THE ANALYSIS OF POWER INFLUENCE ARISING FROM VISCOUS FRICTION ON DYNAMICS OF THE DISK TURBINE

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Introduction.

In the foothill regions of Kazakhstan there is a dense network of spillways for technical and domestic needs. For a solution of the problem of environmentally friendly transformation of energy of water flow, in particular mountain spillways are developed new designs of the water-wheels making the minimum impact on structure of water and allowing to reduce cavitational processes. The main limiting factor of the organization of industrial production of disk water-wheels and their modernizations is lack of techniques of their calculation and design.

Material and research methods.

Traditional (diagonal, radial-axial, propeller, bucket etc.) the water-wheels of micro hydro stations using energy of spillways on pipelines, create intensive cavitation in water. It results not only in fast wear of shovels of turbines (for providing their operational characteristics expensive high-strength materials are used), but also to decrease in quality of water. The last is caused by destruction of natural internal molecular structure of water [7].

In method of calculation of existing traditional turbines for a basis physical interaction of a stream of liquid with a barrier is accepted and a number of assumptions is accepted. It is supposed that liquid isn't squeezed, the stream is continuous, that is a current continuous, established in which speeds in all points and sections remain constants in size and the direction and forces of viscosity of liquid are absent - liquid is accepted ideal [2, 6].

Designs of disk hydrocars [4] which can be used both as pumps, and as turbines are known.

Theoretical researches of dynamics of disk hydrocars generally concern to the solution of tasks on a liquid current in a ring crack between rotating disks of pumps [3, 5]. These decisions are based on Novye-Stokes's equations with various regional conditions. It should be noted that researches of disk pumps showed their advantages in certain scopes before lopastny by efficiency, and also according to noise and cavitational characteristics [1]. These advantages, obviously, belong and to disk turbines. However so far there is no theoretical development for calculation of power characteristics of disk turbines depending on parameters of a spillway and their design.

Results of researching.

For the analysis of power interaction of liquid with working bodies of the disk turbine we will consider the scheme of interaction of water flow with disks of a rotor of the turbine which in the simplified look is represented in figure 1.

Feature of the disk water-wheel is that it ротор1состоит from a set of flat disks 2 d thickness R radius in n quantity connected in the package which has been rigidly established on a shaft of selection of power which is mounted on bearing support in the case 3 with a diffusor 4. Water from a diffusor 4 under the pressure of N and with an expense of Q moves through a slothole opening 5 width C and H height, getting to gaps between disks size b, untwists a turbine rotor at the expense of powers of viscous friction. Unlike traditional turbines in which rotation of a rotor of the turbine is provided at the expense of forces of reaction of the water flow influencing the blades of the turbine as on a barrier. The water which has transferred kinetic energy to a rotor of the turbine, is removed through the central opening by r radius in the axis Z direction.

Let's consider power ofinteraction of a pressure head stream of liquid with disks of a rotor of the turbine in cylindrical system of coordinates (Z, ρ, φ). Let's consider that the turbine has one degree of freedom and can rotate only round an axis Z. Let's accept consolidations ideal, and we won't take into account liquid interaction with constructive elements of fastening of disks on a shaft of selection of power of a rotor of the turbine.

The element of viscous liquid the area of df moves in a gap between disks b on a certain trajectory *6* which is described radius vector ρ. The last is function of an angle of rotation of a rotor of the turbine φ.

Powers of viscous friction (coupling) between a surface of disks and liquid will be the main driving forces providing movement of rotors of the turbine. Elementary power of viscous friction for a single element on known dependences it is possible to write down in a look

$$dF_\mu = 2\frac{\mu}{b}df dv, \tag{2}$$

where μ – the coefficient of viscous friction depending on viscosity of liquid, temperature, a material and a roughness of a surface of disks, pressure sizes, obviously, it can be defined only experimentally; df – the element area (as the element of liquid interacts in a gap at the same time with two disks, the coefficient 2 is accepted); dv - the absolute speed of an element of liquid.

The elementary power of viscous friction will create a torque on a turbine rotor concerning an axis of rotation of Z

$$dM_\mu = dF_\mu \rho = 2\frac{\mu}{b}df dv_\varphi \rho, \tag{3}$$

where dv_φ - a tangential component of speed of an element of liquid.

For definition of a total torque it is necessary to know regional values of speed of liquid on an entrance to gaps between disks and at the exit from them. Taking into account a consumption of liquid on an entrance to gaps between disks of value of tangential speed it is possible to write down in the following look.

$$v_{\varphi R} = \frac{Q}{h(c-nd)}, \tag{4}$$

where d – thickness of disks. Accepting the angular speed of disks of a constant for a certain consumption of liquid and forces of resistance, its value can be calculated on a formula

$$\omega = \frac{Q}{h(c-nd)R'}, \tag{5}$$

here R'– the specified radius, which physical sense consists in the following. If the turbine works in a single mode, the speed of rotation of a rotor will be maximum, and turbine capacity minimum, that is energy of a stream of liquid will be spent only for friction overcoming in bearings. In such mode relative tangential speed of a stream and disks the almost zero. At connection of loading the speed of rotation of a rotor of the turbine falls. Thus there are zones "slippings" of a stream of liquid concerning disks. These zones "slippings" obviously, arise in peripheral areas of disks and are proportional to loadings (useful resistance). The zone "slippings" of elements of liquid comes to an end on some given radius R', further liquid in the tangential direction moves together with disks, without creating the tangent tension and a torque. There is only radial relative movement of liquid to the center of rotation of a rotor.

Expression for the tangential speed of liquid at the exit from an interaction zone with disks of a rotor of the turbine can be written down in a look.

$$v_{\varphi r} = \frac{Qr}{h(c-nd)R'} . \quad (6)$$

Taking into account regional conditions and assumptions the pressure head stream of liquid will disperse a turbine rotor on the given radius of R' and as a first approximation it is possible to write down the equation (3) in an integrated form in the following look.

$$\int_0^M dM_\mu = 2\,\frac{\mu}{b}\int_0^{f'} df \int_{v_{\varphi R}}^{v_{\varphi r}} dv_\varphi R', \quad (7)$$

where $f' = n\pi(R^2 - R'^2)$ – the total area of interaction on which the torque from powers of viscous friction is created.

Considering dependences of (4) and (6), having integrated expression (7), in view of that movement of a rotor of the turbine happens at the expense of counteraction forces to powers of viscous friction, we will receive finally expression for a torque on a turbine rotor from powers of viscous friction.

$$M = \frac{2\mu n\pi(R^2-R'^2)(R'-r)Q}{bh(c-nd)}. \quad (8)$$

Here it should be noted that when idling turbine when there are no forces of the useful resistance, the specified radius will be close to the external radius of the disks R rotor of the turbine. In this case the torque on the turbine is minimum, and the angular speed of rotation is maximum. At the loaded turbine the angular speed of rotation of the turbine falls and, the specified radius respectively decreases, and the torque on the turbine thus increases.

Capacity on a shaft of a rotor of the disk turbine taking into account (5) and (8) will be determined by a following dependence

$$N = M\omega = \frac{2\mu n\pi(R^2 - R'^2)(R'-r)Q^2}{R'b[h(c-nd)]^2}. \quad (9)$$

The schedule of dependence of power of the disk turbine from a consumption of the water *Q [m³/s]* calculated on a formula (9) for the following parameters: *R=0,2м, r = 0,045м, b=0,002м, n=30, c=0.06m, h=0,003м, d =0,0012м* and dynamic viscosity of water at a temperature 10^{o}C equal 1,307 * 10^{-3} Grazed, at R'= *0,18; 0,17; 0,15* m it is presented in figure 2.

From the equation (8), accepting a torque on a turbine shaft equal to a total torque of friction forces and payload and $(R' - r) = (R - r)/2$ as a first approximation is determined the specified radius by the following dependence

$$R' = \sqrt{R^2 - \frac{bh(c-nd)M}{\pi\mu nQ(R-r)}}. \quad (10)$$

In figure 3 the schedule of dependence of the specified radius is submitted to R' from the M torque on a shaft of a rotor of the disk turbine for the parameters stated above, at a consumption of water $Q = 0{,}02\ m^3/s$.

The schedule (fig. 3) shows characteristic influence of size of a torque of M on a rotor of the disk turbine on change of the specified radius, that is on reduction of angular speed of the rotation determined by a formula (5).

Discussions and conclusions

The received analytical dependences allow carrying out the analysis of influence of parameters of pressure head liquid, a design and the geometrical sizes of the disk turbine on its dynamic characteristics. These dependences are applicable for engineering calculations of disk turbines.

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