

# Analysis of thermal stress of the piston during non-stationary heat flow in a turbocharged Diesel engine

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**Abstract.** In the paper, numeric calculations of thermal stresses of the piston in a turbocharged Diesel engine in the initial phase of its work were carried out based on experimental studies and the data resulting from them. The calculations were made using a geometrical model of the piston in a five-cylinder turbocharged Diesel engine with a capacity of about 2300 cm<sup>3</sup>, with a direct fuel injection to the combustion chamber and a power rating of 85 kW. In order to determine the thermal stress, application of own mathematical models of the heat flow in characteristic surfaces of the piston was required to show real processes occurring on the surface of the analysed component. The calculations were performed using a Geostar COSMOS/M program module. A three-dimensional geometric model of the piston was created in this program based on a real component, in order to enable the calculations and analysis of thermal stresses during non-stationary heat flow. Modelling of the thermal stresses of the piston for the engine speed  $n=4250 \text{ min}^{-1}$  and engine load  $\lambda=1.69$  was carried out.

## 1. Introduction

At present, still increasing demands both in terms of exhaust emissions and fuel economy, as well as the possibility to obtain high unit power are made in relation to modern diesel engines. Compliance with these requirements is possible, among others, due to a better understanding of the phenomena occurring inside the internal combustion engine, as well as development of various models for their subsequent multi parameter evaluation. Increasing engine power is accompanied by – apart from an increase in the mechanical loads – also an increase in thermal loads. These loads have a significant influence on such features of the engine as the exhaust blow through to the crankcase, the engine oil wear or the level of pollutants emitted by the engine to the atmosphere. Particularly a piston experiencing deformations variable in the time is exposed to these effects. Already at the stage of preliminary design work, knowledge on the thermal loads the piston will be subjected to is required. Such information can be obtained by suitable modelling of the temperature distribution in the studied cross-sections of the piston's characteristic surfaces. Design works and model tests are very expensive and prolonged. However, application of mathematical models and suitable computer techniques allows for determining the temperature distribution for different materials, sizes and shapes of the piston [12,13,14,15]. Subsequent analysis of this information helps to formulate initial design assumptions, as well as to modernize existing solutions for operation of the engine at various speeds and loads. Very good results in studies on thermal stresses of the piston are provided by application of the finite element method. However, in order to carry out any numerical calculations, some assumptions and use of empirical methods of data gathering for each of the analysed operation conditions of the engine are needed.



## 2. Analysis of the influence of selected parameters on the thermal loads of the piston

Thermal loads of components of a combustion chamber engine are the main factor limiting the performance and durability of Diesel engines [1]. The most loaded elements are the piston head and the surface of the engine cylinder head. Exceeding the permitted temperature of the piston causes various types of material losses and leads to an excessive increase in the diameter of the piston, resulting in its galling (Figure 1). In turn, temperature fluctuations may cause formation of cracks, leading to development of leakages and even preventing further work of the engine.



**Figure 1.** Traces of galling of the guide surface of a piston.

The heat effect during the engine operation causes reversible and irreversible dimensional changes in the piston. This is related closely with the suitable shape of the piston which determines the value of the clearance of the piston-cylinder assembly. The clearance of the piston in the cylinder has a significant influence on the tightness of the combustion chamber and thus on the size of the charge losses. This is particularly important for Diesel engines, where the loss of charge during compression reduces the real excess air ratio in the engine combustion chamber. Moreover, an improper clearance of the piston-cylinder assembly also has a negative impact on oil consumption, the capability to start the engine, and the noise. The piston temperature is significantly affected by the proper position of engine rings. From 60 to 80% of the heat acquired by the piston from the working medium is returned through the rings into the cylinder. Requirements for the location of the rings are incompatible. In order to reduce the thermal loads of the rings, they should be located as far away from the piston head as possible, and in order to reduce the thermal loads of the piston, they must be placed possibly high. For all engine speeds and loads, a distinct reduction in the piston temperature is achieved in all of its points by reducing the distance between the ring and the piston head. Changing the number and height of rings also causes a change in the piston temperature due to changes in the surface areas via which the heat exchange occurs.

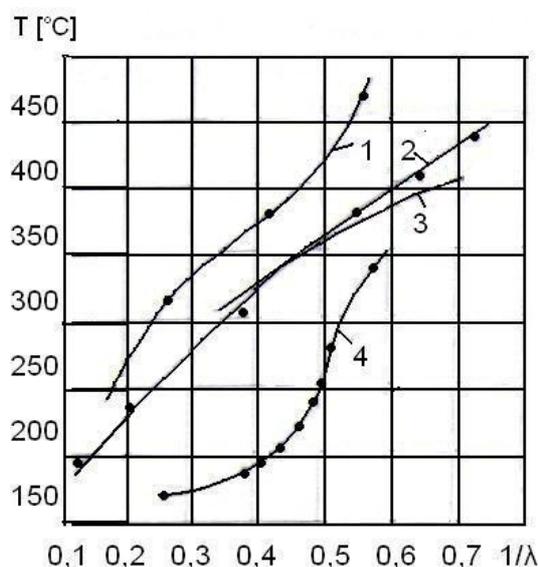
Apart from the design conditions, also the service conditions of an engine affect the thermal loads of pistons [2,3]. The conditions of the engine operation include a change in the effective pressure. Assuming that mechanical efficiency is constant, an increase in the average effective pressure is equivalent to a higher heat emission, which leads to an increase in the piston temperature. The influence of the increase in the average effective pressure on unit heat flux for the parameter characterizing the heat load of the cylinder result from the following dependence:

$$K_c = bc_m^{0.56} \frac{P_e g_e}{D^{0.5} (\eta_v P_d)^{0.435}} \frac{T_d}{T_0} \quad (1)$$

where:

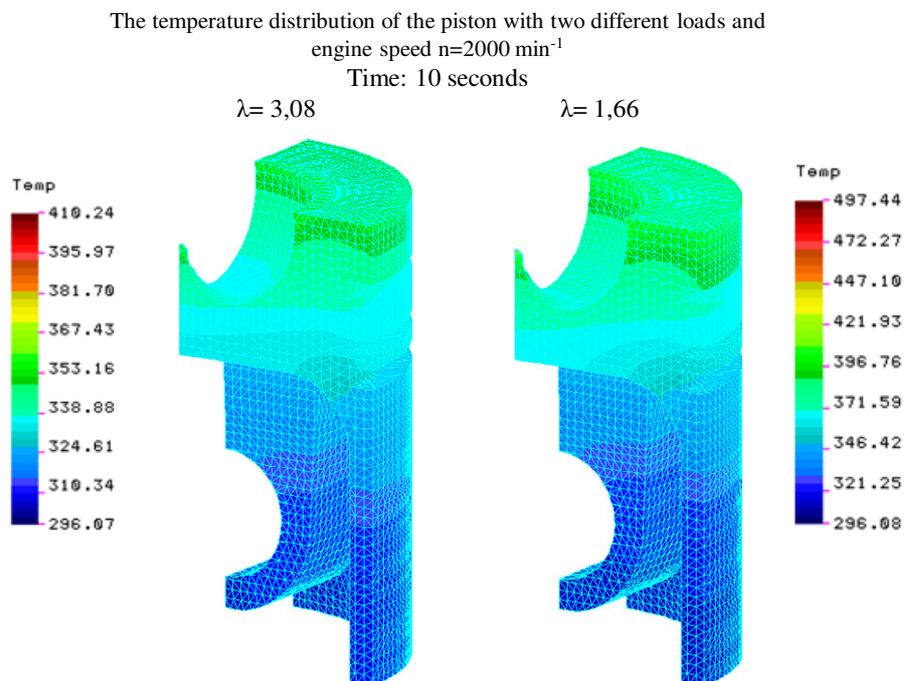
- b - coefficient of the amount of strokes (for four stroke  $b=2$ )
- $\eta_{\square}$  - fill coefficient
- $c_m$  - average velocity of the piston [m/s]
- D - diameter of the cylinder [dm]
- $p_d$  - pressure at the inlet valve [MPa]
- $T_d$  - temperature at the inlet valve [K]
- $p_e$  - average effective pressure [MPa]
- $g_e$  - actual fuel consumption [g/kWh]
- $T_0$  - ambient temperature ( $T_0=293$  K)

The increase in the average effective pressure of a Diesel engine is achieved by injection of a higher dose of fuel per working cycle into a cylinder. Thus, a higher amount of heat is transferred to the piston while reducing the excess air ratio. The lower the excess air ratio  $\lambda$ , the larger relative amount of heat may be released from the fuel in the cylinder. The temperature changes are analogous to the changes in the engine power. Both the mixture enrichment and impoverishment cause a drop in temperature of the piston. With a too rich mixture, a large part of the fuel is not burned. A lean mixture burns slowly, reducing the combustion temperature despite the fact that as a result of chronic combustion, the temperature of the exhaust gas increases. In a Diesel engine, an increase in the engine load leads to a decrease in the excess air ratio  $\lambda$  in the result of supplying more fuel. It leads to an increase in the amount of heat released in the engine's combustion chamber and in the temperature of the elements surrounding it, including the piston (Figure 2) [6].

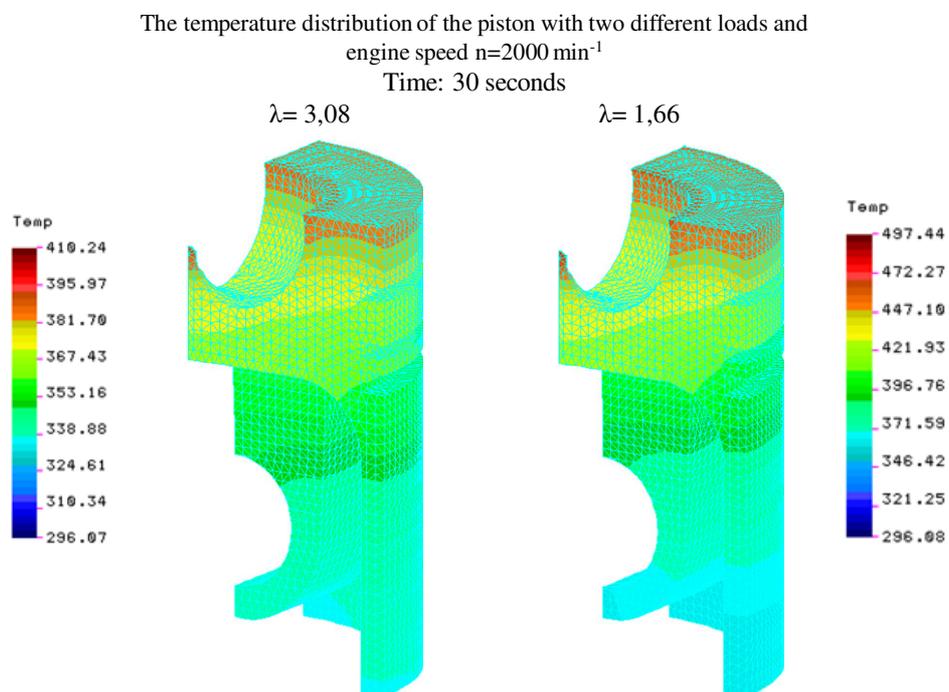


**Figure 2.** Temperature dependence of the piston vs. the excess air ratio for a self-ignition engine: 1 - engine with a pre-chamber; 2, 3 and 4 - direct injection engines.

For comparison, Figures 3 and 4 present modelled piston temperature distributions in a turbocharged Diesel engine with a capacity of about 2300 cm<sup>3</sup> with a direct fuel injection to the combustion chamber and engine power of 85 kW, for two loads corresponding to  $\lambda = 3.08$  and  $\lambda = 1.66$  at the engine speed of 2000 min<sup>-1</sup> after 10 and 30 seconds of its work from the start up. Further information about boundary conditions and modelling of the temperature distribution in the piston can be found in the literature [8,9].

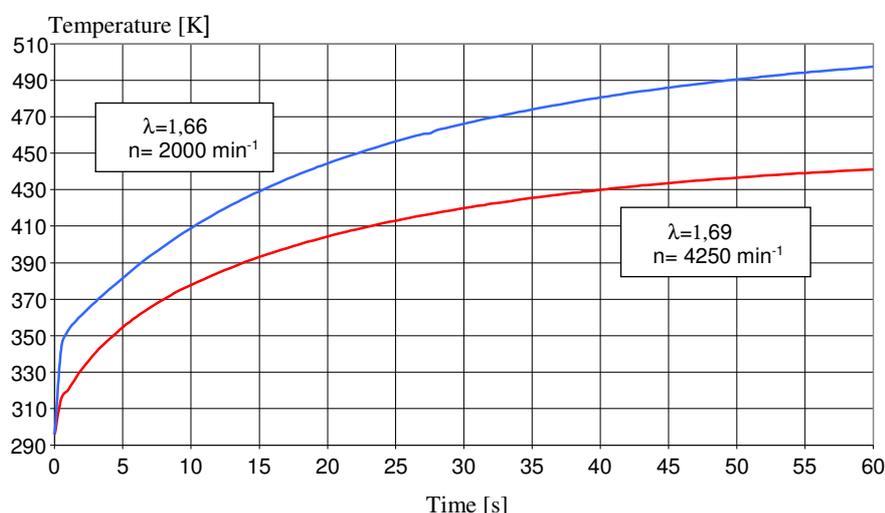


**Figure 3.** The temperature distribution in the piston for two loads of a turbocharged Diesel engine after 10 seconds.



**Figure 4.** The temperature distribution in the piston for two loads of a turbocharged Diesel engine after 30 seconds.

Both the calculations and the research show that the temperature distribution as a function of time has a similar course for both loads. However, for a lower load at the same rotational speed of the engine, the temperature values are lower. Another parameter affecting the thermal load on the piston is the engine speed. The influence of this parameter is quite complicated. On the one hand, an increase in the engine speed is equivalent to an increase in the frequency of combustion in the cylinder, increasing the amount of the heat in the combustion chamber. On the other hand, it affects the change in the engine filling efficiency, and thus the course of the combustion process itself. In a Diesel engine, the amount of fuel being provided into the cylinder does not depend on the engine filling efficiency with air, because for all rotational speeds, the engine sucks in a different but maximum value of the air mass. The amount of the injected fuel depends on the curve of dosing characteristic of the injection pump and on the excess air ratio which depends both on the amount of intake air as well as injected fuel. Figure 5 shows the course of the calculated maximum values of the piston head surface temperature during a 60-second operation of a turbocharged Diesel engine, calculated counted from the engine start for its two rotational speeds  $n = 2000 \text{ min}^{-1}$  and  $n = 4250 \text{ min}^{-1}$ .



**Figure 5.** The course of maximum temperature on the surface of the piston head for two engine speeds.

Based on the calculations, it was found that an almost twofold increase in the rotational speed of a turbocharged Diesel engine for the same load causes lower thermal loads of the piston head [10]. This is due to a higher rate of heat exchange between the piston and its environment, in comparison with the amount of heat generated in the combustion chamber of the engine.

### 3. Modelling of thermal stresses of an engine piston

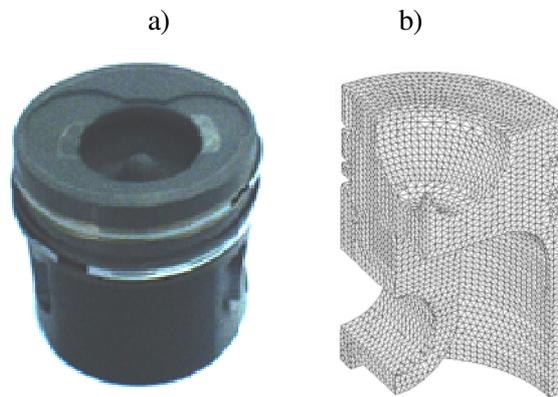
After starting the engine, the piston heats up until it reaches a state of equilibrium, which results from the balance between the heat taken from hot gases in the combustion chamber and transformed into useful work, and the heat transferred to the environment by, among others, the coolant and the combustion gases. Determination of thermal stresses of the piston by means of modelling required to assume equations and mathematical expressions for the calculations, describing the process of heat exchange in such a way in order for the model to reflect the actual processes occurring on the characteristic surfaces. This model was created for the piston on the basis of the differential equation of the heat flow in solids.

$$\frac{\partial T}{\partial t} = a \nabla^2 T + \frac{1}{c_p \rho} \frac{\partial \lambda}{\partial T} \left[ \left( \frac{\partial T}{\partial x} \right)^2 + \left( \frac{\partial T}{\partial y} \right)^2 + \left( \frac{\partial T}{\partial z} \right)^2 \right] + \frac{q_v}{\rho c_p} \quad (2)$$

where:

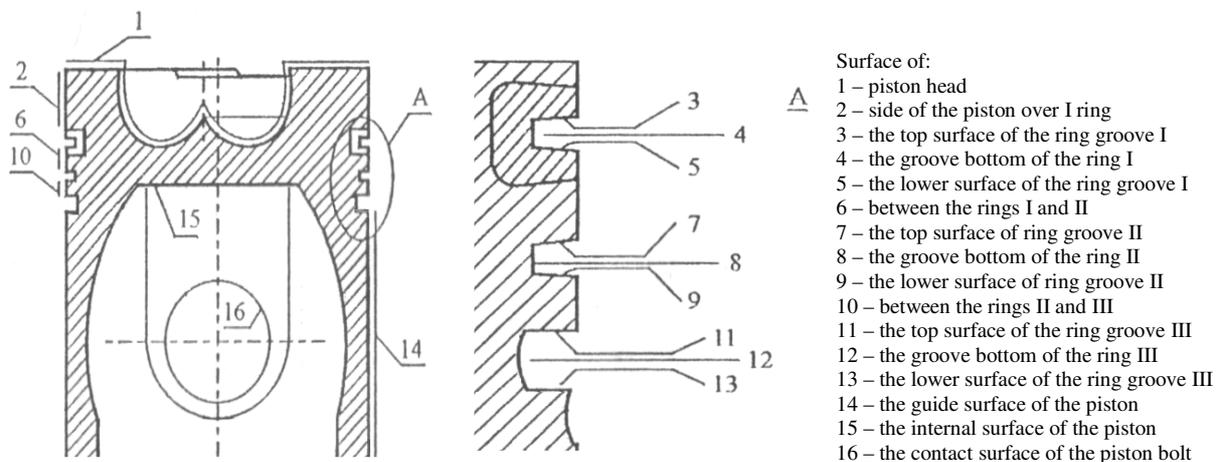
- a - coefficient of temperature compensation
- $c_p$  - specific heat capacity at constant pressure [J/kg K]
- $\rho$  - density [kg/m<sup>3</sup>]
- T - temperature [K]
- $q_v$  - volumetric efficiency of the internal heat source [W/m<sup>3</sup>]
- $\lambda$  - thermal conductivity [W/mK]

The numerical calculations of the stresses were performed by means of the COSMOS/M program based on the knowledge of the temperature distribution for assumed operating conditions of the studied engine. In the program an actual three-dimensional discrete geometric model of the piston was created based on the real component (Figure 6) [7].



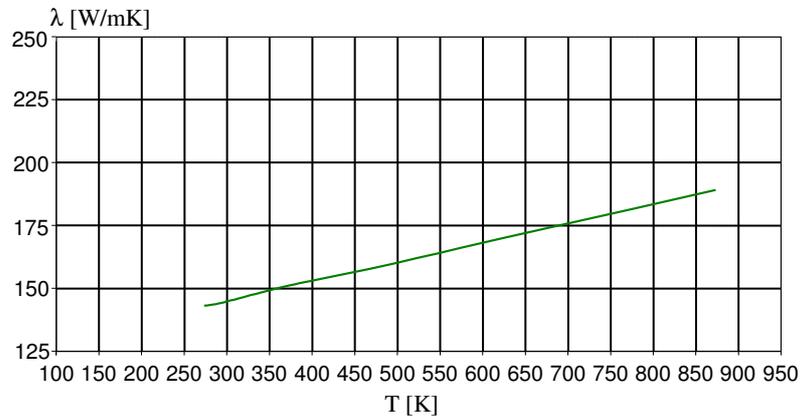
**Figure 6.** Geometric model of the piston: a) real component b) discrete model.

In the model, 16 characteristic the surfaces of the piston were distinguished, for which the temperature distributions and specific values of type III boundary conditions (Fourier conditions) were determined. These conditions determine the temperature of the medium surrounding the piston and the heat transfer coefficients of the characteristic surfaces. These surfaces are shown in Figure 7 [4].



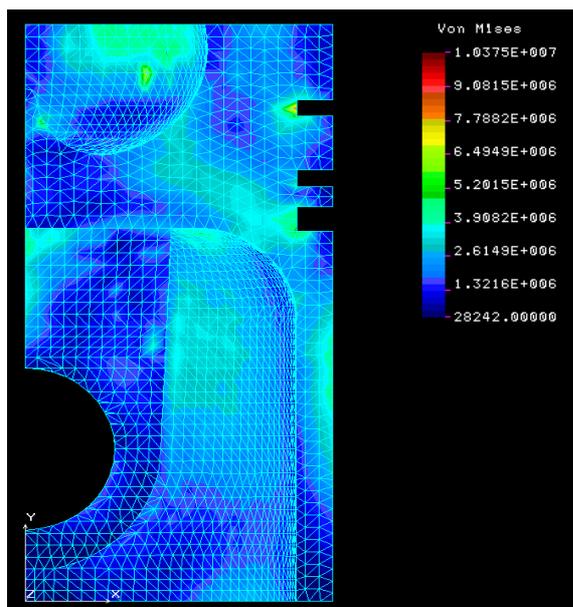
**Figure 7.** Characteristic of surfaces of the piston.

For numerical calculations of the thermal stresses of the piston in the state of a non-stationary heat flow, the material data of the AK 12 silumin were used. Due to the possibility to use the temperature curves in the COSMOS/M program, variable value of the thermal conductivity coefficient of the material as a function of temperature was accepted (fig. 8), among others.

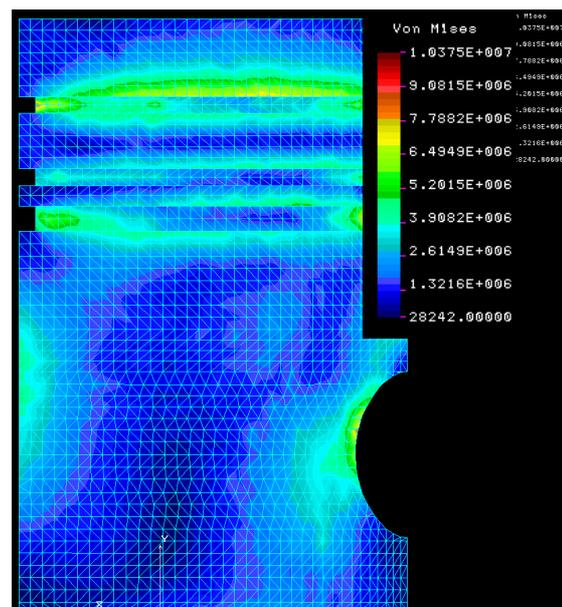


**Figure 8.** The graph of the thermal conductivity coefficient of the AK 12 alloy as a function of temperature

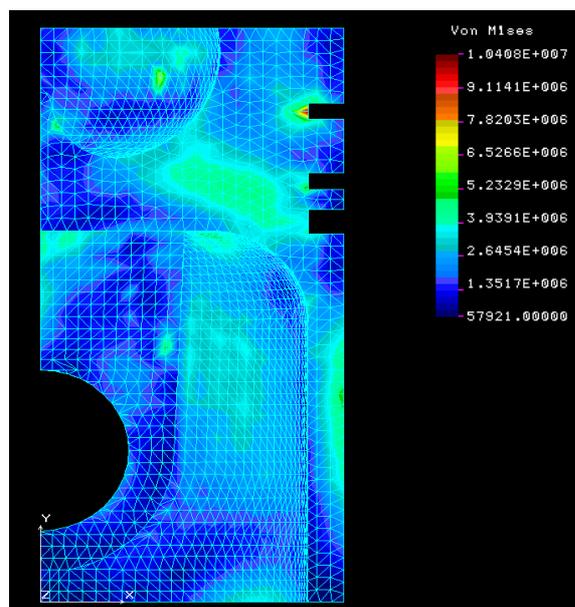
For individual piston surfaces, conditions of heat exchange equivalent to those in the combustion chamber for each cycle of the engine were assumed in the calculations. Based on the recorded indicator diagrams and the calculated total heat transfer coefficient, the temperature of the working medium surrounding the combustion chamber and the values of the heat transfer coefficient for the engine load of  $\lambda = 1.69$  and speed of  $n = 4250 \text{ min}^{-1}$  were determined [5]. Figures 9 and 10 show the exemplary values of thermal stresses of the piston during a non-stationary heat flow, corresponding to 10 and 20 seconds of engine operation measured from its start-up.



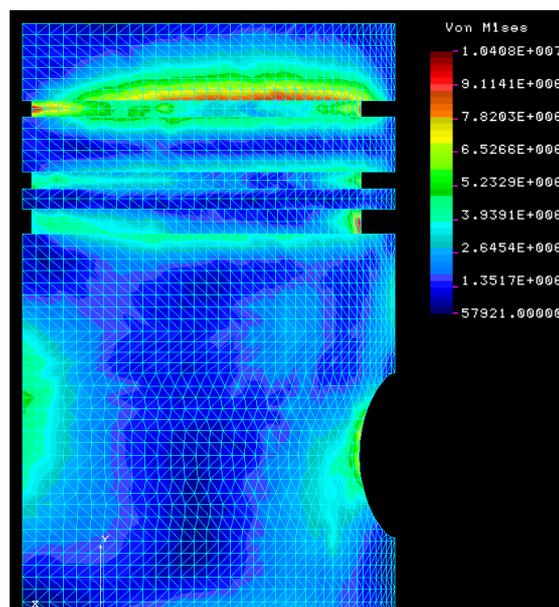
**Figure 9.** Thermal stresses of the piston after 10 seconds - front view.



**Figure 9.** Thermal stresses of the piston after 10 seconds - back view.



**Figure 10.** Thermal stresses of the piston after 20 seconds - front view.



**Figure 10.** Thermal stresses of the piston after 20 seconds - back view.

The calculations carried out indicate that thermal stresses of the piston were increased during the warm-up of the engine. After 10 seconds of the engine operation, the highest thermal stresses occurred only on a small surface of the bottom groove of the third ring. However, after 20 seconds, the highest thermal stresses of the piston could be observed on the upper surface of the groove of the first ring and they included also the bottom side surface of the piston above this ring.

#### 4. Conclusions

Based on the preliminary results of the calculations, it can be concluded that maximum values of thermal stresses of the piston are found mainly in the ring portion and on the surface of the piston head. On the other hand, the lowest values are found in the guide part. The obtained maximum values of thermal stresses for the engine speed of  $n = 4250 \text{ min}^{-1}$  and  $\lambda = 1.69$  did not exceed 12 MPa. In the state of a non-stationary heat flow, distributions of thermal stresses on the individual surfaces of the piston different in the time were obtained. According to the Authors, they represent, together with mechanical stresses, important factors in the design and subsequent operation of the piston for a specified engine kind and type. However, the calculations should be verified experimentally, to enable obtaining results of numerical calculations which would correspond to the real thermal stresses in the engine during warming up in the future [11].

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