Disc friction to specify power balance of a turbopump unit of LPE

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Abstract. The research elaborates the main characteristics of the liquid propellant engines; it determines that some turbopump units have got the frequency of rotor spinning up to 100 000 and even 120 000 r.p.m. The authors analyse the current methodologies to specify losses connected with disc friction. The methodologies determining disc friction coefficient often obtain criterion-empirical character, and due to increasing a number of rotor spinning of turbopump unit of the liquid propellant engines, they reflect the specification reliability inaccurately. The research demonstrates equations to determine disk friction coefficients based on considering the theory of three-dimensional boundary layer to a rotational flow taking into account the turbulent process of velocity distribution. The authors give recommendations on determining an index of velocity distribution degree depending on Reynolds rotation criterion of a disc and plug flow.

1. Introduction

The major part of the existing methodologies to calculate flow parts of turbopump units (TPU) of the liquid propellant engines (LPE) is of criterion-empirical character; it is used for the nod boundary conditions and feeding units of the liquid propellant engines obtaining rotation frequency approximately 20 000 – 40 000 rotations per minute. The current TPU of the liquid propellant engines obtain comparatively high rotation frequency and it can be up to 120 000 rotations per minute. The fact does not consider the degree of the turbulent process of flow velocity distribution in the three-dimensional boundary layer of the elements of the flow range of the feeding units and it requires additional research and specifications of the used dependencies due to changing boundary conditions.

While designing current rocket engines and converting to environmental fuel components, there is a tendency to increase rotation frequency of TPU. That results in increasing in pressure produced by a pumping stage, growing a pump speed coefficient as upgrading the efficiency of the pump and the turbine, decreasing the dimensions such as reducing the axial and radial loads to the rotor and as a consequence it leads to TPU mass reduction in general.

Multiple discrepant values are provided by the numerous experimental and theoretical research conducted by scientists within a range of geometrical and performance parameters of flows in the rotor-stator gap [1-9]. Some methodologies to calculate resistant torque obtains various disadvantages first of all connected with applying empirical coefficients; they do not consider the availability of the radial component of the absolute velocity in the side cave.
One of the main tendencies to develop aerospace engineering is to improve the design quality based on using the current calculation methodologies and mathematical models reflecting the processes in both separate units and in the whole airborne vehicle in general.

While designing, the requirements to TPU configure due to the assignments performed by the power unit (PU), where TPU is an element constituent and with which it assembles as a single block.

The requirements to the PU belong to TPU as well: to provide feeding fuel components with the flow requirement and pressure under high degree of reliability and efficiency in all operational modes of the engine; to provide minimal dimensions and mass; the construction simplicity and minimal cost.

2. Specifying disk friction coefficient

The power balance of a centrifugal-type pump are consider where the power output of the centrifugal-type pump is calculated as

\[ N_{out} = N_{req} - N_{cont} - N_d - N_{leak} - N_{hydr}, \]

where \( N_{req} \) - the power required of the centrifugal-type pump, \( N_{cont} \) - contact friction loss power, \( N_d \) - disc friction power of a rotor wheel, \( N_{leak} \) - lost power to the working fluid leakage through seal groups and \( N_{hydr} \) - hydrodynamic power loss.

Calculates a disk friction coefficient taking into account increasing rotation frequency of TPU of LPE. While realizing turbulent flow, originally, the power law of airfoil shape distribution of speed in the boundary layer [10] is applied

\[ \frac{u}{U} = \left( \frac{y}{\delta} \right)^{m} \]  

(1)

Depending on the shape of the velocity airfoil \( m \), it is necessary to overdetermine the dependence of friction induced shear stress close to the inside wall in the boundary conditions of the turbulent boundary layer \( \tau_0 \). Based on the expression for the friction stress [11, 3]

\[ \frac{\tau_0}{\rho U^2} = 0.01256 \left( \frac{U \delta_{w}^{**}}{v} \right)^{0.25} \]  

(2)

The considered flow in the side cave between rotor disk and TPU body of LPE belongs to rotational flow according to the “solid body” law \( \frac{U}{R} = \omega = const \). The momentum losses thickness rotationally with a certain airfoil shape for the wall is determines as

\[ \delta_{uw}^{**} = 0.04535 \left( \frac{4M^2 - 7L}{1 + H} \right)^{0.4} \left( \frac{2}{J} + \frac{1}{L} \right)^{0.8} \left( \frac{v}{\omega_n} \right)^{0.2} \frac{R^{0.6}}{R} = \]  

(3)

where \( M, L, H, J, K \) – relative characteristic quantities of the dynamic three-dimensional boundary layer [10].

Denote the parameter

\[ A = \left( \frac{18m^3 + 99m^2 - 171m}{m^2 + 18m^2 + 105m + 200} \right)^{\frac{m}{3}} \left( \frac{7m^3 + 76m^2 + 253m + 240}{18m^2 + 72m + 54} \right)^{\frac{1}{3}} \]  

\[ \frac{1}{Re_h^{2}} \]
The momentum losses thickness rotationally with a certain airfoil shape for a disc is

$$
\delta_{od}^{\omega} = \left\{ \frac{5}{3} \cdot \frac{0.01256}{1+H} \frac{(2+1/L)}{\sqrt{3LJ+4L(K-2J)}} \right\}^{0.8} = 
$$

$$
= 4,492 \cdot 10^{-3} \left[ \frac{(m+5)(7m^2+41m+48)}{\sqrt{9m(2m^2+9m+5)(m+1)(m+3)}} \frac{8}{9} \right] \frac{1}{R_d^4} \frac{\omega^4}{\omega_d^4} 
$$

where \( \omega = \omega_0 - \omega_d \), \( \omega_d \) – disk angular velocity, \( \omega_r \)– flow angular velocity, and Reynolds criteria for disk \( \text{Re}_d = \frac{\omega R^2}{\nu} \).

Denote the parameter

$$
B = \left[ \frac{(m+5)(7m^2+41m+48)}{\sqrt{9m(2m^2+9m+5)(m+1)(m+3)}} \frac{8}{9} \right] \frac{1}{R_d^4} \frac{\omega^4}{\omega_d^4} 
$$

Assuming the expressions for the momentum thickness rotationally, the friction induced shear stress on the wall is determined as

$$
\tau_{\omega a}^w = \frac{0.02722 \rho \nu^2 \text{Re}_w^9}{AR^2} \quad \text{(5)}
$$

and on the disc it is

$$
\tau_{\omega a}^d = \frac{0.048515 \rho \nu^2 \text{Re}_d^9}{BR^2} \quad \text{(6)}
$$

The equation for disc friction loss coefficient for a rotor wheel is

$$
C_{f} = \frac{\tau R^2}{\rho \text{Re}^3 \nu^2} \quad \text{(7)}
$$

Therefore, the friction coefficient equation for a rotational wall is

$$
C_{f\omega}^{w} = \frac{0.02722}{A \text{Re}_w^3} \quad \text{(8)}
$$

and for the friction coefficient for a disc rotationally, the equation is

$$
C_{f\omega}^{d} = \frac{0.048515}{B \text{Re}_d^3} \quad \text{(9)}
$$
Should to take into account that when the disc losses is depend on the disc and wall frictions [5], therefore, the disc friction loss coefficient is determined as

\[ C_M = C^{\mu a} + C^{\nu a} \quad \text{(10)} \]

Also should to take into account the following: to determine disc friction loss coefficient and, thus, disc losses should be aware of distributing angular velocity of the plug flow \( \omega_f \) depending on disc angular velocity \( \omega_d \), that requires to solve a differential equation [10]

\[ \frac{d\omega_f}{dR} = -\frac{2\pi}{\rho V_{\text{look}}} \left( \tau_{\omega a}^{\text{wall}} + \tau_{\omega a}^{\text{wall}} \right) - \frac{2\omega_f}{R}, \quad \text{(11)} \]

or taking into account the equation for the shear stress of friction at the wall and the disc, the equation is

\[ \frac{d\omega_f}{dR} = -\frac{2\pi}{\rho V_{\text{look}}} \left( \frac{0.02722 \rho \nu^2 \text{Re}_a^\frac{3}{2}}{AR^2} - \frac{0.048515 \rho \nu^2 \text{Re}_d^\frac{3}{2}}{BR^2} \right) - \frac{2\omega_f}{R}, \quad \text{(12)} \]

where \( \dot{V}_{\text{look}} \) - seal leakage.

A sophisticated assignment is to specify analytical dependence of angular velocity of the plug flow \( \omega_f \) on the disc angular velocity \( \omega_d \). For practically important occasions as the first approximation for the non-leakage flow while determining disc friction coefficient should apply \( \omega_f = 0.5\omega_d \), then expression is resulted as

\[ C_M = \frac{1}{\left(0.5\text{Re}_d\right)^\frac{1}{5}} \left( \frac{0.02722}{A} + \frac{0.048515}{B} \right). \quad \text{(13)} \]

Figure 1 presents the dependence of disc friction coefficient for the turbulent flow if \( \text{Re} > 10^5 \).
Should be emphasized that if \( m = 6 \) function coincides with the dependence by L.A. Dorfman [4]. The dependence by F. Schulz-Grunow [2] is located lower than the obtained dependences. Should to take into account that the obtained equations demonstrate a good convergence with the results, obtained by other researchers and according to experimental data, for the different degrees of distributing the velocity traverse under the turbulence flow.

A separate assignment is to determine an airfoil shape \( m \) depending on the criterion Re of the plug flow and Re of the disc. When the disc friction and thermal efficiency in the rotary cavities are specified depending on the mode parameters (Reynolds criterion) of the earlier obtained airfoil shape degree of the function of distributing flow velocity, recommended to consider \( m = 9 \div 11 \) at the disc and \( m = 7 \div 9 \) at a wall. Reynolds criterion at the disc is determined with angular velocity of a pump wheel disc (4) or a turbine.

3. Conclusion

The obtained equations for the disc friction loss coefficients allow to determine the efficiency loss to the disc friction in the TPU of LPE centrifugal-type pumps. Comparing with the empirical dependencies obtained by other researchers, using degree \( m \) of the turbulent dynamic three-dimensional boundary layer significantly increase the range of the area and reliable determination of the disc friction.

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