THE HEAT TRANSFER COEFFICIENT CALCULATION IN THE ICE CYLINDER BASED ON IN-CYLINDER PRESSURE DATA

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Abstract

In this paper the calculations algorithm of heat-transfer coefficient in the combustion chamber of the internal combustion engine is presented. Developed algorithm is based on the in cylinder pressure data. The proposed algorithm can be helpful to determine the average values of heat-transfer coefficient from working medium to the combustion chamber walls (crown of a cylinder head, cylinder walls and piston head) during combustion process. The calculation method includes modified one zone heat release model in combustion chamber of SI engine. Proposed method consists in closing the energy balance equation by the coefficient which expresses the heat losses to the walls of the combustion chamber. The average value of the heat losses during combustion process is calculated by two steps. Firstly, the integration of the energy balance equation (without specifying the heat losses) leads to designation of the so-called net value of heat released in cylinder. In the next step the amount of the total energy supplied to the cylinder is determined taking into account the chemical energy of the supplied fuel. The difference between the supplied value of chemical energy and heat released net value allows to determine the heat losses average value. In last stage, the heat flow equation leads to calculate the mean value of heat transfer coefficient during combustion process.

Keywords: *heat transfer coefficient, combustion engines, in-cylinder pressure*

1. Introduction

The reciprocating internal combustion engines belong to a group of thermal machines working cyclically. Every single working cycle includes many thermodynamic processes (some of them occur together with the combustion process) of working medium. The end of last thermodynamic process is covering with start of the first process. Physical processes occurring inside the internal combustion engine are very complex. During each working cycle the proper amount of the fuel dose (regarding to engine load) is combusted. The heat released in cylinder is converted to internal engine work. The time of supplying, mixing and burning the air-fuel mixture is of the order of few milliseconds (the higher engine speed the shorter cycle time). The short time of these events causes serious difficulties in a measurement method which will be bring accurate results of thermodynamic parameters against the crank angle.

The in-cylinder pressure can be accurately measured using very fast piezoelectric transducer together with a charge amplifier, a crank angle encoder and a data acquisition device. In-cylinder pressure analysis is nowadays an indispensable tool for investigating the internal processes of reciprocating combustion engine. The in-cylinder pressure as a function of crank angle can be used to calculate the value of heat transfer coefficient. The heat transfer from working medium to the engine internal surfaces plays an important role in engine performance and kind of material use for engine parts.

2. Heat release analysis

There are a two main method which describe the phenomena occurring in the cylinder of combustion engine. The first method applies to macroscale models. In this approach a few zones is investigated (eg. flame, unburned mixture, exhaust) with different degrees of fidelity of real phenomena. These type of mathematical models give a good quantitative results and them main advantage is short computation time. The weak point is the lack of information on the instantaneous values of the analyzed parameters in different places of engine combustion chamber. However, in many cases, the phenomenological macroscale models are sufficient in the engine parameters calculations. The second method based on multidimensional models which enable to determine instantaneous value of internal parameters at chosen places of combustion chamber.

In this paper the single-zone heat release model was used in the computations. It was assumed that at any time of the combustion process, the working medium contained in the cylinder is a homogeneous. Moreover, the thermodynamic state of working medium describe the averaged values of thermodynamic parameters specified for instantaneous value of cylinder volume. The initial condition of working medium is it's the state after compression. What mean, the homogenous air-fuel mixture which is ready to combust. The scheme of energy balance model is presented in Fig. 1.

The following assumptions has been made in order to prepare the energy balance:

– the working medium is considered as an ideal gas with a change of composition during combustion process,
– the specific heat is an function of average temperature in cylinder,
– the change of working medium composition is taken in to account by instantaneous value of fuel mass fraction burned (MFB),
– the heat losses in to the combustion chamber walls are closing the energy balance equation.

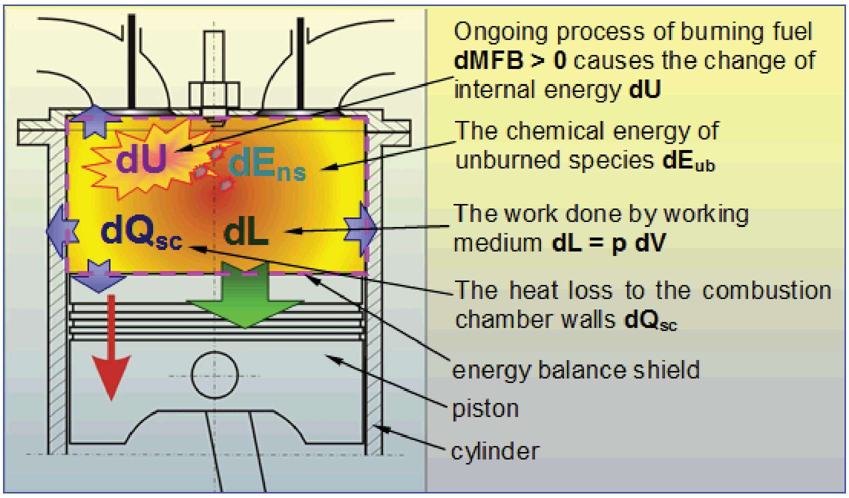

Fig. 1. The scheme of in-cylinder SI engine energy balance

The chemical energy of the fuel is a significant part of the energy balance equation. The energy is released gradually according to combustion process rate, what is expressed by MFB function. Besides, the balance equation includes the change of total internal energy and internal work. The heat losses are closing the balance. Finally, the energy balance equation (differential form) can be described using the first law of thermodynamic, as follow:

$$\frac{dU}{\alpha}+\frac{dL}{\alpha}+\frac{dQ_{sc}}{\alpha}+\frac{dE_{ns}}{\alpha}=0\,, \tag{1}$$

where:

dU – internal energy change [J],
dL – elementary internal work done by working medium [J],
dQ_{sc} – elementary heat losses transferred to combustion chamber walls [J],
dE_{ub} – chemical energy of unburned fuel [J].

The differential form of internal energy can be expressed as:

$$\frac{dU}{d\alpha}=-\frac{dE}{d\alpha}+\frac{1}{\kappa(T,MFB_v)-1}\left(p(\alpha)\frac{dV}{d\alpha}+V(\alpha)\frac{dp}{d\alpha}\right)\text{, where: }\frac{dE}{d\alpha}=m_{p,0}\,LHV_f\,\frac{dx}{d\alpha}. \quad (1a, 1b)$$

And finally the differential (without heat loss) the heat release rate equation (net value) is described as follow:

$$\frac{dE}{d\alpha}=\frac{1}{\left(1-\delta_{ub}\dfrac{LHV_{ub}}{LHV_f}\right)}\left[\frac{1}{\kappa(T,MFB_v)-1}\left(\kappa(T,MFB_v)\;p(\alpha)\frac{dV}{d\alpha}+V(\alpha)\frac{dp}{d\alpha}\right)\right], \quad (2)$$

where:
δ_{ub} – the mass fraction of unburned fuel [–],
LHV_{ub} – lower heating value of unburned species at the working medium [J/kg],
LHV_f – lower heating of the fuel [J/kg],
$\kappa(T, MFB_v)$ – the specific heat ratio as a function of in cylinder temperature and, mass fraction burned (MFBv) [–],
$p(\alpha)$ – instantaneous cylinder pressure [Pa],
$V(\alpha)$ – the engine volume as a function of the crank angle [m^3].

After integration the equation (2) at the range of combustion angle from ap (start of combustion) to a (current value of crank angle during combustion), the net amount of energy realised in the cylinder can be expressed by:

$$E_d(\alpha^*)=\frac{1}{\left(1-\delta_{ub}\dfrac{LHV_{ub}}{LHV_f}\right)}\int_{\alpha_p}^{\alpha^*}\left[\frac{1}{[\kappa(T,MFB_v)-1]}\left(\kappa(T,MFB_v)p(\alpha)\frac{dV}{d\alpha}+V(\alpha)\frac{dp}{d\alpha}\right)\right]d\alpha, \quad (3)$$

where:
α^* – current value of crank angle during combustion process [CA deg],
α_p – value of crank angle at start of combustion process [CA deg].

The upper limit of integration reach the values from the range of [$\alpha_p - \alpha_k$], where the α_k. is the value of crank angle at the end of combustion process. The total amount of net heat released in the cylinder will be reached when $\alpha^* \to \alpha_k$.

The actual heat input to the cylinder during a single working cycle is related to the amount of chemical energy contained at the fuel dose. Therefore, amount of chemical energy supplied to the cylinder can be noted as:

$$E_{x,0}=m_{p,0}\,LHV_f, \quad (4)$$

where $m_{p,0}$, is a fuel dose supplied on single cycle [kg./c. cyl.].

The value of heat losses can be determined by subtracting the amount of net heat released in the cylinder (Eq. 3) from the actual heat supplied during the working cycle (Eq. 4). Then the final equation of the heat losses during combustion process is as below [4]:

$$Q_{sc}=m_{p,0}\,LHV_f-\frac{1}{\left(1-\delta_{ns}\dfrac{LHV_{ub}}{LHV_f}\right)}\int_{\alpha_p}^{\alpha_k}\left[\frac{1}{[\kappa(T,MFB_v)-1]}\left(\kappa(T,MFB_v)p(\alpha)\frac{dV}{d\alpha}+V(\alpha)\frac{dp}{d\alpha}\right)\right]d\alpha. \quad (5)$$

The specific heat ratio employed in this model was calculated as a function of the in-cylinder temperature and the charge composition. Determination of the total heat loss during combustion process, according to the Eq. 5 allows designation of the heat transfer coefficient of the working medium to the wall of the engine combustion chamber.

3. Heat transfer coefficient calculations

There are many papers e.g. [1-3], which describe the methodology how to determine the heat transfer coefficient in internal combustion engine. However, usually each formula presented in the literature gives different values, even is used for the same engine and in the same engine working condition. One of the most popular formula to determine the heat transfer coefficient is the Woschni. For combustion period and expansion process the equation can be written as:

$$\alpha_{sc,W} = 820\, D^{-0.2}\; p(\alpha)^{0.8} \left[2.28\, \overline{w}_{tl} + 3.24 \cdot 10^{-3} \frac{V_s T_2}{p_2 V_2} [p(\alpha) - p_0] \right]^{0.8} T(\alpha)^{-0.53}, \tag{6}$$

where:
$\overline{w}_{tl}$ – mean piston speed [m/s],
$p(\alpha)$ – instantaneous cylinder pressure [MPa],
$T(\alpha)$ – instantaneous cylinder temperature [K],
Vs – swept volume [m^3],
p_0 – motoring pressure [MPa],
p_2, T_2, V_2 – parameters evaluated at start of combustion process.

The heat transfer from the combustion products occurs by convection and radiation. The value of the heat transfer coefficient obtained by Eq. 6 includes convection only. In the spark ignition engine radiation may account for up to 20 per cent of the heat transfer. Therefore it will be possible what the difference in results obtained by Woschni equation is and when using algorithm based on net heat release in cylinder.

In accordance with the principles of heat transfer, the heat flux, transferred from the combustion products to the walls of the working engine space can be described:

$$\dot{Q}_{sc} = A(\alpha)\, \alpha_{sc}\, [T(\alpha) - T_{sc}], \tag{7}$$

where:
$A(\alpha)$ – instantaneous heat exchange surface (crown of a cylinder head, cylinder walls and piston head) [m^2],
α_{sc} – total heat transfer coefficient (includes convection and radiation) [W/m^{2}K],
$T(\alpha)$ – instantaneous cylinder temperature [K],
T_{sc} – average temperature of the heat exchange surface [K].

There is the following correlation between the heat flux $\dot{Q}_{sc}$ and the heat loss rate $dQ_{sc}/d\alpha$:

$$\dot{Q}_{sc} = \frac{dQ_{sc}}{d\alpha}\frac{d\alpha}{dt}, \quad \dot{Q}_{sc} = \omega \frac{dQ_{sc}}{d\alpha}, \tag{8}$$

where ω is the engine angular velocity [rad/s].

Using the Eq. (7) in formula (8), the heat loss rate can be described:

$$\frac{dQ_{sc}}{d\alpha} = \frac{1}{\omega} A(\alpha)\, \alpha_{sc}\, [T(\alpha) - T_{sc}], \tag{9}$$

and after integration the Eq. 9 in range of combustion period it is possible to determine the heat transfer coefficient based on heat losses:

$$Q_{sc} = \frac{1}{\omega} \int_{\alpha_p}^{\alpha_k} \alpha_{sc}\, A(\alpha)\, [T(\alpha) - T_{sc}]\, d\alpha. \tag{10}$$

Finally using Eq. 9 and Eq. 10, the mean value of heat transfer coefficient during combustion process is possible to calculate using following equation:

$$\overline{\alpha}_{sc} = \frac{\omega\, Q_{sc}}{\int_{\alpha_p}^{\alpha_k} A(\alpha)\,[T(\alpha) - T_{sc}]\, d\alpha}\,, \tag{11}$$

where $\overline{\alpha}_{sc}$ is a mean value of heat transfer coefficient taking into account convection and radiation [W/(m^{2}K)].

4. The engine test rig and experimental results

An overview of the engine test bench and its main equipment is presented in Fig. 2. The main components of the experimental set up include:

– naturally aspirated four cylinders SI engine with a capacity of 899 cm^3 and compression ratio equal to 8.8,
– electromagnetic brake,
– electronic management system,
– fuelling system with an accurate mass flow meter,
– measuring devices for temperature and pressure evaluation.

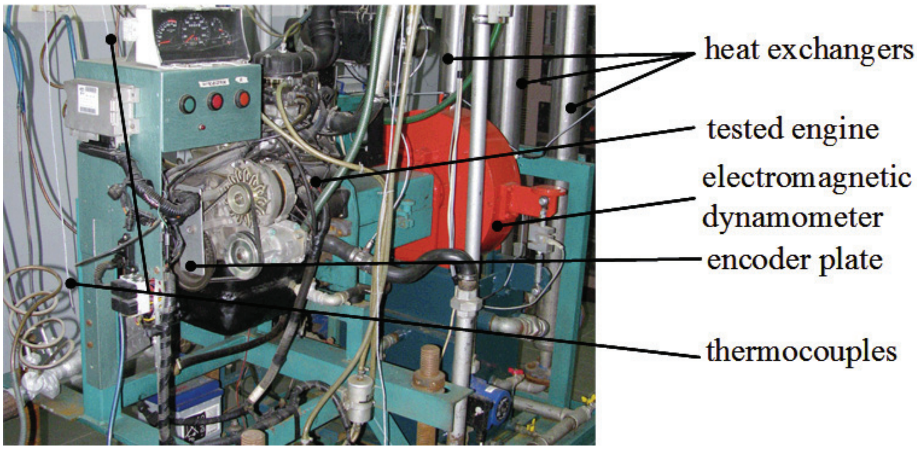

Fig. 2. The test bench with SI engine

The engine test bench has a cooling system for engine lubrication and cooling liquids. Within the cooling system two double-pipe heat exchangers connected to the valves have been employed. Controlling thermostat is located in the primary circuit of the engine cooling system. The settings of the control valves on the secondary side of the heat exchangers allow to adjust the amount of the removed heat, which is then transmitted to the local central heating system. In this way, the stabilization of the engine temperature in the range of 80 ± 5°C can be assured.

A Kistler piezoelectric pressure transducer (type 6117BF17) that had been mounted in a spark plug was used for the acquisition of in-cylinder pressure measurements. This transducer generates analogue voltage signals through the charge amplifier in response to in-cylinder pressure changes. The inlet manifold is equipped with a piezoresistive absolute pressure sensor, which also generates voltage signal. The voltage signals are sampled with sufficient frequency by the data acquisition system (DAQ). An optical encoder with a 0.35 degree Crank Angle (CA°) resolution is used for the engine speed, for the crank angle displacement measurements and to indicate the piston position (at TDC). Furthermore, the optical encoder was used for triggering the data acquisition system. Each data series consisted of one hundred engine working cycles. The MFB_v function was obtained with the Rassweiler and Withrow algorithm to resolve equation (5).

Comparison of the results obtained by experimental algorithm (Eq. 11) presented in this paper with the results calculated by Woschni function (Eq. 6), for 5 values of engine speed from 1000 rpm to 5000 rpm is shown in Fig. 3-7. During experiments the engine load was changing from 5% to 100%. Presented results refer to combustion period only.

The figures below shows a good qualitative agreement of the used methods to determine the heat transfer coefficient in the cylinder of SI engine during the combustion process. The one reason for the high quantitative difference is fact that the Woschni equation is includes only convection heat transfer. On the next figures the similar trend is observed, but in the range of high engine loads and higher engine speed (3000 rpm, 4000 rpm and 5000 rpm) the values obtained using Woschni equation are even 60% higher than values calculated by using heat release algorithm proposed in this paper.

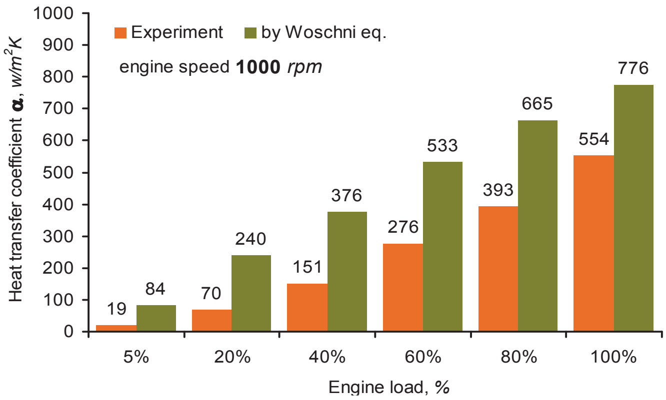

Fig. 3. Comparison of the results obtained by experimental method and Woschni equation, for constant engine speed 1000 rpm and changing load

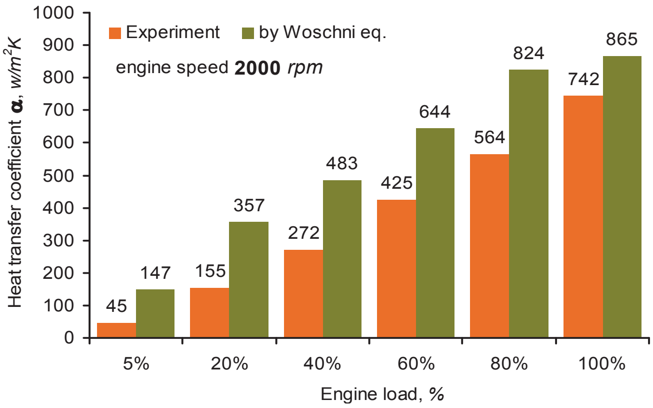

Fig. 4. Comparison of the results obtained by experimental method and Woschni equation, for constant engine speed 2000 rpm and changing load

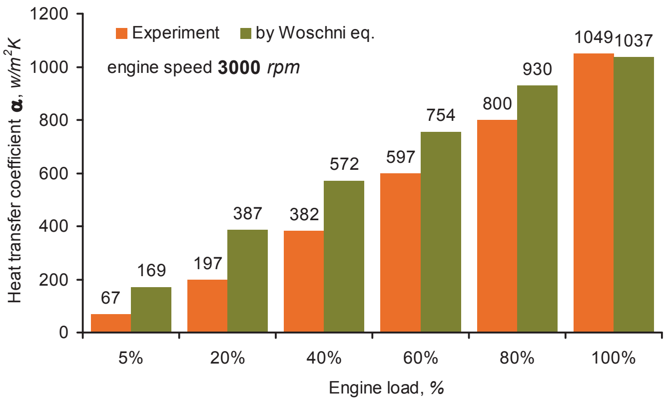

Fig. 5. Comparison of the results obtained by experimental method and Woschni equation, for constant engine speed 3000 rpm and changing load

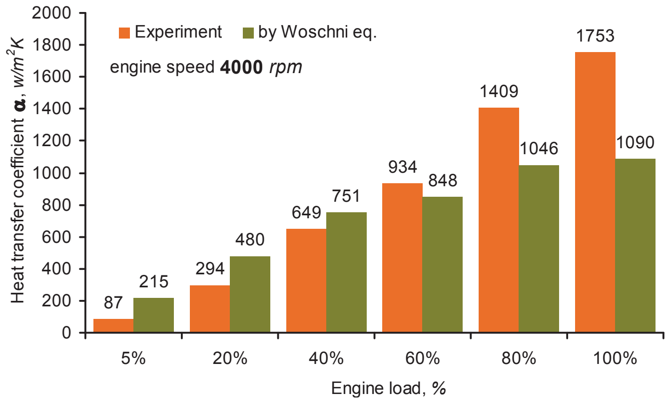

Fig. 6. Comparison of the results obtained by experimental method and Woschni equation, for constant engine speed 4000 rpm and changing load

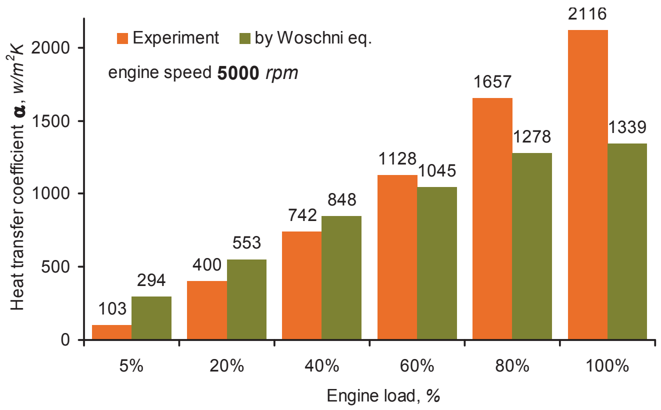

Fig. 7. Comparison of the results obtained by experimental method and Woschni equation, for constant engine speed 5000 rpm and changing load

5. Conclusions

The algorithm of determination the heat transfer coefficient during the combustion process in the cylinder of internal combustion engine has been described and used regarding to experimental data of SI engine.

The results obtained using Woschni equation and proposed algorithm which is based on in cylinder pressure function are qualitatively comparable but mainly for low engine speed (1000 rpm and 2000 rpm). In the range of this engine speed the Woschni equation bring the lower values comparing to presented algorithm. In the range of high engine loads and higher engine speed (3000 rpm, 4000 rpm and 5000 rpm) the values obtained using Woschni equation are even 60% higher than values calculated by using heat release algorithm proposed in this paper.

The heat transfer from the combustion products occurs by convection and radiation. when, the value of the heat transfer coefficient obtained by Woschni equation includes convection only. It is one of the reason which leads to the high difference (for some operating points of ICE) at the obtained results.

Presented in this paper the "heat release rate" way of determination the heat transfer coefficient includes complex algorithm which involves all major physico-chemical phenomena occurring in the system during combustion process. These phenomena have a significant impact on obtained results. Finally, regarding to it, the presented algorithm should bring more realistic values than the universal equations.

Acknowledgements

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