Journal of Engineering Physics and Thermophysics, Vol. 72, No. 1, 1999

CALCULATION OF WORKING LOADS AND EVALUATION OF THE SERVICE LIFE FOR ICEBREAKER STERN BEARINGS

B. F. Klochkov and V. G. Muramovich

UDC 629.12.037.6

Consideration is given to an original method that makes it possible to calculate the load in any cross section of stern bearings of an icebreaker and is suitable for calculating the parameters of a shaft-line installation and for diagnostics of the technical state of the stern bearings.

1. In putting into operation atomic icebreakers with large propeller-shaft torques, it was revealed that the service life of the rubber strips from which the working surface of stern bearings is assembled is substantially shorter than in icebreakers with conventional power plants and smaller propeller-shaft torques. Experience in the operation of the stern bearings of Arktika atomic icebreakers shows that the selected design of the bearing and the material for manufacturing the strips of its assembly do not correspond to the loads in the bearing as far as the service life is concerned. The character of failure for the strips of stern bearings of the above type indicates a strong action on them from the direction of the propeller shaft and an unfavorable load distribution along the bearing length. Strip wear depends on the load that acts in the bearing as the shaft rotates. The wear intensity cannot be explained by dynamic loads due to flexural vibrations of the shaft since the disturbing force acts with a blade frequency that exceeds the intrinsic frequency of the shaft's flexural vibrations by approximately a factor of four; this being so, the flexural-vibration amplitude of the shaft and the dynamic loads in the bearing are small.

The existing branch standard [1] divides the vertical component of the hydrodynamic moment of a propeller into two constituents: a constant constituent and a variable constituent. For even-blade propellers, with which icebreakers are equipped, the variable constituents on opposing blades are practically compensated, and therefore the magnitude of the vertical component for these propellers is governed mainly by the constant constituent of the hydrodynamic moment. This enables us to regard the state of the system of propeller shaft–stern bearing as equilibrium for the prescribed power on the propeller shaft and its rotational speed, which depends on the power and the magnitude of the torque required torque.

2. The propeller will be regardded as a solid. The work of the external load applied to the propeller located on the propeller-shaft cantilever is expended on changing the internal energy of the system of propeller shaft–stern bearing. The load that acts on the system in question is the constant constituent of the hydrodynamic moment. Its work is expended on changing the energy of the flexural strain of the propeller shaft and the compressive strain of the strips of the stern bearing. Then, using the law of conservation of energy, we have

$$A_{\mathrm{m}} = U_{\mathrm{fl}} + U_{\mathrm{compr}} . \tag{1}$$

We regard the propeller shaft as a simply supported circular beam of constant cross section with a cantilever on which the propeller is located. We take position 1 in Fig. 1 as the position of the static equilibrium of the fixed shaft. The constant constituent of the hydrodynamic moment, whose magnitude depends on the torque of the power plant and the rotational speed of the shaft, causes bending of the shaft and strain in the bearing. As a result the system in question will move to a new position of equilibrium 2 in Fig. 1.

Admiral S. O. Makarov State Naval Academy, St. Petersburg. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 72, No. 1, pp. 156-160, January-February, 1999. Original article submitted January 3, 1997; revision submitted January 20, 1998.

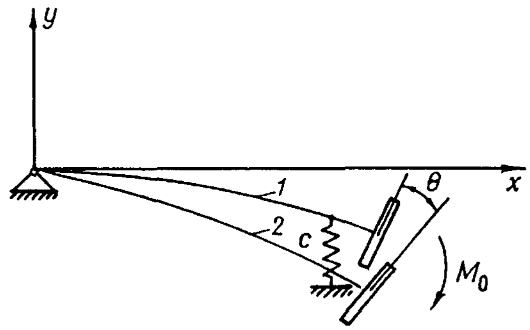

Fig. 1. Design scheme of a propeller shaft.

Thus, the hydrodynamic moment is an external load that acts on the system, and the restoring moments of the bending force for the shaft and the reaction of its support are a balancing load. The work of the internal forces serves as a measure of the potential-energy change of the system.

The maximum vertical component of the hydrodynamic moment [1] is determined by the relation $M_{vert} = M_0 + M_z$.

As has already been mentioned, the variable constituent of the vertical component is small and cannot have a substantial effect on the working load or on the wear of bearing strips, and therefore it will be disregarded. In this case, the work of the constant constituent of the vertical component of the propeller's hydrodynamic moment is expressed as $A_m = M_0 \Theta$.

It is known that the potential energy of flexural strain in a beam of a constant cross section that is a shaft of length l depends on its flexural rigidity and is represented as

$$U_{fl} = \frac{EI}{2} \int_0^l (y'')^2 \, dx .$$

The area of the spot of contact between the propeller shaft and the working surface of the stern bearing varies very insignificantly under changes in the load, and therefore the rigidity of the bearing can be regarded as a constant quantity. This means that the potential energy of the compressive strain of the bearing strips will depend on this strain in the following manner:

$$U_{compr} = \frac{1}{2} c \, \Delta^2 .$$

Introducing the expressions for the work and the energies into Eq. (1), we obtain

$$M_0 \Theta = \frac{EI}{2} \int_0^l (y'')^2 \, dx + \frac{1}{2} c \, \Delta^2 .$$

Then the compressive strain of an annular series of strips of the stern-bearing assembly is determined by the expression

$$\Delta = \left(\frac{2M_0 \Theta - EI \int_0^l (y'')^2 \, dx}{c} \right)^{1/2} .$$

Taking into account that the reaction in compression is determined by the relation $Q = -c\Delta$, we obtain the reaction in the bearing cross section in question for the steady-state rotational speed of the propeller shaft in ice-free water:

$$Q = - c^{-1/2} \left(2M_0 \Theta - EI \int_0^l (y'')^2 \, dx \right)^{1/2} .$$

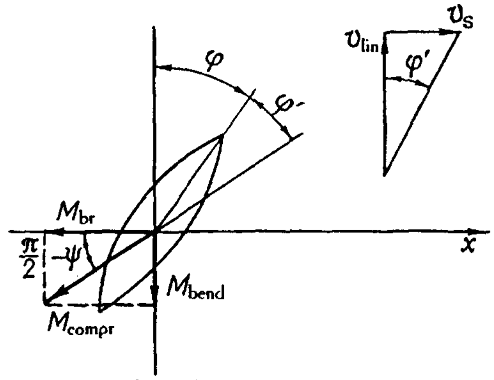

Fig. 2. Determination of the bending moment in the interaction of a propeller with ice.

In icebreaker operation in ice, an additional bending moment due to the interaction of the propeller blades with the ice develops on the propeller shafts. Here, on the blades an additional moment of resistance appears (see Fig. 2) that causes a decrease in the rotational speed of the shafts and an increase in the bending moments on them. The additional moment of resistance increases with the depth of cutting of the ice by the propeller blades, their rotational speed, and the ship velocity. Taking into account that the disk-area ratio of the icebreaker propeller, i.e., the ratio of the sum of the areas of the rectified blades to the area of the propeller disk, is large (0.85), the propeller blades can be considered to interact continuously with the cut ice floe so that the variable constituent does not make a marked contribution to the additional moment of resistance, which can be considered constant. From the scheme in Fig. 2, it can be seen that the additional moment of resistance can be resolved into two components: a braking component that decreases the rotational speed of the propeller shaft and a bending component that causes additional bending of the shaft. Thus, determination of the working loads in the stern bearing in icebreaker operation in ice reduces to finding the components of the additional moment of resistance.

The helix angle ψ of the helical line left by the blades of the propeller shaft in the ice floe that interacts with it depends on the pitch angle φ of the blades, the linear velocity v_{lin} of the propeller-blade's edge, and the ship velocity v_s. Using the scheme of Fig. 2, we can determine it in the following manner:

$$\psi = \varphi + \varphi' = \varphi + \arctan\frac{v_s}{v_{lin}}.$$

Since, in the interaction of the propeller with the ice, the angular speed ω of the shaft changes, the kinetic moment K of the shaft–propeller system also changes; here, $\delta K = I_x \delta\omega$.

The time τ in which the change δK in the kinetic moment of the system occurs, can be determined from the following considerations. The propeller geometry is such that no more than two of its four blades can interact simultaneously with the ice, and the second blade begins to interact with the ice after a time interval that is one-fourth of the time of a turn of the propeller. The moment of resistance will attain its largest magnitude when the second blade of the propeller enters the ice floe, i.e., a quarter of a turn after the beginning of the interaction. Consequently, $\tau = 1/(4\pi)$.

Using the theorem on the change in the kinetic moment, we obtain the average braking moment that causes a change in the rotational speed ω of the propeller shaft, $M_{br} = \delta K/\tau$, and the additional bending moment on the shaft

$$M_{fl} = M_{br} \tan\left(\frac{\pi}{2} - \psi\right).$$

The total bending moment on the propeller shaft is the sum of the free-water bending moment and the additional bending moment caused by the interaction of the propeller blades with the ice.

The method presented makes it possible to calculate the working loads in any cross section of a stern bearing.

The residual thickness of the layer of antifriction coating of a stern bearing is one parameter that can be used for diagnosing the technical state of the bearing and evaluating its service life. During operation the thickness decreases constantly because of intense wear of the antifriction material, which is explained by the high contact pressure, particularly at the trailing edge of the bearing. The rate of antifriction-material wear is known to depend on the contact pressure and the slip velocity of the friction couple. Experimental investigations and experience in ship operation enable us to consider the wear rate of the antifriction materials employed to be practically the same and to be characterized by a wear factor $\alpha = 2 \cdot 10^{-3}$ $mm^2 \cdot h^{-1} \cdot N^{-1}$ [2]. The same source yields formulas for calculating the contact pressure q in a stern bearing as a function of the static reaction P and the bearing geometry:

$$q = P/(Ld) , \tag{2}$$

and for calculating the wear H of the antifriction material of the bearing in t hours of its operation: $H = \alpha q t$.

Icebreaker stern bearings are loaded nonuniformly and, according to [3], react to the main load only by a portion of the length that is equal to two diameters d of the propeller shaft. This enables us to introduce the concept of the effective length of the stern bearing $L_{\text{eff}} = 2d$.

The working load Q, on which strip wear depends, is a function of the power N on the propeller shaft and the number n of revolutions of it per minute, i.e., $Q = Q(N, n)$. Then, replacing the static reaction P by the working load Q and the length L of the stern bearing by its effective length L_{eff} in formula (2), we obtain a formula for calculating the working contact pressure in the bearing: $q_w = Q(N, n)/(2d^2)$. In this case, the expression

$$H = \alpha \frac{Q(N, n)}{2d^2} t ,$$

which makes it possible to determine the wear of the strips of an icebreaker stern bearing from the working load calculated from operation or to evaluate its service life up to a prescribed magnitude of the wear, will be a mathematical model of the wear of an icebreaker stern bearing.

3. In 1990, working loads in a stern bearing of the shipboard line of the shaft of an Arktika atomic icebreaker were determined experimentally. The following working loads were obtained: for operation at 100% power in ice-free water, the maximum working load was 210 kN on the trailing edge of the stern bearing and 75 kN on the leading edge of the same bearing; for operation in ice, the maximum working load was 805 and 360 kN, respectively. The working loads for the same regimes calculated by the procedure presented were 212 and 77 kN for free-water operation and 807 and 362 kN for operation in ice.

Thus, the measured and calculated loads are in satisfactory agreement. This permits the conclusion of the possibility of using the method presented to calculate installation parameters for the shaft line of an icebreaker in the design stage and to diagnose its technical state.

NOTATION

A_m, bending-moment work; U_{fl}, potential energy of the flexural strain of the shaft; U_{compr}, potenmtial energy of compression of bearing strips; M_{vert}, maximum vertical component of the propeller's hydrodynamic moment; M_0, constant constituent of this moment; M_z, variable constituent of this moment; Θ, angle of rotation of the propeller plane; l, length of the cantilever of the shaft; E, modulus of elasticity of the propeller-shaft material; I, moment of inertia of the shaft cross section; y, transverse displacement of the shaft cross section; $y'' = d^2y/dx^2$, second derivative of y with respect to x; x, longitudinal coordinate of the shaft cross section; c, stiffness coefficient of the stern bearing; Δ, its strain; Q, reaction of the bearing; ψ, helix angle of the helical line in the cut ice floe; φ, pitch angle of the propeller blades; v_{lin}, linear velocity of the edge of the propeller blade; v_s, ship velocity; φ', auxiliary angle in Fig. 2; ω, angular speed of the propeller shaft; K, kinetic moment of the shaft–propeller system; δK, change in the kinetic moment in the time τ; I_x, rotational moment of inertia of the shaft with the propeller; $\delta\omega$, change in the angular speed in the time τ; τ, time of a quarter of a turn of the shaft; n, number of turns of the shaft per minute; M_{br}, braking component of the moment of resistance; M_{bend}, bending component of the moment of resistance; α, wear factor; q, contact pressure in the stern bearing; P, static reaction of the bearing; L, bearing length; L, bearing diameter; H, wear of the bearing material; t, operating time for the bearing; L_{eff}, effective length of the bearing; N, propeller-shaft power; q_w, working contact pressure in the bearing.

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