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CALCULATION TEMPERATURE AND PRESSURE OF THE ROTARY ENGINE

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Abstract. **The principles of calculating the temperature and pressure in the working chambers of the rotary vane engine with an external supply of heat are considered. The mathematical model for calculating the pressure and the temperature in the chamber with heat transfer between the working fluid and the chamber walls is built. The plots of the dependence of the pressure and the temperature in the chamber on the angle of rotation of the output shaft at the minimum and maximum temperature of the walls are obtained.**

Keywords: **the rotary vane, the heat transfer with the walls, the chamber temperature, the chamber pressure.**

I INTRODUCTION

The main areas of economic development in the XXI century is the search for promising energy conversion technologies based on high-performance thermodynamic cycles for the use of renewable energy sources. In the opinion of many foreign experts, promising direction is the development and widespread adoption of power generation systems based on external combustion engines [6]. One of such engines is a rotary-vane engine with an external supply of heat (RVE), developed the staff of the Pskov State University Y.N. Lukyanov, M.A. Donchenko and others [4,5].

Vanes group of the RVE (Fig. 1) consists of a cylindrical housing 1 in which installed two coaxial rotor (external and internal), two sealings housing 2 and two end covers 4. The outer rotor and the inner rotor are consists of two shafts 6, 7 respectively, the four pistons 5, four pressure plates 8 and four seals 3. Two rotors form a four working chambers of variable volume. A four power strokes are proceeds in each of the chambers consistently: receipt of the working fluid to the heater through the windows in the end cap 4, the expansion of the working fluid, post working fluid to the cooler through the windows in the end cap 4 and the compression of the working fluid.

Thus, working chamber the engines consists by the following components: two shafts 6, 7, two end caps 4, two pistons 5 and the housing 1.

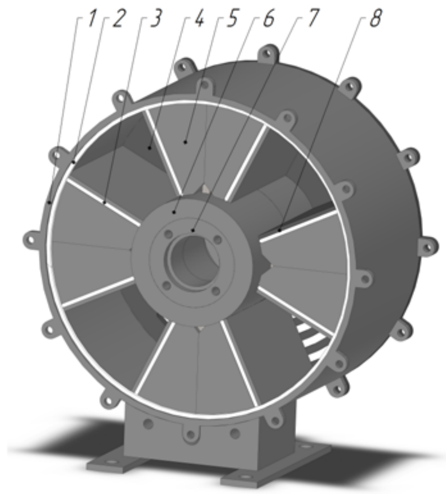

Fig. 1. Vanes group RVE:
1 – housing; 2 – sealings housing; 3 – sealings pistons; 4 – end covers; 5 – pistons; 6, 7 – shafts; 8 – pressure plates.

When designing a new heat engine, one of the first stages of calculation is necessary to determine the temperature and pressure in working chambers.

Existing methods of calculating the parameters of the operating cycle heat engine with external heat supply based usually on the isothermal mathematical model [1]. In some works at calculations of a running cycle the processes compression and expansion in the isolated volume consider adiabatic [7]. However, it should note that when using a heat engine occurs at constant heat transfer between the working fluid and the walls of the chamber, which has noticeable effect on the temperature and pressure of the working fluid in the chamber. In this paper, we propose a

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mathematical model to calculate the main parameters of the operating cycle engines with an external supply of heat for cycles of expansion and contraction of the working fluid in the isolated volume, which takes into account the heat transfer between the working fluid and the parts that make up the working chamber.

II DIFFERENTIAL EQUATIONS PRESSURE AND TEMPERATURE CHANGES IN THE CHAMBER

The processes of expansion and contraction of the working fluid in the isolated volume described by the first law of thermodynamics, written for the case of constant mass [2]:

$$Mc_V dT + pdV + dQ_w = 0, \tag{1}$$

where M – the mass of the working fluid in the chamber; c_V– heat capacity at constant volume of the working fluid; T– the temperature of the working fluid in the chamber; p– the pressure of the working fluid in the chamber; V– chamber volume; dQ_w– heat given (received) the working fluid by heat exchange with the walls of the chamber.

Considering in order that the working fluid in the chamber is ideal gas we write the equation of state ideal gas:

$$pV = MRT, \tag{2}$$

where R – gas constant.

After a series of transformations of the equations (1) and (2) in accordance with [2], we obtain two differential equations:

$$\frac{dT(\alpha)}{d\alpha} = -\frac{(k-1)T(\alpha)}{V(\alpha)}\frac{dV(\alpha)}{d\alpha} - \frac{1}{Mc_V}\frac{dQ_w(\alpha)}{d\alpha} \tag{3}$$

$$\frac{dp(\alpha)}{d\alpha} = -\frac{kp(\alpha)}{V(\alpha)}\frac{dV(\alpha)}{d\alpha} - \frac{p(\alpha)}{Mc_V T(\alpha(}\frac{dQ_w(\alpha)}{d\alpha}, \tag{4}$$

where k – adiabatic exponent; α– angle of rotation of the output shaft of the engine.

Equation (3), is a differential equation for the variation of temperature in the chamber. Equation (4), is a differential equation of the pressure change in the chamber.

The volume of the working chambers RVE is given by:

$$V(\alpha) = (\psi(\alpha) - \psi_p)c, \tag{5}$$

where $\psi(\alpha)$ – angle between the axes of the pistons; ψ_p– the angular size of the piston; c– design parameter.

According to [2], the angle between the axes of the pistons is determined by the expression:

$$\psi(\alpha) = 2(a + b\cos 2\alpha), \tag{6}$$

where $a = \pi/4$; $b = \pi/4 - \psi_{\min}/2$ and where ψ_{min} – the minimum value of the angle ψ.

With regard to (5), (6) can be written $\frac{1}{V(\alpha)}\frac{dV(\alpha)}{d\alpha} = -\frac{4b\sin 2\alpha}{2a + 2b\cos 2\alpha - \psi_n}$.

Consider the summand $\frac{dQ_w(\alpha)}{d\alpha}$. According to [2]:

$$\frac{dQ_w(\alpha)}{d\alpha} = \frac{\beta F(T(\alpha) - T_w)}{\omega}, \tag{7}$$

where β – the average value of heat transfer coefficient; F– heat exchange surface area; T_w– surface temperature of parts forming a working chamber; $\omega = \frac{d\alpha}{dt}$ – angular velocity.

If the temperature of the surfaces of parts constituting working chamber different, the heat dQ_w is determined by the following relation:

$$\frac{dQ_w(\alpha)}{d\alpha} = \sum_{i=1}^{n}\frac{\beta_i F_i(T(\alpha) - T_{wi})}{\omega}, \tag{8}$$

where β_i – the average value of heat transfer coefficient surface separate part; F_{ic}– heat exchange surface area separate part; T_{wi} – surface temperature separate part; n – number of surfaces, forming a working chamber; i – serial number surface forming a working chamber.

For RVE relation (8) takes the form:

$$\frac{dQ_w(\alpha)}{d\alpha} = \frac{\beta_1 F_1(T(\alpha) - T_{w1})}{\omega} + \frac{\beta_2 F_2(T(\alpha) - T_{w2})}{\omega} + 2\frac{\beta_3 F_3(T(\alpha) - T_{w3})}{\omega} + 2\frac{\beta_4 F_4(T(\alpha) - T_{w4})}{\omega}, \tag{9}$$

where β_1, F_1, T_{w1} – heat transfer coefficient, surface area and surface temperature housing 1 (see. Fig. 1); β_2, F_2, T_{w2}– heat transfer coefficient, surface area and surface temperature shafts 6, 7 (see. Fig. 1); β_3, F_3, T_{w3} – heat transfer coefficient, surface area and surface temperature end covers 4 (see. Fig. 1); β_4, F_4, T_{w4}– heat transfer coefficient, surface area and surface temperature pistons 5 (see. Fig. 1).

Adding equations (3) and (4) in accordance with equation (8) yields:

$$\frac{dT(\alpha)}{d\alpha}=-\frac{(k-1)T(\alpha)}{V(\alpha)}\frac{dV(\alpha)}{d\alpha}-\\-\frac{1}{M\cdot c_V}\sum_{i=1}^{n}\frac{\beta_i F_i\left(T(\alpha)-T_{wi}\right)}{\omega} \quad (9)$$

$$\frac{dp(\alpha)}{d\alpha}=-\frac{k\cdot p(\alpha)}{V(\alpha)}\frac{dV(\alpha)}{d\alpha}-\\-\frac{p(\alpha)}{Mc_V T(\alpha)}\sum_{i=1}^{n}\frac{\beta_i F_i\left(T(\alpha)-T_{wi}\right)}{\omega} \quad (10)$$

Equations (9) and (10) must be supplemented by the initial conditions.

The main difficulty is the determination of the unknown β_i and T_{wi}, appearing in the equations (9, 10).

The heat transfer coefficient is a complex function of various quantities characterizing the process of heat transfer. In general this relationship can be expressed as [3]:

$$\beta = f\left(u, T_w, T, \lambda, c_p, \rho, \mu, a, \Phi, l_1, l_2 \ldots\right), \quad (11)$$

where ω – the rate of the working fluid, T_w – wall temperature, T– the temperature of the working fluid, λ– thermal conductivity, c_p– heat capacity at constant pressure of the working fluid, ρ – the density of the working fluid, μ – the dynamic viscosity of the working fluid, a – thermal diffusivity, Φ – function shape of the body, l_1, l_2 – the dimensions of the body.

In most cases, analytically determine the coefficient of heat transfer is not possible. That is why, it is determined experimentally or by numerical simulation. Method of determining the heat transfer coefficient for parts vanes group RVE numerical simulation methods is given in [8].

Each working chamber of the engine is formed by the following components (see. Fig. 1): the two shafts 6, 7, two end caps 4, the housing 1 and the piston 5.

Wall temperature parts, forming a working chamber T_{wi}, varies during engine operation from T_{wimin} to T_{wimax}.

III DETERMINATION OF THE TEMPERATURE OF PARTS OF THE WALLS FORMING THE CHAMBER

The minimum temperature T_{wi} of the wall partsforming a working chamber, have at the time the engine is started and it is equal to the ambient temperature T_{atm}. After which they exchange a heat with the working fluid warmed and gradually heated to a temperature T_{wimax}, at which further operation of the engine is not changed.

For an approximate determination T_{wimax} can be assumed that the heat exchange of the working fluid with the walls working parts forming the chamber stationary. Then, as the temperature of the working fluid should be adopted mean-temperature working fluid in one cycle:

$$T_{ave}=\int_{\alpha=0}^{2\pi} T(\alpha)_{lim}\,d\alpha, \quad (12)$$

where $T(\alpha)_{lim}$ – the temperature of the working fluid calculated without heat exchange with the walls of the working fluid.

Further, it is necessary to determine the approximate density of the heat flow through the parts forming a working chamber. Methods for determining the density of heat flux through parts with by stationary heat transfer depends on the shape of the items (a flat, cylindrical or spherical) and given in [3]. Knowing the density of the heat flow in details and the heat transfer conditions on its borders, we can determine the temperature of the wall part which is in contact with the working fluid.

Next, determine the temperature field in the body parts 1 (see. Fig. 1) for the design scheme shown in Fig. 2.

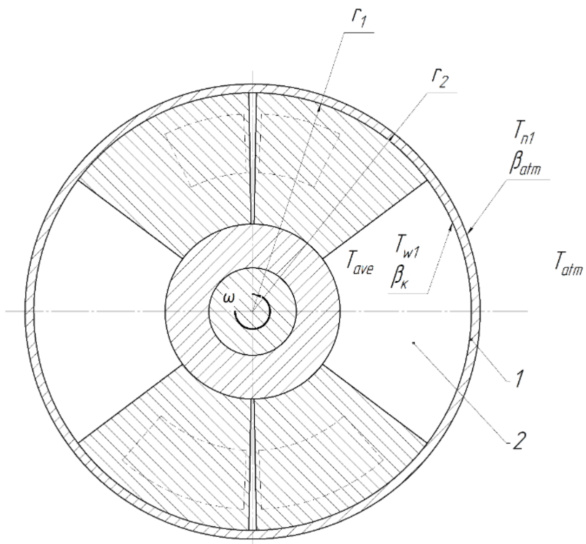

Fig. 2. Design scheme:
1 – housing; 2 – working chamber.

Heat flow density on the inner and outer surfaces of the housing 1 is given by [3]:

$$q_{in}=\frac{\lambda\left(T_{w1}-T_{n1}\right)}{r_1\ln\left(\frac{r_2}{r_1}\right)} \quad (13)$$

$$q_{out} = \frac{\lambda(T_{w1} - T_{n1})}{r_2 \ln\left(\frac{r_2}{r_1}\right)}, \quad (14)$$

where q_{in} – heat flow density on the inner surface of the housing; q_{out}– heat flow density on the outer surface of the housing; T_{w1}– the temperature of the inner surface of the housing; T_{n1}– temperature of the outer surface of the housing; r_1– the inner radius of the housing; r_2– the outer radius of housing; λ – thermal conductivity.

Adding to the equation (13), (14) the boundary conditions of the 3rd kind, written for the outer and inner surfaces of the case, we obtain a system of equations:

$$\begin{cases} q_{in} = \dfrac{\lambda(T_{w1} - T_{n1})}{r_1 \ln\left(\frac{r_2}{r_1}\right)} \\ q_{out} = \dfrac{\lambda(T_{w1} - T_{n1})}{r_2 \ln\left(\frac{r_2}{r_1}\right)} \\ q_{in} = \beta_h(T_{w1} - T_{ave}) \\ q_{out} = \beta_{atm}(T_{n1} - T_{atm}) \end{cases}, \quad (15)$$

where β_h – heat transfer coefficient of from the working fluid to the inner surface of the housing; β_{atm} – heat transfer coefficient of from the outer surface of the housing to the environment.

From equations (15) is defined T_{w1}, which is the maximum wall temperature during operation of the engine. To determine the maximum wall temperature of the other parts forming the working chamber is necessary to solve a system of equations such as (15).

We shall show the calculation of the maximum temperature of the wall housing parts 1 (Figure 1) for the layout RVE 10 kW.

The input parameters for the calculation are: T_{ave}=398,2 K; T_{atm}=293 K; r_1=125 mm; r_2=130 mm; material - steel 40; λ=51 W/(m*K); working body - the air.

The numerical value of the heat transfer coefficient β_h defined in [8] and is equal to β_h=38,34 W/(m^2*K). The numerical value of the heat transfer coefficient β_{atm} determined by the method described in [3] and is equal to β_{atm}=10 W/(m^2*K).

For a given input parameters temperature on the inner surface of the housing is equal to T_{w1}=376,5 K.

Numerical values wall temperatures for all the items forming the working chamber shown in the table.

THE NUMERICAL VALUES OF THE WALL TEMPERATURE

Part Name	Part number, i	The wall temperature T_{wi}, K	
		minimal, T_{wimin}, K	maximum, T_{wimax}, K
Housing 1 (Figure 1)	1	293	376,5
The rotors 6, 7 (Figure 1)	2	293	388,7
The end cap 4 (Figure 1)	3	293	377,2
Piston 5 (Figure 1)	4	293	398,2

IV NUMERICAL SIMULATION

The numerical solution of differential equations (9) and (10) was carried out in the system with the help of Mathcad functions Odesolve designed to solve linear differential equations for the highest derivative Runge-Kutta methods.

Let us show the results of calculations for the compression cycle in RVE power of 10 kW. Compression cycle in RVE runs at an angle of rotation of the output shaft $0^\circ \le \alpha \le 45^\circ$. Pressure and temperature in the chamber at the beginning of the compression cycle are respectively, p_0=1 and T_0=293 K.

The input parameters for the calculation are:k=1,35; R=287 J/(kg·K); c_v=82 J/(kg·K); M=1,22 10^{-3} kg; ω=180 rpm; β_1=38,34 W/(m^2·K); F_1=0,019 m^2; β_2=38,34 W/(m^2·K); F_2=0,0077 m^2; β_3=35 W/(m^2·K); F_1=0,008 m^2; β_1=36,7 W/(m^2·K); F_1=0,009 m^2.

Solve the equation (9), (10) with T_{wi} of equal T_{wimin}, which corresponds to engine start, and with T_{wi} of equal T_{wimax}, which corresponds to the steady operation of the engine.

Graphs of temperature and pressure in the working chamber during compression of the working fluid at temperatures of walls T_{wimin} and T_{wimax} are shown in Figure 3.

It may be noted that a change in temperature of the walls of the working chamber from T_{wimin} to T_{wimax} the temperature in the chamber at the end of the compression stroke is changed from $T(45^\circ)$=367,7 K to $T(45^\circ)$=377,4 K, that is approximately 2,7%, the pressure is changed from $p(45^\circ)$=2,44 atm to $p(45^\circ)$=2,51 atm, that is approximately 2,8%. Thus we can conclude that transient temperature of the chamber walls during operation RVE layout power of 10 kW leads to a change in temperature and pressure in the working chamber is not more than 3%.

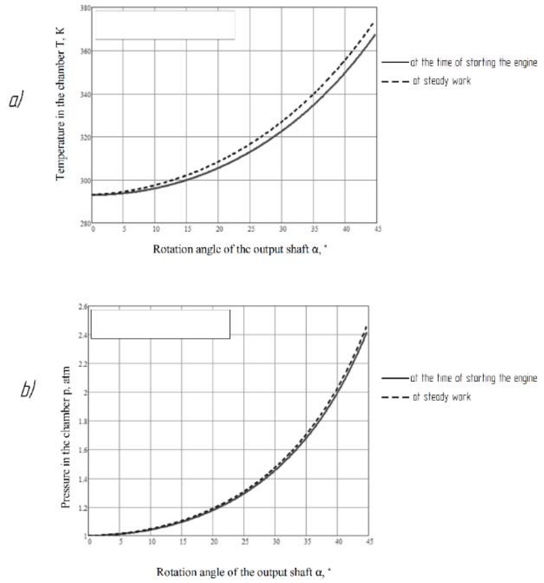

Fig. 3. Graphs of temperature and pressure:
a) the chamber temperature dependence of the rotation angle of the drive shaft; b) the chamber pressure dependence of the angle of rotation of the drive shaft.

V CONCLUSIONS

In this paper, the mathematical model for calculating the pressure and temperature in the chamber during the processes of expansion and contraction of the working fluid in an isolated volume considering heat transfer the working fluid from the chamber walls. Is given plot the schedule of dependence pressure and temperature in the chamber of the rotation angle of the output shaft at the minimum and maximum wall temperature. Found that transient temperature of the chamber walls during operation RVE layout power of 10 kW leads to a change in temperature and pressure in the working chamber is not more than 3%.

These calculations are needed to calculate the thermal stress of the engine parts, as well as to describe the gas exchange occurring in the rotary-vane engine.

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