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Development of a Rotordynamic Analysis Model for Rotor Shaft of SMART

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Abstract

A rotordynamic analysis model for rotor shaft assembly of SMART MCP was develo rotor shaft assembly consists of vertical spinning shaft, impeller, water lubricated canned motor. The analysis model includes journal bearing model, gap model be motor stator and rotor, motor dynamic model, and impeller dynamic model. Reynold is applied to predict the stiffness and damping of the axially grooved journal be solution is obtained by finite different method. Black's equation is used to ca stiffness, damping, and added mass for the small gap filled with water between the rotor of motor. Dynamic parameters of impeller are calculated using Childs' equa depicts the hydraulic imbalance forces. Electromagnetic force of canned motor is using Iwata's model. The developed analysis model was applied to investigate t speeds, vibration mode shapes, and damped responses at bearings of the conceptual MCP rotor shaft.

1. Introduction

MCP design of SMART is currently underway. The MCP plays an important circulating the primary coolant through the internals of SMART. It has to wor failure under high temperature (310 ℃) and high pressure (15 MPa). Therefore related with rotor vibration is important. It has complex structure[1] consisting spinning shaft, canned motor, water lubricated bearings, and impeller. Especially bearings are special in design with a groove along the axial direction to assure its with the circulating water through the inside of pump. Hydraulic force through th electromagnetic force induced by the motor, internally circulating fluid in the unbalance force by impeller are possible sources of serious vibration which may damage of the MCP rotor shaft. Rotordynamics handled with only structural vibrati in the beginning. Then, in the early 1960s, hydrodynamic bearings were considered in order to evaluate the stability problems of rotor. However, the interaction forces between fluid and structure such as liquid effect of gap and the excitation forces of the impeller were still not payed attention to. So nowadays adequate rotordynamic model should describe all of those phenomena and that of MCP should also do. The rotordyanmic analysis model of MCP can't be solved by theoretical method. There are two kinds of method to solve it, transfer matrix method and FEM. The FEM could establish an accurate model of rotor-bearing system with complex external forces while transfer matrix method might lose some eigenvalues and sometimes it diverge in calculation. Therefore the stability of the MCP rotor shaft will be predicted using FEM(Fig. 1).

2. Rotordynamic Analysis Model

2.1 Analysis Model of Axially Grooved Bearing

Bearings and their seal have great influence to the vibration behavior and the the rotor systems. Therefore figuring out the dynamic phenomena of bearing is t understand rotor dynamics. The structure and working conditions of the MCP be different from those of plain one. The MCP bearings have several axial groove journal(Fig. 2). The axial grooves of the journal improve performance of load ca also enhances stability of journal bearings. This kind of structure is usually found bearings which have inward-pumping spiral grooves. Analysis procedure for perfor spiral-groove journal bearing was developed simultaneously and independently by Chow[2] and by Hirs[3]. Bearing characteristics can be calculated from the distribution of the oil film and the pressure distribution is obtained through solvin equation(1) with certain boundary conditions.

$$\frac{1}{R}\frac{\partial}{\partial\theta}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial\theta}\right)+\frac{\partial}{\partial\theta}\left(\frac{h^3}{12\mu}\frac{\partial p}{\partial z}\right)=\frac{1}{2}\,\omega\frac{\partial h}{\partial\theta}+\frac{\partial h}{\partial t} \quad (1)$$

Reynolds equation which solution can be obtained by finite difference method(FDM us reasonable prediction of pressure distribution[4]. So to speak, we can get no static characteristics but also dynamic stiffness and damping coefficients of MCP be the Reynolds equation with film thickness(h) and first order expansion of pressure

$$h=h_o+\Delta x cos(\theta)+\Delta y cos(\theta) \quad (2)$$

$$p=p_o+p_x\Delta x+p_y\Delta y+p_{\dot{x}}\cdot\Delta\dot{x}+p_{\dot{y}}\cdot\Delta\dot{y} \quad (3)$$

2.2 Dynamic Model of Liquid Annular Seal

The gap filled with water between the rotor and stator of the motor acts as a which provides moderated support for the shaft. The bearing effect of the gap should be evaluated accurately when the vibration characteristics of pump shaft system is analyzed. Black[5] made an approximation equations(4) which can be used for determining stiffness and damping coefficients as well as added mass. He confirmed the accuracy of his equations by experiments, which is well-known and widely used in pump rotor dynamic analysis. From his equations the dynamic coefficients of gap are calculated if geometry and flow rate are provided.

$$\frac{\lambda}{\pi R_j P}\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = -\begin{bmatrix} \mu_o - \frac{1}{4}\mu_2\omega^2 T^2 & \frac{1}{2}\mu_1\omega T \\ -\frac{1}{2}\mu_1\omega T & \mu_o - \frac{1}{4}\mu_2\omega^2 T^2 \end{bmatrix}\begin{Bmatrix} X \\ Y \end{Bmatrix} - \begin{bmatrix} \mu_1 T & \mu_2\omega T^2 \\ -\mu_2\omega T^2 & \mu_1 T \end{bmatrix}\begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix} - \begin{bmatrix} \mu_2 T^2 & 0 \\ 0 & \mu_2 T^2 \end{bmatrix}\begin{Bmatrix} \ddot{X} \\ \ddot{Y} \end{Bmatrix} \tag{4}$$

Where the μ_0, μ_1, μ_2, are empirical coefficients which can be obtained in his pape ω, and T are the radius of journal, ambient pressure, rotational speed, and pa through the gap respectively.

2.3 Electromagnetic Force of Canned Motor

The electromagnetic force should not be the dominant force for general pum However, it may cause considerable influence to the MCP rotor system because th and damping effect in water-lubricated journal bearing installed vertically are co smaller than those of horizontal pump. Iwata's equation is applied in order to calcu

$$-\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = \begin{bmatrix} K & 0 \\ 0 & K \end{bmatrix}\begin{Bmatrix} x \\ y \end{Bmatrix} - \begin{bmatrix} -\frac{\pi B_o^2 RL}{2\mu_p d} & 0 \\ 0 & -\frac{\pi B_o^2 RL}{2\mu_p d} \end{bmatrix}\begin{Bmatrix} x \\ y \end{Bmatrix} \tag{5}$$

Where, B_0, R, L, μ_p, and d are the magnitude of magnetic flux, radius of core, len gap, permeability, and clearence of gap respectively.

2.4 Dynamic Model of Impeller

For MCP as a vertical pump, the hydraulic force induced through impeller is static load and dominant factor related to vibration and noise. Though it can be dis horizontal pump, it can not be neglected in the SMART MCP body dynamic load nature not so simple as unbalance force. It significantly depends on flow rate and conditions. Therefore the estimation of it is difficult. Childs developed an analysi calculate it and it can generally be modeled as following equations(6).

$$-\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = -\begin{bmatrix} K & -k \\ -k & K \end{bmatrix}\begin{Bmatrix} X \\ Y \end{Bmatrix} - \begin{bmatrix} C & c \\ -c & C \end{bmatrix}\begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix} - \begin{bmatrix} M & m_c \\ -m_c & M \end{bmatrix}\begin{Bmatrix} \ddot{X} \\ \ddot{Y} \end{Bmatrix} \tag{6}$$

Where, K and k are equivalent stiffness, C and c equivalent damping, M and m_c equivalent mass of impeller.

2.5 Rotordynamics Analysis of MCP Rotor Shaft

SMART MCP rotor is supported on water-lubricated light load long journal bearin are so complex that it is difficult to get the static and dynamic characteristics o The case of MCP is more flexible than normal pump because it has a long rotor fixed on upper flange of the reactor, so the foundation may have considerable influ bearing stiffness and should be considered. MCP rotor also has to bear the hydr from impeller and the electromagnetic force from the canned motor. The model f rotor developed by FEM shows high performance on predicting bending and shear transverse and rotary inertia, axial force and gyroscopic effect. In addition, FEM superior to any other numerical methods because it represents a very general ap structural dynamics. Especially for MCP rotor systems in which motion of case m negligible, it can provide us with a good solution. In FEM model of rotor system beam elements are used to describe rotor structure. Through the solution of FEM get critical speed, stable response and stability criteria of rotor system. The MCP FEM consists of 36 elements, 3 journal bearings, 2 gaps, impeller, motor, and thr which withstands the axial force of shaft. The vibration mode shapes of the rotor shown in Fig. 3. The results of analysis show that there are two critical speeds rpm as shown in Fig. 4. The damped responses of journal bearings are represented Fig. 5 shows that the maximum amplitude of vibration near the critical speeds is μm and the responses of journal bearings near operating speed are around 20 μm. it is judged that although there are two critical speeds within the operating stability of the MCP rotor shaft is assured since the maximum amplitude is acceptable level.

3. Conclusions

An analysis model to investigate the rotor shaft dynamic behavior of the SMA was developed. A preliminary evaluation of the rotor shaft dynamic character performed using the conceptual design data. The results show that the MCP rotor maintain its stability during operation. However, a design optimization of the ro required to eliminate the critical speed observed within the operating range.

Acknowledgment

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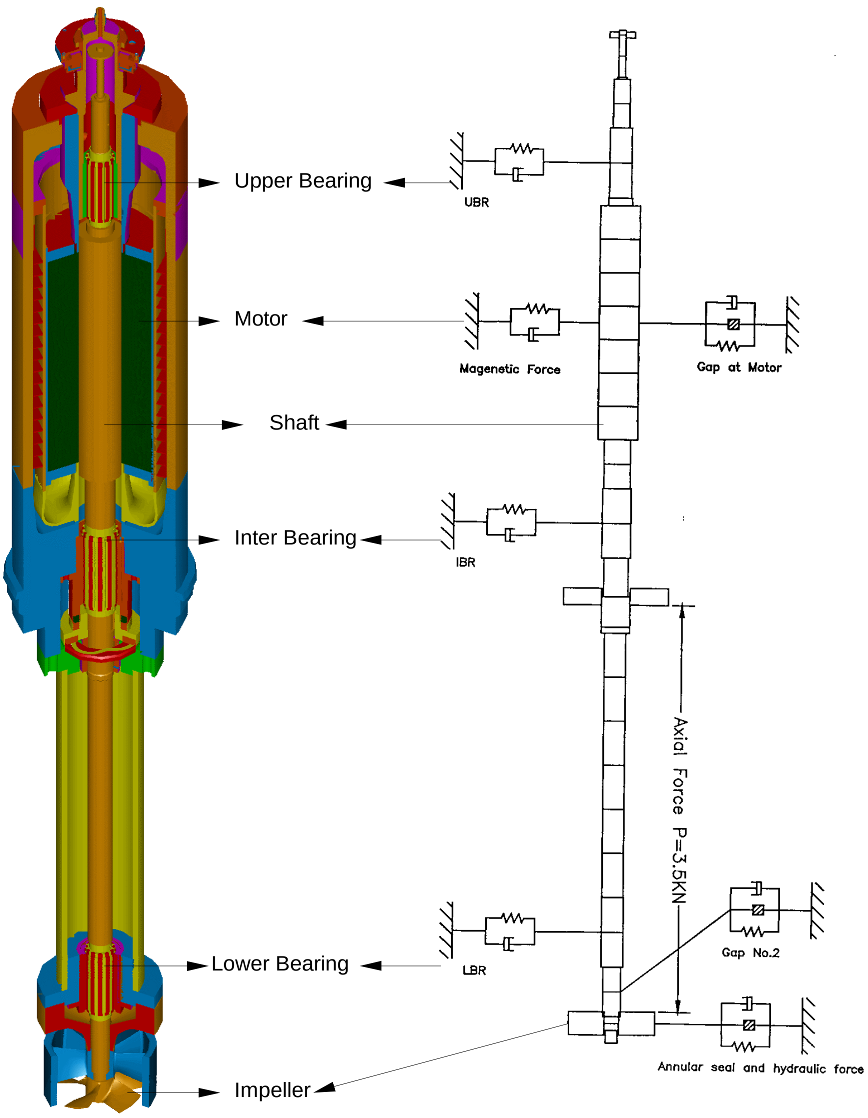

Fig. 1 SMART MCP Structure and Its Rotordynamic Analysis Model

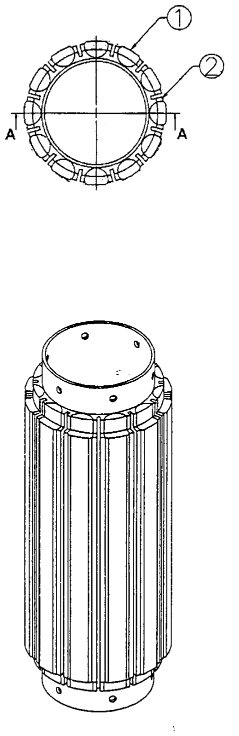

Fig.2 Structure of Journal Bearing

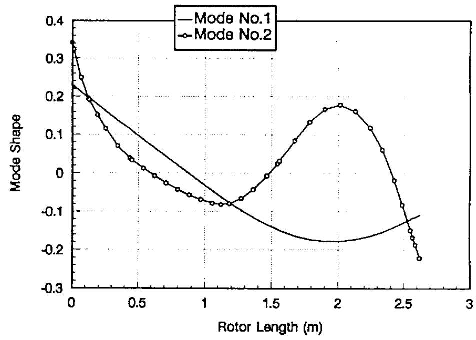

Fig.3 Mode Shapes of MCP Rotor Shaft

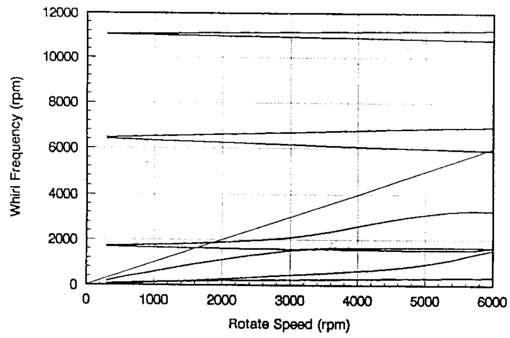

Fig.4 Critical Speeds of MCP Rotor Shaft

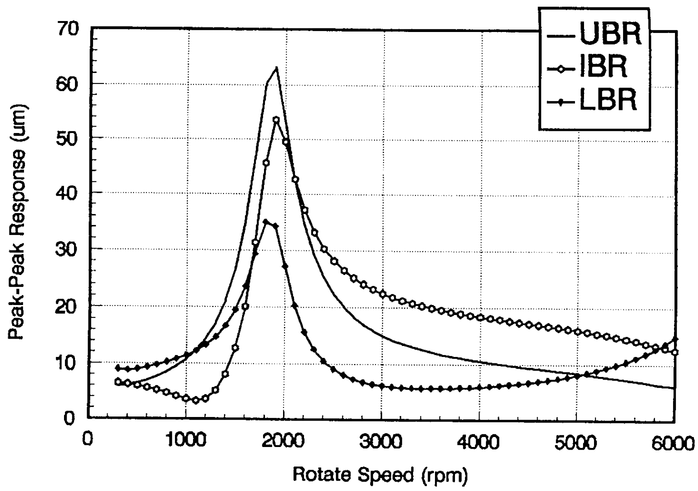

Fig. 5 Damped Responses at Journal Bearing