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MODEL OF HEAT EXCHANGE IN PULSED COOLING SYSTEM OF FIRE PUMP STATION ENGINE

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Abstract. It is established that one of the possible constructive solutions for the problem of intensifying the heat exchange process is thermal pulse loading of the cooling system. The created model of pulse cooling of the fire pumping station engine was implemented using the MatLab/Simulinc complex. The necessity of the developed model in a fairly wide range of coolant flow is proved. The dependence of the relative heat transfer coefficient on the Struhal number related to the pulse frequency, which was varied by changing the frequency of modulation of the coolant flow as well as its flow rate, was investigated.

Keywords: fire pumping station, cooling system, pulsating flow mode, pulsation valve

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Introduction

To ensure the fire safety of people in buildings and structures, a set of engineering systems is used, one of which includes an internal fire-fighting water supply, which is a complex of devices and structures for supplying water to the epicenter of fire with required pressure and sufficient flow. Such system includes a number of pipelines and devices that provide water supply, as well as a special (fire) pump station (FPS). The choice of FPS characteristics is determined by the proposed fire extinguishing method, the presence of a water source, as well as the architectural and planning characteristics of the object in question. Stationary FPS are located in buildings, structures – in modules or blocks. Mobile FPS can be moved and are used by fire protection units to draw water from fire reservoirs and then supply it to the facility [1].

Diesel fuel pumps are a complex mounted on a single frame, the main components of which are a diesel engine, usually with a liquid cooling system, and a fire-fighting centrifugal pump of considerable capacity. During the operation of a diesel engine, a significant amount of heat is released, which must be removed from the engine cylinder block [2].

The FPS engine cylinder block is a casing, the functioning of which assumes an uneven thermal effect, as a result of which the temperature stresses arising from various structural elements manifest themselves in the form of deformations, therefore, the task of intensifying the cooling process seems to be very relevant. One of the possible solutions applied to isotropic shells is thermal pulse loading, which is considered in [3, 4]. Thus, in [5] it is stated, that the increase in the time-averaged heat transfer coefficient in the pulsating flow mode and small Reynolds numbers reaches 3,5 times with a relative amplitude of velocity fluctuations equal to four.

Flow pulsations are usually created using a piston pump by periodically bypassing the flow [6]. However, due to the frequency modulation of the motor and the pumping of liquid through the pipeline, additional energy costs are required [7]. Another method of pulsation generation involves the use of an electromagnetic valve or a reciprocating pump, which makes the energy saving effect insufficient [8]. Therefore, increasing the heat exchange efficiency of the engine cooling system and reducing energy consumption by the pulsation device posed new problems in the development of a pulsation heat exchange model [9].

Thus, the task is to develop a model of pulsation heat transfer in an orthotropic shell with the consideration of a moving pulsed local heat source in relation to optimizing the cooling system of the FPS engine.

Materials and research methods

In [10], a cooling scheme was proposed that implements a model of pulsation heat exchange of a diesel engine cooling system. A device that increases the efficiency of the FPS engine cylinder block cooling process is shown in fig. 1.

The water cooler 5, the cylinder block 1 and the first hydraulic accumulator 6 form a pulsating cooling water circulation system.

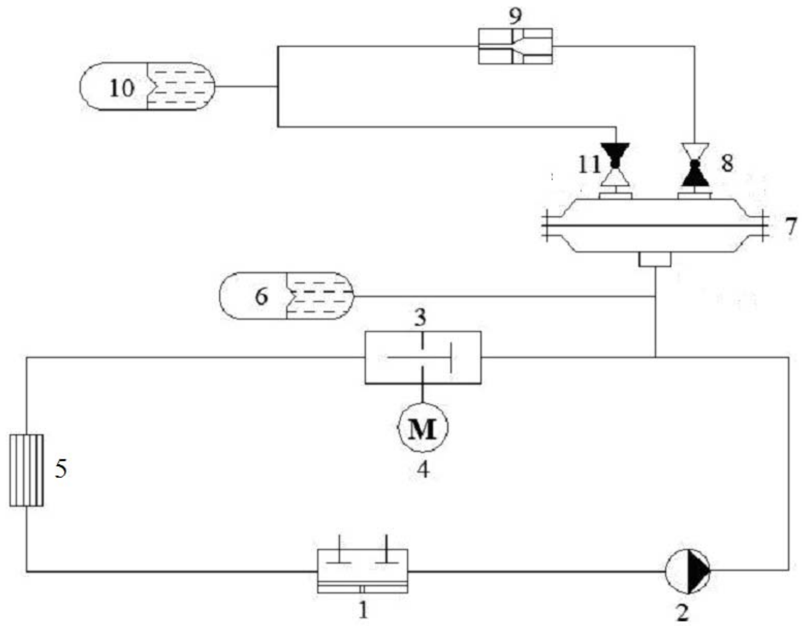


Fig. 1. **Diagram of the pulse cooling system of the PNS engine:**
1 – cylinder block; 2 – centrifugal water pump; 3 – pulsation valve; 4 – electric drive motor; 5 – local heat exchanger; 6 – the first hydraulic accumulator; 7 – diaphragm booster; 8 – the first check valve; 9 – conical tube; 10 – second hydraulic accumulator; 11 – second check valve

The cooling water, moving through the pipeline from the centrifugal water pump 2 through the pulsation valve 3, enters the cooled cylinder block 1. The first hydraulic accumulator 6 is connected in the line between the centrifugal water pump 2 and the pulsation valve 3. To smooth out possible changes in coolant pressure in the pipeline, a pulsation valve 3 is used, which is controlled by an electric drive motor.

The coolant flowing through the cavities of the cylinder block 1 is supplied at the inlet of the centrifugal water pump 2, and passing through the pulsation valve 3, creates a pressure drop to form a pulsating flow.

The pulsating flow generated is piped into the cooling circuit of the cylinder liner. The heat exchanger 5 is cooled by water supplied to the pumping from the main line. The cooled water is piped into the cylinder liner 1 of the diesel engine, forming a closed pulsating cycle.

Cooling water is supplied at high speed to the elastic membrane of the membrane supercharger 7. The liquid that creates the pressure drop enters under pressure from the membrane supercharger 7, then into the lower cavity of the membrane supercharger 7 to deform the elastic membrane. After deformation of the elastic membrane, the liquid in the cavity of the diaphragm blower 7 is pushed out through the first valve 8 and enters the conical tube 9 through the first one-way valve 8. The static pressure decreases, and the second hydraulic accumulator 10, connected to a conical tube 9, compensates for fluctuations in liquid pressure 9. After increasing the speed, the liquid continues to flow through the pipeline to the second one-way valve 11. After passing through the second one-way valve 11, the coolant flow flows back into the diaphragm blower 7 and acts on the elastic membrane. As the temperature increases, the pressure in the pulsating circulation system increases due to the action of the elastic membrane, the speed of the pulsating liquid increases due to the increase in pressure, and the heat exchange efficiency will also increase after the speed increases.

The key element of the pulse cooling system of the FPS engine, shown in fig. 1, is a membrane amplifier that provides pulse modulation of the coolant flow. An approximate diagram of one of the design options for such a device was described in [11] and shown in fig. 2.

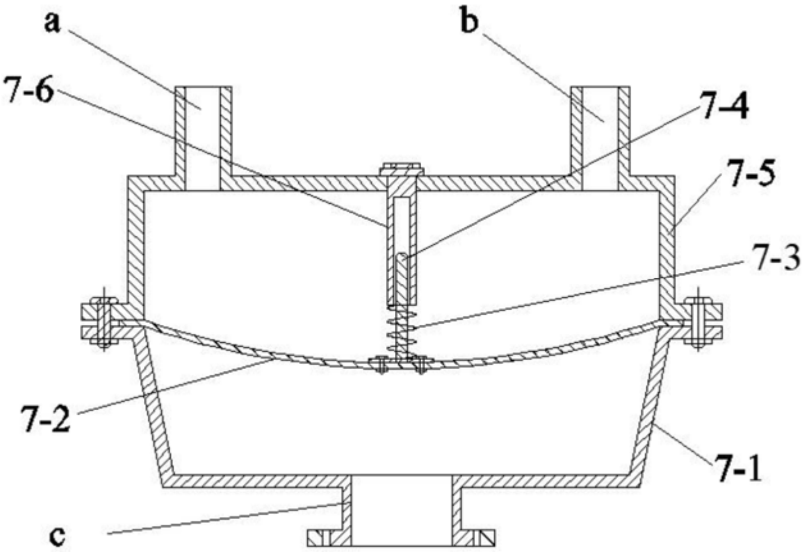


Fig. 2. **Membrane amplifier circuit**

As shown in fig. 2, the outer part of the membrane amplifier 7 contains a lower cavity 7-1 having an inlet opening **c**, and an upper cavity 7-5 having a first outlet opening **a** and a second outlet opening **b**. The lower cavity 7-1 retains the same height. The upper cavity 7-5 is a cylindrical shell without a bottom.

The diameter of the lower part of the housing of the membrane amplifier 7-1 is smaller than the diameter of its upper part. In turn, the upper part of the housing of the membrane amplifier 7-5 has a cylindrical shape. The diameter of the housing 7-5 is equal to the diameter of the upper part of the housing 7-1. The housing of the diaphragm amplifier 7-1 in its lower part is connected to the pipeline through the flange connection **c**.

A diaphragm 7-2 made of elastic material, the dimensions of which exceed the dimensions of the housing 7-5, is tightly fixed between the lower part of the housing of the diaphragm amplifier 7-1 and its upper part 7-5. The sealing of the diaphragm 7-2 ensures no coolant leaks.

In the experiment described in [10], the results of a study of the comparative cooling efficiency of engine cylinder blocks in the presence or absence of a membrane amplifier were obtained. The effect of a membrane amplifier on the pulsating pressure in a pipeline at the same pulsation frequency was also studied for three possible pulsation frequencies: f_1=1,43 Hz; f_2=1,70 Hz; f_3=2,85 Hz at a coolant flow rate of 0,182 l/s and 0,22 l/s in the pipeline.

Mathematical model of the direct problem solving method

The formulation of the direct problem of heat transfer between a coolant and a solid surface involves solving the equation of thermal conductivity when setting initial and limiting conditions. When switching to a stationary body heating mode, the generating module of the mathematical model generates changes in the amount of heat flow, which causes changes in limiting conditions. The function of such a module is to create a time dependence of temperatures in the cross-section of the studied shell in relation to possible standard shapes, for example, a plate or a hollow cylinder.

The mathematical model of the direct method of solving the problem involves solving the equation of thermal conductivity under the influence of a pulse-modulated heat flow on the outer surface of the shell.

The condition of the heat exchange problem between the coolant and the surface of a solid body assumes a change in the heat flux density in a short period of time according to the parabolic law:

$$\frac{q}{q_{\max}} = \frac{4 \cdot t}{t_{\max}} \cdot \left(1 - \frac{t}{t_{\max}}\right).$$

The solution of the heat conduction equation for a point located on the surface for a pulsating heat flow looks like this:

$$T_s(t) = T_o + \frac{4 \cdot q_{\max}}{\sqrt{\frac{\lambda \cdot \rho \cdot c}{t_{\max}}}} \cdot \left[\sqrt{\frac{G(2)}{G(2{,}5)} \cdot \left(\frac{t}{t_{\max}}\right)^3} - \sqrt{\frac{G(3)}{G(3{,}5)} \cdot \left(\frac{t}{t_{\max}}\right)^5} \right],$$

where *G(x)* is the gamma function, which is calculated based on the Stirling algorithm, and the Fourier criterion is set as the dimensionless time for this model:

$$t = Fo = \frac{a \cdot \tau}{R^2},$$

where *a* is the coefficient of thermal conductivity of the coolant; R is the characteristic linear channel size.

Mathematical model of the inverse problem solving method

A flat orthotropic shell of non-negative Gaussian curvature is considered, along which a pulsed heat carrier moves according to the law $x_u = x(\tau)$; $y_u = y(\tau)$. Then the equations of thermal conductivity look so:

$$h^2 \cdot \nabla_\lambda^2 \cdot T_1 - \mu_1 \cdot T_1 - \mu_3 \cdot T_2 - \frac{h^2}{a} \cdot \frac{\partial T_1}{\partial \tau} = -(\mu_1 \cdot t_1 + \mu_2 \cdot t_2 + W_1), \quad (1)$$

$$h^2 \cdot \nabla_\lambda^2 \cdot T_2 - (1 + \mu_1) \cdot T_2 - 3 \cdot \mu_3 \cdot T_2 - \frac{h^2}{a} \cdot \frac{\partial T_1}{\partial \tau} = -3 \cdot (\mu_1 \cdot t_{21} + \mu_2 \cdot t_1 + W_2), \quad (2)$$

where

$$T_1(x,y,\tau) = \frac{1}{2 \cdot h} \cdot \int_{-h}^{h} t(x,y,z,\tau) \cdot dz; \quad T_1(x,y,\tau) = \frac{1}{2 \cdot h} \cdot \int_{-h}^{h} t(x,y,z,\tau) \cdot dz..$$

Applying to equations (1) and (2) the Fourier transform in spatial coordinates *x* and *y* and the Laplace transform in time coordinate τ, using a modified form of the SIMPLE algorithm, it is necessary to make a transition to a system of linear algebraic equations in transformate space, the solution of which allows us to form a model of the pulsation heat transfer process in an orthotropic shell. The formed model of pulse cooling of the FPS engine was implemented using the MatLab/Simulinc software and hardware complex, the features of which are described in detail in [12].

The solution of the inverse problem of heating a flat orthotropic shell of non-negative Gaussian curvature, along which a pulsed heat carrier moves, is based on the application of the finite volume method on spaced grids. The third-order precision QUICK procedure was used to solve the differential equations describing the convective component of heat transfer. In order to take into account the possible influence of the diffusion components of the orthotropic shell cooling process, the central differences of the second order of accuracy were used. This made it possible to take into account the influence of the evaporation process of the coolant on fluctuations in the pressure of its flow based on the calculation of the components' volume fractions [13].

To solve the differential equations, a grid was used that was uneven in both axial and radial coordinates. The tightening of the calculation nodes occurs near the axis of the coolant flow, the section of the shell and the boundary layer near its outer walls. The area of numerical simulation has the shape of a cylinder with a diameter of 10 diameters of an orthotropic shell. The first calculation node was located at a distance of $y=U/v=0,3\div0,5$ from the wall, where U – the speed of coolant movement on the shell section.

The correct choice of the condensation constant for the boundary layer made it possible to calculate at least 10 nodes. The digital model was formed in a grid of 200×256 calculated nodes, while the ratio x/(2R)=10 was observed, where R is the inner radius of the orthotropic shell. When reducing the number of calculated nodes located on the abscissa axis, it is necessary to maintain the optimal ratio of linear dimensions of elementary volumes. A control check of the generated model was also carried out on a grid containing 300 × 400 elementary volumes.

Results of modeling

The dependence of the relative heat transfer coefficient $ER=Nu/Nu_o$ on the Struhal number ($St_D=f\cdot D/U$), which is related to the pulse frequency f, is investigated. Here Nu is the heat transfer coefficient when using a frequency-modulated coolant flow; Nu_o is the heat transfer coefficient when using a stationary flow; f is the frequency of flow modulation; D is the inner diameter of the pipeline; U is the average coolant flow rate. The mode of movement of the coolant at the pipeline inlet was determined during the preliminary calculation of the coolant flow for a pipe with a diameter of $D = 15$ mm and a length of $20\ D$.

The dependence of the relative heat transfer coefficient $ER=Nu/Nu_o$ on the frequency of modulation of the coolant flow is shown in fig. 3.

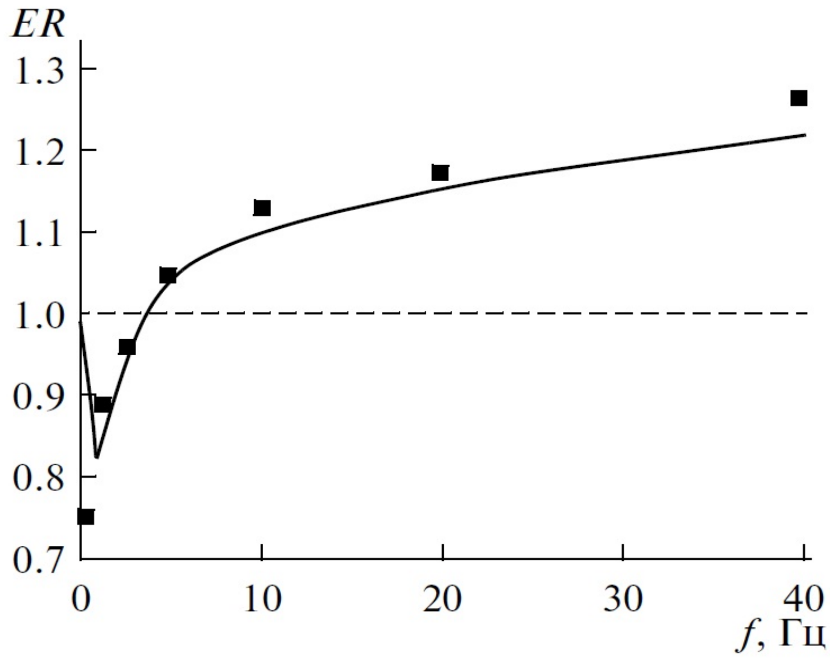
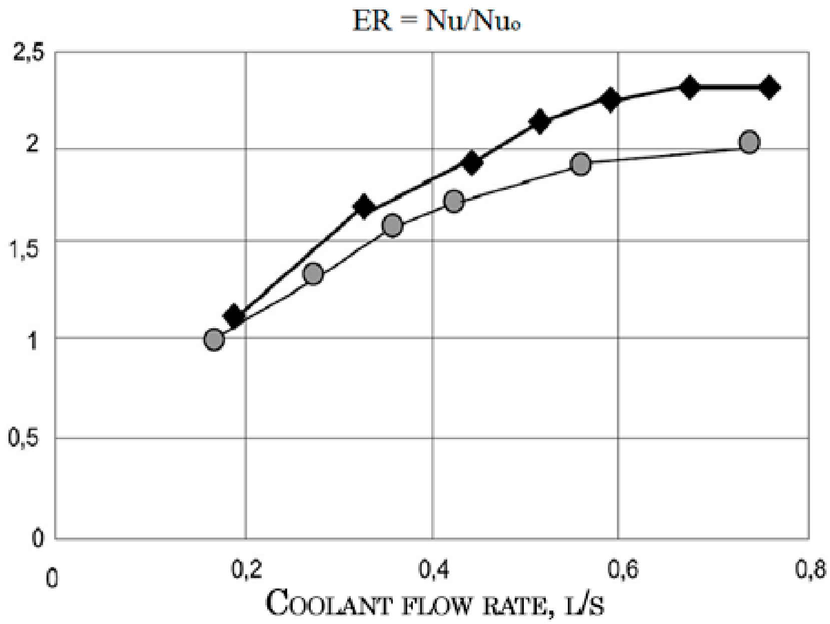


Fig. 3. **Comparison of calculated (line) and experimental (strokes) data ratios of heat transfer coefficients [14], averaged over time and over the surface for different flow modulation frequencies**

In the infra-frequency range *(f < 5 Hz)*, there is a decrease in the intensity of the heat exchange process. As the frequency f increases, a certain intensification of the heat exchange process is observed (*ER* > 1,2 for the frequency f = 40 Hz). It can be noted that the results of numerical calculations correspond satisfactorily with the experiments [14] in the entire range of investigated frequencies f.

The experimental data presented in [15] made it possible to assess the adequacy of the developed model of the pulse cooling system of the FPS engine in a fairly wide range of coolant flow. The dependence of the relative heat transfer coefficient $ER=Nu/Nu_o$ on the coolant flow rate is shown in fig. 4.



■ – pulsed flow; ● – stationary flow

Fig. 4. **Comparison of calculated (line) and experimental (strokes) data on the ratios of heat transfer coefficients [15] for various liquid flow rates**

For both pulsed and stationary coolant flow, its average flow rate was 0,37 liters/sec. The liquid consumption during the full-scale experiment ranged from 0,17 to 0,75 liters/sec. As a result, it is possible to identify the dependence of the value of the heat transfer coefficient on the coolant flow rate. Assuming that the coefficient of thermal conductivity of the wall material near the cooled surface remains constant within the limits of liquid temperature fluctuations, it can be concluded that the change in the heat transfer coefficient occurs mainly due to processes occurring in the boundary layer, which are influenced, among other things, by the nature of the movement of the coolant.

Conclusion

It has been established that one of the possible constructive solutions in relation to the task of intensifying the heat exchange process is the thermal pulse loading of the engine cooling system.

The dependence of the relative heat transfer coefficient on the Struhal number related to the pulse frequency, which was varied by changing the frequency of modulation of the coolant flow, as well as its flow rate, was investigated.

It is shown that the increase in the time-averaged heat transfer coefficient in the pulsating flow mode and small Reynolds numbers reaches 3,5 times with a relative amplitude of velocity fluctuations equal to four.

A cooling schematic that implements a model of pulsation heat exchange of a diesel engine cooling system, the main element of which is a membrane amplifier that provides pulsed modulation of the coolant flow is proposed.

The presented model of pulse cooling of the FPS engine was implemented using the MatLab/Simulinc software and hardware complex and proved the adequacy of the developed model in a fairly wide range of coolant flow rates.

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