Evaluation of Rotordynamic Stability of a Steam Turbine Due to Labyrinth Seal Force*

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Abstract

This paper describes the evaluation of unstable vibration caused by the seal force, which is known as "Steam Whirl" in a steam turbine. Stability of a steam turbine is evaluated by complex eigenvalue analysis of rotordynamics model considering the dynamics of seals, rotor, bearings and pedestals. A commercial CFD program is employed to estimate the dynamic coefficients of labyrinth seal. The labyrinth seal of a large scales steam turbine is taken as an object of analysis and a 3D model with eccentric rotor is solved to obtain the rotordynamic force components. The rotordynamic force is derived by integrating the pressure on the rotor surface. Evaluation formula is formed from the results of numerical calculation, which is used to predict the dynamic coefficient of each seal in a steam turbine. Then rotordynamics model of total system including seal is constructed and stability is evaluated by complex eigenvalue analysis. This procedure is applied to the design of steam turbines and enables the optimization of the turbine structure considering the efficiency and stability.

Key words: Labyrinth Seal, Steam Whirl, CFD, Instability, Steam Turbine

1. Introduction

Labyrinth seals are used in various kinds of turbo machines to reduce internal leakage flow. The working fluid or, the gas passing through the rotor shaft labyrinth seals, often generates driving force components that may increase the unstable vibration of the rotor. In a steam turbine this unstable vibration is known as "Steam whirl." Recently precise prediction of the instability by steam whirl is getting more important because the demand for high efficiency increases the number of the stages and, as a result, the rotor becomes more flexible and the tolerance for steam whirl decreases.

For predicting the instability of rotor-bearing-seal-system, the complex eigenvalue analysis technique is used. Analysis model consists of beam elements as rotors and spring and damping elements as seals and bearings. It is important to know the accurate rotordynamic force components of seals and bearings for precise evaluation of the stability.

In 1980's the analytical methods employing bulk flow models were developed to estimate dynamic coefficients of seals. They are still used in the rotating machinery industries and have helped engineers to design stable rotating machinery. The bulk flow models are so simplified that the program applying them can be run on a personal computer with standard memory. However, they require the empirical correction to have the analysis predictions match the experimental results [1-6].

Recently the progress of computer technologies makes it possible to utilize a CFD program for predicting the rotordynamic force of rotating machinery seals. Rhode [7,8], Ishii [9] and Kwanka [10] calculated the rotordynamic force of gas labyrinth seals using the CFD programs they developed. Moore [11] used SCISEAL, a CFD code specially developed for the analysis of the seals, and compared the results with experiment results and the bulk flow programs. Hirano [12] showed that a general-purpose commercial CFD code could be used to calculate the rotordynamic force of labyrinth seals in a steam turbine and a compressor. A 3D model with eccentric rotor was solved to obtain the rotordynamic force. The leakage flow and rotordynamic force from a CFD code were compared with the results by an existing bulk flow analysis computer program [4-6].

The precise estimation of the bearing dynamic properties is also important to predict stability of the system accurately. Hisa [13], Sakakida [14] and Ikeda [15] carried out the excitation tests on the large-scale test rig and measured the dynamic properties of various types and sizes of bearings. The types of bearings tested were 2 lobe and tilting pad bearing. The diameters are 508mm(20 inches), 584mm(23inches) and 813mm(32 inches).

This paper describes our approach to accurate prediction of stability of a steam turbine due to steam whirl. Firstly, the analytical model of rotor-bearing-seal system for stability evaluation is shown. Secondly, the analysis of seal dynamics is shown. A general-purpose commercial CFD code is used to estimate the seal force. The dependency of the seal force on the parameters such as the preswirl, the pressure difference, etc is evaluated. Evaluation formula is formed from the results of numerical calculation, which is used to predict the dynamic coefficient of each seal in a steam turbine.

Next, the dynamic properties of the bearing are described. The tilting pad bearings are usually used to support the high-pressure (HP) or intermediate pressure (IP) turbine rotors, where steam whirl is sometimes concerned about. For the calculation of the dynamic properties of tilting pad bearing, the effect of the flexibility of the contact point between the backside of the pad and the housing of the bearing is considered. The calculated results were compared with the test results.

Finally, the stability of a steam turbine is shown. Analytical model including the seals and the bearings dynamic properties described above is solved and the stability is evaluated.

2. Analytical Model

Figure 1 shows an analytical model used for stability analysis. It consists of rotors, seals, bearings and pedestals. Rotors are modeled as mass and beam elements. Seals are modeled as spring elements, which have direct and cross coupled terms. Bearings are modeled as spring and damping elements. Pedestals are modeled as spring, damping and mass elements.

Fig. 1 Analytical Model of rotor-bearing-seal system

Figure 2 shows a flow of stability analysis. The finite element model of total system is

constructed combining the models of the rotors, seals, bearings and pedestals. Then the stability is evaluated by complex eigenvalue analysis. The real part of eigenvalue shows the stability of the system.

Fig. 2 Procedure of Stability Analysis

3. Seal Force Evaluation

3.1 Test

A general-purpose commercial CFD code is used to calculate the seal force of the labyrinth seal. First, the test is carried out to validate the application of CFD code to the seal force analysis. Figures 3 and 4 show the photo of the test rig and the labyrinth seal tested, respectively. This test uses the air as fluid. The seal has 5 fins and 4 chambers. It is designed to have equivalent Reynolds Number to that used in a steam turbine. The Reynolds number of the seal is approximately 2×10^4 , where the representative length is the radial clearance of the seal and the representative velocity is the average velocity at the tip gap in the axial direction. The rotor has a dynamic eccentricity, i.e. there is a difference between the geometrical center and the rotational center of the rotor. The pressure in a chamber fluctuates in synchronization with the rotation of the rotor. The pressure in the chambers is measured and compared with calculated result by the CFD code. Differentiation scheme and the turbulent model used in CFD are MARS and $k-\varepsilon$ /High Reynolds, respectively.

Fig.3 Test Rig for Seal Force

Fig.4 Tested Labyrinth Seal

Figure 5 shows the average pressure of each chamber measured on the stator surface and the pressure calculated by CFD. The horizontal axis shows the pressure difference, which is the difference of the pressure between the inlet and the outlet of the seal. Figure 6 shows the amplitude of the pressure fluctuation in the circumferential direction, which causes the seal force. Figure 7 shows the peak position of pressure fluctuation. The peak

position indicates the angle of the pressure peak from the direction of eccentricity. In these figures CFD and Test indicate the results obtained by the numerical analysis and the test, respectively. The numbers #1-#4 indicate the number of the chambers. The rotation speed of the rotor is $4,000\text{min}^{-1}$. The results by test and calculation show good agreement. And the propriety of the application of CFD to the seal force analysis is confirmed.

Fig. 5 Comparison of test and CFD -- average pressure

Fig. 6 Comparison of test and CFD -- pressure fluctuation

Fig. 7 Comparison of test and CFD -- peak position

3.2 Seal force of steam turbine

The calculations on the flow in the labyrinth seals of the steam turbine are carried out by the CFD code. One example shown in this section is the typical labyrinth seal used in the gland seal between the high pressure (HP) and intermediate pressure (IP) section as shown in Fig. 8. Figure 9 shows the analytical model and boundary condition. This seal is Hi-Lo type seal, where high section and low section are aligned one after another. Table 1 shows the condition and specification of the seal. This is typical example of the seal used in the first segment of the gland seal between HP and IP sections. Figure 10 shows the mesh used

for the calculation. Total number of mesh is about 1.5 million. In this analysis the rotor has static eccentricity and no whirl. The eccentricity (ϵ) is 0.078mm, which is equal to 10 % of seal clearance. Differentiation scheme and the turbulent model used in CFD are MARS and k- ϵ /High Reynolds, respectively. And the fluid is assumed to be compressible and ideal gas. The mol weight is 19.7 so that the density is derived correctly by ideal gas equation under the condition of the steam in the seal.

Table 1. Specification and Condition of the Seal

Inlet Pressure	14.7MPa
Outlet Pressure	13.0MPa
Rotation Speed	3600min ⁻¹
Number of Fins	11
Clearance	0.78mm
Diameter of high section	Φ550mm

Fig. 8 Gland seal between HP and IP sections

Fig.9 Analytical model and boundary condition

Fig.10 Mesh used for CFD

Figure 11 shows the velocity vector. There are two vortices in each chamber. The pattern of the vortices in the chamber with the odd number is different from that with the even number.

Figure 12 shows the pressure distribution on the rotor surface. The pressure drop in the high section is larger than that in the low section. The seal force can be obtained by integrating the pressure distribution on the rotor surface. Defining r-axis in the direction of eccentricity and t-axis perpendicular to r-axis, the seal forces in r and t axis are expressed by

following equations.

$$F_r = \int_{z_1}^{z_2} \int_0^{2\pi} p(z, \theta) \cos(\theta) R d\theta dz$$

$$F_t = \int_{z_1}^{z_2} \int_0^{2\pi} p(z, \theta) \sin(\theta) R d\theta dz$$

where

$p(z, \theta)$: pressure distribution on the rotor surface

z : coordinate in the axial direction

θ : coordinate in the circumferential direction,
 $\theta=0$ in the direction of eccentricity

R : Radius of rotor

Dynamic coefficient of the seal can be obtained by dividing the force by the eccentricity. The stiffness is defined by the following equations.

$$k_r = -\frac{F_r}{\varepsilon} = -\frac{1}{\varepsilon} \int_{z_1}^{z_2} \int_0^{2\pi} p(z, \theta) \cos(\theta) R d\theta dz$$

$$k_t = \frac{F_t}{\varepsilon} = \frac{1}{\varepsilon} \int_{z_1}^{z_2} \int_0^{2\pi} p(z, \theta) \sin(\theta) R d\theta dz$$

Then k_r is the direct stiffness and k_t is the cross coupled stiffness. The force that works on the rotor is expressed as follows:

$$\begin{pmatrix} F_x \\ F_y \end{pmatrix} = - \begin{pmatrix} k_r & k_t \\ -k_t & k_r \end{pmatrix} \begin{pmatrix} x \\ y \end{pmatrix}$$

, where x, y are the displacements of the rotor in the vertical and horizontal direction. Damping coefficients are not considered.

The force becomes destabilizing force when k_r is negative or k_t is positive. Usually the effect of k_t is much more significant.

Figure 13 shows the effect of the preswirl to the stiffness of the seal. The horizontal axis is the preswirl ratio, which is the ratio of the circumferential velocity at the inlet to the rotor surface speed. The relation between the preswirl and k_r, k_t is almost linear. The effect of preswirl on k_t is large; k_t at preswirl ratio 1.0 is about 5 times larger than that at 0.5. Because the cross-coupled term k_t has a significant role on the stability of the rotor-seal-bearing system, the preswirl and the stability have a close relationship.

Fig. 11 Velocity Vector

Fig. 12 Pressure distribution

Fig. 13 k_r, k_t vs preswirl

3.3 Evaluation Formula

It is not practical to apply CFD for the calculation of the seal force of every seal in a steam turbine, because it takes a week ~ 10 days to calculate the seal force by CFD and a steam turbine has more than 30 seals.

Numerical calculations are carried out to investigate the dependency of the parameter such as the preswirl, the pressure difference, the inlet pressure, the clearance etc. Evaluation formula is formed from the results as experimental formula is formed from the test results. This formula is used to predict the dynamic coefficients of each seal in a steam turbine. They are incorporated in the rotor dynamics model of the total system. Then the stability is evaluated by complex eigenvalue analysis.

4. Bearing Dynamic Properties

4.1 Test of the bearing

The precise estimation of the bearing dynamic properties is also important to predict stability of the system accurately. To investigate the dynamic properties of the bearings, excitation tests were carried out on the large-scale test rig. Figures 14 and 15 show the photo and the cross sectional view of the test rig.

This test rig has the test bearing at its central part and both end of the test rotor is supported by the 406mm (16in.) diameter elliptical bearings. The test rotor can be rotated by the 2400kW steam turbine up to the maximum speed of 66.6rps.

The test bearing is fixed rigidly to the bearing casing and the bearing casing is supported by air bellows in horizontal and vertical direction. Air pressure in each bellows can be arbitrarily changed and a vertical load up to 980kN is applied to the test bearing upward by the vertical bellows. The bearing casing is provided with two hydraulic exciters at the position of ± 45 degree from vertical direction in the lower half. Entire test bearing can be excited by a sinusoidal wave with its exciting force being constant. The maximum value of exciting force is 90kN and its frequency range is 10 - 100Hz.

Spring coefficients of the air bellows were measured when there was no oil film. Radial and tangential spring coefficients of vertical bellows are about 2 kN/mm and 6 kN/mm respectively, and those of horizontal bellows are both under 1 kN/mm. On the other hand, spring coefficients of oil film are about 1000 kN/mm. Therefore stiffness of air bellows is under 1 % of stiffness of oil film and air bellows do not affect the measurement of dynamic coefficients of oil film.

Tests for 2-lobe and tilting pad bearings are carried out so far. Tilting pad bearings include 6-pad load-on-pad (6LOP) type and 4-pad load-between-pad (4LBP) type.

Fig.14 Bearing test rig

Fig.15 Cross section of bearing test rig

Fig.16 Comparison of dynamic properties between test and calculation

4.2 Comparison between test and calculation

Test results are compared with the calculation results. Figure 16 shows comparison of dynamic properties of 4LBP type tilting pad bearing. P_m in the figure is specific load of the bearing. The specification and cross section of the bearing are shown in the Table 2 and Fig. 17, respectively. The calculation of oil film characteristics is based on Reynolds equation. The effect of the turbulent flow is considered.

Moreover the flexibility of the pivot, which is the contact point of the backside of the pad and the housing, is also considered. In the large-scale steam turbine application the bearings are used with high specific load to reduce the bearing loss. In this case the flexibility of the pivot has a significant effect on the bearing characteristics. In Fig.16 "Calculation with Pivot" shows the calculation result considering the effect of the pivot flexibility and "Calculation without Pivot" corresponds to the result without the consideration. The deformation and stiffness of the pivot are calculated based on Herz contact theory. The difference between two calculation results becomes larger as the specific load becomes higher. And the consideration of the pivot flexibility enables the precise prediction of the bearing dynamic properties.

Table 2 Specification of 4LBP bearing

Diameter: D	580 mm
Length to diameter ratio: L/D	0.55
Clearance ratio: $2 * c/D$	0.0013
Preload factor	0
Type of lubricant	ISO - VG32
Pad angular extension	72 deg
Pivot location (Offset)	0.6

Fig.17 Cross section of test bearing

5. Stability

The stability of the steam turbine is evaluated following the procedure shown in Fig. 2. The stabilities of the typical steam turbines are shown in Fig.18 and confirmed to be in the stable region. They are operated silently as expected.

Fig. 18 Stability of steam turbines

6. Conclusion

It is important to know the rotordynamic force of the seals and bearings for the accurate prediction of stability of a steam turbine due to steam whirl. A general-purpose commercial CFD code is used to calculate the seal force of the labyrinth seal. The application of CFD to the seal force analysis is validated by comparing the calculation results with test results. Then the CFD is applied to the calculation of the seal force in an actual steam turbine. Evaluation formula is formed from the results of numerical calculation, which is used to predict the dynamic coefficient of each seal in a steam turbine.

The bearing is also important to predict stability of the system accurately. To investigate the dynamic properties of the bearings, excitation tests were carried out on the large-scale test rig. In the large-scale steam turbine application where the bearing specific load is high, the consideration of the flexibility of the pivot, which is the contact point of the backside of the pad and the housing, is also important.

The stabilities of steam turbines are evaluated by the complex eigenvalue analysis of the finite element model of the rotor-seal-bearing system. They are confirmed to be stable region.

The procedure to evaluate the rotordynamic stability utilizing CFD results is developed and applied to the design of steam turbines. It enables the optimization of the turbine structure considering the efficiency and stability.

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