

## PRELIMINARY SCHEME DESIGN AND ANALYSIS OF CATCHER BEARING FOR HTR-10GT

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**ABSTRACT:** The electromagnetic bearing was chosen to support the rotor in the helium turbine and generator system of 10MW high temperature gas-cooled reactor (HTR-10GT), and the catcher bearing was applied in the HTR-10GT as the backup protector. When the electromagnetic bearing doesn't work suddenly for the power broken, the catcher bearing is used to support the falling rotor with high rotating speed. The rotor system of the turbine and generator will be protected by the catcher bearing. The design of catcher bearing is the ultimate safeguard for the HTR-10GT. This rotor is vertically mounted to hold the turbine machine, compressors and the power generator together. The rotor's length is about 7 m, its weight is about 1500 kg and the rotating speed is 15000 r/min. The design of the catcher bearing is one of the challenging problems in the whole system. It is very important for HTR-10GT to make success. The research status of the catcher bearing was summarized in the paper. A sort of preliminary scheme of catcher bearing structure was proposed. The falling course was analyzed. The stress of the generator rotor and catcher bearing was computed by the finite element method and MSC.Dytran. The scheme of catcher bearing and the analytical results offer the important theoretical base for the protector design of the HTR-10GT.

**KEY WORDS:** HTR-10GT, Electromagnetic Bearing, Catcher Bearing, Finite Element Method, Rotor Dynamics

### 0. INTRODUCTION

The 10MW high-temperature gas-cooled test reactor (HTR-10) is the first module high-temperature gas-cooled test reactor in the world. It was constructed by INET at Tsinghua University of China. The steam cycle has been applied in the power conversion unit (PCU) of the HTR-10. This technology is mature and reliable. In order to sufficiently use the quantity of heat produced by the high-temperature reactor, a direct gas cycle to replace the current steam cycle was studied in the HTR-10 phase II (HTR-10GT). Helium was chosen as the medium to achieve a direct helium cycle<sup>[1]</sup>.

If oil bearings are used as the supporting assembly in the PCU for the direct helium cycle, oil leaks from the bearings would be likely to occur. Ordinary bearings are not adaptable for shortcomings such as wear, maintenance or frequent replacement. Electromagnetic bearings are replacing ordinary bearings as the perfect sustaining assembly for the PCU because they have several advantages: they are free of contact, do not require lubrication, are not subject to the contamination of wear, have endurance, and control performance very well<sup>[2]</sup>.

The need to avoid contamination of the primary system, along with other perceived advantages, has

led to the selection of electromagnetic bearings in most ongoing commercial-scale gas cooled reactor designs. However, one implication of magnetic bearings is the requirement to provide backup support to mitigate the effects of failures or overload conditions. The catcher bearing is used to support the rotor when the magnetic bearing fails to work. Especially when the rotor is rotating with high speed, the magnetic bearing suddenly doesn't support the rotor for the power off. So it is very important for the catcher bearing to protect the rotor system.

The weight of the HTR-10GT rotor is about 1500 kg and the rotating speed is 15000 r/min. Several critical speeds can be passed. It is a large, vertical, high-speed and flexible rotor. This is a challenging project in theory and practice. The research status of the catcher bearing was summarized in the paper. A sort of preliminary scheme of catcher bearing structure was proposed and analyzed. The scheme of catcher bearing and the analytical results offer the important theoretical base for the protector design of the HTR-10GT.

## 1. SURVEY OF CATCHER BEARING

The magnetic bearing system needs a back-up rotor support system (catcher bearing) in case the electromagnetic bearing fails or the machine does not operate. The catcher bearing located at the outer-bound of the magnetic bearing is also called an "auxiliary bearing" or a back-up bearing. Being assembled on the stator, the catcher bearing is not active during the normal operation of the magnetic bearing. The clearance of the catcher bearing is smaller than that of the magnetic bearing to protect it. Typically, fifty percent of the magnetic bearing clearance is used for the catcher bearing clearance<sup>[3, 4]</sup>.

The diversity in both requirements and applications has led to the development of several auxiliary bearing concepts, each having its own advantages and disadvantages. In general, the catcher bearing concepts in use today fall into six general categories. These are plain bearings, rolling element bearings, planetary bearings, zero-clearance auxiliary bearings, ceramic bearing and gas bearing. Each of the catcher bearing designs has its own set of advantages and disadvantages. These are summarized in Table 1.

TABLE 1. Advantages and Disadvantages of Auxiliary Bearing Concepts

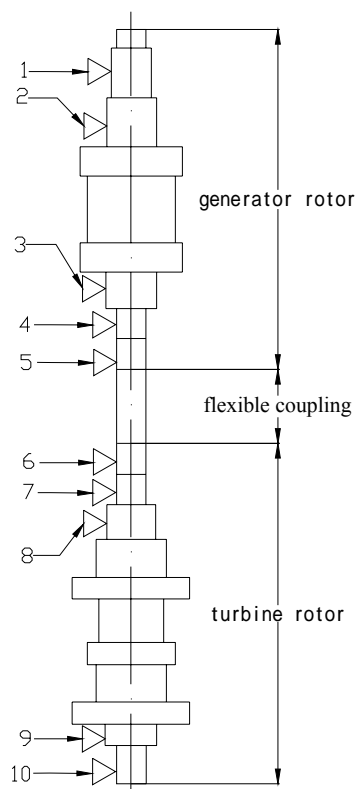
Technology	Advantages	Disadvantages
Plain Catcher Bearing	<ol style="list-style-type: none"> <li>1) Low-cost</li> <li>2) Passive, no moving parts in bearing</li> <li>3) Reduced potential for deterioration in standby mode</li> <li>4) Condition, wear may be assessed by measuring clearance with EMB's</li> </ol>	<ol style="list-style-type: none"> <li>1) Higher friction coefficients, heat generation during rundown</li> </ol>
Rolling Element	<ol style="list-style-type: none"> <li>1) Low-cost</li> </ol>	<ol style="list-style-type: none"> <li>1) Potential for bearing/cage</li> </ol>

Catcher Bearing	<ol style="list-style-type: none"> <li>2) Low friction coefficients, heat generation during rundown</li> <li>3) Potentially minimum volume with combined radial/thrust bearing</li> </ol>	<p>damage during acceleration</p> <ol style="list-style-type: none"> <li>2) Potential for deterioration in standby mode; contamination must be avoided</li> <li>3) Windage induced rotation must be prevented in standby mode</li> </ol>
Planetary Catcher Bearing	<ol style="list-style-type: none"> <li>1) Reduced DN for given rotor diameter and speed</li> <li>2) Low friction coefficients, heat generation during rundown</li> </ol>	<ol style="list-style-type: none"> <li>1) Greater complexity and cost</li> <li>2) Contamination must be avoided</li> <li>3) Windage induced rotation must be prevented in standby mode</li> <li>4) Potential for acceleration damage (reduced relative to rolling element bearings)</li> </ol>
Zero Clearance Catcher Bearing	<ol style="list-style-type: none"> <li>1) Eliminates rotor-bearing gap during rundown</li> <li>2) Extended run time capability</li> <li>3) Reduced DN for given rotor diameter and speed</li> <li>4) Low friction coefficients, heat generation during rundown</li> </ol>	<ol style="list-style-type: none"> <li>1) Greatest complexity and cost</li> <li>2) Actuation failures should be considered</li> <li>3) Contamination must be avoided</li> <li>4) Windage induced rotation must be prevented in standby mode</li> <li>5) Potential for acceleration damage (reduced relative to rolling element bearings)</li> </ol>
Ceramic Catcher Bearing	<ol style="list-style-type: none"> <li>1) Endures high temperature and wear, not to be eroded, high hardness, low density and the coefficient heat-expand</li> <li>2) Well performance of self-lubricated</li> </ol>	<ol style="list-style-type: none"> <li>1) Lower life than steel bearing's in low speed and heavy load</li> <li>2) Complicated technics, much difficulties, and high cost</li> <li>3) Unstable performance and bad tenacity for porosity and not uniformity of ceramic</li> <li>4) Theory of ceramic bearing is imperfect</li> </ol>
Gas Catcher Bearing	<ol style="list-style-type: none"> <li>1) Gas as lubricant, low expense</li> <li>2) Low friction very much and noise and libration</li> </ol>	<ol style="list-style-type: none"> <li>1) Bears low weight</li> <li>2) Speed is limited by unstable whirling motion</li> </ol>

<p>3) Works at higher and lower temperature, high rotary precision, long life</p>	<p>3) Bad work condition                  4) Frictional force possible may be large for impurity in gas, bad reliability                  5) High processing precision and cost</p>
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The use of rolling element bearing for catcher bearing is widespread in rotating machinery incorporates magnetic bearing. The function of the catcher bearing is to prevent rotor/stator contact, for which the inner race can experience a high impact force and rapid angular acceleration. Rapid deterioration of the catcher bearing can result from rotor impacts and high-speed touchdowns. It is therefore important to ascertain the influence of catcher bearing design parameters on the number of touchdowns that can be tolerated before replacement is required. A prerequisite is to understand the dynamic behavior of the system during a touchdown event, and this is also a necessary step before attempting to predict any thermal transients within the catcher bearing.

**2. PRELIMINARY SCHEME DESIGN**



*FIG. 1 THE STRUCTURE OF HTR-10GT ROTOR*

Catcher bearings are a part of electromagnetic bearing and intended to support the HTR-10GT rotor in the following cases: 1) Scheduled HTR-10GT shutdown when electromagnetic bearing are de-energized; 2) Failure of electromagnetic bearing during HTR-10GT operation to ensure rotor rundown till shutdown; 3) Dynamic loads exceeding electromagnetic bearing load-carrying capacity.

The preliminary scheme design has been finished by INET and OKBM, Russia. Figure 1 gives the preliminary structure of HTR-10GT rotor. The generator is fixed on the top of the rotor, and the turbine machine is fixed on the bottom of the rotor. The position of the radial magnetic bearing lies in areas 2, 3, 8 and 9 shown in Figure 1. The position of the axial magnetic bearing lies in areas 4 and 7 shown in Figure 4. The position of the radial catcher bearing lies in areas 1 and 10 shown in Figure 1. The position of the radial-axial magnetic bearing lies in areas 5 and 6 shown in Figure 1.

The basic parameter is as follow:

1) Radial gaps in the central rotor position

- a) between magnetic circuits of stator and rotor of radial magnetic bearing: 0.7mm;
- b) between radial catcher bearing and rotor: 0.15mm.

2) Axial gaps in the central position of the thrust disk of axial EMB

- a) between thrust disk and stator magnetic circuits of axial magnetic bearing: 1.0 mm;
- b) between axial catcher bearing and rotor: 0.3 mm.

3) Rotation speed of the TM rotor

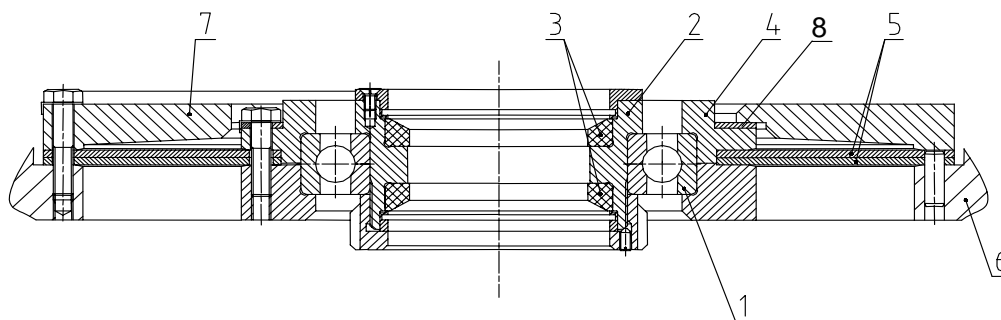
- a) nominal:  $250 \text{ s}^{-1}$  (15000 r/min);
- b) maximal:  $300 \text{ s}^{-1}$  (18000 r/min).

## 2.1 Design of radial-axial catcher bearing

Main working element of radial-axial catcher bearing (Figure 2) is radial-axial single-row four-point ball bearing pos. 1 with detachable internal yoke and massive bronze separator. To eliminate primary helium contamination the internal contacting surfaces of the ball bearing (balls and their raceways in yokes) are covered by dry lubricant.

Sleeve pos. 2 with graphite rings pos. 3 which form the plain bearing are pressure into internal yoke of the ball bearing. When catcher bearing starts operating the graphite rings exclude self-welding of bearing contacting surfaces and the rotor due to short-term slipping of the rotor relative to graphite rings of the plain bearing in immovable internal yoke of the ball bearing.

External yoke of ball bearing pos. 1 is installed in casing pos. 4. The casing suspension is made of flat circular springs pos. 5 fixed in HTR-10GT stator pos. 6. Package of springs pos. 5 has axial mounting downward tension which is carried out by pressure flange pos. 7 and adjustment gasket pos.8.



1 – ball bearing; 2 – sleeve; 3 – graphite ring; 4 – casing; 5 – spring;  
6 – stator; 7 – pressure flange; 8 – adjustment gasket

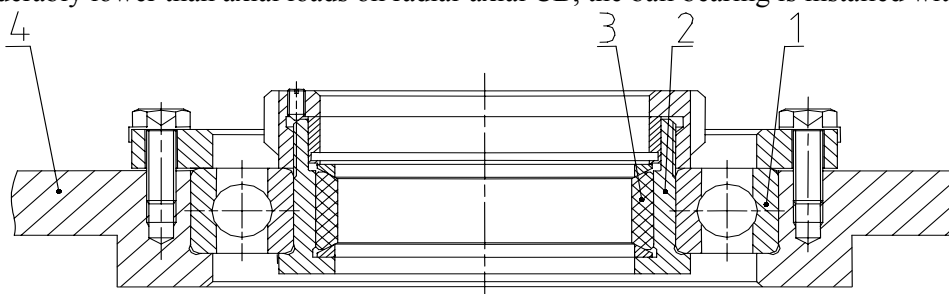
*FIG. 2 RADIAL-AXIAL CATCHER BEARING*

If axial magnetic bearing fails the package of springs prevents axial displacement of the falling rotor and returns the rotor upwards by the value of spring contraction travel. The spring contraction travel does not exceed allowable change of the gap in axial EMB between the thrust disk and electromagnet. Elastic reaction of the springs reduces dynamic loads on the ball bearing.

## 2.2 Design of radial catcher bearing

General view of a radial catcher bearing is shown in Figure 3. Main elements and structural decisions of the radial magnetic bearing correspond to radial-axial catcher bearing. The radial catcher bearing consists of ball bearing pos. 1, sleeve pos. 2 with graphite ring pos. 3 which form a plain bearing are pressed in the internal yoke of ball bearing pos. 1. The ball bearing is similar to the one used in radial-axial catcher bearing. Diametrical gap between the rotor and internal surface of radial catcher bearing amounts to 0.3 mm. In case of radial magnetic bearing failure the HTR-10GT rotor will be displaced in radial direction by the value of this gap. This ensures free rotation of turbine and compressor blade systems installed with radial gap of 0.4 mm.

External yoke of ball bearing pos. 1 is installed in casing pos. 4. Since radial loads on radial CB are considerably lower than axial loads on radial-axial CB, the ball bearing is installed without springs.



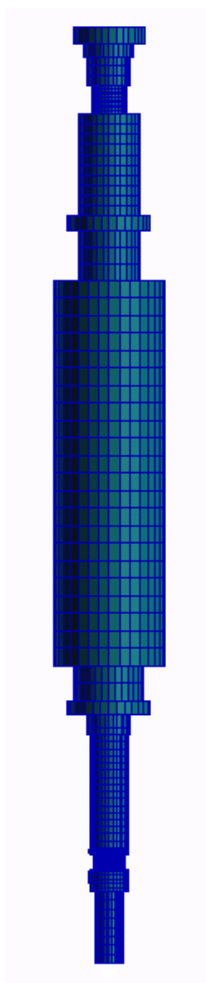
1 – ball bearing; 2 – sleeve; 3 – graphite ring; 4 – casing

*FIG. 3 RADIAL CATCHER BEARING*

## 3. ANALYSIS OF FALLING COURSE

The touchdown process is usually divided into four distinct regimes of motion: free-fall, impact, sliding-whirling, and rolling<sup>[5]</sup>.

It is difficult to analyze the falling course of the rotor. The preliminary analysis of generator rotor was done in the special condition by the finite element method and MSC.Dytran software.



*FIG. 4 FINITE  
ELEMENT MODEL*

The finite element model of the generator rotor is showed in figure 4. The rotor falls into the catcher bearing at the speed of 18000r/min when the magnetic bearing fails to support the rotor.

When the rotor falls without excursion, the coefficient of friction is supposed to 0.1. The stress of the generator rotor is 1160MPa. The stress of the radial-axial catcher bearing is 945MPa. The stress of the radial catcher bearing is 0.

When the rotor falls with the excursion, the coefficient of friction is supposed to 0.05. The angle of the excursion is  $0.03^{\circ}$ . The stress of the generator rotor is 1800MPa. The stress of the radial-axial catcher bearing is 2000MPa. The stress of the radial catcher bearing is 1200MPa. If the coefficient of friction is 0.1, the maximal stress of the rotor or catcher bearing exceeds 2000MPa. So the excursion will bring serious result. The preliminary result will offer the important reference for the design of catcher bearing.

#### 4. CONCLUSION

The research status of the catcher bearing was summarized in the paper. The preliminary scheme of catcher bearing structure was proposed. The falling course was analyzed, and the stress of the generator rotor and catcher bearing was computed. The scheme of catcher bearing and the analytical results offer the important theoretical base for the catcher bearing design of HTR-10GT.

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#### AUTHOR INTRODUCTION

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