CI ENGINE AS A CASE STUDY OF THERMOMECHANICAL FE ANALYSIS OF THE PISTON – PISTON RINGS – CYLINDER SYSTEM

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Abstract

The compression-ignition engine as a case study of a methodology of the numerical modelling and simulation of the piston – piston rings – cylinder system was presented in the paper. Thermomechanical FE analysis, taking into account thermal and mechanical loads, was carried out using the MSC.Marc/Mentat software. The mechanical loads included loads due to inertial and gas forces as well. A three dimensional solid geometrical model of the considered set was developed using AutoCAD software, whereas the finite element mesh was generated using Altair HyperMesh. Kinematic boundary conditions – the vertical displacement and acceleration of the piston – were described by the corresponding curves as a function of time for selected engine speed. Changes in pressure on the piston crown were estimated based on data from the engine manufacturer and the corresponded indicator diagram available in the literature. The results of thermomechanical FE analysis were presented in the form of stress and/or displacement contours. The main aim of the analysis was to determine the deformation of the piston depending of on the piston material. Two types of material were compared – the actual one PA12 aluminum alloy and the new composite material with low hysteresis. The second material was characterized by slight differences of the coefficient of thermal expansion for heating and cooling.

Keywords: *engine piston, composite piston, thermomechanical analysis, FE analysis*

1. Introduction

The engine pistons are the most loaded elements of the internal combustion engine. Their purpose is to satisfy the requirements concerning durability and functionality. Therefore, a new type of material with high strength properties at high temperatures is still searched. In addition, the new materials should be characterized by a low hysteresis – the differences of the coefficient of thermal expansion for heating and cooling are not supposed to be significant. It allows increasing piston resistance to fatigue damage and thermal shock.

A piston of the S12U compression-ignition engine was selected as a representative for the study. Such engines are installed in one of the popular Polish tanks – PT-91 *Twardy*. The engine is a 12-cylinder vee engine but the V-configuration is not fully symmetric. The engine is equipped with one row of cylinders including the master connecting rods and the second row including the articulated connecting rods. The row with the master connecting rods was taken into consideration in the current study. Selected technical data for the S12U engine is provided in Tab. 1.

2. The load of the piston – piston rings – cylinder system

Load of the piston – piston rings – cylinder system can be divided into mechanical and thermal loads [1]. The mechanical load by inertial forces is caused by the to-and-fro motion of the piston. The load of gas forces is time changing according to the cycle of engine operation and depends on the thermodynamic changes occurring in the cylinder. The periodic changes in temperature of the

Tab. 1. Selected technical data for S12U engine [5]

Parameter	Unit	Value
Nominal Engine Speed	(rpm)	2000
Bore	(mm)	150
Stroke	(mm)	180.0[1)] 186.7[2)]
Displacement	(ccm)	38,880
Power	(kW) (KM)	625 850

[1)] for the row with master connecting rods
[2)] for the row with articulated connecting rods

combustion chamber walls do not exceed 1% of the working medium temperature changes and temperature fluctuations in the depth of the walls are insignificant. Hence, the quasi-steady state of the heat transfer was assumed for the thermal load calculation. Moreover, due to the thermal inertia, the temperature field loading the system was treated as mean and constant values for the full cycle.

Vertical displacement x of a moving piston in the cylinder was described by the curve resulting from the kinematics of the considered crank-piston system (Fig. 1). On the basis of the figure, displacement x can be determined as follow:

$$x = L + R_c - (R_c \cos\alpha + L\cos\beta)\,. \tag{1}$$

Making some transformations [1], we can simplify the above relation to the following form:

$$x = R_c\left[(1-\cos\alpha) + \frac{\lambda}{2}(1-\cos^2\alpha)\right], \tag{2}$$

where λ is equal to R_c / L, and α is a crank angle.

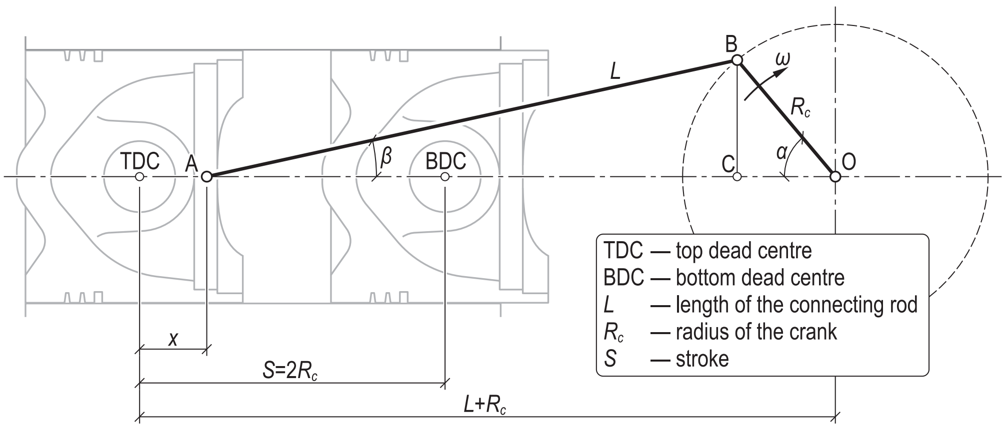

Fig. 1. A scheme of the engine crankshaft for the considered row of cylinders (based on [1])

Acceleration of the piston a can be formulated as a second time derivative of the displacement x

$$a = \ddot{x} = R_k\,\omega^2\,(\cos\alpha + \lambda\cos 2\alpha)\,, \tag{3}$$

where ω is an angular velocity of the crankshaft related to the engine revolutions per minute (rpm).

Curves describing kinematic conditions – displacement x and acceleration a of the piston – are depicted in Fig. 2. Such curves were defined as a function of the crank angle. For the current study, they were converted to the time-function curves for the nominal engine speed 2000 rpm.

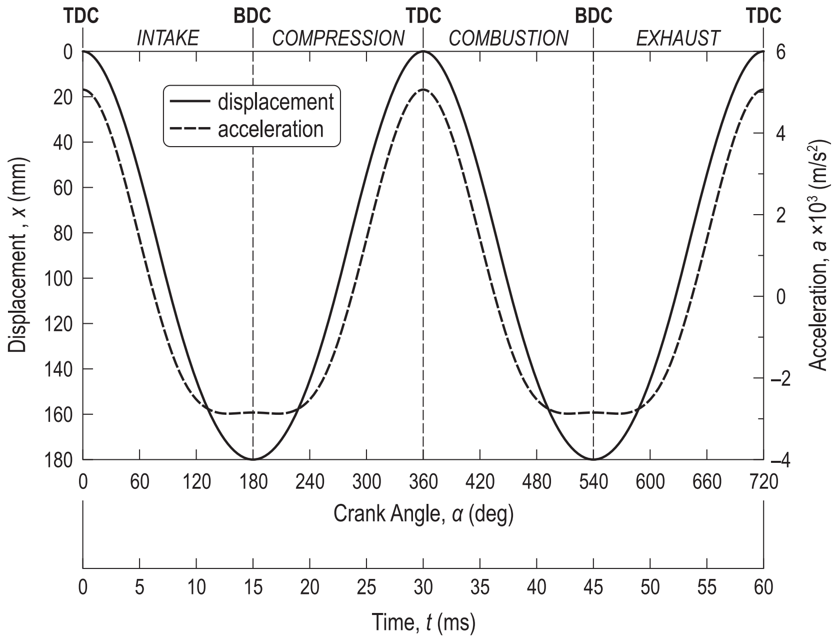

Fig. 2. Changes in displacement and acceleration of S12U engine piston for the nominal engine speed (2000 rpm)

The inertial force acting on the piston was defined as:

$$B = -m\,a\,, \tag{4}$$

where m is the mass of the piston.

Forces from the gas pressure in the cylinder are the second type of the piston loads. Fig. 3 shows a scheme of the engine crankshaft loaded by force P, which is the resultant of gas P_g and inertial B force. Force P can be decomposed into normal force N – perpendicular to the cylinder bearing surface, and force S along the axis of the connecting rod. Normal force N was defined as follow:

$$N = P\,\mathrm{tg}\beta = P\frac{2\lambda\sin\alpha}{2-\lambda^2\sin^2\alpha}\,. \tag{5}$$

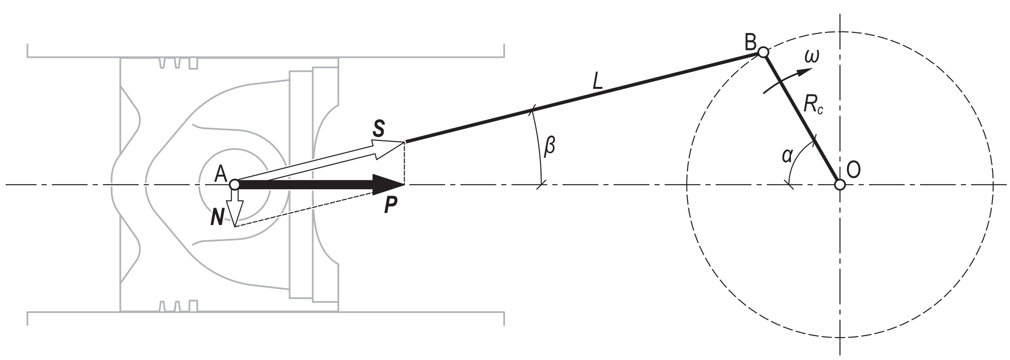

Fig. 3. The distribution of forces in the crankshaft (based on [1])

Temporary gas force P_g was determined based on the actual pressure acting on the piston crown. Changes in pressure were estimated based on data from the engine manufacturer and the corresponding indicator diagram [3]. In the current study, it was limited to the section of the indicator diagram near the top dead centre, where the value of the gas forces is much higher than the inertial force. Moreover, it was assumed to omit the gas forces outside the considered section of the diagram.

Figure 4 presents a summary graph taking into account all loads acting on the considered system –gas force P_g, inertial force B and normal force N.

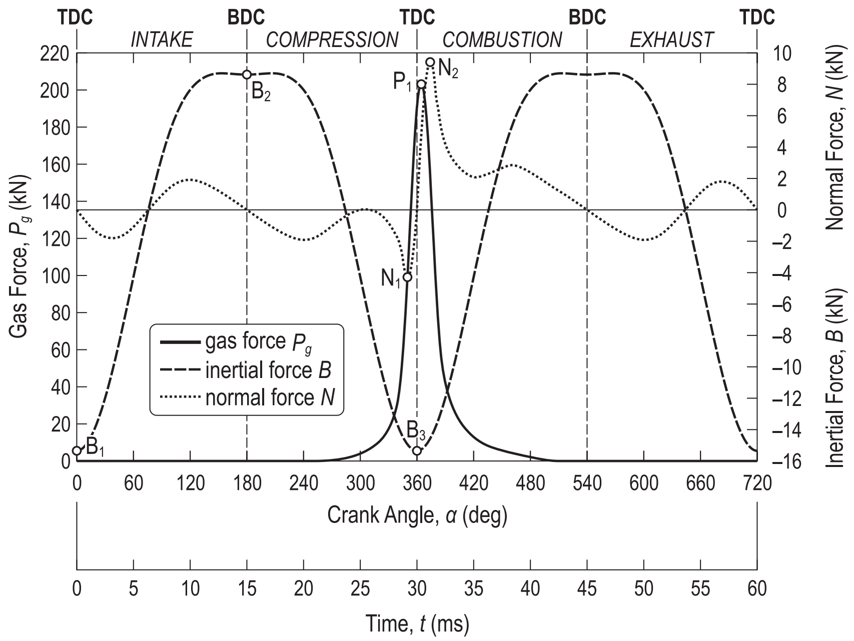

Fig. 4. Changes in the gas, inertial and normal forces for the engine speed of 2000 rpm

3. Development of geometrical and FE model

A geometrical model of the piston – piston rings – cylinder system was developed on the basis of manufacturer's documentation and measurement of the actual object. Ovalization of the piston was taken into account in the study. The 3D solid model was generated using AutoCAD software. A symmetry for considered problem was assumed. Vertical *yz* plane symmetry, perpendicular to the axis of the piston pin was defined. It allowed developing only a half model of the system, depicted in Fig. 5a.

A finite element model of the system (Fig. 5b) was developed using Altair HyperMesh software. 4-node *tetra4* elements were applied for the piston FE model, whereas the 8-node *hex8* elements were used for other components. The total number of finite elements and nodes was equal to 101,864 and 29,104, respectively.

The assumed symmetry of the system was carried out using the appropriate contact option. A property of a rigid body was declared for *yz* symmetry plane. Using this type of contact option, in the case of thermo-mechanical or thermal analysis, guarantees no flow of the heat through the surface (plane) of symmetry. Contact between respective components of the FE model including mechanical properties was defined using contact tables. An average value of the friction coefficient equal to 0.05 was determined based on literature data [7–10].

Mechanical boundary conditions resulting from the construction of the cylinder liner and its attachment to the engine block were applied by fixing of appropriate nodes of the model, corresponded to places of the actual attachment. Kinematic boundary condition – vertical displacement of the piston – was defined according to the methodology described in the previous chapter. Vertical displacement was assigned to the additional node located in the axis of the pin rod. This node was connected to the nodes located on the surface of the pin hole with a one-dimensional elements called *NODAL TIES*. The inertial force was applied using a *GRAVITY LOAD* option. The gas force load was implemented using a *FACE LOAD* option. The load was distributed evenly on the outer faces of solid elements corresponding to the piston crown. The influence of the pressure on the inner surface of the cylinder liner was not taken into consideration. A *LOAD POINT* option was used in order to load the piston – and cylinder liner consequently – by normal force. A force vector was attached to the additional node, located in the axis of the pin rod. A magnitude of the vector was time-changing a according to the curve depicted in Fig. 4.

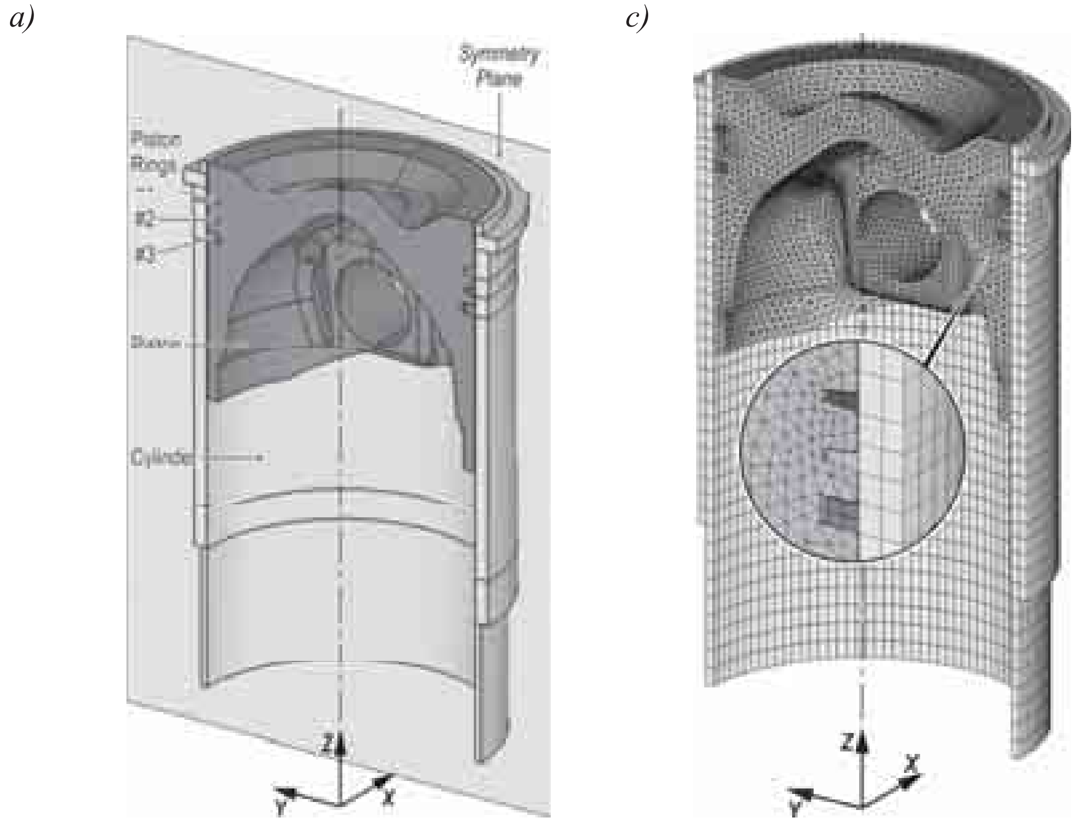

Fig. 5. Geometrical (a) and the FE (b) model of the piston – piston rings – cylinder system

Thermal boundary conditions applied in the FE model describe exactly the thermal load of the piston taking into account heat transfer between the piston walls, piston rings and the cylinder liner. A steady state of thermal loads was assumed for each component of the considered system. A methodology described in [2–4, 11] was used in order to determine thermal loads. The adopted thermal boundary conditions were presented in Fig. 6.

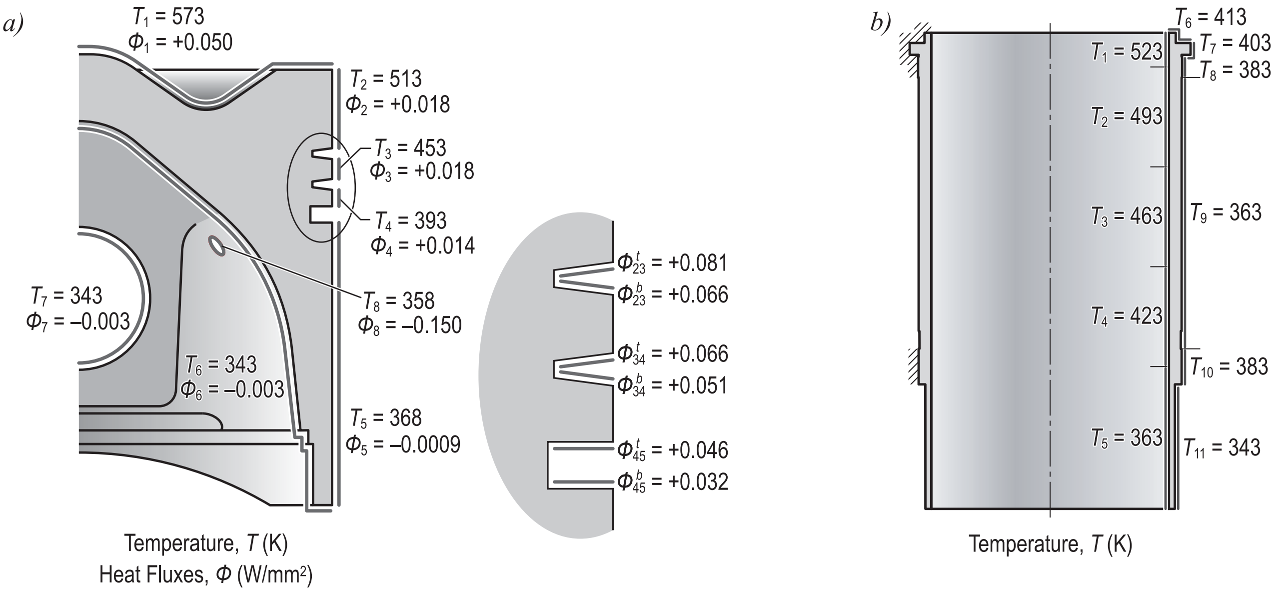

Fig. 6. Average values of thermal boundary conditions for S12U engine piston (a) and the cylinder liner (b) according to [2–4, 11]

Material properties for each component of the system were provided in Tab. 2. Diagrams of changes of the coefficient of thermal expansion for both PA12 aluminum alloy and the new composite material, were depicted in Fig. 7.

Tab. 2. Average material properties for the piston, piston rings and cylinder liner [2, 4, 6, 12, 13]. Provided values in FEA units

Parameter	Piston (PA12 Aluminum Alloy)	Piston Rings and Cylinder (Cast Iron)
Young's modulus, E (MPa)	69 000	152 000
Poisson's ratio, ν (–)	0.330	0.211
Density, ρ (Mg/mm^3)	$2.710 \cdot 10^{-9}$	$7.150 \cdot 10^{-9}$
Thermal conductivity, κ (mW·mm^{-1}·K^{-1})	171	47
Specific heat, c (mJ·Mg^{-1}·K^{-1})	$890 \cdot 10^6$	$460 \cdot 10^6$
Coefficient of thermal expansion, α (K^{-1})	see Fig. 7	$10.5 \cdot 10^{-6}$

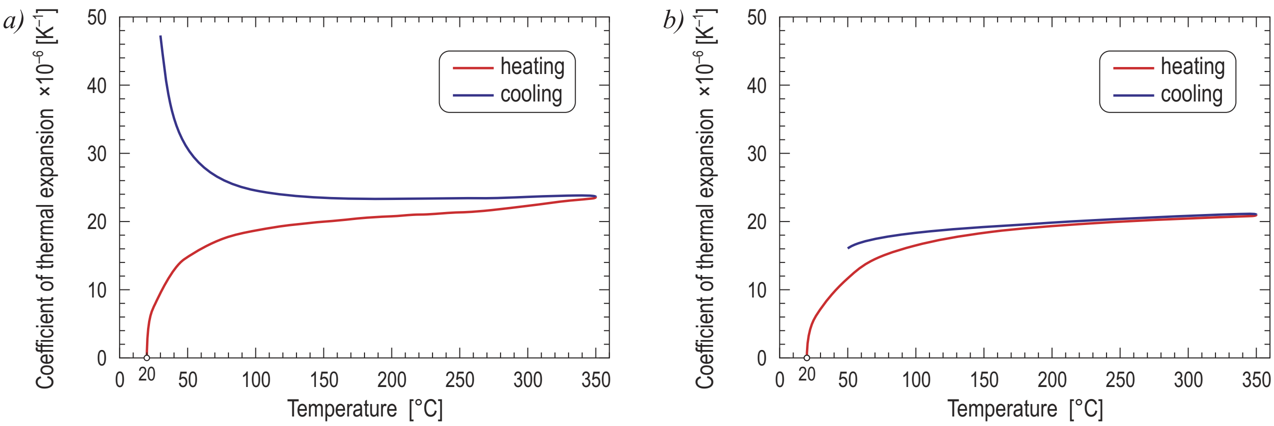

Fig. 7. Changes in coefficient of thermal expansion for PA12 aluminum alloy (a) and the new composite material (b)

4. Results of thermomechanical FE analysis

Selected results of the thermomechanical FE analysis in the form of contour bands for two types of the piston material – PA12 aluminum alloy and the new composite material with low hysteresis – were compared and presented in Fig. 8 and 9. 20-times scale factor for the deformed model was used in order to better visualize the deformation.

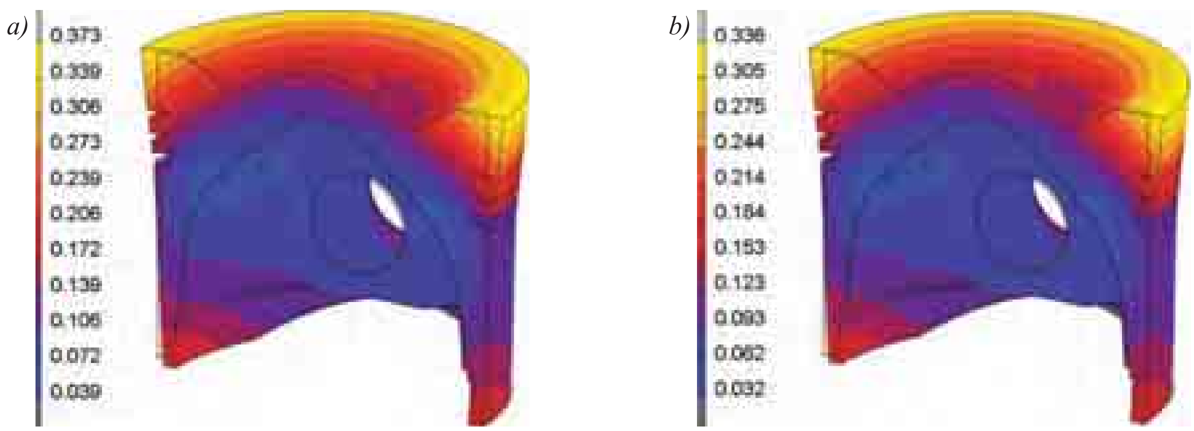

Fig. 8. Contour bands of displacement [mm] for the piston made of PA12 aluminum alloy (a) and the new composite material (b) under thermal load

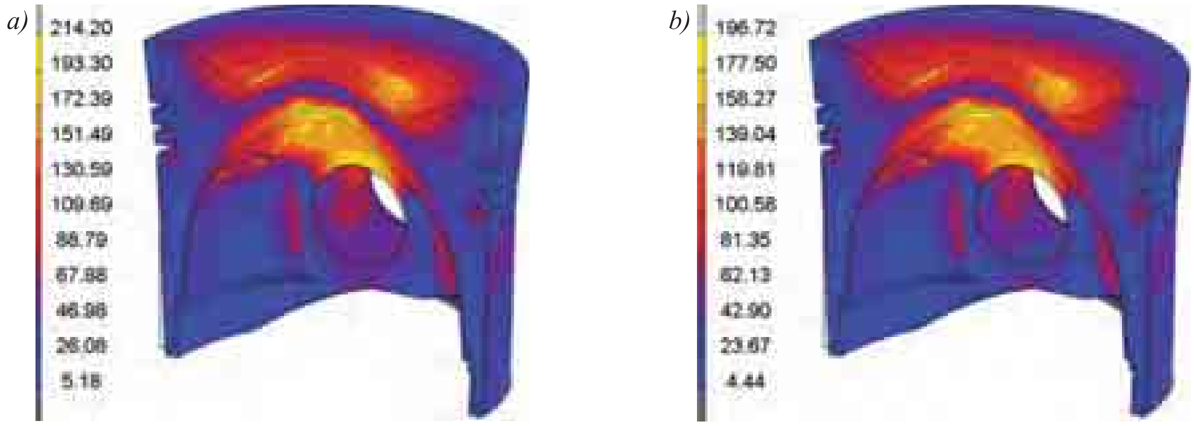

Fig. 9. Contour bands of equivalent stress [MPa] for the piston made of PA12 aluminum alloy (a) and the new composite material (b) under thermal load

The displacements registered for the composite piston reaches values of about 8–13% lower than in the case of a standard piston made of PA12 aluminum alloy. Deformations caused by heating of the piston led to the increase of equivalent stress values up to 197 MPa for the composite piston, which is approximately 7–10% lower than for the standard piston.

Figure 10 and 11 shows the displacement and the equivalent stress contours respectively, for the composite piston – piston rings – cylinder system at the position just behind the top dead centre (point P_1 in Fig. 4). At this point, the piston crown is under the maximum working pressure. Furthermore, the piston is loaded by inertial and normal force.

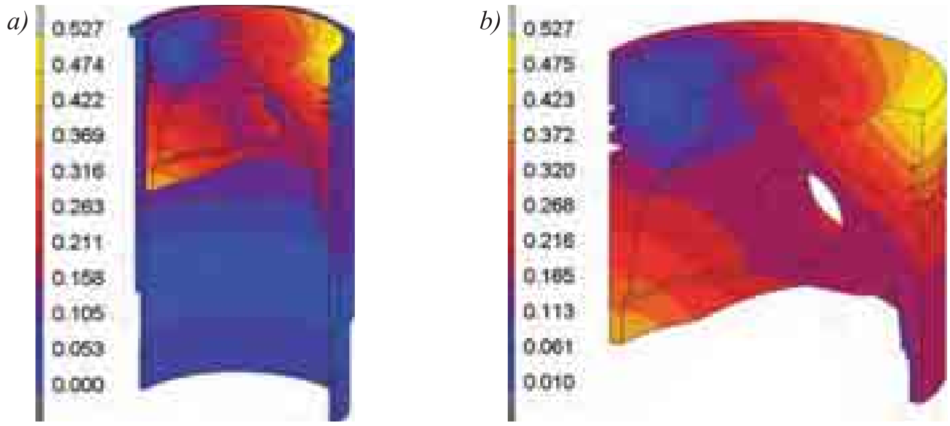

Fig. 10. Contour bands of displacement [mm] for the piston – piston rings – cylinder system (a) and for the piston (b) made of the new composite material at the position just behind the top dead centre (TDC)

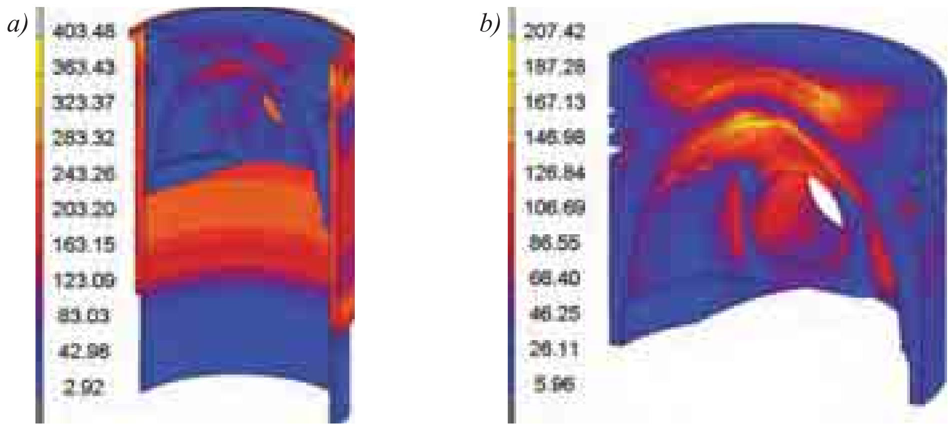

Fig. 11. Contour bands of equivalent stress [MPa] for the piston – piston rings – cylinder system (a) and for the piston (b) made of the new composite material at the position just behind the top dead centre (TDC)

7. Conclusions

The compression-ignition engine as a case study of a methodology of the numerical modelling and simulation of the piston – piston rings – cylinder was presented in the paper. Two types of the piston material were compared – the actual one PA12 aluminum alloy and the new composite material with low hysteresis. The results of comparative analysis demonstrate much more favourable properties of the piston made of the composite material. The obtained deformations and values of stresses are about 10% lower than for the actual piston. The thermal load has a significant influence on the state of stress in the piston. An effect of inertial, gas, and normal forces is slight in comparison to the thermal one.

Acknowledgements

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