# Thermal-Stress State of the Piston During Transient Diesel Operation, Synthesis of the Piston Profile

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**Abstract.** This paper deals with simulating the thermal and stress states of the piston and cylinder of a two-stroke diesel engine type D100 for a locomotive using two different methods. The methods are based on the use of various models of piston-cylinder arrangement: one of them uses assembly of the piston-cylinder arrangement and includes all components, but the other uses two models separately: the cylinder and the piston and rings assembly. It was shown that for both methods, the piston temperature fields differ slightly both in zone of the combustion chamber and on the inner surface, and the maximum difference was 8 K in the zone of the first piston ring, but the difference in the cylinder temperature fields is significant, especially in the middle section where the difference reaches 35 K. Transient thermal-stress states of the piston were determined for three programs of engine starting from the cold state (with various initial temperatures equal to 20 °C; 0 °C; −20 °C) and heating it up to its maximum operating mode, and for one program with cooling from maximum mode to idling, every time with stepped loading. Warming up of oil and antifreeze were taken into account. It is shown that temperatures and stresses reach their peaks and then drop in some zones of the piston. The stress reaches a maximum level of 380 MPa at the center of the piston surface from the combustion chamber side when heated with an initial temperature −20 °C, which is two times more than the stress at steady-state maximum operating mode (192 MPa). The profile of the piston side surface was synthesized taking into account the heating rate. The piston design with different materials of the piston head – VCh60-2, 25H2G2FL, 12DH1MFL – was proposed for D49 and D80 engines. The mean temperature of the piston hottest zone drops by about 40 K in proposed design. The stress also decreased.

**Keywords:** Diesel · Piston · Cylinder · Transient mode · Heat transfer coefficient · Temperature · Stress

## 1 Introduction

The resource requirements for the pistons of modern internal combustion engines are constantly increasing. Two factors most strongly affect the durability of the piston are the load and the wear. In the previous works [1, 2] the authors of this study proposed an


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alternative piston design for the D100 engine and the technique of calculating the boundary conditions for modeling the stress state of the piston. This work continues the research started. The study analyzed the thermal and stress state of the piston-cylinder arrangement (PCA) parts during accelerating and unloading engine. According to the results of studies [3], the stress in the piston under transient conditions can reaches a value of 2…3 times more in comparison with the steady-state maximum mode. This has negative affect on piston strength and lifetime. The issues of working capacity and lifetime of the piston are solved in different ways, for example, by controlling the fuel supply, but first of all, by choosing (optimizing) the design and choosing the material. Important design parameters include the profile of the side surface. A correctly created profile should ensure the absence of contact between very heated piston head and sleeve at all engine operating modes [4–6] as well as the optimal contact surface of the piston-sleeve friction pair along the skirt [7–9]. On the other hand, it is known that in ICE the temperature of the combustion chamber (CC) surface affects the indicator parameters, the temperature of the working fluid and the heat transfer coefficient from the working fluid to the walls of the CC. At different temperatures of the CC surface, the equivalent boundary conditions (BC) will be different.

## 2 Research Goals

1. To develop the algorithm for calculating the thermal boundary conditions on the surfaces of the piston and cylinder during transient diesel operation.
2. To determine temperature, strain and stress in piston of a D100 type two-stroke engine in transient modes.
3. To optimize the profile of the piston side surface, taking into account its operation in pair with the cylinder liner.
4. To propose an alternative piston design for D49 and D80 diesel engines operated at Ukrzaliznytsia.

## 3 Analysis of the Transient Thermal and Stress States of the Piston. Optimization of the Piston Side Surface for the Engine Type D100

### 3.1 Statement of the Problem

The four transients were considered, three of which – from cold engine with initial component temperatures of −20 °C, 0 °C and +20 °C to maximum operating mode (hereinafter – variants 1, 2, 3), and variant 4 – from maximum mode to idle. The type of loading is stepped. Figure 1 shows the piston geometric parameters of diesel engine type D100 at 20 °C, that determine the initial profile. Here is indicated: $R_p$ – the piston radius, mm; $h_p$ – the piston height, beginning from CC, mm; $\theta$ – the angular coordinate with beginning ($\theta = 0°$) from wrist pin axis, deg.

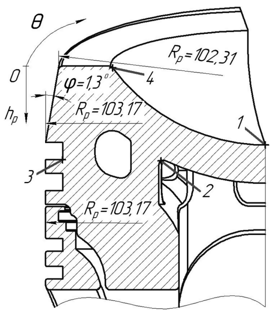


**Fig. 1.** The geometric parameters of the piston

Based on the researches carried-out for piston of a diesel type D100, the design of alternative piston for diesel engines of the D49 and D80 families was proposed and investigated.

## 3.2 Materials

All parts of piston-cylinder arrangement of D100 type engine are made of inoculated or high-strength cast iron with spherical graphite that has hardness HB = 207…255 [10]. The exception is the piston skirt, which is made of aluminum alloy AK4. For modeling the piston head, cylinder, rings and wrist pin, the VCh60-2 cast iron was chosen [11, 12], physical properties of which are presented in Table 1.

**Table 1.** The physical properties of VCh60-2 cast iron at different temperatures

| Temperature T, °C | Thermal conductivity λ, W/(m K) | Specific heat $C_p$, J/(kg K) | Thermal expansion coefficient α, 1/K·$10^{-6}$ | Elasticity modulus E, MPa |
|---|---|---|---|---|
| 20 | 54.5 | 390 | 10 | 180000 |
| 370 | 46.2 | 505 | 11.5 | 163650 |
| 537 | 42.4 | 523 | 12.25 | 152410 |
| 650 | 40.3 | 568 | 12.45 | 142500 |

The oil used for the 10D100 engine is M14V2. Its thermophysical properties – density $\rho$, specific heat $C_p$, thermal conductivity $\lambda$, kinematic viscosity $v$ – can be calculated using the following formulas:

$$\rho = \exp(6.818 - 0.000718 \cdot T), \mathrm{kg/m^3}; \tag{1}$$

$$C_p = \exp(7.475 + 0.002272 \cdot T),\ \mathrm{J/(kg\ K)}; \tag{2}$$

$$\lambda = 0.1427 \cdot \exp(-0.0009971 \cdot T),\ \mathrm{W/(m\ K)}; \tag{3}$$

$$\upsilon = 2.29042 \cdot T^{-2.60745},\ \mathrm{m^2/s} \tag{4}$$

As a coolant, the antifreeze «−65 °C» was chosen, specific heat of which is determined by formula (5)

$$C_p = 7.6106 \cdot T + 769,\ \mathrm{J/(kg\,K)}. \tag{5}$$

### 3.3 Algorithm for Calculating Thermal BC for Transient Engine Operating Modes

The problem of determining the heat exchange boundary conditions on the surfaces of diesel combustion chamber for transient operating modes is associated with following:

- significant influence of temperature on the properties of both the structural elements that limit volume of the combustion chamber and the cooling liquids (oil, antifreeze);
- time-varying heat transfer boundaries;
- time-varying parameters of gas in combustion chamber.

To correctly determine the thermal loads in the specified operating conditions, it is necessary to ensure the thermal balance of the system "gas – combustion chamber parts – coolants". Since the task is unsteady, the heat balance must be achieved at each calculated time, given the variable amount of heat generated and removed from the system.

**Heat Transferring to Oil and Cooling Liquid.** The change in the temperature of oil and antifreeze during the *i*-th time interval depends on the amount of heat $\Delta Q_i$ transferred during this interval and can be determined by the formula:

$$\Delta T_i = \Delta Q_i/(mC_i), \tag{6}$$

where, $m$ is the mass of oil or antifreeze;

$C_i$ is the specific heat of oil or antifreeze at $i$-th time interval.

Then the temperature at the next solving time interval will be:

$$T_{i+1} = T_i + \Delta T_i, \tag{7}$$

where, Ti is the temperature at the current i-th interval.

**Heat Transferring on the Surfaces of Combustion Chamber.** Due to the piston movement the inner surface of the cylinder changes as well as the state of the working fluid, therefore the following technique [13] was applied for calculating the BC for a multi-zone surface. The Fig. 2 shows the principal scheme of heat exchange process that occurs at $i$-th time moment on multi-zone boundary surface $A$ with time-varied BC – the local heat transfer coefficients $\alpha_{Aji}$ and local bulk temperatures $T_{Aji}$, and on single-zone surface $B$ with constant BC $\alpha_B$, $T_B$.

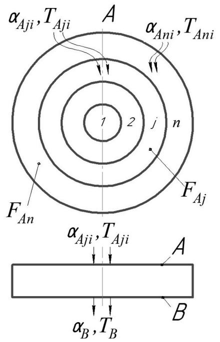


**Fig. 2.** The scheme of solving the BC

The surface $A$ with total area $F_A$ is divided on $n$ number of zones $A_j$ with areas $F_{Aj}$. The areas $F_A$ and $F_{Aj}$ are not constant and changed with time. The total amount of heat $Q_A$ transferred through surface $A$ equal to

$$Q_A = \sum_{j=1}^{n} \sum_{i=1}^{m} \alpha_{Aji} \cdot F_{Aji} \cdot \left(T_{Aji} - \overline{T}_{Aj}\right) \cdot \Delta t = \overline{\alpha}_A \cdot F_A \cdot \left(\overline{T}_{A.e} - \overline{T}_A\right) \cdot \tau, \qquad (8)$$

and the local amount of heat $Q_{Aj}$, transferred through $j$-th zone of surface $A$, equal to

$$Q_{Aj} = \sum_{i=1}^{m} \alpha_{Aji} \cdot F_{Aji} \cdot \left(T_{Aji} - \overline{T}_{Aj}\right) \cdot \Delta t = \overline{\alpha}_{Aj} \cdot F_{Aj} \cdot \left(\overline{T}_{Aj.e} - \overline{T}_{Aj}\right) \cdot \tau, \qquad (9)$$

where, $m$ is a number of time intervals $i$; $\tau$ is the total time of described process;

$F_{Aji}$ is the current area of the $j$-th zone on the surface $A$ at the time $i$;
$\overline{T}_A$, $\overline{T}_{Aj}$ is the mean temperature of the surface $A$ and $j$-th zone of surface $A$ during the time $\tau$;
$\overline{\alpha}_A$ and $\overline{\alpha}_{Aj}$ is the mean value of the heat transfer coefficient at surface $A$ and $j$-th zone of surface $A$ during the time $\tau$;
$\overline{T}_{A.e}$ and $\overline{T}_{Aj.e}$ is the equivalent bulk temperature of the ambient acts on surface $A$ and $j$-th zone of surface $A$ during time $\tau$;
$\Delta t$ is the time step.

Then the mean equivalent BC for surface $A$ and its zones can be determined by next equations:

$$\overline{\alpha}_A = \frac{Q_A}{\sum_{j=1}^{m} \sum_{i=1}^{n} F_{Aji} \cdot \left(T_{Aji} - \overline{T}_{Aj}\right) \cdot dt}; \qquad (10)$$

$$\overline{\alpha}_{Aj} = \frac{Q_{Aj}}{\sum_{i=1}^{n} F_{Aji} \cdot \left(T_{Aji} - \overline{T}_{Aj}\right) \cdot dt}; \qquad (11)$$

$$\overline{T}_{A \cdot e} = \frac{Q_A}{\overline{\alpha_A} \cdot F_A \cdot \tau} + \overline{T}_A; \quad (12)$$

$$\overline{T}_{Aj.e} = \frac{Q_{Aj}}{\overline{\alpha}_{Aj} \cdot F_{Aj} \cdot \tau} + \overline{T}_{Aj}. \quad (13)$$

**The Algorithm.** The special algorithm was developed to calculate the boundary conditions for heat exchange surfaces the schematic diagram of which is presented at Fig. 3.

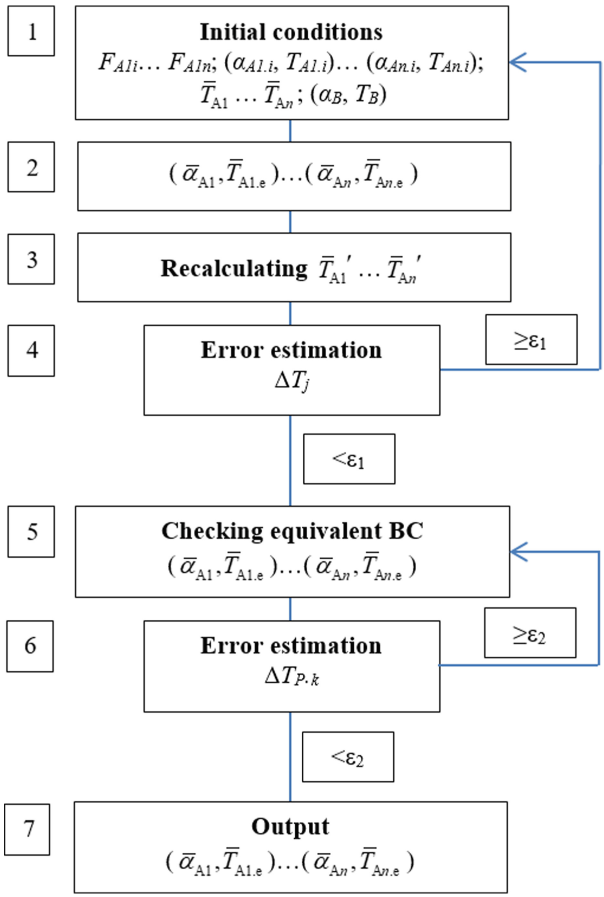


**Fig. 3.** Algorithm of determining an equivalent BC for A surface

The initiating of the algorithm is begins from defying areas $F_{A1i} \dots F_{A1n}$ of all $j$-th zones (from 1 to $n$) and the corresponding time-varied boundary conditions ($\alpha_{A1.i}$, $T_{A1.i}$) … ($\alpha_{An.i}$, $T_{An.i}$) at the current time moment $i$, also the mean surface temperatures

$\overline{T}_{A1} \ldots \overline{T}_{An}$ for all *j*-th zones and BC ($\alpha_B$, $T_B$) for surface *B* – block 1. The block 2 calculates the equivalent BC, $(\overline{\alpha}_{A1}, \overline{T}_{A1.e}) \ldots (\overline{\alpha}_{An}, \overline{T}_{An.e})$, using (9), (11), (13). The block 3 is associated with CAE module that provides thermal analysis of structure according to equivalent BC received from block 2, and then recalculates the surfaces mean temperatures $\overline{T}'_{A1} \ldots \overline{T}'_{An}$. The next is block 4 that compares the levels of the recalculated temperature $\overline{T}'_{Aj}$ with the initially defined $\overline{T}_{Aj}$ at each zone. If the case of temperature differences $\Delta T_j = \overline{T}_{Aj} - \overline{T}'_{Aj}$ exceeding the accuracy level, the algorithm redefines the initial data in block 1 and repeats. The blocks 5 and 6 are for checking the correctness of equivalent BC. The procedure is carried out in to steps. The equivalent BC $(\overline{\alpha}_{Aj}, \overline{T}_{Aj \cdot e})$ are used in steady-state thermal analysis of PCA to calculate the initial condition for further transient thermal analysis with time-varied BC ($\alpha_{Aj.i}$, $T_{Aj.i}$) corresponded to steady-state operating mode of the engine. The error estimated as temperature differences in local points *P.1…P.k* of the piston surface at start (initial state) and end (time $\tau$) of the transient analysis – $\Delta T_{P.k} = T_{P.k\ start} - T_{P.k\ end}$. The block 7 gives the final equivalent boundary conditions that ensure needed accuracy of thermal state for PCA components.

For the piston internal surface *B*, cooled by oil, the thermal loads are also variable over time. However, modeling the processes of heat transferring through this surface in CAE module, as for a region divided by a *n* number of zones with variable heat exchange boundaries, is extremely difficult due to the complex configuration of the piston crown. Therefore, it is recommended to complete the determination of equivalent BC on surface B after block 4.

This algorithm can be also used when the time-varied BC with variable heat exchange boundaries are defined on both the surface of combustion chamber and the surface of piston crown. In this case, the determining of BCs must be carried out in parallel for both surfaces with matching the results.

### 3.4 Simulating the Steady Operating Mode of Diesel Engine

This task is solved by high level CAE software.

Two models are investigated in the work: the first one is the assembly that consists of all parts of PCA (Fig. 4); the second one is presented by two separate sub-models – piston and rings assembly and the cylinder (Fig. 5). For the first model, the calculated surfaces are the piston surface from side of CC and the inner surface of the cylinder. For the second model, the calculated surfaces are the piston surface from side of CC (one surface), the rings surfaces from the side of cylinder (four surfaces) and the inner surface of the cylinder divided on 10 zones.

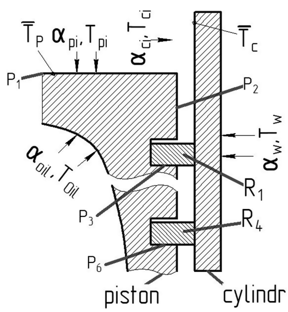


**Fig. 4.** The model 1

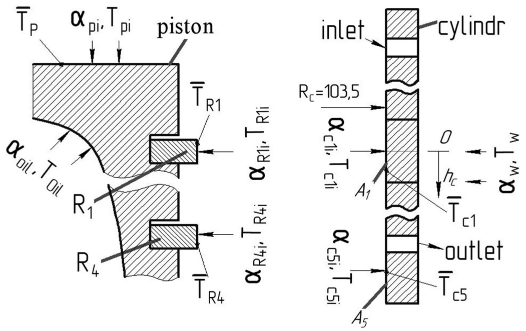


**Fig. 5.** The model 2

The time-varied BC in combustion chamber on the piston surfaces ($\alpha_{pi}$, $T_{pi}$), on the rings ($\alpha_{R1i}$, $T_{R1i}$) … ($\alpha_{R4i}$, $T_{R4i}$) and on the cylinder surface ($\alpha_{c1i}$, $T_{c1i}$) … ($\alpha_{c5i}$, $T_{c5i}$) are determined in result of preliminary workflow calculation [2].

According to [14] and based on workflow calculation [2] the mean emission of heat to the cooling system at maximum mode is defined as $Q_w$ = 1000 kW, and to the oil system – $Q_{oil}$ = 420 kW. The mass of the antifreeze $m_w$ = 1250 kg, and the oil mass $m_{oil}$ = 1250 kg.

The heat transfer coefficient from the cylinder wall to the antifreeze is 18000 W/(m$^2$ K) [15]. The heat transfer coefficient between the oil and the inner surface of the piston head is calculated as described in [13].

The thermal analysis results for the PCA components are presented in Table 2 and on the Figs. 6 and 7. From Table 2 and Fig. 6 it is seen, that the temperature fields on the piston head and the mean temperatures of the piston surface $\overline{T}_p$ from CC side differ slightly. The maximum difference is 8 K in the grove for 1-st ring (point $P_3$, Fig. 4) – 414 K in model 1 versus 422 K in model 2. The difference in temperatures of the piston skirt is less – only 4 K at point $P_5$ and 3 K at point $P_6$. However, the temperature field of the cylinder differ significantly (Fig. 7). In middle zone this difference reaches 35 K.

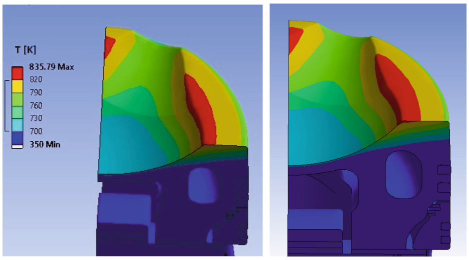


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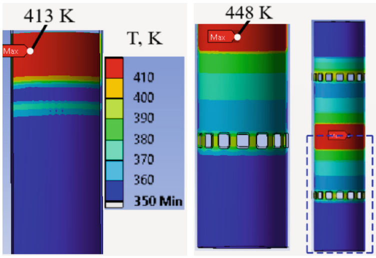
**Fig. 6.** The piston temperature field: (a) model 1; (b) model 2




a b

**Fig. 7.** The cylinder temperature field: (a) model 1; (b) model 2

The other difference between the models is next. Use of the model 2 it is necessary to determine and redefine the BCs for (n + m + 1) surfaces while, as for model 1 – only one surface. So, volume of work with model 2 is more. Summarizing the noted above, it is recommended to use model 1 for thermal stress analysis of the piston.

**Table 2.** The piston temperature in different points (Figs. 4 and 5)

| The temperature of piston, K | | |
|---|---|---|
| Point or zone number | Model 1 | Model 2 |
| $\overline{T}_p$ | 775 | 776 |
| $P_1$ | 710 | 712 |
| $P_2$ | 668 | 670 |
| $P_3$ | 414 | 422 |
| $P_4$ | 376 | 382 |
| $P_5$ | 366 | 370 |
| $P_6$ | 364 | 367 |

The amount of heat transferred at maximum mode to inner surface of the cylinder from the ring – is 634 J/cycle, from combustion products – is 6260 J/cycle. In view of small heat transferring through the rings (about 10% according to Table 2) and usability for analysis of heating rate the model 2 is chosen for the cylinder.

### 3.5 Analysis of the Transient Modes

Using the method described above, the equivalent time-dependent BC for the piston were obtained for heating variants 1…3, noted in (Sect. 3.1). The Table 3 shows results for variant 1, where: $Q_p$ is the amount of heat transferred to the piston per 1 cycle (1 revolution of the crankshaft); $\overline{\alpha}_p$, $\overline{T}_{pe}$ are the equivalent heat transfer coefficient and the mean temperature.

**Table 3.** Varying the equivalent boundary conditions at the piston surface during transient mode

| Mean temperature of the piston and amount of heat transferred to the piston | | | | |
|---|---|---|---|---|
| $\tau$, *sec* | $\overline{T}_p, K$ | $\overline{Q}_p, J$ | $\overline{\alpha}_p$, W/(m$^2$ K) | $\overline{T}_{\ni p}, K$ |
| 0 | 253 | 2700 | 1850 | 870 |
| 80 | 775 | 925 | 1890 | 980 |
| 150 | 815 | 771 | 1892 | 987 |
| 500 | 775 | 925 | 1890 | 980 |
| 1000 | 775 | 925 | 1890 | 980 |

Figures 8 and 9 show the graphs of temperature and stress varying in point P.1 (see Fig. 1) during heating up. The analysis is carried-out taking into account the acting pressure of 10 MPa and lateral force of 22800 N. Figures 10 and 11 show the loading variants 1, 2, 3, 4 according to (Sect. 3.1).

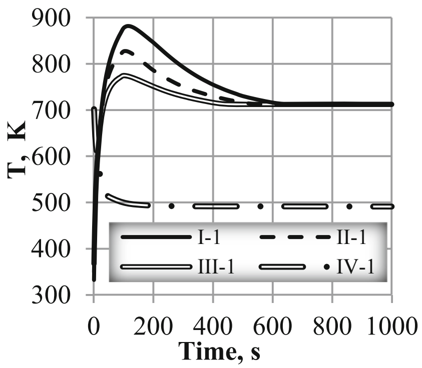



**Fig. 8.** Varying the piston temperature during heating up

The Fig. 9 shows that the maximum stress during heating up at two times more than in steady state mode. In variant I the maximum stress in point P.1–380 MPa (at

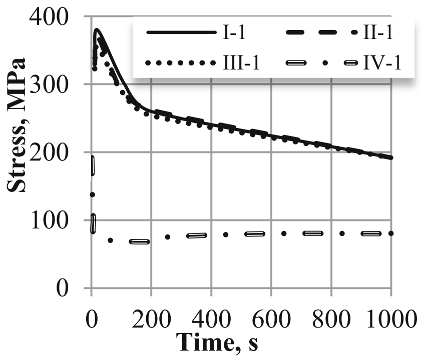



**Fig. 9.** Varying the stress in the piston during heating up

$\tau$ = 20 s) when in steady-state it is no more 192 MPa. Variant IV does not generates the high level of stress.

The additional analysis was carried out to examine the effect of transition time from initial conditions (cold engine) to maximum operating mode on stress owerswing. For this, the three additional variants *V*, *VI*, *VII*, were simulated with transition time from mode to mode equal to 50, 100 and 200 s respectively. The maximum stress in piston head drops (Fig. 10) from 380 MPa in variant *I* to 300 MPa in variant VII.

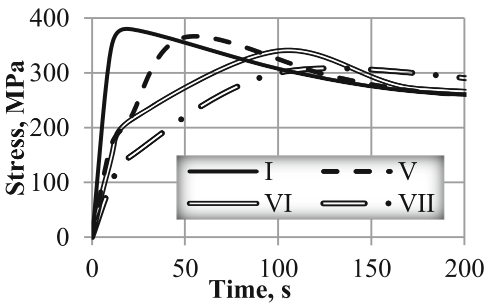



**Fig. 10.** Stress varying in piston head at different transition time

### 3.6 Synthesis of the Piston Profile

The modeling of thermal, stress and strain states under the heating by variants 1…3 allowed us to synthesize the profile of the piston outer surface and the cylinder inner surface. In each variant the piston-to-cylinder clearance is exanimated. Figure 11 shows the profiles of the piston and cylinder liner. It is shown that at $\tau$ = 200 s the points are appeared on the piston surface at distance $h_p$ = 34 mm. These points penetrate into cylinder at TDC position. It is necessary to decrease diameter in this cross-section and change in the surface inclination angle $\varphi$ of the piston head. After corresponding changes of the piston geometric parameters the dangerous zone is not

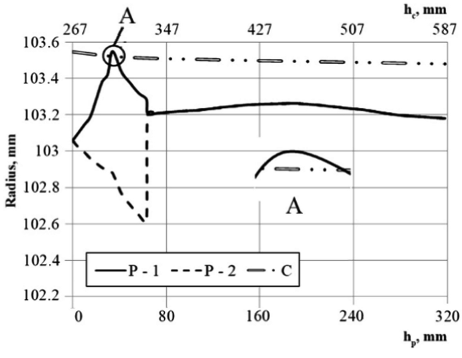



**Fig. 11.** Varying the piston and cylinder liner profiles in cross-section normal to the wrist pin axis. Heating time – 200 s

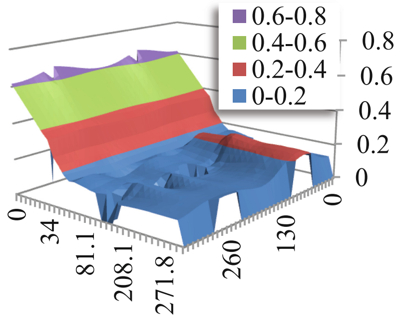



**Fig. 12.** Varying the piston profile at maximum operating mode

appeared and the minimum piston-to-cylinder clearance no less 0.13 mm at angle $\theta$ = 90° and operating time $\tau$ = 1000 s. Figure 12 shows the deformation of piston (the profile) at maximum operating mode.

## 4 Development of the Piston Design for Engine Type D49 and D80

The Fig. 13a, b shows the design of the original and proposed piston respectively. In proposed design, the outer surfaces are kept without changes, but inner zone was modified and circulated oil cooling was replaced by jet.

In line with [14], the piston head of prototype is manufactured from steel 20Kh3MVF and the piston skirt – from aluminum alloy AK4. The different materials for piston head were examined in proposed design. These materials are VCh60-2, 25Kh2G2FL, 12DKh1MFL. The thermal-stress state of the piston was determined by [13], including workflow calculation, choosing the $\alpha$-equation [2], determining the thermal BC. Table 4 presents the BC in zones shown on Fig. 13.

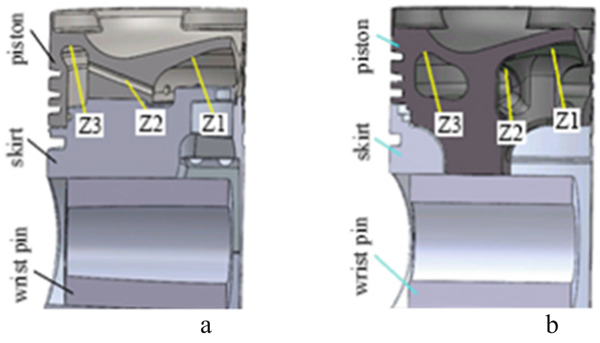



**Fig. 13.** The D80s pistons design: a – original; b – proposed

The pistons stress states are presented on Fig. 14 and in Table 5. It is shown that proposed design is 2 kg lighter than the original, if materials 25Kh2G2FL, 12DKh1MFL are used, and 4 kg lighter, if material VCh60-2 is used. The proposed design from cast iron is colder, the mean temperature is about 40 K less (675…685 K versus 715 K). As a result, the stress in proposed design from material VCh60-2 was dropped about in 3 times (156 MPa versus 475 MPa).

**Table 4.** The boundary conditions in chosen zones

| Zone | $T_{oil}$, K | Original | Proposed |
|---|---|---|---|
| | | $\alpha_{oil}$, W/($m^2$ K) | $\alpha_{oil}$ $\alpha_{oil}$, W/($m^2$ K) |
| Z1 | 350 | 530 | 1000 |
| Z2 | 350 | 540 | 550 |
| Z3 | 350 | 330 | 550 |

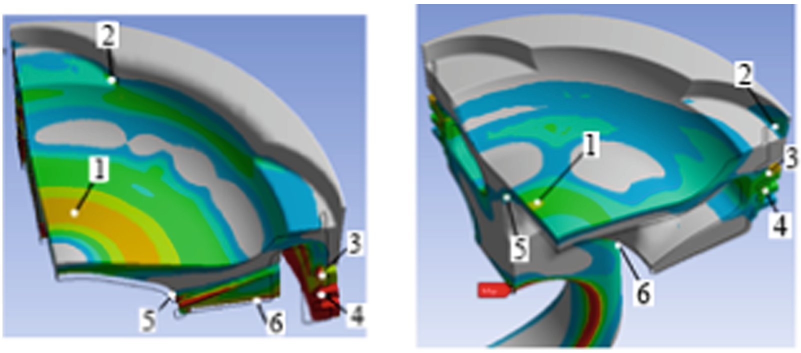



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**Fig. 14.** Stress fields: a – original; b – proposed

**Table 5.** Comparing the thermal-stress states of the original and proposed piston design

| Point num. | Original design | | Proposed design | | | |
|---|---|---|---|---|---|---|
| | 20Kh3MVF | | VCh60-2 | | 25Ch2G2FL | |
| | T, K | σ, MPa | T, K | σ, MPa | T, K | σ, MPa |
| 1 | 710 | 142 | 650 | 112 | 660 | 134 |
| 2 | 740 | 110 | 705 | 85 | 710 | 100 |
| 3 | 430 | 475 | 470 | 156 | 460 | 190 |
| 4 | 400 | 305 | 450 | 192 | 440 | 232 |
| 5 | 475 | 257 | 530 | 126 | 530 | 117 |
| 6 | 460 | 685 | 400 | 220 | 400 | 245 |

## 5 Conclusions

The paper deals with choosing the model for calculating equivalent BC at diesel transient operating mode for piston-cylinder arrangement.

The analysis results show that both the temperature and the stress of the piston reach their maximum level and then decreased when engine heated up. In variant *I* in central point of piston the temperature reaches level of 885 K (at 110 s) and the stress is 380 MPa (at 20 s), and then these levels decrease up to 710 K and 192 MPa.

The transition time has significant effect on maximum level of stress: 380 MPa at stepped loading, while only 300 MPa when transition time is 200 s.

The piston design was modified to avoid penetration of piston into cylinder in high temperature zones at all operating modes.

The proposed piston design for D80 engine is colder than the original.

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