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Research of Optimal Operating Regimes of Power Micro Gas Turbine Installation with Heat Recuperation

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In this paper, power micro gas turbine heat recovery unit operation modes have been studied. An approach to determine the optimal unit turbocompressor rotor speed, significantly influencing its design concept, was offered. According to the results of mathematical modeling, dependences of the power micro gas turbine unit characteristics on turbocompressor rotor speed have been obtained.

Keywords: **power micro gas turbine unit, turbocompressor, optimal operation mode, mathematical modeling**.

When calculating optimal operating modes for the micro gas turbine unit (MGTU), it is reasonable to assess the turbocompressor rotor maximum speed, significantly influencing the turbocompressor's design and its geometrical and power parameters. The fact is that an increase in the rotor speed increases the compressor's compression ratio, reduces dimensions and moment of inertia of the turbocompressor rotor, which favorably influences MGTU's dynamic characteristics generally.

However, increasing the rotational speed, firstly, is limited by tensile strength of structural materials, and secondly, by the relative velocity value at the inlet of the compressor impeller, which - in all MGTU operation modes – should remain subsonic.

Maximum relative velocity at the inlet of the impeller is identically determined by the circumferential and absolute velocities at the periphery of the leading edge of the impeller vane. Circumferential velocity component is identically determined by the rotor speed and the diameter of the leading edge at the periphery, and the absolute velocity value and direction are uniquely determined by the air flow through the compressor and by direction of the absolute velocity vector at the inlet of the impeller.

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Thus, structurally compressor shall consist of Stationary Blade Row (SBR) and a centrifugal or an axial-centrifugal impeller, then the relative velocity vector can control rotation speed, flow rate and the blades angle at the outlet of the SBR [1].

To determine the impeller rotation frequency, we will use kinematic ratios at the inlet of the compressor impeller at periphery, and the "flow rate through the compressor" equation [2].

From the velocity triangle (Fig. 1), we can find out the relative velocity vector value by the cosine theorem:

$$w_1^2 = u_1^2 + c_1^2 - 2u_1c_1\cos\alpha_1, \qquad (1)$$

where w_1, u_1, c_1 are relative, circumferential and absolute velocities at the inlet of the impeller at the peripheral diameter D_1, and α_1 is the angle of the blade at the periphery of the SBR.

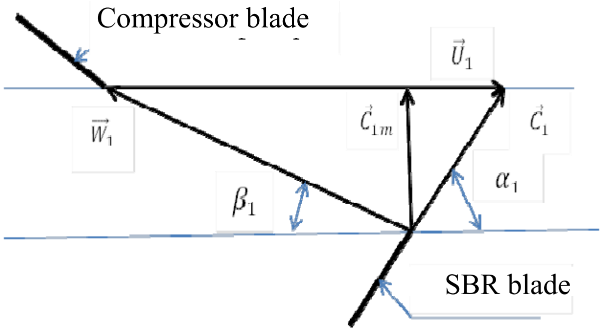

Fig.1. Velocity diagram at the impeller vane periphery

Let's divide the equation (1) by the squared speed of sound:

$$a^2 = kRT \cdot \tau(M_{c1m}/\sin\alpha_1), \qquad (2)$$

after conversion of which, we obtain a quadratic equation of the circumferential velocity:

$$u_1^2 - \frac{2M_{c1m}a(M_{c1m})}{tg\,\alpha_1} + \left(\frac{M_{c1m}^2}{(\sin\alpha_1)^2} - M_{w1}\right) \times (a(M_{c1m}))^2 = 0, \qquad (3)$$

where M_{c1m}, M_{w1} is Mach number in the meridional and relative motion, k = 1,4 is an isentrope indicator, R = 287,3 $\frac{J}{kg\cdot K}$ is a gas constant, $\tau(M_{c1m}/\sin\alpha_1)$ is a gas- dynamic function of the temperature ratio.

The solution of equation (3) defines the maximum peripheral speed:

$$u_1 = \frac{a(M_{c1m})}{tg\,\alpha_1}M_{c1m} + \sqrt{\left(\frac{a(M_{c1m})}{tg\,\alpha_1}\right)^2 M_{c1m}^2 - \left(\frac{M_{c1m}}{(\sin\alpha_1)^2} - M_{w1}\right)^2 (a(M_{c1m}))^2} \qquad (4)$$

at the impeller blade leading edge periphery, as "-" sign gives negative circumferential speed value.

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It should be noted that knowing the circumferential speed does not allow one strictly determining the speed of turbocompressor rotor rotation $\omega = \frac{2u_1}{D_1}$, which depends on the impeller inlet diameter D_1.

To select a single value of the peripheral speed, we use the flow rate equation:

$$\dot{m}_{\text{в}} = \frac{A(k)\sigma_1 P_1^* F_m q(M_{c1m}/K_m)}{\sqrt{RT_1^*}}, \tag{5}$$

where $A(k) = \sqrt{k(\frac{2}{k+1})^{\frac{k+1}{k-1}}}$ is a gas-dynamic factor, σ_1 is a pressure recovery coefficient, $q(M_{c1m}/K_m)$ is a gas-dynamic mass flow density function, K_m is a coefficient reflecting the non-uniformity of axial velocity components and flow obstruction by blades,

$$F_m = \frac{\pi D_1^2}{4}\left(1 - \bar{d}_{\text{вт}}^2\right) = u_1^2 \frac{\pi(1-\bar{d}_{\text{вт}}^2)}{\omega^2} \tag{6}$$

meridian section area , $\bar{d}_{\text{вт}}$ is relative diameter of the sleeve.

After substituting (6) in (5), and enabling the resulting expression with respect to the squared circumferential speed, we get the following:

$$u_1^2 = \frac{\dot{m}_{\text{в}}\sqrt{RT_1^*}\,\omega^2}{A(k)\sigma_1 P_1^* \pi(1-\bar{d}_{\text{вт}}^2) q(M_{c1m}/K_m)}. \tag{7}$$

Let’s square expression (4) and equate it to the right side of (7), and then let‘s transform the resulting equation, so that the left side of the equation contained only original and required parameters. We obtain two equivalent complexes:

$$\frac{\dot{m}_{\text{в}}\omega^2}{\sigma_1 P_1^*\left(1-\bar{d}_{\text{вт}}^2\right)\sqrt{T_1^*}} = f(M_{c1m}, M_{w1}, \alpha_1) =$$

$$= \left[\frac{M_{c1m}}{tg\,\alpha_1} + \sqrt{\left(\frac{M_{c1m}}{tg\,\alpha_1}\right)^2 - \left(\frac{M_{c1m}}{sin\,\alpha_1}\right) - M_{w1}}\right]^2 \times$$

$$\pi k\sqrt{R}\tau\left(\frac{M_{c1m}}{\sin\alpha_1}\right) q\left(\frac{M_{c1m}}{K_m}\right). \tag{8}$$

Let’s solve equation (8) with respect to the turbocompressor rotor speed:

$$\omega = \sqrt{\frac{[f(M_{c1m}, M_{w1}, \alpha_1)]\sigma_1 P_1^*(1-\bar{d}_{\text{вт}}^2)\sqrt{T_1^*}}{\dot{m}_{\text{в}}}}. \tag{9}$$

Obviously, the maximum rotational speed of the rotor will be reached when function $f(M_{c1m}, M_{w1}, \alpha_1)$ has a maximum value. Dependences of function $f(M_{c1m}, M_{w1}, \alpha_1)$ on Mach number M_{c1m} at different values of M_{w1} *and* $\alpha_1 = 75°$ are shown in Fig. 2, and at different values of α_1 and at $M_{w1} = 0{,}9$ are shown in Fig. 3.

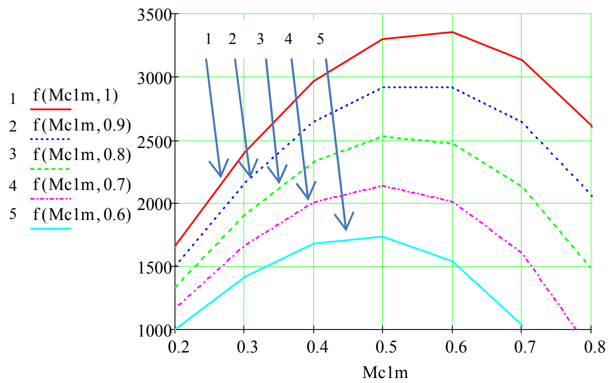

Fig. 2. Dependence of function $f(M_{c1m}, M_{w1}, \alpha_1)$ on M_{c1m} at $\alpha_1 = 75°$ and different values of M_{w1}

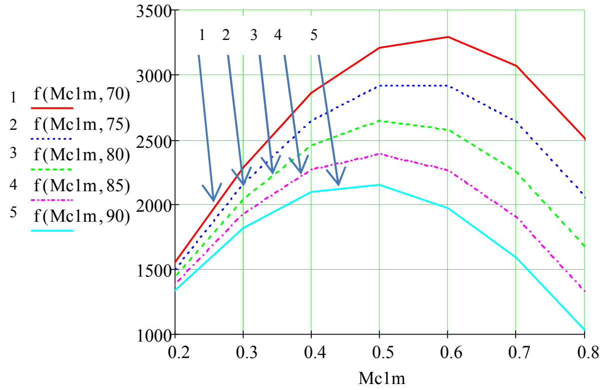

Fig. 3. Dependence of function $f(M_{c1m}, M_{w1}, \alpha_1)$ on M_{c1m} at $M_{w1} = 0.9$ and different values of $\alpha_{1_{w1}}$

Analysis of the graphs in Figures 2 and 3 shows that for providing subsonic flow at the inlet of the compressor impeller, $M_{w1} = 0{,}9$ should be set, and blade angle of the SBR should be selected from interval from 75° to 90°, in spite of the fact that with decreasing angle α_1 rotor speed increases. The fact is that with decreasing α_1, inlet flow backspin in the direction of impeller rotation increases, which leads to lower degree of compressor compression, and increases the pressure loss at SBR that reduces the pressure at the inlet of the impeller. This reduces the total pressure at the outlet of the compressor and the degree of pressure reduction in the turbine, which leads to an increase in air flow and reducing the rotor speed. Obviously, the rotor speed increases with a decrease in the relative diameter of the sleeve and with increasing air temperature. Dependence of the rotor speed at different values of the relative diameter of the sleeve is shown in Figure 4. It should be noted that rotation speed results obtained do not depend on other variable MGTU parameters.

For increasing MGTU efficiency, it is expedient to use part of heat energy after the turbine to increase the heat content of the air entering the combustion chamber [3]. With the increase in the heat content of air, fuel consumption reduces, but the air flow increases due to the increased pressure loss at the recuperator and reduced expansion ratio at the turbine.

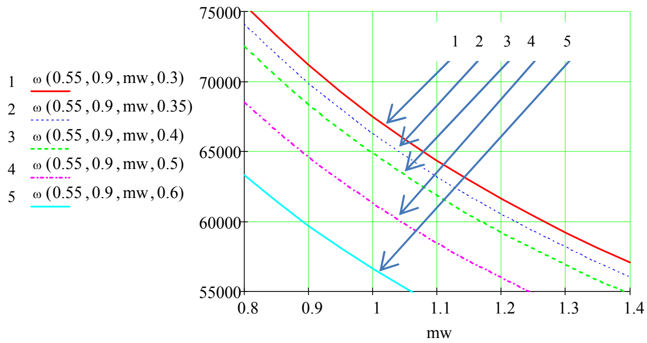

Fig. 4. Dependence of the turbocompressor rotor limit speed on flow at different values of the relative diameter of the sleeve at $\alpha_1 = 75°$

As a result of the calculation of the MGTU heat recovery with recuperation, dependences of the air flow and hourly fuel consumption depending on the regeneration degree shown in Fig. 5 and 6 have been defined.

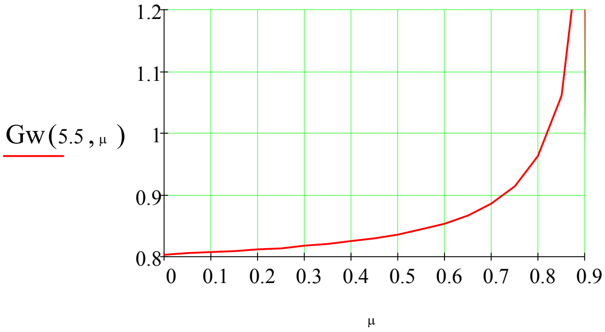

Fig. 5. Air flow dependence on recuperation degree

To find out the maximum MGTU turbocompressor rotor speed, it is enough to find out the degree of regeneration corresponding to the minimum hourly fuel consumption acc. to the schedule (Fig. 6). Then, based on recuperation degree, we can find out the air flow acc. to the schedule (Fig. 5). And finally, based on schedule (Fig. 4), we can determine the MGTU turbocompressor rotor maximum speed.

For further research, a centripetal turbine has been chosen for the following reasons:

1. At low capacities, centripetal turbines' efficiency is higher than axial turbines' efficiency.

2. In centripetal turbines' stages, we can provide a bit bigger thermal gradient, because at identical stresses in the impeller, circumferential velocities in centripetal turbines may be bigger than in the axial ones.

3. Impellers of centripetal turbines are easier to manufacture and are more reliable due to the small number of blades and a simple configuration.

The turbine parameters should be defined based on a number of requirements: high efficiency, the necessary strength, constructability and simple design. In a number of these requirements, often conflicting, obtaining the turbine high efficiency is one of the main tasks.

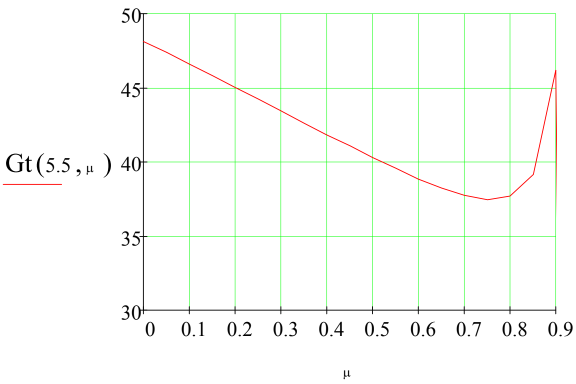

Fig. 6. Dependence of hourly fuel consumption on the regeneration degree

Let's estimate the effect of turbocompressor rotor speed on turbine geometry through the modal parameter:

$$\bar{u}_1 = \frac{u_1}{c_{ад}} = \frac{\pi d_1 n}{\sqrt{2c_p T_0^*\left[1-\left(\frac{p_2}{p_0^*}\right)^{\frac{k-1}{k}}\right]}}, \qquad (10)$$

where $c_{ад}$ is nominal adiabatic speed, u_1 is circumferential speed on the diameter d_1, d_1 is inlet diameter of the turbine wheel, T_0^*, p_0^* are total pressure and temperature of combustion products entering the turbine, p_2 is static pressure at the turbine outlet.

The modal parameter value optimal from the point of view of obtaining the maximum centripetal turbine efficiency can be found out by the formula [4]:

$$\bar{u}_{1\,опт} = \frac{\psi}{\sqrt{\mu^2\left(\frac{(\cos\beta_2)^2}{m^2}-\psi^2\right)+\frac{(1-m^2\psi^4)\varphi^2(\cos\alpha_1)^2}{m^2\psi^2(1-\varphi^2)}}}, \qquad (11)$$

where φ, ψ are velocity coefficients in the nozzle unit and in the impeller, μ is a radiality coefficient, α_1 is a flow angle in absolute motion at the inlet to the impeller, β_2 is a flow angle in relative motion at the output of the impeller, m is a value characterizing elements of the velocities triangle at the turbine outlet:

$$m_{опт} = \frac{1}{\psi^2}\left[1-\sqrt{\frac{\mu^2(1-\varphi^2)(1-(\cos\beta_2)^2\psi^2)}{(\cos\alpha_1)^2\varphi^2+\mu^2(1-\varphi^2)}}\right]. \qquad (12)$$

Analysis carried out in [3] shows that after defining $\bar{u}_{1\,опт}$ acc. to formula (11), the following value may be used in the centripetal turbine calculation:

$$\bar{u}_{1\,расч} = \bar{u}_{1\,опт} - (0 \div 0{,}1)\ , \qquad (13)$$

which implies the turbine impeller maximum circumferential speed reduction, which is beneficial to its strength characteristics.

Let's express diameter d_1 from (10), replacing $\bar{u}_{1\,опт}$ with $\bar{u}_{1\,расч}$:

$$d_1 = \frac{\bar{u}_{1\,расч} c_{ад}}{\pi n}. \qquad (14)$$

Assuming the turbocompressor rotor speed for different angles α_1 and β_2, we can find out diameter d_1 (and consequently, approximate dimensions of MGTU), wherein the turbine efficiency will be maximized.

Figure 7 shows results of calculating d_1 for the input data given in [1], and the three combinations of angles α_1 and β_2. Value $\bar{u}_{1\,расч}$ was determined by the formula:

$$\bar{u}_{1\,расч} = \bar{u}_{1\,опт} - \ 0{,}08. \qquad (15)$$

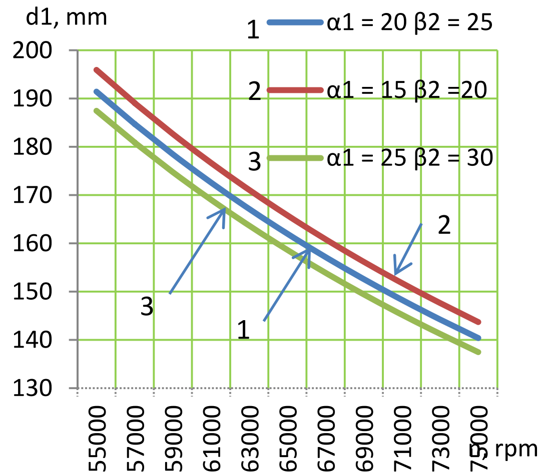

Fig. 7. Influence of α_1 , β_2, n on the turbine impeller geometrical dimensions provided ensuring maximum efficiency

For the case $\alpha_1 = 15°$ and $\beta_2 = 20°$ optimal circumferential velocity was $u_{1\,опт} = 564\ m/s$, for $\alpha_1 = 20°$ and $\beta_2 = 25°$ $u_{1\,опт} = 551\ m/s$, for $\alpha_1 = 25°$ and $\beta_2 = 30°$ $u_{1\,опт} = 540\ m/s$. These values can be considered satisfactory as for the turbine impeller strength.

Thus, the following conclusion can be made based on the results of the mathematical modeling:

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1. We have obtained ratios enabling – at the stage on thermal calculation – estimating the limiting turbocompressor rotor speed providing subsonic flow at the inlet to the compressor in all MGTU operating modes.
2. We have offered a method for calculating the limiting turbocompressor rotor speed of MGTU with regeneration, providing the minimum hourly fuel consumption.
3. We have given the methodology for assessing the maximum circumferential velocity and the corresponding turbine impeller input diameter providing the highest efficiency.

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