HOANG NGUYEN, D.Sc.,M.E.
Department of Basic Engineering Sciences
Gdynia Maritime University

# A simulation model of heat exchange in the ship diesel engine cylinder - - environment system

SUMMARY

*In the paper a model of heat exchange in the diesel engine cylinder – environment system was presented together with its application to 6ZA40S engine. The model, a component of the general model of energy transformation process in the engine, makes it possible to calculate instantaneous values of heat flow through cylinder walls. It can operate in real time mode within the entire range of engine load.*

*The model was verified by comparing the simulation results and data from engine bed test measurements as well as by assessing time course correctness of selected working parameters.*

## INTRODUCTION

The subject of this work is a part of simulation model of the energy transformation process occurring in a ship main diesel engine of medium speed [2], i.e. the model of heat exchange within engine cylinder - environment system. It deals with heat transmission through walls of the cylinder liner, cylinder head and piston crown from working medium within the cylinder to cooling medium and further to the engine environment.

The process is unsteady due to cyclic operation of the engine cylinder system. Temperature values of the working medium and heat exchanging surfaces are periodically changeable. The model makes it possible to calculate instantaneous values of heat flow through the cylinder walls.

A general physical and mathematical model of the process in question is presented as well as its application to 6ZA40S diesel engine. The real time mode was applied to the process simulation.

## GENERAL MODEL

To specify the model the following statements and assumptions were made :

* The heat exchange process occurs in a ship propulsion diesel engine of medium speed.
* This is four-stroke, trunk-piston, turbo-charged engine.
* The cylinders and cylinder heads are fresh-water cooled.
* The engine operates in the ship propulsion system equipped with a controllable pitch propeller.
* The engine operates at constant rotational speed and its load is controlled by changing the propeller pitch.
* The engine is fitted with a multirange speed governor.
* Working medium within the cylinder changes its internal energy due to absorption of heat from burning the fuel injected to the cylinder, heat exchange with the surrounding walls, gas exchange and by doing work.
* The heat exchange process between working medium and cooling water through the cylinder walls is unsteady as the heat exchange surface area, wall temperature and overall heat-transfer coefficient from the side of working medium, ($\alpha_o$), is likely to change in function of the crank angle.
* Cooling water temperature at outlet from the engine is controlled by means of a system fitted with PI controller. The control is realized by changing the rate of water flow through the cooler.
* The overall heat-transfer coefficient from the side of cooling water, ($\alpha_w$), can be assumed constant due to small changes of its flow velocity within the engine

Moreover, the following simplifying assumptions were made :

- Temperature, pressure and gas constant values of the working medium at a given time instant are the same in every point of the working chamber.
- The heat exchange area is limited to its three most important components, namely :
  - bottom plate area of cylinder cover together with valves
  - piston crown area and
  - changeable area of cylinder liner being in contact with the working medium.
- Average values of wall thickness of cylinder liner, cylinder cover bottom plate, piston crown and cylinder block are assumed constant.

- Heat flow occurs only in perpendicular direction to the particular surfaces.
- The cylinder liner is a flat plate, and properties of cylinder wall material do not depend on temperature, time and geometrical coordinates.
- Inertia is not taken into account in the state change process of working medium and in producing toxic compounds within the cylinder.

## MODELLING METHOD

The heat exchange process was modelled with the use of the finite element method (FEM). The cylinder liner was divided into *n* finite elements (FE) within the range from the bottom dead centre (BDC) to top dead centre (TDC) (Fig.1).

For the finite element a mathematical model of heat flow through cylinder liner was elaborated.

On the basis of the model investigations its dynamic behaviour was approximated by means of a 1st order inertial element of variable time constants depending on values of the coefficient $\alpha_o$.

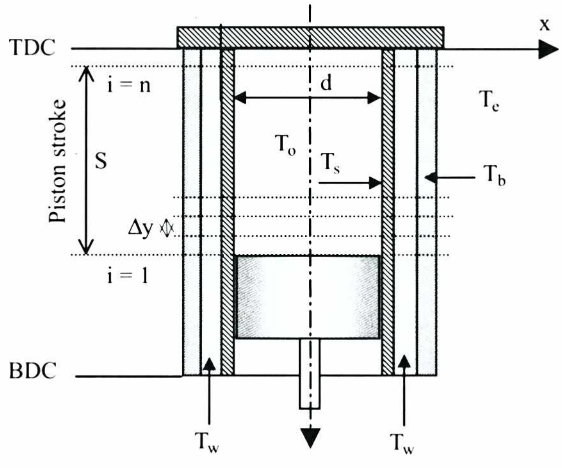


*Fig.1. Schematic diagram of the modelled cylinder and its division into n finite elements*
*Notation : d- piston diameter; $T_b$, $T_o$, $T_c$, $T_s$, $T_w$ - temperature of cylinder block, working medium, environment, cylinder wall and cooling water at inlet to the cylinder, respectively; S - piston stroke; i - consecutive FE number, where $i \le n$ (n - total number of FEs); $\Delta y$ - FE height*

### Model of heat flow through the cylinder liner

The model in question is presented in Fig.2. Walls of the cylinder liner were divided into FEs of $\Delta x$ thickness.

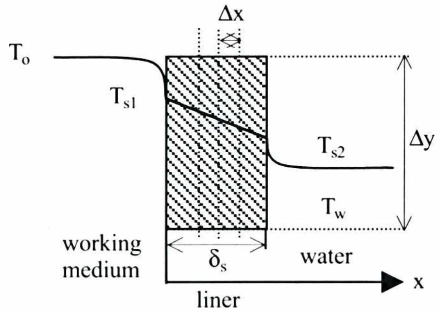


*Fig.2. Schematic model of heat transfer through cylinder liner wall*
*Notation : $T_{s1}$, $T_{s2}$ - temperature of internal and external cylinder liner surface, respectively; $\delta_s$ - liner wall thickness; $\Delta x$ - FE thickness*

The differential equation of unsteady heat transfer through the wall is of the following form :

$$\frac{\partial T_s}{\partial \tau} = a_s \frac{\partial^2 T_s}{\partial x^2} \tag{1}$$

with 3rd kind boundary conditions :

$$\begin{aligned} \alpha_o (T_o - T_{s1}) &= -\beta_s \left( \frac{\partial T_s}{\partial x} \right)_{x=0} \\ \alpha_w (T_{s2} - T_w) &= -\beta_s \left( \frac{\partial T_s}{\partial x} \right)_{x=\delta_s} \end{aligned} \tag{2}$$

where :

$a_s$ - wall conductivity temperature
$\beta_s$ - thermal conductivity through wall.

As values of working medium temperature and velocity within the cylinder substantially differ in different phases of working cycle, values of the coefficient $\alpha_o$ were calculated in function of the crank angle in accordance with the empirical formula given in [5], as follows :

$$\alpha_o(\varphi) = 130\, d^{-0.2} p_o^{0.8}\, T_o^{-0.53} \left[ C_1 C_m + C_2 \frac{V_{ps} T_1}{p_1 V_1} (p_o - p_{ed}) \right]^{0.8} \tag{3}$$

where :

$\varphi$ [deg] - crank angle
d [m] - cylinder diameter
$p_o$ [bar] - working medium pressure in the cylinder, at the $\varphi$ crank angle
$V_{ps}$ [$m^3$] - piston stroke volume
$C_m$ [m/s] - mean piston speed
$p_{ed}$ [bar] - pressure in the cylinder when the engine is external driving
$T_1$, $p_1$, $V_1$ - working medium parameters at the beginning of compression
$C_1 = 6.18$ - for compression, combustion and expansion stroke,
2.28 - for exhaust stroke
$C_2 = 0.00324$ [m/s K] - for engines of direct injection combustion chamber type
0.00622 [m/s K] - for engines of the divided combustion chamber type.

It can be observed that the above presented heat exchange model is non-linear.

Temperature value in j-th node and (k+1)-th time instant was determined by means of the finite difference method, as follows :

$$\begin{aligned} T_s(j, k+1) = \frac{1}{N} \big[ & T_s(j+1, k) + \\ & + (N-2) T_s(j,k) + T_s(j-1,k) \big] \end{aligned} \tag{4}$$

where :

$$N = \frac{\Delta x^2}{a_s \Delta\tau} \ge 2$$

The cylinder cover bottom plate and piston crown was divided into FEs in the same way.

The FE was approximated by applying a 1st order inertia element as the calculation procedure was assumed to be carried out in real time mode and by using non-specialized computers.

Transfer function of the element can be described by the following equations :

$$
\begin{aligned}
T_{s1}(s) &= \frac{K_{11}}{A_1(\alpha_o)s+1}T_o(s)+\frac{K_{12}}{A_2(\alpha_o)s+1}T_w(s) \\
T_{s2}(s) &= \frac{K_{21}}{A_2(\alpha_o)s+1}T_o(s)+\frac{K_{22}}{A_1(\alpha_o)s+1}T_w(s)
\end{aligned}
\tag{5}
$$

where :

$s$ - complex variable of transfer function
$K_{ij}$ - constant values
$A_1, A_2$ - time constants dependent on the heat transfer coefficient $\alpha_0$.

The dependence of the time constants on the coefficient $\alpha_0$ in equations (5) can be determined on the basis of investigations of the „exact" model described by equations (1) - (4).

Parameters of the working medium in the cylinder (Fig.3) can be determined by means of the energy and mass balance equations (6) and (7) :

$$
\frac{d(m\cdot u)_o}{d\tau}=\frac{dQ_{fo}}{d\tau}-\frac{dQ_c}{d\tau}-p_o\frac{dV_o}{d\tau}+h_{in}\frac{dm_{in}}{d\tau}-h_{out}\frac{dm_{out}}{d\tau} \tag{6}
$$

$$
\frac{dm_o}{d\tau}=\frac{dm_{in}}{d\tau}-\frac{dm_{out}}{d\tau} \tag{7}
$$

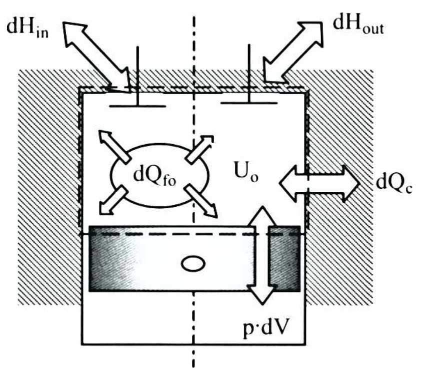


***Fig.3.*** *Schematic diagram of energy balance of the „cylinder" thermodynamic sub-system*
***Notation :*** *$dH_{in}$ - input enthalpy increment; $dH_{out}$ - output enthalpy increment; $dQ_{fo}$ - heat emitted during fuel combustion; $dQ_c$ - heat exchanged with walls; $p{\cdot}dV$ - work done by working medium; $U_o$ - working medium internal energy*

The differential equation of working medium temperature can be presented in the following form :

$$
\frac{dT_o}{d\tau}=\frac{1}{m_o\,c_v}\left(\frac{dQ_{fo}}{d\tau}-\frac{dQ_c}{d\tau}-p_o\frac{dV_o}{d\tau}+ h_{in}\frac{dm_{in}}{d\tau}-h_{out}\frac{dm_{out}}{d\tau}-u_o\frac{dm_o}{d\tau}-m_o\frac{\partial u}{\partial F}\frac{dF}{d\tau}\right) \tag{8}
$$

where :

F - ratio of burned fuel mass and air mass contained in the working medium.

A calculation model was elaborated of the working medium flow rate through inlet and exhaust valves in function of their opening state and pressure within the cylinders, suction and exhaust manifolds.

The cylinder operation cycle was divided into 4 phases :

⇒ Phase I – only inlet valves opened; inlet of fresh air to the cylinder
⇒ Phase II – all valves closed; compression, combustion and expansion of the working medium within the cylinder
⇒ Phase III – only exhaust valves opened; outlet of exhaust gas from the cylinder.
⇒ Phase IV – all valves opened; simultaneous outlet of the exhaust gas and inlet of fresh air.

Example balance equations for two first phases are below presented :

**Phase I**

**a)** if pressure in the inlet manifold is greater than that within the cylinder : $p_{im} > p_{cyl}$

$$
\left.
\begin{aligned}
&\frac{dm_o}{d\tau}=\dot{m}_+ \\
&\frac{dT_o}{d\tau}=\frac{1}{m_o\,c_v}\cdot\left[-\frac{dQ_c}{d\tau}+h_{im}\dot{m}_+-p_o\frac{dV_o}{d\tau}+ -u_o\frac{dm_o}{d\tau}-m_o\left(\frac{\partial u}{\partial F}\right)\frac{dF}{d\tau}\right] \\
&\frac{dF}{d\tau}=-\frac{(F+1)F}{m_o}\cdot\frac{dm_o}{d\tau}
\end{aligned}
\right\} \tag{9}
$$

**b)** if pressure in the inlet manifold is smaller than that within the cylinder : $p_{im} < p_{cyl}$

$$
\left.
\begin{aligned}
&\frac{dm_o}{d\tau}=-\dot{m}_- \\
&\frac{dF}{d\tau}=0 \\
&\frac{dT_o}{d\tau}=\frac{1}{m_o\,c_v}\left[-\frac{dQ_c}{d\tau}-h_o\dot{m}_--p_o\frac{dV_o}{d\tau}-u_o\frac{dm_o}{d\tau}\right]
\end{aligned}
\right\} \tag{10}
$$

where :

$\dot{m}_+, \dot{m}_-$ - working medium flow rate through inlet valves from the manifold to the cylinder and back, respectively,
$h_{im}$ - working medium enthalpy in the inlet manifold.

**Phase II**

During this phase the following is valid :

$$
\frac{dm_{in}}{d\tau}=0 \quad \text{and} \quad \frac{dm_{out}}{d\tau}=0
$$

Two transformation stages of the working medium within the cylinder can be distinguished :

- $1^{st}$ of polytropic compression and expansion (without any change of mass and composition of working medium)
- $2^{nd}$ of chemical composition change of working medium, resulting from combustion of the fuel injected into the cylinder.

The combustion process was assumed instantaneous and the enthalpy increase of the delivered fuel, $h_{fo}$, negigible.

So, the relevant balance equations are of the following form :

$$\left.\begin{array}{l} \dfrac{dm_{out}}{d\tau}=0 \qquad \dfrac{dm_o}{d\tau}=\dfrac{dm_{fo}}{d\tau} \\[2ex] \dfrac{dT_o}{d\tau}=\dfrac{1}{m_o c_v}\left(-\dfrac{dQ_c}{d\tau}+\dfrac{dQ_{fo}}{d\tau}-p_o\dfrac{dV_o}{d\tau}+\right. \\[2ex] \qquad \left.-u_o\dfrac{dm_o}{d\tau}-m_o\left(\dfrac{\partial u}{\partial F}\right)_o\dfrac{dF}{d\tau}\right) \\[2ex] \dfrac{dF}{d\tau}=\dfrac{F+1}{m_o}\left[(F+1)\dfrac{dm_{fo}}{d\tau}-F\dfrac{dm_o}{d\tau}\right] \end{array}\right\} \quad (11)$$

To determine the heat flow emitted during combustion the Wiebe's dimensionless combustion function was applied [5] :

$$B=1-e^{C\left(\frac{\varphi-\varphi_{cb}}{\varphi_{ce}-\varphi_{cb}}\right)^{w+1}} \quad (12)$$

where :

- $B$ - share of the heat emitted up to the crank angle $\varphi$ in the value of the total heat emitted during one cycle
- $e$ - Neper number
- $w$ - shape exponent of combustion function
- $C$ - constant
- $\varphi_{ce}$ - crank angle at the end of combustion
- $\varphi_{cb}$ - crank angle at the beginning of combustion.

The combustion heat flow is determined by the following formula :

$$\frac{dQ_{fo}}{d\tau}=m_{foc}W\eta_{com}\frac{dB}{d\tau} \quad (13)$$

where :

- $m_{foc}$ - fuel mass per operation cycle
- $W$ - fuel calorific value
- $\eta_{com}$ - combustion effectiveness.

In order to apply the Wiebe function it is necessary to determine values of the following parameters for each engine load :

- the ignition delay $\Delta\varphi_{id} = \varphi_{cb} - \varphi_{ib}$
- the combustion angle $\Delta\varphi_{com} = \varphi_{ce} - \varphi_{cb}$

where : $\varphi_{ib}$ - crank angle at the beginning of fuel injection.

Approximate assessment of values of the parameters is troublesome and often providing misleading results.

Therefore a method was applied consisting in determination of the so-called reference point $(\Delta\varphi_{id}, \Delta\varphi_{com})_{rp}$ for which Wiebe function parameters were determined at each engine load [3,4]. Influence of the following parameters on the Wiebe's function coefficients was accounted for :

- igintion delay $\Delta\varphi_{id}$
- excess air number $\lambda$
- engine load expressed by three parameters :
  - initial pressure $p_{300}$ (for the crank angle of 300 deg)
  - initial temperature $T_{300}$ (for the crank angle of 300 deg)
  - residual exhaust gas content $\gamma$.

The following equations make it possible to calculate the Wiebe's equation coefficients for a new working point :

→ for the beginning of combustion : $\varphi_{cb} = \varphi_{ib} + f \cdot (\Delta\varphi_{id})_{rp}$

→ for the end of combustion : $\varphi_{ce} = \varphi_{cb} + g \cdot (\Delta\varphi_{com})_{rp}$

where :

- f - influence factor for combustion beginning
- g - influence factor for combustion end.

$$\left.\begin{array}{l} f=\dfrac{\Delta\varphi_{id}}{(\Delta\varphi_{id})_{rp}}=f_{id}\cdot f_\lambda\cdot f_p\cdot f_T\cdot f_\gamma \\[3ex] g=\dfrac{\Delta\varphi_{com}}{(\Delta\varphi_{com})_{rp}}=g_{id}\cdot g_\lambda\cdot g_p\cdot g_T\cdot g_\gamma \end{array}\right. \quad (14)$$

For the group of diesel engines the following parameters of the reference point were established [3], used to calculate the Wiebe's equation coefficients for the remaining engine working points :

| | |
|---|---|
| - rotational speed [rpm] | 510 |
| - effective pressure [MPa] | 1.9 |
| - air excess number $\lambda$ [-] | 1.4 |
| - residual exhaust gas content $\gamma$ [%] | 2.63 |
| - initial pressure $p_{300}$ [MPa] | 2.5 |
| - initial temperature $T_{300}$ [K] | 613 |
| - ignition delay angle $\Delta\varphi_{id}$ [°] | 12 |
| - combustion beginning $\varphi_{cb}$ [°] | 348 |
| - combustion angle $\Delta\varphi_{com}$ [°] | 40 |
| - combustion end angle $\varphi_{ce}$ [°] | 388 |
| - Wiebe's equation exponent [-] | 1.2 |

Relationships which express dependence of the Wiebe function influence factors on some engine working parameters, are given in Tab.1.

***Tab. 1.*** *Dependence of the combustion function influence factors f and g on some engine operation parameters [5]*

| Type of influence | Factor f | Factor g |
|---|---|---|
| Influence of ignition delay | $f_{id}=\dfrac{430-\Delta\varphi_{id}}{430-(\Delta\varphi_{id})_{rp}}$ | $g_{id}=1$ |
| Influence of excess air number | $f_\lambda=\dfrac{2.2\lambda^2-3.74\lambda+2.54}{2.2\lambda_{rp}^2-3.74\lambda_{rp}+2.54}$ | $g_\lambda=\dfrac{2\lambda^2-3.4\lambda+2.4}{2\lambda_{rp}^2-3.4\lambda_{rp}+2.4}$ |
| Influence of initial pressure $p_{300}$ | $f_p=\left(\dfrac{p_{300}}{p_{rp,300}}\right)^{-0.47}$ | $g_p=\left(\dfrac{p_{300}}{p_{rp,300}}\right)^{-0.28}$ |
| Influence of initial temperature $T_{300}$ | $f_T=2.16\dfrac{T_{rp,300}}{T_{300}}-1.16$ | $g_T=1.33\dfrac{T_{rp,300}}{T_{300}}-0.33$ |
| Influence of residual exhaust gas content | $f_\gamma=0.088\dfrac{T}{T_{rp}}+0.912$ | $g_\gamma=0.237\dfrac{T}{T_{rp}}+0.763$ |

During thermodynamic calculations of the engine continuous determination of working medium calorific parameters (i.e. specific heat, internal energy, enthalpy) is required for particular sub-systems. Their values were determined, within the assumed reference system, by applying the Benson's model [1].

By means of the author's model it was possible to determine, at the time instant $(\tau+\Delta\tau)$, values of the working medium parameters within the cylinder, temperature distribution in the cylinder liner walls and instantaneous heat flow through surrounding walls.

## RESULTS OF THE MODEL INVESTIGATIONS

Example heat exchange calculations by means of the simulation model in question were carried out for 6ZA40S main diesel engine installed in a fishing trawler of B672 series. The simulation results were assessed both for static and dynamic working conditions.

### Static (steady-state) conditions

Values of bed-state working parameters of the engine were obtained from investigations of the simulation model under engine bed test loading conditions. In Tab.2 some their results are compared with results of the engine bed test measurements under different loads [7].

From the data given in Tab.2. it results that the simulation results are close to the measured ones at different engine loads. At the same load the model "developes" the power close to that of the real engine (item 1) at the constant rated speed of 510 rpm (item 2). Values of effective pressure (item 3) or combustion pressure (item 4) are also very close. Relative error does not exceed 5%. Simulation results of heat exchange of the cylinder-environment system are also similar to the measured data of the real engine. The difference between the measured and calculated temperature increment of cooling water behind the cylinders (item 5) amounts to 0-1°C. Zero increment of the measured temperature at lower loads results from a specific accuracy of the measuring instruments used to the real engine. The greater engine load the greater heat amount taken away by the cylinder cooling water and its temperature increment. The smaller increment of the cooling water temperature measured behind the engine (item 6), at increasing load from 50% to 75%, would result only from measurement error.

*Tab.2. Comparision of the measured [7]and calculated results for 6ZA40S diesel engine in bed-test conditions*

| No | Item | | Desired load factor | | | | |
|---|---|---|---|---|---|---|---|
| | | | % | 100 | 90 | 75 | 50 |
| 1 | Engine power | measurement | kW | 3350 | 3011 | 2525 | 1678 |
| | | calculation | kW | 3330 | 2964 | 2526 | 1704 |
| 2 | Engine rotational speed | measurement | $min^{-1}$ | 508.5 | 508 | 511 | 509.3 |
| | | calculation | $min^{-1}$ | 510 | 510 | 510 | 510.7 |
| 3 | Mean effective pressure | measurement | MPa | 1.9 | 1.7 | 1.4 | 0.9 |
| | | calculation | MPa | 1.89 | 1.68 | 1.4 | 0.92 |
| 4 | Combustion pressure | measurement | MPa | 15.8 | 15 | 12.3 | 9.3 |
| | | calculation | MPa | 16 | 14.6 | 12.5 | 8.9 |
| 5 | Temperature increase of cooling water behind cylinders | measurement | °C | 1.5 | 0.5 | 0 | 0 |
| | | calculation | °C | 1.35 | 1.25 | 1.08 | 0.75 |
| 6 | Temperature increase of cooling water behind engine | measurement | °C | 4 | 3 | 1 | 2 |
| | | calculation | °C | 3.5 | 3.2 | 2.8 | 1.7 |

### Dynamic (transient- state) conditions

#### Simulative calculations of the heat exchange process in the cylinder

The simulations were carried out at 50% and 100% engine load and its rated speed of 510 rpm.

Course diagrams of calculated working medium pressure and temperature in the cylinder are presented in Fig.4.

In Fig.5. courses of calculated temperature of the cylinder liner wall, cooling water and the last FE of the cylinder block (i = n) are presented in function of time. Initial temperature of the walls and cooling water was assumed the same as that of cooling water at inlet to the engine (350 K).

The dinstinct stationary temperature oscillations which can be observed on the inner surface of the cylinder liner, almost decay on its outer surface (see the expanded fragments of curve 1 and 2 in Fig.5a).

The average temperature of the inner surface reached the stationary value of 506 K (233°C) at 100 % load. The temperature oscillated between 200 °C and 250 °C.

The temperature oscillation amplitude of the inner surface amounted to about 2 K, and that of the outer surface to 0.02 K. The cooling water temperature and that of the cylinder block did not reveal oscillations.

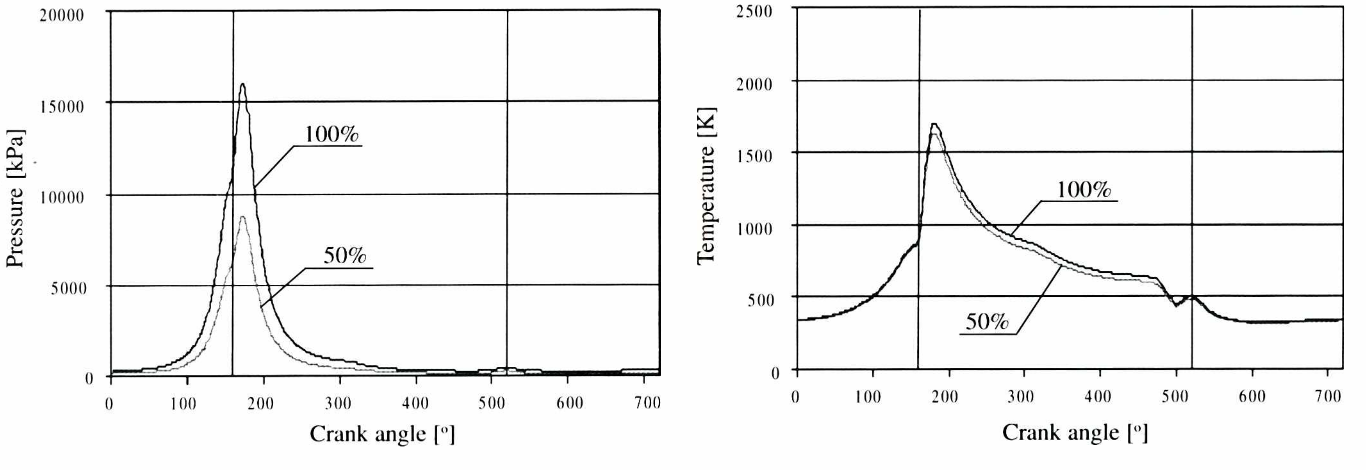


*Fig.4. Course diagrams of working medium pressure and temperature in the cylinder versus crank angle, calculated for 100 % engine load and 50 % engine load*

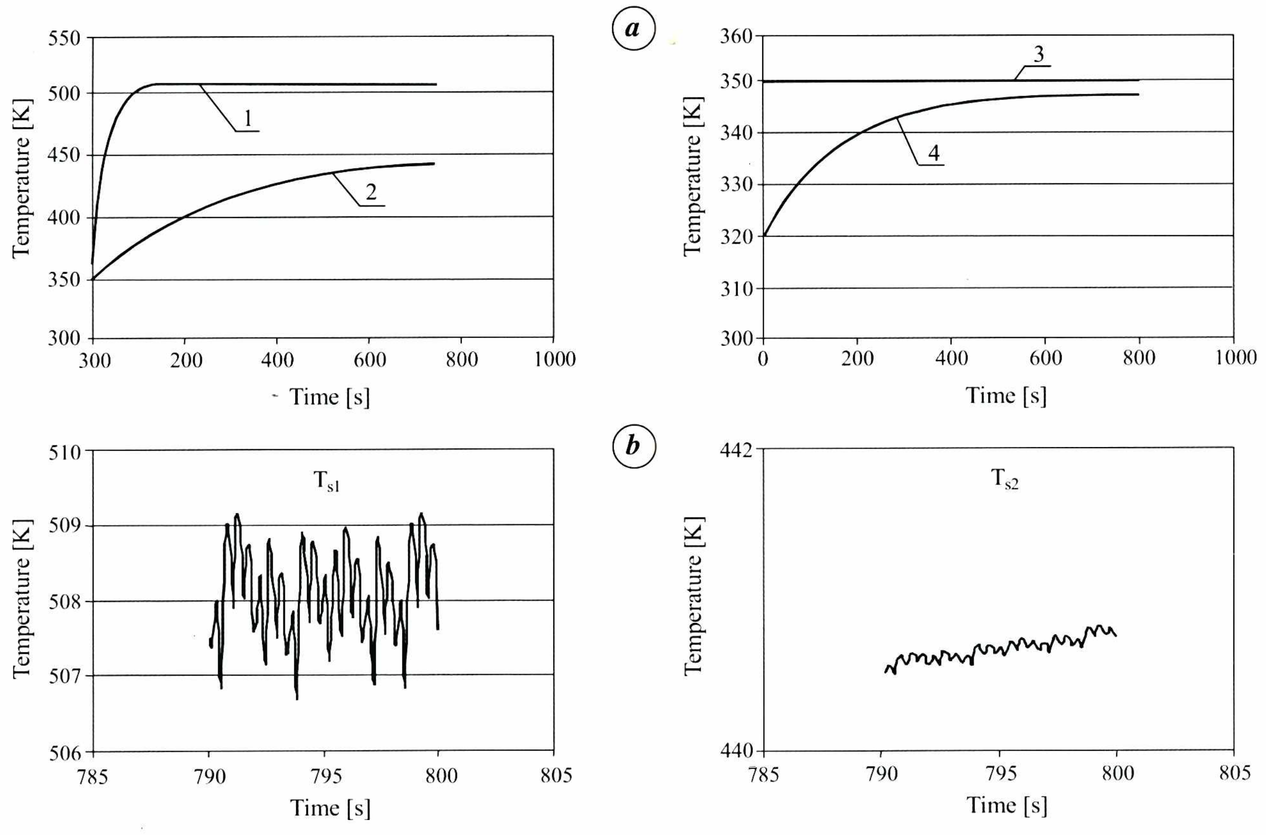


***Fig.5.*** *Course diagrams of temperature versus time of:*
*1 - internal surface of cylinder liner wall, 2- external surface of cylinder liner wall, 3 - cooling water,*
*4 - the last control element of cylinder block (i = n) - in response to periodical change of working medium temperature*
*(calculated for 100 % engine load and 510 rpm engine speed).*
***a)*** *Approximate curves,* ***b)*** *Expanded fragments of $T_{s1}$ and $T_{s2}$ curves*

## CONCLUSIONS

- The presented simulation model makes it possible to investigate heat transfer through the cylinder walls in steady - and transient - state conditions.
- It can operate in real time mode within the entire range of engine loading.
- Accuracy of the model applied to steady-state operation of the engine is acceptable as the appropriate calculation and measurement results (see Tab.2) differ to each other by no more than ±3%.
- Modelling results of the investigated dynamic processes can be deemed phenomenologically correct (see Fig.4 and 5).

***Appraised by Jan K. Wlodarski, Prof.,D.Sc.***

**NOMENCLATURE**

a - temperature conductivity [m²/s]
$c_v$ - specific heat [J/kgK]
F - ratio of burned fuel mass and air mass [-]
h - specific enthalpy [J/kg]
i - FE successive number
j - node successive number
$K_{ij}$ - constant coefficients
m - mass [kg]
p - pressure [Pa]
Q - heat, work [J]
$Q_c$ - heat exchanged with combustion chamber walls [J]
s - transfer function complex variable
T - temperature [K]
u - specific internal energy [J/kg]
U - internal energy [J]
V - volume [m³]
α - heat transfer coefficient [W/m² K]
$\alpha_o$ - heat transfer coefficient from the side of working medium [W/m² K]
$\alpha_w$ - heat transfer coefficient from the side of cooling water [W/m² K]
β - thermal conductivity [W/mK],
γ - residual exhaust gas content [%]
Δx - FE thickness [m]
Δy - FE height [m]
λ - excess air number [-]
τ - time [s]
φ - crank angle [deg]

**Indices**

b - cylinder block
cb - combustion beginning
ce - combustion end
com - combustion
e - environment
fo - fuel oil
ib - injection beginning
id - ignition delay
im - inlet manifold
in - inlet
o - of working medium parameters at the φ crank angle
out - outlet
rp - reference point
s - wall
$s_1$ - internal surface of cylinder liner
$s_2$ - external surface of cylinder liner
w - cooling water
λ - excess air number
γ - residual exhaust gas content

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