AUXILIARY BEARING DESIGN CONSIDERATIONS FOR GAS COOLED REACTORS

S.R. PENFIELD, Jr.
Technology Insights,
Carthage, Tennessee

XA0103038

E. RODWELL
EPRI,
Palo Alto, California

United States of America

Abstract:

The need to avoid contamination of the primary system, along with other perceived advantages, has led to the selection of electromagnetic bearings (EMBs) in most ongoing commercial-scale gas cooled reactor (GCR) designs. However, one implication of magnetic bearings is the requirement to provide backup support to mitigate the effects of failures or overload conditions. The demands on these auxiliary or "catcher" bearings have been substantially escalated by the recent development of direct Brayton cycle GCR concepts. Conversely, there has been only limited directed research in the area of auxiliary bearings, particularly for vertically oriented turbomachines. This paper explores the current state-of-the-art for auxiliary bearings and the implications for current GCR designs.

1. INTRODUCTION

With continued progress in power electronics and controls technologies, magnetic bearings are finding increased acceptance in specialized applications where their unique characteristics offer particular advantages. One such application is high-temperature gas-cooled reactors (GCRs), which employ high-purity helium as the working fluid. In GCRs, magnetic bearings provide a practical means for eliminating the necessity of rotating seals and for avoiding the potential contamination associated with conventional bearing lubricants. Well-designed magnetic bearing installations are also demonstrating high reliability in large conventional turbomachinery applications, such as pipeline compressors and turbo-expanders.

Given the above, there are compelling incentives for magnetic bearings in GCRs. However, a consequence of their selection is the need to provide backup support to mitigate the consequences of primary bearing failures. Further, while auxiliary bearings are a common feature in all machines with magnetic bearings, there is little in the present literature addressing their design and implementation. In addition, GCR applications pose particular challenges for the following reasons:

1. The tribological characteristics of materials in dry helium tend to be more difficult due to the potential for self-welding
2. The more recent evolution of direct Brayton cycle GCR's has increased the size of the turbomachinery and, thus, heightened the technical challenges associated with both magnetic and auxiliary bearings.
3. GCR designs with vertical turbomachines pose even greater demands on the axial auxiliary bearing.

With increasing worldwide interest in GCR's, EPRI has launched a supporting initiative with the initial objective of exploring areas where focused research could best facilitate

GCR development. The technology of catcher bearings is one such area, and this paper summarizes preliminary results of the EPRI sponsored evaluation. While current GCR turbomachine designs include both horizontal and vertical variants, the focus of the preliminary evaluation has been on vertical machines, due to their more demanding technical requirements and the relative timing of the associated project initiatives.

The remainder of the paper begins with a review of key parameters that will drive the design of auxiliary bearings for the two commercial-scale GCR projects. Next, the principal design considerations associated with auxiliary bearings are outlined and various auxiliary bearing types are described, along with their advantages and disadvantages. With this background, the requirements for GCR auxiliary bearings are compared with the existing experience base. Finally, some preliminary observations are offered regarding an integrated approach to bearing design and the need for additional research and development.

2. TURBOMACHINE CHARACTERISTICS IN CURRENT GCR PROJECTS

At present, the two active GCR projects of commercial scale are the Pebble-Bed Modular Reactor (PBMR) Project, based in South Africa and the US/Russian Federation Gas-Turbine Modular Helium Reactor (GT MHR) Project, which is largely being developed in Russia. The designs being advanced by both of these projects apply the direct gas-turbine (Brayton) cycle for power conversion, with the primary coolant, helium, serving as the working fluid. Additional details of the PBMR and GT-MHR designs have been reported elsewhere in Refs. [1] and [2].

Key parameters of the PBMR and GT-MHR turbomachines are provided in Table I. Note that, where the GT-MHR utilizes a single shaft turbocompressor-generator (TC-Gen), the PBMR incorporates separate low-pressure (LP) and high-pressure (HP) turbocompressors (TC), plus a power turbine-generator (PT-Gen). Common features include the vertical orientation of the machinery, the working fluid, dry helium, and the use of magnetic bearings for support. Both the GT-MHR's turbocompressor-generator and the PBMR's power turbine-generator rotate at the 50 Hz synchronous speed of 3000 rpm; however, the PBMR's turbocompressors will operate at higher speeds. As would be expected from the relative configurations and power outputs of the two designs, the mass of the GT-MHR TC-Gen is substantially larger than that of the PBMR PT-Gen.

TABLE I. SUMMARY OF TURBOMACHINE CHARACTERISTICS IN THE PBMR AND GT-MHR

Parameter	**PBMR** [3]			**GT-MHR** [4]
Nominal Power (MWt/MWe)	265/110			600/285
Component	LP TC	HP TC	PT-Gen	TC-Gen
Orientation	Vertical	Vertical	Vertical	Vertical
Speed (rpm)	15,000	18,000	3000	3000
Mass (ton)	[TBD]	[TBD]	[~50]	105

3. AUXILIARY BEARING DESIGN CONSIDERATIONS

The requirements for auxiliary bearings evolve from a number of interrelated considerations. These include the basic characteristics of the machine in which they are installed and the characteristics of the system of which the machine is a part. In addition, there may be external environmental considerations (e.g. seismic) that affect both the machine and the system. In many cases, the considerations related to auxiliary bearing design are iterative in nature. That is, particular features of a machine or system may influence the selection of a particular auxiliary bearing concept and, conversely, the selection of a particular auxiliary bearing concept may significantly influence the design of the machine. A summary of some of the important considerations in auxiliary bearing design is given in Table II.

3.1. Auxiliary Bearing Options

As evidenced by Table II, auxiliary bearing design involves many diverse and often competing technical issues, which must be balanced in the requirements for any given application. 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 auxiliary bearing concepts in use today fall into four general categories. These are plain bearings, rolling element bearings, planetary bearings and zero-clearance auxiliary bearings.

3.1.1. Plain Auxiliary Bearings

Plain bearings represent the simplest and, potentially, the least expensive of the auxiliary bearing options. In their most basic form, they might comprise interfacing surfaces on the rotor and stator, which are configured to come into first contact in the event that electromagnetic bearing (EMB) support is lost or the capacity of the EMB's is exceeded.

More commonly, plain auxiliary bearings incorporate a tribologically compatible and replaceable wearing surface on either the rotor or stator or on both. This can be as simple as a plain bronze sleeve (radial) or thrust washer (axial), or can be much more sophisticated, involving highly engineered material pairs, and/or the use of compliant mountings to mitigate rotordynamics effects. Depending upon the characteristics and/or requirements of the application, lubrication provisions can be met by the working fluid, built into the materials themselves or provided from external sources.

An example of a sophisticated plain auxiliary bearing system is the rotor delevitation system (RDS) developed by Federal-Mogul (F-M) [5]. The F-M RDS employs special sintered bearing alloys that are impregnated with various lubricants, selected on the basis of particular design requirements. For smaller, simpler machines, the sintered components are mounted on sleeves or thrust washers. For larger and more complex machines, such as the 23MW motor-compressor developed for NAM (a partnership of Exxon and Shell in the Netherlands), the sintered material is mounted on individual bushing pads that comprise sectors of the auxiliary bearing (Fig. 1). In turn, the auxiliary bearing sectors are mounted on compliant springs (for damping) within an outer retaining ring that may be split for ease of maintenance. If required, the bushing pads in the corresponding axial auxiliary bearing (Fig. 2) may also be compliantly mounted.

TABLE II. AUXILIARY BEARING DESIGN CONSIDERATIONS

CHARACTERISTIC	CATCHER BEARING (CB) IMPLICATIONS
Machine Related	
Rotor Mass	Primary influence on CB sizing. Also affects CB design via rotordynamics and rundown times.
Shaft Orientation	Gravitational loads shared by multiple radial CBs in horizontal machines, usually concentrated in a single axial CB in vertical machines. Smaller influence on rotordynamic loads. Becomes more significant as size of machine increases.
Speed (rpm)	High speeds, particularly when combined with large shaft diameters, may reach DN (bore diameter x rpm) limits of CBs. Also influences rundown times.
Rotordynamics	Rotordynamics must be acceptable both on the EMBs and CBs, plus any combination of the two within the design basis (see Reliability/Risk Criteria).
EMB Limitations	When capacity of EMBs is exceeded, CBs must absorb excess loads.
System Related	
Reliability/Risk Criteria	Significant influence on the EMB design and the likelihood of partial vs. full EMB failures. This, in turn, is a primary determinant of the number and type (related to full vs. partial EMB failure) of demands on the CB during its lifetime. May determine the design objectives of the CB (i.e. to avoid vs. mitigate damage to the machine). Also potentially influences CB margins.
Number of Drops	Related to reliability/risk criteria. Likely to be the primary factor limiting life of CBs and potentially lead to need for in-service evaluation and/or replacement provisions.
Operating Environment	Includes temperatures, pressures and characteristics of working fluids, including contaminants. These factors would potentially impact CB materials selections. Also includes limitations placed by environmental requirements on CB options, e.g. lubricant limitations in GCRs.
Overspeed Criteria	Would affect CB design if an assumption of coincident EMB failure were imposed or if overspeed events result in loads that exceed the capacity of the EMBs.
System Induced Loads	These include mostly dynamic loads induced by operation of the system that might be a factor during a rundown event. An example would be unbalanced pressure loads across a turbine, which could potentially exceed the weight of the rotor.
Externally Induced Loads	Examples would be seismic requirements and shock loads from external explosions.
Rundown Time	In large applications, such as GCRs, primarily determined by the system response to EMB failure, plus active braking provisions. Absent system influences, rundown time would be determined by inertial energy stored in the rotor, plus deceleration forces, such as those associated with internal machine friction and windage.
Maintenance Criteria	For example, the extent to which CBs are available for inspection and/or replacement in standby service or following a rundown event.
Bearing Related	
Heat Generation and Dissipation	For a given application, the CB type would have an important influence on the heat generated during rundown and the provisions that are required to manage it.
Lubrication	Lubrication provisions for a given CB design will have a significant influence on heat generation during a rundown event. They may also influence the applicability or desirability of specific bearing design for a given application
Materials Characteristics	The characteristics of materials will influence the size of the CBs and the compatibility of the CBs with the machine and system.
Cooling Provisions	CB cooling provisions would allow higher loads and/or extended rundown periods.
Gap Control	In a zero clearance design, the capability exists to actively control the gap between the auxiliary bearings and supported shaft, providing additional control over machine rotordynamics during rundown.

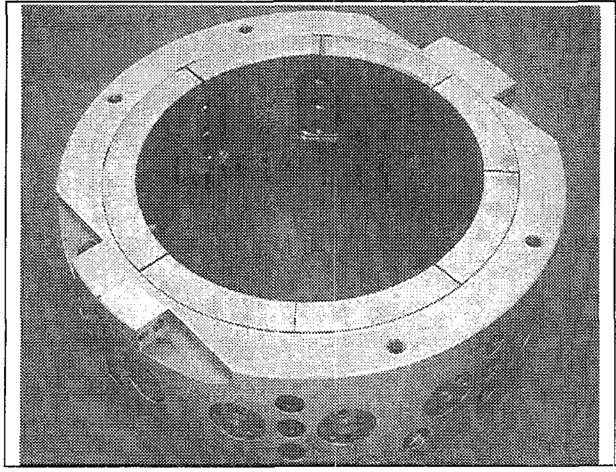
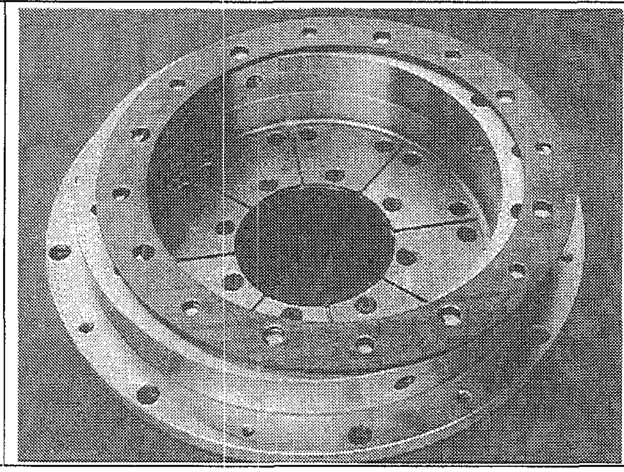
 

Figure 1. NAM Radial Auxiliary Bearing [5] Figure 2 NAM Axial Auxiliary Bearing [5].

3.1.2. Rolling Element Auxiliary Bearings

At present, the majority of large machines supported on EMB's incorporate rolling element auxiliary bearings for backup. The most common configuration is dual full-complement angular contact ball bearings mounted in a face-to-face preloaded configuration in order to prevent windage-induced motion under standby conditions. The angular contact design supports both radial and axial loads. Balls are preferred over rollers in such backup applications to minimize the potential for skidding during acceleration. Full-complement bearings (which are cageless) avoid the issue of cage-related failure modes, which were observed in earlier qualification tests for a GCR circulator (see Section 4.2.). In some applications, a clutch- or plain bearing-like material is provided at the rotor-bearing interface to moderate the initial acceleration and/or avoid rotor damage.

3.1.3. Planetary Auxiliary Bearings

In planetary auxiliary bearing concepts, an external ring surrounding the rotating shaft serves as a carrier for three or more individual rolling elements (planets) (Fig. 3). The number of planets and the type of rolling element depend upon the requirements of the specific application. The corresponding interface on the rotating shaft might be a hardened sleeve or a tribologically engineered surface, similar to that used in a plain bearing design, also depending on application requirements. The primary uses of planetary designs are with large diameter shafts and/or applications in which the rated speed (DN rating) of the bearing becomes an issue. This is because the planetary configuration reduces the effective DN requirement in accordance with the following formula:

$$\mathbf{DN = Rotor\ OD\ (mm) \times rpm \times} \left[\frac{\mathbf{Bore}}{\mathbf{OD}}\right]_{\mathbf{Roller}}$$

In practice the ratio reduces the effective DN speed of the bearing by 0.5 or more [6].

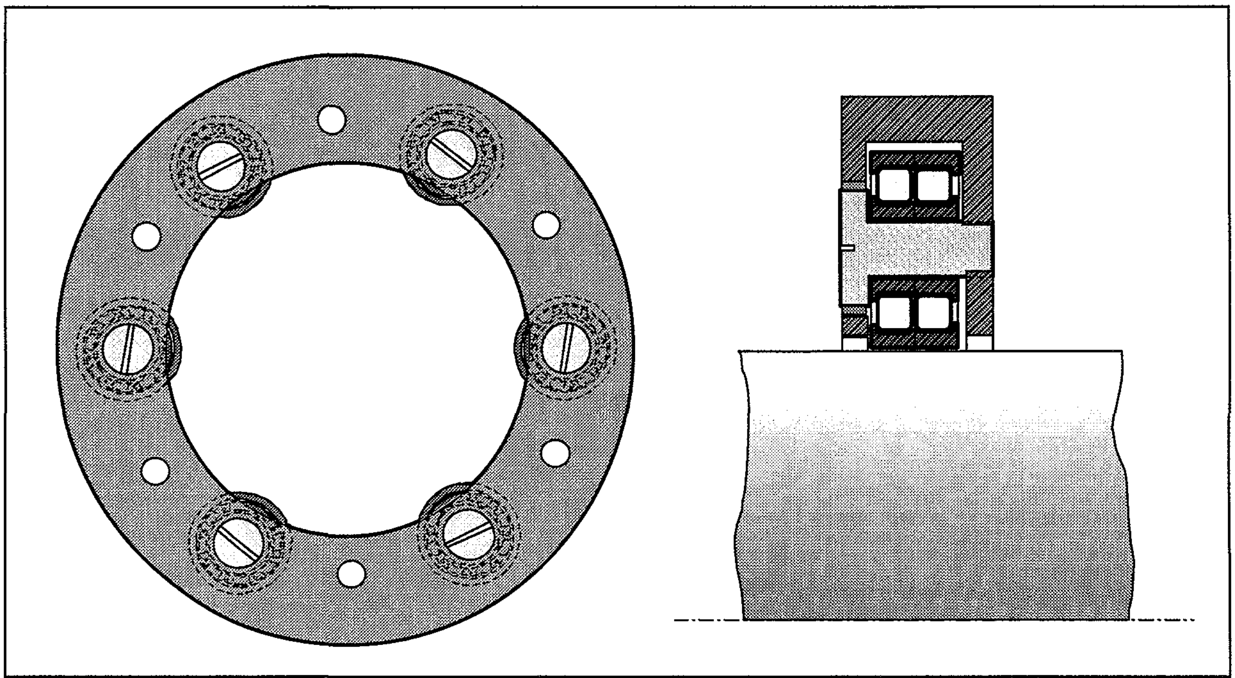


Figure 2. Planetary Auxiliary Bearing.

3.1.4. Zero-Clearance Auxiliary Bearing (ZCAB) [6][7]

The zero clearance auxiliary bearing (ZCAB), developed by Mohawk Innovative Technology Inc. (MITI) (Fig. 4), is a specialized variant of the planetary auxiliary bearing concept. Its particular objectives are to support the shaft with zero clearance when the auxiliary bearing is actuated and to provide a "get home" capability for certain defense-related aviation and marine applications.

In the ZCAB design, the azimuthal position of the planetary elements (typically five to eight) is maintained by a pair of drive rings, incorporating radial slots. The slots engage the individual shafts upon which the planetary elements are mounted and allow limited movement of the planetary elements toward and away from the machine rotor. The drive rings along with their planetary elements are, in turn, mounted within a separate support assembly that incorporates two support plates with spiral circumferential slots. These spiral circumferential slots also engage the individual shafts upon which the planetary elements are mounted. As the support plates are rotated relative to the drive rings, the planetary elements are moved closer to or further from the shaft. Through this means, the nominal clearance between the machine rotor and auxiliary bearings is closed when the auxiliary bearings are actuated. By reversing the process, control can be returned to the magnetic bearings upon their restoration. As is the practice with other auxiliary bearing designs, the support assembly is usually compliantly mounted to provide an additional measure of damping.

The relative movement between the drive rings and support plates and, thus, the clearance between the auxiliary bearings and machine rotor can be actively or passively controlled in either direction. In the specific example depicted in Figure 4, the ZCAB is held in the open position by a spring mechanism (F_S). In the event of failure of the radial EMB, or a loading in excess of its capacity, the rotor will impact one or more of the planetary elements. This results in a tangential force (F_t), which moves the planetary element(s) and, thus, drive rings in the direction shown. This, in turn, causes the planetary elements to move inward and engage the rotor.

In instances where both axial and radial loads must be reacted, the outer ring of the planets can be modified to an elliptical form, which acts in conjunction with a corresponding groove in the shaft to withstand axial loads. An example of such an interface is provided in Fig 5 [8].

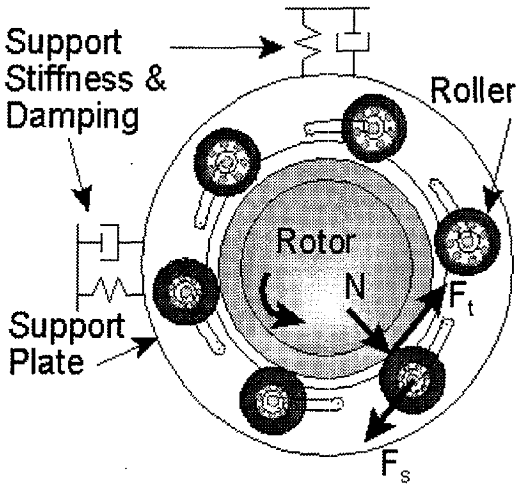


Figure 3. Zero-Clearance Auxiliary Bearing (ZCAB) [6].

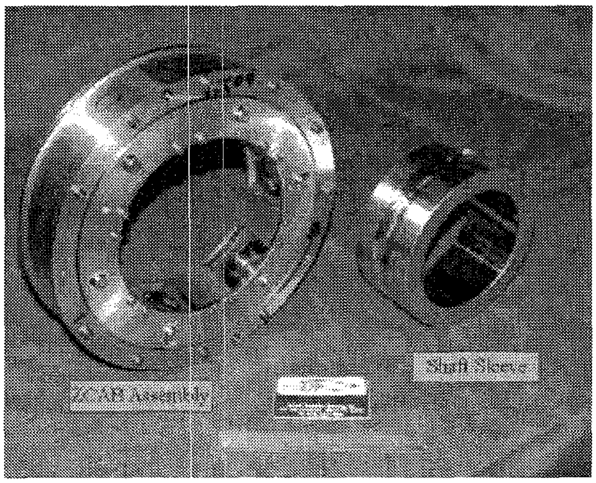


Figure 4. ZCAB Providing Both Axial and Radial Support [8].

3.1.5. *Hybrid Bearings*

In addition to the individual bearing types described in the Sections 3.1.1 through 3.1.4, various permutations and/or combinations of these concepts can be considered. For example, one bearing type can be used for axial support while another is used for radial support. Additionally, the materials associated with the plain bearing types could be utilized in the interface between the rotating shaft and the various rolling element or planetary designs.

3.2. Advantages And Disadvantages Of Auxiliary Bearings

Each of the auxiliary bearing designs described in Section 3.1 has its own set of advantages and disadvantages. These are summarized in Table III.

Note that there is a fundamental tradeoff between plain auxiliary bearings and the low friction bearing types, which include both simple rolling elements and planetary configurations. The plain bearing types have significant advantages in terms of their simplicity, durability in standby service over extended periods of time and the ability to assess their condition without direct access or removal. However, these advantages come at the cost of greater friction and heat generation, which must be managed during rundown. The significance of this issue increases with higher loads and stored energy (primarily driven by rotor mass and speed). Furthermore, the tribological characteristics of plain auxiliary bearings have not yet been verified for the dry helium environment of the GCR designs.

By contrast, the low friction bearing types, and particularly the planetary designs, minimize the additional heat generated during a rundown event. However, the initial acceleration of the bearing is a very violent event, potentially leading to bearing damage or even failure. In addition, these bearings must survive long periods of standby service, prior to being called into service. During this period, undesired motion of the bearings must be prevented and damage by corrosion and/or contamination must be avoided. Furthermore, there is only limited potential for assessing the condition of the bearings, either as a preventative maintenance measure or following their use. One positive factor specific to GCR's is that the dry helium environment will tend to limit corrosion in standby service.

The advantages and disadvantages of planetary auxiliary bearings are similar to those for simple rolling element bearings, but with the additional advantage that the DN rating required of the individual planetary bearing elements is reduced. This desirable feature must be weighed against their additional complexity and cost.

The tradeoffs associated with the ZCAB concept are similar to those for other planetary designs; however, the ZCAB concept provides an additional capability to eliminate the rotor-bearing gap and allow enhanced control over shaft rotordynamics during a rundown event. As previously noted, the ZCAB design has also been developed to provide an extended runtime or "get home" capability in critical aviation and marine applications. However, both the extended runtime feature and the potential for ZCAB auxiliary bearings to provide axial support would need further evaluation in the context of GCR requirements.

4. AUXILIARY BEARING EXPERIENCE BASE AND IMPLICATIONS FOR GCR'S

As of the present, there are several hundred large turbomachines operating on magnetic bearings, with a cumulative experience base well in excess of 5 million operating hours [9]. The majority of these large machines are industrial compressors and turboexpanders, which are used in both the production and transportation of liquid and gaseous fuels. The size of these machines ranges up to 29 MW, with rotational speeds (not coincident) as high as 70,000 rpm.

While the existing experience base is substantial, there is little actual experience with vertical machines. Further, there is no field experience and only limited research specific to the conditions found in GCR's. To date, the largest vertically oriented machine on magnetic bearings is Korea Electric Power Company's Yoshino hydropower installation. The Yoshino machine is a 6 MW Francis turbine driving a generator, operating at 600 rpm [10, 11]. The weight of the rotor is 35 tons, about 2/3 of the weight of the PBMR rotor (~50 tons) and 1/3 of the weight of the GT-MHR rotor (105 tons).

There have been two related experimental programs specifically addressing auxiliary bearing issues for GCR's. James Howden Company conducted both of these programs, which supported both a proposed retrofit of the Fort St. Vrain power plant gas circulators with magnetic bearings and the gas circulator design for a modular steam cycle high temperature gas-cooled reactor (MHTGR)[12,13].

TABLE III. ADVANTAGES AND DISADVANTAGES OF AUXILIARY BEARING CONCEPTS

Technology	Advantages	Disadvantages
Plain Auxiliary Bearings	• Low-cost • Passive, no moving parts in bearing • Reduced potential for deterioration in standby mode • Condition, wear may be assessed by measuring clearance with EMB's	• Higher friction coefficients, heat generation during rundown
Rolling Element Auxiliary Bearings	• Low-cost • Low friction coefficients, heat generation during rundown • Potentially minimum volume with combined radial/thrust bearing	• Potential for bearing/cage damage during acceleration • Potential for deterioration in standby mode; contamination must be avoided • Windage induced rotation must be prevented in standby mode
Planetary Auxiliary Bearings	• Reduced DN for given rotor diameter and speed • Low friction coefficients, heat generation during rundown	• Greater complexity and cost • Contamination must be avoided • Windage induced rotation must be prevented in standby mode • Potential for acceleration damage (reduced relative to rolling element bearings)
Zero Clearance Auxiliary Bearings	• Eliminates rotor-bearing gap during rundown • Extended run time capability • Reduced DN for given rotor diameter and speed • Low friction coefficients, heat generation during rundown	• Greatest complexity and cost • Actuation failures should be considered • Contamination must be avoided • Windage induced rotation must be prevented in standby mode • Potential for acceleration damage (reduced relative to rolling element bearings)

4.1. Applicability of Experience in Horizontal Machines

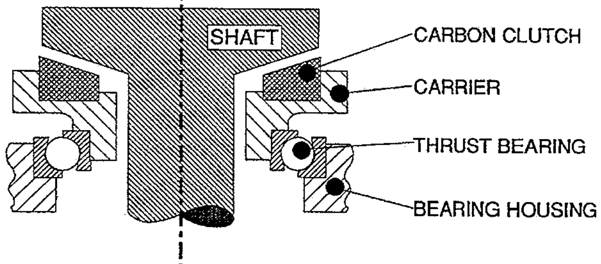
For the auxiliary bearing, the primary difference between horizontal and vertical machines is the direction and distribution of loads. In a horizontal machine, the weight of the rotor is distributed over at least two radial auxiliary bearings if a complete failure of the magnetic bearing system occurs. In a vertical machine, the entire weight of the rotor is usually concentrated on a single axial auxiliary bearing. This concentration of load affects both the initial impact on the auxiliary bearing and the heat generated during the rundown, both of which must be sustained at one location.

By contrast, rotordynamic differences are likely to be of secondary importance, when comparing otherwise identical horizontal and vertical machines. Since rotordynamic response is determined by the physical characteristics of the shaft and the stiffness and damping of the supports, there would be little change in key dynamic parameters, such as critical speeds, if the location and characteristics of the auxiliary radial supports were comparable. One notable orientation-related difference is the reduced potential for backward whirling in horizontal machines that results from the stabilizing influence of gravity. However, experience indicates that the tendency for backward whirling in vertical machines can be controlled by appropriate design of the auxiliary bearings (See Section 4.2).

Given the rotordynamic similarities between horizontal and vertical machines, the sophisticated rotordynamic analysis tools now used to predict the behavior of horizontal machines can also be used for vertical applications. This includes the ability to predict and avoid the incidence of backward whirl. One caveat that applies to both orientations, however, is that the physical characteristics of the rotor components must be well understood. This can be problematical in some instances, such as accurately predicting the effective stiffness and internal damping of compressor, turbine, generator and coupling design features.

4.2. Howden Catcher Bearing System Development Tests

Initial design work by James Howden & Co. in the 1985 timeframe [14] resulted in the selection of rolling element bearings to back up the magnetic bearings in the gas circulator for the MHTGR. Balls were chosen over cylindrical rolling elements to avoid the potential for skidding at the ends of the rolling elements in the dry helium environment. The upper auxiliary bearing, which was required to provide both axial and radial support employed an angular contact design. The radial auxiliary bearing at the lower end of the circulator was of a deep groove ball bearing design that would provide radial support only. To moderate the initial acceleration transient imposed by axial loads on the upper auxiliary bearing, a graphite ring was incorporated at the rotor-bearing interface. The resulting radial/axial auxiliary bearing design for the MHTGR is schematically depicted in Figure 6.



Figire 5. MHTGR Axial Auxiliary Bearing Configuration.

Given the lack of experience with auxiliary bearings in vertical machines, two separate research programs were conducted between 1988 and 1991, addressing the axial and radial support characteristics of the bearings, respectively. These are described in the following sections.

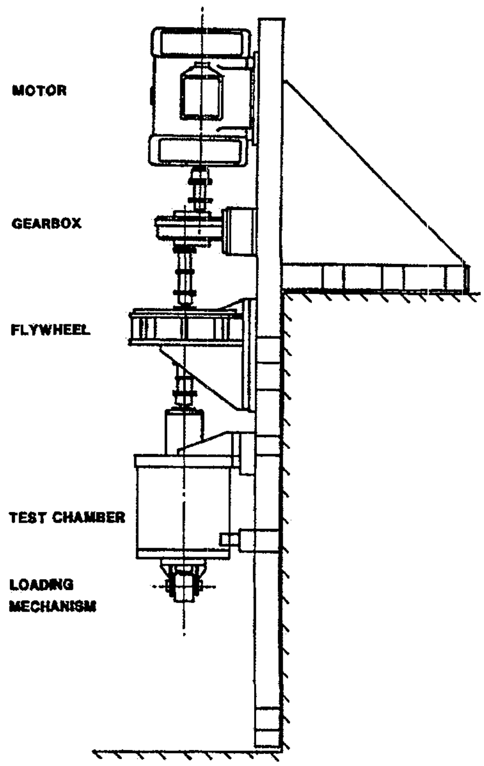


Figure 6. Axial Auxiliary Bearing Test.

4.2.1. Axial Support Tests [12]

In addition to the MHTGR circulator, the axial support tests were initially intended to support the prospective retrofit of the Fort St. Vrain High-Temperature Gas-Cooled Reactor (HTGR) circulators with magnetic bearings. On this basis, the maximum loads selected for the tests were 100 kN impact and 33 kN steady-state. Also the maximum speed for the tests was initially set at 10,500 rpm, based on Fort St. Vrain requirements. However, this was subsequently reduced to 6000 rpm, following the permanent closure of the Fort St. Vrain plant. A further objective of the tests was to achieve 20 drops on the axial auxiliary bearing from a height of 0.5 mm. This represented a factor of two margin relative to the MHTGR circulator design criteria, which was 10 drops.

The rig utilized for the axial bearing tests is graphically depicted in Figure 7. The test shaft was driven by an inverter/motor combination through a speed-up gearbox. A flywheel was incorporated to provide some inertia on the shaft and, when combined with continued operation of the motor, was adequate to represent the energy input during the bearing acceleration transient. The catcher bearing assembly test article was enclosed in a hermetically sealed enclosure, which contained dry helium corresponding to the MHTGR requirements.

A total of 7 test series were run with the number of drops in each ranging from six to thirty. The test series explored various options for the axial auxiliary bearing design, which are identified along with test results in Table IV. The significant conclusions of the axial auxiliary bearing tests are summarized in Table V.

4.2.2. Radial Support Tests [13]

While the axial tests were primarily concerned with the initial acceleration of the combined radial/axial auxiliary bearing at the upper end of the circulator, the radial tests were focused on the potential for backward whirling instabilities in the lower radial bearing. Simplified mathematical models of such instabilities had predicted the potential for very high dynamic loadings associated with movement of the rotor within the nominal clearance of 0.5 mm provided at the interface between the rotor and auxiliary bearings.

Based on preliminary scaling calculations completed during 1989, a suitably designed quarter-scale model was deemed adequate to model the behavior in the MHTGR circulator. The analytical evaluation also included a full assessment of the conditions required to initiate the feared orbital motion in the model. Paradoxically, it was found that initiation of the orbit motion could not be assured unless artificially induced.

The quarter-scale model developed through the analytical study is depicted in Figure 8. It comprises a rotor driven by an air motor located at the top of the rig, connected by a double universal drive. The rotor is initially spun up on ball bearings at the top and bottom of the rig. A release mechanism is provided that removes lateral support from both the top and bottom bearings simultaneously. Following release, the upper bearing support continues to provide axial support. A permanent magnet device is incorporated to ensure that the orbit motion is always initiated.

Initial tests quickly revealed that the orbit frequency quickly reached 38 Hz and was independent of shaft speed over a range of 5-20 Hz. It was concluded that the orbit frequency was unrelated to the dynamic characteristics of the system, but instead arrested by a balance between friction-induced acceleration forces and damping. However, the level of damping observed was substantially higher than expected. The source of the excess damping was at first a mystery, but finally identified as being the result of rolling friction, a normally negligible component in a linear system. However, it was subsequently shown that when the shaft is rolling within a track that has been "rolled up into a circle", the normally small rolling resistance force becomes amplified by a factor equal to R/(R-r), where R is the hole radius and r is the shaft radius. The amplification factor in the model was approximately 200, and accounted for the excess damping. With this revelation, the results of the quarter-scale tests can be summarized as in Table VI.

5. DISCUSSION AND CONCLUSIONS

The large size and vertical orientation of the PBMR and GT-MHR turbomachines pose particular challenges in the design and development of auxiliary bearings. Field experience is limited, with only one large vertical machine, incorporating a 35-ton, 600-rpm rotor, presently operating on magnetic bearings. That compares with the ~50 ton PBMR rotor and the 105 ton GT-MHR rotor, both of which will operate at 3000 rpm. The remaining experience is derived from EMB-equipped flywheels and limited testing in support of GCR circulators, both of which relate to smaller, albeit higher speed machines.

TABLE IV. HOWDEN AXIAL AUXILIARY BEARING TEST SUMMARY

Feature	Options Tested	Test Results
Bearings	RHP 7224 • 25 degree contact angle • Inner ring centered solid brass cage	Series 1: Bearing failure attributed to lubrication breakdown. Series 2: Severe cage wear after 27 drops. Evidence of local heating at cage/inner race interface. However, bearing remained functional.
	SKF 7224BM • 40 degree contact angle • Rolling element centered solid brass cage	Series 3: Bearing undamaged after 30 drops. Series 4-7: Cage cracking or fracture after 6-18 drops, however, running elements of bearing remained undamaged, functional. Cage fractures attributed to difference in processing vs. bearing in Series 3.
Lubricants	Kluber Barrierta 1MI	Lubrication failure, discontinued after first test series
	Dupont Krytox 250AC (High temperature grease)	Acceptable in remaining test series
Rubbing Materials	Electrographite (Morganite EY308	Cracking observed after first test series (10 drops), severe cracking after 3 test series, totaling 67 drops
	Carbon fibre reinforced graphitized carbon (Dunlop CB7 or similar)	No observable damage after two test series (16 drops)
Cone Angle	15.0 degrees	Insignificant radial support
	27.5 degrees	No significant improvement in radial support. Increased angle resulted in faster acceleration, greater damage potential

TABLE V. SIGNIFICANT RESULTS AND CONCLUSIONS OF HOWDEN AXIAL AUXILIARY BEARING TESTS

1. Axial auxiliary bearing design for the MHTGR circulator successfully qualified for 10-drop criteria.
2. Both of the angular contact ball bearings tested were found acceptable
• Rolling element-centered cage (SKF bearing) judged to be preferable
• Sand blasting of cage in Series 3 bearing apparently reduced residual stresses from manufacture, avoided cracking seen in other test series with SKF bearings
• Higher strength cage materials should be considered
3. Clutch cone angles do not provide significant radial support. Higher cone angles may aggravate acceleration-related damage.
4. The observed friction coefficients in the carbon based clutch materials were significantly lower than expected. It is possible that carbon debris generated at the interface forms a rolling lubricant, reducing the friction.
5. Both graphite and carbon reinforced carbon clutches were satisfactory, with the latter preferred.

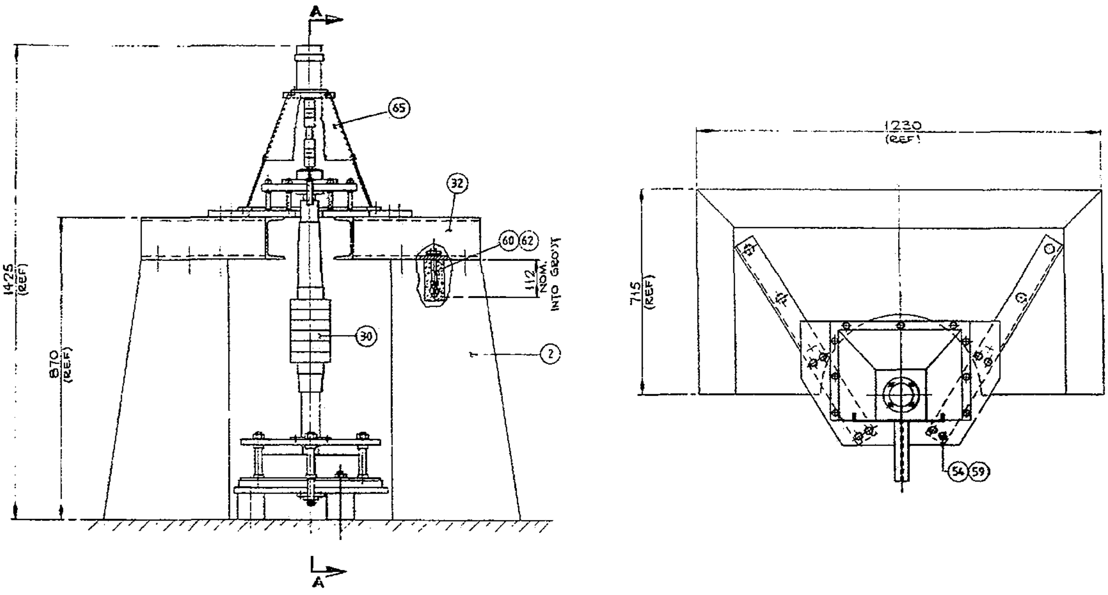


Figure 7. Radial Auxiliary Bearing Test Rig.

TABLE VI. SIGNIFICANT RESULTS AND CONCLUSIONS OF HOWDEN RADIAL AUXILIARY BEARING TESTS

1. Orbital motion of the rotor is a real phenomenon that can be induced for a wide range of operating conditions.
2. Arrestment of orbital speed was unrelated to any dynamic phenomenon; rather, the orbital speed would increase rapidly until damping energy balanced the input energy due to friction.
3. Energy loss due to structural and shaft damping processes by themselves cannot limit the orbit speed and resulting radial loads to tolerable levels, unless the bearing friction coefficient is impracticably low or an extraordinary level of damping is incorporated.
4. Orbit arrestment speeds were consistently below that explained by structural damping alone. Follow-up analyses indicated that rolling resistance at the shaft-bearing interface accounted for the unexplained energy absorption.
• Normally negligible, but becomes the dominant factor in a close conformity rolling geometry
• Resulting force scales with the ratio of bearing diameter to bearing clearance
5. Rolling resistance loss can limit the orbit speed and radial forces to tolerable levels and can provide an engineering basis for control of rotor behavior on failure of the magnetic bearing system.

Not withstanding the above, the technical basis for radial auxiliary bearings in large vertical machines appears to be established. Over 5 million hours of operation have accumulated in large horizontal machines on magnetic bearings, and the rotordynamic issues that are the principal concern with the vertical radial bearings are common to both horizontal and vertical machines. Rotordynamic analysis tools presently available are believed to be adequate for either orientation. Further, the Howden vertical rotor tests, described in Section 4.2, provide additional assurance that the rotordynamic responses in a large vertical machine can be adequately managed by appropriate design. For the radial auxiliary bearings, there appears to be no compelling advantage for either the plain bearings or low-friction designs that are likely to be considered for this size application. The principal tradeoff will be in terms of minimal heat generation (low friction) vs. low maintenance (plain).

The axial auxiliary bearing, however, remains a significant challenge, due to the high loads and associated heat generation that are likely to be encountered during the rundown process. This is aggravated by the fact that the generator must be disconnected from the grid at the start of the rundown transient and that provisions for dynamic braking are both difficult and expensive. These factors clearly favor low-friction designs, and it is likely that development and testing of the axial auxiliary bearings will be required.

Finally, the reliability of magnetic bearings continues to improve, leveraging on the continuing major advancements in both computers and power electronics. Given a level of redundancy appropriate to the size of the GCR machines, it should be possible to develop a bearing support system that would place minimal demands on the auxiliary bearings. In this context, the number of rundowns and the criteria for success (e.g. damage avoidance vs. damage mitigation) are open to further discussion.

Consideration of the above leads to the following conclusions:

1. The design of the primary EMB support system should be such that the likelihood of a rundown on the auxiliary bearings is minimized, preferably being not likely to occur within the lifetime of the plant.
2. The technical basis for rotordynamic design of the radial auxiliary bearings is presently in hand and is adequately supported by computer modeling techniques. While it appears that either plain or planetary auxiliary bearing designs could be applied, management of the heat generated during rundown must be addressed for either type.
3. The key to the design of the axial auxiliary bearings in the GCR designs will be management of the heat generated during the rundown process. This is presently viewed as problematical in these machines. Of the auxiliary bearing designs described in this paper, this would tend to favor the rolling element concepts, given their lower friction coefficients. Active cooling provisions are likely to be required during the rundown process.
4. Testing of axial auxiliary bearing design alternatives is recommended, preferably at full-scale.

REFERENCES

[1] Nicholls, D.R. "Status of the Pebble Bed Modular Reactor", Presented at the EPRI Helium Gas Reactor Workshop, Palo Alto, CA, December 7-8, 1999.

[2] Simon, W. "Status and Current Activities of the International GT-MHR Program", Presented at the EPRI Helium Gas Reactor Workshop, Palo Alto, CA, December 7-8, 1999.

[3] http://www.pbmr.com/pebbed3/Simulate/800.html, October 2000.

[4] Rodwell, E. "Helium Gas Turbine Reactor Technical Challenges", TP-114690, EPRI, Palo Alto, CA, January 2000.

[5] *Federal Mogul News Update*, Federal-Mogul Magnetic Bearings, Mystic, CT, Number 4, 1999.

[6] Chen, H.M. Walton, J. Heshmat, H. "Test of a Zero Clearance Auxiliary Bearing", *Proc. of the MAG '97 Industrial Conference and Exhibition on Magnetic Bearings*, MITI, Albany, NY, August 21-22, 1997.

[7] "Zero Clearance Auxiliary Bearing Developed", http://www.albany.net/~miti/feb99.htm, downloaded, October 20, 2000.
[8] "Auxiliary Bearing Developed for 140mm Shaft with 1000 Pound Load at 18,000 RPM", http://www.albany.net/~miti/sep99.htm, downloaded October 20, 2000.
[9] *Turbomachines Reference List*, S2M, St. Marcel, France, June 9, 2000.
[10] "35 Tons Load Capacity", *Actidyne News*, S2M, St. Marcel, France, September 1998, p. 3.
[11] Yamaishi, K. "Applications and Performance of Magnetic Bearing for Water Turbine and Generator", Nippon Koei Co. Ltd. Yokohama, Japan, Circa 1997.
[12] Rennie, J.A. *Catcher Bearing System Development and Test, Thrust Bearing Tests – Final Report*, TR7404, Prepared by James Howden & Co. Renfrew, Scotland for EPRI, Palo Alto, CA, June 14, 1991.
[13] Fraser, W.M, Catcher Bearing Development and Test, Final Report on Dynamic Behaviour of a Gas Circulator Rotor During Run-Down in Catcher Bearings, TN7411, Prepared by James Howden & Co. Renfrew, Scotland for EPRI, Palo Alto, CA, May 30, 1991.
[14] Lewis, M.W.J. "Helium Gas Circulator Catcher Bearing Selection and Design", NCT Report 2585/451 prepared for James Howden & Co. Renfrew, Scotland, August 1985.