MASS FRACTION BURNED ANALYSIS

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Abstract
Common algorithms for mass fraction burned computations are presented and compared. Some potential problems of these calculations are also discussed. The comparison shows that the evaluation of polytropic indices, necessary for current procedure may lead to some ambiguity in MFB determination.

1. Introduction

An internal combustion engine gains its energy from the heat released during the combustion of the oxidizer-fuel mixture. Engine processes leading to the conversion of the chemical energy contained in the fuel are complex phenomena that occur in transient thermodynamic conditions and the reliable evaluation of these processes is the key to engine optimization and effective control - combustion processes inside the engine cylinder “dictate” engine power, efficiency and emissions.
The optimization methods may be based on cylinder pressure feedback control. In-cylinder pressure changes are the crucial parameters - combustion descriptors - affecting performance, thermal efficiency and emission of spark-ignition engine. Standard engine control systems rely on calibration tables. Their values are taken from an analysis of an engine under fixed test conditions.

The extensive review of advanced engine controls for modern SI engine prepared by Delphi & GM is given in [11]. According to the authors, cylinder-pressure-based feedback control has potential to optimize spark timing, A/F ratio and EGR for the best engine operation. It may also offer improved detection of knocking combustion and cylinder misfire.

The overall long-term goal of contemporary scientific studies in the field of internal combustion engines is to develop a neural-network-based control system - artificial neural network which derives its name from the human brain with its capability of learning. Such a system should be better than present day controllers, both in terms of fuel efficiency and exhaust emission levels. The successful development of artificial neural network algorithms for nonlinear signal processing and control of SI engines is led by Isermann’s group of Darmstadt University of Technology. The energy conversion during a combustion cycle can be described by the Mass Fraction Burned (MFB) at a specific crank angle degree (CAD) and the MFB, in turn, can be used by feedback control systems. In IC engine, the MFB depends on engine geometry, engine speed, A/F, ignition angle, residual mass etc. Measuring and controlling the result can compensate many variables that affect the combustion process compensated on an individual cylinder basis.

2. Mass fraction burned

Mass fraction burned (MFB) in each individual engine cycle is a normalized quantity with a scale of 0 to 1, describing the process of chemical energy release as a function of crank angle. The determination of MBF is commonly based on burn rate analysis – a procedure developed by Rassweiler and Withrow (publicated in 1938). It is still widely used though its evident approximating character [3] because of its relative simplicity and computational efficiency.

The Rassweiler and Withrow procedure is based on the assumption that, during engine combustion, the pressure rise Δp (at crank angle increment $\Delta\varphi$; see Fig.1) consists of two parts: pressure rise due combustion (Δp_c) and pressure change due to volume change (Δp_v):

$$\Delta p = \Delta p_c + \Delta p_v$$

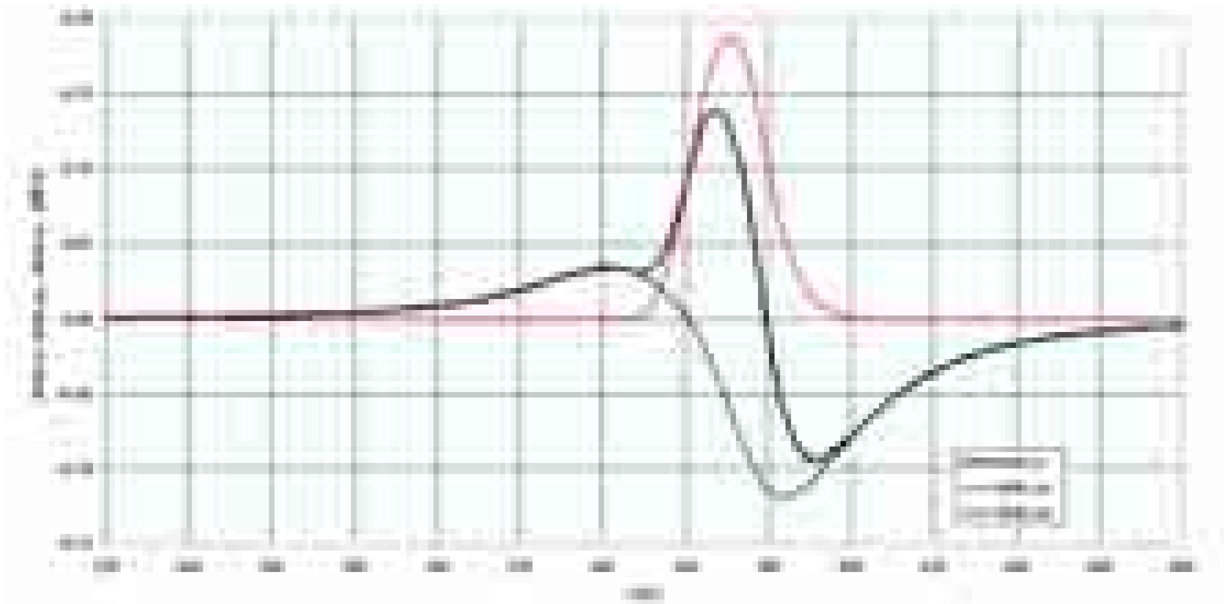

Fig. 1. *Components of cylinder pressure rise*

Assuming that the pressure rise Δp_c is proportional to the heat added to the in-cylinder medium during the crank angle interval, the mass fraction burned at the end of the considered i-th interval may be calculated as [3]:

$$MFB = \frac{m_b(i)}{m_b(total)} = \frac{\sum_0^i \Delta p_c}{\sum_0^N \Delta p_c}$$

where 0 denotes the start of combustion, N – end of combustion (N is the total number of crank intervals).

For the MFB calculations knowledge of Δp_c is necessary. It is accepted that the cylinder pressure and volume in the absence of combustion are related by the polytropic equation:

$$pV^n = idem$$

and a pressure change due to volume variation is given by:

$$\Delta p_v = p_{i+1} - p_i = p_i\left[\left(\frac{V_{i+1}}{V_i}\right)^n - 1\right]$$

The MFB profile calculated according to Rassweiler and Withrow procedure is presented in Fig.2 (generated using [8]).

Methods based on fitting the recorded pressure trace to a polytropic process have the benefit that no additional data (beside cylinder pressure and CAD) need to be recorded. The drawback of these methods is the difficulty to select the proper crank angle interval to fit. It should be remembered that polytropic equation is some kind of experimental relation describing the actual behavior of a gas and the constant polytropic index is an approximation that is valid only for a certain limited range for which the data must be correlated. That is why the precise determination of this region is crucial for MBF analysis. It includes determination of the start and end of combustion.

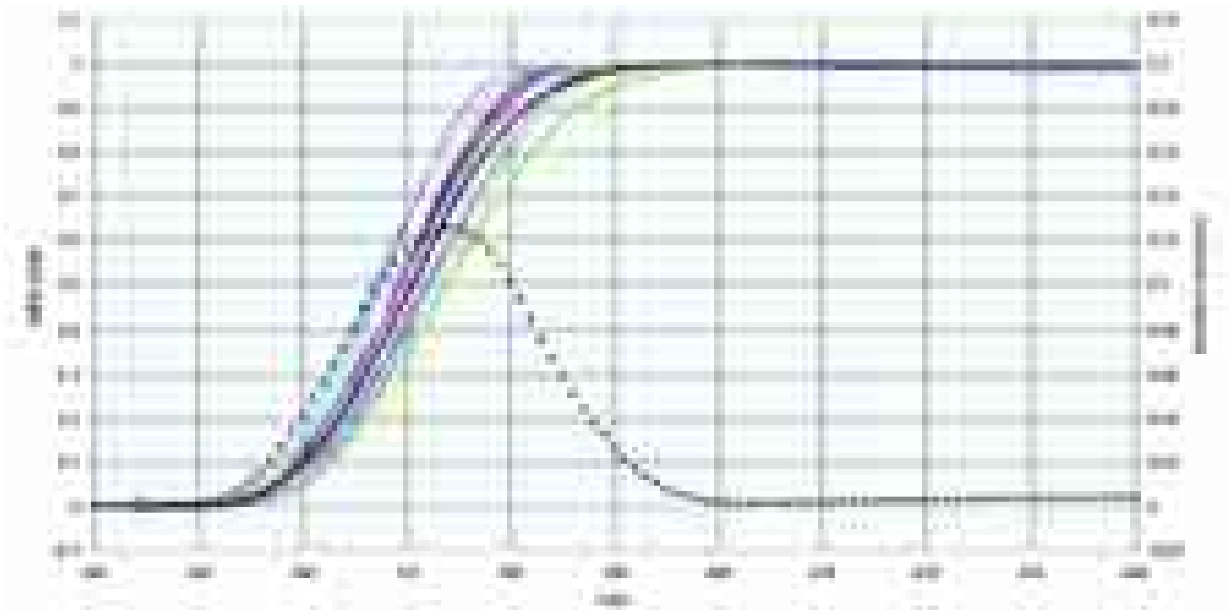

Fig. 2. *Mass Fraction Burned - Rassweiler and Withrow procedure*

3. Start and end of combustion

The fitting of polytropic index for compression should be carried out just before the start of combustion and the expansion index - close to the end of combustion but after this event. Because of stochastic variation (see Fig.3) of the in-cylinder thermodynamic parameters it is practical to perform the fitting of polytropic indices (for compression and expansion) in some crank angle interval (window) of 5 ÷ 50 CAD (the influence of window length will be considered).

3.1. Start of combustion

The knowledge of crank angle position corresponding to the start of combustion required for burned mass analysis of SI engine may by gained in relatively reliable way from the recorded data of high tension coil current (Fig.4). The spark inflames a small region of gas in the vicinity of the electrode gap and flame kernel will then develop into a propagating flame. The first phase of the discharge (called breakdown, preceding arc and glow phase; arc and glow phase are much longer: μs- and ms-regime, respectively) is characterized by a very high–energy conversion efficiency. This conversion takes place in a very short time and the chemical reactions can be observed a few nanoseconds (i.e. fraction of crank angle degree) after spark onset [3] and an ignition takes place at the boundary of the plasma where heat and radicals get into contact with the fresh mixture.

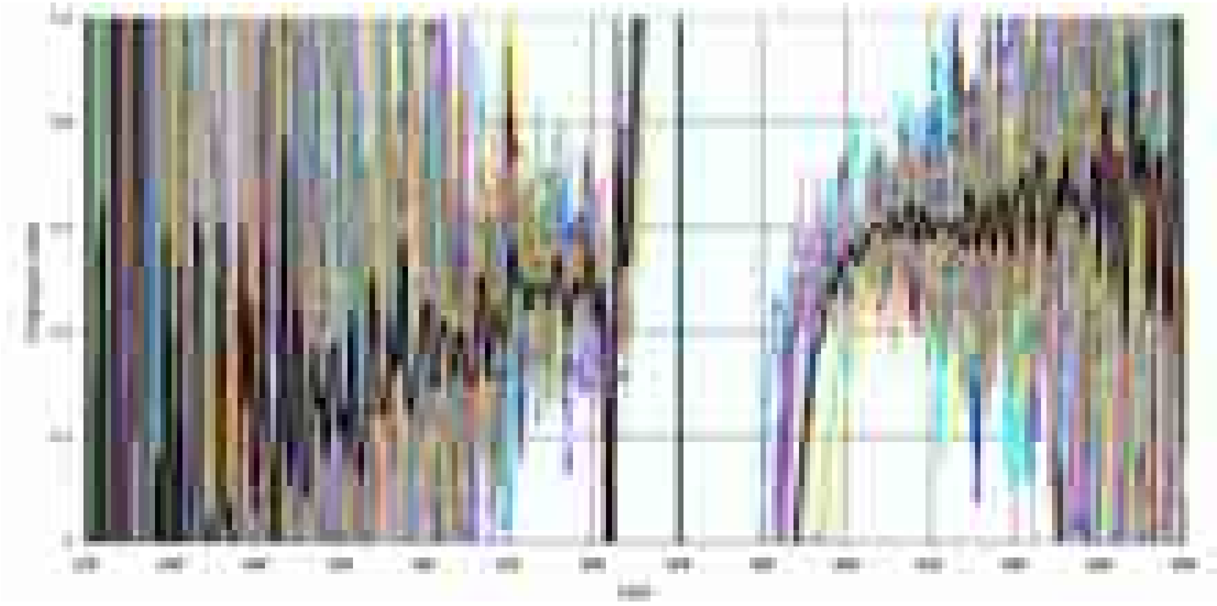

Fig. 3*. Polytropic index variations*

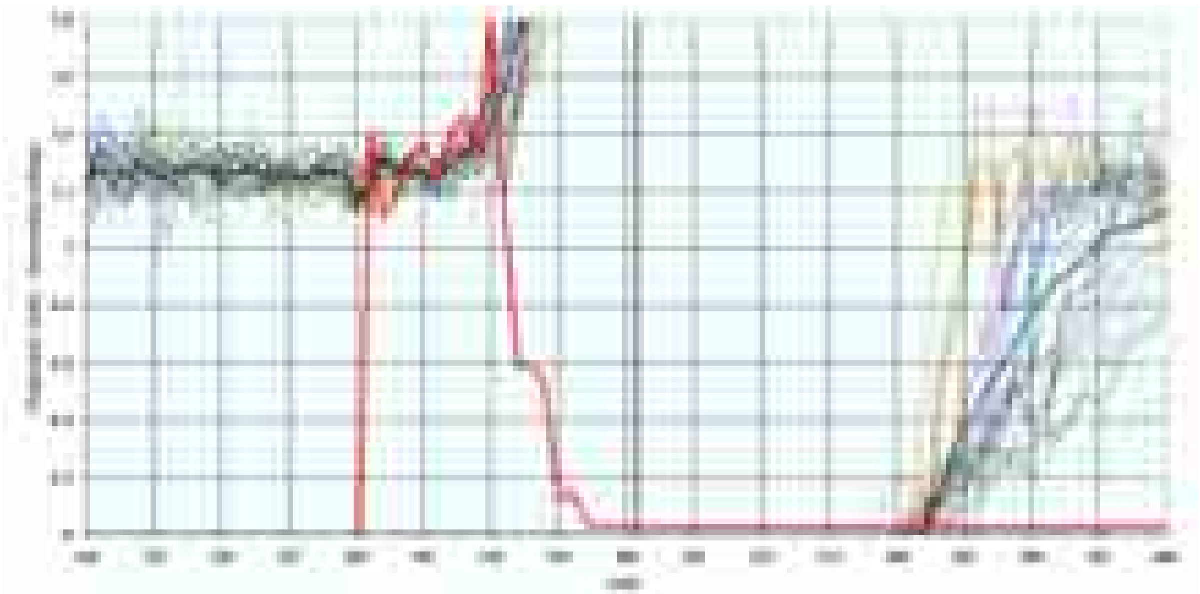

Fig. 4. *The spark signal (not to scale) shown at the background of polytropic index*

3.2. End of combustion

The knowledge of crank position corresponding to the end of combustion process is necessary both for determination of expansion index and the total number N of summation intervals. Precise determination of this event is difficult because of asymptotic character of the

final stage of combustion itself. There is no one method for the estimation of the end of combustion (a method suggested in [1] as the most common is based in the determination of the crank that provides a maximum value of the expression $pV^{1,15}$).
The method applied in this study is based on the First Law of Thermodynamics for the closed system:

$$\frac{dQ_{ch}}{d\varphi} = \frac{dU}{d\varphi} + \frac{dL}{d\varphi} + \frac{dQ_{nt}}{d\varphi}$$

The event of the end of combustion corresponds to the minimal value of the above equation. Taking into consideration the positive values work and heat transfer derivatives, the end of combustion should correspond to the minimal value of sensible internal energy change:

$$\left(\frac{dU}{d\varphi}\right)_{\min} = \left(m \cdot c_v \cdot \frac{dT}{d\varphi}\right)_{\min}$$

Assuming invariability of cylinder medium mass and individual gas constant:

$$\frac{dT}{d\varphi} = \frac{1}{m \cdot R} \cdot d(pV)$$

we arrive at the equation indicating that EOC corresponds the minimal value of the derivative of the pV product:

$$\left(\frac{dU}{d\varphi}\right)_{\min} = \frac{c_v}{R} \cdot \frac{d(pV)_{\min}}{d\varphi}$$

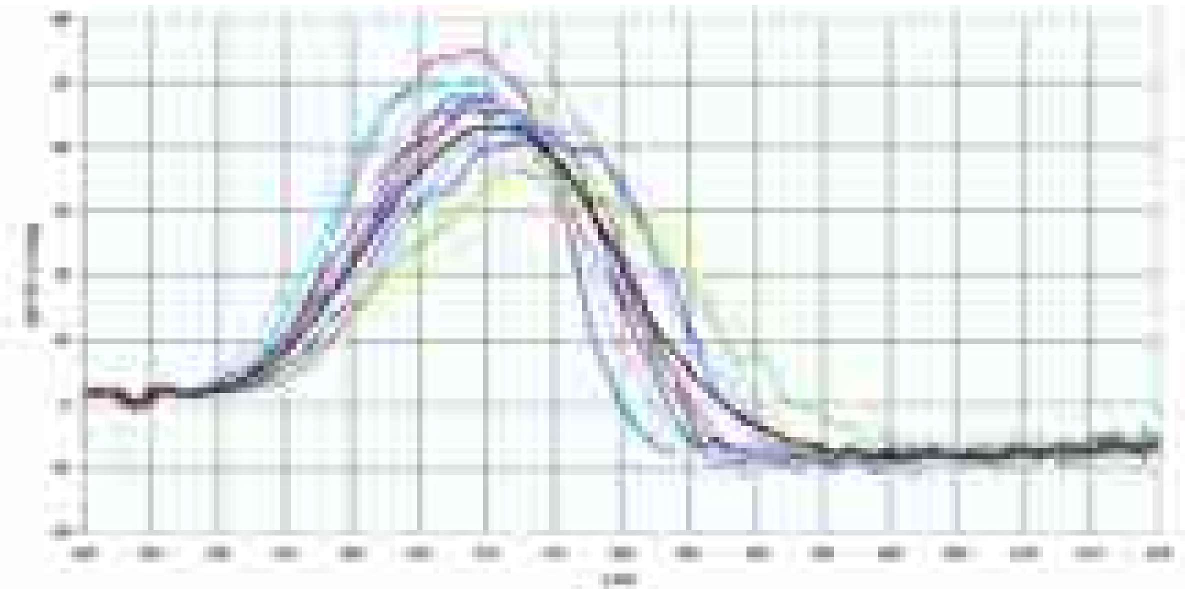

Fig. 5. *Determination of EOC*

4. Other MFB algorithms

4.1. Pressure Ratio Management

The algorithm proposed by Matekunas et al. [11] is similar to the principle used by Rassweiler and Withrow - it assumes that MFB can be determined from the fractional pressure rise due to combustion. The pressure ratio (PR) is defined as:

$$PR = \frac{p}{p_{mot}}$$

Where p is the fired cylinder pressure and p_{mot} is motored pressure (the pressure that would exist if combustion did not occur) calculated from precombustion pressure curve. The PR has unity value before combustion and rises to a final pressure ratio during combustion and the increase in the final pressure ratio is called the modified pressure ratio (MPR):

$$MFB = FPR - 1$$

The fractional rise in the PR (normalized with its maximum) may be used as an estimate of the MFB:

$$MFB \approx MPR = \frac{p}{p_{mot}} - 1 = \frac{p - p_m}{p_m}$$

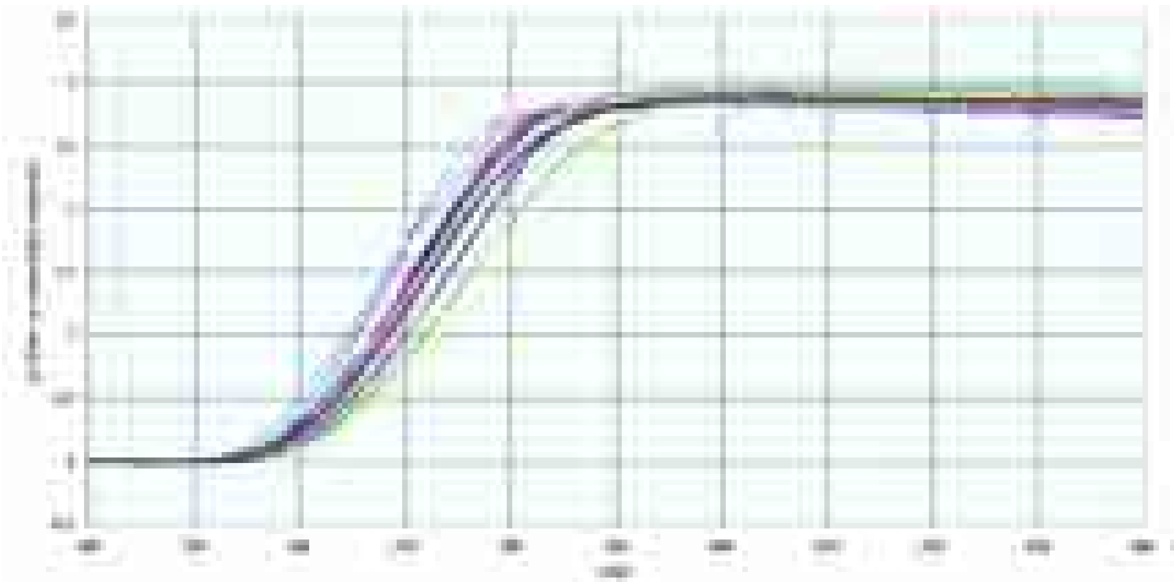

Fig. 6. *Modified Pressure Ratio profile*

Such an algorithm was also used by L. Eriksson [2] and in [7,9,10].

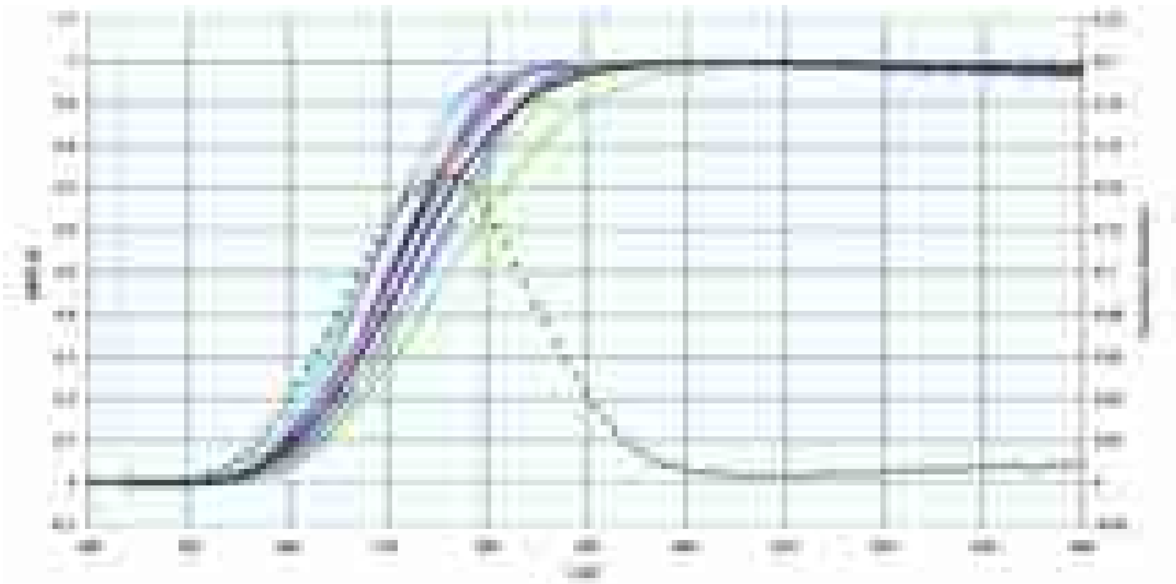

__Fig. 7.__ MFB profile based on MPR

4.2. Procedure applied by Isermann and Muller

Isermann and Müller use the approximation of the MFB by the equation (proposed by F.D. McCuiston, G.A. Lavoie and C.W. Kauffman [6]):

$$MFB \approx \frac{p \cdot V^{n} - p_{SOC} \cdot V_{SOC}^{n}}{p_{EOC} \cdot V_{EOC}^{n} - p_{SOC} \cdot V_{SOC}^{n}}$$

where p, V – cylinder pressure and volume at specified crank angle and n – polytropic index. The indices EOC and SOC denote the crank angle locations at the end of combustion (EOC) and before ignition start of combustion (SOC). This method is based on the observation that the heat added to the system (released between SOC and EOC):

$$Q = p_{EOC} \cdot V_{EOC}^{n} - p_{SOC} \cdot V_{SOC}^{n}$$

During combustion, the heat released is proportional to pV^n normalized to the value of the denominator of MFB expression.

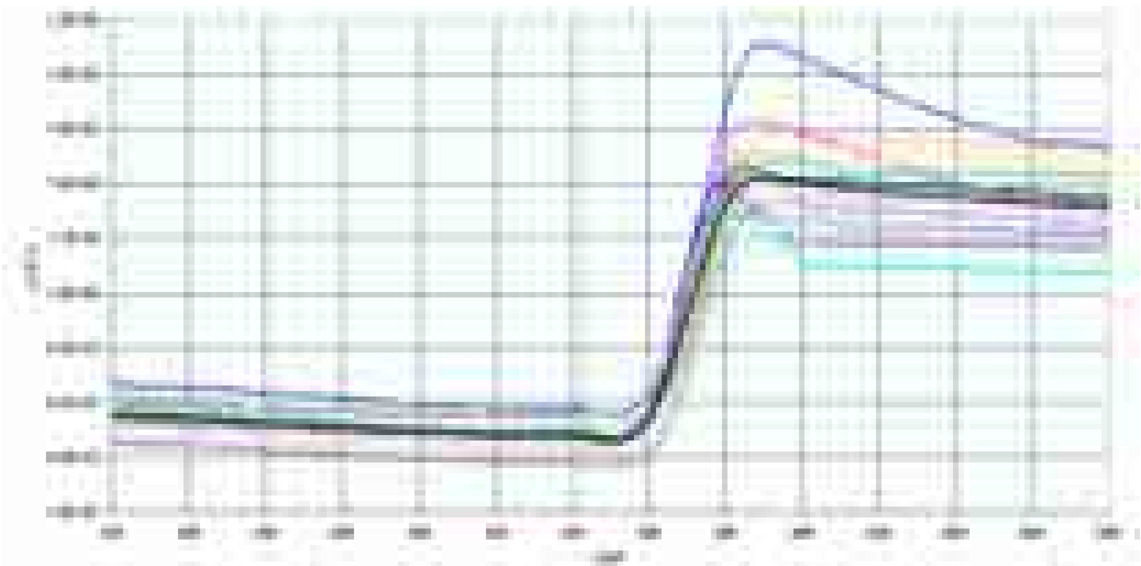

Fig. 8. *The pV^n value changes during heat addition*

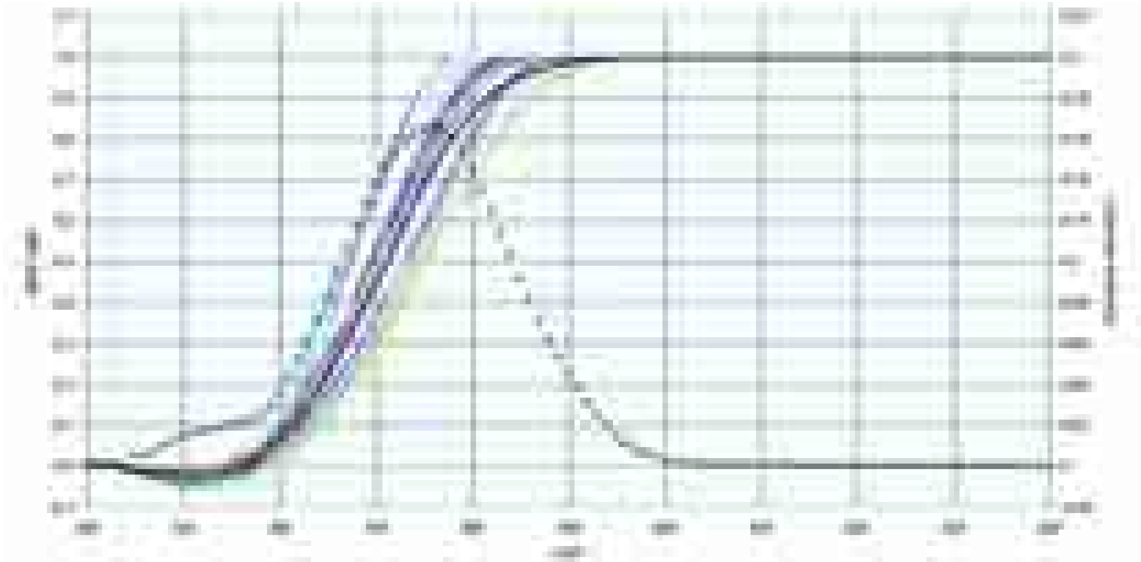

Fig. 9. *MFB approximation used by Isermann and Müller*

5. Comparison of MFB algorithms

Popular MFB algorithms presented above were compared in Fig.10. The most interesting point of MFB profile is its half value – it is suggested that the optimum efficiency of engine combustion is achieved if 50 % energy conversion occurs at 8 CAD after TDC [5]. The figure shows that the differences of the crank angle location of MFB=0.5 are up to 3 degrees of crankshaft angle. This very crank angle location of 50% energy conversion should be calculated and controlled by modifying the point of ignition accordingly [4].

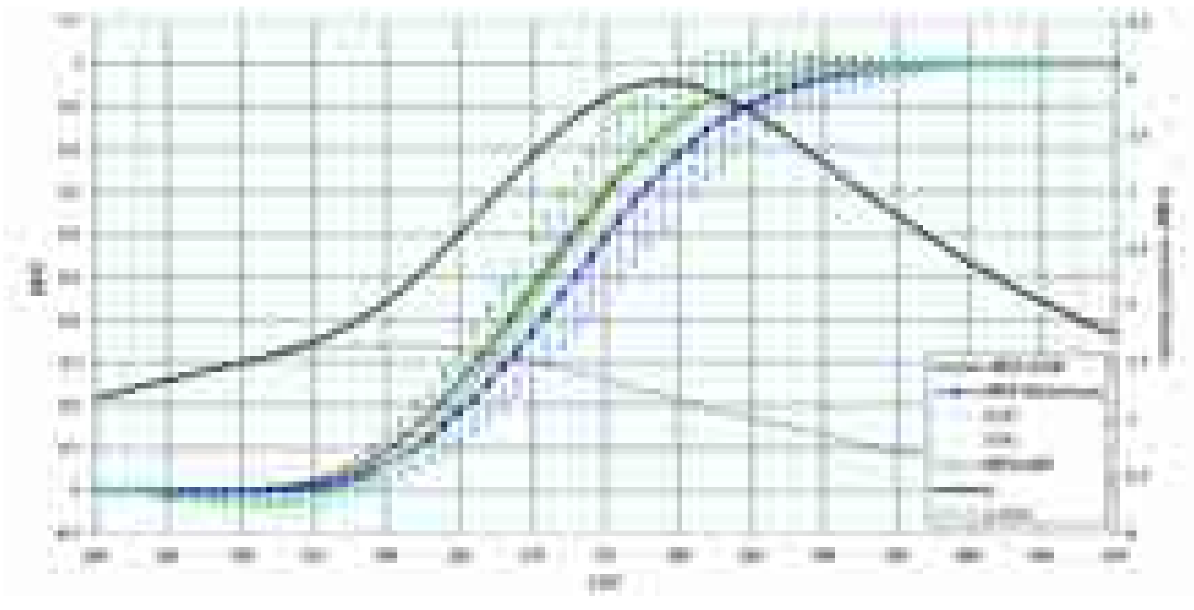

Fig. 10*. Comparison of MFB algorithms*

Shown differences and some ambiguity in reliable evaluation of suitable polytropic index (necessary for presented MFB algorithms) may make this goal more difficult to reach.

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