Developments and Tests on Retainer Bearings for Large Active Magnetic Bearings

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1. Reactor Coolant Gas Circulators with Magnetic Bearings and the Retainer Bearing Test Facility

Radial and axial active magnetic bearings due to their specific features and operating characteristics can meet technical requirements beyond the scope of conventional bearings. With these bearings rotating machines can be arranged encapsulated in the process medium without any risk of interaction with lubrication oil or stationary component parts during normal operation. They are thus predestined for use in gas-cooled high temperature reactors. This is true in particular because the demand of freedom from maintenance is met here in a way not reached in mechanical engineering before.

Fig. 1 is a cross section through a coolant gas circulator prototype being developed at present. It includes the same proven assemblies used in the circulators for gas-cooled reactors designed and built by ABB to date /1/. A basically new feature is the vertical circulator axis with rotor weights up to several tons and, naturally, the non-contacting active magnetic bearings /2/.

The circulator is designed as a withdrawable unit for a reactor service life of forty years, with an overhung impeller arranged at the top. Drive motor and cooling device are integrated. The complete unit is arranged in the helium loop of the reactor and is under operating pressure.

An essential prerequisite for the use of active magnetic bearings are retainer bearings, needed for preventing undesired contacts between the rotor and fixed parts of the machine when the magnets are de-energized. In view of the large dimensions and the shaft weights to be kept under control design and construction of the retainer bearing are the subject of a research and development project backed by experiments.

Fig. 2 depicts the functioning mode of the retainer bearing test facility with active magnetic bearings designed by ABB and set up at HRB in Jülich. Its configuration and shaft dimensions have been chosen such that they correspond to the conditions of the prototype circulator as far as possible. The locations shown in red and dark blue illustrate the principle: if the active magnetic bearings fail the shaft will drop onto conical parts which themselves run in rolling-contact bearings.

Within a few hundred ms the retainer bearing is accelerated to the shaft speed when the shaft is dropped. The shaft may then safely coast to a halt according to the power consumed in the impeller wheel or with additional electric braking by the motor.

The advantage of the active magnetic bearing in the retainer bearing test facility is not only that shaft drops take place under realistic boundary conditions. It is also possible in addition to make rotor loads associated with larger rotor dimensions act upon the retainer bearings via magnetic forces. And finally it is possible to re-energize the magnetic bearing during an experiment in the event of an impending failure of a retainer bearing. The rotor will then return to the levitation position and further damage of conical parts and retainer bearing components is avoided.

Fig. 3 is a view of the retainer bearing test facility. The switch cabinets house the retainer electronic devices for the magnetic bearing and the converter as well as the instrumentation. The actual rotating system is inside the container coated with light paint; this container also allows experiments in a helium atmosphere. Nearly 100 rotor drops have been carried out to date.

2. Design Principles for Retainer Bearings

Fig. 4 illustrates the design features. In the axial-radial retainer bearing the shaft forces are directed via friction cones into the retainer bearing bushing which in this case runs in three rolling-contact bearings. Depending on the failure mode the shaft can be supported by the pairs of cones RKI or RKII.

In a radial retainer bearing radial shaft motions are guided by the cylindrical friction surfaces RSIV. For centering of the shaft the pair of cones RKIII may be used.

The retainer bearing units thus basically consist of rolling-contact bearings and friction surfaces. By means of the friction coefficient μ a major influence on the angular acceleration of the retainer bearing bushing is exerted right at the start of the drop. Fairly smooth transitions from standstill to rated rotational speed are to be aimed at in order to keep the ball and ball retainer ring accelerations occurring inside the rolling-contact bearings within acceptable limits.

The friction surfaces furthermore have the additional basic task of acting as an emergency bearing and of bridging a potential rolling-contact bearing failure.

The friction cones may be made of various materials, such as e.g. sintered materials based on bronze, metallic and ceramic coatings, and metallic materials of high surface hardness. For the technology of helium-cooled reactors the considerably differing friction behaviour - compared with normal atmospheric conditions - is the main factor for the choice of materials. Generally the values for the friction coefficient μ in helium under reactor conditions are much higher than in air.

Fig. 5 shows an axial-radial retainer bearing unit in assembled condition with mounted telemetering instrumentation for the non-contacting transmission of temperatures measured in the rotating part. A part of the friction cone with a radial slot arranged for collecting abraded material is clearly visible. The balls, running grooves and retainer borings shown represent the parts of a dry-lubrication bearing after exposure to loads. The matt luster of the balls and the traces of tarnishing are clear indications that ball bearing wear has already started.

As the presentation of the test results to date will also show, great importance is attached to the tribology of rolling-contact bearings. For the application of retainer bearings in machines with active magnetic bearings used for nuclear reactors dry lubrication has a wide application potential. In view of a reactor service life of 40 years the maintenance requirements and disadvantages involved in the limited stability time of lubricating grease justify almost any effort for developing serviceable dry-lubrication retainer bearings with reproducible behaviour.

In our opinion the application of the active magnetic bearings technology for large machines in nuclear engineering depends decisively on the number of safely guaranteed drops into the retainer bearings. For conventional mechanical engineering this restrictions does not apply to the same extent, since reactor-related and conservative availability considerations for active magnetic bearings and their components need not be made. The difficulties involved in the handling of irradiated machine components do not exist, and the bearings are always accessible. For conventional engineering the operational and economic benefits of active magnetic bearings are therefore much more distinct.

3. Phenomenology of the Test Parameters

Fig. 6 shows the typical curves of a number of test parameters for a drop from an initial speed of 6100 r.p.m.

From the curve trace "ARF bush speed" it can be seen that the run-up of the axial-radial retainer bearing (ARF) occurs approximately 1.1 s. For the radial retainer bearing (RF) an analog run-up with a time delay of approximately 0.5 s takes place according to the curve trace "RF bush speed".

The curve of the axial force taken up by the ARF is most interesting. At a rotor weight of 1320 kg in the test facility an additional magnetic load was introduced for this test. As the curve trace "axial force" shows an axial load of approx. 25 kN was present after the dynamic processes involved in the drop had died down.

The shock load occurring at the first moment of the drop amounting to 104 kN was four times as large as the quasi-static load occurring subsequently.

The curve trace "Rotor Displacement" for the axial shaft position clearly reflects the dynamic dropping process. The elasticity of the design chosen by us permits an overshooting of more than 0.2 mm beyond the axial "drop path" of the rotor amounting to approx. 0.7 mm in the equilibrium condition.

In Fig. 7 the experimental findings for the run-up time of the ARF cone with retainer bearing bushing are depicted as a function of the drop speed of the shaft. The scattering of the measured values as shown reflects, apart from the effect of the damage to the rolling contact bearing which increases with the number of drops the assumption that the friction processes during the drop are much more complex than might be expected according to the classical friction law.

With Fig. 8 the important influencing quantity temperature increase in the friction cone comes in. It also rises considerably with rising drop speed. If the plotted data are adapted by means of a power function one obtains nearly the theoretically expected speed exponent 2 for the approximation of the ΔT figures ($\ln \Delta T = - 12.6 + 1.93 \ln n$).

In Fig. 9 the evolution of the temperatures in the 5 rolling-contact bearings L1 to L5 of the entire retainer bearing arrangement for a drop from approximately 6000 r.p.m. is shown. The shaft coastdown time amounts to slightly more than 15 minutes. Distinct maximum temperature differentials of up to 12 K appear between internal and external ring of the ball bearings with a lubricating grease film.

For dry-lubrication bearings the respective ΔT values are higher by a factor 3 to 5. The corresponding material expansions are a major input variable for the detail design of circulators, because they directly affect the tolerances and clearances between rotating and fixed machine parts.

Since additional reserves for temperature gradients in the circulator resulting from operational influences must always be taken into consideration, it is quite obvious that this is one of the major problem areas for the design of retainer bearings for large machines.

This brief survey makes clear that even with the use of commercial rolling-contact bearings with a grease film no satisfactory results could be observed at first during the drops. For a successful application considerable modifications and adaptations of the design parameters are necessary. This concerns in particular:

- Bearing play, pressure angle and osculation
- Ball bearing spacer guidance, design and material
- Surface configuration of the elements involved in the rolling movement
- Type, quantity and method of application of lubricants
- Damping of the bearings.

In case of dry bearings special attention must be paid to some of the above-mentioned parameters acc. to Fig. 10, since the temperature developments are much more pronounced during a drop, and the impact of the tribological properties is much more serious. Therefore experimental parameter optimizations are necessary which are being carried out by us at present.

The interactions of the various material gliding upon each other in the rolling-contact bearing are a key problem, with the proportion of the masses to be accelerated in the starting phase exerting a great influence. Dry lubrication films can help to avoid the rapid destruction of the surface structure of balls, rings and retainers in a helium atmosphere with metallic contacts and cold welding.

From a phenomenological point of view the bearing load is best determined by a comparison of two surface peak-to-valley measurements. Fig. 11 shows two extremes: rolling-contact bearing surfaces in new condition and after failure due to overloading. The shown orders of magnitude for the surface roughness illustrate the experimental tolerance, with the failure value for the surface roughness of a rolling-contact bearing being dependent on the dry lubricant used. It is our objective to discover recipes and procedures by means of which, in a similar way as with a film of grease, the metallic contact is reproducibly prevented for a large number of drops. A prerequisite for this is the development of suitable non-destructive inspection procedures for in-process manufacturing control.

4. Metrology Assisting the Development of Retainer Bearings

Irrespective of the influences of dry lubrication discussed so far there is a large improvement potential in the configuration of the heat removal when the retainer bearings are loaded. There are structural possibilities available as well as the possibility to exert influence through forced cooling.

For solving the difficult optimization tasks we are assisted in the retainer bearing test facility by the sophisticated instrumentation which has been installed. In Fig. 12 the significant measured variables and the means for recording them are illustrated. The temperature measuring devices based on complex telemetric equipment shall be specially mentioned, since thermocouples can be placed in the immediate vicinity of the rotating heat sources, and there are no disturbing factors when the thermoelectric voltage is picked up.

In Fig. 13 the design data of the retainer bearing test facility are compiled. The large variation scope of the individual parameters arose from the requirements involved in the different circulator variants of future HTR lines, from the heating reactor up to the HTR 500.

We thus have a universally useful test equipment at our disposal allowing us to investigate even special features of active magnetic bearings.

The success already achieved with the detail design of retainer bearings for coolant gas circulators is an indication of the fact that the retainer bearing test facility may in the medium term also be used for optimizing retainer bearings for other applications. Under this aspect the R&D funds invested will have a "spin off" effect beyond the current objective. We wish to thank the Federal Ministry of Research and Technology for their sponsorship of the experimental work.

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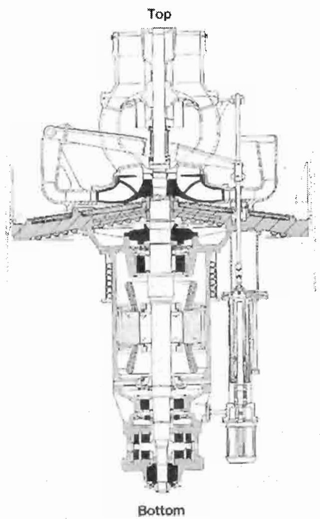

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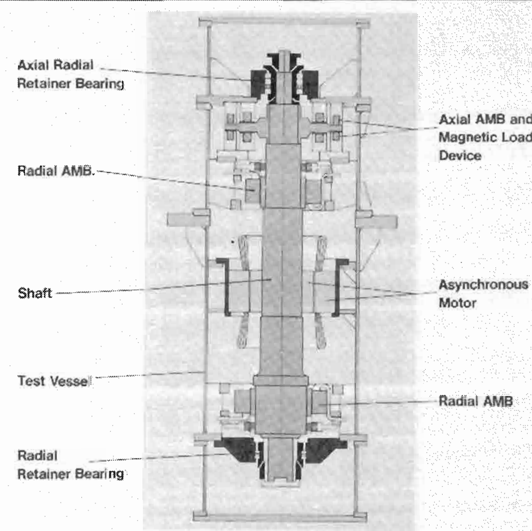
Cross Section of the NWA-Circulator for the HTR 500

Fig. 1



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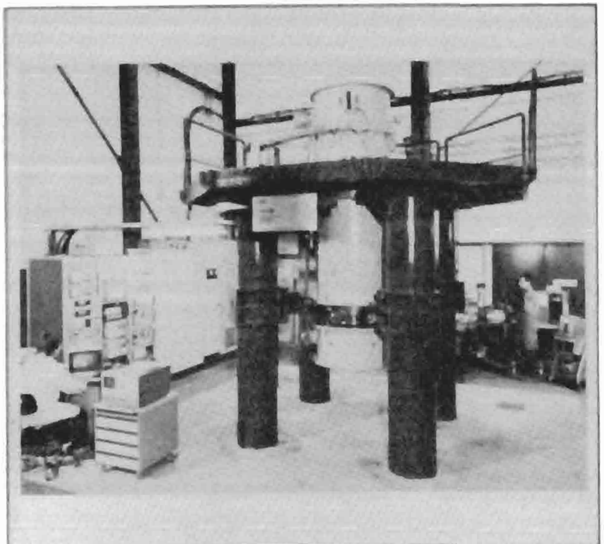
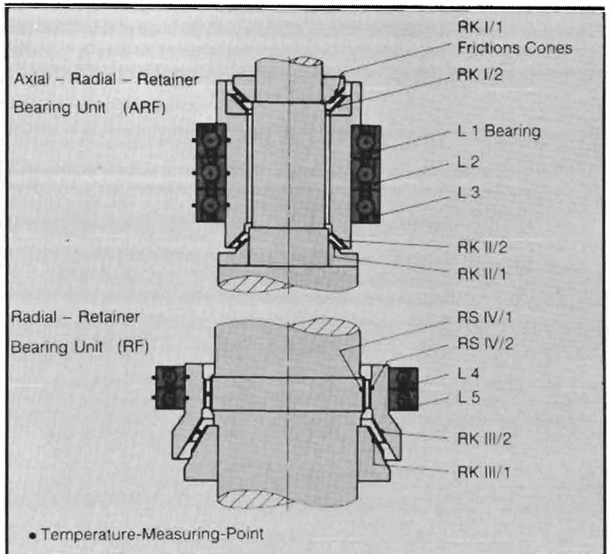
Arrangement for the Retainer Bearing Test Facility, Schematic

Fig. 2

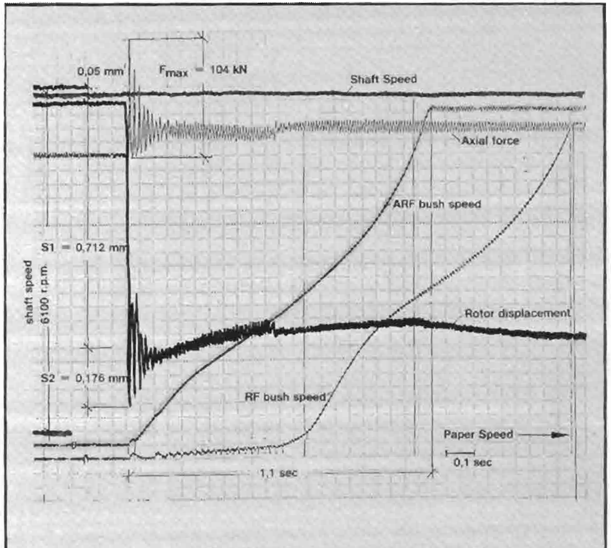


BBC — Retainer bearing test facility with active magnetic bearings — Fig. 3

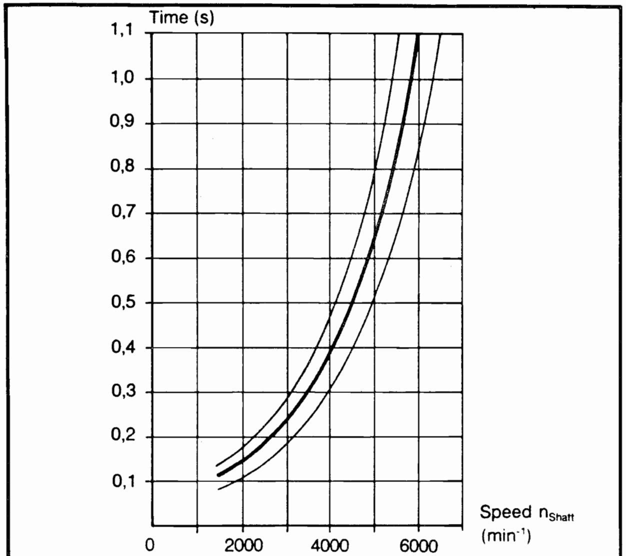


HRB — Schematic of the Retainer Bearing Elem.– Ball Bearings and Friction Cones — Fig. 4 — 88.2-59

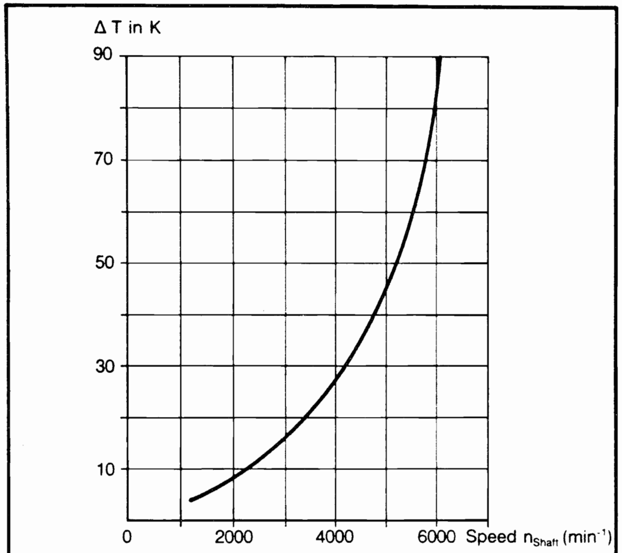
HRB | Complete Axial – Radial Retainer Bearing Unit and Ball Bearing Details | Fig. 5 | 88.2–62



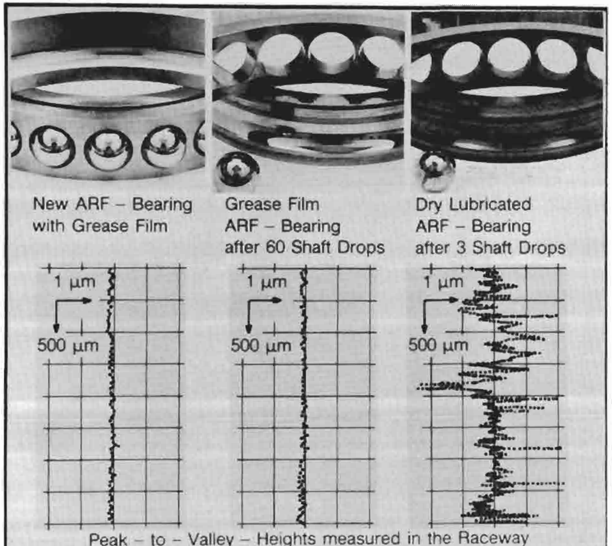
BBC BROWN BOVERI | Test Parameters at the Beginning of a Shaft Drop | Fig. 6



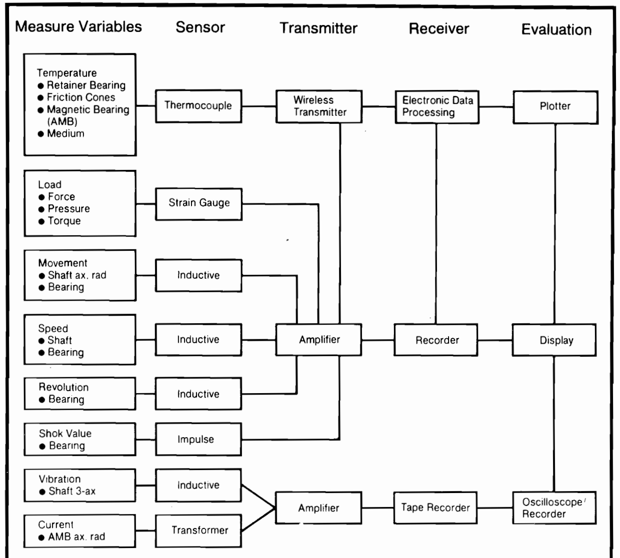
HRB | Cone Run-up Time as a Function of Initial Shaft Drop Speed | Fig. 7 | 88.2-61



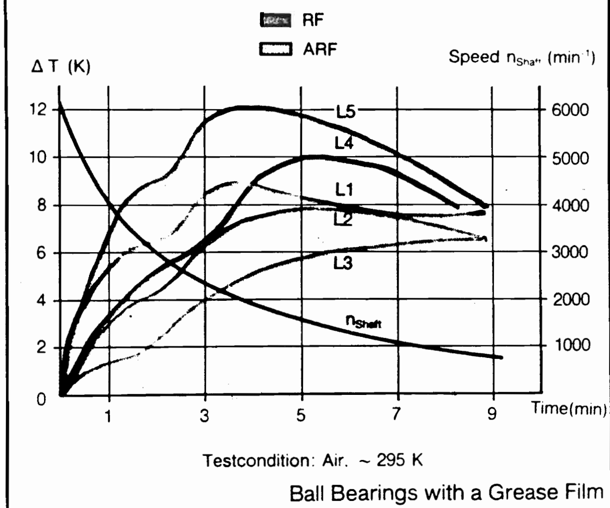
HRB | Temperature Rise in the Friction Cones as a Function of Initial Shaft Drop Speed | Fig. 8 | 88.2-65



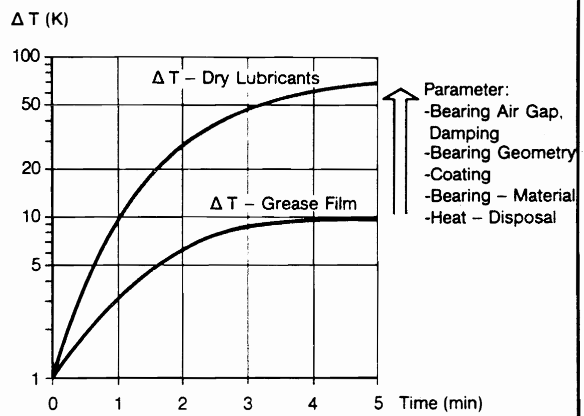
HRB | Peak-To-Valley-Heights after Various Ball Bearing Loads | Fig. 11 | 88.2–60



HRB | Retainer Bearings Test Bed Measuring Techniques | Fig. 12 | 88.2–67



HRB | Typical Temperature Rise Curves Δ T During Coastdown from ∼ 6000 r.p.m | Fig. 9 88.2-64



HRB | Parameters Affecting the Temperature Differences Δ T in Ball Bearings | Fig. 10 88.2-66

Test Bearing -Dimensions (max.)		di ~ 300 mm
Speed		$\leq$ 7.200 min^{-1}
Loads	-Pressure	300 kN
	-Tensile	100 kN
	-Moment of Torsion	4000 Nm
Kind of Loads		axial + radial
Pressure		$0 < p < 3$ bar
Temperature		333K (393K)
Medium		He, N_2, Air, Vakuum
Engine Power		Pmax = 300 kW
Shaft Mass		2000 kg
Testbox – Dimensions		Ø 1 m Height 3 m

HRB

Significant Desgin Data of the Retainer Bearings Test Bed

Fig. 13

88.2–63