THE METHOD OF LONG-LIFE CALCULATION FOR A FRICTION COUPLE "ROTOR – HYBRID BEARING"

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Abstract. Reliability of rotating machinery is determined to a considerable degree by the bearing units. For several applications the requirements in rotation speed, bearing load and maximal vibration level are so extreme that neither rolling-element bearings nor fluid-film bearings could provide necessary performance characteristics during all regimes of operation. Hybrid bearings, which are a combination of rolling-element and fluid-film bearings, can improve performance characteristics and reliability of the rotor-bearing systems. The article presents the approach for to formation of the method of resource calculation for hybrid bearing with speed separation. The results show that the resource of the slide bearing increases significantly when it is used in combination with a rolling-element bearing compared to its single setting.

1 INTRODUCTION

When designing rotary machines with multiple start-ups and shutdowns (turbochargers of chemical fuel cell electric vehicles (FCEVs), turbo expanders and different types of pumps for cryogenic engineering and petrochemical industry), the task of ensuring their high reliability is of great importance. It imposes more stringent requirements on the bearings of the rotors of such machines. Most of rotary machines with a rotation rate of 10^5 rpm use a fluide-film bearing (FFB) as a support of the rotor. Fluid-film bearing unlike rolling-element bearings (REB) are not limited in high speed. But at the point of "start-up" and "shutdown" the wear of the FFB is certain to happen because of the dry frictions at these points. When none of the existing types of bearings meets the technical requirements of a rotor machine, the possible solution of the problem is to combine the bearings with different functional principle. It will increase the reliability of a bearing due to the separation and duplication of their functions.

Hybrid bearings are able to eliminate many disadvantages of particular bearing type from the point of view of load capacity, dynamic characteristics, reliability, and life time. The positive effects of hybrid bearings are achieved by separating and/or duplicating bearing functions on different operating regimes. This makes possible to increase demands on operational requirements, thus the efficiency of rotating equipment could be improved.

There are two basic designs of hybrid bearings. Notation suggested in [1, 2, 3] is used in this work for describing different hybrid bearings. The first design is hybrid bearing with load separation – PL bearing (figure 1a), a shows the scheme of PLEX (Parallel Load, Externally Fed) bearing. In PL bearing REB is less loaded on the main operating mode. Discharge of REB is due to increase in hydrodynamic or hydrostatic reaction in FFB. The second design is hybrid bearings with speed separation – PS bearing. Figure 1b shows the scheme of PSEX (Parallel Speed, Externally Fed) bearing. Well-known squeeze-film dampers are the special case of PSEX bearings. In PS bearing shaft rotates in REB during the start-up and shut-down. As rotating speed increases REB is partially or fully switched off from the operation and shaft starts to rotate in FFB. Switching off of REB is achieved by combining certain geometric and operating parameters, or by using special switching devices. Lubricant supply in FFB can be also provided through the shaft. In this case one can distinguish two more designs – PSIN (Parallel Speed, Internally Fed) bearing, (figure 1c) and PLIN (Parallel Load, Internally Fed). In all concepts the fact that hydrostatic bearing turns on only during the main regime allows to manage without an external pressure supply.

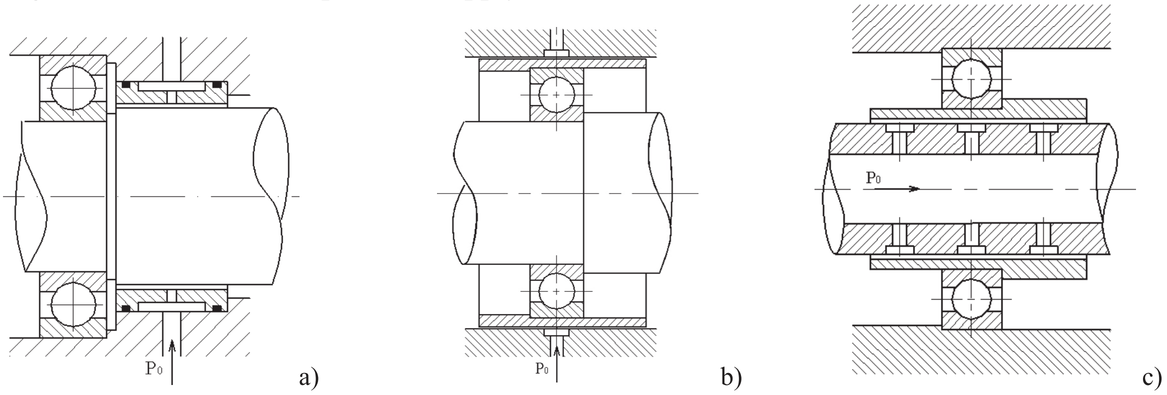

Figure 1: Schemes of the hybrid bearings

The mechanism of the PSEX work assumes the improvement of characteristics of the entire bearing in the following fields [4, 5].

1. *Decreasing of the damaging moments* which affect the sleeve of a rolling element bearing during the transition regimes. It will lead to its long-life improvement. One of the main reasons for losing the efficiency is a sleeve wear during the transition regimes (start-up, shutdown, a contact of a shaft with a sleeve as a result of unstable motion or stressful loads.) The mathematical models of the FFB wear assume that the wear depends on the compressive forces in a contact and speed in relation to the motion of contacting elements. When putting a FFB in the PSEX during the transition regimes, the sleeve of the FFB is affected not by the shaft torque, but the friction torque in the REB, that leads to the speed reduction in a contact, reduction of the distance of friction_and thus, to the increase of the FFB and entire PSEX long-life, as the FFB long-life isn't practically limited.

2. *Reducing of the centrifugal loads in a rolling element bearing* due to the speed separation. The mechanism of speed separation assumes the speed reduction of the relative motion of the REB elements and sometimes its complete switching off that leads to its unlimited long-life and opportunity of using at its maximum speed.

3. *Removing from the self-excited oscillation modes* typical for the rotor motion in the fluid film bearings: during the unstable rotor motion, the risk of significant damages (e.g., micro

welding in the FFB sleeve) is practically eliminated due to the possibility of an additional roll in the REB; the mechanism of the development of self-excited oscillations, for example, the unstable part of the fluid film reaction, which causes the turbulence of the rotor in the FFB, is applied to the REB outer ring, which, due to the possibility of an additional roll neutralizes or reduces the negative impact of the reaction of the REB on the rotor.

4**.** *Increasing of the abilities to provide the necessary dynamic characteristics of the bearing* to ensure the transition of the resonant modes with permissible oscillation amplitudes. By choosing appropriate geometrical parameters of REBs and FFBs and taking into consideration the operational ones, we can achieve the required stiffness and damping to provide reasonable amplitude-frequency characteristics of the entire rotor-bearing system.

5. *Improving of the reliability of the bearing* by increasing its efficiency in case of emergency and under the unstable operation due to the duplication of functions of the REB and FFB. In case of emergency in a turbine set (for example, a tearing off of a blade), there can be an abrupt rotor unbalance change and the dynamic loads on the bearings become higher. In case of the FFB failure to transfer the load to the housing and rotor spinning, the REB starts to operate. And for some time it can operate at a rotary speed exceeding its maximum speed and eliminate a significant damage of the entire machine.

2 MATHEMATICAL MODEL

The declared advantage for reducing the damaging moments leads to the necessity to describe this effect, which requires modeling of the coupled problems defining the forces and a friction torque in REBs and FFBs, the dynamics of the rotational motion of the bearing and the FFB sleeve wear.

The defining of the FFB sleeve long-life is based on the solution of the wear equation [6], which can be generally represented as:

$$I = a_0\left(\frac{p}{HB_{1,2}}\right)^{a_1}\left(\frac{\lambda^*}{h}\right)^{a_2}\left(\frac{E_{1,2}}{\sigma_{0_{1,2}}}\right)^{a_3}(1+\propto K)^{b_1}\left(1+\beta\frac{HB_a}{HB_{1,2}}\right)^{b_2}\left(1+\gamma\frac{Sd_a}{V}\right)^{b_3} \tag{1}$$

where $\frac{p}{HB_{1,2}}$ is a dimensionless group, describing a stress state in a contact (p is a load per unit area, MPa; $HB_{1,2}$ is the hardness of the working surfaces of the shaft and bearing, MPa); $\frac{\lambda^*}{h}$ is a complex, defining the contact mechanism (when there are mechanical impurities in the fluid film $\lambda^* = (R_{a1}^2 + R_{a2}^2)^{1/2}$, here $R_{a1,2}$ is a mean square deviation of the surface roughness mcm; when there are mechanical impurities $\lambda^* = d_a$, here d_a is a given diameter of particles, and under condition $d_a > (R_{a1}^2 + R_{a2}^2)^{1/2}$; $\frac{E_{1,2}}{\sigma_0}$ is a complex regarding plasticity and endurance strength of materials $E_{1,2}$ is a modulus of elasticity, MPa; $\sigma_{0_{1,2}}$ is the endurance strength of materials, MPa; $(1+\propto K)$ is a concentration criterion of the mechanical impurities ($\propto$ is a coefficient, regarding the degree of influence of particles concentration on the wear; K is a concentration 1 kg abrasive in 1 kg fluid); $1+\beta\frac{HB_a}{HB_{1,2}}$ is a criterion of the hardness of impurities ($\pi_т$) (β is a coefficient, regarding the degree of influence of particles concentration on the wear; HB_a is the hardness of mechanical impurities, MPa); $1+\gamma\frac{Sd_a}{V}$ is a criterion of particles shape ($\pi_ф$) (γ is a coefficient, regarding the degree of influence of the particle geometry on the couple wear, S is a particle surface, V is a particle volume); a_0 is a coefficient, regarding physical-mechanical properties; a_i, b_i are indexes, defined on the basis of the available information on the test of materials.

In the transition regime the fluid film bearing in a hybrid bearing is identical to the friction couple operating in a semi-fluid mode. In relation to the fluid film bearings of the semi-fluid friction, the wear rate of a sleeve made of anti-friction material can be simplified to [7]:

$$I_h = 2{,}29 \cdot 10^6 K \cos^2\varphi_T \psi p_m V_S E_{\text{пр}}, \tag{2}$$

where $V_S = \frac{\pi dn}{60}$ is a sliding speed in a contact; $E_{\text{пр}} = \frac{2}{\frac{1-\varepsilon_1^2}{E_1}+\frac{1-\varepsilon_2^2}{E_2}}$ is a given modulus of elasticity, respectively E_1, E_2 are modulus of elasticity of the material for a shaft and FFB sleeve; $p_m = \frac{F_r}{dl}$ is the medium pressure in a bearing; $\psi = \frac{2h}{d}$ is a relative clearance; K is a coefficient, depending on properties of the materials for a shaft and a sleeve and lubrication conditions; φ_{T} is an angle between a vertical axes and a maximum pressure point on the FFB surface (figure 2).

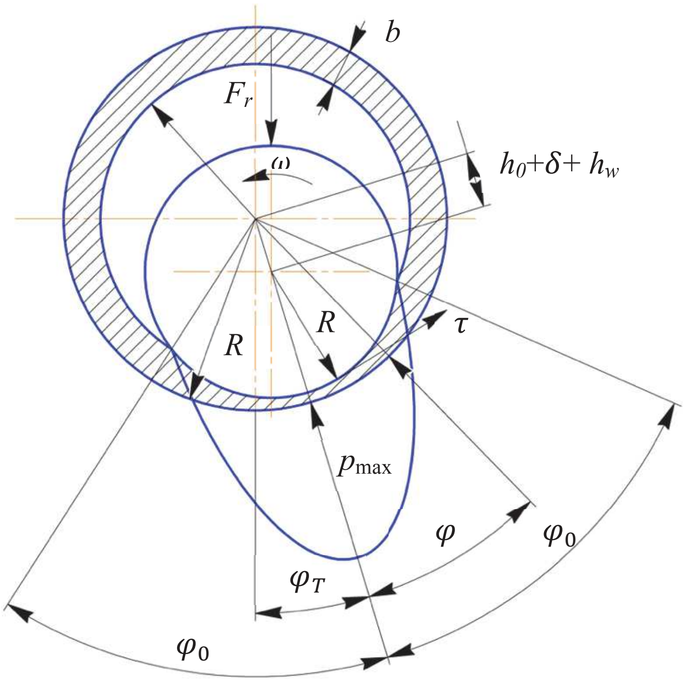

Figure 2: Pressure Distribution in a Fluid Film Bearing

The FFB long-life (hours) and, respectively, of the entire PSEX:

$$L_h \leq \frac{[h]}{I} \tag{3}$$

where $[h]$ is a permissible clearance.

The sliding speed in a contact depends on the rotation speed of the REB outer ring, with which a space sleeve is rigidly bound. The space sleeve is a response surface for the FFB. From the point of view of the dynamics, the PSEX can be represented in the form of a dualmass six-degree-of-freedom oscillator (figure 3): flat motions and rotor spinning, which has been replaced by a point mass off-centered about the axis of rotation, and flat motions and rotation of the outer ring of the REB. To solve the problem of defining the function of speed separation between the shaft and outer ring of the REB, it is necessary to compose the equation of the rotary motion between the shaft and outer ring of the FFB according to the law of conservation of angular momentum:

$$I\frac{d\omega}{dt} = M_{dr} - M_{fr}^{REB}\ ;$$
$$i\frac{d\Omega}{dt} = M_{fr}^{REB} - M_{fr}^{FFB}\ , \tag{4}$$

where I, i are polar moments of inertia of the rotor and outer ring of the REB in combination with a space sleeve; ω, Ω are angular rates of a rotor and outer ring, respectively; M_{dr} is a moment of driving forces; M_{fr}^{REB} is a REB friction torque; M_{rf}^{FFB} is a FFB friction torque.

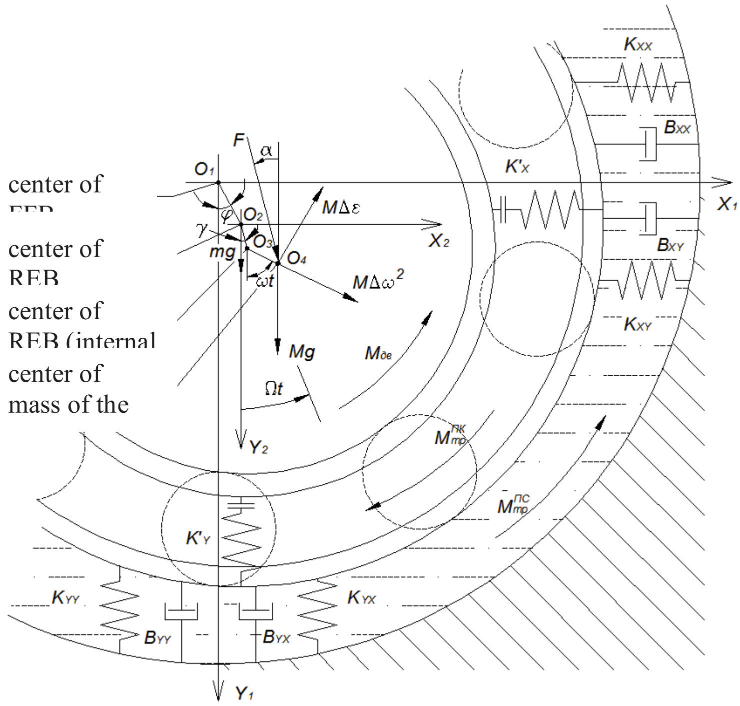

Figure 3: Dynamic Model of the PSEX

The restoring force of the REB depends on the elastic properties of rings and rolling elements, and its defining is based on Hertz's contact deformations theory [8]. The rolling elements are deformed under the influence of an external force $F_{\sum}$ on the value of δ (figure 4). The dependence of the rolling elements reaction on their deformation under the action of external force:

$$R^{REB} = K'\delta^{3/2} \tag{5}$$

where K' is a coefficient of proportionality depending on the material and shape of the contacting surfaces (it can be described as the coefficient of the REB nonlinear stiffness with dimension of N/м$^{3/2}$) [8].

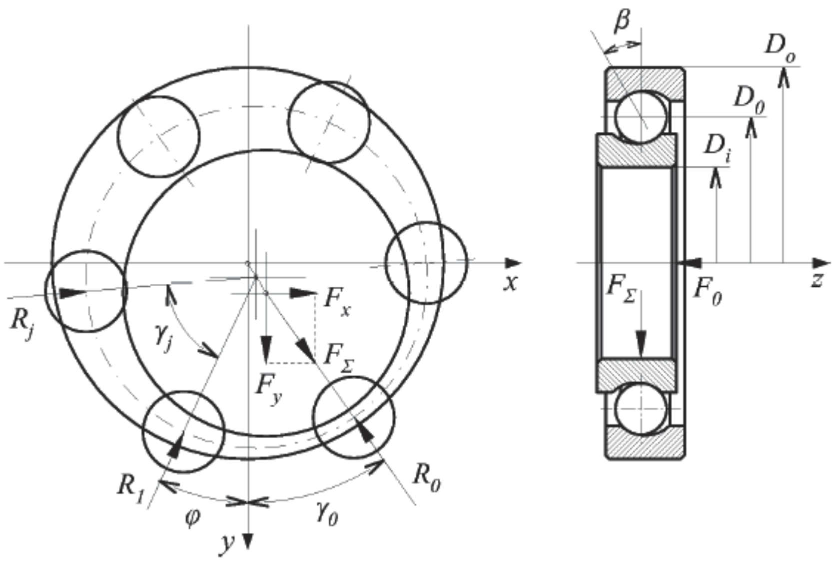

Figure 4: Scheme of the Ball Rolling Element Bearing

The mathematical description of friction forces is based on the semi-empirical method developed and applied at the leading enterprises of the bearing industry [7]:

$$M_{\text{тр}}^{\text{ПК}} = 0.9 \cdot 10^{-6} \left(\frac{F_\Sigma}{C_0}\right)^{0.55} (3F_a - 0.1F_r)D_0 + 2 \cdot 10^{-10}(\nu n)^{2/3} D_0^3) \qquad (6)$$

where $F_\Sigma = (XVF_r + YF_a)K_\text{Б}K_\text{Т}$ is an equivalent load on the REB; F_r, F_a are radial and axial loads; X, Y are coefficients recording different damaging action of radial and axial loads. They are defined according to the ratio F_r/C_0, , here C_0 is a static load capacity of the REB; V is a coefficient of rotation; $K_\text{Б}$, $K_\text{Т}$ are safety and temperature coefficients; ν is the kinematic viscosity; n is a frequency of rotation.

The model of a fluid film bearing (figure 5) is based on the equations of hydrodynamic lubrication theory [9,10]. The defining of pressure fields is based on the simultaneous solving of the Reynolds equations for the case of two-dimensional flow of a compressible fluid regarding the effect of turbulence (7), the equations of energy, equations of state, turbulence models, functions of radial clearance and the ratios of speeds on the surface of the shaft.

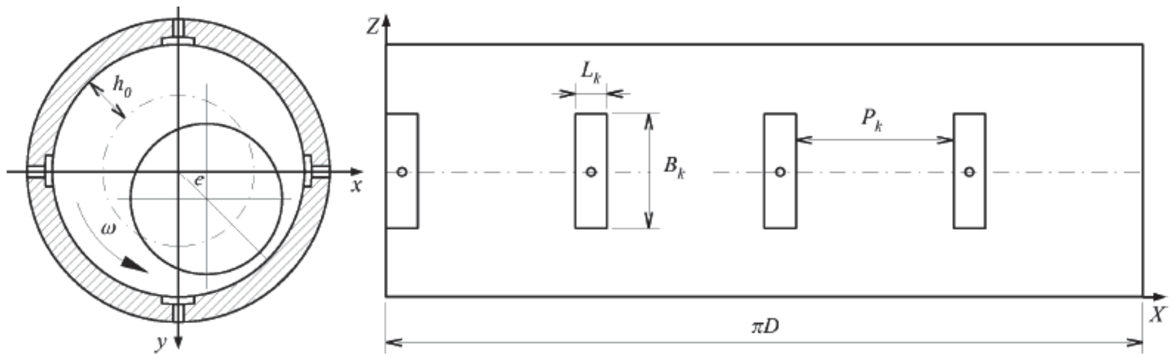

Figure 5: Scheme of the Hydrostatic FFB

The following example is a single line equation:

$$\frac{\partial}{\partial x}\left[\frac{h^3\rho}{\mu K_x}\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial z}\left[\frac{h^3\rho}{\mu K_z}\frac{\partial p}{\partial z}\right]=6\frac{\partial}{\partial x}(\rho U h)-12\rho V+12h\frac{\partial \rho}{\partial t}. \tag{7}$$

Reaction and friction torque of the fluid film bearing:

$$R_X=-\int_0^L\int_0^{\pi D} p\cos\alpha\, dx\, dz;\ R_Y=-\int_0^L\int_0^{\pi D} p\sin\alpha\, dx\, dz;$$
$$M_{\text{тр}}^{\text{ПС}}=-\frac{D}{2}\int_0^L\int_0^{\pi D}\tau\sin\alpha\, dx\, dz;\ \tau=\frac{h}{2}\frac{\partial p}{\partial x}+\frac{\mu U}{h}. \tag{2}$$

3 RESULTS OF MODELING AND DISCUSSION

The method of the PSEX long-life defining was based on the assumption that since the operation speed of the REB tends to zero and one of the main factors of its destruction – the emergence of large stresses on the outer ring due to centrifugal forces F_{cf} (Figure 6) - is excluded, the main factor that defines the PSEX life is a FFB sleve wear. Thus, the method of the FFB sleeve wear long-life defining has been offered which is based on the assumption of proportionality of wear and torque. The mechanism of PSEX operation assumes that at the start mode the FFB sleeve is affected by the REB friction torque, but not by the machine torque, and also the distance of friction becomes less. The simultaneous solution of the equations of rotary motion and wear dynamics allows to get the dependency of the resource on various working parameters of the PSEX and to compare it with the FFB single setting (Figure 7).

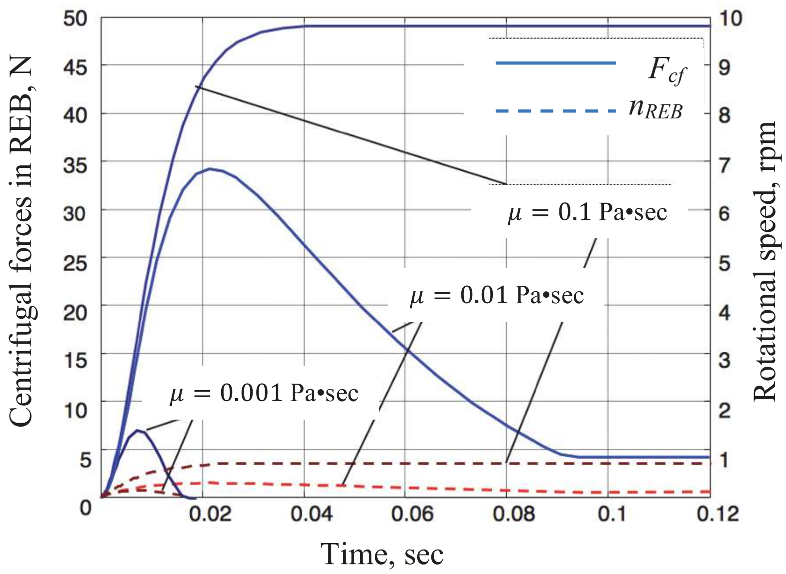

Figure 6: Speed separation in a hybrid bearing

The results show that the FFB long-life improves significantly when it is used in a combination with a rolling-element bearing compared to its single setting. It means that under low loads the resource improvement can be up to 7 times higher compared to a FFB single setting, and under high loads up to 2...3 times.

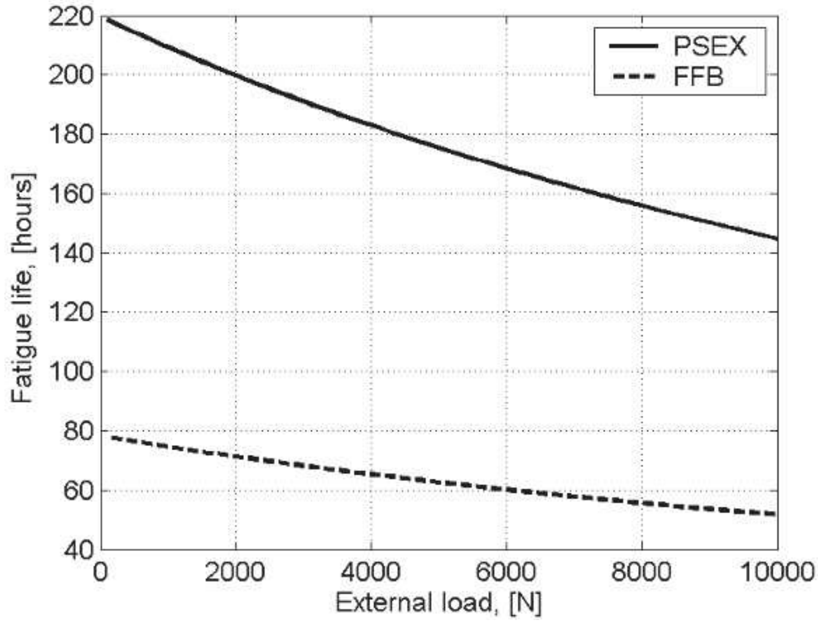

Figure 7: Long-life improvement in a hybrid bearing

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