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Experimental Study Of Heat Transfer In The Cooling Jacket Of ICE KAMAZ

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Abstract- The research of thermal state for the loaded parts of engines becomes more relevant due to the increasing degree of ICE forcing. One way to maintain optimal thermal condition is the use of a managed water pump of a cooling system. Foreign experts devote a lot of works to this method, but they pay insufficient attention to the heat transfer beyond jacket space. These issues were actively studied during the 70s of the last century, but during the obtaining of the known equations the change of coolant consumption that does not give a complete picture of the heat exchange in a cooling jacket was not taken into account. The studies of heat transfer to the coolant during the rotation speed control of centrifugal pumps were performed in STC OJSC KAMAZ within a joint KNRTU-KAI project. The object of the study was a promising gas piston engine, where the rotation of the pump was ensured by an independent electric engine. The engine had 4 thermally measured bushes of the right semiblock, as well as the cylinder block head. After the processing of experimental data the criteria equations of heat transfer in the cooling jacket of ICE KAMAZ were obtained.

Keywords: heat exchange, ICE, cooling system, heating, thermal state of the engine, coolant flow

Introduction

At the present stage of engine industry development it is increasingly important to maintain an optimal thermal state of specially stressed parts in order to increase the service life. A lot of works was devoted to it recently.A lot of works was devoted to it recently. There are two approaches to the assessment of ICE elements thermal state: the calculations in local three-dimensional problem formulation, and the calculations in a complex one-dimensional formulation.

In general, the works in three-dimensional formulation are based on an assessment of thermal stress state of the cylinder block head (CBH), piston or heat exchange in the cooling jacket of an engine.

The study of the CBH temperature field within the diesel engine GW4D20 was described in Baoksin Zhao et al. works [1]. In order to estimate the thermal stress of the cylinder block head, they simulated the temperature field.

Zhilin A.A. and Zharov A.V. analyzed not only the CBH, but also a coil channel between the sleeve and the cylinder block (CB), and identified the areas in the CBH channels, where the film boiling may occur [2].

At the same time one-dimensional modeling of the thermal state, which allows to evaluate not only the temperature of characteristic of nodes, but also the influence of the coolant temperature on various indicators becomes more and more popular. Zhukov V.A. studied the dependence of the energy and environmental performance of an engine on the thermal and physical properties of the motor coolant in [3]. The engine efficiency increase due to the use of the cooling system (CS) application with a high pressure was studied in [4], where the operating temperature of the coolant increases by pressure increase. Another study results are presented in [5]. The feature of this work is the breaking of elements apart during the modeling process. The functional model includes a cooling system, lubrication system, combustion (released heat) and friction. The impact of all selected systems on the temperatures of the revealed details is simulated during the warm-up. The CS is described in [6- 8] with the intelligent control of a pump and an electronic thermostat. These studies demonstrated the need to regulate the flow of a coolant to maintain optimal thermal state of particularly stressed parts. Another important factor is the engine operation performance increase as a whole by increasing the coolant temperature and the use of an electric motor for the pump drive of CS [7.9].

The Research Institute of energy-efficient technologies KNRTU-KAI studies the CS characteristics with intelligent pump control. The main tool for the numerical experiments is the functional CS model, but in order to assess the impact of coolant flow on the thermal state of the ICE thermal loaded parts a precise determination of heat transfer equation in a cooling jacket is necessary. Similar equations are obtained for different motors since the 70-ies [10 and 11], however, since the degree of engine speeding up increased substantially, primarily due to the use of more efficient turbochargers. Most of the equations presented in [10 and 11], are criterial ones, but the results of ICE "KAMAZ" cooling system calculations with intelligent pump drive on a functional model in LMSAMESim give considerable discrepancy with the experimental data.

For the correct description of heat exchange processes in the cooling jacket of the engines "KAMAZ" the task of getting a criterion equation of heat exchange in the cooling jacket of ICE "KAMAZ" is set.

Main part

In order to be able to estimate the thermal state of the engine in the functional model of ICE KAMAZ cooling system a series of experiments on a motorized stand of OJSC KAMAZ STC is performed. The tests were conducted using the gas engine KAMAZ 830.11-320 running on methane. In addition to standard thermal and mechanical parameters measured on the bench, the pressure and temperature sensors are added in the channels of CS jacket, the thermal measurement of CBH and bushes is performed liners and cylinder head, the coolant flow meter is installed. Tests were conducted at a liquid flow along a large cooling circle. It is also necessary to note that the water pump drive is carried out from an independent electric motor the rotation speed of which varied during the experiment.

A feature of the heat transfer equations in presented earlier sources [10 and 11] is that they are received, provided that the flow of coolant in the cooling jacket channels is directly dependent on KV rotation speed. Thus, the equation does not summarize the data at different flow rates of coolant at one of the engine operation modes. Besides, the rotation speed of the pump is conditioned by the pressure in the cooling jacket cavity, which substantially affect the thermal physical properties of the liquid and the process of heat exchange during phase transitions.

In order to study the heat emission in the cooling jacket of ICE KAMAZ a series of tests is performed at certain rotation frequency of a crankshaft and load on the shaft during engine warming (Table 1).

Table 1. Moment on the crankshaft of KAMAZ ICE 830.11-320

KV frequency, rpm	Load, %					
	0	20	40	60	80	100
1000	0	235	468	703	937	1171
1400	0	274	548	822	1096	1370
1800	0	243	486	730	973	1217
2200	0	204	408	613	817	1022

Within the above presented modes, the pump rotation frequency was changed in accordance with Table 2.

Table 2. Pump rotation frequency (rpm), depending on a crankshaft rotation frequency

KV frequency, rpm	Pump rotation frequency, %			
	100	75	50	25
1000	1596	1197	798	399
1400	2234	1676	1117	559
1800	2873	2155	1437	718
2200	3511	2633	1756	878

During the experiment, the engine was warmed up from 20 °C to 80 °C (which corresponds to the opening of the thermostats). At that the engine load, the engine speed and the speed of the pump were maintained within constant values.

In order to evaluate the heat exchange in the cooling jacket the bush (wall) temperature was averaged according to the formula:

$$t_w = \frac{\sum_{i=1}^{12} t_{wi} * F_{wi}}{F}$$

Geometrical parameters required for bush temperature averaging along the area are summarized in Table 3.

Table3.Geometrical parameters

Parameter	Designation	Value
The height of thermocouple first belt	l_1	16 mm
The distance between the first and second zone of thermocouples	l_2	53,7 mm
The distance between the second and third zone of thermocouples	l_3	53,7 mm
Bush external radius (fig.1)	r	68,45 mm
The rectangle width is equal to	a	10,8 mm
External diameter of ring channel (fig.1)	D	151 mm

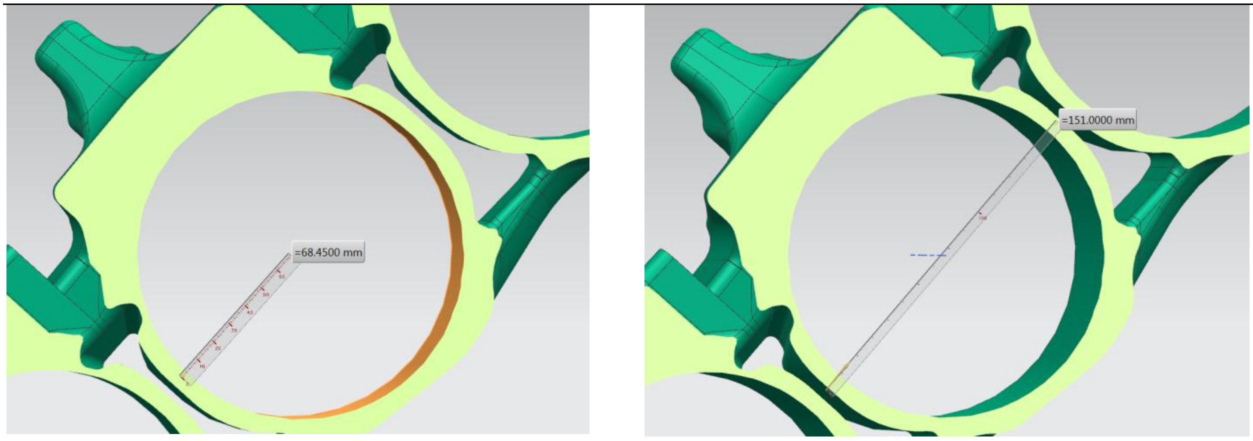

Figure1. Geometric parameters of ring channel

In order to determine the flow of coolant in the space beyond jacket the results of previous hudraulic tests of ICE KAMAZ cooling jacket were used. It was found that the distribution of the cost along half-blocks has a permanent character [12], at a known total flow rate. It is possible to calculate the costs through the oil cooler (OC), the jumper between the half-blocks and through the whole right semiblock.

The water temperature at the inlet to a right semiblock is calculated by the enthalpies of the incoming flows from the cooler and the left semiblock (LSB):

$$t_{\text{вх}} = \frac{G_1 * Cp * t_{\text{мо}} + G_2 * Cp * t_{\text{ЛПБ}}}{G_3 * Cp}$$

At an average temperature of the coolant t_{cp} the thermal physical properties of the heat carrier are calculated, and taking into account the geometrical parameters the water velocity, Reynolds numbers, Prandtl and Nusselt numbers are calculated.

The result of the calculations according to the abovementioned formulas revealed that the average heat transfer coefficients by modes take the values from $2*10^3$up to $5*10^3$Wt/m^2/°C, which indicates a sufficient cooling efficiency. The reduction of the water pump rotation frequency involves the decrease of heat transfer in the cooling jacket at warming (Figure 2), but the temperature of the thermal loaded parts do not exceed the allowable values [13].

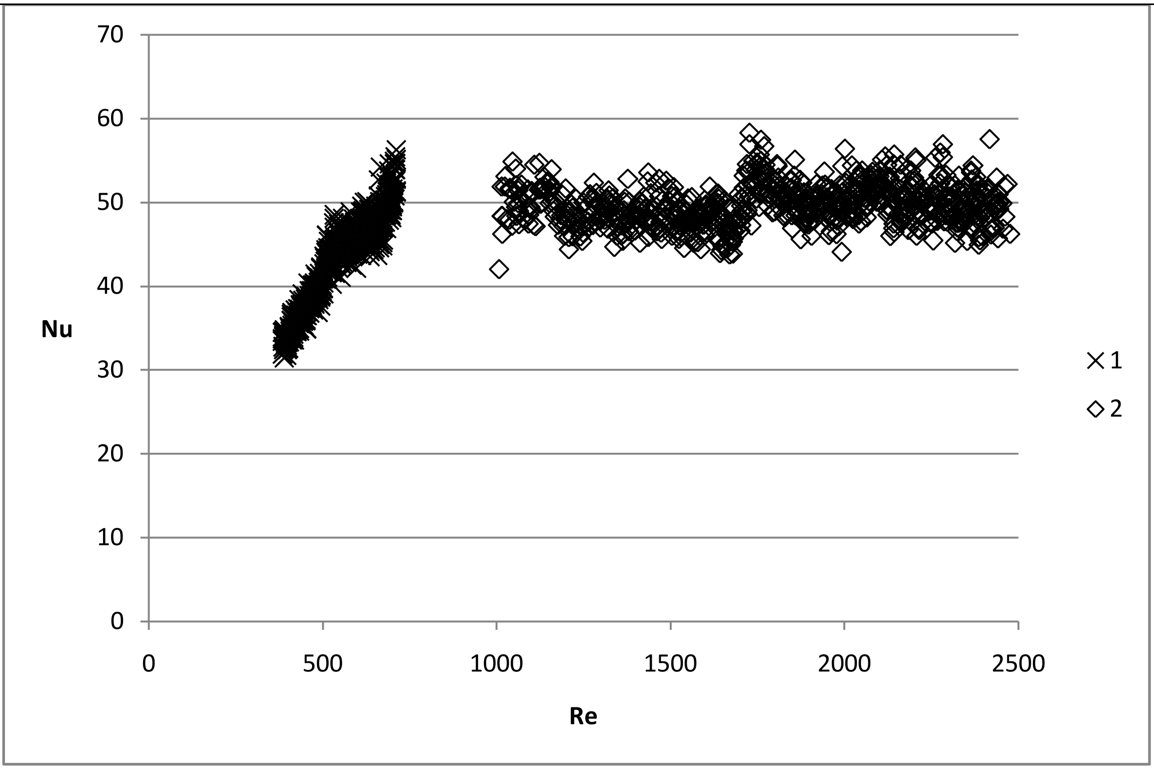

Figure2. The experimental values of the Nusselt numbers at 1000 rpm KV and 20% load: 1 -25% of pump rotation speed; 2 -75% of pump rotation speed

Data analysis

In order to obtain a proper functional model of ICE CS KAMAZ a criterion equation describing heat transfer in beyond jacket space of an engine was obtained. A conventional technique was used for this.

The result of summary data clearly reveals 2 heat exchange modes depending on the mode of the coolant flow. - laminar and transient (turbulent one) [14].

In the case of transient flow (Figure 3-5) the influence of thermal properties is described by the simplex $\mathrm{Pr}^{0.43}\left(\frac{Pr}{Pr_w}\right)^{0.25}$(the most common one to evaluate the effect of thermal physical properties on heat transfer for turbulent fluid flow). In this case the experimental data are summarized by the following equation:

$$Nu = 5{,}3 * Re^{0.25} * \mathrm{Pr}^{0.43} * \left(\frac{Pr}{Pr_w}\right)^{0.25} \qquad (1)$$

Figure 3 shows the ratio of the left and right sides of the equation1, which clearly shows that the extent of 0,25 fairly well describes the effect of the coolant flow rate, as all points are located almost horizontally.

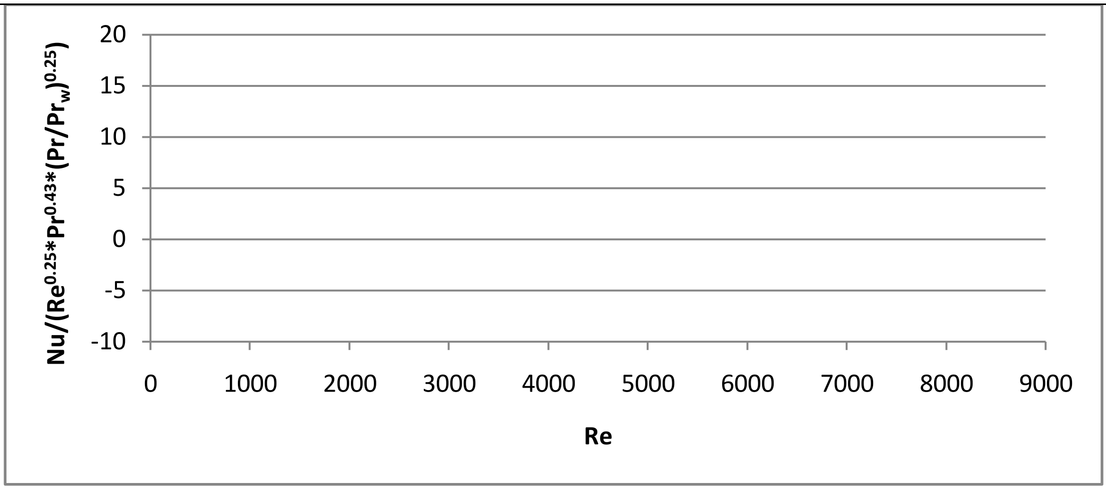

Figure3. The ratio of Nusselt numbers experimental values to the right part of the equation 1

The known equations for the calculation of heat exchange in the space beyond jacket are presented below.

Petrichenko's formula:

$$Nu = 1.58 * Re^{0.5} * \mathrm{Pr}^{0.33} * \left(\frac{Pr}{Pr_w}\right)^{0.25} \qquad (2)$$

Buznik's formula:

$$Nu = 0.017 * Pe^{0.8} * \mathrm{Pr}^{0.4} * \left(\frac{d_2}{d_1}\right)^{0.18} \qquad (3)$$

Novennikov's formula:

$$Nu = 0.7004 * Re^{0.6}\mathrm{Pr}^{0.43} * \left(\frac{Pr}{Pr_w}\right)^{0.25} * \left(\frac{d_{экв}}{d_1}\right)^{0.85} * \varepsilon_l \qquad (4)$$

Mikheeva's formula:

$$Nu = 0.021 * Re^{0.8} * \mathrm{Pr}^{0.43} * \left(\frac{Pr}{Pr_w}\right)^{0.25} \qquad (5)$$

Rosenblit's formula:

$$Nu = 0.395 * Re^{0.64} * \mathrm{Pr}^{0.32} * \left(\frac{w_{виб}}{w_{п}}\right)^{0.23} \qquad (6)$$

where$w_{виб} = 8.34 * 10^{-10} * \left(\frac{2n}{i}\right)^3 + 7.25 * 10^{-6} * \left(\frac{2n}{i}\right)^2 - 1.385 * 10^{-3} * \frac{2n}{i} + 54 * 10^{-2}$, n – KV rotation frequency, i–tact.

The comparison of values obtained using different formulae is presented on figures 4 and 5. Figure 4 demonstrates the engine heating points at KV rotation frequency of 1800 rmp and the load of 40% and 75% from the centrifugal pump rotation frequency (2155 rpm). The best coincidence with experimental values is provided by the calculations using the equations 1 and 3.

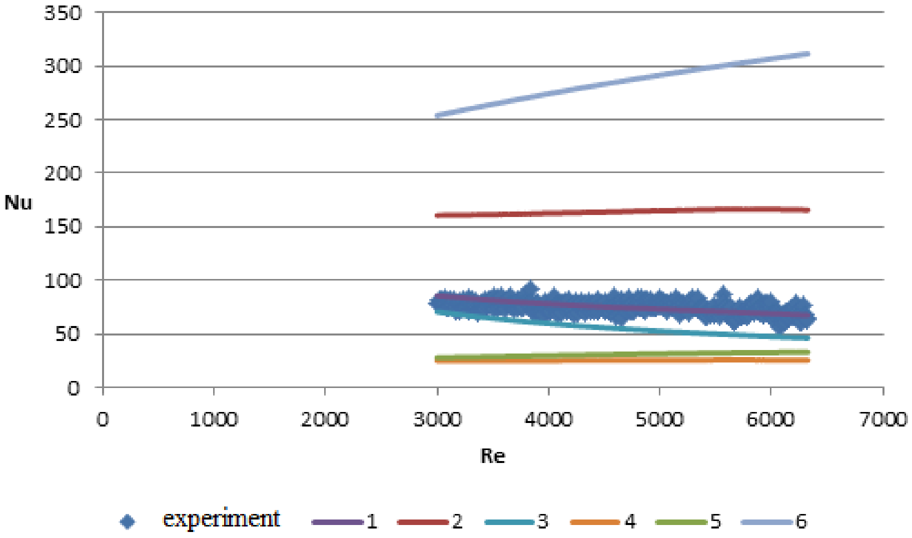

Fig4. The comparison of calculated and experimental data at the KV frequency of 1800 rpm, 40% of load and 75% of pump rotation frequency

A more complete picture is presented on Figure 5, which shows all the modes with the transitional flow regime. Formula 3 also provides quite a good description of heat transfer, but the best coincidence is observed at the formula 1.

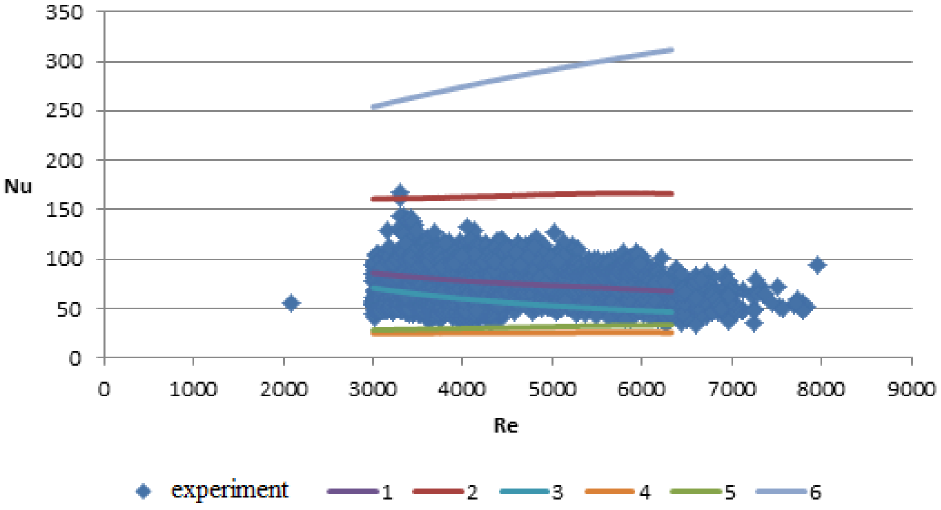

Figure 5. The comparison of calculated and experimental data for the transient mode of coolant flow

At laminar flow mode the effect of thermal physical properties of a coolant is characterized by the simplex$\Pr^{1/3}\left(\frac{Pr}{Pr_w}\right)^{0.25}$.A greater impact of the flow rate on heat transfer should be noted compared to the transitional regime of fluid flow. At that the experimental data are summarized in this formula:

$$Nu = 2{,}6 * Re^{0.33} * \Pr^{1/3} * \left(\frac{Pr}{Pr_w}\right)^{0.25} \quad (7)$$

The horizontal location of all points on the graph (Figure 6) provides the correct judgement concerning Reynolds number degree.

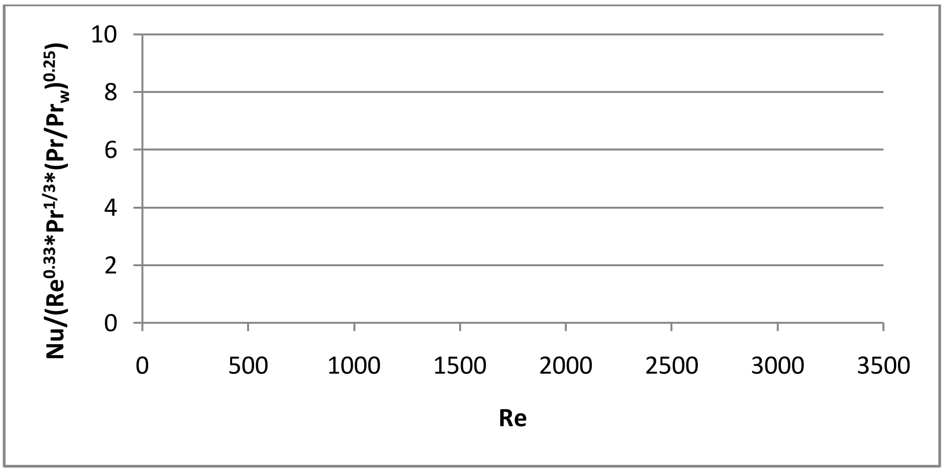

Figure 6. The ratio of Nusselt number experimental values to the right part of the equation 7.

The results are compared similarly to the transitional mode of coolant flow. These results were obtained by different formulas (in Figure 7), at the KV rotation speed of 1000 rpm, the load of 60% and 75% from the rotational speed of a centrifugal pump (1197 rpm). The greatest convergence of formula 7 and formula 3 is observed here.

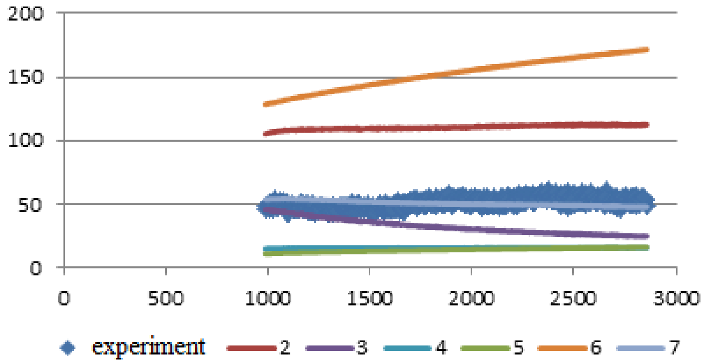

Figure 7. The comparison of the calculated and experimental data at the KV frequency of 1000 rpm, 60% of load and 75% pump rotation frequency

When comparing the calculated values with the full range of the experimental data, a fairly high convergence for formula 7 is observed (Figure 8).

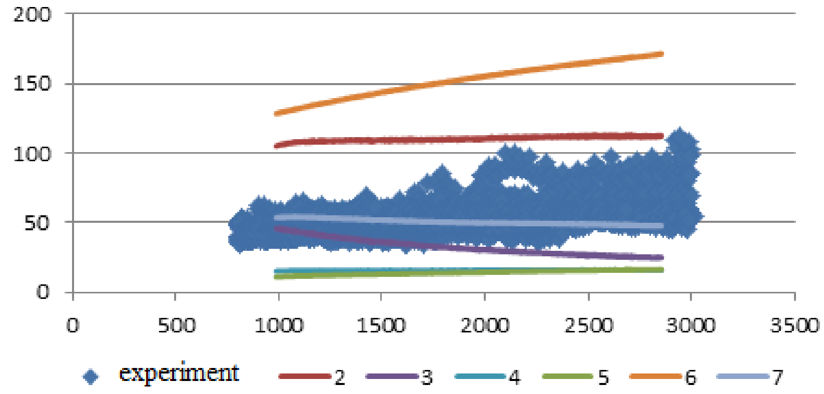

Figure 8. Comparison of the calculated and experimental data for the laminar mode of coolant flow

Conclusion

This paper analyzes the results of heat transfer experimental studies in the cooling jacket of ICE. The criterial equations of heat transfer in laminar and transitional regime of fluid flow are obtained on the basis of these data. At that the average error of the calculated values in comparison with the experimental ones makes 18.4% for transient mode, and 11.4% for laminar mode. The use of the obtained equations is suitable for the calculations in the one-dimensional formulation in order to estimate the average temperatures of cylinder-piston group parts. Besides, the formulae are applied to estimate the thermal state of a bush not only in heating modes, but in steady engine operation modes. It should be also noted that the assessment of maximum temperatures is possible during the study of CHG temperature dependence and the average temperature of a bush.

Summary

The experiments also showed the increase the engine mechanical efficiency at the reduction of coolant consumption by coolant temperature, oil and parts increase. The reduction of coolant flow at low load modes has a positive effect not only on the efficiency of an engine, but also on ecological values (the emissions of hydrocarbons and carbon monoxide are reduced). Also, one should emphasized that the flow of coolant decline reduces an engine warm-up time and has a positive effect on the durability of thermally loaded parts as part optimum temperatures are reached much faster.

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