Modelling and Experimental Validation of Turbulent Flame Propagation in Spark Ignition Engines

نمذجة نظرية وتحقيق تجريبي لتقدم اللهب الدوامى فى محركات الاشعال بالشرارة

BY

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الخلاصة: فى هذة الدراسة تم تطوير وتعديل نموذج نظرى تم تأسيسة بجامعة ليدز بالمملكة المتحدة لدورة الاحتراق بمحركات الاشعال بالشرارة ذات غرف الاحتراق الأحادية، وذلك بهدف الوصول الى نتائج أكثر دقة من هذا النموذج. وقد تضمنت هذة التحسينات اضافة نموذج نظرى تم تأسيسة لحساب كمية الشحنة المتسربة خلال فراغات المكبس والشنابر.

وللتحقق من التعديلات والتحسينات التى تمت على هذا النموذج أستخدمت نتائج تم الحصول عليها تجريبيا من محرك اشعال بالشرارة ثنائى الأشواط تم تجهيزة لتصوير تقدم اللهب أثناء اجراء الاحتراق. وقد تم الحصول على هذة النتائج عند ظروف التشغيل الآتية: نسبة انضغاط (٨.٣)، سرعة محرك (١١٠٠ لفة/دقيقة)، ضغط الدخول للشحنة (٠.٨٥ بار) و نسبة تكافؤ (١.٠). وللتحقق من جدوى التعديلات التى تم ادخالها على هذا النموذج تم تطبيقة على دورات أخرى عند ظروف تشغيل مختلفة.

وقد اتضح أن التعديلات والتحسينات التى أدخلت على النموذج النظرى لدورة الاحتراق بمحركات الاشعال بالشرارة أدت الى تحسين ملموس فى النتائج التى يتم الوصول اليها من هذا النموذج وذلك مقارنة بالنتائج التى يتم الحصول عليها من المحرك.

ABSTRACT

This work is concerned with modelling and experimental validation of turbulent flame propagation in spark ignition engines. An existing engine cycle model, developed in Leeds for single chamber spark ignition engines, has been modified in order to obtain more realistic model predictions. The most significant improvement involved the incorporation of a sub-model for blowby and piston top land crevice flow.

To optimise and then verify the modified model predictions, the model was initially applied to data derived from experiments using an optically accessed two stroke spark ignition engine. The modified model was optimized an engine cycle at the running conditions compression ratio 8.3, equivalence ratio 1.0, engine speed 1100 rpm and intake manifold pressure of 0.85 bar. Thereafter, its predictive performance was verified for other cycles at varied engine operating conditions.

The inclusion of the blowby routine was shown to greatly improve the predictions of the model, compared with the experimental data. Inclusion of blowby is particularly important in modelling ported two stroke engines, due to the inefficiency of the piston rings early in the compression stroke.

KEY WORDS
SI engine cycle model, turbulent combustion in spark ignition engines.

1. INTRODUCTION

The new and changing requirements of spark ignition engines have greatly increased the complexity of engineering engine design to give optimum operating characteristic within the limitations of emissions and fuel economy standards and fuel octane levels.

The improvements in available computing facilities in recent years have greatly increased the importance and capabilities of engine cycle simulation models. These models provide an important aid to improving engine design. They reduce the extent of the experimental development studies necessary, such studies require sophisticated equipment and considerable expense. They also provide a good approach to understanding of the complex events that occur during the engine cycle. In an engine cycle model, the combustion process is the most important process to be modelled. It is in this process that the fuel is burned, the engine power is developed and pollutants are produced.

The objective of the work described in this paper was to study the combustion in spark ignition engines through the modelling of the turbulent flame propagation process.

2. CHARACTERISTICS OF FLAME PROPAGATION IN S.I.Es.

In spark ignition engines, immediately following the spark, there is a period during which very little heat release due to combustion is observed [1, 2, 3]. Following this period flame propagation from the initial spark ignited flame kernel proceeds through the unburned charge in the combustion chamber.

Flame is defined as a rapid, self-sustaining, chemical reaction occurring in a discrete reaction zone [4]. The speed by which the reaction zone moves relative to unburned gas ahead the flame front is known as the burning velocity. In quiescent mixture, the reaction zone thickness is thin, the flame surface is smooth, and flame propagation is laminar. When the combustible mixture is turbulent, the reaction zone becomes thicker than laminar case, the flame front is no longer smooth and the burning velocity becomes several times its equivalent laminar value, depending on the intensity of turbulence.

The laminar burning velocity is determined by the chemical and thermodynamic properties of the mixture. Laminar burning velocity data has normally been generated from experiments conducted in constant volume bombs or on burners [5 - 8]. Such data are rarely at the required engine pressures and temperatures, it is necessary to adopt extrapolations using relationships such as those presented in references [6] and [7].

The turbulent burning velocity in spark ignition engines is generally calculated using one of many available correlations [2, 8-10]. The common approach behind these correlations is to isolate the effects of turbulence from those of mixture composition and temperature by correlating the ratio of turbulent-to-laminar burning velocity with turbulence intensity

Most of these correlations have been derived from experimental measurements in burners and constant volume bombs, in which the length scales were larger than in engines. Nevertheless it is usually considered that such correlations are applicable to engines [11].

A number of the correlations available (for the turbulent burning velocity) in literature have been previously incorporated into the Leeds engine model by Hynes [11]. However the preliminary assessment for the usefulness of these correlations for the model, revealed preferred correlations are those of Bradley and his co-workers [10], since they incorporate flame straining effects and the "effective" r.m.s. turbulent velocity for developing turbulent flames. These workers have demonstrated the dependence of the turbulent burning velocity on the dimensionless Karlovitz flame stretch factor, K, and the Lewis number, L_e. The latter represents the ratio of transport coefficients for energy and mass and is given by $k/C_p D$ (k is the thermal conductivity, C_p is the constant pressure specific heat and D is the diffusion coefficient of the deficient reactants). The parameter K may be regarded as a ratio of chemical to eddy lifetime and is defined by $(u'/\lambda)(\delta_l / u_l)$. It was readily shown [10] that, for isotropic turbulence

$$K = 0.157 \, (u'/u_l)^2 \, R_L^{-0.5} \qquad (1)$$

in which R_L is the turbulent Reynolds number and is given by $u'L/\upsilon$ (υ is the kinematic viscosity).

Shown in Fig. 1, is the relationship between u_t/u_l and u'_k/u_l suggested by the Leeds turbulent burning velocity correlations. The full line curves show the effect of flame straining in terms of Karlovitz stretch factor. The values of (R_L/L_e^2) are represented by the chain dotted curves. The influence of length scale has been incorporated in R_L which represents a dimensionless mean eddy lifetime. The values of all parameters relate to the unburned gas ahead of the flame front.

The propagation of a fully established turbulent flame will be influenced by the full spectrum of the turbulence. However, an explosion in a gaseous medium of isotropic turbulence that initiates from a point ignition source is first affected only by the highest frequencies of turbulent spectrum. As the flame grows, progressively, increasingly lower frequencies become influential. For an elapsed time from flame initiation of t, the r.m.s. turbulent velocity effective in influencing flame propagation, u'_k, is given by:

$$(u'_k)^2 = u' \int_{\bar{f}_k}^{\infty} \bar{S}(\bar{F}) \, d\bar{F} \qquad (2)$$

where, $\bar{S}(\bar{F})$ is dimensionless power spectral density function given by

$$\bar{S}(\bar{F}) = S(f) \, / \, (u')^2 \, \tau_k \qquad (3)$$

The dimensionless power spectrum $\bar{S}(\bar{F})$ based on laser doppler anemometry measurements in Leeds, is set out in Fig. 2(a). In Eq. 3, the dimensionless frequency $\bar{F} = f\tau_k$ and the infinite frequency is associated with zero time. Results of integrating Eq. 2 are shown in Fig. 2(b), while values of u'_k/u' is plotted against the dimensionless time t_k / τ_k [12].

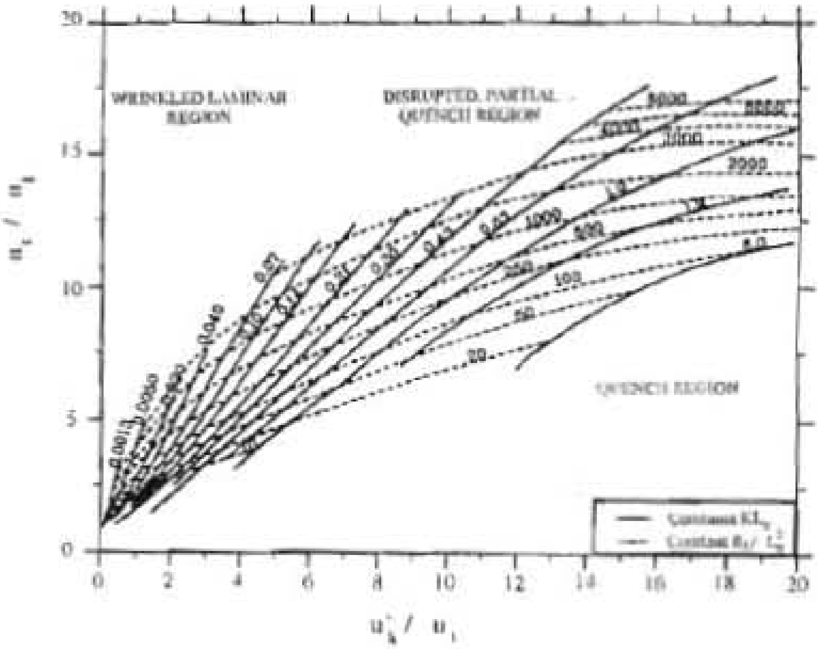


Fig.1 The relationship between u_t/u_l and u'_k/u_l suggested by the Leeds turbulent burning velocity correlations

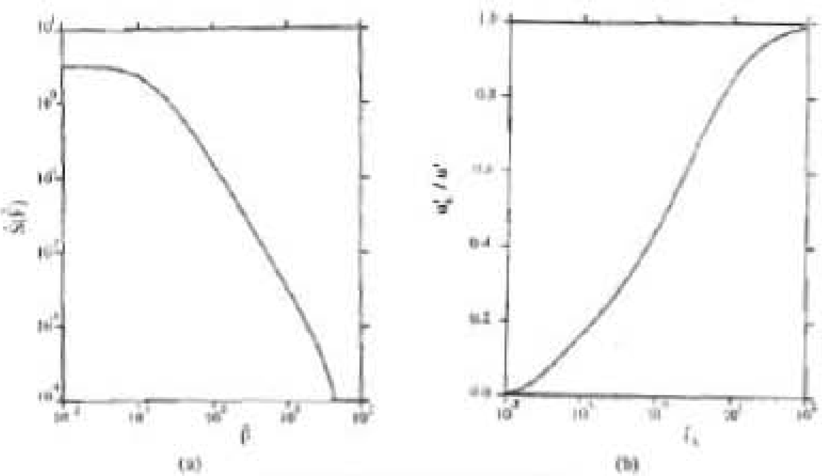


Fig. 2 Developing turbulence intensity

3. ENGINE CYCLE MODEL

The University of Leeds Department of Mechanical Engineering has a longstanding interest in spark ignition engine modelling [11,13,14]. A thermodynamic cycle model has been developed for single chamber engine. This model was initially based on the air-standard Otto cycle, on the premise of thermodynamical equilibrium of the combustion products. The model was initially used [13] to demonstrate the higher thermal efficiencies obtained by lean burning and to show that higher compression ratios could be used to compensate for the lower power output associated with burning lean mixtures. Later, the program was modified [14] so that it could simulate power modulation by throttling at a constant equivalence ratio. Further extensive modifications to the program were then made [11] to allow the combustion rate to be expressed as a function of turbulent burning velocity and wall heat transfer effects were incorporated. A full description of the Leeds simulation model is given in Reference [11]. To assist understanding of the present paper, a brief outline of the combustion process in the Leeds engine cycle model is included below.

3.1 Combustion Process

In the model it was assumed that, during the combustion process, the working fluid is composed of a mixture of combustion products of "frozen" combustion and/or fresh reactants. This comprised two distinct regions in the chamber, one of fresh reactants (including any residual gases from a previous cycle) and the other of combustion products, separated by a thick reaction zone (flame front). The two regions existed at identical pressure but at different temperature; no heat transfer was allowed between the two regions. The combustion process in the model was divided into two periods: ignition and combustion.

<u>The ignition period</u> The ignition process was considered to be initiated by spark. Following the spark, a spherical kernel of diameter equal to the Taylor microscale (λ) was assumed to ignite at its center, and to burn at a rate given by the laminar burning velocity (u_l). The time required to burn such a kernel was calculated using the following equation [15].

$$\tau = 1.8\, C_{ig}\, [L / u']^{1/3}\, [\lambda / u_l]^{2/3} \quad \ldots\ldots \quad (4)$$

C_{ig} is an engine constant, L was taken as 0.1 times the cylinder height and λ was found from the relation given by Taylor [16]

$$\lambda / l_e = A\, \nu / u' \quad \ldots\ldots \quad (5)$$

where, l_e is the laminar flame thickness and λ is the Taylor microscale.

Following burning the kernel of radius (λ), the program considered an expansion of the burned charge and compression of the unburned charge. Heat transfer rate from the unburned charge during the time was calculated and the revised temperature and pressure of the unburned charge was found. Based in these revised values of unburned gas temperature and pressure, the turbulent parameters and burning velocity were re-calculated

<u>**The combustion period**</u> After ignition, the flame was assumed to travel through the mixture in the combustion chamber, entraining the unburned gas ahead of it. The flame propagation model was considered to be spherical and the entrainment rate of the unburned gas was given by [17]:

$$d\,M_e/dt = \rho\,A_f\,u_e \qquad (6)$$

where A_f is the instantaneous mean flame surface area and u_e is the entrainment velocity (u_t). The turbulent burning velocity could be found, on the basis of estimated value of u'_k and u_l, from the Leeds burning velocity correlation diagram. The entrainment was considered in an incremental fashion with a spherical flame growth. The mass entrained during each increment was given by:-

$$\Delta m_{ei} = \rho\,u_{ei}\,\Delta V_f \qquad (7)$$

It was then assumed that some of this mass entrained would burn during this increment, together with mass entrained during previous increments. After burning it would expand, compressing the burnt and unburned regions; this process was considered to be isentropic and resulted in a movement of the flame front forward of a distance equal to $u_{gi}\,\Delta t_i$, where u_{gi} is the unburned gas velocity ahead of the flame front. Once entrained, the unburned charge was assumed to burn at an exponential rate [17].

The process was considered computationally in an incremental manner. The mass was considered to be burned instantaneously at constant volume, then allowed to expand, compressing the 'old' burnt charge. The burnt regions were mixed (the unburned charge behind and in front of the flame front was treated as one region for the purpose of pressure equalization). Heat transfer was considered in this calculations using the Woschni correlation [18]. The process was repeated until the inflammed volume was greater than 99 % of the total cylinder volume.

3.2 Current Modifications of the Leeds Cycle Model

Work is being undertaken to verify the Leeds cycle model. Some success has been reported [11], but imperfect matching of experimental data and model predictions has been noted. Therefore, in the work reported in this paper, three modifications were conducted to the Leeds cycle model in order to obtain more realistic prediction of the model against the experimental data. These modifications concerned piston ring blowby and piston top land crevice flow, flame front area calculation, and heat transfer calculation. These modifications will be discussed now.

<u>**First modification**</u> The piston's top-land crevice volume, bounded by the piston top-land, piston top ring and cylinder wall, is an important factor in engine design. A significant part of the mass contained in the cylinder is trapped in this small volume at critical stages of the cycle. This mass is unavailable for normal combustion, being compressed into the crevices; it is hence of great importance in cycle modelling.

In the current study, the Leeds cycle model has been extended to include the effect of the blowby and piston top land crevice flow by incorporating a sub-program of blowby and piston top land crevice flow [3]. This sub-program was developed from the simple orifice and volume method proposed by Ting and Mayers [19].

Second modification Generally in SI engine models, the flame is considered to propagate through the combustion chamber either in spherical or cylindrical manner with a regular front area.

In this study, a "shape factor" option was included in the model in attempt to obtain a more realistic estimation of the flame front area and hence of the burning rate in the model. This factor is defined as the ratio between projected area of smoothed flame measured on the film to that calculated on spherical growth basis. Typical values of this factor for a number of engine running conditions are given in Reference [3].

Third modification The influence of heat transfer on model predictions was investigated by Hynes [11]. Based on comparisons between model predictions with and without heat transfer, he showed that the heat losses during combustion and expansion processes from the cylinder charge has little effect on cylinder pressure but a significant effect on the charge temperature. However, since it has been suggested by many authors [20,21] that heat transfer has a significant effect on the cylinder pressure as well as the charge temperature, this effect was also assessed in the current study.

Therefore, the Leeds' program was modified by adopting the well known Annand correlation [22] for calculating the coefficient of heat transfer in addition to the previously existing [11] Woschni correlation [18]. The principal reason for adopting Annand correlation was that it included a radiation term.

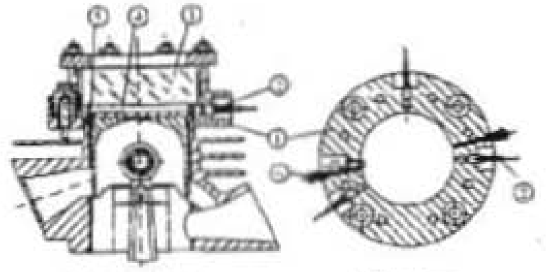
4. EXPERIMENTAL ENGINE DATA

A modified single cylinder two stroke Yamaha engine was employed in the current study to collect the engine experimental data required for the verification and validation of the current modelling work. The engine had a specially designed cylinder head which allowed full optical access to the combustion chamber through a top window as shown in Fig.3. Full details of this engine can be found in Reference [23]. The advantage of this engine was that at high compression ratio the clearance height was small and hence the flame spread could be regarded as two dimensional.

The techniques adopted in this study included the use of a high speed cine' camera and the collection of simultaneous cylinder pressure records using an on-line computer. This included an electronic ignition system, a shaft encoder, a peizo-electric pressure transducer with charge amplifier, an on line data acquisition system and a a Hitachi 164 high speed cine' camera, capable of 10000 frames per second. Details of the instrumentation is given in Reference [3] and Fig.4 shows an outline of the techniques.

It may be noted that, for purpose of synchronization, spark and TDC signals together with the pressure signal were converted to digital form using a Micro-Consultants very high speed analog to digital converter (ADC - 8 channels with 400 kHz maximum rate) unit. The engine was fired every 4th cycle to facilitate breathing and help prevent the engine accelerating beyond the required speed. The system provided spark and TDC signals to the on-line computer.

High speed natural light films of combustion process were recorded at the engine operating conditions given in Table 1. The films were analyzed using a back projecting screen. This allowed slow motion viewing of the film as well as examination frame by frame. For selected cycles the flame position in each frame was traced onto transparent paper. Using a Summa Graphics bit pad, the flame traces were digitized into X-Y coordinates and stored.



1- Cylinder head
2- Pressure transducer
3- Quartz window
4- Flat mirror
5- Gasket
6- Spark electrode

Fig.3 Yamaha engine

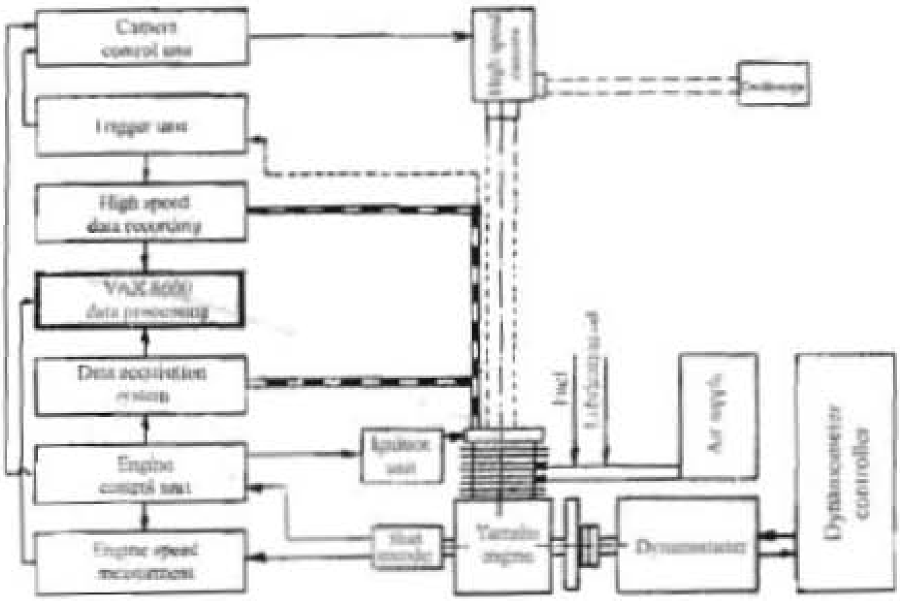


Fig.4 Outline of the engine and the experimental technique

in a data file. These data could be used subsequently to calculate the flame radii and flame speeds for each cycle, using an existing computer program [11]. A diagnostic computer program for analyzing spark ignition engine experimental data [19] was used to calculate engine parameters as well as the burning rate during combustion process. Therefore, using the above mentioned experimental technique for data collection and analysis, it was possible to obtain the in-cylinder pressure, the flame speed and the flame radius for the selected Yamaha engine cycles given in Table 1. These experimental data were used in the optimisation and validation of the current modelling work.

TABLE 1- Engine operating conditions

Cycle No.	Equiv. ratio	Comp. ratio	Speed, rpm	Spark adv., degree bTDC
1	1.0	8.3	1100	32.0
2	1.0	6.2	1100	32.0
3	1.0	8.3	1400	32.0
4	0.9	8.3	1400	36.0

5. PARAMETRIC STUDY USING LEEDS ENGINE CYCLE MODEL

As mentioned before, the present study is principally concerned with the incorporation of a blowby routine into the Leeds engine cycle model, in order to obtain a more realistic prediction of the model as compared with the experimental data. However, as a preliminary, a parametric study has also been undertaken to highlight the effects of varying model parameters on predicted engine combustion for a number of engine conditions. Initially, engine variables were maintained constant corresponding to the first cycle in Table 1.

5.1 Turbulence Parameters

The turbulence parameters which have significant effects on the model prediction are: integral length scale, turbulence intensity, and chemical lifetime. Effects of changing these parameters are outlined below

Integral length scale The integral length scale is the characteristic average length of the large eddies within a turbulent flow while the small eddies associated with velocity fluctuations at high frequencies are measured by the Taylor microscale [24]. The integral length scale at ignition has been assumed proportional to the instantaneous cylinder height. For turbulence in a bomb it has been shown [25] that the length scale is approximately one tenth of the vessel diameter. In an engine, the constant of proportionality was taken as 0.15 by [1] and as 0.1 by [26], while [27] suggested 0.2 for this constant. However, to investigate the effect of integral length scale at ignition on the engine parameters, the program was run for a different values of the constant k_l. Shown in Fig.5 are the model prediction for the in-cylinder pressure using different values of the integral length scale at ignition. It is clear in this figure that value of the constant of proportionality (k_l) has a significant effect on the in-cylinder pressure

Turbulence intensity The turbulence intensity, u', is defined as the root mean square of the fluctuating velocity of the turbulent flow. For conventional single chamber engines, at the time of ignition, this parameter has usually been assumed to be proportional to the mean

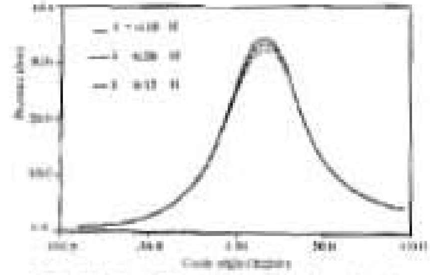


Fig. 5 Effect of integral length scale at ignition on the model prediction

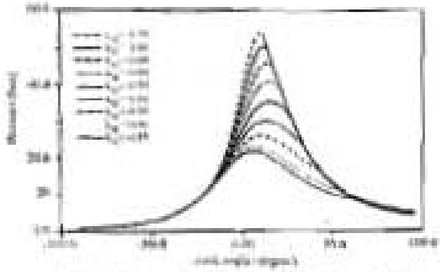


Fig. 6 Effect of turbulence intensity at ignition on the model prediction

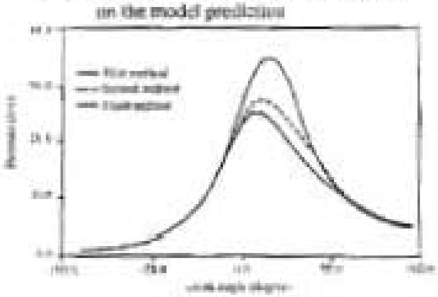


Fig. 7 Effect of the manner in which u' and L vary during the combustion process on the model prediction

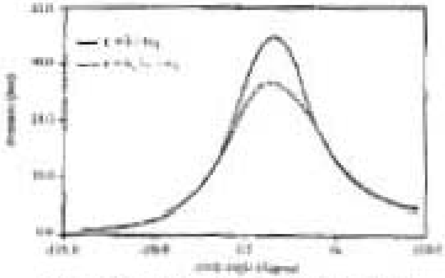


Fig. 8 Effect of chemical lifetime on the model prediction

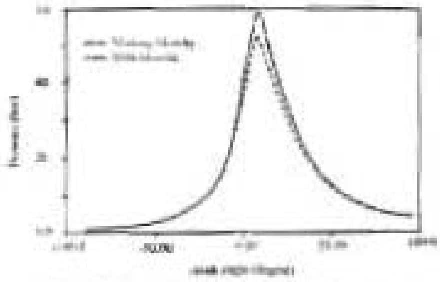


Fig. 9 Effect of blowby on the model prediction

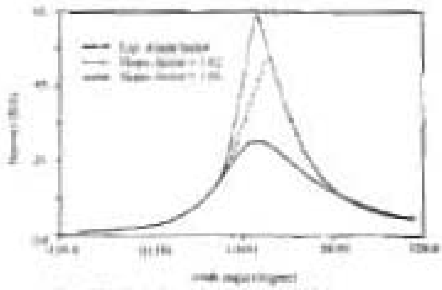


Fig. 10 Model predictions at different values of shape factor

piston speed. Therefore for the thermodynamic models which incorporate turbulent rate of burning the value of the constant of proportionality k_w will greatly affect u' at ignition which in turn affects the model prediction.

In literature, there are a variety of figures for the constant k_w. Results from LDA measurement gave the range of 1-1.1 for this constant [28, 29]. Recent results using LDA [26] showed that this constant varies between 0.3 and 0.7 with changes in engine speed and flow rate. A literature review [30] suggests that this constant may have a value in the range 0.4 - 1.3.

In order to examine the effect of this constant on predicted engine performance, the program was run for a range of values of it. Shown in Fig. 6 are the model prediction for the in-cylinder pressure at different values of k_w (from 0.35 to 1.1). It is clear that there is a great effect of the value of k_w on the in-cylinder pressure.

The Leeds' program allows a number of variation of turbulence intensity during combustion process to be selected [3,31], including one based on the "rapid distortion theory" [32]. According to this theory the turbulence intensity during combustion process is governed by conservation of angular momentum as:-

$$u' = u'_{ig}(\rho / \rho_{ig})^{1/3} \qquad (8)$$

A preliminary comparisons between Leeds model output and experimental data suggested that this theory is inappropriate for the turbulence parameters calculation in the case of relatively quiescent disc-shaped chambers. To demonstrate the effect of the manner in which u' and L vary during the combustion process, the model was run using three methods; the resultant model output is shown in Fig. 7. In this figure, the first method refers to the rapid distortion theory, second method refers to the constant variables assumption, and third method refers to the adoption of Hall and Bracco's expression [26]. The significant differences between the model prediction for the in-cylinder pressure in this figure highlight the importance of the careful choice of the turbulence expression in the model. In the current modelling work the Hall and Bracco expression has been adopted. This is because this expression has been obtained experimentally from an engine similar to the Yamaha engine used to validate the current work.

Chemical lifetime. The chemical lifetime is defined as the time that an eddy takes to burn and is given by -

$$\tau = l_e / u_l \qquad (9)$$

where l_e is the characteristic radius of an eddy and u_l is the laminar burning velocity, at which the eddy will burn inwards from peripheral ignition sites.

The chemical lifetime is a parameter which plays an important role in the burning rate behind the flame front. If this time is greater than the eddy lifetime (the time that eddy will exist), then an eddy will break up before combustion is complete; this can lead to quenching of the flame [11,33,34]. In an earlier study with a 4-stroke engine, the value of l_e in Eq. 8 was assumed to be proportional to valve lift [17].

Tabaczynski [1], quoting reference [11] considered the turbulence structure in some detail. An eddy, size L, was assumed to contain a number of dissipative vortices, of size of order of the Kolmogorov scale. These were considered to be separated by quiescent regions of dimension of order Taylor microscale. Diffusion of a laminar nature was assumed to occur in these regions, hence the time to burn each microcell was given by -

$$\tau = \lambda / u_l \qquad (10)$$

The chemical lifetime can also be expressed as the ratio of flame thickness to laminar burning velocity [11]. Reference [35] suggested that the flame thickness would be approximately given by υ / u_l.

However to show the effect of the chemical life time on the model prediction, the model was run for two values of chemical life time and the model output is given in Fig. 8.

5.2 Blowby and Piston Top Land Crevice Flow

To show the effect of blowby and piston top land crevice flow on the model prediction, the program was run for the cycle under investigation with and without blowby and the model output data are set out in Fig.9. It is clear that including blowby in the program has reduced the in-cylinder pressure. This is attributed to the reduction in mass available for combustion close to top dead center in case of using the blowby option; about eight percent of the in-cylinder mass is computed to be in the piston ring crevices at critical stages in this cycle.

5.3 Shape Factor

Effects of the shape factor on the model prediction of the cycle under study are shown in Fig. 10. In this figure three sets of computations for different values of the shape factor are displayed. The significant differences between the model predictions highlight the importance of using this factor in the model.

5.4 Heat Transfer

Effect of heat transfer correlation on model predictions for the in-cylinder pressure is shown in Fig. 11. In this figure, the model predictions of two runs for the cycle under study (at the operating conditions set out in Table 1) are displayed. In the first run, the Woschni correlation [18] was used to calculate the heat transfer coefficient in the model, while the Annand correlation [22] was used in the second run. It is clear in this figure that the model prediction for the in-cylinder pressure varies with the correlation of heat transfer coefficient.

6. EXPERIMENTAL VALIDATION OF CURRENT MODELLING

To optimise the values of the constants used and then validate the modified model predictions, the model was initially applied to data derived from experiments with a Yamaha engine [3]. The cycle used for this verification is the first cycle in Table 1.

For this cycle, the modified computer program was run. The input to the program included the engine operating conditions for this cycle as well as the experimental value of shape factor for this cycle. The ratio between turbulence intensity to mean piston speed at ignition was set at 0.6 and the integral length scale at ignition was considered to be 0.1 times cylinder height.

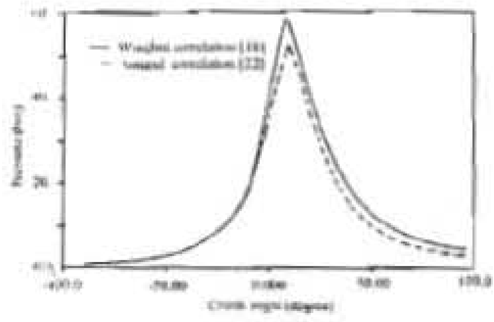


Fig. 11 Effect of heat transfer correlation on the model prediction

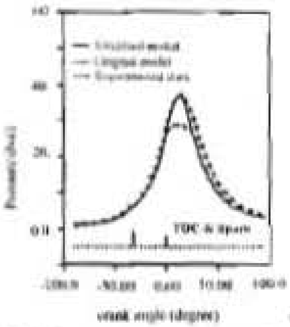


Fig. 12 (a) Comparison between model predictions and experimental data for in-cylinder pressure (1st cycle)

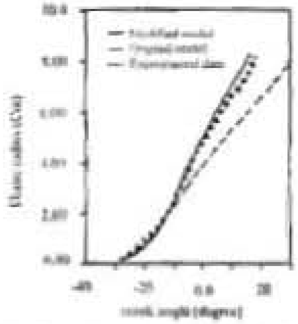


Fig. 12 (b) Comparison between model predictions and experimental data for flame radius (1st cycle)

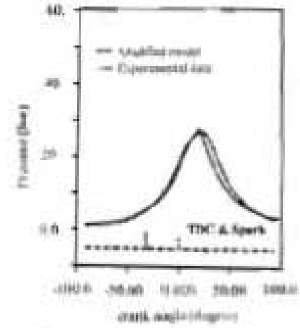


Fig. 13 Comparison between modified model prediction and experimental pressure data (2nd cycle)

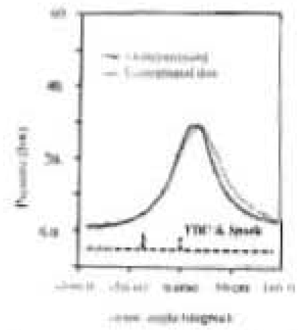


Fig. 14 Comparison between modified model prediction and experimental pressure data (3rd cycle)

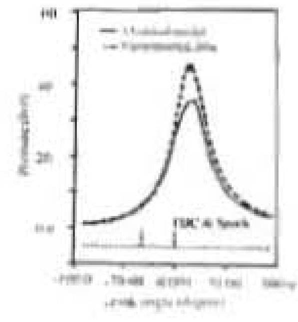


Fig. 15 Comparison between modified model prediction and experimental pressure data (4th cycle)

Based on a comparison between initial model results and the experimental data for the cycle under study, it was noted that the calculated induction time was not the same as that observed on the film; however, it should be noted that to obtain the correct start of the combustion process in the model, the measured induction time and initial kernel volume must be fed to the program.

Shown in Figs. 12 are the original model predictions, the modified model predictions and the experimental data for the cycle under study. From this figure, it is clear that the modified model prediction for the displayed parameters (pressure, flame radius) for this cycle are in good agreement with the experimental data for the same cycle. On the other hand, considerable discrepancy between the original model prediction and the experimental data for the same cycle can be noticed. The effects of turbulence in increasing flame speed are generally attributed to two mechanisms [2]. The first mechanism considers the effect of turbulence eddies of a scale less than the thickness of the flame front. These eddies are assumed to increase the local heat and mass transfer rates along the flame front, thereby increasing the local rate of flame propagation. A second mechanism concerns the effects of larger turbulent eddies. These eddies are assumed to have no effect on flame speed, but to distort the flame front so that its area is increased. The increase in flame speed is then proportional to the increased area of the flame front.

Following the optimization of the modified model for one cycle, in order to validate the modified model for the Yamaha engine, the model has been applied to the other three cycles listed in Table 1. It may be noted that each of these cycles has the same engine operating conditions as the first cycle, apart from one parameter in each case. The input to the program for each cycle was the engine operating conditions for that cycle together with the experimental values of the shape factor and the initial flame kernel size.

Comparison between the modified model output and the experimental data for the second cycle is given in Fig 13. The engine operating conditions of this cycle were the same as the first cycle except that the compression ratio was 6.2. It is clear from this figure that, during the early stages of combustion, the modified model predicted slower burning than observed in the engine data. Later in the cycle, the predicted combustion becomes faster than the experiment. Reasons of these differences may be attributed to difference in flame development associated with a more 3-dimcational chamber gometry when the compression ratio is reduced.

Another comparison between modified model predictions and experimental data is shown in Fig. 14. This comparison has been undertaken for the third cycle, which has the same operating conditions as the first except that the engine speed was increased to 1400 rpm. For this cycle, good agreement was obtained between the model and experiment from the start of combustion upto the top dead centre position; then the modelled burning seems to become slower than that observed in the experiment. This behavior may be associated with cyclic variation noted in the experiment.

Finally, comparisons for the fourth cycle are shown in Fig. 15. This cycle is characterized by a lower equivalence ratio, of 0.9. Although good agreement between the model and experiment was obtained during the early stages of combustion, considerable differences emerged later in the cycle.
In general, fair agreement between the trends of the modified model output data and the experimental data were obtained for the range of engine parameters used in the current comparisons.

7. CONCLUSIONS

In this study the Leeds engine cycle model has been extended and modified. In this model the burning velocity approach has been used to model the progressive combustion in an engine. This method is considered to be more appropriate for modelling over a wide range of conditions than purely empirical methods. The following conclusions are drawn from the current modelling work.

1. Turbulence parameters at ignition have very significant effects on model predictions. Increasing the value of turbulence intensity at ignition increased the turbulence level and burning rate throughout the combustion process. However, this value needs to be measured in the engine concerned if the turbulence level is to be reliable.

2. The assumed variation of turbulence intensity (u') and integral length scale (L) during the combustion process is of great importance. It was found that the relationships based on the Bracco's experimental data were appropriate for the engine under study; Braco's data were generated for an engine similar to that used in the current study.

3. Including the "shape factor" in the model significantly changed both trends and values of model predictions. However, further experimental work is required to develop a generalized expression for this factor.

4. Blowby and piston top-land crevice flow have significant effects on modelled predictions of cylinder pressure.

5. Although generally good qualitative agreement was obtained between the modified model predictions and experimental data, over a range of engine operating conditions, further work is needed to obtain more reliable quantitative agreement. This work should concentrate on determination of correct turbulence and heat transfer constants to be used in the model.

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