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ABSTRACT

Simultaneous measurements of early flame speed and local measurements of the major parameters controlling the process are presented. The early flame growth rate was captured with heat release analysis of the cylinder pressure. The local concentration of fuel or residual gas were measured with laser induced fluorescence (LIF) on isooctane/3-pentanone or water. Local velocity measurements were performed with laser doppler velocimetry (LDV).

The results show a significant cycle to cycle correlation between early flame growth rate and several parameters. The experiments were arranged to suppress all but one important factor at a time. When the engine was run without fuel or residual gas fluctuations, the cycle to cycle variations of turbulence were able to explain 50 % of the flame growth rate fluctuations. With a significantly increased fluctuation of F/A, obtained with port fuelling, 65% of the growth rate fluctuation could be explained with local F/A measurements. With a homogeneous fuel/air-mixture but with a high concentration of residual, a correlation could be obtained between local residual concentration and combustion.

INTRODUCTION

The spark ignition engine is established as the major type used for automobile applications. Its success can be attributed to the combination of several good characteristics and the ability to adapt to changes. For instance the requirement on exhaust emissions was meet with the addition of a three-way catalyst. But there has since the birth of the engine been a fundamental problem, the cycle to cycle variation. This limits the engine performance and gives rise to increased emissions.

This paper will first describe how the cycle to cycle variations are detected and the reasons for focusing on the early part of the combustion. Then experimental results are presented that shows a correlation between early combustion rate and fluctuations in gas composition and fluid flow from cycle to cycle. A discussion on the relative importance of fluid flow and gas composition then follows. Finally, a discussion on the unexplained variation in early combustion will end the presentation.

CYCLE TO CYCLE VARIATIONS

INDICATORS OF CYCLE TO CYCLE VARIATIONS- The combustion process in a spark ignition engine is not repetitive from engine cycle to engine cycle. This can easily be noted if the pressure trace in the cylinder is measured. The peak pressure obtained can change ñ 30% from cycle to cycle in a well functioning engine.

Historically it is the cylinder pressure that has been used to measure the fluctuations. This has led to the use of pressure related parameters to quantify the fluctuation intensity. The maximum pressure and its crank angle location are frequently used parameters [27]. The variation in indicated mean effective pressure, imep, produced per engine cycle is also a well-used parameter. The standard deviation is usually normalized with the average value to give a coefficient of variation, COV_{imep}. These parameters have the benefit of requiring no modeling and the COV_{imep} shows how much torque fluctuation that the transmission etc. must tolerate. COV_{imep} is thus a good parameter to be used for transmission design and a general indicator of engine behavior.

The major drawback with the parameters derived from the pressure directly is the lack of knowledge on the ongoing process. The only reasonable way the pressure can change from cycle to cycle is variation in the combustion process. Hence, there is every reason to use the pressure in a heat release model and analyze the heat release function instead of the pressure trace, especially if the origins of fluctuations are of interest. If even more detailed information on the combustion process is required, the heat release calculation should be replaced by some form of flame location detection.

EARLY AND LATER PARTS OF COMBUSTION- In general, the combustion in a spark ignition can be expected to fluctuate during the entire flame propagation process. The fuel and residual gases are generally not well mixed with air and hence the laminar flame speed will differ depending on location and time. The same argument can be used for the

flow situation. The level of turbulence can not be expected to be homogeneous and the mean flow situation will also change from cycle to cycle and from location to location. But even though fluctuations are expected for large flames, these flames will have the benefit of integrating out inhomogeneity in the fuel, residual and flow fields. The very small flame in the early part of the combustion does not have this possibility to average out the flame speed setting parameters. Thus, this part of the combustion is expected to have the greatest problem with cycle to cycle variation. This is also supported in the literature [1], [8], [11], [24], [26], [27], [29], [38], [40], [42], [43], [54], [60], [67], [68], [70].

Consequently, this work was focused on the fluctuation of the early combustion and the possible parameters that can effect this process.

MEASURE OF COMBUSTION RATE

AVAILABLE TECHNIQUES- The paper presented is intended as a contribution to our understanding of cycle to cycle variations in the combustion event in spark ignition engines. To enable a study of the importance of different parameters to the combustion rate, there is, however, a need for a measure of combustion rate. This measure should be sensitive to combustion fluctuations only and have the ability to differentiate between local and global effects. It should be easily obtained from a working engine with a simple measurement technique without interfering with the combustion or other engine operating parameters. At present there exists no simple measurement technique that can give the rate of heat release during the entire flame propagation event with acceptable accuracy. The present options are:

1. Use of the cylinder pressure in a heat release code.
2. Detecting the arrival time of flame at discrete positions with ion probes [39] or optical probes [68], [69].
3. Detecting the flame passage over a thin line of light, [56].
4. Flame photography by using some optical technique, [1], [3], [7], [26], [44], [58], [71].

The results presented in this paper use the cylinder pressure in a heat release analysis to determine the rate of early (and later) combustion rate. The output from this analysis is the duration in crank angle degrees from ignition to a percentage of heat released. The details for the measurement system and beat release program and a list of the code used can be found in [32]. Figure 1 shows the mean combustion duration for a lean operating case. It can be noted that combustion is progressing a substantial time without producing any significant heat. The ignition was set at 30 degrees BTDC.

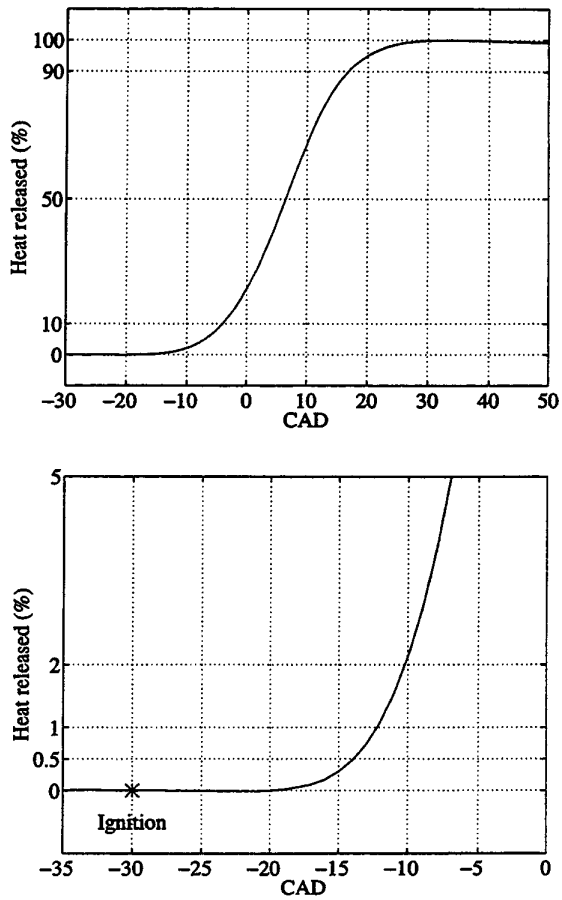

Figure 1: The heat released for a lean operating condition. The early part of the event is enlarged in the lower figure.

This average combustion is, however, not repeated from cycle to cycle. Figure 2 shows the heat released in the early part for a few individual cycles. Clearly the rate of heat release varies from cycle to cycle.

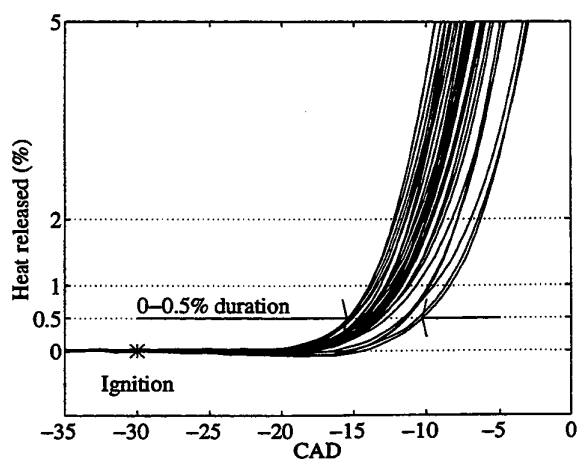

Figure 2: Early combustion in 30 individual cycles with the fluctuation in crank angle position for 0.5% heat released.

PARAMETERS GIVING CYCLE TO CYCLE COMBUSTION VARIATIONS

Cycle to cycle variations in combustion can be attributed to the cycle to cycle variations of any of the parameters known to effect combustion. A number of models have been used to describe the flame propagation [1], [2], [5], [6], [9], [10], [28], [29], [41], [55], [61], [62], [63], [65], [66]. Most divide the effects of chemistry and flow fields into two separate flame velocities, the laminar and the turbulent. The laminar flame speed has been found to respond to:

- Fuel/air-ratio, ϕ
- Inert gas concentration which can be residual gases, exhaust gas recycled (EGR), water vapor or any other gas without oxygen or fuel.
- Temperature
- Pressure
- Stress

The flow field effect is generally expressed as a flame speed increase from the laminar case. The ratio of turbulent to laminar flame speed is in common use. The parameters often used to describe this enhancement are:

- Turbulence intensity, u'
- One or several turbulence length scales. The most often frequently used is the integral length scale, l_I
- Flame size

In some cases are also the local stress and fractal dimension [45] included in the description of the turbulence enhancement, [21], [22], [44], [46], [48] [50], [51]. These parameters can, however, often be expressed in terms of a length scale and turbulence level.

Apart from these parameters the mean velocity in the vicinity of the spark plug is used in models which describe the very early flame propagation. The mean velocity does not effect the flame speed directly, but instead alters the location of the flame. This flame location in the very early part of the combustion alters the contact area between flame and walls, when the flame is small compared to the spark plug size. Increased wall contact area, in which the spark plug is included, will increase the heat loss from the flame and hence cools it. This in turn reduces the flame speed as the energy balance is altered [57].

The effect of most of these parameters on cycle to cycle variation will be presented. The effects of precombustion pressure and temperature has not previously been published and is thus presented in greater detail. The other parameters have been presented previously [30], [31], [33], [34], [35]. A brief summery of results is included here, but for more details the papers are recommended.

LAMINAR FLAME SPEED

PRESSURE

Measurement technique- The cylinder pressure was measured with a conventional quartz transducer connected to a charge amplifier. A more detailed description is given in [36].

Evaluation procedure- A special version of the heat release program was developed to enable a printout of the cylinder pressure in the crank angle interval 100 CAD ATDC to TDC for each engine cycle. The heat release program was then used with the three different options for the DC level locking of pressure in each cycle. The simplest and fastest way to treat the cylinder pressure is to assume that small variations exist from cycle to cycle. This corresponds to the solid line in Figure 3. A standard deviation of cylinder pressure in the range of 0.02-0.3 bar then results, see Figure 4. This corresponds to a COV_p of 1%. If the assumption is made that the cylinder pressure is constant from cycle to cycle during the inlet stroke, a clear improvement is obtained. The standard deviation is reduced with an order of magnitude and the COV_p is less than 0.1 % close to TDC. The same trend is obtained with the assumption of a known γ during the early part of the compression stroke. In the selected engine operating condition, (1200 rpm, λ=1.7, WOT, Skip fire 13) these two assumptions give equal errors. In other conditions slightly different results can be obtained. Pressure pegging is preferred when the composition of the fuel/ air-mixture is expected to change significantly from cycle to cycle. If variations in cylinder pressure can be expected, like for instance with unstable tuned inlet manifolds, the λ method is preferred.

Results- A scatter plot of the duration 0-0.5% heat release as a function of the cylinder pressure at ignition is shown in Figure 5. In this case the γ method was used to peg the cylinder pressure and 100 engine cycles were recorded. Figure 6 shows the correlation obtained with the three alternatives for locking the pressure level. The pressure at different CAD was in this case used as explaining parameter. It can be noted that all correlation coefficients are low irrespective of crank angle used. It is also noted that the unpegged pressure gives the lowest correlation.

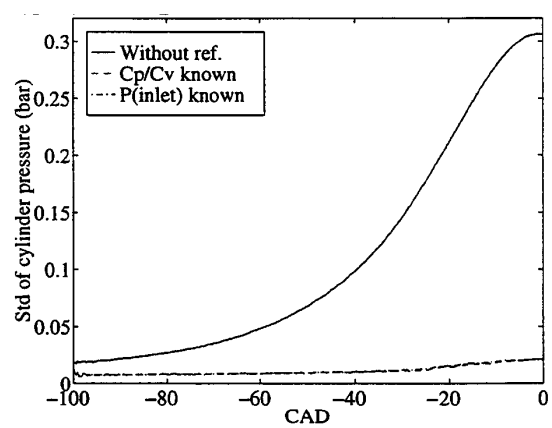

Figure 3: Standard deviation for the cylinder pressure during the compression stoke with the use of different DC-level locking methods.

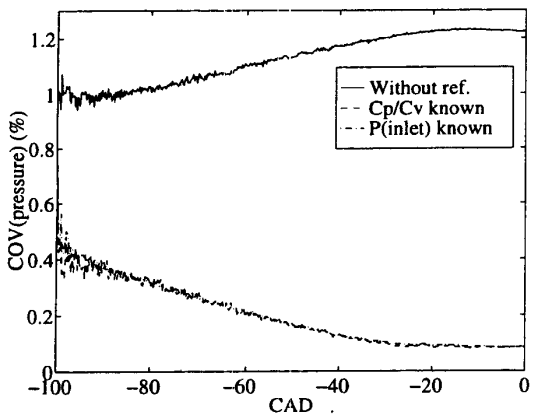

Figure 4: COV_p for the cylinder pressure during the compression stoke with the use of different DC-level locking methods.

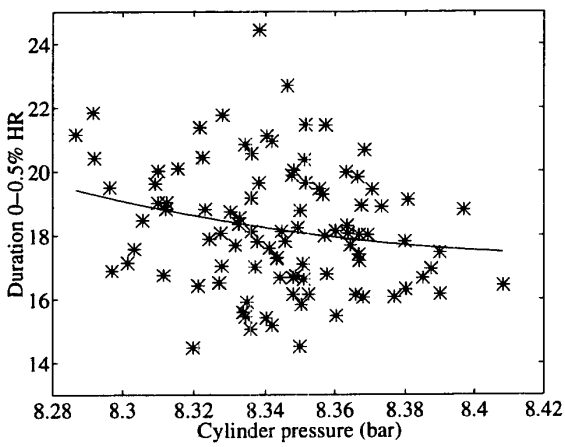

Figure 5: Duration of the early combustion, 0-0.5% HR, for different pressure levels in the cylinder.

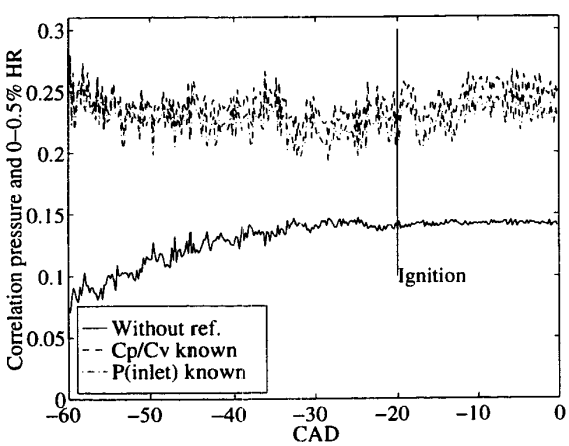

Figure 6: Correlation between cylinder pressure at different crank angle locations and the duration of 0-0.5% HR. Different pressure pegging alternatives are shown.

One interesting thing is the slope of the regression line in Figure 5. According to the experimental results, a small but detectable decrease in the duration of 0-0.5% HR shows up for an increased pressure. This should be compared to the equations governing the laminar flame speed response to increased pressure. Gülder proposed a exponent of -0.5 [23] whereas Ryan [59] gave a value of -0.565 for ϕ=0.85 and 1 -0.623 for ϕ=1.0. Both references used methane as fuel thus similar to the used natural gas.

If we assume that the equations were valid and that only the pressure changes, one standard deviation of pressure with they γ pegging would give

$$\frac{\left(\frac{p \pm dp}{p_0}\right)^{\beta}}{\left(\frac{p}{p_0}\right)^{\beta}} = \frac{\left(\frac{8.34 \pm 0.02}{1}\right)^{-0.5}}{\left(\frac{8.34}{1}\right)^{-0.5}} = \frac{0.9988}{1.0012}$$

change of laminar flame speed. This is a very small change and can by no means explain the change of flame speed detected in the engine. There is, however, a possible secondary effect of pressure fluctuations. If we assume that the variations in pressure take place before or during inlet valve closing, the temperature in the cylinder will also vary. As the temperature coefficient in the expression determining laminar flame speed is higher than the pressure corrections and also of positive value, this could be the reason behind the unexpected slope in Figure 5.

TEMPERATURE

Measurement technique- To determine the temperature in the cylinder, Dual Broadband Rotational Coherent Anti-Stokes Raman Spectroscopy (DB-RCARS) was applied This technique uses the population distributions in the rotational energy levels for nitrogen and oxygen to determine the temperature. To obtain the temperature information, three laser beams are focused and crossed in a measurement volume. From this volume four beams result. The generated signal beam contains the temperature information and is therefore lead into a spectrograph where a spectrum is recorded. This spectrum is later compared to theoretical spectra for different temperatures and Oxygen/Nitrogen concentration ratios. The best fitted theoretical spectrum then gives the temperature and concentration ratio. The experimental technique, developed by the Division of Combustion Physics, and the fitting process will not be described further in this paper, instead the results will be used, treating the DBR-CARS as a "black box". Some practical aspects and comparisons with cylinder pressure will however, be discussed.

The laser system used to generate the necessary three laser beams was a pulsed Nd:YAG in combination with a dye laser. This type of laser gives a very short laser pulse (= 10 ns) with a repetition rate of 10 Hz. This repetition rate is well suited to an engine operation speed of 1200 rpm. One laser pulse is then obtained for each combustion event. The limitation to one laser shot per cycle means that any measurements of the temperature change during the engine cycle must be made with averaging methods. For N number of crank angle positions N number of engine runs must be performed. This leads to a high requirement on the repetitiveness of the experimental set-up. Four engine runs were used with a change of triggering position of the CARS.

Evaluation procedure- The CARS spectrum evaluation is a complex and very time consuming task. This was done at the Division of Combustion Physics. See [4] for details

Results- The cylinder pressure was recorded simultaneously with the CARS measurements. This can be used to detect whether there exists any correlation between fluctuations in cylinder pressure and temperature on a cycle to cycle basis. A small variation was found in the measured temperature from engine cycle to cycle. Figure 7 shows the temperature measured with CARS and the temperature estimated from the pressure and volume traces obtained from the cylinder pressure evaluation program. A good agreement was found. The polytropic index, γ, evaluated from the cylinder pressure and volume was 1.394. When the CARS temperatures and cylinder pressures were used a value of 1.392 was obtained. In the calibration of the heat release program a value of 1.395 for 300 K is often used for the lean operation point at which the CARS measurements were performed at (λ=1.7). A linear decrease with 8.13% per 1000 K is in the HR-code. This would give an index of 1.362 at the highest temperature measured with CARS (707 K). Both cylinder pressure and CARS thus overestimate the polytropic index. This could be due to an incorrect setting of top dead center location for the pressure trace. Figure 8 shows the standard deviation of CARS temperature and this normalized with the mean temperature, COV_T as a function of temperature. The standard deviation increases with temperature, but the COV_T is very constant.

There is an interesting question concerning this deviation. Is this a deviation which can be attributed entirely to experimental scatter or does in fact the temperature change from cycle to cycle? This could then be one reason for cycle to cycle variations in the early combustion, as the laminar flame speed is sensitive to temperature fluctuations. There exist cycle to cycle variations in the cylinder pressure, but the magnitude is very small. Unfortunately this level of cycle to cycle variation depends on the pegging technique used to lock the DC-level of the pressure trace. With the three methods presented, the correlation between single cycle temperature measurements and the pressure at different crank angle positions during the cycle is shown in Figure 9. The result show that the correlation is increased with the use of a pegging procedure to reduce the cycle to cycle variations in cylinder pressure. Although the value of the correlation coefficient is small, this indicates that a pegging procedure is reasonable.

In general the correlation between temperature and pressure is low and is approaching insignificant values. This can be attributed to one of the following reasons:

- The pressure measurement has a low signal to noise ratio and is hence unreliable.
- Fluctuations in temperature could be introduced by variations in wall cooling, etc., which do not show up in cylinder pressure measurements.
- The CARS measurements contain experimental scatter.

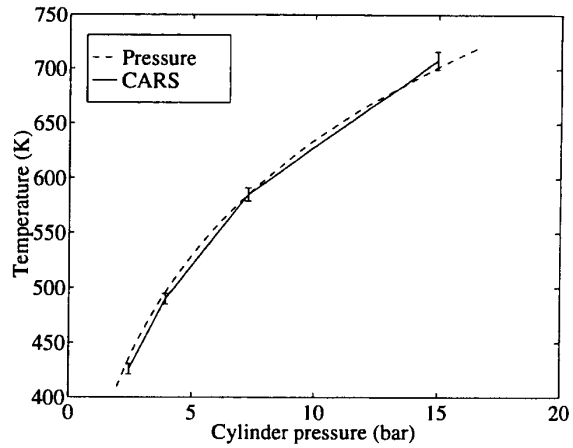

Figure 7: Temperature estimated from pressure and measured with CARS.

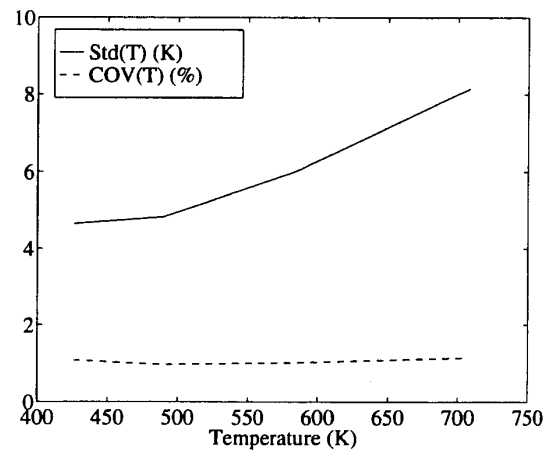

Figure 8: The standard deviation of temperature measured with CARS and the standard deviation divided by mean temperature for the four measurements.

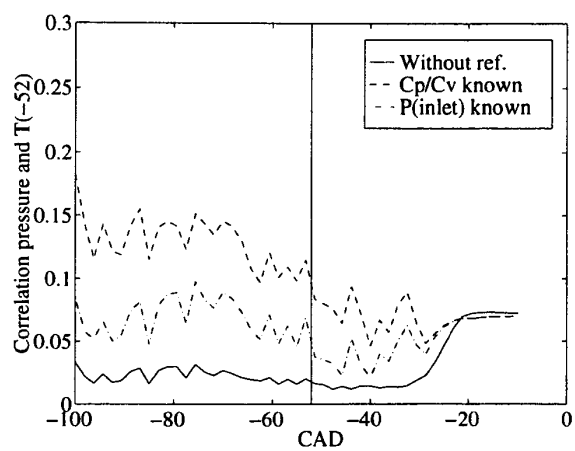

Figure 9: Correlation between measured temperature and the pressure at different crank angle positions, using different cylinder pressure pegging techniques. The vertical line indicates the time for temperature measurement.

The first statement is less likely as the cycle to cycle variations in the cylinder pressure are very low. This indicates that the pressure measurements are reliable. The second statement is also unlikely as the temperature in the cylinder follows the cylinder pressure according to the polytropic compression expression. The remaining cause must then be experimental scatter. Hence the cycle to cycle variations in temperature should be lower than the temperature fluctuations obtained with CARS. An estimation of the temperature fluctuations can be obtained by using the pressure variations. With the assumption of an polytropic compression, ΔT would equal $\Delta p^{\gamma-1/\gamma}$. With a deviation of pressure of 0.02 bar we then would have a temperature deviation of 0.34 K. This is significantly lower than the values obtained with CARS.

The variations in cylinder temperature can be used to estimate the combustion fluctuation expected from a change of laminar flame speed. If the expression presented by Gülder[23] is used, a laminar flame speed at the location 34 CAD BTDC would change

$$\frac{\left(\frac{T \pm dT}{T_0}\right)^{\alpha}}{\left(\frac{T}{T_0}\right)^{\alpha}} = \frac{\left(\frac{707.76 \pm 8.14}{300}\right)^{2.33}}{\left(\frac{707.76}{300}\right)^{2.33}} = \frac{1.027}{0.973}$$

and if the expression by Ryan [59] is used instead the variation would be

$$\frac{\exp\left(\frac{-b_3}{T \pm dT}\right)}{\exp\left(\frac{-b_3}{T}\right)} = \frac{\exp\left(\frac{-2043.5}{707.76 \pm 8.14}\right)}{\exp\left(\frac{-2043.5}{707.76}\right)} = \frac{1.037}{0.967}$$

when the variations measured with CARS are used. If only a fraction of this variation is expected to be actually present, a change of less than 1% is thus expected.

The results from the pressure and temperature measurements show that both these parameters experience only very small fluctuations from cycle to cycle. This means that their impact on the laminar flame speed and hence the flame propagation can be neglected. Therefore will these parameters not be included in any list of potential sources for cycle to cycle variations of combustion hereafter.

FUEL/AIR RATIO

Measurement technique- To obtain information on the fuel/air-ratio in the combustion chamber, Laser Induced Fluorescence (LIF) as applied. A vertical laser sheet was passed through the optically accessible engine. The fuel was doped with a tracer, 3-pentanone, which upon excitation with laser light at a wavelength of 248 nm emitted fluorescence. The emitted fluorescence was collected with an image intensified CCD camera. To calibrate the fluorescence intensity to fuel/air-ratio, the engine was run with a thoroughly premixed mixture and the exhaust gas composition was analyzed to determine λ (=1/ϕ). The LIF system was developed at the Division of Combustion Physics [52]. A detailed description of the system can be found in [53]. Figure 10 shows the schematic layout.

Evaluation procedure- The LIF images acquired were post processed in several steps in which the image was compensated for background level, fluctuating laser intensity, laser mode and compared to reference images with known fuel concentration.

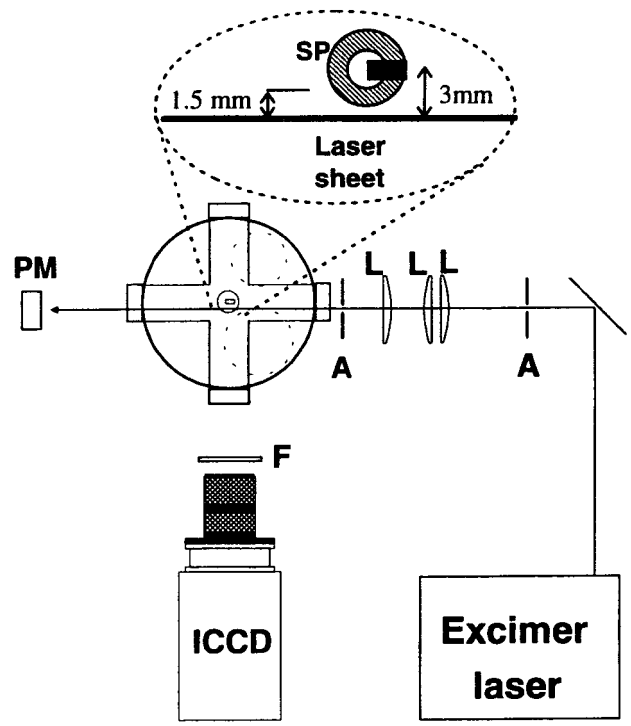

Figure 10: Experimental set-up of the Laser Induced Fluorescence for fuel concentration measurements and the processing of LIF images to obtain quantitative fuel distributions.

Results- The simultaneous LIF and cylinder pressure measurements enabled a cycle to cycle correlation between fuel concentration in the vicinity of the spark plug and the rate of early heat release. The correlation coefficient was found to depend on the fluctuations of fuel concentration. With a close to homogeneous fuel distribution, only weak correlation could be found, but when the engine was operated with a high degree of inhomogeneity, the correlation was increased to 0.8, see Figure 11. This is a high value and shows that the LIF technique can be used to accurately measure the fuel concentration in the vicinity of the spark plug. It also shows that the early flame development is strongly effected by fluctuating fuel concentration. This is expected as the laminar flame speed declines fast for lean or rich mixtures. More results can be found in [34]. Figure 12 shows that the local information on fuel concentration is of most significance. When the average ϕ within circular areas with increasing radius was used, the correlation to early combustion peaked at R= 4.35 mm.

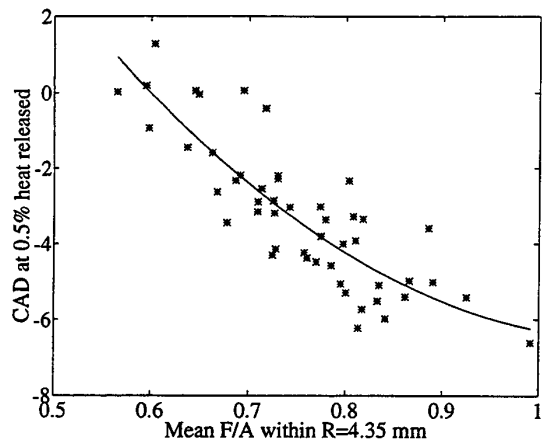

Figure 11: Scatter plot of 50 individual cycles with changing fuel concentration and early combustion rate.

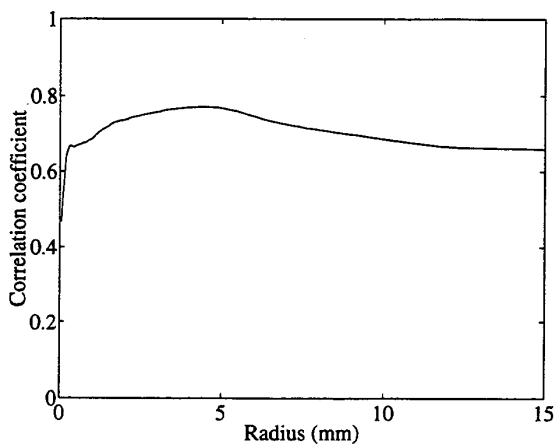

Figure 12: Correlation between fuel concentration within a specified radius and early rate of heat release.

RESIDUAL GASES

Measurement technique- The residual gases were also measured using LIF. Here the laser was used for two-photon excitation of water vapor. As the combustion process generates substantial amounts of water, the residual gases will also have a high water concentration. The fuel used was natural gas, which does not fluoresce. In these experiments, no means of calibrating the water signal to a known residual gas concentration could be used, and hence only semi-quantitative results were obtained. The experimental set-up was identical to the one used for the fuel-LIF.

Evaluation procedure- The evaluation procedure from the fuel-LIF was used with minimum changes for the water-LIF. The lower signal level from water, however, gave a more unreliable signal.

Results- The simultaneous cylinder pressure and water-LIF measurements were performed at part load with throttling [35]. This type of operating conditions was chosen to introduce significant levels of residual gas. The cycle to cycle correlation between water signal and early rate of heat release in this case depended on the percentage of residuals in the cylinder. Figure 13 shows a scatter plot of the early combustion rate versus water signal. The correlation between combustion rate and water signal is shown in Figure 14 when the water signal within a different radius was used. As can be seen, the correlation is less than the value obtained by fuel-LIF. This can depend on a lower signal level and the use of a narrower laser sheet located further away from the spark plug. This results in loss of information.

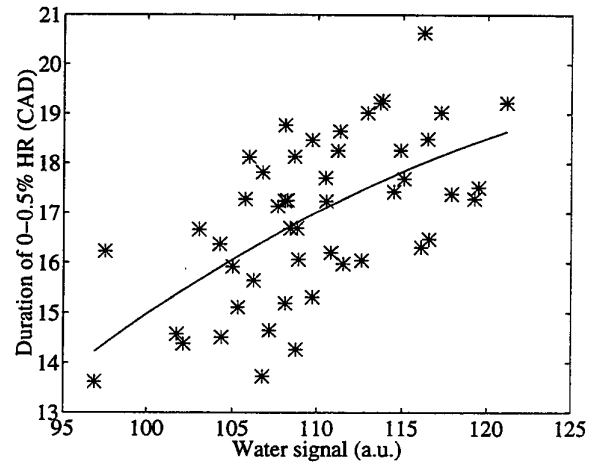

Figure 13: Duration of 0-0.5% heat released vs. average water signal within a radius of 2.9 mm from the spark plug.

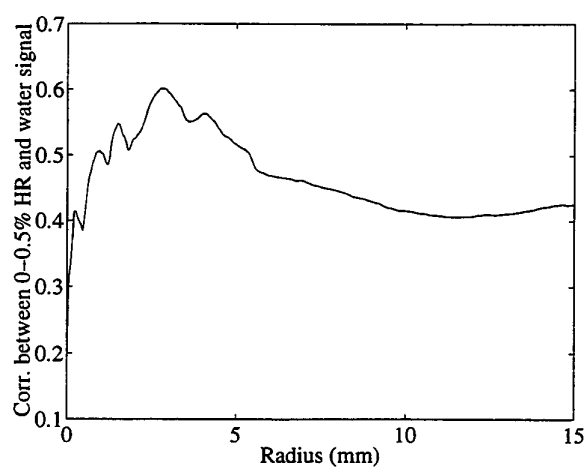

Figure 14: Correlation between 0-0.5% HR and average water signal within a specified radius.

FLUID FLOW

MEAN VELOCITY- The bulk flow pattern does not effect the flame speed directly. Instead variations in the mean velocity from cycle to cycle can convect the small flame kernel away form the cold spark plug electrodes or it can increase the wall contact area. If the flame is moved away from the spark plug less heat loss will result and hence a faster laminar flame speed can be expected [57].

Measurement technique- The mean velocity was measured with Laser Doppler Velocimentry (LDV). This technique crosses two laser beams to generate a measurement volume. Scattered light from small particles, added to the fluid flow, is then collected in any direction. The system was delivered from the supplier DANTEC, and was used without any modifications. Two velocity components were measured simultaneously.

Evaluation procedure- The output from the LDV system are bursts (velocity registrations) with a known velocity and arrival time. The nature of LDV will give a random time between the velocity samples as a burst result when a particle passes through the beam crossing. The passages of particles cannot be controlled. To convert the random arrival time data into time equidistant samples a combined resampling and low-pass filtering was adopted. This procedure used a moving window to estimate the average velocity at a given crank angle position. The procedure simply calculated the average velocity of all samples present at the current position ñ half the window width. A window function was also included to obtain a better frequency response of the mean velocity trace. The window function gives a higher weight to samples close to the center of the window.

Results- Simultaneous measurements of mean velocity and cylinder pressure gave a correlation coefficient of 0.5 between the duration of 0 to 0.5% heat released and mean velocity for the optimum measurement volume location in the vicinity of the spark plug. The correlation was found to be insensitive to changes in air-fuel ratio as indicated in Figure 15. However, the duration and standard deviation of 0-0.5% heat released changed significantly with a leaner mixture, as shown in Figure 16 [30].

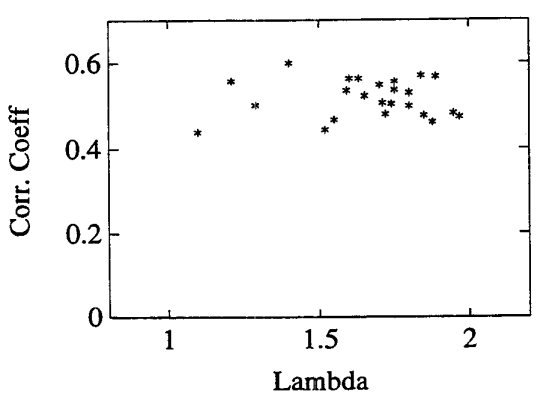

Figure 15: Correlation between mean velocity close to spark plug and duration of 0-0.5% heat released.

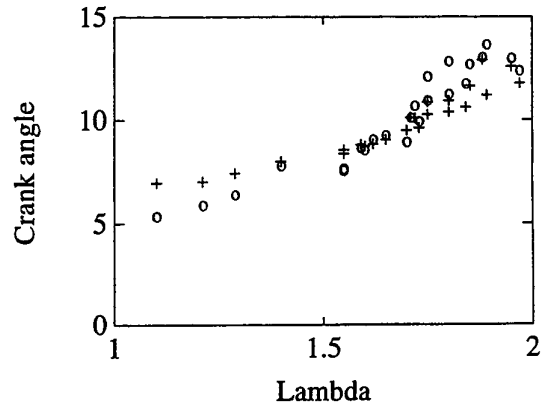

Figure 16: Duration of the early flame development (plus) and the standard deviation (circle magnified a factor ten) for the runs in Figure 15. Ignition timing at 20 BTDC.

TURBULENCE

Measurement technique- The same technique as for the mean velocity, LDV, was used to obtain information on the high frequency velocity fluctuations, known as turbulence. There was, however, a slightly increased demand on the LDV system as the data rate must be sufficiently high to enable a low-noise estimation of the turbulence.

Evaluation procedure- The turbulence was estimated as the root mean square (RMS) of the deviation between velocity samples and mean velocity trace within a window. Normally the window width was set to 10 crank angle degrees, but this width is less critical than the one selected for mean velocity filtering.

Results- A scatter plot of the crank angle position of 0.5% heat released as a function of turbulence is shown in Figure 17. There is clearly a correlation between the two variables and the correlation coefficient reached approximately 0.7. Also the correlation between turbulence and 0-0.5% heat released was measured with different air/fuel-ratios. In this case, as well as with the mean velocity, the correlation was rather insensitive to #, as shown in Figure 18 [30], [31].

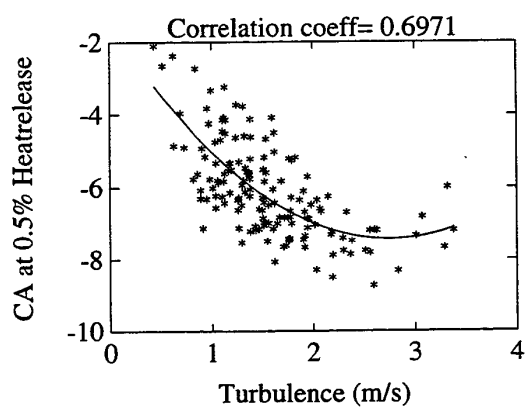

Figure 17: Crank angle position of 0.5 % heat released as function of turbulence for 150 individual cycles with fitted curve

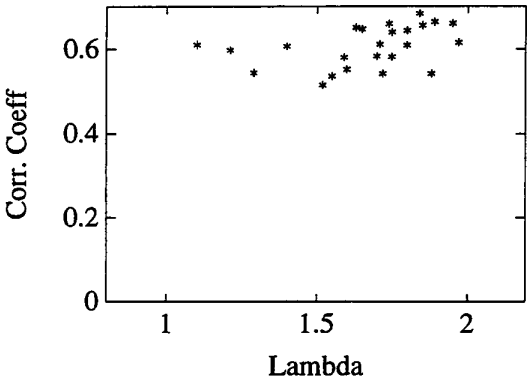

Figure 18: Correlation between turbulence close to spark plug and duration of 0-0.5% heat released.

TIME SCALE OF FLOW- Not only the turbulence intensity is used to describe the turbulent flame speed. Most models use some form of length or time scale information to describe the size of the turbulent eddies. One possible approach to measure the frequency content of the flow for each cycle will be described in this section.

Measurement technique- The measurement technique was also in this case single point LDV, but to ensure a possibility to frequency resolve the turbulence, an extremely high data rate is required to reduce the probability of data losses during the cycle.

Evaluation procedures- The separation of turbulence into different frequencies was done both in the time and frequency domains. The time domain filtering was done by simply resampling the velocity trance with different window widths. Then the RMS was calculated between two consecutive mean velocity traces and not between samples and mean velocity trace. Figure 19 shows a velocity trace and different mean velocities.

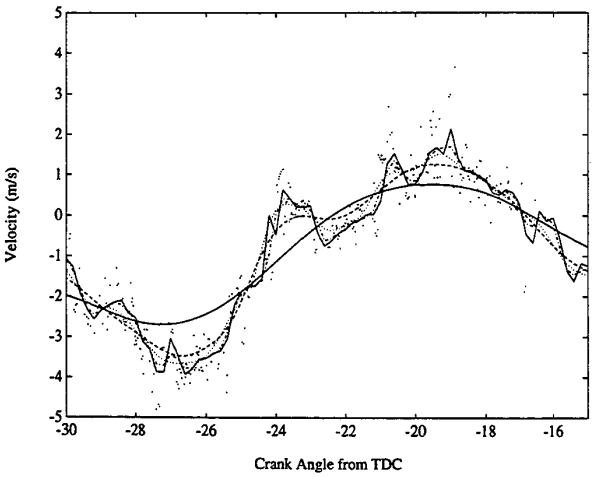

Figure 19: Plot of individual velocity registrations (dots) for one individual cycle with low-pass filtered mean velocity traces.

The separation was also done in the frequency domain by using a fast fourier transform (FFT) in the program package MATLAB. Here the velocity trace resampled with a 0.6 CAD window was analyzed.

Results- The result from the frequency domain analysis is given in Figure 20. A clear peak in correlation was found in the frequency range 4-6 kHz and thus the conclusion was drawn that this is the frequency interval that has the highest significance for early combustion. Later evaluation of the very early flame speed, evaluated from Schlieren images [47], showed similar trends [33], [64].

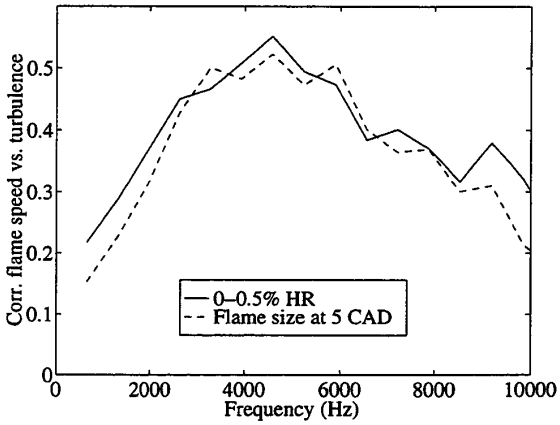

Figure 20: Correlation between the individual FFT coefficients, indicating the energy at different frequencies, and the duration of early flame development.

LENGTH SCALE OF FLOW

Lack of measurement technique- The turbulence scales are normally not given in the time domain. More common is to use some form of length scale to characterize the turbulent spectra. The average turbulent integral length scale has been measured with the use of two-point LDV in several engine configurations [14], [16], [17], [19], [20]. However, there is no reason to believe that the length scales would be stable from cycle to cycle when the turbulence level has shown to vary significantly. However, at present there is no available technique for measuring an integral length scale in each cycle of a working engine. The most common whole field velocimeter, Particle Image Velocimetry (PIV), has a limited number of interrogation areas, or measurement cells available. If a video based PIV system should be used, a maximum of 64 by 64 measurements can be obtained. Thus in any direction only 64 samples are available. Film based PIV cannot be considered as an alternative when approximately 200 individual engine cycles are required for each engine run to get a reasonable statistical confidence interval of the results. The evaluation time would be extensive. A promising alternative is Doppler Global Velocimetry (DGV), but this technique still requires refinement before it can be used in an engine [37].

Indirect evaluation with the use of time scales- The lack of measurement technique for the length scale on a cycle to cycle basis leads to the use of an indirect evaluation using Taylor's hypothesis and a time scale according to

$$l = \frac{\overline{U}}{f}$$

where l is length scale information, $\overline{U}$ mean velocity and f frequency. The hypothesis is only valid if the velocity fluctuation (turbulence) is much smaller that the mean velocity. This is not the case in the engine and the uncertainty can therefore be considered large. Integral length scale estimations from measured integrals time scales and Taylor's hypothesis have been shown to give results within a factor of 2.5 from the length scales directly measured [12], [13].

Results- The correlation between the duration of 0-0.5% heat release and the length scale information obtained from the frequency resolved turbulence analysis and Taylor's hypothesis is shown in Figure 21. The smaller flame measured with the Schlieren technique shows a lower correlation to large eddies. This is also expected as large eddies is less effective in wrinkling small flames.

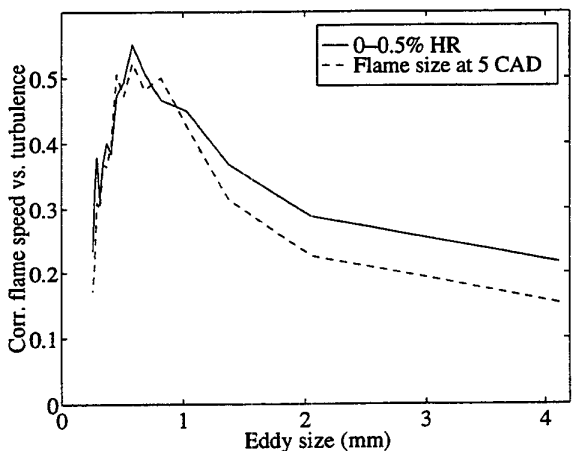

Figure 21: Correlation between eddy size and duration of 0-0.5% heat released.

THE RELATIVE IMPORTANCE OF THE DIFFERENT PARAMETERS FOR CYCLE TO CYCLE VARIATIONS

SELECTED OPERATING CONDITION FOR EXPERIMENTS - The results presented concerning cycle to cycle variations have all been measured in the same engine configuration. The operating conditions have, however, been changed to stress the engine to give large variations in the parameters studied, but to suppress fluctuations of any unmeasured parameter.

The flow and spark characteristics measurements were performed with the engine running with skip fire, i.e. with the ignition cut during a few cycles before the measurement cycle. The engine was fed with a premixed natural gas/air mixture during these measurements. This selection of operating condition almost eliminates the effects of fluctuating residual gas concentration and fuel air ratio. Hence, the only remaining causes for cycle to cycle variations should be flow fluctuations. If accurate measurements of these variables were available, a full explanation of the combustion fluctuations should be attainable.

The fuel concentration measurements were taken with the engine operating in a skip fire mode to reduce the influence of residuals. The fluid flow in the cylinder can not be stabilized and will as a consequence act as "noise" during the correlation between fuel concentration and combustion rate. To increase the inhomogeneities in the cylinder, the fuel was injected close to the inlet valve. Fuel injection against both open and closed valve were tried.

The residual gas measurements were done with the engine operating at part load without skip fire. The fuel was in this case well premixed far upstream the inlet valve. The fluctuation due to fluid flow was still present.

The precombustion temperature measurements were performed in the last motored cycle during skip fire. This choice was not the optimal one as no correlation between temperature and combustion was possible. The reason for this choice of measurement position was interference between ignition system and trigger system of the CARS laser.

The choice of operating conditions for the different type of experiments is summarized in Table 1

THE FLUCTUATION OF 0-0.5% HEAT RELEASED AND THE CORRELATION OBTAINED- The fluctuation of 0-0.5% heat released duration has not been kept constant during the different experiments. The lowest values were reached with skip fire and a stoichiometric premixed fuel/air-mixture. This gave a standard deviation of 0.5-0.6 CAD. With a lean mixture the deviation increased to 1.3-1.5 CAD depending on ignition timing. The correlation between duration of 0-0.5% heat released and mean velocity was 0.5. The turbulence gave approximately 0.6-0.7. Both correlation coefficients were found to be modestly increased as # increased.

The fuel concentration experiments gave a standard deviation of 0-0.5% heat released of 0.8-1.9 CAD depending on injection strategy with an average λ of 1/(0.76-0.86)=1.16-1.32 . Here the correlation coefficient depended on the deviation of 0-0.5% heat released. A large deviation gave an increased correlation coefficient.

The residual gas measurements experienced a similar trend. The standard deviation of 0-0.5% heat released became 0.8-1 CAD for a stoichiometric case at part load. The operating conditions with λ=1.5 showed values of 1.18-1.6 CAD with an increase for lower loads. Also in this case the correlation between residual gas concentration and duration of 0-0.5% heat released was best for conditions with an unstable engine operation.

The fluctuations of 0-0.5% heat released, the correlation between this and some parameters and the estimated fluctuations that can be explained are summarized in Table 1. The explained variation given in this table is estimated according to

$$CAD_{explained} = R^2\, CAD_{std}$$

where R^2 is the correlation coefficient squared [15]. The unexplained deviation is thus ($1-R^2$) times the standard deviation.

Table 1: Operating conditions for the different studies presented

Parameters studied	Parameter eliminated	Uncontrolled parameters	Skip /fire	Fuel preparation	Engine load	Engine speed
Fluid flow	Fuel conc.,	Spark	3:3	Premixed	WOT	700-1000
Fluid flow and Spark	Fuel conc., residuals	None obvious	3:3	Premixed	WOT	700
Fuel conc.	Residuals	Fluid flow, Spark	3:3	Port injection	WOT	700
Residuals	Fuel conc.,	Fluid flow, Spark	none	Premixed	0.3 bar-WOT	700
Temp.	Fuel conc., Residuals	Fluid flow,Spark	3.3	Premixed	WOT	1200
Pressure	Fuel conc., Residuals	Fluid flow, Spark	3:3	Premixed	WOT	1200

Table 2: Summary of correlation obtained between different parameters and the duration of 0-0.5% heat released.

Case	Ref.	Pin	λ	x_b	Std 0-0.5%	Correlation to 0-0.5% HR				expl.	unexpl.
#	#	Bar	-	a.u.	CAD	mean vel	turb	λ	x_b	CAD	CAD
1	[30]	1	1.0	-	0.5	0.5	-	-	-	0.125	0.375
2	[30]	1	1.0	-	0.5	-	0.6	-	-	0.18	0.32
3	[30]	1	1.3	-	0.6	0.5	-	-	-	0.15	0.45
4	[30]	1	1.3	-	0.6	-	0.6	-	-	0.216	0.384
5	[30]	1	1.8	-	1.3	0.5	-	-	-	0.325	0.975
6	[30]	1	1.8	-	1.3	-	0.7	-	-	0.637	0.663
7	[31]	1	1.0	-	0.5	0.7		-	-	0.245	0.255
8	[31]	1	1.3	-	0.6	0.75		-	-	0.338	0.262
9	[31]	1	1.8	-	1.3	0.8		-	-	0.832	0.468
10	[34]	1	1.32	-	1.92	-	-	0.8	-	1.23	0.69
11	[34]	1	1.25	-	1.5	-	-	0.65	-	0.634	0.866
12	[34]	1	1.22	-	1.0	-	-	0.62	-	0.384	0.616
13	[35]	0.3	1	120	1.58	-	-	-	0.6	0.569	1.011
14	[35]	0.65	1.5	45	1.20	-	-	-	0.41	0.202	0.998
15	[35]	1.0	1.5	35	1.17	-	-	-	0.3	0.105	1.065

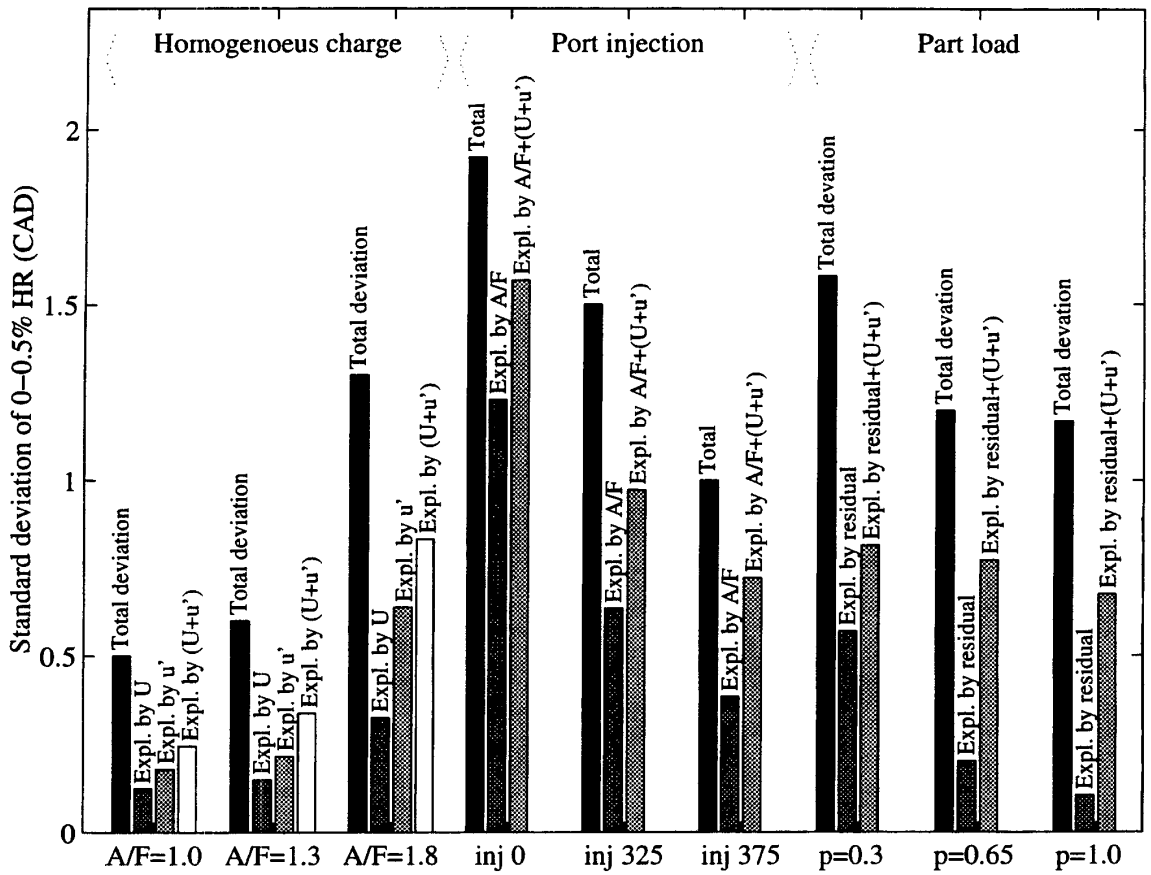

Figure 22: Deviation of 0-0.5% heat released and the ability of different variables to explain this variation. The three lefts sets of bars show total variation, variation explained by mean velocity, turbulence and both of them. The six sets of bars at right show total variation, variation explained by measured fuel or residual fluctuation and the variation explained by fuel/residual in combination with both flow parameters. With port injection, the engine was run at l=1.3. The first part load case was run at l=1.0 and the other two at l=1.55.

Table 2 shows that the mean velocity and turbulence separately can explain less than half the variations for any λ. The turbulence with a higher correlation to 0-0.5% heat release, can explain slightly more. As mean velocity and turbulence were measured at the same time, a multiple regression model can also be used. The multiple correlation coefficient obtained from this analysis was higher than the one obtained from turbulence or mean velocity alone. Hence a greater part of the fluctuations can be explained. It is, however, worth noting that the sum of explained deviation from case 1 and 2 is higher than the one from case 7. This shows that there exists a correlation between mean velocity and turbulence and they are as consequently not independent.

The introduction of fuel or residual gas inhomogeneity increases the fluctuation of 0-0.5% heat released significantly. In the worst case with an extremely inhomogeneous fuel charge the standard deviation of 0-0.5% heat released was more than tripled. This increased fluctuation could be well explained by the fluctuations in fuel concentration in the vicinity of the spark plug. The unexplained deviation was 0.6-0.8 CAD. This is approximately the variation detected with premixed fuel charge.

The residual gas measurements were less successful in explaining the increase in fluctuations with the engine operating at part load. In this case 1 CAD deviation remained unexplained. This is most likely an effect of the less accurate water-LIF technique and the reduced measurement volume used, compared to fuel-LIF.

Total explanation for the cycle to cycle fluctuations with, inhomogeneous charge- It is tempting to use the explained deviation from the flow measurements also in the results of the inhomogeneity experiments. The results from this summary of explaining factors are shown in Figure 22, It must be pointed out, however, that no information is available on the correlation between flow parameters and local concentrations resulting from charge inhomogeneity. A better mixing is expected if the turbulence level is high. Hence a correlation between charge inhomogeneity and turbulence is expected. There is a known correlation between the change inhomogeneity and the fluctuation in local concentration close to the spark plug. This expected correlation between turbulence and local concentration

means that the total explanation indicated in Figure 22 is most likely an overestimation as local concentration

and turbulence is expected to. However, the information from Figure 22 can still be of some interest.

Can then the general question of relative importance of fluid flow, fuel concentration and residual concentration be answered by these results? Unfortunately not as the engine used is only weakly related to a modern four valve spark ignition engine. This means that the combustion process in this engine will not necessarily be similar to the one in a production unit. Preliminary measurements of correlation between flow and combustion in a four-valve pentroof engine have revealed results very similar to the ones presented here but it is still to early to say that the results are generally applicable.

One definite result from the experiments is that the major factor determining the fluctuation of early combustion changes for different engine operating conditions. If the fuel injection gives a well-premixed charge and if the load is high, the major parameter is the fluid flow. If the fuel is less premixed, fuel fluctuations will dominate. At very low loads, the residual concentration will be the major parameter to generate cycle to cycle variations. A normal spark ignition engine will operate at part load, with the fuel semi-premixed. This means that the relative importance of the three parameters generally is unknown, and will change even with a slight variation in engine operating conditions.

DISCUSSION

THE CORRELATIONS OBTAINED AND THE UNEXPLAINED VARIATIONS- Figure 22 is to a large extent the summary of the data presented. The results presented in this figure show that a substational fraction of the fluctuations in early combustion can be explained by the variations in fluid flow and charge inhomogeneity from cycle to cycle. The total explanation is in most cases in the order of 50% to 85% with the assumption of no correlation between fluid flow and inhomogeneity. What then is hidden in the remaining unexplained 15-50%?

First of all it must be stressed that the explained variation is obtained from experimental data. Experimental results do always contain uncertainties and lack of information. The velocity measurements were done in a single point and only two of three components were recorded at a time. This is not enough information of the time dependent three-dimensional flow field in the vicinity of the spark plug. The flow inside the spark gap has been reported to be significantly different from the bulk flow outside the gap [25]. Hence will the early flame kernel experience different flow situation during its growth. The turbulence estimation from a limited number of samples will also introduce an uncertainty.

The inhomogeniety effects, measured with LIF, also have uncertainties. The two-dimensional image of fuel concentration can not give full information on the three-dimensional situation in the engine. In the case of fuel-LIF this problem with lacking information can, however, not be severe as the correlation obtained between fuel concentration and combustion rate is high.

The results from the correlation performed between early combustion rate and fuel concentration within areas of increasing size showed that the optimum size was in the order of 4 mm, see Figure 12. We can assume that this radius is a measure of the "integral length scale of fuel inhomogeneity". The distance from the laser sheet to spark plug center was in this case 3 mm. This gives us a total distance from spark plug center of 5 mm. An integral scale is defined as the maximum distance between two points with a correlation between the value of some parameter, measured simultaneously at the two points. The volume within which we have information can thus be estimated to be of 2*5 mm times the height of the laser sheet. This volume covers the major part of the location of very early flame kernel development, including the spark plug.

In the case of water-LIF, the correlation peaked at approximately 3 mm, see Figure 14. The distance from the spark plug center to laser sheet was in this case 4.5 mm. This gives almost the same "integral length scale of inhomogeneity" but the center of the probed volume was in this case further away from the spark plug. This means that the information was of lower value. The lower height of the laser sheet also reduces the information obtained.

What parameters are then left unmeasured? The major unmeasured parameter is the length scale of turbulent flow. Most models which describe the turbulent flame propagation use some form of length scale as input. There is no reason to believe that the (integral) length is constant from cycle to cycle, but how large fluctuations that really exist are still unknown. The importance of this presumed fluctuation to the combustion rate is also unknown. A second unmeasured parameter is the flame contact area to the cold electrodes. The Schlieren images have some information on the electrode contact area, but due to the very time consuming post-processing, this information has not been extracted from the raw images. Preliminary correlation between flame size and flame locating have shown only a weak relationship.

SUGGESTED FUTURE WORK

- A major drawback with the presented results is the lack of simultaneous experiments with all three major contributors to cycle to cycle variations; fluid flow, fuel and residual gas inhomogeneities. The only way to determine the crosscorrelation between these parameters is to perform simultaneous measurements.
- The experiments were conducted in a less production-like geometry to simplify the optical access necessary for all measurements. Any general applicable result is difficult to obtain in a combustion engine and thus a more production like geometry would be much preferred.
- The engine speed was very low during most experiments. The effect of engine speed on in-cylinder mixing of fuel and residuals would be of interest.
- The tracer/fuel combination for fuel visualization had one single boiling point. This is not the appearance of conventional fuels which instead has a destillation curve. Measurements of several different tracers would probably show interesting phenomena.
- The measurements of fuel and residuals were limited to a region close to the spark plug. Some form of global

measure of the total amount of fuel and/or residuals would supplement this information. This could be done with a less focused, thick, laser beam or with the use of multi-pass arrangements.

- Most results were correlated against the duration of 0-0.5% heat released, obtained from the cylinder pressure. Simultaneous measurements of flow and growth rate of very small flames were carried out, but simultaneous measurements of fuel and/or residual gas concentration and very early flame growth are still lacking.

CONCLUSIONS

The presented experimental findings show that different important contributors to the cycle to cycle variation of combustion in a spark ignition engine are present, depending on operating condition.

If the fuel and air are well mixed and the engine load is high, the dominant cause of variations in the early combustion is the flow situation in the vicinity of the spark plug

Fuel concentration measurements with LIF in the vicinity of the spark plug revealed an inhomogeneous charge if port fuel injection was used. A dramatically increased fluctuation of early combustion from cycle to cycle, could be explained by the fluctuating fuel concentration.

Water content measurements with LIF showed a correlation between early combustion rate and residual gas concentration in the vicinity of the spark plug. The increased fluctuation in this case could only to a smaller extent be explained by the LIF measurements.

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