Transactions of the 15th International Conference on
Structural Mechanics in Reactor Technology (SMiRT-15)
Seoul, Korea, August 15-20, 1999

A Rotordynamic Analysis Model for Rotor Shaft of SMART MCP

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ABSTRACT

A rotordynamic analysis model has been developed in order to predict the dynamic behavior of the SMART MCP rotor shaft. The analysis model includes the journal bearing model, the gap model between the motor stator and rotor, the motor dynamic model, and the impeller dynamic model. The developed analysis model is applied to investigate the critical speeds, vibration mode shapes, and damped responses for the conceptually designed SMART MCP rotor shaft. A stability analysis of the rotor shaft is carried out on these results.

1. INTRODUCTION

The main purpose of the SMART MCP is to circulate the primary coolant continuously through the internals of reactor for ten years. The MCP has to work without failure under bad environments like high temperature and high pressure. It has a complex structure consisting of a vertically spinning shaft, a canned motor, water lubricated bearings, and an impeller[1,2,3]. In particular, the journal bearings are special in design, having a groove along the axial direction to assure bearing lubrication of the circulating water through the inside of pump. Hydraulic force through the impeller, electromagnetic force induced by the motor, internally circulating fluid in the MCP, and an unbalanced force by the impeller are possible sources of serious vibration which may lead to the damage of the MCP rotor shaft. Therefore the pump integrity related with rotor vibration should be evaluated and verified before the pump is used in field. Rotordynamics handled only the structural vibrations of the rotor in the starting stage of research. Then, in the early 1960s, hydrodynamic bearings were considered in order to evaluate the stability problems of the rotor. However, the interaction forces between fluid and structure such as the liquid effect of gap and the excitation forces of the impeller were still not given attention. Nowadays, at last, an adequate rotordynamic model can describe all of those phenomena. The rotordynamic analysis model of SMART MCP includes structural vibration analysis, a hydrodynamic bearing model, and fluid-interaction phenomena. It can't be solved by theoretical methods. There are two kinds of methods to solve it, the transfer matrix method and the FEM. The FEM could establish an accurate model of a rotor-bearing system with complex external forces while the transfer matrix method might lose some eigenvalues and sometimes may diverge in calculation.

In this paper, the stability of the MCP rotor shaft is predicted using FEM(Fig. 1).

2. ROTORDYNAMIC ANALYSIS MODEL

2.1 Analysis Model of Axially Grooved Bearing

Bearings and their seals have great influence on the vibration behavior and the stability of the rotor system. Figuring out the dynamic phenomena of the bearings is the key to understand rotor dynamics. The structural and working conditions of the MCP bearings are different from those of plain ones. The MCP bearings have several axial grooves on the journal(Fig. 2) and rotate under very high temperature. The axial grooves of the journal improve performance of the load capacity and also enhance stability of journal bearings. This kind of structure is usually found in gas-film bearings having inward-pumping spiral grooves. Analysis procedures for performance of spiral-groove journal bearings were developed simultaneously and independently by Vohr and Chow[4] and by Hirs[5]. The bearing characteristics can be calculated from the pressure distribution of the oil film and the pressure distribution is obtained through solving Reynolds Eq.(1) with certain boundary conditions.

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

Here, R is the radius of journal, h is the clearance, μ is the viscosity of lubricant, p is the pressure, ω is the rotational speed of the rotor, θ is the tangential coordinate, and G_θ, G_Z are the turbulent flow coefficient of tangential and axial coordinate, respectively. The Eq.(1) whose solution can be obtained by finite difference method(FDM) provides us with a reasonable prediction of pressure distribution. So to speak, we can get the static characteristics of journal bearing such as the dynamic stiffness and damping coefficients.

2.2 Dynamic Model of Liquid Annular Gap and Impeller

The gap filled with water between the rotor and stator of the motor acts as a bearing, which provides moderated support for the shaft. The bearing effect of the gap should be evaluated accurately when the vibration characteristics of the pump shaft system is analyzed. For the MCP as a vertical pump, the hydraulic force induced through the impeller is the main static load and dominant factor related to vibration and noise. Though it can be disregarded in the horizontal pump, it can not be neglected in the SMART MCP. The fluid-structure forces on gap and impeller are simplified by Hirs and Childs[6].

$$-\begin{Bmatrix} F_x \\ F_y \end{Bmatrix}=-\begin{bmatrix} K & -k \\ -k & K \end{bmatrix}\begin{Bmatrix} X \\ Y \end{Bmatrix}-\begin{bmatrix} C & c \\ -c & C \end{bmatrix}\begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix}-\begin{bmatrix} M & m \\ -m & M \end{bmatrix}\begin{Bmatrix} \ddot{X} \\ \ddot{Y} \end{Bmatrix} \tag{2}$$

Where, F_x and F_y are the linearized fluid-structure reaction forces in X and Y directions, respectively. K and k are of stiffness coefficients, C and c are damping coefficients, M and m are mass coefficients.

The SMART MCP has some gaps which form flow path in pump without leak tightening seal. The gaps are filled with water and may be regarded as smooth plain seal. Therefore, if geometry and flow rate are provided, rotordynamic coefficients can be obtained by well known analytic model.

The dynamic coefficients of impeller can be calculated by Eq.(2) and the hydraulic imbalance induced from some imperfection in the flow path through impeller blade can be expressed following

$$F_{HI} = \alpha K_{HI}\ Db^{*}\omega^{2} \tag{3}$$

Where F_{HI} is the rotating load magnitude, D is the diameter of impeller, b^* is the exit width of the impeller including the side walls, and K_{HI} and $\alpha = \Delta P/\omega^2$ are the empirical coefficients for specific impeller.

2.3 Electromagnetic Force of Canned Motor

The electromagnetic force should not be the dominant force for general pump rotors. However, it may cause considerable influence to the MCP rotor system because the stiffness and damping effect in the water-lubricated journal bearing installed vertically are comparatively smaller than those of the horizontal pump. Iwata's equation is applied to calculate the dynamic coefficients of canned motor.

$$-\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = \begin{bmatrix} K & 0 \\ 0 & K \end{bmatrix}\begin{Bmatrix} x \\ y \end{Bmatrix} - \begin{vmatrix} -\dfrac{\pi B_o^2 RL}{2\mu_p d} & 0 \\ 0 & -\dfrac{\pi B_o^2 RL}{2\mu_p d} \end{vmatrix}\begin{Bmatrix} x \\ y \end{Bmatrix} \tag{3}$$

Where, $B_{0,}$ R, L, μ_p, and d are the magnitude of magnetic flux, radius of core, length of core, permeability, and the clearance of gap between motor stator and rotor, respectively. The displacement x, y are shown in Fig. 3.

2.4 Rotordynamics Analysis of MCP Rotor Shaft

The SMART MCP rotor shaft is supported on a water-lubricated light load long journal bearings and thrust bearing. The MCP rotor shaft also has to bear the hydraulic force from the impeller and the electromagnetic force from the canned motor. The case of MCP is more flexible than a normal pump because it has a long rotor and is only fixed on the upper flange of the reactor. So the foundation may considerably influence on the bearing stiffness and should be considered to predict the dynamic characteristics of rotor shaft. But in this paper, only flexible rotor shaft without MCP case are modeled by FEM because FEM shows high performance on predicting bending and shear deflection, transverse and rotary inertia, axial force and gyroscopic effect. In addition, FEM is probably superior to any other numerical method. Especially for the MCP rotor systems including pump case whose analysis model will be completed to next stage, it can provide us with a good solution. In the FEM model of SMART MCP rotor shaft systems, slender beam elements are used to describe rotor structure. Through the solution of FEM, we get critical speeds, stable responses and stability criteria of a MCP rotor shaft. The FEM model of MCP rotor shaft consists of 36 elements, 3 journal bearings, 2 gaps, impeller, motor, and thrust bearing which withstands the axial force of the shaft. The main parameters used in the analysis model of SMART MCP rotor shaft are shown in Table 1.

3. ANALYSIS RESULTS

Among the natural frequencies(Fig. 4) obtained from the analysis model of rotor shaft, the first ones(1B, 1F) lie in very low speed. The second ones(2B, 2F) are appeared near 1800 rpm and the frequency difference between backward and forward natural frequency is not more than a few rpm. The third ones(3B, 3F) are presented in high speed over 5000 rpm. The first vibration mode doesn't effect on the rotor shaft stability because there isn't no critical speed within the operating speeds, 900 rpm and 3600 rpm. The damping coefficients(Fig. 5) corresponding to the first vibration mode(Fig. 6) are so heavy that the level of vibration amplitude may be suppressed. However, the second vibration mode is inevitably met when pump starts up and shuts down though rotor shaft doesn't rotate at the speeds of the second natural frequencies. In addition, the damping value of the second vibration mode is so small that excessive vibration response may be expected. Bending vibration mode is appeared in the third natural frequency with almost zero damping coefficient. It is recommended to separate the operating speeds far from the third natural frequency because large bending vibration mode with small damping value may arise high vibration response like Fig. 7.

4. CONCLUSIONS

The dynamic analysis model of the rotor shaft for SMART MCP has been completed. The preliminary evaluation results performed with the conceptual design data show that the MCP rotor shaft has critical speeds within the operating range. Having the small vibration responses at those critical speeds, the rotor shaft passes the critical speeds easily when pump starts up and shuts down. It can be concluded that the rotor shaft will maintain its stability during operation because the operating speeds are different from the natural speeds of the rotor shaft. These results were calculated with the preliminary data. Therefore, it is required to judge the stability of rotor shaft using sufficient data.

ACKNOWLEDGEMENT

This work has been performed under the nuclear research and development program sponsored by the Ministry of Science and Technology, Korea.

REFERENCES

1. J. S. Park, et al, "The Conceptual Design of SMART MCP," *Proc. of the Korean Nuclear Society Spring Meeting*, Vol. 2, 885-890(in Korean), Seoul, Korea, May 1998.
2. D. H. Koo, et al, "Design and Analysis of MCP for SMART(System-integrated Modular Advanced Reactor)," *KIEE Summer Annual Conference '98*, pp 228-231(in Korean), Korea, July 1998.
3. J. S. Park, et al, "Development of a Rotordynamic Analysis Model for Rotor Shaft of SMART MCP," *Proc. of the Korean Nuclear Society Autumn Meeting*, Seoul, Korea, October 1998.
4. J. H. Vohr & C. Y. Chow, "Characteristics of Herringbone Grooved Gas Lubricated Journal Bearings," *Journal of Basic Engineering*, 87:568-576.
5. G. G. Hirs, "The Load Capacity and Stability Characteristics of Hydrodynamic Grooved Journal Bearings," *American Society Lubrication Engineering Transaction*, 8:296-305.
6. Dara Childs, *Turbomachinery Rotordynamics Phenomena, Modeling and Analysis*, A Wiley-Interscience Publication, 1993.

7. H. F. Black et al, "Effects of high Pressure Ring Seals on Pump Rotor Vibrations," *ASME Paper*, No. 71-WA/FE-38, 1971.
8. C. W. Lee, *Vibration Analysis of Rotors*, Kluwer Academic Publishers, 1993.

Table 1 Parameters of SMART MCP rotor shaft

Parameters	Value
Shaft Length	4170 mm
Number of Journal Bearing	3
Number of Thrust Bearing	1
Number of Impeller	1
Operating Speeds	900/3600 rpm
Power Consumption	190 kW
Flow Rate of Water	1982 m^3/h
Length of UJB and IJB	256 mm
Length of LJB	228 mm
Number of Pad of UJB and IJB	22
Number of Pad of LJB	20
Clearance of Journal Bearing	0.15 mm

UJB : Upper Journal Bearing, IJB : Intermediate Journal Bearing
LJB : Lower Journal Bearing

Table 2 Critical speed of SMART MCP rotor shaft

Vibration mode	Critical speed(rpm)
2F	1800
3F	5000
4F	5700

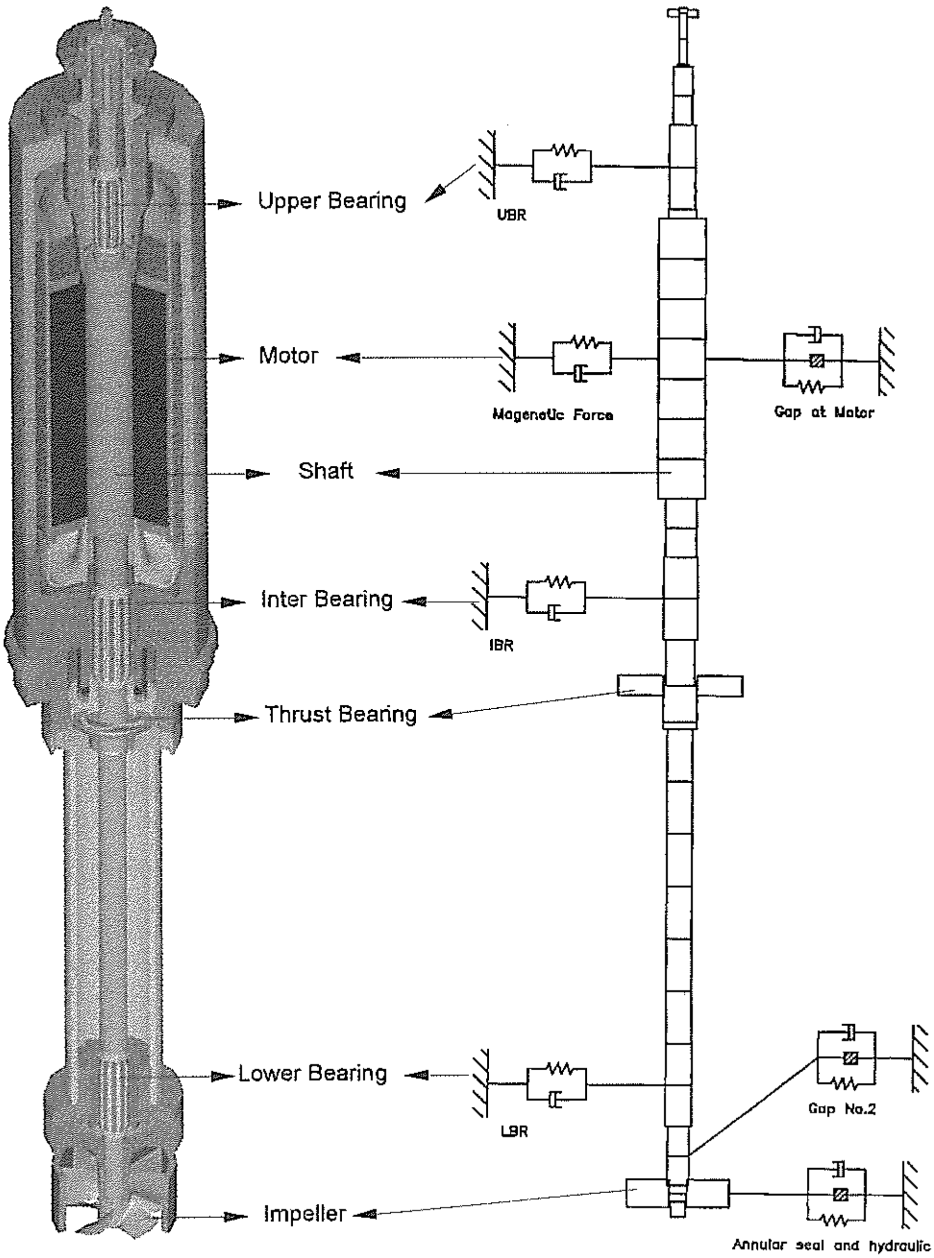

Fig. 1 SMART MCP structure and its rotordynamic analysis model

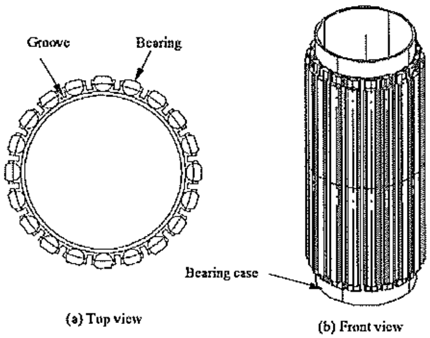

Fig. 2 The journal bearing of SMART MCP

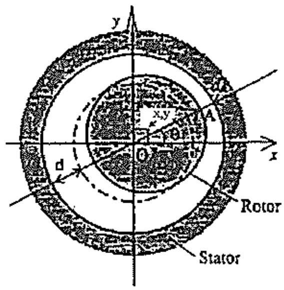

Fig. 3 Canned motor pump

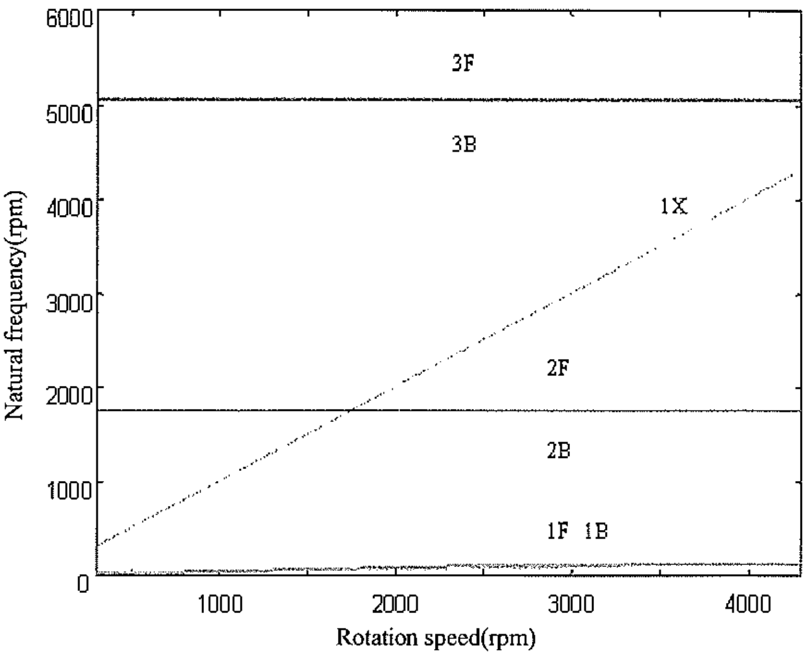

Fig. 4 Natural frequencies of SMART MCP rotor shaft

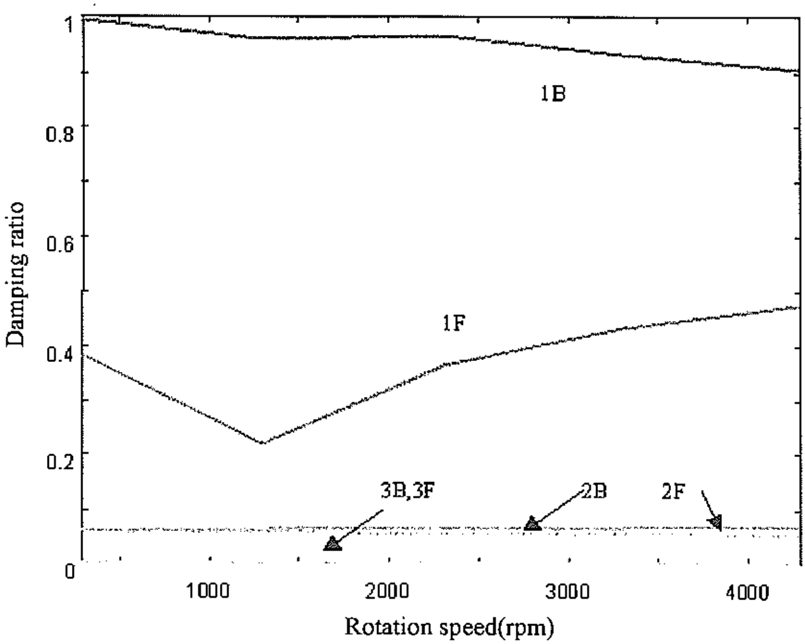

Fig. 5 Damping ratio at each natural speeds

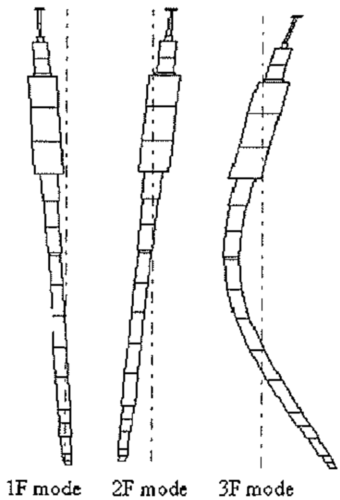

Fig.6 Vibration modes of MCP

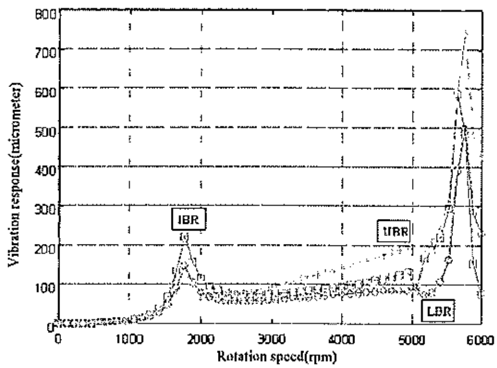

Fig. 7 Damped response