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A New Method to Determine the Start and End of Combustion in an Internal Combustion Engine Using Entropy Changes

Mohand TAZEROUT and Olivier LE CORRE
Ecole des Mines de Nantes
4 rue Alfred Kastler, BP 20722, 44 307 Nantes Cedex 3 - France
E-mail: lecorre@emn.fr

A. RAMESH*
Indian Institute of Technology Madras
(Presently at Ecole des Mines de Nantes, France)

Abstract

Simulation studies indicated that the start and end of combustion in an internal combustion engine could be determined from the points of minimum and maximum entropy in the cycle. This method was further used to predict the beginning and end of the combustion process from experimentally obtained pressure crank angle data from a natural gas operated, single cylinder, spark ignition engine. The end of combustion always matched well with the point of maximum entropy. The start of combustion could be determined easily from the rate of change of entropy, which showed a sharp change at ignition. Results closely agreed with those obtained from a heat release analysis.

Key words: combustion analysis, entropy, spark ignition engine, natural gas engine, entropy variation

1. Introduction

An analysis of the cylinder pressure – crankangle data of an internal combustion engine can yield information on the starting and ending of the combustion process. It is well known that these parameters have a significant influence on the thermal efficiency and peak cylinder pressure of an engine (Hsu 1984). Various methods of determining the start and end of combustion in internal combustion engines have been proposed in literature over the years. The use of log (pressure) versus log (volume) plots provides a direct way to determine these quantities (Young and Lienesch 1978, Varaprasada Rao et al. 1993). Such a plot obtained using experimental data is seen in *Figure 1*. The compression and expansion processes are nearly straight lines indicating that the corresponding polytropic exponents are almost constant. The start and end of combustion are determined as the points indicated in *Figure 1,* where the trend deviates from a near linear behaviour. These points have to be located with care by drawing tangents and hence a high accuracy cannot be expected. Other diagrams involving related quantities such as pV^{γ}, log(p), log(V), their derivatives and polytropic exponent plotted versus crank angle have also been used instead (Varaprasada Rao et al. 1993, Harrington 1975). Often the heat release rate is estimated by a first law analysis of the averaged pressure crank angle data (Eyzat and Guibet 1968, Gupta et al. 1996, Hayes et al. 1986, Gatowski et al. 1984). These procedures are based on classical single zone models with varying degrees of complexity.

Calculations generally involve assumptions for the temperature of the cylinder wall, properties of the working substance etc.. The heat transfer coefficient is generally calculated using well-accepted correlations. In their review, Varaprasada Rao et al. 1993, found certain discrepancies between the results obtained with one or the other of the approaches mentioned earlier. However, no conclusions were drawn regarding the accuracy of these approaches.

* Corresponding Author

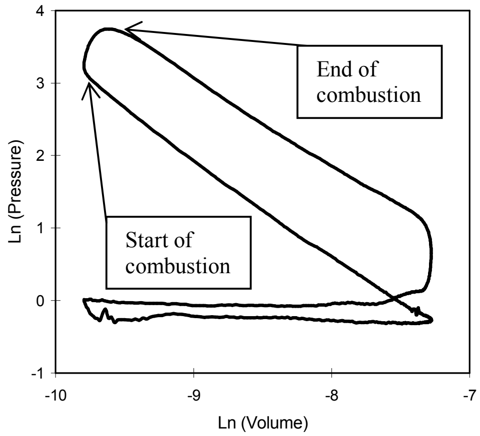

Figure 1. Variation of ln(Pressure) with Log(Volume) obtained from experimental data

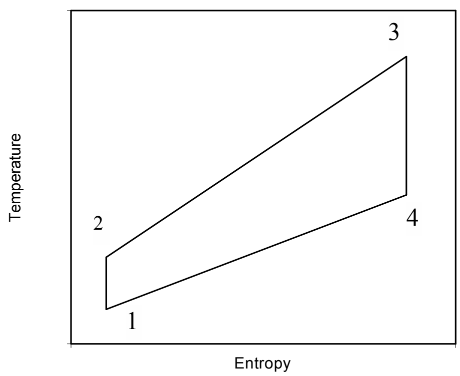

Figure 2. Temperature – entropy variation in an ideal Otto cycle

The entropy variation in a ideal Otto cycle is seen in *Figure 2*. It is seen that the start and end of combustion are quite well defined as points 2 and 3, i.e. minimum and maximum entropies respectively. However, in an actual engine, the compression and expansion processes are not isentropic due to the influence of heat transfer and irreversibilities. Hence, whether the entropy in the cycle can in some way be related to the combustion process is to be studied. The rate of heat transfer and rate of combustion will have a significant influence on the actual entropy variation in the cycle.

2. Present Work

A new approach to determine the start and end of combustion in a spark ignition engine, using the experimentally obtained Pressure – crank angle data is presented here. The method uses the calculated values of entropy of the cylinder contents during the closed period of the cycle. At first a simple simulation program was used to ascertain the relationship between the entropy and heat release variations. The simulation program used a heat release law as the input and hence the start and end of combustion was well defined. The pressure crank angle output of the simulation program was used to calculate the entropy variations and this was related to the start and end of combustion. This exercise demonstrated the suitability of the method. Hence, the influence of variations in parameters like heat transfer coefficient, wall temperature and compression ratio on the suitability of this method were studied. In the next step, the method was applied to analyse experimentally obtained pressure crank angle data from an SI engine run on natural gas. These results are presented and discussed.

3. Simulation and Heat Release Programs

A simple program to simulate the closed period of the engine cycle was developed. This program assumes air to be the working medium and the specific heat to be a function of temperature as given below (Hayes et al. 1986).

$$\frac{c_p}{R} = \left[3635.9 - \frac{1337.36T}{1000} + \frac{3294.21T^2}{10^6} - \frac{1911.42T^3}{10^9} + \frac{27.5462T^4}{10^{12}}\right]$$

for T<1000 K

$$\frac{c_p}{R} = \left[3044.73 - \frac{1338.05.T}{1000} + \frac{488.256.T^2}{10^6} - \frac{855.475.T^3}{10^{10}} + \frac{570.1327.T^4}{10^{15}}\right]$$

for T>1000 K

(1)

where c_p is the specific heat, R is the gas constant [$Jkg^{-1}K^{-1}$], T is the temperature [K].

Heat transfer coefficient h [$Wm^{-2}K^{-1}$] was calculated by using the correlation given below (Hohenberg 1979).

$$h = C_1.V_c^{-0.06}.p^{0.8}.T^{-0.4}.(v_p+C_2)^{0.8} \quad (2)$$

where C_1 and C_2 are constants with values 130 and 1.4 respectively. V_c is the cylinder volume in m^3, p is the pressure in bar, T is the temperature in K and v_p is the mean piston speed in m/s. The combustion chamber wall temperature was assumed to be a constant. Temperature at inlet valve closure was given as an input to the simulation program and was used to calculate the mass in the cylinder. Heat release rate was assumed to vary according to the cosine law as given below (Ferguson 1986)

$$x = 0 \text{ when } \theta < \theta_s$$

$$x = \frac{1}{2}\left[1 - \cos\left(\frac{\Pi(\theta - \theta_s)}{\theta_b}\right)\right] \text{ when } \theta_s < \theta < (\theta_s + \theta_b)$$

$$x = 1 \text{ when } \theta > (\theta_s + \theta_b)$$

(3)

Where θ is any crank angle. θ_s is the crank angle at start of combustion, θ_b is the crank angle at the end of combustion and x is the mass fraction of fuel burned at crank angle θ. This model allowed the start and end of combustion to be varied easily and hence was useful in this analysis. Entropy was calculated from the pressure and temperature variations. Heat release analysis of the experimentally obtained pressure crank angle data was done using a simple software developed for the purpose. Calculations were done based on an assumed value of the polytropic index during compression and expansion. The polytropic index was taken as 1.32. The equation used for the calculation of the heat release rate is given below.

$$\Delta Q = \frac{n}{(n-1)} p.\Delta V + \frac{1}{(n-1)} V.\Delta p \quad (4)$$

Where Q is the heat released [J], p is the pressure [Pa] and n is the polytropic index. This method was used as it did not involve estimations of heat transfer, assumption of wall temperature and gas property relations. The heat release was normalised based on its maximum value. The entropy of the medium in the cylinder was estimated during the closed period of the cycle by using the variation of the temperature and pressure as given in the equation below.

$$\Delta s = c_p . \ln \frac{T}{T_0} - R . \ln \frac{p}{p_0} \quad (5)$$

Where s is the specific entropy in $Jkg^{-1}K^{-1}$, T_0 and p_0 are reference temperature and pressure. The reference condition was taken as the point where the inlet valve closed.

Temperature was calculated using pressure and volume variations. The temperature at the point where the inlet valve closed was assumed. This indirectly evaluated the mass trapped. It was seen that the value of this temperature did not significantly affect trends of entropy variations, which are important in this approach. Specific heat at constant pressure needed for the evaluation of entropy, was calculated using Eq. 1. The datum for entropy was taken as the crank angle at inlet valve closure.

4. Experimental Set up and Experiments

A single cylinder spark ignition engine run on natural gas was used for the experiments. It was coupled to an electrical dynamometer and was run at a constant speed of 1500 rpm. The details of the engine are given in TABLE I.

Cylinder pressure versus crank angle data was obtained using a flush mounted AVL make pressure transducer. The data was acquired on a personal computer. Since pressure signals obtained from piezo electric transducers do not indicate absolute values but only changes, the obtained data had to be corrected. This was done by assuming that the cylinder pressure in a small crank angle window near the bottom dead centre of the suction stroke is equal to the measured mean intake manifold pressure in the same crank angle window. The intake manifold pressure was measured using a piezo resistive transducer capable of giving the absolute pressure. Data from several cycles were averaged to get the mean pressure crank angle variation, which was used as input in calculations.

TABLE I. ENGINE DETAILS

Type	Spark Ignition, Air Cooled
Bore	95. 3 mm
Stroke	88.8 mm
Compression ratio	12.37
Rated power	5.5 kW @1500 rpm

5. Results and Discussion

5.1 Simulation

The simulation program was run with the main inputs given in TABLE II. *Figures 3 and 4* indicate the variation of entropy with crank angle and temperature with entropy.

TABLE II. INPUTS FOR THE SIMULATION PROGRAM

Bore	95. 3 mm
Stroke	88.8 mm
Compression ratio	9 to 12.5
Speed	1500 rpm
Start of combustion	15° to 45° before TDC
End of combustion	15° to 45° after TDC
Temperature at IVC	270 to 340 K
Pressure at IVC	0.3 to 0.8 bar
Wall temperature	350 to 550 K
Heat transfer coefficient	0 to 200 % of actual

As the compression proceeds from the point where the inlet valve closes, the temperature increases. As long as the gas temperature is lower than the cylinder wall temperature, heat transfer is to the gas and hence the entropy increases (from 1 to 2 in *Figures 3 and 4*). The entropy decreases when the gas temperature exceeds the wall temperature (from 2 to 3). When the heat release rate due to combustion is higher than the heat transfer rate to the cylinder wall, entropy begins to rise (point 3 in *Figures 3 and 4*). Again when the heat transfer rate exceeds the combustion rate, i.e. towards the end of the combustion process, the entropy starts to decrease. Thus a peak value is reached at state 4 indicated in *Figures 3 and 4*.

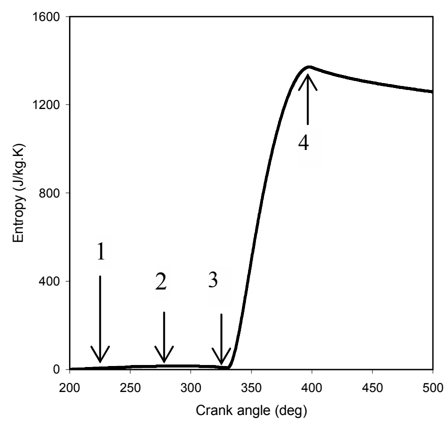

Figure 3. Variation of entropy with crank angle

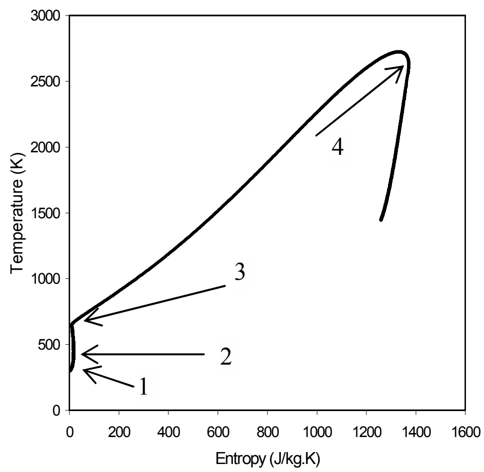

Figure 4. Variation of temperature with entropy

An examination of the results indicated that the point of minimum entropy coincided with the beginning of the combustion process. Maximum entropy was always very close to the end of the combustion process. Results were obtained with parameters like wall temperature, compression ratio, and temperature of the gas at the beginning of the compression stroke and heat release rate being varied. The influence of the above parameters on the observed trends was negligible. However, the heat transfer rate had a small influence on crank angle of occurrence of the maximum of entropy. Heat transfer coefficient that was predicted by the correlation given in Eq. 3 (Hohenberg 1979) was changed using a multiplying factor. Its influence on the entropy maximum and minimum crank angles is seen in *Figure* 5. The entropy minimum point, which occurs near the start of combustion, is not significantly affected by the heat transfer coefficient. This is because the heat transfer is negligible at this condition on account of the low gas temperature. However the crank angle at which maximum entropy occurs slightly deviates from the actual end of combustion as seen in *Figure* 5. On the whole it was concluded that the beginning and end of combustion can be obtained by determining the crank angles at which entropy minimum and maximum occur.

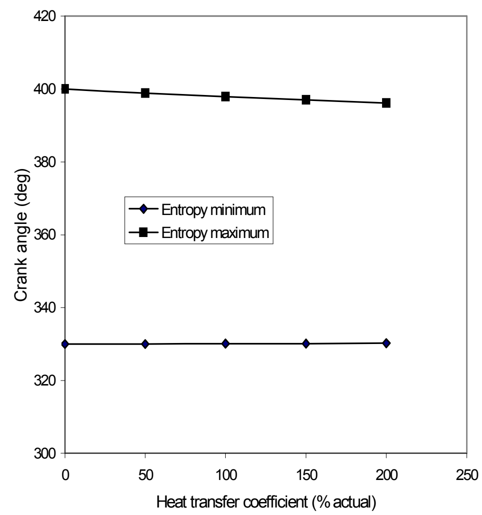

Figure 5. Variation of entropy maximum and minimum points with heat transfer coefficient

5.2 Experiments

Experiments were conducted at a constant speed of 1500 rev/min. The throttle opening was also fixed; Equivalence ratio was varied and at each operating condition the spark timing was adjusted to MBT (minimum advance for best torque). Heat release and entropy were obtained during the closed period of the cycle as mentioned earlier. The variation of the heat release at different equivalence ratios is seen in *Figure 6*. It is clear that the combustion duration becomes longer as the mixture is made leaner. Further the heat release starts earlier with lean mixtures due the effect of the advanced spark timing. The variation of the calculated entropy at the same equivalence ratios is presented in *Figure 7*. The entropy rises during the combustion process and then starts to decline near the end of combustion. Thus it reaches a maximum value very close to the end of combustion. In all cases the entropy reaches a minimum during the compression stroke. This point certainly does not coincide with the beginning of the combustion process. An increase in the equivalence ratio increases the peak value of the entropy as expected.

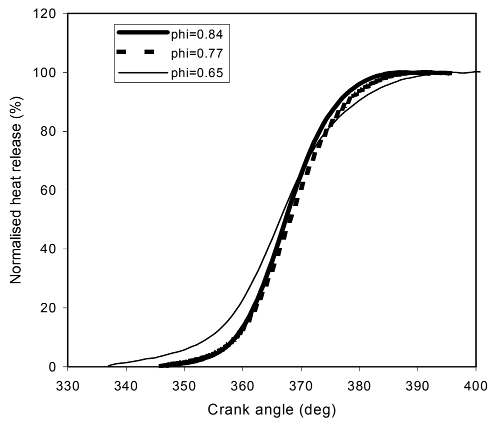

Figure 6. Variation of normalised heat release with crank angle

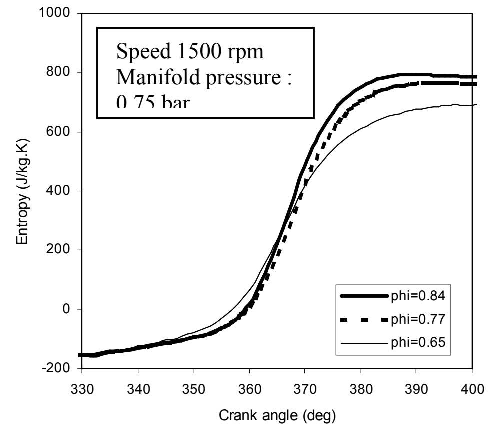

Figure 7. Variation of entropy with crank angle

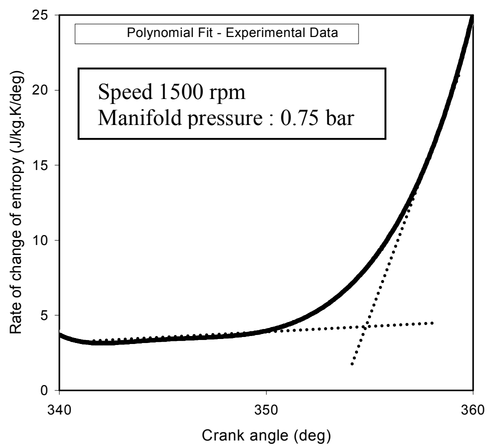

Figure 8. Variation of entropy rate with crank angle at an equivalence ratio of 0.84

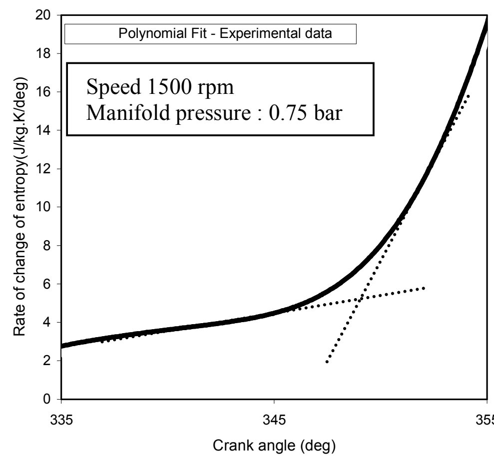

Figure 9. Variation of entropy rate with crank angle at an equivalence ratio of 0.7

An attempt was made to determine the relation between the rate of change of entropy and the beginning of combustion. This exercise was done at all equivalence ratios studied. However *Figures 8 and 9* indicate the variation at equivalence ratios of 0.84 and 0.7 only as samples. There was a rapid change in the entropy rate near the start of combustion in all cases. Similar trends were seen at all equivalence ratios and hence these have not been presented. These points were obtained from the tangents drawn on the curve in *Figures 8 and 9* as shown. Thus it was possible to detect the beginning of the combustion process from the rate of change in entropy. The end of combustion could be detected using the maximum of entropy.

Normally the start and end of combustion are detected from the heat release curves. The start of combustion is taken as the point where 2% or 5% of the heat release has been completed. End of combustion is detected as the points where 95% or 98% of the heat release have been completed.

The crank angles at spark, 2% heat release and 5% heat release have been compared with the start of combustion detected from the rate of change in entropy in *Figure 10*. It is clear that the method of using entropy rate to detect start of combustion produces results, which are reasonable. The values predicted lie between 2% and 5% heat release. It was seen that the method of detecting start of combustion using entropy is not very reliable at very lean mixtures where the combustion rate is low. Results obtained from heat release are compared with the crank angle at maximum value of entropy in *Figure 11*. There is a drastic change in the end of combustion angle between equivalence ratios of 0.7 and 0.65 as there is a significant difference in the MBT spark timings between these operating points. Always the point of maximum entropy lies very close to the crank angle where the maximum heat release occurs. Thus the point of maximum entropy effectively detects the end of combustion. Combustion duration calculated based on crank angles at different heat release values and using the present approach has been presented in *Figure 12* for comparison. The results lie between those of 2% to maximum heat release and 5% to maximum heat release.

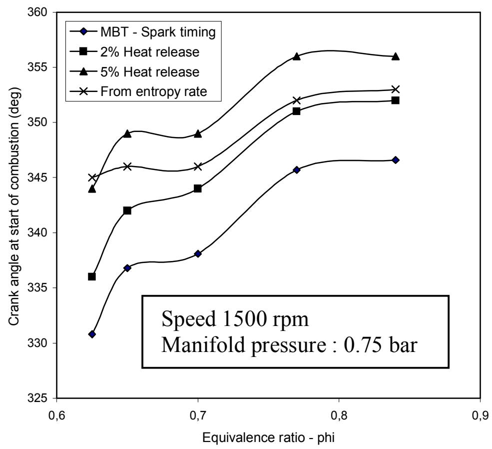

Figure 10. Variation of start of combustion with equivalence ratio

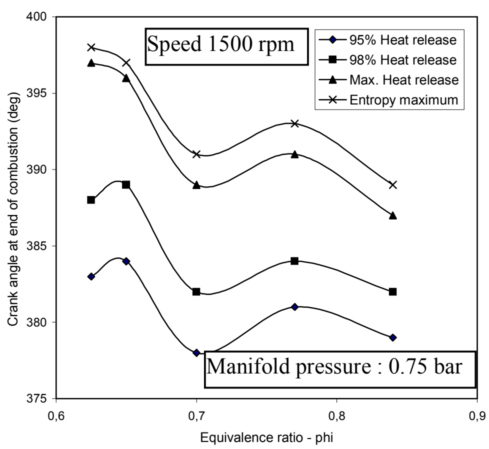

Figure 11. Variation of end of combustion with equivalence ratio

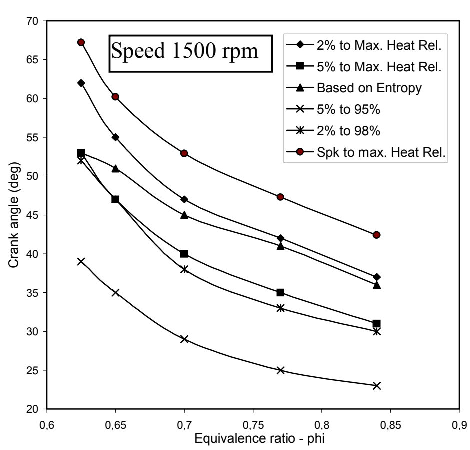

Figure 12. Variation of combustion duration with equivalence ratio

6. Conclusions

The following are the broad conclusions drawn based on this work:

- Results of the simple simulation program indicated that it is possible to detect the start and end of combustion by observing the entropy variation of the cylinder contents. Entropy reaches a minimum close to the start of combustion and a maximum near the end of the combustion process. Normal engine variables do not affect this trend significantly.
- Experimental results indicated that the point of maximum entropy of the cylinder gases closely matched the end of combustion predicted by the heat release curve.
- The start of combustion did not coincide with the minimum of entropy in the case of the experimental results
- It was possible to detect the beginning of the combustion process from experimental data from the rate of change of entropy with crank angle, which showed a rapid change at that point.
- Start of combustion predicted using the present method lies between 2% and 5% of total heat release. The end of combustion observed by this method is slightly later by about 1 to 2 degree crank angle than that predicted by maximum of total heat release.

Thus it is concluded that the method presented in this paper can be used to determine the start and end of combustion from experimental pressure crank angle data. However, its suitability has to be further investigated on different engines.

Nomenclature

c_p	Specific heat at constant pressure $Jkg^{-1}K^{-1}$
C_1, C_2	Constants in the heat release equation
h	Heat transfer coefficient $Wm^{-2}K^{-1}$
IVC	Inlet valve Closure
MBT	Minimum advance for Best Torque
n	Polytropic index
P,p	Pressure in bar or Pa as indicated
P_0	Reference pressure Pa
Q	Heat released J
R	Gas constant $Jkg^{-1}K^{-1}$
s	entropy $Jkg^{-1}K^{-1}$
SAE	Society of Automotive Engineers
T	Temperatrue K
TDC	Top Dead Center
T_0	Reference temperature K
V	Volume m^3
VC	Cylinder volume m^3
v_p	Mean piston speed m/s
x	Mass fraction burned

θ Any crank angle deg

θ_s Crank angle at start of combustion deg

θ_b Crank angle at end of combustion deg

γ Ratio of specific heats

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