DYNAMICS ANALYSIS OF VERY FLEXIBLE ROTOR

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ABSTRACT: In the project of 10MW high temperature gas-cooled reactor (HTR-10 GT), helium turbine will be use in the first circle of the high temperature gas-cooled reactor. The preliminary design of its power conversion unit (PCU) tries to use active magnetic bearing (AMB) to sustain the 1.5 ton gas turbine and generator to run under a high speed of 15,000rpm, over the second bending critical frequency [1]. A small dynamics similar prototype, which includes a very flexible rotor and a set of magnetic bearing device, is set up to test the feasibility of running through the bending critical frequency [2]. This paper will give an overview on the dynamic analysis of this very flexible rotor in the prototype device. Dynamics analysis is one of the most important aspects of rotor design. Proper modeling of rotor's dynamics behavior plays a crucial role in the control design of rotating machines and erroneous dynamics design selection can cause uncontrollability or severe machine failure. In this paper, dynamics analysis will perform the critical speed analysis and force response analysis (unbalance response), which will take into account the gyroscopic effect by use of finite element analysis method.

KEYWORDS: Rotor dynamics; Dynamics analysis; Flexible rotor

0. INTRODUCTION

Structural vibration limits the performance of turbomachinery in various applications, such as power generation equipment and rocket engines turbo-pumps. Most of conventional mechanical machine run under the critical speed to reduce or avoid the risk of unexpected vibration. But to some special reason, the generator and turbine rotors of HTR-10 GT PCU have to run over the second bending frequency [1]. Any kind of unbalance force may lead to instability of the whole system and inability to predict, these instabilities during the design stage can only lead to costly delays and shutdowns after system installation.

The dynamics problem of very flexible rotor must be well studied before designing the magnetic bearing of HTR-10 GT PCU. A small dynamics similar prototype device was set up to help to prove the theoretical capability of rotor running over second bending frequency under magnetic bearing levitation, and till now, most of the preliminary experiments and analysis are concern about this prototype device. The dynamics analysis described in this paper is focusing on the rotor of this prototype device as well.

The small testing shaft of the prototype is a vertical rotor, which has three disks on it, two of them, located on both side, are used for thrust bearing, and the other middle located one is used for dynamics balance. At both end of this shaft, there are radial magnetic bearings. The picture and the diagrammatic sketch of this small testing shaft are showed in figure 1. The length of this shaft is 613mm, the diameter of the disk is 100mm, and the diameter of the shaft is about 24mm.

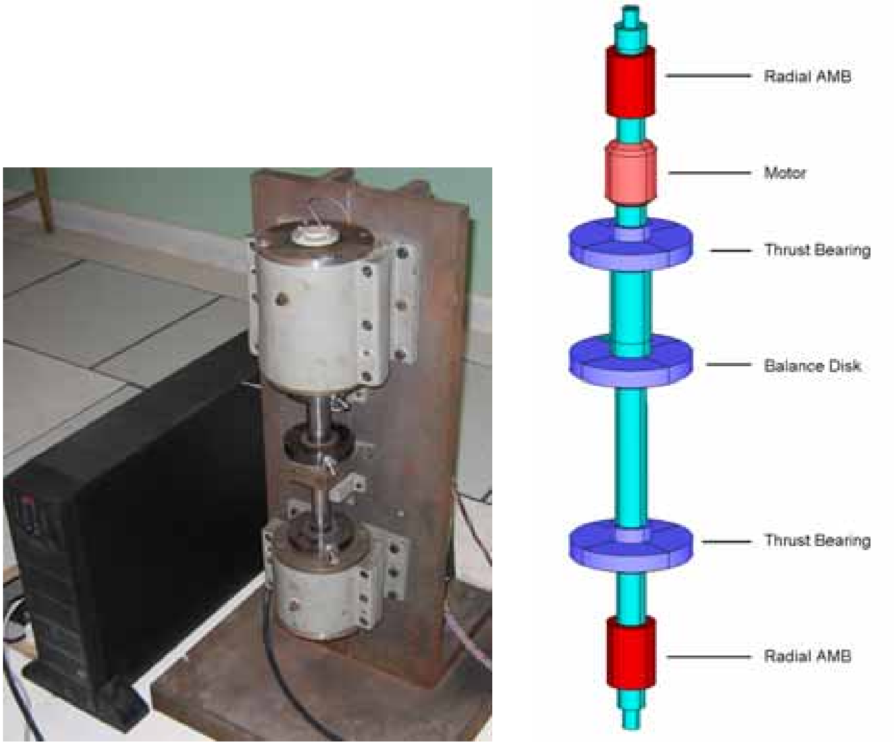

FIGURE 1. The small testing shaft

1. ROTOR DYNAMICS ANALYSIS

1.1 MODELING

There are two methods are both wildly used to solve the rotor dynamics problem, transfer matrix and finite element. Here, finite element method is used.

When modeling this shaft, one dimension variable cross-section beam element is used, in order to include the gyroscopic effect. The FEA model use 212 elements, 213 nodes totally to reshape this rotor.

Consider the three disks, the lamination stack of radial magnetic bearing and the motor are all assembled to the shaft with large magnitude of interference, this rotor is modeled as a single variable cross-section shaft.

These three disks, lamination and mandrel are made in different kind of steel, so of course, they have different material properties. But, fortunately, these materials have similar characters, so assume the whole shaft has the same material property when modeling. Ignore some detail structural of this shaft. Finally, common linear isotropy steel material property and linear FEA analysis method are used to solve the modal and harmonic analysis in this paper.

The magnetic bearings are simulated as four constant stiffness spring elements. One side of the spring element is connected to the middle point of the magnetic bearing. And the other side is connecting to a fixed boundary condition, shown in figure 2. The stiffness and damping of the magnetic bearing can be adjusted by change the spring element’s real constant property. Here in the analysis, the stiffness was set to be : $K=3\times10^4$N/m, which is obtained from experiment and is much smaller than traditional bearing, and the damping was set to be zero for simplifying the analysis.

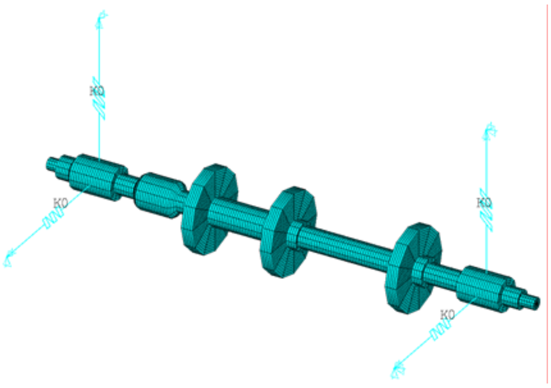

FIGURE 2. FEA modal of the rotor

TABLE 1 The rotor system parameters are showed in the following table.

Density:	$\rho = 7800 \mathrm{kg/m^3}$	Radius of disc:	$R_d = 50\mathrm{mm}$
Young's modulus:	$E = 2.0 \times 10^{11} \mathrm{N/m}$	Radius of axis:	$R_s = 12\mathrm{mm}$
Poisson's coefficient:	$\mu = 0.3$	Radius of lamination:	$R_l = 20\mathrm{mm}$
Mass of axis:	$M = 11.12\mathrm{kg}$	Length of axis:	$L = 613\mathrm{mm}$

1.2 MODAL ANALYSIS

First of all, modal analysis of this rotor is processed. From the result of modal analysis, modal frequency and modal shape can be well know, which are useful for the sensors distribution design of magnetic bearing. The results are shown in figure 3. In this result, the gyroscopic effect has not been included yet.

Because of the low bearing stiffness, there are two low translatory critical frequencies. These two critical frequencies reflect only the relationship of the rotor's mass and the stiffness. The other two bending frequency are much more concerned about.

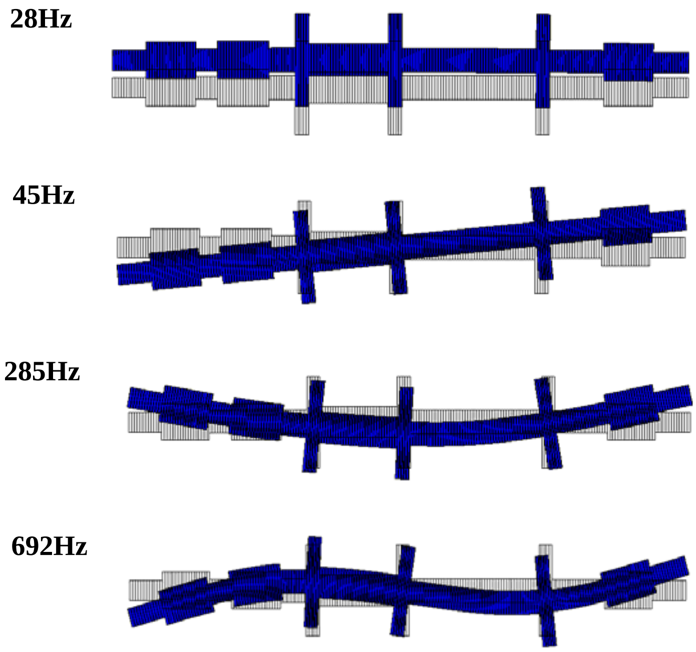

FIGURE 3. Modal frequency and modal shape

Use the method of rotor identification, the basic modal frequency and modal shape can be test through experiment [2]. In figure 4, the modal shape of modal analysis and experimental identification are compared.

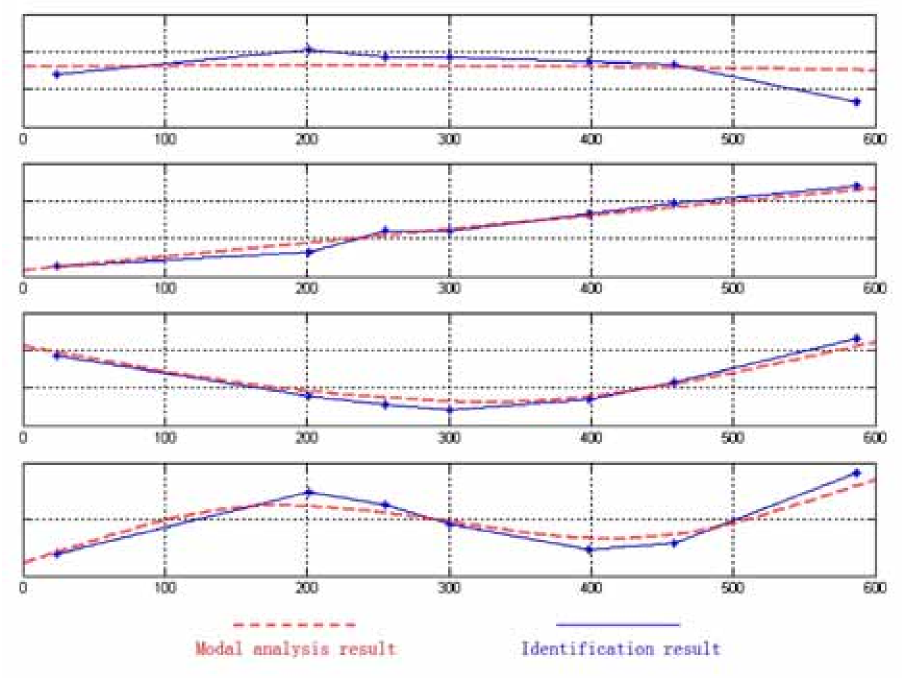

FIGURE 4. Modal shape compare of modal analysis and identification

In this figure, symbol "*" show the identification sensors' distribution. And the amplitude of the modal shape has already been normalized through energy.

Since the magnetic bearing is a kind of small stiffness bearing, the first two modal of the rotor are the so-called rigid-body-modes, where one of them is translational motions and the other is angular motion [3].

1.3 FORCE RESPONSE ANALYSIS

Force response analysis can simulate the force response of the rotor, which can help to identify the main modal frequency. To a complex rotor, the modal analysis can give a serial of eigenfrequency, but only some of them are just the right modal frequency value we need. The harmonic analysis can extract the eigenvector of the modal analysis result, integrate them together, and show the amplitude frequency response characteristics of some special place of the object rotor.

Assume the unbalance force is mostly supply by the middle disk, and the force is 20N. Figure 5 shows the compare of the top sensor respond of amplitude result and the analysis simulation result.

In this figure, identification result shows the amplitude increase at the front four eigenfrequency. And the harmonic analysis result shows the similar tendency here.

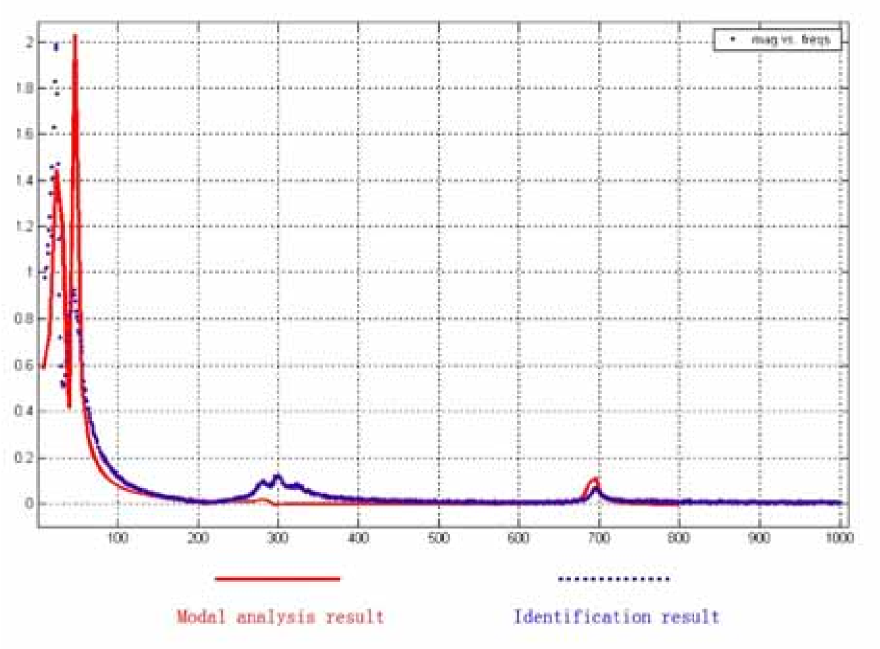

FIGURE 5. Harmonic analysis of the shaft

1.4 CRITICAL SPEED ANALYSIS

The basic differences between the dynamic behavior of a non-rotating body and a rotating one are caused by gyroscopic properties. Typical natural vibrations of a spinning rotor manifest themselves as a "whirling" of the rotor axis, which whirls in the same sense to the rotor spin in a forward whirl or opposite to it in a backward whirl. When the gyroscopic effect and the rotor whirling are considered into the modal and harmonic analysis, much more accurate resonant critical frequency result can be obtain, that is very necessary for the controller design.

Figure 6 shows the eigenfrequency change as the rotor speed grows.

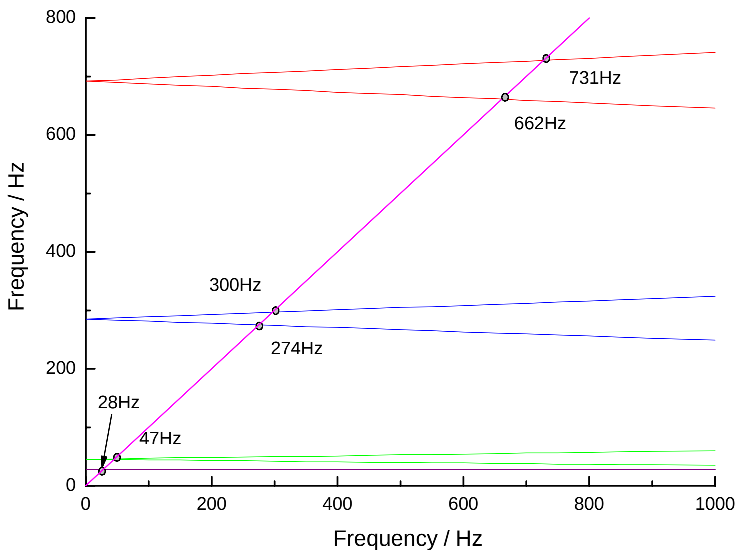

FIGURE 6. Critical speed analysis with gyroscopic effect

When taking into account gyroscopic effects, the critical speed result has not been well proved yet. But this result is similar to the result provided by transfer matrix method [4]. And this result shows that this testing rotor's critical speed has no much change when the running speed grows up.

2. CONCLUSION

The FEA method in this paper has well perform the modal frequency, modal shape, forced response and critical speed analysis of a testing rotor of HTR-10 GT PCU. Some of the results, such as modal shape and vibration response have been examined to be correct.

This procedure will be implemented and produced reasonable results for the HTR-10 GT rotors.

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