Unbalance Force Analysis of Primary Pump Shaft in PGSFR

Sung-Kyun Kim [a*], Gyeong-Hoi Koo [a]

[a] *Korea Atomic Energy Research Institute, Dukjin-Dong, Yuseong-Gu, Daejeon, South Korea*
[*] *Corresponding author: sungkyun@kaeri.re.kr*

1. Introduction

All rotating shafts, even in the absence of external loads, have always eccentric distance due to the manufacturing deviations, material defects, and installation error. This unbalance mass resulting in the eccentric distance of the rotating structure causes resonance at critical speeds. The resonance is the state at which the harmonic loads are excited at their natural frequencies causing shafts to vibrate excessively. This vibration of large amplitude causes shafts to bend and twists significantly and leads to permanent failure. Hence, the determination of these rotor-dynamic characteristics of rotating shaft in the mechanical design stage is much important.

In this paper following issues are addressed:

- Describe how to calculate the unbalance force.
- Predict the natural frequency variations and identify critical speeds within or near the operating speed range of a shaft for consideration of the unbalance bearing stiffness and damping.
- Determine the available support positions of a pump shaft to avoid resonance within operating speed.
- Perform an unbalance response analysis of a shaft in order to calculate shaft displacement and quantify the forces acting on the shaft support that are caused due to shaft unbalance.

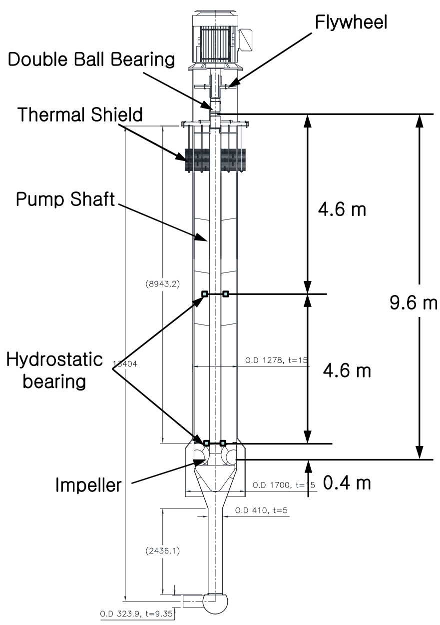

Fig. 1 Layout of primary pump of PGSFR

2. Methods and Results

In this section, the unbalance of a pump shaft calculated by following ISO 1940-1[1] is described. In addition, its rotor-dynamic characteristics obtained from ANSYS [3] are addressed here.

2.1 Calculation of Unbalance Force

International standard ISO 1940-1 gives a rule in order to calculate an acceptable residual unbalance, having that gross unbalance deficiencies and useless and expensive balancing works are avoided. ISO 1940-1 specification presents various G-grades of allowable unbalance for different types of rotating machinery. For pumps, a standard value has been G 6.3. "G" grade designations represent the magnitude of the product ($e_{per} \times \Omega$)-permissible rotor eccentricity times rotational speed expressed in mm/sec. Meanwhile, permissible residual unbalance U_{per} can be derived on the basis of a selected quality grade G by the following equation:

$$U_{per} = 1000 \frac{(e_{per} \cdot \Omega) \cdot m}{\Omega} \tag{1}$$

where ,
U_{per} is the numerical value of the permissible residual unbalance (g · mm).
$(e_{per} \cdot \Omega)$ is the numerical value of the selected balance quality grade (mm/s).
m is the numerical value of the shaft mass (kg).
Ω is the numerical value of the angular velocity of the service speed (rad/s).

For the primary pump shaft of PGSFR, the calculated value of the permissible residual unbalance is

$$U_{per} = 1000 \frac{(e_{per} \cdot \Omega) \cdot m}{\Omega} = 63.092 \times 10^3 g \cdot mm.$$

2.2 Rotor-dynamic Analysis

In order to perform the rotor-dynamic analysis for the primary pump shaft, following assumptions have been made.

- Mass of the impeller is a point mass.
- Bearings are fixed in space.
- Bearing stiffness and damping are symmetric.
- Pump shaft has uniform diameter and thickness.
- Shaft material is 316SS and temperature of pump shaft is 390 °C.

Fig. 2 shows the FEM model for the rotor-dynamic analysis. Previously on the FEM shaft model, bearing #3 was added above 1 m from bearing #2 to satisfy that the critical speed fulfills 1.5 times operating speed (1125 rpm). However, since the more a number of bearings increase the more mechanical design of the pump is complicated, we decided to increase the diameter of a shaft from 130 mm to 230 mm instead of increasing a number of bearings. Fig. 3 shows the Campbell diagram for the modified pump shaft. This figure shows that the critical speed of shaft is above the design criteria.

In the FEM model, the unbalance force is applied at the impeller position which is the mass center of pump shaft. For boundary conditions, all nodes are fixed about an axial rotation and also all bearing nodes are constrained about all degree of freedom. For the bearing No.1 axial displacement is constrained because of a thrust bearing load.

Fig. 4 shows the bearing deflections along with shaft rotating speed. As shown in this figure, the deflections increase with an increase in rotating speed. It is shown that the maximum deflection in the operation speed 750 rpm is less 0.1 mm. This value is below the hydrostatic bearing clearance (0.75 mm)[2].

Fig. 5 shows each bearing loads along with shaft rotating speed. Among bearings bearing #2 is got the maximum bearing load, while bearings bearing #1 got the minimum bearing load.

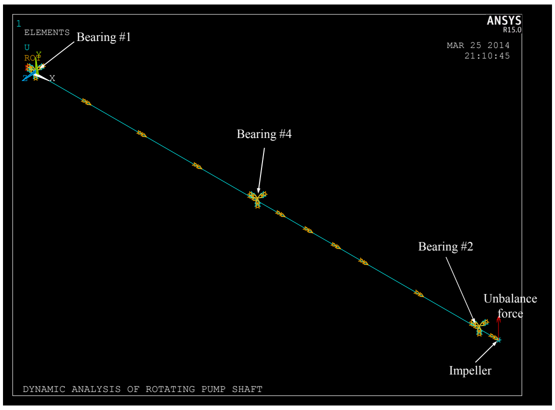

Fig. 2 Finite element model and boundary conditions for the unbalance force analysis

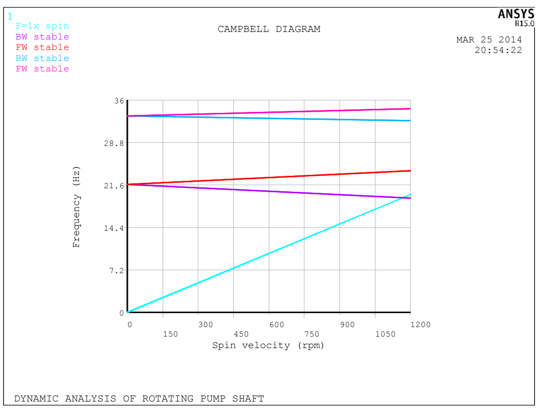

Fig. 3 Campbell diagram of pump shaft

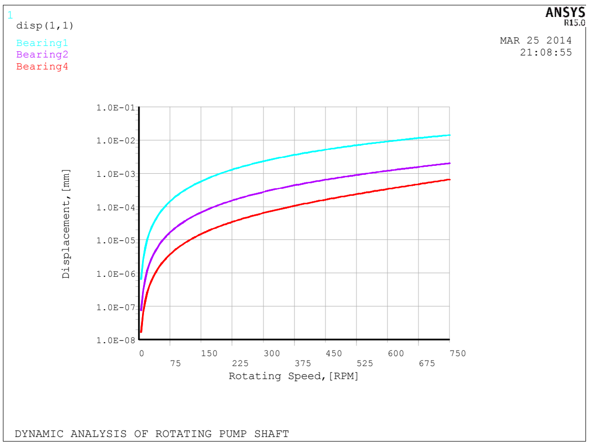

Fig. 4 Displacements at each bearing locations with respect to rotating shaft speed

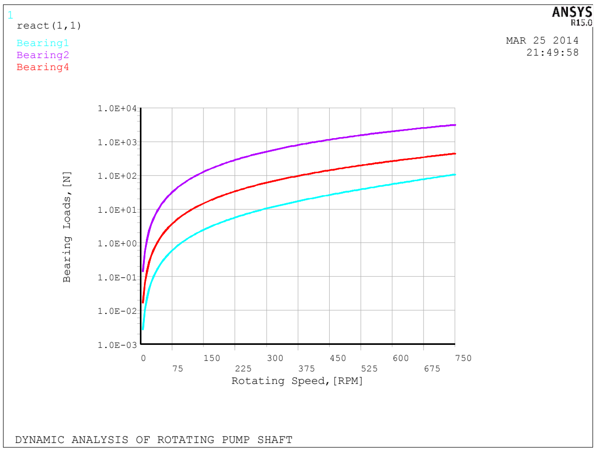

Fig. 5 Bearing loads at each bearing locations with respect to rotating shaft speed

3. Conclusions

In this paper, rotor-dynamic characteristics for the primary pump shaft were obtained. On the basis of the results, we decided to almost double the shaft diameter. By virtue of this design change, the revised design can be satisfied with the critical speed design criteria as well as can decrease a number of support bearings. In the results of unbalance analysis, it was figured that the maximum bearing deflections and loads are within the design criteria.

REFERENCES

[1] ISO 1940-1, Mechanical vibration-Balance quality requirements for rotors in a constant (rigid) state, 2003.

[2] R. Fischer, Presentation of PGSFR cooling pump shaft analysis, 2013.

[3] ANSYS Inc. Manual, 2012.

[4] Hydrostatic Journal Bearings, Notes 12(b), Luis San Andres, Texas A&M University.

[5] C. W. Lee, Vibration Analysis of Rotors, Kluwer Academic Publishers, London, 1993.

[6] W. B. Rowe, Hydrostatic and Hybrid Bearing Design, Butterworths, 1983.