Analysis of heat transfer coefficients for simulation of the heat exchange between oil and the internal cavity faces of the isolated piston at variable gravity conditions

A Bilohub¹, N Van Duong¹, F Sirenko¹, I Bilousov², R Symonenko³ and I Zajačko⁴

¹Zhukovsky National aerospace university “Kharkiv Aviation Institute”
²Kherson State Maritime Academy, Marine Engineering Faculty, 20 Ushakova avenue, Kherson, 73000, Ukraine
³State Road Transport Research Institute, Kyiv, Ukraine
⁴University of Žilina, Faculty of Mechanical Engineering, Univerzitná 1, Žilina, 010 26, Slovakia

E-mail: dongcomaybay@gmail.com

Abstract: The paper deals with the problems related to the development of a new piston design for replacement during repair in 2-stroke D-100 engines for locomotives. The paper addresses the design, the mass of which is 35 % less than the mass of the prototype. The results of thermal calculation, which served the initial data for the temperature and heat transfer coefficients evaluation over the heat-receiving surface of the combustion chamber of the piston, coincide with the experiment of prof. G. Rosenblit. The integral boundary conditions were refined by calculating five successive engine cycles, αP = 1830 W m⁻² K⁻¹ and TqP = 970 K were selected for further research. Next, heat transfer from the piston internal cavity to the oil, which is supplied through a nozzle in the small head of the connecting rod, was studied. The study included a 3D simulation of the cooling oil movement from the nozzle to the inner surface, both in the stationary mode (Earth’s gravity field) and taking into account the variable acceleration of the piston at the design mode. In this case, the location of the holes in the nozzle, the flow parameters and the properties of the oil were simulated. The study showed that taking piston acceleration into account leads to a slight difference in heat transfer coefficient in the center of the piston (830–800 W m⁻² K⁻¹) and rather large at the periphery (600 ... 420 W m⁻² K⁻¹). It is shown that the nozzle with 4 holes and the angle of inclination of the side holes φ = 55° gives a more efficient cooling.

1. Introduction

Ukrainian railroads as well as many other railroads of post-soviet countries have numerous main-line locomotives. The locomotives are powered with two-stroke opposed-piston diesel engines 2D100 and 10D100 have been manufactured by Kharkiv Factory of Transportation Machine-building of Malyshev. The 10D100 piston engine (figure 1) is a modification of 2D100 with augmented parameters. The power augmentation was achieved by charging the cylinder with higher pressure, which in its turn caused the redesign of crank-rod mechanism. In doing so, the pistons of 10D100 piston engine can be installed for 2D100 [1].
However, at present, the engines have consumed the lion’s portion of their overall mileage and require deep modification, which must be focused on reducing the emissions rate and enhancing the fuel efficiency and reliability.

One of the assemblies that needs modification is a cylinder/piston pair, namely piston. Lighter piston may reduce the gasoline consumption as well as wear out of the pair.

Concluding the above described, one proposed the alternative design of the piston (figure 2) [2].

![Figure 1](image1.png)  ![Figure 2](image2.png)

**Figure 1.** Original design of 10D100 piston.  **Figure 2.** Improved design for 10D100 piston.

The expected mass of the new proposed piston will be 26 kg against 37.8…39.2 kg of the top piston and 38.4…40.0 kg of the bottom one. The drastic reduction of piston mass effects on the inertia of reciprocating masses (figure 3).

The simulation shown that the new light piston for 10D100 (35 % less weight) has 2 % less mechanical losses against the initial design [2].

![Figure 3](image3.png)

**Figure 3.** Inertia of reciprocating masses of 10D100 piston engine.

To verify the simulation results it is essential to evaluate the thermal-stress state of the piston with high precision.

The problem of thermal state evaluation belongs to essential problems that must be solved when dealing with the design or modification of existing piston engines. For mow, the finite element
simulation is the most precise option to get the thermal and thermal-stress states of components of piston engine. The software that implements the finite-element methods has become user-friendly high-precision tool for solving the problem. The key to accurate temperature field is hidden in the boundary conditions that describe the heat exchange between red-hot gases and piston crown, oil film and internal face of the piston, side surface of the piston and cylinder, and piston and piston rings.

Goal of the research: simulation of the oil cooling of the piston of 10D100 piston engine. Analysis of oil cooling in the entire expected range of maintenance conditions.

2. Heat exchange between the red-hot gases and piston crown
The flow of heat exchange evaluation consists of two stages and starts from selection of formulas for heat transfer coefficient evaluation and ends up with the estimation of the achieved results.

The comparison of formulas that describe the heat exchange between the red-hot gasses and the piston crown had been made in [3]. The Rosenblit formula proved itself to be the best candidate for this kind of engines. The results can be checked from figures 4 and 5.

![Figure 4](image1.png)  ![Figure 5](image2.png)

**Figure 4.** Temperature of red-hot gasses vs crank angle.

**Figure 5.** Heat transfer coefficient of red-hot gasses vs crank angle.

| Equivalent BC | Point 1 – point with the maximum temperature of the CC of the piston | Point 2 – center of CC |
|---------------|---------------------------------------------------------------------|------------------------|
| $T_{eq} p = 950$ K| $\alpha_p = 1830$ W m$^{-2}$ K$^{-1}$ | $\alpha_p = 1830$ W m$^{-2}$ K$^{-1}$ |
|               | Point 1 – $\Delta T = 1.5$ K                                       | Point 2 – $\Delta T = 1$ K |
|               | $T_{eq} p = 970$ K                                                | $\alpha_p = 1830$ W m$^{-2}$ K$^{-1}$ |
|               | Point 1 – $\Delta T = 0.5$ K                                       | Point 2 – $\Delta T = 0.2$ K |
|               | $T_{eq} p = 1000$ K                                               | $\alpha_p = 1830$ W m$^{-2}$ K$^{-1}$ |
|               | Point 1 – $\Delta T = 2.5$ K                                       | Point 2 – $\Delta T = 2$ K |

If the second stage of the flow fails, then one has to get back to the first stage and repeat the simulation from the scratch with the updated boundary conditions.

To check the simulation results, one used the method described in [4]. It allows estimation of temperature distribution across the piston crows.
Let us start solving the non-steady state problem by the total check of $\alpha$ and $T$ till the change of temperature in check points for the last 5 cycles becomes less than 1 K. The results are presented in table 1.

The results for the rated mode are $\alpha_P = 1830$ W m$^{-2}$ K$^{-1}$ (average heat exchange between the red-hot gases and piston crown) and $T_{eqP} = 970$ K. The results were obtained from the heat flux conservation law that describes the heat flux from the red-hot gases to piston crown.

According to the described above, the average results will next need to be updated because of the heat exchange in the piston.

3. Heat exchange between the IPS (inner piston surface) and the oil film

To achieve the optimal temperature state of the parts of the combustion chamber the designers usually need to intensify the heat exchange between the IPS and oil film [5]. When dealing with pistons this usually means the forced oil cooling. Modern methods of estimating and reducing the thermal stress of pistons using mathematical modeling are the most effective tool at the design stage. However, the successful usage of mathematical models requires adequate methods for determining the boundary conditions of heat transfer over the surfaces of the analyzed parts. Evaluation of heat transfer from the cooling surfaces is a challenging problem, which is solved for each engine individually, based on the design features of the piston, the engine speed and its degree of boost. A widely used method for piston cooling simulation is all about faking piston being stationary while the oil movement is determined from the flow rate and gravity with regard to viscosity [6–9]. However, this method is not accurate enough to predict the temperature fields of complex pistons (figure 2) or pistons with jet-inertial cooling. The movement of oil in the internal cavity of the piston depends not only on the above-mentioned factors, but also on the movement of the piston. An oil cooling model with a moving piston was developed in [10]. The model of the moving piston with its grid is a challenging task and this simulation takes a long time and outputs the results with a large error. To solve this problem, one assumes that the piston does not move and replaces its speed and acceleration with the movement of oil.

The corresponding piston scheme is shown in figure 6.

![Figure 6](image)

**Figure 6.** Design scheme, where: 1 – piston; 2 – jet; $\varphi$ – slope of bores in a side face of the piston, deg; $\Omega_1$, $\Omega_2$ – diameter of central bore and bores in a side face of the piston.

4. Simulation of oil movement in the piston

The simulation was held with the ANSYS Workbench. First step of the simulation was the design of the piston internal cavity (figure 7). The internal cavity was meshed with ANSYS ICEM CFD.
The simulation was made with ANSYS CFX. Notoriously, the piston velocity $V_p$ and acceleration $a_p$ were calculated as [11]:

$$V_p = \omega \cdot R \cdot (\sin \varphi + \lambda / 2 \cdot \sin (2 \cdot \varphi)), \quad (1)$$

$$a_p = \omega^2 \cdot R \cdot (\cos \varphi + \lambda \cdot \cos (2 \cdot \varphi)). \quad (2)$$

It looks obvious that when oil leaves the jet, it has the same acceleration to piston $a_p$. The relative velocity of oil and piston correspond to the velocity of oil leaving the nozzle. In the internal cavity of the piston, outside the oil jet, oil moves with the gravity acceleration. Hence, as piston is immovable, the oil in the internal cavity of the piston moves with velocity $V_p$ and relative acceleration $a_{o-p}$. The relative acceleration $a_{o-p}$ was calculated as:

$$a_{o-p} = a_p - g, \quad (3)$$

where: $a_{o-p}$ is a relative acceleration of oil against the piston internal cavity; $g$ – gravity acceleration; $V_o$ – velocity of oil stream leaving the jets. The gravity acceleration is negligibly small, so let us overlook it. Then $a_{o-p} = a_p$ and is up directed along vertical axis (figure 8).

5. Material properties
Piston crown is made of strong ductile iron, which conductivity is $\lambda = 45$ W m$^{-1}$ K$^{-1}$ [12]. The oil that is used for the engine is 10D100-M13B2. Its properties are presented in table 2 [13].
Table 2. Properties of M14B2 oil.

| Temperature (K) | Density (kg m⁻³) | Heat conductivity λ (W m⁻¹K⁻¹) | Specific heat $C_p$ (J kg⁻¹K⁻¹) | Dynamic viscosity $\mu$ (Pa s) |
|-----------------|------------------|---------------------------------|---------------------------------|-------------------------------|
| 330             | 877              | 0.1348                          | 2000                            | 0.053                         |
| 350             | 865              | 0.1322                          | 2100                            | 0.0239                        |
| 370             | 852              | 0.1295                          | 2200                            | 0.01289                       |

6. Results of heat exchange between internal cavity of the piston and oil calculation

The required initial data for the design model of 10D100 diesel engine: the jet in the small end of the connecting rod is 3 mm diameter [14–20]. Following the formulas in [5], the oil flow rate through the jet $W_{oil} = 550$ l h⁻¹ with velocity $V_1 = 22$ m s⁻¹.

Table 3. Results of initial data calculation for the condition when the gravity acceleration is equal to $a_{O-P}$.

| Acceleration φ (º) | n | $\Theta_1$ (mm) | $\Theta_2$ (mm) | $W_{oil}$ (l h⁻¹) | $V_1$ (m s⁻¹) | $V_2$ (m s⁻¹) | $\alpha_{1av}$ (W m⁻²K⁻¹) | $\alpha_{2av}$ (W m⁻²K⁻¹) | $\alpha_{3av}$ (W m⁻²K⁻¹) |
|---------------------|---|-----------------|-----------------|------------------|--------------|--------------|----------------------|----------------------|----------------------|
|                    |   |                 |                 |                  |              |              |                      |                      |                      |
| -9.81               | 55| 4               | 2               | 1.4              | 520          | 22           | 16                   | 800                  | 460                  | 420                  |
| $a_{O-P}$           | 55| 4               | 2               | 1.4              | 520          | 22           | 16                   | 830                  | 600                  | 520                  |

In the table 3: $n$ is number of nozzles in the jet (1 central nozzle); $W_{oil}$ is oil flow rate, l h⁻¹; $V_1$, $V_2$ are velocities of oil while leaving the central and side bores, m s⁻¹; $\alpha_{1av}$, $\alpha_{2av}$, $\alpha_{3av}$ are heat transfer coefficients that describe the heat exchange between oil and piston walls (3 zones) (figure 9).

Figure 9. Heat transfer coefficient vs crank angle. AT1, AT2, AT3 are average heat transfer coefficient in zones 1, 2 and 3 when the piston acceleration is variable AT1g, AT2g, AT3g are average heat transfer coefficient in the zones when the piston acceleration is constant and is equal to 9.81 m² s⁻¹.

The simulation results of heat exchange between the heated-up piston crown and cooling oil are shown in figures 10 and 11. In the simulation the only factors to be considered were gravity and time-dependent piston acceleration.
Figure 10. Heat transfer coefficient when the acceleration is constant and equal to gravity acceleration.

Figure 11. Heat transfer coefficients when the acceleration varies and is equal to $\alpha_{\phi, p}$ (the scale is same to figure 10).

Table 4. Calculation results.

| Oil temp. (K) | $\phi$ (deg) | n | $\Omega_1$ (mm) | $\Omega_2$ (mm) | $W_{\text{in}}$ (l h$^{-1}$) | $V_1$ (m s$^{-1}$) | $V_2$ (m s$^{-1}$) | $\alpha_{1\text{av}}$ (W m$^{-2}$K$^{-1}$) | $\alpha_{2\text{av}}$ (W m$^{-2}$K$^{-1}$) | $\alpha_{3\text{av}}$ (W m$^{-2}$K$^{-1}$) |
|--------------|--------------|---|----------------|----------------|------------------|----------------|----------------|---------------------------------|---------------------------------|---------------------------------|
| 350          | 30           | 4 | 2              | 1.4            | 520              | 22             | 16             | 870                            | 590                             | 400                             |
| 350          | 40           | 2 | 1              | 1.4            | 520              | 22             | 16             | 890                            | 610                             | 410                             |
| 350          | 50           | 4 | 2              | 1.4            | 520              | 22             | 16             | 850                            | 630                             | 490                             |
| **350**      | **55**       | **4** | **2**       | **1.4**        | **520**          | **22**         | **16**         | **830**                        | **600**                         | **520**                         |
| 350          | 60           | 4 | 2              | 1.4            | 520              | 22             | 16             | 870                            | 600                             | 500                             |
| 350          | 50           | 6 | 2              | 1              | 520              | 22             | 16             | 880                            | 590                             | 410                             |
| 350          | 55           | 6 | 2              | 1              | 520              | 22             | 16             | 880                            | 570                             | 405                             |
| 350          | 55           | 4 | 2              | 1.4            | 700              | 30             | 22             | 1050                           | 720                             | 530                             |
| 350          | 55           | 4 | 2              | 1.4            | 350              | 15             | 11             | 720                            | 550                             | 460                             |
| 350          | 55           | 4 | 1.7            | 1.2            | 520              | 30             | 22             | 920                            | 650                             | 500                             |
| 350          | 55           | 4 | 2.4            | 1.7            | 520              | 15             | 11             | 760                            | 600                             | 470                             |
| 330          | 55           | 4 | 2              | 1.4            | 520              | 30             | 22             | 640                            | 470                             | 400                             |
| 370          | 55           | 4 | 2              | 1.4            | 520              | 30             | 22             | 1200                           | 750                             | 590                             |
The average heat transfer coefficient from wall to oil differs slightly in zone 1 when gravity and acceleration are considered, about 4%, but in zones 2 and 3, the difference is significant, especially in zone 2 (23%). It is clear that not only the average, but also the current (instantaneous) value is different.

Table 4 shows the results of research related to changes in design parameters – the number of holes and the angle of inclination of the side bores in the nozzle.

The results in table 4 reveal that piston cooling is efficient when the bores of a side face of the piston are tilted at an angle $\varphi = 55^\circ$ and their number is $n = 4$. The greater oil exhaust from the bore, the higher cooling efficiency, but this requires a more powerful oil pump.

In figure 12 one can see the effect of oil temperature, in particular when $T_{oil} = 370$ K, the piston is cooler, especially in the center of the combustion chamber, which is caused by a decrease in viscosity and an increase in consumption.

![Figure 12. Piston temperature field at different oil temperatures.](image)

7. Conclusions and recommendations

To simulate the temperature fields of the piston with oil jet cooling, it is necessary to take into account the dynamics of the process, namely the movement of the piston with variable acceleration.

It is proven that the nozzle with one central and three side bores tilted at an $\varphi = 55^\circ$ provides more efficient cooling. Moreover, when the oil temperature is $T_{oil} = 370$ K, the piston is cooler than in the other cases, especially in the center of the combustion chamber.

In the next steps of the research, we expect to clarify the thermal boundary conditions on other surfaces of the piston and to simulate the resource taking into account the data obtained. It is assumed to obtain data for calculating the life of the piston of the old design for comparative analysis.

8. References

[1] Volodin A I 1990 *Lokomotivnyie dvigateli vnutrennego sgoraniya* (*Locomotive internal combustion engines*) (Moscow: Transport) p 256 (in Russian)

[2] Belogub A V, Nguyen V Z, Linkov O Yu and Kravchenko S A 2016 Razrabotka konstruktsii «legkogo» porshnya dlya dizeley tipa D100 *Dvigateli vnutrennego sgoraniya* (*Internal Combustion Engines*) 1 50–55 (in Russian)

[3] Nguyen V Z and Belogub A V 2018 Raschet protsessa teplootdachi v dizelnom dvigatele tipa D-100 s ispolzovaniyem izvestnykh $\alpha$-formuly *Dvigateli vnutrennego sgoraniya* (*Internal Combustion Engines*) 2 14–21 (in Russian)

[4] Nguyen V Z and Belogub A V 2019 Opredeleniye granichnykh usloviy dlya rascheta termo napryazhennogo sostoyaniya porshnya *Aviatsionno-kosmicheskaya tekhnika i tehnologiya* 1 (153) 39–47

[5] Rozenblit G B 1977 Teploperedacha v dizelyakh (Heat transfer in diesel engines) *Mashinostroyeniye* (Moscow) p 216 (in Russian)
[6] Myagkov L L and Mikhaylov Yu V 2012 Primenenie metodiki rascheta stryunogo okhlazdeniya dlya opredeleniya parametrov sistemy okhlazdeniya porshnya sportivnogo mototsikla (Application of jet cooling calculation methods for determining the parameters of a piston cooling system of a sports motorcycle) Nauka i obrazovaniye 3 1–16 (in Russian)

[7] Chaynov N D, Myagkov L L and Karenkov A V 2006 Raschet intensivnosti maslyanogo okhlazdeniya porshney DVS Mashinostroyeniye 7 42–52

[8] Varghese M B, Goyal S K and Agarwal A K 2005 Numerical and experimental investigation of oil jet cooled piston SAE Technical Papers 2005-01-1382, https://doi.org/10.4271/2005-01-1382

[9] Nasif G G 2014 CFD Simulation of Oil Jets with Application to Piston Cooling PhD thesis

[10] Wendling L, Karyofyli V, Frings M, Hopf A, Elgeti S and Behr M 2017 7th GACM Colloquium on Computational Mechanics for Young Scientists from Academia and Industry (University of Stuttgart) p 700–704

[11] Kolchin A I and Demidov V P 2008 Raschet avtomobilnyh i traktornykh dvigateley (Calculation of automobile and tractor engines) (Moscow: Vyssh. Shk.) p 496 (in Russian)

[12] Heat capacity of cast iron 2019 Available from: http://thermalinfo.ru/svojstva-materialov/metally-i-splavy/teploemkost-chuguna-entalpiya-i-sostav

[13] Physical properties of engine and turbine oils 2019 Available from: http://www.highexpert.ru/content/liquids/oil.html

[14] Avrunin A G 1970 Teplovoznyy dizeli 2D100 i 10D100 (Locomotive diesel engines 2D100 and 10D100) (Moscow: Transportation Public) p 320 (in Russian)

[15] Kuric I, Mateichyk V, Smieszek M and Tsumian M et al 2018 The peculiarities of monitoring road vehicle performance and environmental impact MATEC Web of Conferences 244 03003

[16] Kuric I, Tlach V, Ságaová Z, Cisar M and Gritsuk I 2018 Measurement of industrial robot pose repeatability MATEC Web of Conferences 244 01015

[17] Gritsuk I, Gutarevych Y, Mateichyk V and Volkov V 2016 Improving the processes of preheating and heating after the vehicular engine start by using heating system with phase-transitional thermal accumulator SAE Technical Paper 2016-01-0204, https://doi.org/10.4271/2016-01-0204

[18] Kuric I, Bulej V, Sága M and Pokorný P 2017 Development of simulation software for mobile robot path planning within multilayer map system based on metric and topological maps International Journal of Advanced Robotic Systems 14 1–14, https://doi.org/10.1177/1729881417743029

[19] Sapietová A, Sága M, Kuric I and vácík Š 2018 Application of optimization algorithms for robot systems designing International Journal of Advanced Robotic Systems 15 1–10, https://doi.org/10.1177/1729881417754152

[20] Gritsuk I, Volkov V, Mateichyk V and Grytsuk Y. et al 2018 Information model of V2I system of the vehicle technical condition remote monitoring and control in operation conditions SAE Technical Paper 2018-01-002, https://doi.org/10.4271/2018-01-0024