due to piston motion in addition to a random turbulent component. Therefore ensemble averaging has been a valuable procedure for flow and turbulence $(u')$ velocities at any instant of the engine cycle. This technique has been applied by Witze(19). An alternative procedure is to employ a high pass filter to remove the low frequency periodic component and process the remaining random signal using time averaging techniques. This has been adopted by Semenov(20) using analogue processing equipment, and by Dent and Salama employing digital data processing techniques(6).

Semenov used a hot wire anemometer in conjunction with a resistance thermometer for determining gas temperature. Correction for the varying gas temperature in the motored engine cylinder was applied automatically through an analogue network, so that wind tunnel calibration of the wire could be used. The temperature corrected continuous output voltage from the anemometer bridge was ganged every cycle to produce a sample $(\delta)$ having a width of $24^\circ$ crank angle around the crank angle $\delta$ of interest. The time average of the flow velocity was obtained over 20 to 40 consecutive cycles. The high pass filtered signal $(u)$ was fed either through a suitably selected narrow band filter (for spectral analysis) on to an r.m.s. voltmeter or directly to the r.m.s. meter to yield $u'$.

Dent and Salama(6) recorded the anemometer output voltage on FM tape for subsequent computer processing. A special purpose signal processing computer with high speed analogue to digital converter (400 kHz) and Fast Fourier Transform capability was used. The instantaneous voltage signal was high pass filtered and sampled at the crank angle of interest over a $10^\circ$ crank angle sampled windows using the storage facilities in the computer, a turbulence trace was synthesised from individual $10^\circ$ crank angle samples over 70 to 400 consecutive engine cycles. The removal of the low frequency periodicity in the synthesised data resulted in the data being stationary. This was verified using the Run Test(21) to evaluate significance to a level of 95% certainty. The synthesised record was then used to establish the parameters $F(f); R_2; \lambda_u$ and $L_u$. From further digital computer processing, the hot wire calibration was obtained and hence $\lambda_u, L_u, U$ and $u'$. The engine turbulence parameters determined by Dent and Salama(6) corroborate the experimental findings of other investigators(20)(22) and the predictions of Tabaczynski(23) for $\lambda_u$ and $\lambda_u$. While the method discussed above yields results of acceptable accuracy, the effort to attain this goal is considerable in terms of data processing. This is a considerable disadvantage when hot wire anemometry is to be used for detailed mapping of in-cylinder flows and engine development studies.

6. AN ENGINE TURBULENCE ANALYSER

A scheme for real time evaluation of the two turbulence parameters which appear to closely characterise the engine combustion process - intensity of turbulence $(u')$ and micro scale $(\lambda_u)$ has been developed and is briefly outlined here.

The method requires a suitable calibration of the hot wire as discussed in section 3 to evaluate $u'$ and $(\lambda_u)$. Howells(24) outlines the possibilities of using the micro time scale $\lambda_u$ with an uncalibrated wire to infer in-cylinder turbulence behaviour will be briefly discussed later.

The engine under study is equipped with crank and camshaft encoders. The crankshaft encoder produces an electrical pulse every $2^\circ$ crank angle of the shaft rotation while the camshaft encoder produces a single pulse at a fixed point in the engine cycle. These signals, along with the high pass filtered turbulence signal, were fed to the analyser described below.

Evaluation of the r.m.s. value of the fluctuating voltage $(e)$ representative of turbulence $(u)$ is

$$e' = \left[ \frac{1}{T} \int_0^T e^2 \, dt \right]^{1/2}$$

(14)

The turbulence is sampled over a small interval $(\Delta t)$ during each cycle, at a particular point (crank angle) of the cycle, and the process repeated over $n$ successive cycles. Therefore for a gated signal of interval $(\Delta t)$ over $n$ cycles equation (14) can be written as

$$e' = \left[ \frac{1}{n(\Delta t)} \sum_{i=1}^{n} e^2 \, dt \right]^{1/2}$$

(15)

The cam pulse enables each engine cycle to be counted, while the $2^\circ$ crank pulses enable the sampling window position and width to be determined. Suitable electronics have been devised to open an analogue gate to accept turbulence signals $(e)$ over interval $(\Delta t)$ for $n$ consecutive cycles. The signal $(e)$ is squared and integrated over interval $(\Delta t)$ and then passed to a dividing circuit to produce

$$\Delta t \int e^2 \, dt / \Delta t$$

The signal is then passed to a voltage to frequency converter after which it is divided by $n$ before digital display as $e'$. The total number of cycles sampled $(n)$ can be varied between 250 and 1000.
The micro scale \( \lambda_x \) has been determined by the zero crossing method proposed by Townsend and discussed by Hine(3), which can be described by

\[
\lambda_x = \frac{\sqrt{2} \bar{U}}{\pi N_o}
\]  

(16)

where \( N_o \) is the number of zero crossings of the turbulence signal per unit time.

A schematic of the system for the evaluation of \( e^{12} \) and \( N_o \) is shown in Fig. 5, along with a gated signal of input \( e \). Typical results for \( \lambda_x \) and \( \lambda_t \) are shown in Fig. 6 for \( n = 250 \) cycles. Comparison has been made with the results of Dent and Salama(6) in Fig. 6, and in all cases agreement is better than 4%.

Spectrum of turbulence can readily be obtained by passing the turbulence signal through suitable narrow band filters ahead of the input gate to the system. The ensemble average of the voltage representative of the mean velocity \( \bar{U} \) over the engine cycle can be obtained by circuitry similar to that used for processing \( \bar{U} \). Calibration data for the hot wire is necessary to obtain absolute values of the mean and turbulent velocity.

Before leaving this section it is important to note that the hot wire anemometer can be used in an uncalibrated mode to determine a micro time scale \( \lambda_t \). From equations (5) and (16) we see that

\[
\lambda_t = \frac{\sqrt{2}}{\pi N_o}
\]  

(17)

Hence, in the engine development context where effects of changes in engine variables on turbulent mixing need evaluation, equation (17) and the zero crossing technique may be used to evaluate relative values of \( \lambda_x \). Such a study has been undertaken by Haghgoole et al.(24).

7. APPLICATION OF HOT WIRE ANEMOMETRY TO CONCENTRATION MEASUREMENT IN BINARY FLOWS

The essential features of the method are based upon the convective heat transfer characteristics of electrically heated wires in a binary gas flow, which are controlled by:

- The gas flow velocity over the wire
- The temperatures of the wire and the flow over it
- Thermal properties of the convective gas flow.

The thermal properties of binary gas mixtures are dependent on: the temperature, pressure and composition (mass fraction) of the mixture, and the temperature of the hot wire element. If the gas pressure and temperature and the hot wire temperature (resistance) are known, the convective heat loss from the wire (which is dependent on the measured output voltage of the anemometer bridge) will vary with mixture flow velocity and the mass fraction of the mixture. If two hot wire elements are operated at differing temperatures (resistances) in a binary gas flow, and in close proximity to each other, so that they sense the same mean flow velocity but do not cause thermal interference, the anemometer bridge voltages will be dependent on the mixture concentration (mass fraction) alone, at the particular flow condition. A fairly straightforward but tedious calibration procedure(25)(26) in a binary gas tunnel may be used to relate the output voltages of two anemometer bridges, mass fraction of the gas mixture, and its velocity. A typical calibration map of this type is shown in Fig. 7.

8. CONCENTRATION MAPPING OF SIMULATED ENGINE AIR-FUEL MIXING

The two wire method for concentration measurement discussed above requires significantly high operating wire temperatures for good sensitivity. Because of this, operation in an engine cylinder can reduce probe life at the elevated temperatures and pressures near t.d.c. under motored conditions. This, coupled with the tedious calibration procedure, led to the adoption of the engine flow simulation technique discussed below.

A transient gas jet (g) injected into a steady swirl flow of air (a) in a model, will have the same trajectory and spread as a liquid fuel spray (f) injected into a swirling air charge in the combustion chamber of an engine, when the following conditions of similarity between the model (M) and the engine (E) are satisfied.

- Linear scale \( L_E \), \( L_M \)
- Mean air swirl velocity profiles in the combustion space \( U_E \), \( U_M \)
- Equality of jet momentum flux

\[
\frac{\rho_f \cdot U_f^2}{\rho_a \cdot U_a^2} = \frac{\rho_f \cdot U_f^2}{\rho_a \cdot U_a^2}
\]

- Injection periods \( t_E \), \( t_M \)

An analysis of the above conditions(27) yields the following relationships.

\[
\frac{d_M}{d_E} = \left( \frac{\rho_f}{\rho_a} \right)^{\frac{1}{2}} \left( \frac{\rho_a}{\rho_f} \right)^{\frac{1}{2}} \left( \frac{L_M}{L_E} \right)^{\frac{1}{2}}
\]

(18)

\[
\frac{U_M}{U_E} = \frac{U_f}{U_a}
\]

(19)

\[
\frac{t_M}{t_E} = \left( \frac{L_M}{L_E} \right)^{\frac{1}{2}} \frac{U_f}{U_a}
\]

(20)

Equations (18), (19) and (20) enable the nozzle orifice diameter in the model \( d_M \), the gas jet velocity \( U_f \), the mean swirl velocity of the air in the model \( U_M \) and the gas jet injection period in the model \( t_M \) to be evaluated.

8.1 Experimental Techniques

A steady flow model of fixed geometry, representative of the engine combustion bowl, was
used to create a swirl flow with a prescribed velocity profile, similar to that in the engine combustion chamber under simulation — Fig. 8.

On entering the model, the metered air flow is deflected by a vane to produce a suitable velocity profile across the flow channel, which is representative of the air swirl in a 150\degree sector of the engine combustion chamber under study. The velocity profiles of the air swirl in the engine are determined by hot wire anemometry under motoring conditions (27)(28). Optimum opening of the vane, and location of air holes in it, were determined experimentally to obtain the required air swirl velocity profile.

Jet orifices located in the centre column of the model, or alternatively at the circumference of the combustion space, were used for injection of the fuel (gas) jet into the steady air swirl flow. Pulsed jets for transient analysis were created by using a rotary valve in the gas supply line, in close proximity to the jet nozzle.

Mapping of the gas concentration profiles under transient conditions requires good spatial resolution and fast response of the detecting transducer, hence the adaptation of the hot wire anemometer for this application.

8.2 Experimental Results
Tests were undertaken using the flow model described to simulate the published data of Stock (29) and Rife and Heywood (30) which include extensive high speed schlieren and direct photography of spray development in a cross flow of air. Further engine experiments were also conducted (25) and the results reported in reference (27). A selection of results are shown in Fig. 9, from which it is seen that the anemometry technique can provide an indication of local fuel concentration.

A limitation of the technique is that it requires probing the test volume at a large number of locations, it therefore requires experimental conditions to be maintained stable for long periods of time with good cyclic repeatability of the gas injection process. This can be reduced by completely automating the probe traversing and data acquisition processes. The technique does not enable the phase change of the atomised droplets in the engine fuel spray to be accounted for. Evaporation of the fuel spray in an engine cylinder under medium to high load operating conditions can be extremely fast, so that the assumption of a gas jet representing the atomised and evaporating fuel spray is a reasonable one. However, at low loads and during cold starting conditions, this approximation would be in error. In order to overcome these limitations the double exposure holographic interferometric method to be discussed below was considered.

9. APPLICATION OF HOLOGRAPHIC INTERFEROMETRY TO AIR-FUEL RATIO MEASUREMENT

9.1 Optical Interference
When beams of light intersect they obey the principle of superposition, and, because of the wave-like nature of light, this summation produces patterns of varying intensity of illumation known as interference.

The superposition of two wave trains, assuming for simplicity that each has the same frequency, results in a wave train similar to those of the component waves, but with amplitude now a function of the phase relationship and amplitudes of the component wave trains. The phase relationship depends on the distances travelled by the two wave trains, and their wavelengths in the media traversed. When the wave trains are in phase their amplitudes are additive, and when the wave trains are out of phase, the resultant amplitude is the difference in the amplitudes of the component wave trains. Between these limits, the resultant amplitude varies in a sinusoidal manner. If the wave trains are superimposed on a screen in such a way that the difference in optical path length travelled by the wave trains varies over the screen, bright and dark fringes of light will be seen, these being determined by the phase relationship between the two waves.

In interferometry it is the phase relationship between two wave trains that is made use of. It can be seen that with a large and random distribution of phase relationships, as will occur with white light, there will, on average, tend to be as many in phase as out of phase wave trains, so that there is a cancelling out of the phase information, and the resultant intensity is given by the sum of the individual intensities, but there is no interference, thus, the phase information in the light beam is lost. This demonstrates the important point concerning the coherence of a light source. In an ordinary light source such as a tungsten filament lamp, the atoms radiate independently, producing light waves which are incoherent both spatially (wave trains have random phasing) and temporally (wave trains are emitted with differing wavelengths). Temporal coherence can be achieved by using a monochromatic light source such as a mercury lamp with a green filter (0.53 \mu m wavelength), spatial coherence can be obtained by using a pinhole (which produces a plane wave train). The result is a spatial coherence length of the order of 10^{-3} m. In contrast, all the light produced by a laser is both spatially and temporally coherent. The spatial coherence length of a stabilised helium-neon laser is approximately 10 m, while that of a pulsed ruby laser is of the order of 1 m. In interferometry, the design and quality of the optical system must be such that it operates within the coherence length of the light source used. The advantage of the laser source is readily apparent.

9.2 Holographic Interferometry — Holographic Recording
Holography is based entirely on the excellent coherence properties of laser light sources. In the off axis system shown schematically in Fig. 10, the photographic plate is simultaneously exposed to the object and reference beams. The object beam is directed through the test volume containing a vapourising fuel spray. The coherent light passing through every point of the spray is impeded in its progress through it by differing amounts (depending on the local density within the
spray), and is therefore out of phase with the coherent reference beam by differing amounts. The object and reference beams interfere at the photographic plate and these interference patterns, recorded as variations in light intensity on the photographic emulsion, relate to differing regions of the spray relative to the photographic plate.

The photographic plate records only intensity which is proportional to the sum of the squares of the amplitudes of the object and reference beams. When the photographic negative is illuminated with a laser beam (here assumed to be of the same frequency and in the same direction as the reference beam), the negative - the hologram - behaves as a complex diffraction grating, so that the emerging wavefronts are dispersed to reconstruct the virtual and real images of the object volume - Fig. 10.

The zero-order undeflected beam contains no phase information about the object. The positive first order diffracted beam is responsible for reconstruction of the virtual image of the object, located behind the hologram in the position of the actual test volume. The negative first order diffracted beam contains information about the object in a pseudoscopic form, and results in the real image at a location in front of the hologram.

9.3 Double Exposure Holographic Interferometry

In the double exposure technique, a hologram is made of the test volume in the absence of the event to be studied - reference state exposure. Before processing the plate, a second exposure is made with the plate undisturbed, but during which the test volume has been modified by the event under study - the object state exposure. The result is two overlapping wavefront systems, which form an interference pattern indicative of the modifications to the test volume by the event under study. In engine studies the double exposure technique is undertaken with a pulsed laser.

9.4 Engine Interferometry System

The engine facility and experimental techniques have been described in detail elsewhere(31)(32)(33). A schematic of the optical access to the motored cylinder of the modified petrol engine of 86 mm bore is shown in Fig. 11. Because of the large (50 mm) diameter window in the cylinder head and the investigation of quiescent chamber direct injection engines, a reed valve system was used so that the cylinder 'breathing' was on a two stroke cycle. Fuel injection to the test cylinder (pentane or hexane) was from a standard diesel injector and jerk pump.

Interferograms were obtained using the double exposure method described in the previous section, with a 300 mJ pulsed ruby laser. Precise firing of the laser was controlled by an electronic logic control system actuated from the engine crank shaft.

Reconstruction of the holographic interferograms were carried out using a continuous wave helium-neon laser. The reconstructed virtual image was either imaged onto photographic plate for further enlargement and study with a back lit projection box, or imaged onto the vidicon tube of a television camera for video recording and later display on a large television monitor.

A typical interferogram for the injection of a liquid pentane jet into the engine test cylinder is shown in Fig. 12. The fringe structure is clearly evident. It should be noted as the axis of the jet is approached, the fringe density increases rapidly due to the presence of a liquid 'core' in the jet. This is further illustrated by a hologram of a liquid pentane injection - Fig. 13 - under identical conditions to those used for the interferogram. The recording was made with a single exposure of the plate with injection present - and is essentially a photograph of the jet. However, the fuel vapour is transparent to the laser beam, and small droplets (< 5 μm) were not resolved. The dense core of liquid fuel can clearly be seen, and its similarity to that observed with double exposure interferometry is very noticeable.

9.5 Inversion Of Fringe Count - Axisymmetric Case - Non-Reactive Jet

For a vapourising but non-burning axisymmetric jet, assuming refraction effects are negligible, the observed fringe system is the integrated result of interference caused by the object and reference beams. For a slice dz across a vapourising fuel jet, for which the refractive index variation is axisymmetric, the fringe number S(y) at y for the integrated effect along the optical path shown in Fig. 14 is given by the expression

\[
S(y) = \frac{\lambda}{4} \int_{0}^{R} \frac{(n_r - n_o) }{ (r^2 - y^2)^{3/2}} \,dr
\]

(21)

where \(\lambda\) is the wavelength of the light source, \(n_r\) is the refractive index of the fluid within the jet at radius \(r\), and \(n_o\) is the refractive index at a reference condition - state of the air outside the jet. The variation of the fringe number \(S(y)\) decreases from a maximum at the centreline of the spray to zero at its edge.

Equation (21) is a form of the Abel integral equation, and provided \((n_r - n_o) = 0\) for \(r > R\) can be inverted to yield

\[
(n_r - n_o) = -\frac{\lambda}{\pi} \int_{0}^{R} \frac{[dS(y)/dy] \, dy}{[y^2 - r^2]^3/2}
\]

(22)

Fringe counts were undertaken at a number of axial stations along the jet. At each axial station equation (22) was evaluated numerically to yield \((n_r - n_o)\).

9.6 Evaluation Of Local Air-Fuel Ratio

The Gladstone-Dale equation relates \(n_r\) to mixture density \(\rho_r\) at location \(r\) and can be expressed as

\[
n_r = \rho_r k_r + 1
\]

(23)

The Gladstone-Dale constant \(k_r\) for the binary mixture of air and fuel vapour (assumed to behave as an ideal gas) will be dependent on the mass fractions of fuel vapour \((m_f)\) and air \((m_o)\) at
location \( r \). Hence equation (23) can be expressed as

\[
n_r = \rho_r \left[ m_{jr} k_j + m_{or} k_o \right] + 1 \tag{24}
\]

where \( m_{jr} + m_{or} = 1 \). Therefore

\[
n_r - n_o = \frac{\rho_r}{\rho_o} \left[ \frac{k_j}{k_o} - 1 \right] - 1 \tag{25}
\]

Assuming constant pressure throughout the test volume, expressing the mixture volume at \( r \) as the sum of the partial volumes of air and fuel vapor at a mean temperature \( T_{sr} \), and application of the equation of state for the mixture — produces after some algebraic rearrangement — the mass fraction of the fuel vapor at \( r \) which is

\[
m_{jr} = \frac{(T_o/T_r - 1) - (n_r - n_o)}{\rho_o k_o} \tag{26}
\]

In equation (26), \( n_r - n_o \) is obtained from fringe inversion — equation (22) — \( \rho_o \), \( m_{jr} \), \( \rho_o \) and \( T_o \) are known, \( k_o \) and \( k_j \) are obtained from tabulated data.

\( T_r \) is considered to be the mass mean temperature of the vapourising fuel at a reference saturation state temperature \( T_s \)(32), and the surrounding air at the reference state \( T_o \). Hence

\[
T_r = m_{jr} \cdot T_s + (1 - m_{jr}) \cdot T_o \tag{27}
\]

Equations (26) and (27) are solved by iteration to yield the unknowns \( m_{jr} \) and \( T_r \).

The fuel concentration \( m_{jr} \) has been determined from the interferogram shown in Fig. 12, and the results presented in Fig. 15. The difficulty in resolving fringes close to the axis of the spray is apparent.

The problem of fringe resolution is acute in diesel engine applications — this can be seen from equations (21) and (23). From (23) the refractive index \( n_r \) is seen to increase in proportion with cylinder density (hence compression ratio), and from (21) it is seen that the fringe count — number of fringes is directly related to \( n_r \). This difficulty is most severe in the study of non-reactive vapourising sprays where chamber density is high, to overcome this problem the similarity principles developed in section 8 have been extended(32)(33) to consider similarity in spray evaporation behaviour, thereby enabling the evaporation process in the high compression diesel engine to be simulated at a lower pressure in a test engine, where interferometry may be employed to determine local air-fuel ratio.

The essential feature of the evaporation similarity between the actual and test engine is the equality of the mass transfer flux \( \dot{m}^m \) for each system, and the dependence of \( \dot{m}^m \) on the droplet Reynolds Number \( \left( \text{Re}_D \right) \) and the Transfer Number \( \left( \text{B} \right) \). \( \text{Re}_D \) is based on the fuel jet discharge velocity — which is directly related to the injection pressure — and a mean droplet size which is considered to be the Sauter Mean Diameter (S.M.D.) — which represents the ratio of the atomised fuel spray volume to its surface area. The S.M.D. is itself related to the fuel properties and injection parameters(34). The Transfer Number \( B \) for evaporation of the fuel droplets represents the ratio of the enthalpy potential available for mass transfer (evaporation) to the enthalpy of vapourisation of the fuel — which must be exceeded before the process of evaporation can proceed. The basis of the scaling relationships outlined here have been evaluated experimentally in the test engine discussed in section 9.4. The engine was operated at two compression ratios with two fuels of differing boiling points, with two differing injection nozzle orifice diameters. The similarity of local air-fuel ratio obtained in these two experiments is shown in Fig. 15.

9.7 Combustion Bomb Studies

The principles of holographic interferometry have been applied(33) to the study of a burning fuel spray following transient injection and autoignition of the spray in a bomb. A bomb was chosen in preference to the engine discussed in 9.4, because of its superior optical access. In order to achieve delay periods representative of an engine, diethyl ether doped with nitromethane was used as the fuel. An impingement plate was used in the bomb so that the effects of wall jet development on air entrainment could be studied, as the transient jet impinging on the plate. The relationship of chemical composition to refractive index, density, temperature and optical fringe shift was established as indicated below.

Chemical equilibrium was assumed in the burning regions of the spray, and the following reactions considered.

\[
\begin{align*}
C_2H_5O_2 & + q(0_2 + 3.76 N_2) \rightarrow uCO_2 + (x - u) CO \\
& \quad + \text{H}_2 + \left( \frac{y}{2} - u \right) \text{H}_2 \\
& \quad + \left( q + \frac{z}{2} - \frac{(x + u + v)}{2} \right) 0_2 + 3.76 qN_2
\end{align*}
\tag{28}
\]

\[
\begin{align*}
\text{CO}_2 & \rightarrow \text{CO} + \frac{q}{2} 0_2 \\
\text{H}_20 & \rightarrow \text{H}_2 + \frac{q}{2} 0_2
\end{align*}
\tag{29}
\]

Reactions with equivalence ratio \( \alpha > 1.5 \) result in the absence of \( \text{CO}_2 \) in the products. Beyond the rich limit of combustion \( \phi > 3 \) for diethyl ether fuel, vapour coexists with the products of combustion.

For a given equivalence ratio \( \phi \), temperature, pressure and fuel type, equations (28) to (30) and appropriate equilibrium constant data enable the concentration \( \text{f}_r \) of species \( r \) and the density \( \rho_r \) to be evaluated. The refractive index \( n_{rb} \) is given by

\[
n_{rb} = \rho_{rb} \sum \frac{C_{1r} k_{1r}}{\text{ALL} i} + 1 \tag{31}
\]
where \( k_j \) is the Gladstone-Dale constant for the species \( j \). Thus a table or chart of \( T_r, (n-1) \) and \( \delta \) can be prepared.

The refractive index at every point in the jet is calculated from the fringe count using the procedure discussed earlier and the assumption of an axisymmetric burning jet. However in this case, the fringe counts are assumed to decrease up to the point on the radius where the refractive index calculated would be such that the temperature of the products of combustion would reach a value corresponding to the adiabatic flame temperature for the mixture fraction beyond this point, and up to the axis of the jet, the fringe count increases, see Fig. 16. The values of refractive index and mixture fraction at any point are used to find the temperature from the data of \( T_r, (n-1) \) and \( \delta \). This procedure is used up to the rich limit, when mixture fraction exceeds the rich limit. Above this limit are considered to be present. The concentration \( C_f \) of fuel vapour is given by

\[
C_f = \left( C_r(r, x) - C_{RL} \right) / \left( 1 - C_{RL} \right)
\]

where \( C_{RL} \) refers to the mixture fraction at the rich limit. If the combined Gladstone-Dale constant and average molecular weight of the products of combustion at the rich limit are \( k_B \) and \( M_b \) respectively, then the temperature at any point in the jet can be obtained from

\[
T_r = \frac{k_r}{k_w} \left[ \frac{C_r + (1 - C_r) k_B}{k_w + (n_r - n_w) c_w} \right] \left[ \frac{1 + \frac{C_f}{M_f} - \frac{1 - C_r}{M_r}}{M_f - M_b} \right]
\]

where \( r, w, f \) refer to the surroundings, considered point in the jet and fuel vapour. The entire scheme of calculation is summarised in Ref. 33.

Fig. 17 shows radial variation of temperature at various axial stations from the injector orifice. Near the nozzle the temperature is high, and away from it the temperature drops. At 1.5 ms after the start of injection, the temperature along the axis of the jet is low and remains so along the whole length of the jet indicating a cool central core to the jet. The temperature increases in the radial direction, reaches a peak and decreases sharply towards the edge of the jet. By 2.5 ms after the start of injection, the central regions of the jet have reached temperatures of the order of the adiabatic flame temperature along the length of the jet. This appears to indicate strong air entrainment into the tail and central regions of the jet. By 3.5 ms this entrainment has caused a further decrease in the temperature of the central regions of the jet from about half the jet length to its tip, the presence of the impingement plate causes a considerable drop in temperature. The tail half of the jet remains at remarkably high temperatures indicating reduced entrainment in that region.

Concentrations of \( H_2O \) vapour, \( CO \) and \( CO_2 \) vary according to the completion of combustion, Fig. 18. At a distance of 19.5 mm from the nozzle \( H_2O \) and \( CO_2 \) increase and then decrease from the periphery to the axis of the jet. The maximum is reached at the location where the mixture fraction is approximately stoichiometric. This peak in the profile disappears at larger intervals of time (2.5 ms) due to mixing and entrainment of air. The concentration of oxygen decreases from the periphery towards the axis and is absent near the axis, during the period just following ignition. As time progresses, the concentration of oxygen near the axis increases due to entrainment. Close to the wall, the concentrations of \( H_2O \) and \( CO_2 \) are high for times as large as 4.5 ms after injection, suggesting that combustion is complete. Fig. 19 shows the variation of mass averaged temperature and combustion products over the whole jet at instants of time following injection.

9.8 Consideration of Asymmetric Non- Reactive Gas Jet Studies

The applications of interferometry discussed in the earlier sections have been limited to the study of axisymmetric fuel spray vapourisation and combustion. In actual engines, the fuel spray development departs from the idealised axisymmetric configuration due to:

- Wall impingement of the jet and the proximity of cylinder head and piston surfaces causing recirculation effects.
- Effects of bulk air motion - swirl in high speed direct injection diesel engines cause deflection of the fuel spray axis, and the combined effects of impingement and swirl also produce a strong flow of a fuel rich mixture along the combustion chamber wall.

Application of double exposure holographic interferometry and data inversion to yield fuel concentration in an asymmetric jet flow will be briefly touched on, and results presented.

In the earlier discussion of holographic recording and reconstruction, it was seen that actual holograms have a field of view limited by the size of the photographic plate used for recording. This, coupled with the need to avoid distortion of the fringe field during analysis by viewing at angles that do not deviate significantly from the object beam direction, requires that sufficient angular coverage of the object field is obtained during holographic recording (180° when no planes of symmetry are present). In the context of the transient asymmetric jet flow field this requires simultaneous recording of double exposure interferograms grouped around the test volume.

For a generalised three-dimensional concentration field, the fringe number \( S(x,y,z) \), resulting from the interference caused by the mixing of object and reference beams, will have the form

\[
S(x,y,z) = \frac{1}{L_2} \int_{L_0}^{L_1} \left[ n(x,y,z) - n_0 \right] dL
\]

where \( n_0 \) is the refractive index at a reference state outside the concentration field. For a plane of constant \( z \), equation (34) can be rewritten as
\[ S(y', x') = \frac{1}{x_0} \int_{x_0}^{x_1} f(x, y, z_0) \, dx \]  

(35)

where \( f(x, y, z_0) = \frac{n(x, y, z_0)}{n_0} - 1 \) and \( x' \) and \( y' \) are measured in a coordinate system rotated by an angle \( \phi \) about the z axis.

The inversion of equation (35) follows the detailed procedure developed by Mazurka[35] based on the earlier work of Maldonado[36], which in essence expresses \( f(x, y) \) as a series of orthogonal functions, in which the coefficients of the series contain the summation of the experimental fringe number data over the data plane at \( z_0 \).

The experimental configuration is shown schematically in Fig. 20. Propane was injected into the cavity through a 2 mm orifice for a duration of 6 ms. Three interferograms were taken, one viewing the open back of the cavity (using the nozzle axis plane for a reference), a second was taken at 90° from the first giving a side view of the cavity, and a third interferogram midway between the first two. These interferograms were recorded simultaneously 1 ms after the end of injection.

From the contours shown in Fig. 21 the effect of cavity walls on air entrainment is apparent. Because of the late firing of the laser, the leaner regions of the jet will be closer to the injector orifice, this is also enhanced by the availability of air. The jet impinges on the side walls of the cavity at about 30% of its length. This impedes air entrainment hence richer zones appear downstream of the impingement point (3.25 cm). The effect of the inclination of the jet axis can also be seen in the very rich zone close to the top surface of the cavity. Nearer the tip of the jet, the enhanced air entrainment of the wall jet appears to have an effect in reducing the concentration of propane.

**REFERENCES**


FIG. 1 VARIATION OF MEAN VELOCITY AND INTENSITY OF TURBULENCE, MOTORED TESTS.

FIG. 2 VARIATION OF EXHAUST SMOKE WITH TURBULENCE INTENSITY AND MICROSCALE (D.I. DIESEL ENGINE WITH SWIRL).

FIG. 3 EXHAUST SMOKE VARIATION WITH MIXING RATE (QUIESCENT D.I. DIESEL ENGINE).

FIG. 4 UNBURNED HYDROCARBON VARIATION WITH MIXING RATE (QUIESCENT D.I. DIESEL ENGINE).

FIG. 5 SCHEMATIC OF ELECTRONICS FOR $e^2$ AND $n_o$.

FIG. 6 COMPARISON OF DATA WITH TURBULENCE ANALYSER.

FIG. 7 TYPICAL CALIBRATION MAP FOR TWO WIRE PROBE IN BINARY GAS TUNNEL.
FIG. 8 SCHEMATIC OF ENGINE SWIRL FLOW MODEL.

FIG. 9(a) COMPARISON - TRAJECTORY AND SPREAD ENGINE FUEL SPRAY AND ENGINE SIMULATION (STEADY FLOW).

FIG. 9(b) ENGINE SIMULATION LOCAL A/F RATIO (STEADY FLOW).

FIG. 10 SCHEMATIC ILLUSTRATING HOLOGRAPHIC RECORDING (a), AND RECONSTRUCTION (b).

FIG. 11 SCHEMATIC OF ENGINE OPTICS.

FIG. 12 INTERFEROGRAM OF PENTANE INJECTION INTO ENGINE CYLINDER (1100 r.p.m.).

FIG. 13 HOLOGRAM OF PENTANE INJECTION INTO ENGINE CYLINDER (1100 r.p.m.).

FIG. 14 SCHEMATIC ILLUSTRATING FRINGE INTERPRETATION.

FIG. 15 VARIATION OF A/F RATIO IN PENTANE (FIG 12) AND HEXANE JETS OBERING SIMILARITY CRITERIA.
FIG. 19 MASS AVERAGED TEMPERATURE AND PRODUCT CONCENTRATION IN A BURNING TRANSIENT JET.

Recirculation cavity

FIG. 20 PROPANE INJECTION INTO A CAVITY.

FIG. 21 PROPANE CONCENTRATION CONTOURS AT VARIOUS PLANE PERPENDICULAR TO JET AXIS AT 1 MS AFTER INJECTION (RELATIVE CONCENTRATION SCALE 0 TO 10).

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