Special Rotordynamics: Finite Element Model for Pump Rotor

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Abstract. An approach to the finite element modal analysis for the real pump rotor is presented. The first eigenfrequencies and eigenmodes of the initial, simplified and battered rotor-seal-bearing system are compared.

1. Introduction

Rotor of the simulated pump rotates in the two hydrodynamic bearings as well as inside the eight ring seals. Fluid seeping through their clearances creates ten places of their hydroelastic interaction with the rotor. Result is the cylindrical whirl of rotor with half velocity of rotation.

It is necessary to simulate primarily the flexible rotor including its blade wheels at the nominal velocity of rotation (167Hz) and next the interaction between the rotor and seals and bearings as well as with neighbour elements of arrangement. For the first, the simplest way is to use the finite element method inside programs such as Ansys [1-4] and to perform the modal analysis of non-rotating rotor under free boundary conditions (figure 1). There is a physical correspondence for “free rotor”. Firstly, it is a full ascent on the liquid layer i.e. the rotor doesn’t contact with any stator elements, and secondly, it is an absence or a significant flaccid on the end couplings to neighbour parts of the pump.

![Figure 1. Finite element model of free rotor for pump (35319 elements).](image)

2. Simulation of free rotor for pump

The natural oscillations are calculated at the permitted small separation between the impeller wheels and antifriction-technological bushings. Such separation is forbidden only for edge mates between the shaft and two end bushings or screw-nuts. Also the relative slipping with friction coefficient of 0.1 is allowed between the wheels and shaft as well as between the intermediate bushings and shaft due to an absence of initial clearances (between them). The remaining couplings or contact pairs between the intermediate bushings and between these bushings and wheels have small initial clearances and, as a...
result, relative slipping with the zero friction coefficient.

The accepted boundary conditions including local (contacts inside the rotor) and global (free ends of the rotor) lead to the following result (figure 2). As seen, the lowest frequency is outside the operating range of the pump and corresponds to the bending vibrations of the rotor.

![Figure 2. First eigenfrequency and eigenmode of free rotor.](image)

It is also clear the high-frequency wheels with blades can be represented as some inertial-equivalent disks of the simple cylindrical shape. Such substitution allows to reduce significantly a dimension of the finite element model and to simplify further calculations by the switching from wheels with blades and adjacent bushings to inertial equivalent disks (figure 3).

![Figure 3. Photography and solid model of blades wheel and its inertial equivalent disk.](image)

Figure 4 illustrates a finite element model of the rotor with inertia-equivalent disks. The disks arrange on the pump shaft so that their inertia centers coincide approximately with places of inertia centers of the wheels. This simplified model has much fewer finite elements. The couplings conditions on coincident cylindrical surfaces of the model are the same. For parallel surfaces to the rotor axis, there is a gapless contact with possibility of the separation during vibrations in the normal direction and the slipping along the axis with friction coefficient of 0.1. For perpendicular surfaces to the rotor axis, there is the same only with zero friction coefficient.

![Figure 4. Finite element model of free rotor with inertial equivalent disks](image)

(Total due to reduced dimension is 4105 elements)
Verification calculation of the lowest natural frequency and mode is figure 5. Well accuracy of the finite element model on base of rotor with inertia-equivalent disks are achieved. So, the simplified model is applicable for further use.

3. Rotor simulation for new pump

Fulfilled finite element simulation of rotor without clearances between the wheels and shaft as well as between the intermediate bushings and shaft shows the rotor can be considered as rigid on some springs needed instead of a fluid which is in the seals and bearings clearances. The total nominal hydraulic stiffness generated by the fluid in ten annular clearances $K_h = m (\frac{1}{2} \omega_n)^2 = 322.5 \cdot 10^3 \text{N/m}$, where $m$ is rotor mass, $\omega_n = 2\pi \times 167 \text{s}^{-1}$ is nominal angular speed.

Finite-element simulation of the rotor with free ends and ten attached hydro-supports under the assumption $k_h = 32.25 \cdot 10^3 \text{N/m}$ is stiffness of each hydro-support yields a slight increase of the first eigenfrequency (figure 6).

Existence of spline-involute joints on the rotor ends corresponds to the boundary conditions of “restraint” type. Results of finite element analysis for the rotor with the restrained ends and the attached analogous springs are shown on figures 7 and 8.
Here, the bending natural frequency decreases more significant and effect of the hydro-stiffness is more noticeable. Moreover, due to the 20Hz hydraulic increasing, the standard threshold remains unattainable in the balancing theory between the rigid and flexible rotors (0.8×226≈181>167Hz). It is still acceptable to consider the rotor as a solid on elastic supports.

However, the rotor can overcome mentioned threshold in operation due to the shaft thinning by friction on seals and bearings surfaces or due to a critical wear of joints and couplings. Below, the second is assumed, i.e. critical looseness for the end joints which are modeled by the boundary conditions of “hinging” type as well as for the shaft-disks couplings which are represented with the zero friction coefficients inside them. For this state, the natural vibrations of the rotor take a form as in figure 9.

**Conclusion**

If the lowest rotor frequency falls away into an operating range due to wear, then the rectilinear shape of the rotor whirling will change to the bending mode during of pump operation. It is dangerous because the resonance comes with the stability loss due to the rotor unbalance.

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