Effect of Bearing Housings on Centrifugal Pump Rotor Dynamics

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Abstract: The article deals with the effect of a bearing housing on rotor dynamics of a barrel casing centrifugal boiler feed pump rotor. The calculation of the rotor model including the bearing housing has been performed by the method of initial parameters. The calculation of a rotor solid model including the bearing housing has been performed by the finite element method. Results of both calculations highlight the need to add bearing housings into dynamic analyses of the pump rotor. The calculation performed by modern software packages is more a time-taking process, at the same time it is a preferred one due to a graphic editor that is employed for creating a numerical model. When it is necessary to view many variants of design parameters, programs for beam modeling should be used.

1. Introduction

The centrifugal pump rotor is one of the most complex and vital assembly units that determines reliability of the whole pump to a large extent.

Major components of a rotor are a shaft, impellers, shaft couplings, shaft sleeves, parts of mechanical seals, balancing devices, etc. Depending on a pump design the rotor is manufactured with different mutual arrangement of parts. A cantilever pump impeller is placed on the shaft end. As a rule, rotor in a single-stage double entry pump and ring-section type pumps is kept by symmetrically located outer bearings.

Bearings are placed in bearing housings. Bearing housings could have different shapes and arrangements. Figure 1 shows bearing housing designs of the barrel casing centrifugal boiler feed pump [1] intended to feed water supply from a deaerator into a plant unit steam generator.
Rotor dynamics of the barrel casing centrifugal boiler feed pump rotor depends on various entities, the most important of which are hydrodynamic processes within plain bearings and annular seals, and the static stiffness of bearing housings.

The inclusion of the stiffness of bearing housing designs will allow to determine rotor dynamics of the barrel casing centrifugal boiler feed pump rotor more precisely.

When calculating rotor dynamics in conjunction with bearing housings it is possible to use both modern production software and programs for beam modeling. But when using such programs there are two key questions: execution time consumed by modern production software during a calculation, and accuracy of calculation being performed by beam modeling programs.

Following mathematical models of bearing elements of the design were used for research of system dynamics.

2. Plain Bearings

Plain journal bearings comprise the round cross-section shaft area (a shaft journal), rotating inside of a bearing shell which is round, as a rule. The diameter of the shaft journal is usually 99.8 – 99.9% of the bearing shell diameter, and resulting clearance is partially-filled with lubrication fluid.

A lubricating film in hydrodynamic plain bearings offers elastic property. Said another way, as a dynamic load is applied to the bearing, for example, as an unbalance consequence, the shaft journal has to move along round orbit about the static balance position, which it could be placed in case of only a static load presence. Effective stiffness and damping related to this elasticity of lubricating film have impact on critical vibration frequencies and system stability. Based on the above stated it is important to assess bearing dynamic characteristics during the design phase.

In study [2] Sommerfeld criterion (number) is used to determine the stiffness coefficient of hydrodynamic plain bearings.

\[ S = \frac{\mu \cdot \omega \cdot D \cdot L}{W} \left( \frac{R}{h} \right)^2 \]

where

- \( \mu \) - viscosity of the lubricating medium, N·f/m²;
- \( \omega \) - angular frequency of rotation, rad/s;
- \( D \) - shaft diameter within the bearing;
- \( L \) - bearing axial length, m;
- \( W \) - bearing load, N;
- \( R \) - shaft radius within the bearing, m;
- \( h \) - radial clearance, m.

Depending on L/D and Sommerfeld number, stiffness and damping coefficients of the plain bearing are defined by the formula:

\[ c_{xx}, c_{xy}, c_{yx}, c_{yy} = \frac{W \cdot \bar{K}_{xx}}{h}, \frac{W \cdot \bar{K}_{xy}}{h}, \frac{W \cdot \bar{K}_{yx}}{h}, \frac{W \cdot \bar{K}_{yy}}{h} \]
\[ d_{xx}, d_{xy}, d_{yx}, d_{yy} = \frac{W \cdot B_{xx}}{\omega \cdot h}, \frac{W \cdot B_{xy}}{\omega \cdot h}, \frac{W \cdot B_{yx}}{\omega \cdot h}, \frac{W \cdot B_{yy}}{\omega \cdot h} \]

where \( K_{xx}, K_{xy}, K_{yx}, K_{yy}, B_{xx}, B_{xy}, B_{yx}, B_{yy} \) are dimensionless coefficients varied with the Sommerfeld number.

3. The Bearing Housing
Stiffness of the bearing housing design C is defined in principle by the formula:

\[ C = \frac{F}{\Delta x} \]

where \( F \) – force applied to the bearing shell, N;
\( \Delta x \) – displacement of the bearing induced by force \( F \), m.

Software ANSYS Workbench 14.5 [3] was used to perform calculations. Table 1 lists stiffness values of bearing housings of the barrel casing centrifugal boiler feed pump.

| Location of the bearing | Loading                               | Stiffness of the design C, N/m |
|-------------------------|---------------------------------------|-------------------------------|
| The pump bearing, drive end | in horizontal direction                | 4.02 \cdot 10^9               |
|                         | in vertical direction                 | 4.50 \cdot 10^9               |
| The pump bearing, non-drive end | in horizontal direction                | 2.50 \cdot 10^8               |
|                         | in vertical direction                 | 3.95 \cdot 10^8               |

Mass of the drive end bearing housing is 127 kg, mass of the non-drive end bearing housing is 70 kg.

4. Analysis of Pump Rotor Vibration by the Method of Initial Parameters. Evaluation of the Bearing Housing Effect

When evaluating free bending vibration of the rotor subsystem, the pump rotor is simulated as a circular beam of a piecewise constant stiffness and its mass per meter with mass of shaft mounted parts (impellers, shaft sleeves, half-coupling, etc.) being discretely placed on the beam.

Bounds between the rotor and the pump casing are simulated by elastic supports at points of the bearing and annular seal location. Bearing housings are simulated by mass and stiffness of the bearing housing design.

Figure 2 shows such simulated model of the pump rotor. This is an unbranched frame structure comprising of the chain-arranged set sections of different types.

The calculation of rotor free vibration is performed by the method of initial parameters in a matrix form [4-6]. The calculation of the stiffness (Nm) of pump annular seals \( C \) is performed in accordance to the procedure [7].

Table 2 lists values of the first three natural frequencies of pump rotor vibration including the bearing housing design and without including it that have been obtained by the method of initial parameters.

| Natural frequencies of vibration | \( f_1 \) | \( f_2 \) | \( f_3 \) |
|----------------------------------|----------|----------|----------|
| Without including the bearing housing design, Hz | 79.10 | 123.85 | 21.56 |
| Including the bearing housing design, Hz         | 74.20 | 119.76 | 209.11 |
| Relative impact of the bearing housing, %        | -6.2   | -3.3    | -4.3     |
Figure 2. Simulation model of the barrel casing centrifugal boiler feed pump.
5. Analysis of Pump Rotor Vibration by the Finite Element Method

Figure 3 shows the rotor solid model of the barrel casing centrifugal boiler feed pump.

(a)

Figure 3. Rotor solid model: (a) without bearing housing; (b) with bearing housing.

Table 3 gives values of first three natural frequencies of rotor bending vibration including the pump bearing housing design and without including it that have been obtained by the finite element method (FEM).

Table 3. Values of first three natural frequencies of rotor bending vibration that have been obtained by FEM.

| Natural frequencies of vibration | $f_1$  | $f_2$  | $f_3$  |
|---------------------------------|--------|--------|--------|
| Without including the bearing housing design, Hz | 84.47  | 132.83 | 207.89 |
| Including the bearing housing design, Hz | 80.47  | 129.26 | 202.33 |
| Relative impact of the bearing housing, % | -4.7   | -2.7   | -2.7   |

6. Analysis of the Bearing Housing Effect

Equivalent stiffness of the “bearing housing - lubricating film” system $K_{eq}$ is determined by the formula:

$$K_{eq} = \frac{K_{b,h} \cdot K_{l,f}}{K_{b,h} + K_{l,f}}$$

where

- $K_{b,h}$ is the stiffness of the bearing housing design;
- $K_{l,f}$ is the stiffness of the lubricating film.

High-stiffness bearing housing designs has less effect on pump rotor dynamics than low-stiffness ones. When engineering calculation of rotor dynamics is performed, we can neglect the stiffness of bearing housings if following inequation is satisfied:

$$K_{b,h} > 10 \cdot K_{l,f}$$
7. Conclusions

According to performed numerical studies it was found that rotor dynamics of the barrel casing centrifugal pump depend on bearing housing designs.

Production software with the graphic editor for creating simulation models should be used to perform calculations of the bearing housing stiffness. Then obtained stiffness values should be used in calculations by means of beam models. Therewith numerical calculations of rotor dynamics are in good agreement with simulation results. Simulation analysis is a time-consuming process; this could be used as a final, validation method.

When it is necessary to view many variants of design parameters (in consideration of their variation range while in operation or for viewing design variants being developed) we could use programs for beam modeling.

References
[1] Martsykovskyy V A, Vorona P N 1987 Pump for NPP (Energoatomizdat, Moscow)
[2] J W Lund and K K Thomson 1987 A Calculation Method and Data for the Dynamic Coefficients of Oil Lubricated Journal Bearings Proceedings of the ASME Design and Engineering Conference (Minneapolis) p 1
[3] Programm Complex ANSYS 14.5, license agreement 673888
[4] Babakov I M 1965 Vibration Theory (Nauka)
[5] Iovich V A 1981 Transfer Matrix in Elastodynamics: Guide, 2nd enlarged edition (Mashinostroenie, Moscow)
[6] Simonovskiy V I 2006 Centrifugal Machine Rotordynamics (Sumy State University, Sumy)
[7] Martsykovskyy V A 1980 Noncontacting Seals of Rotary Machines (Mashinostroenie Moscow)