The Twelfth International Symposium on Magnetic Bearings (ISMB 12)
Wuhan, China, August 22-25, 2010

Principle Test of Active Magnetic Bearings for the Helium Turbomachine of HTR-10GT

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Abstract : The 10MW high temperature gas-cooled test reactor coupled with gas-turbine circle (HTR-10GT) is considered as one of the most important development fields of the future HTR technology due to its simple system, high efficiency and potential economical competitiveness. In the power convention unit (PCU) of the HTR-10GT, the active magnetic bearings (AMB) are adopted to support the vertical high-speed helium turbomachine shaft, its performance affect the layout of gas turbine system related with safety and economics of the nuclear power plant. In order to make a principle test for the design of AMBs of HTR-10GT, a small-scale analogue test rig has been designed and built to simulate the helium turbomachine rotor in compliance with the dynamic similarity principles. Through the test, the advanced control algorithms are validated, design and operation experiences for the next full scale experiment in the near future are accumulated.

Keywords: Active Magnetic Bearings, Helium Turbomachine, HTR-10GT, Test Fig

Introduction

The high temperature gas-cooled reactor (HTGR) is a representative of the next generation of nuclear system [1]. For HTGR technology, the power conversion system currently trends towards utilizing gas turbine generator system worldwide because of its high efficiency and simple structure. For example, the projects of PBMR, GT-MHR, GTHTR300, etc. are designed to utilize gas turbine generator coupled with HTGR [2-4].

In the bearing system of the gas turbine generator, AMB, which are often regarded as substitutes for oil-lubricated bearings, have a number of obvious advantages: elimination of the lubricating oil unit, high reliability of the system, and ease of adjustment and operation of the machines on which they are installed, etc [5]. For all the projects of PBMR, GT-MHR, GTHTR300, etc., AMB was the first choice to support the shaft of gas turbine.

The 10MW high temperature gas-cooled test reactor coupled with gas-turbine circle (HTR-10GT) has been carried out by the Institute of Nuclear and New Energy Technology (INET) of Tsinghua University in China since year 2002 [6]. In the power convention unit (PCU) of the HTR-10GT, the contact-free and no-lubricating active magnetic bearings (AMB) are also exclusively adopted to support the vertical high-speed turbine machine shaft [7]. However, at present, no experience on AMB equipment running in the HTR system of helium environment, especially for the large flexible PCU rotors, has been performed actually. Therefore, the design and experiment of high performance AMBs as well as high reliability and safety are very important for the HTR engineering applications.

Based on the previous studies and small tests of the AMBs [8], the final engineering design of the AMBs for the HTR-10GT turbine compressor rotor has been finished by the INET recently. In this paper, a small AMB flexible test system in compliance with the dynamic similarity principle of the helium turbine compressor rotor of the HTR-10GT was introduced, which is set up to validate the advanced control algorithms and accumulate design and operation experiences for the next full scale experiment in the near future.

System Description

The overview structure of the PCU system of the HTR-10GT is shown in Fig.1. At the latest design process, the turbine and compressors mounted on a same rotor are connected to the generator rotor through an elastic coupling, which are all vertically arranged inside the PCU pressure vessel.

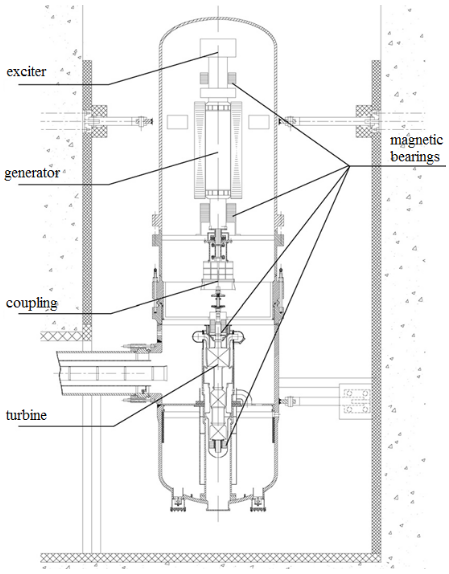


Fig. 1: Overall view of the HTR-10GT PCU system.

The turbine and compressor rotor, with a length of 3.5 m and a weight of 540 kg, is designed to be running at a rated rotation speed of 15,000 r/min.

After successfully building and testing an early small principle AMB experimental machine, the control arithmetic about how to passing the bending critical speed has been preliminarily validated, as well as accumulated experiences for engineering design, theoretical analysis and operations of the AMBs.

Due to the development of new technologies of the controller, sensor, amplifier and structure design for the HTR-10GT AMBs, a new small-scale analogue test rig has been designed and built to simulate the helium turbine compressor rotor in compliance with the dynamic similarity principles, which is based and updated on the early one.

On this small AMB test rig, we are interested in the control design about passing the bending critical speeds as well as how to attenuate vibration and increase system stability and robustness, which is of extreme importance to the flexible rotor control. Besides, the locations of the bearings and sensors are also studied in order to find an optimal collocation for the AMBs.

The structure of the small-scale AMB test rig is shown in Fig. 2. There are four radial magnetic bearing actuators designed in this system which are employed to support the rotor. Magnetic bearing A and B are the main supported bearings, while magnetic bearing C and D are the auxiliary bearings, which can study the control effect of different AMB combinations. In this paper, only magnetic bearing A and B are considered.

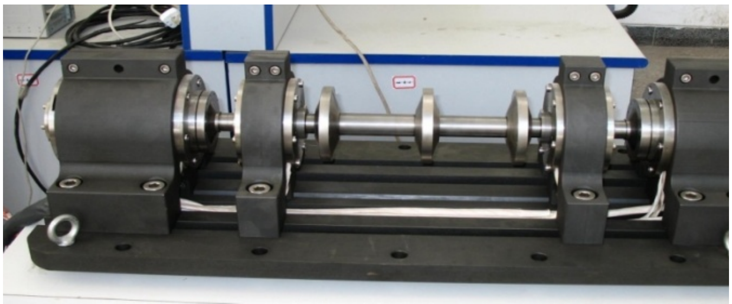


Fig. 2: Structure of the small AMB test rig.

The first and second bending critical frequencies are designed as 112 Hz and 237 Hz respectively, which are higher than those of the actual turbine rotor. It is deliberate to increase some difficult considering the difference between small rotor in experiment and large rotor in the PCU. The main parameters of the test rig are listed in Table 1.

Table 1: Main parameters of the small test rig

Parameter	Value
Parameters of the rotor	
Rotor length, [mm]	1058
Rotor weight, [kg]	15.8
Rated speed, [r/min]	0~24000
Radial electromagnetic bearing	
Maximum radial capacity of each bearing, [N]	300
Interior diameter of stator magnetic, [mm]	40
Maximum current, [A]	6
Number of coil turns for one pole	160
Radial gap between bearing and rotor, [mm]	0.3
Radial gap between catcher bearing and rotor,[mm]	0.15
Axial electromagnetic bearing	
Maximum radial capacity of each bearing, [N]	800
Outer diameter of stator magnetic, [mm]	100
Maximum current, [A]	6
Number of oil turns	250
Radial gap between bearing and rotor, [mm]	1.0
Radial gap between catcher bearing and rotor,[mm]	0.5

Modal Analysis and Control design

A finite element method is used to analyze the mode of the rotor. Fig. 3 gives the results of the modal analysis, which are useful for the bearings and sensors distribution design of magnetic bearing.

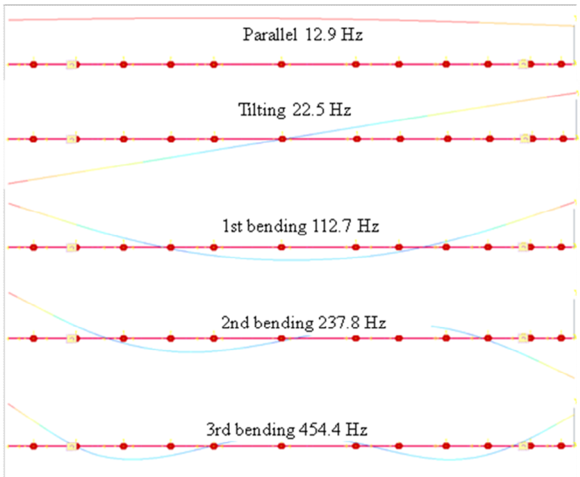


Fig. 3: Modal frequencies and modal shapes.

The magnetic bearings are simulated as four constant stiffness spring elements. The stiffness and damping of the magnetic bearing can be adjusted by change the spring element's real constant property. In the analysis, the stiffness was set to be 5×10^5 N/m, while the damping was set to be zero for simplifying the analysis. Besides, the gyroscopic effect has not been included yet.

If 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, which is very necessary for the controller design. Fig. 4 illustrates the eigen frequency change as the rotor speed grows.

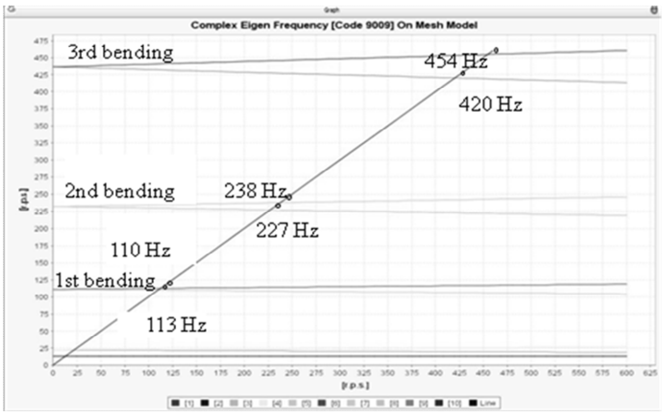


Fig. 4: Critical frequency with gyroscopic effect.

It can be seen that, when running speed grows up, a forward whirl and a backward whirl will appear from the original one eigen frequency if taking into account the gyroscopic effects. As for the first two bending frequencies, there are only small departures to their standstill values, so the gyroscopic effects can be omitted for simplifying the design of the control algorithmic.

Firstly, a PID controller has been realized to achieve the designed stiffness and damping of the AMB system according to a rigid rotor model. Then, the system identification was carried out to get the fine parameters, such as the bending frequencies, force-current and force-displacement factors, etc. Based on the system identification, a kind of phase compensation method was performed and integrated with the PID algorithmic to control the flexible modal vibrations when passing the critical speeds.

Besides, in order to attenuate the synchronous disturbances due to unbalance force of the rotor, an adaptive unbalance vibration control (AUVC) method based on the adaptive feedforward algorithm [9] was designed. Fig. 5 shows the proposed AUVC structure for eliminating synchronous disturbances in magnetic bearing systems [10].

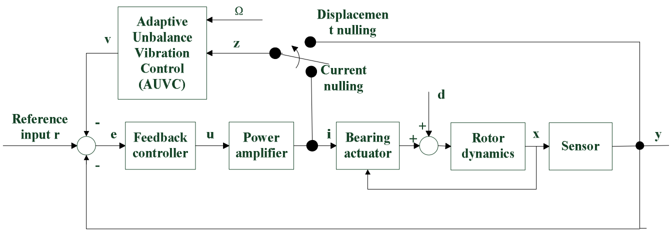


Fig.5 Block diagram of AUVC structure

By constructing the required frequency compensation signal with proper amplitude and phase automatically by means of the operating speed signal through different feedforward approaches, two types of control methods was realized: the “displacement nulling” control can effectively cancel the rotor vibration amplitude within the system bandwidth, while the “current nulling” method can let the rotor rotate around its inertia axis and eliminate the disturbances of the currents in the electromagnet windings to significantly attenuate the mechanical vibration.

Experiments

On the small test rig, the passing bending critical speed experiment was carried out elaborately. The PID controller along with the phase compensators around the first two bending critical frequencies of 113 Hz and 238 Hz has perfect control performance. The rotor exceeded the bending critical speeds safely and smoothly, running stably up to at approx. 450 Hz, shown in Fig.6.

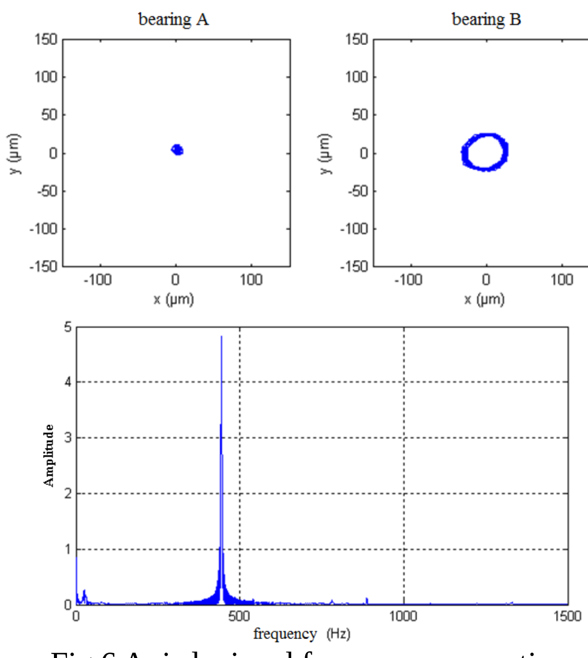
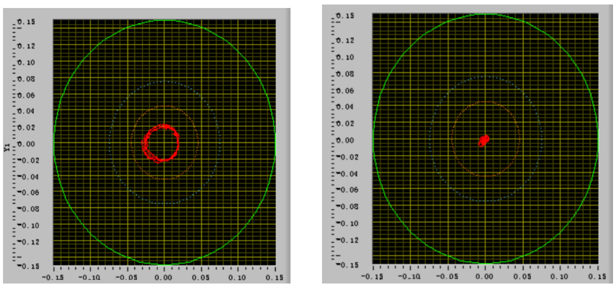


Fig.6 Axis loci and frequency properties

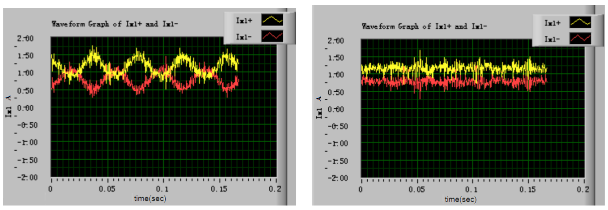
The experiments of investigating the effects of the AUVC method have also been performed on this small test rig. Fig.7 gives the axis locus of the bearing A for the case with or without “displacement nulling” method at a speed of 3000 r/min, while Fig.8 shows the current of the coil winding for the case with or without “current nulling” method at the same rotation speed.

It can be obviously seen that the AUVC method can effectively attenuate the rotor vibration amplitude and the coil current to achieve performance improvements compared with the conventional feedback control method.



no “displacement nulling” “displacement nulling”

Fig.7 Axis locus of bearing A



no “current nulling” “current nulling”

Fig.8 Coil winding current of bearing A

Full Scale Test Rig

After the small experiment was completed, the design and building of the full scale test rig for the turbine compressor rotor and AMBs is just ongoing. The construct work of the full test rig will be finished at the end of the year 2010. Fig.9 illustrates the preliminary design of the full scale test rig.

The main design principles for the analogue rotor are listed as the follows:

(1) Its first two bending critical speeds are closed to the actual ones with a maximum error less than 15%.

(2)Its first two bending modals are consistent with those of the turbine rotor.

(3)Its length (3568 mm) and weight (645 kg) are the same as the turbine rotor.

(4) Its stiffness distribution is coincident with the turbine rotor.

(5)The location and structure of the AMBs are identical to the actual ones..

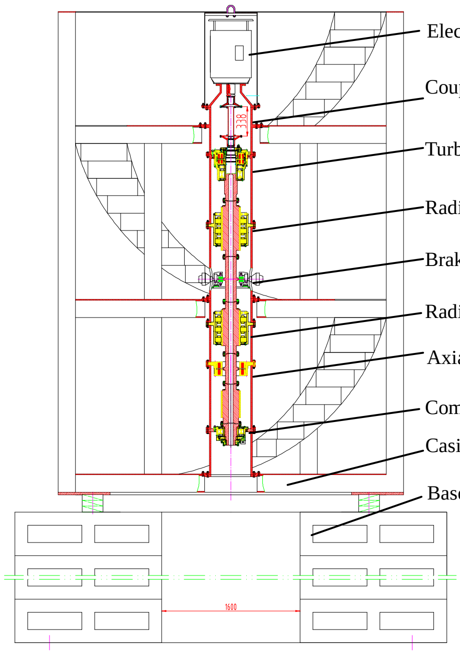


Fig.9 Full scale test rig

The whole setup will be installed in a concrete hole with a diameter of 5 m and a height of 10 m. A high speed electric motor with frequency converter up to 300 Hz is settled at the top, connecting the turbine rotor by a flexible coupling. Like the actual turbine compressor rotor, the analogue rotor also consists of different segments, which are connected by bolts. A braking device is located at the middle of the rotor, which can act emergency stop within a setting value of 30 s. Besides, two radial and one axial force simulators are designed to provide dynamic loadings in the experiments.

Conclusions

Active magnetic bearing (AMB) is an important component to support helium turbomachine in PCU of HTR-10GT, which is a pebble-bed high temperature gas-cooled test reactor together with direct gas turbine designed and built by the INET in China. The experiments of a small AMB flexible rotor test rig are discussed detailed in this paper. The design of the control methods of passing the bending critical speeds, as well as attenuating synchronous disturbances, has been validated on the small test rig. The further design and experiments for a full scale turbine test rig are being carried out continuously till the whole experimental system installed at the end of year 2010. These tests will provide good experience and design references for the application of magnetic bearings of the HTR-10GT in the near future.

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