THERMODYNAMIC ANALYSIS OF SPARK IGNITION ENGINE PRESSURE DATA

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Abstract

This paper presents heat release model, based on in-cylinder pressure data from spark ignition engine, which uses instantaneous properties of working fluid to calculate cumulative heat release characteristics used to quantify combustion development.

Keywords: internal combustion engine, thermodynamic properties, heat release analysis.

1. Introduction

The cylinder pressure can give valuable information about the combustion process and the analysis of cylinder pressure data over the closed part of engine cycle is a classical tool for engine studies. It is expected that it will also be interesting to use pressure sensor in production engine. The present work develops and applies heat release analysis for SI engine pressure data. Mass fraction burned (MFB) is used to interpolate the in-cylinder gas properties for the burned and unburned charge.

2. Equation of heat release rate

Cylinder pressure built by combustion process is also affected by the changes in the volume of combustion chamber (due to piston travel) and chemical composition (and hence thermodynamic properties), heat transfer and gas flows. To examine (and further to develop combustion process), it is necessary to relate each term to measured in-cylinder pressure changes and then combine them to quantify the combustion effects.

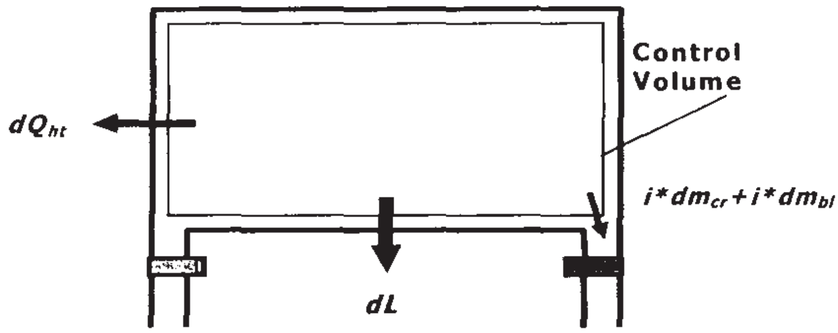

Fig. 1. Control volume in engine combustion chamber

If the First Law of Thermodynamic is applied to a control volume (Fig. 1), the heat released by combustion of fuel is given by the equation:

$$dQ_{ch} = dU + dQ_{ht} + dL + i'dm_{cr} + idm_{bl} \tag{1}$$

Where: dQ_{ch} is the change in chemical energy released from the fuel;

dU denotes the change in internal energy of the mass in control volume (CV);

dQ_{ht} represents heat losses to the cylinder wall across the CV boundary;

dL is elementary work.

The last two terms represent the changes in the enthalpy due to flows of the mass out and into the CV. We assume that (as in [6]):

$dm_{cr}>0$ when the fluid flows out the CV into the crevices (the gas is trapped in crevices until maximum pressure) and $dm_{cr}<0$ when the gas returns to the CV;

dm_{bl} represents the non-return leak flow (blow by) from CV to engine crankcase;

i' is evaluated at cylinder conditions ($dm_{cv}>0$) or at crevice conditions ($dm_{cv}>0$).

The change of the mass in CV is given by: $dm = -dm_{cr} - dm_{bl}$

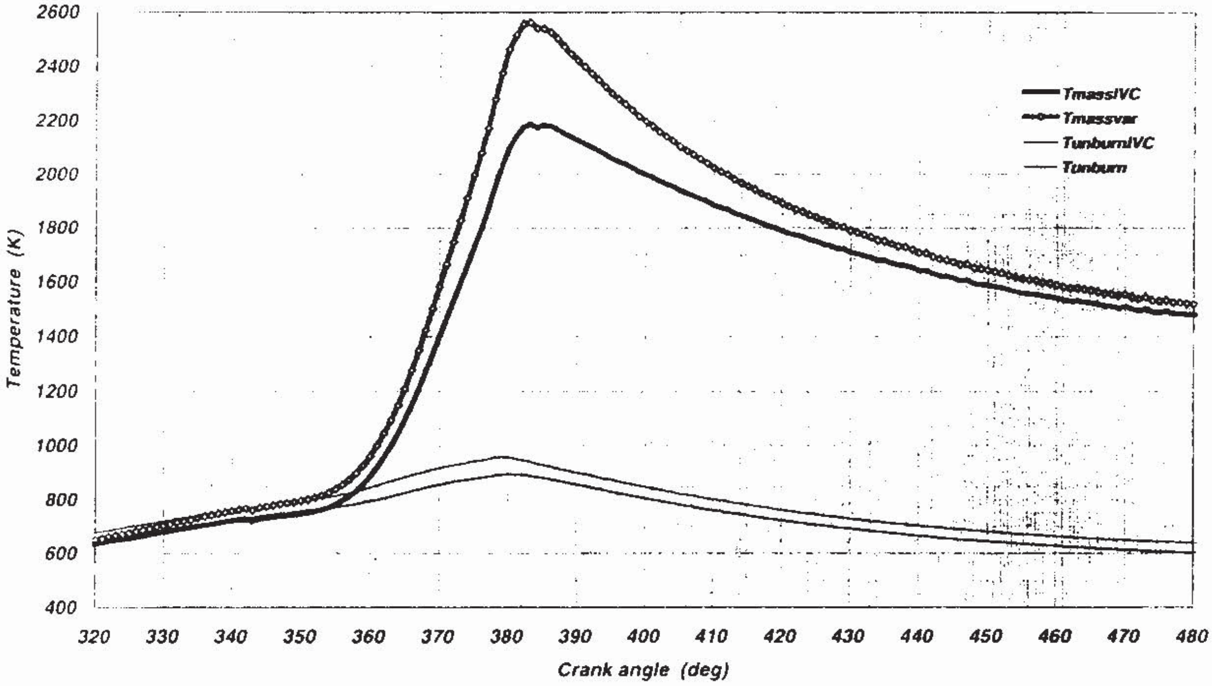

Fig. 2. Influence of mass changes on mean temperature of cylinder charge

With the ideal gas model, acceptable for SI engine [10], the change in internal energy can be expressed as: $$dU = udm + mdu = udm + mc_v dT \qquad (2)$$

and the Eq.(1) rewritten:

$$\begin{aligned} dQ_{ch} &= udm + mc_v dT + dQ_{ht} + dL + i'dm_{cr} + idm_{bl} = \\ &= mc_v dT + pdv + dQ_{ht} + (i' - u)dm_{cr} + (i - u)dm_{bl} \end{aligned} \qquad (3)$$

The mean temperature of the working fluid and its change are determined from the ideal gas equation of state: $T = \dfrac{pV}{mR}$ (4) $\qquad dT = \dfrac{1}{mR}(Vdp + pdV - \dfrac{pV}{m}dm - \dfrac{pV}{R}dR)$ (5)

Finally, we arrive at heat release rate equation:

$$\begin{aligned} dQ_{ch} &= \frac{C_v}{R}Vdp + (\frac{C_v}{R} + 1)pdV + dQ_{ht} - \frac{C_v}{R}mTdR + \\ &(C_v T - i' - u)dm_{cr} + (cvT + i - u)dm_{bl} \end{aligned} \qquad (6)$$

Applying the concept of net heat release rate dQ_{net} [6,7], which represents the work and internal energy change, the chemical energy or gross heat release rate can be expressed as:

$$dQ_{ch} = dQ_{net} + dQ_{ht} + dQ_{cr} + dQ_{bl} \qquad (7)$$

Each term of the above equation requires the knowledge of instantaneous thermodynamic properties of the working fluid. MFB [8,9] is used to interpolate the in-cylinder gas properties for the burned and unburned charge (Fig. 3).

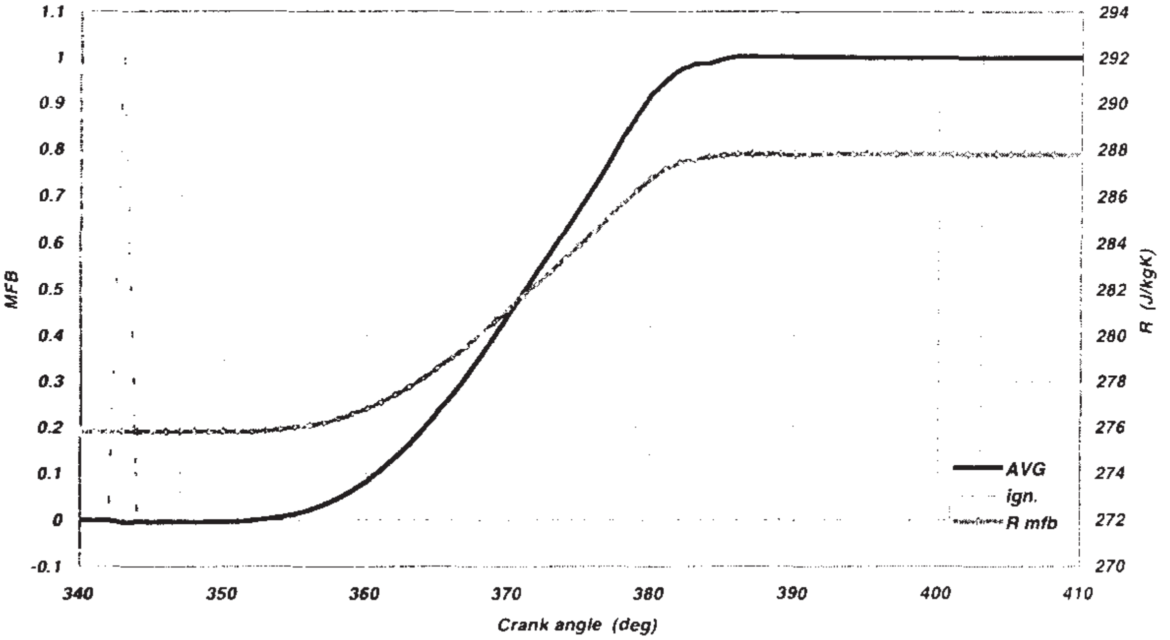

Fig. 3. Mass fraction burned and gas constant of cylinder charge

3. Specific heats model

The specific heats of working fluid are represented by specific heat ratio which is frequently modelled as a linear function of temperature. In the well known single-zone heat release model developed by Gatowski et al. [6], the c_p/c_v ratio is represented by a linear function of mean charge temperature: $\kappa(T) = a + bT$. Such a function (applied also e.g.in [1,4]) underestimates the value of the c_p/c_v ratio (Fig. 3) and that is why Heywood and Cheung [3] proposed to approximate this ratio by a constant value during the combustion period.

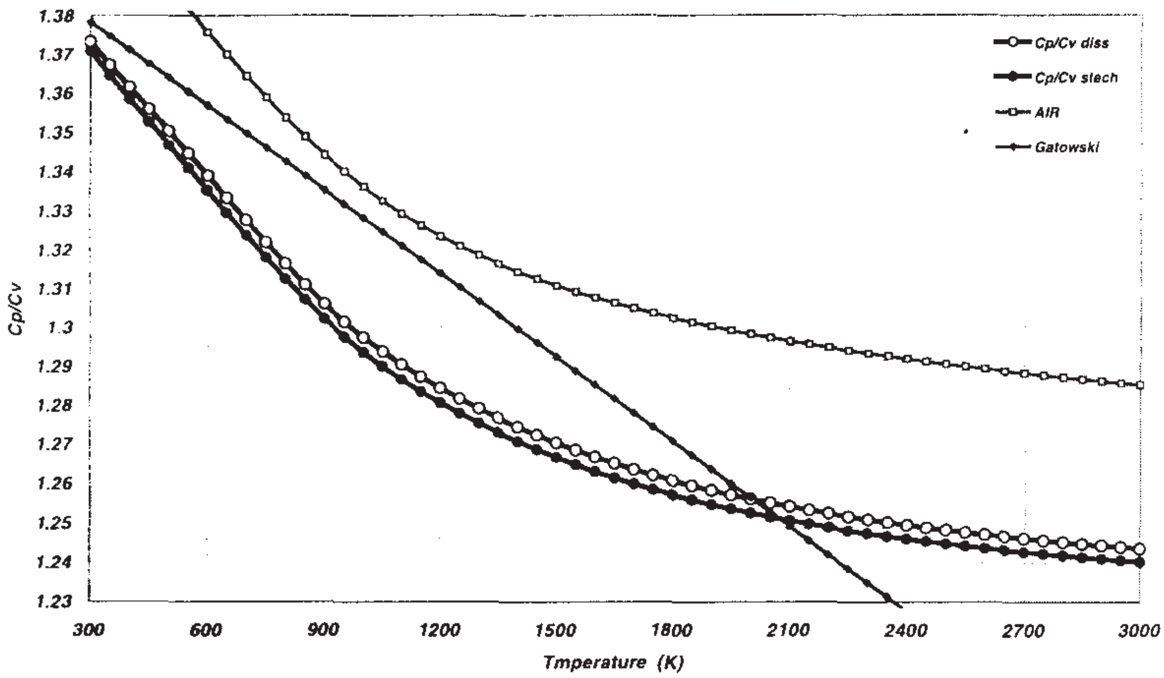

Fig. 4. Specific heat ratios calculated with linear and polynomial models

In the present study, the polynomial model [10] was applied and the MFB was used to interpolate the specific heats for the unburned and burned charge (Fig. 5).

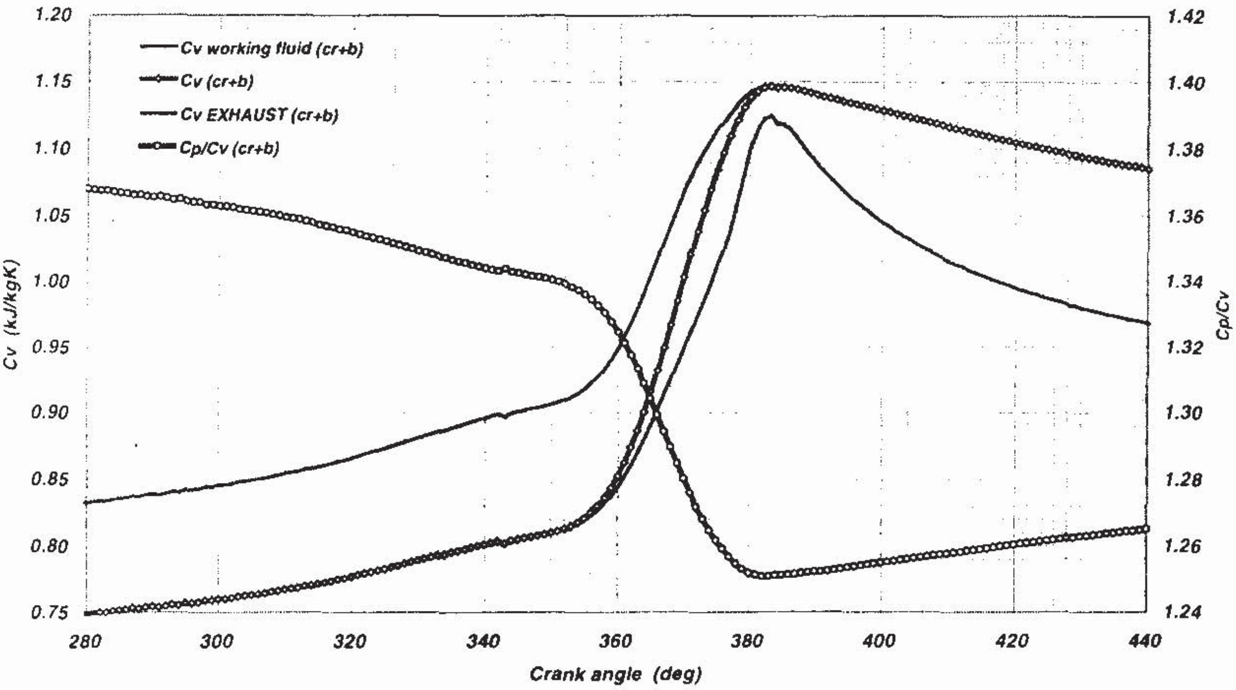

Fig. 5. Specific heat ratio corrected with MFB

4. Crevice and blow by model

There are several crevice volumes in the engine combustion chamber (e.g. top land above the top ring between the cylinder and piston, around the spark plug etc.). Crevice volumes are difficult to estimate with precision because they can vary with engine temperature. The crevice volume used in present analysis is the sum of all relevant crevices in the combustion chamber. It is assumed that the crevice walls are cold and the temperature of the thin layer of the gas in crevices is close to the wall temperature. The mass in the crevices is given by Gatowski et al. [6]:

$$m_{cr} = \frac{pV_{cr}}{RTw}$$

and

$$dm_{cr} = \left(\frac{V_{cr}}{RTw}\right)dp$$

where T_w – constant temperature of the wall;

V_{cr}– resultant volume of the crevices (ca. 2-5% of cylinder clearance volume).

Some gas escapes (blows by) through the piston rings to the space behind (to the crankcase). The assumption of constant specific heats (for that mass loss only) allows the derivation of relatively simple equation - the change of charge mass due to blows by can be expressed using the choked flow equation [5]:

$$\frac{dm_{bl}}{dt} = A\sqrt{\kappa\left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}}\sqrt{\frac{p}{v}}$$

where: A denotes the overall cross sectional area of leak flow;

p and v are cylinder pressure and specific volume.

The throat area A is proportional to ring gap and piston clearance. Its value is not known with certainty. In the present work, its magnitude was matched by simultaneously integrating the flow equation (it was assumed that maximum leak flow can not exceed 2 g/s). Mass changes in the considered control volume are illustrated in Fig. 6.

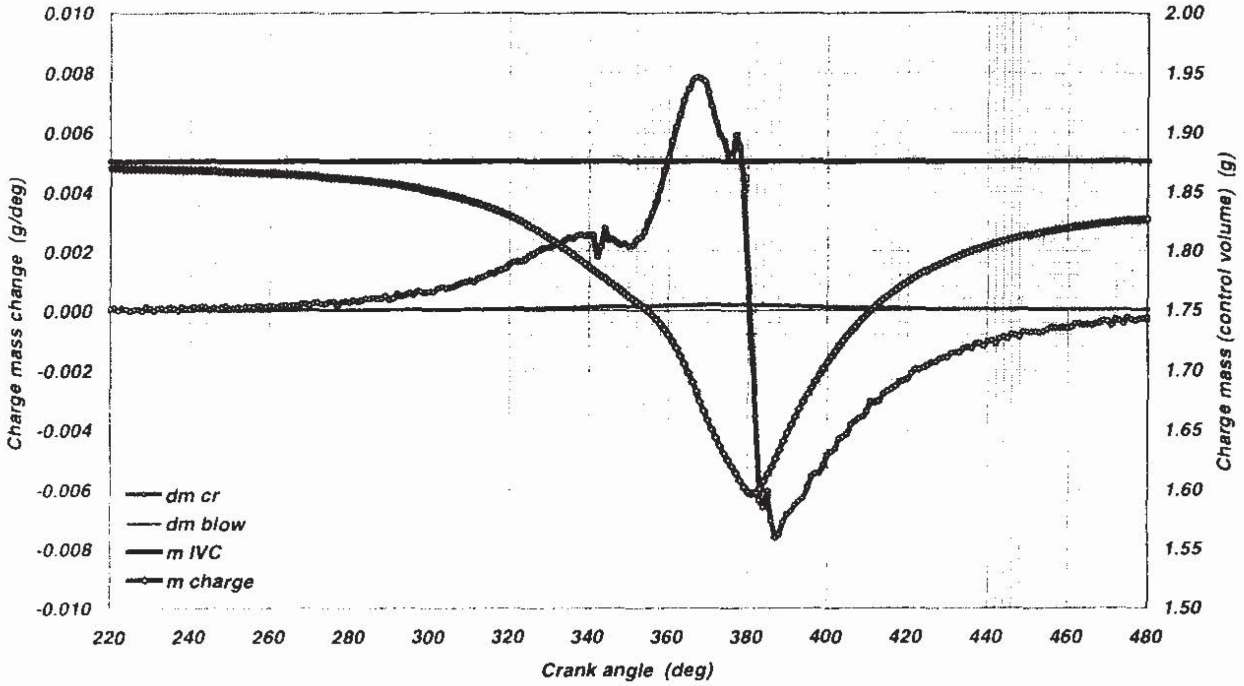

Fig. 6. Crevice and leak flows

5. Heat transfer model

The heat transfer from the in-cylinder gas occurs by convection and radiation but is usually assumed that the radiation component is "included" in the convection heat transfer correlation derived from experiments (the greatest uncertainty is caused by assuming the wrong rate of heat transfer [2]).

In the present analysis, the heat transfer coefficient (HTC) is calculated using the dimensionless correlation of the from:

$$Nu = 0.035\,\mathrm{Re}^{0.8}$$

assumed by Woschni [12],

where: $$Ne = \frac{h \cdot D}{k}$$ - Nusselt number

$$\mathrm{Re} = \frac{\rho w D}{\mu}$$ - Reynolds number

h - heat transfer coefficient

k - gas thermal conductivity

D – characteristic dimension (bore diameter)

μ - gas dynamic viscosity

ρ - gas density

w – characteristic velocity

$$w = C_1 C_m + C_2 \frac{V_s T_r}{p_r V_r}(p - pm)$$

with V_s – swept volume

V_r, T_r, p_r – parameters evaluated at reference conditions (inlet value closure)

p_m – motored pressure.

Woschni applied transport properties of air taken as a in-cylinder working fluid. In the present study k and μ (Fig.7) are evaluated for a given chemical composition and thermodynamic parameters of cylinder charge [11].
The comparison of heat transfer coefficient calculated according to presented procedure and that of Woschni equation for HTC is given in Fig. 8.

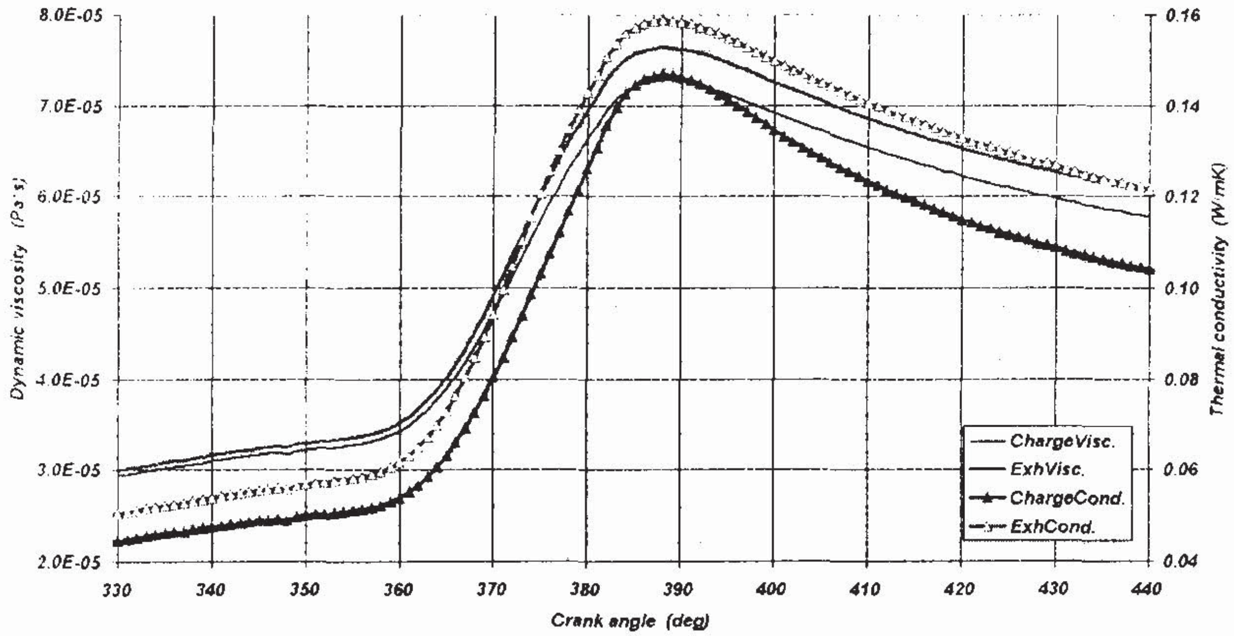

Fig. 7. Transport properties of in-cylinder gas

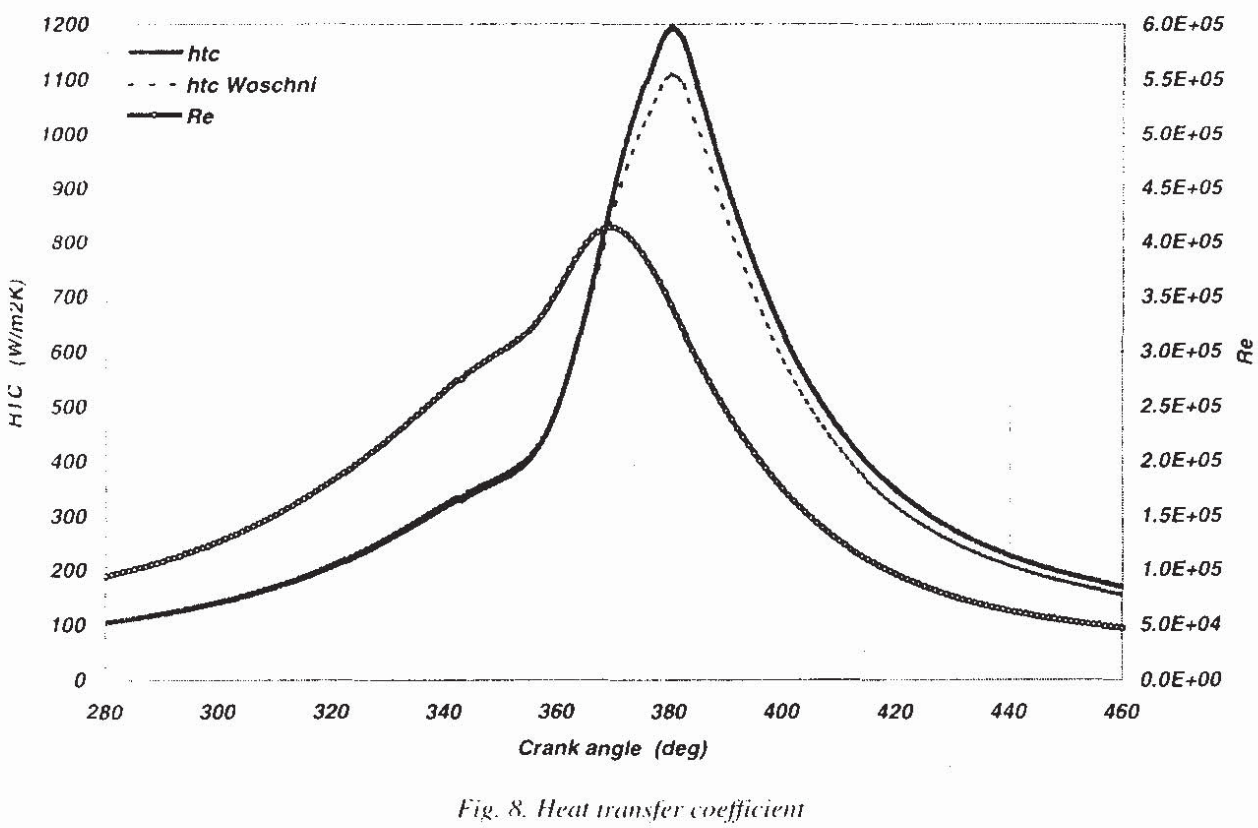

Fig. 8. Heat transfer coefficient

6. Gross heat release rate and cumulative heat release

An example of the use of Eq.7 to analyze a pressure data (Fig.9a) versus crank angle for a spark ignition engine is shown in Fig.9b. The figure presents gross heat release rate components (Eq.7) and cumulative (integral) heat release. The curve at the top of the figure (denoted as ***Ed*MFB***) is the mass of the fuel delivered times its lower heating value (***Ed***) times MFB. The difference between the final value of the gross heat release Q_{ch} (being the measure of both the energy added to the engine cycle and its utilization) and ***Ed*** represents the combustion inefficiency (few percent of ***Ed***) and the error of applied model of gross heat release rate.

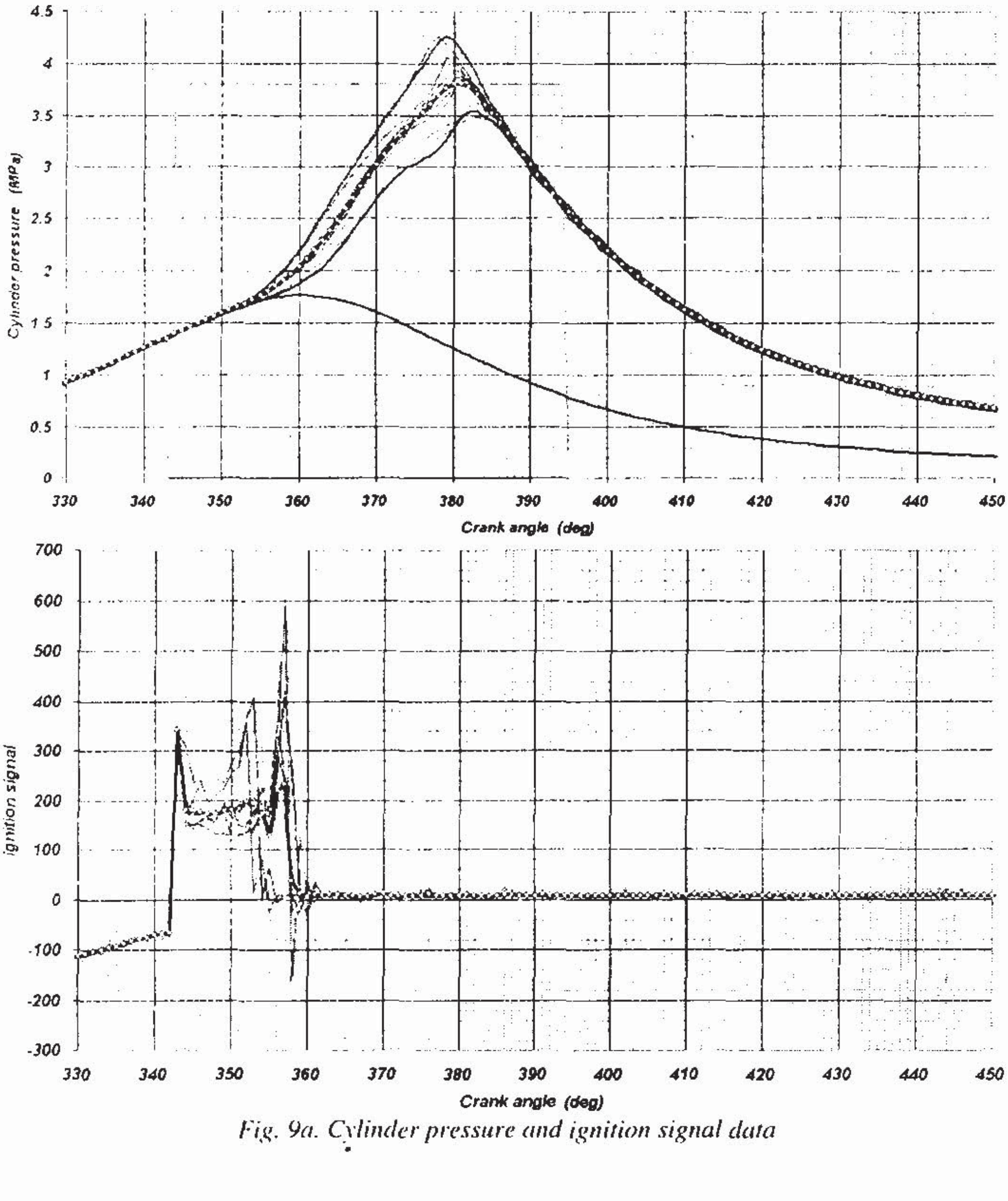

Fig. 9a. Cylinder pressure and ignition signal data

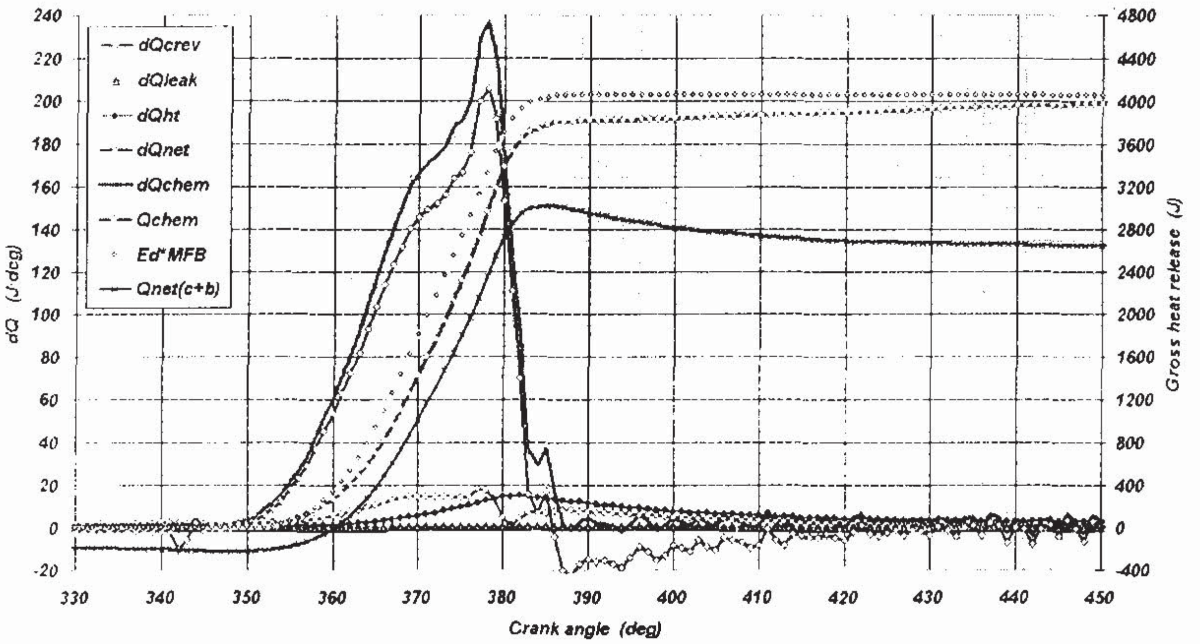

Fig.9b. Gross heat release rate and chemical energy release

There are several possible causes for such discrepancies. The most plausible (besides the inaccuracies in pressure data) are deficiencies in the heat transfer model (the in-cylinder heat transfer varies with position and crank angle; dimensionless correlation neglects spatial variations) and approximations in the thermodynamic model of in-cylinder processes (incomplete energy balance, ideal gas assumption etc).

References

[1] *M.F.J. Brunt, A. Harjit and A.L. Emtage: The calculation of heat release energy from engine cylinder pressure data. 1998. SAE Technical Paper 981052.*

[2] *M.F.J. Brunt, K.C. Platss:. Calculation of heat release in direct injection diesel engine. 1999. SAE Technical Paper 199-01-0187.*

[3] *H. M. Cheung and J. B. Heywood:. Evaluation of a one-zone burn-rate analysis procedure using production SI engine pressure data. 1993. SAE Technical Paper 932749*

[4] *L. Eriksson: Spark Advance Modeling and Control. Printed by Linus & Linnea AB, Linkoping, Sweden 1999, ISBN 91-7219-479-0 ISSN 0345-7524*

[5] *C.R. Ferguson, A.T. Kirkpatrick: Internal Combustion Engines Applied Thermosciences. John Wiley & Sons, 2001*

[6] *J. A. Gatowski, E. N. Balles, K. M. Chun, F. E. Nelson, J. A. Ekchian, and J. B. Heywood. Heat release analysis of engine pressure data. SAE Technical Paper 841359, 1984.*

[7] *J. B. Heywood. Internal Combustion Engine Fundamentals. McGraw-Hill series in mechanical engineering. McGraw-Hill, 1988.*

[8] *K. Z. Mendera, A. SPYRA, M., SMEREKA: Mass fraction burned algorithm based on the pV product", Journal of KONES Internal Combustion Engines Vol.9 nr 3-4, 2002.*

[9] *K. Z. Mendera, A. SPYRA, M., SMEREKA: Mass Fraction Burned Analysis", Journal of KONES Internal Combustion Engines Vol.9 nr 3-4, 2002.*

[10] *K. Z. Mendera: Thermodynamic properties of internal combustion engine working fluid. Paper proposed for Kones 2004.*

[11] *K. Z. Mendera: Termodynamika silnika spalinowego. (W przygotowaniu)*

[12] *G. Woschni: A universally applicable equation for the instantaneous heat transfer coefficient in the internal combustion engine. SAE Technical Paper 670931, 1967.*