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The influence of the main properties of the liquid on the temperature indicators of the inertial heat generator

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In the investigated inertial heat generator, the electric motor imparts rotational motion to the heated fluid. This causes centrifugal forces that move it in the radial direction and create increased pressure on the periphery of the rotor cavity. The rotor is installed above the surface of the liquid and is made in the form of a semi-cylindrical drum with conical skirt. Part of the conical skirt is always in the liquid, and on the periphery of the drum, there are chokes with calibrated holes. Throttling of fluid pressure drops contributes to an increase in its temperature. Experiments showed that the increase in the temperature of the working fluid is influenced not only by the angular velocity of rotation of the rotor and the area of the orifices of the chokes, but also by its basic properties. This is because their density, specific gravity and viscosity are different.

Keywords: inertia, heat generator, throttling, pressure, temperature

INTRODUCTION

The intensification of the use of energy resources is accompanied by an increase in heat consumption of various sectors of the economy [1]. The various processes, associated with the consumption of heat without conversion to other forms of energy, can be divided into two main categories according to the purpose of the consumed heat: consumption of heat for household needs and consumption of heat for technological needs. Currently, the share of domestic needs accounts for about 70%, and the share of technological needs - for only 30% of the total heat consumption in the country [2].

The heat supply system of Kazakhstan has developed in the framework of the implementation of the centralized heat supply approach. The length of heating networks is 12 thousand km in two-pipe terms. Boiler and heat networks are in state ownership. The main problem of the heating system is equipment wear, which leads to high losses during heat transfer through heat networks. Only 75% of the heat produced reaches the consumer [3]. Under these conditions, the development and study of heat generators excluding the enormous heat losses associated with centralized heat supply is an urgent task.

RESULTS AND DISCUSSION

The investigated inertial heat-generating installation is aimed at converting electrical energy into thermal energy, due to pushing the fluid through the throttle holes. A general view of the heating system is shown in Fig. 1. The electric motor 1 rotates the rotor 5, which is made in the form of a semi-cylindrical drum with a conical skirt, installed above the surface of the water. Part of the conical skirt is always in the liquid, and on the side of the drum, there are chokes with calibrated holes. When the rotor rotates, the fluid along a conical skirt enters the drum cavity due to inertial forces, creating increased pressure on its cylindrical wall, which promotes the pushing of fluid through the throttle holes and, as a consequence, its heating [4, 5].

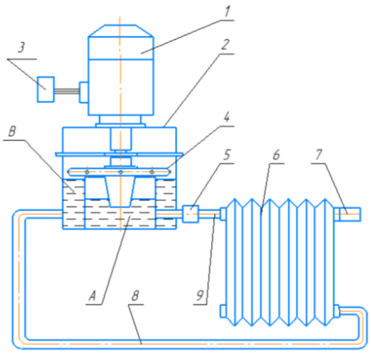

Fig. 1. General view of the inertial heating system: 1- electric motor; 2- case; 3- frequency converter "WESPER"; 4 - rotor; 5 - throttle; 6 - radiator; 7 – temperature relay; 8 - drain line; 9 - pressure line

The purpose of the present work is to establish the influence of the diameters of the throttle holes of the rotor on the temperature indicators of the thermal system when using different types of heat carriers.

The rotor speed is determined by the selected engine parameters and has a constant value,

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respectively, and the fluid pressure at the throttle holes, obtained by inertial forces, has also a constant value [6]. However, when conducting experimental studies to obtain different engine speeds, the «WESPER» frequency converter was used, which allowed adjusting the pressure at the throttle orifices of the rotor within a broad range (Fig. 2).

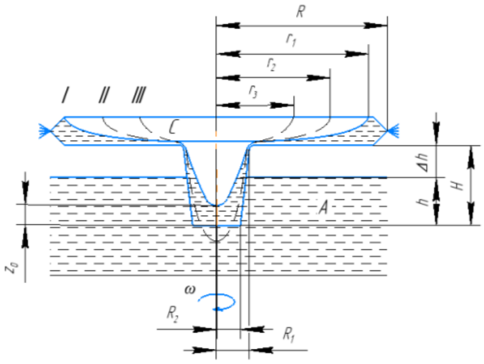

Fig. 2. The state of the fluid in the rotor cavity, depending on the number of revolutions.

The pressure (p) of fluid at the throttle orifices of the rotor is determined by the following expression

$$p = \frac{\gamma\omega^2(R^2 - r_i^2)}{2g}, \qquad (1)$$

where γ is the volumetric weight of the liquid, N/m^3; R - the outer radius of the fluid ring in the rotor, m; r_i - the inner radius of the ring of fluid in the rotor, m; g - gravitational acceleration, m/s^2; ω - the angular velocity of rotation of the rotor, *rad/s*.

It follows from the formula that the smaller the inner radius of the liquid ring r_i, the higher is the pressure at the throttle orifices of the rotor and when $r_i = 0$ the liquid pressure is at its maximum.

The mass of fluid Δm_1 flowing in time Δt through the throttle openings of the rotor is determined by the expression:

$$\Delta m_1 = \rho v_1 \Sigma S_1 \Delta t, \qquad (2)$$

where ρ is the density of the liquid; ΣS_1 - total cross-sectional area of the throttle holes; v- fluid velocity passing through the throttle holes.

The mass of fluid Δm_2 flowing through the inlet of the skirt during Δt is determined by the expression:

$$\Delta m_2 = \rho v_2 S_2 \Delta t, \qquad (3)$$

where S_2 is the inlet area of the rotor skirt.

Since the steady-state operation of the heat generator is ensured by the principle of continuity of fluid flow [4], when the mass Δm_1 of the flowing fluid through the throttle orifices is provided by the mass Δm_2 of incoming fluid through the conical skirt of the rotor, by equating expressions (2) and (3), we have:

$$v_1 \Sigma S_1 = v_2 S_2, \qquad (4)$$

It follows that for the same time interval an equal amount of liquid flows in and out the generator rotor. In cases of use of throttling holes with a small diameter, the possibilities of inertial heat generation installation are not fully realized, since the flow rate and liquid turnover in the heating system will be low. With large diameters of the throttle holes of the rotor it is impossible to create a large fluid pressure in front of the throttles, which will lead to simple pumping without increasing its temperature.

To establish the flow rate at various pressures and total areas of the throttle holes, and assess their influence on the heating temperature of the heat carrier at the Department of Thermal Physics, a full-size experimental stand was made (Fig. 3).

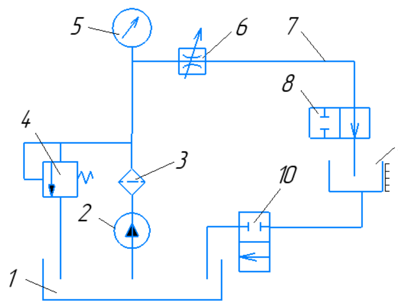

Fig. 3. Scheme of the stand: 1 - tank, 2 - plate pump G12-31M, 3 - slit plate filter 8-80-1K, 4 - safety valve ПГ54 - 22, 5 - pressure gauge, 6 - throttle, 7 - line, 8, 10 - two-position distributors, 9 - measuring tank.

The experimental stand has a lamellar pump 2, which through the filter 3, the throttle 6 and the on-off distributor 8 feeds the liquid into the measuring barrels 9. To regulate the pressure of the fluid on the stand, a pressure valve 4 was used, which was connected in parallel to the discharge line 7 and allowed changing the pressure in the range from 0 to 1.0 MPa. As a throttle, special nozzles with diameters of 1.75, 2.0 and 2.25 mm were used, which were made so that the static inlet pressure differed from the total pressure by no more than 1% [7]. The investigated liquid types were: water, antifreeze and spun oil. A thermometer and a stopwatch were used to measure the temperature of the liquid. Tap water, antifreeze, and spindle oil were used as heat carriers. To clarify the viscosity of the coolants at different temperatures, a vibration viscometer series SV-A was used, the results of which are presented in table 1.

Table 1. Heat carrier viscosity

Temperature T (°C)	Dynamic indicator η (m·Pa·s)		
	water	antifreeze	spindle oil
25	2.22	4.06	23.3
30	2.02	3.98	19.3
35	1.95	3.81	15.8
40	1.84	3.62	13.3

In addition to temperature, pressure affects fluid viscosity. However, up to 20 MPa, the effect of pressure on the viscosity of the fluid is insignificant [8]. Considering that the pressure in the heat generator varies in the range of 0–3.0 MPa, we neglected the effect of pressure on the viscosity of the coolants.

Experiments allowed us to establish the dependence of temperature on the fluid pressure and flow rate per unit of time. Sequence of the experimental study:

- in line 7, a nozzle with a calibrated orifice (choke) is installed;
- the safety valve is fully open;
- the pump is turned on and through the throttle 6, distributors 8, 10 and valve 4 the fluid is discharged into the tank 1;
- by adjusting the safety valve 4, we gradually increase the pressure in line 7, while the flow rate through the valve decreases and increases through the distributors;
- at the beginning of the operation of the safety valve 4, the maximum flow rate through the throttle orifices is achieved at the set pressure;
- distributors 8, 10 are closed and through a measuring tank 9, we set the maximum flow rate of the throttle at the adjusted pressure.

Repeating the experiment for different diameters of nozzles with calibrated holes and changing the type of coolant, we obtained the dependences of the filling time of the measuring tank at different pressures.

Fig. 4 shows the results of the experimental dependences of the filling time of a measuring tank with a volume of 0.0047 m^3 for various types of liquids (water - fig. 4, *a*; antifreeze - fig. 4, *b*; spindle oil - fig. 4, *c*) at different pressures *P* of the fluid in magical and throttle bore diameters *d*. It follows from them that with increasing pressure in the mains, the time for filling the measuring tank for all diameters of the nozzles is reduced.

If you add trend lines depending on Figure 4, *a*) and mentally extend them, then they will converge, which would mean the absence of the effect of the hole diameter at high pressures on the fluid flow. However, after reaching the maximum throughput of the throttle holes, a further increase in pressure will not affect the filling time. Lines will be parallel.

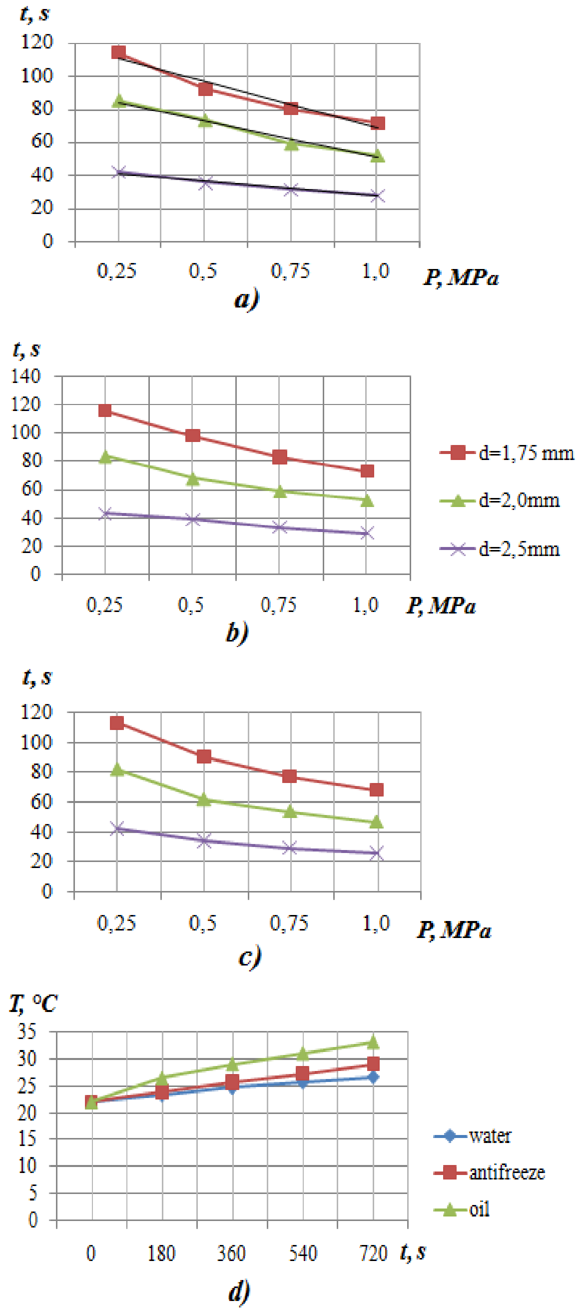

Fig. 4. Experimental dependencies

It follows from the table that water has the lowest dynamic viscosity, and oil has the highest. Therefore, the time needed to fill the measuring tank should increase in the following sequence: water < antifreeze < oil. However, in the course of the experiment it was established that the time for filling the measuring container had the sequence oil < water < antifreeze. This is because a deviation of the experimental results occurred due to the use of a pump designed to work on mineral oils with a kinematic viscosity of 17×10^{-6} to 40×10^{-5} m^2/s.

When determining the temperature change of different types of coolants with throttling time through a nozzle with a diameter of 2.5 mm, it was found that the temperature increase of liquids with high viscosity is much higher than that of liquids with lower viscosity (Fig. 4, d). According to the data in the table, the viscosity of tap water is 1.47 times lower than the antifreeze viscosities, and 8 times lower with respect to the spun oil, while the temperature of water heating below the antifreeze is 1.46 times and oil 2.1 times. From this it follows that viscous liquids are rationally used as a coolant for the heat generator under consideration.

CONCLUSIONS

Processing the results of experimental studies has established that the diameters of the throttle holes affect the flow rate and temperature of the liquid.

It is revealed that the temperature of heating viscous liquids for the same period of time is significantly higher compared with liquids of low viscosity.

Studies have shown that for using liquids with greater viscosity, it is necessary to isolate the heat carrier in the heat generator from the general heating system. Heat removal shall be carried out through heating pipes installed in the body of the heat generator.

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