Radial Shear Force Permanent Magnet Bearing System with Zero-Power Axial Control and Passive Radial Damping

Johan K. Fremerey

Institut für Grenzflächenforschung und Vakuumphysik,
KFA-Jülich GmbH, Postfach 1913, D-5170 Jülich
F.R.G.

Summary

The bearing system described was developed starting from the idea of a simple radial magnetic floatation system with axial pin stops, i.e., mechanical contacts, as devised by L.I. Mendelsohn for application in watt hour meters (General Electric Patent, 1953). The Mendelsohn system provides positive radial and, without the stops, negative axial bearing stiffnesses due to magnetic forces acting between cylindrical magnets with axial magnetization. The magnets are mounted to the rotor and stator assemblies in a face-to-face configuration for mutual attraction. A novel stabilization unit has been developed making the Mendelsohn system a non-contacting device. The unit compensates for the axial force instability and additionally provides radial damping forces. Axial stabilization is accomplished by interaction of electric coils with the end faces of two axially spaced, cylindrical permanent magnets fixed to the rotor shaft, while radial damping is provided by interaction of the rotor magnets with a copper disk projecting into the axial flux between the rotor magnets. A high-speed neutron beam chopper cascade (four units) equipped with the new bearing system is running in continuous operation at ILL-Grenoble since 1985. A turbomolecular pump prototype for industrial application was set up in 1987.

Introduction

The design of a magnetic bearing system is a challenging and enjoyable task in view of the manifold structures that can be realized by various combinations of permanent- and electromagnets, attractive and repulsive forces, radial, axial, or mixed control, coupling and transformation of orthogonal parameters, analog and digital electronics, etc. Each application demands careful analysis of the individual system requirements and optimum adaptation of the bearing parameters toward those requirements. On the other hand, transfer of a magnetic bearing device from the laboratory status into an industrial product sensibly depends on parameters such as manufacturing cost, compatibility with existing systems, specific styling, etc., which do not necessarily meet the ideas of the designer.

The magnetic bearing system described in the present paper has been developed at KFA-Jülich for high-speed rotor application in laboratory as well as in industrial environments. No active components are used for radial force control of the system, and axial control becomes active only in case of dynamic axial load. Due to the simplicity of the bearing and electronics structure, manufacturing cost is substantially lower than that of the available active radial control systems. A large potential for laboratory and industrial applications which do not require the dynamic load capacities of a multiaxis active control system may be exploited by the introduction of passive radial bearing concepts.

The shear force permanent magnet bearing concept

Magnetic forces generated by a tangential displacement of opposing magnetic poles are defined here as magnetic shear forces. In particular, the restoring force arising from a radial displacement of the rotor shaft in Figure 1 is a radial magnetic shear force, and the corresponding bearing structure will be termed a radial shear force permanent magnet (PM) bearing.

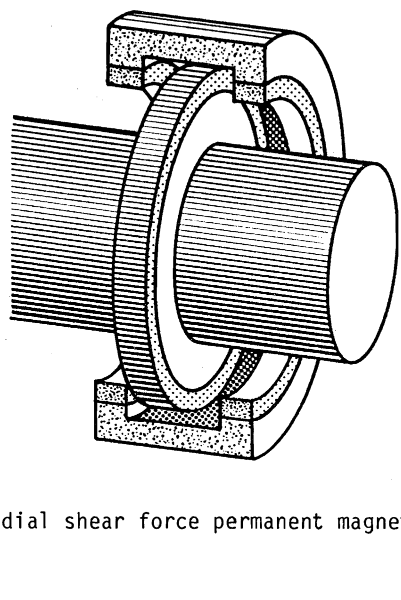

Figure 1. Radial shear force permanent magnet (PM) bearing

This type of bearing has been proposed some 35 years ago by Mendelsohn [1] for application in watt hour meters. The effect of axial unbalance of the rotor magnet between the adjacent stator magnets was eliminated by axial pin stops on either shaft end. The stops were adjusted so as to keep the rotor magnet close to the axial magnetic force equilibrium position.

By using state-of-the-art permanent magnet material such as samarium-cobalt or neodymium-iron-boron alloys, appreciable bearing forces and stiffnesses can be generated by a magnet configuration according to Figure 1. Radial load capacities of more than 100 times the magnet weight can be provided without pulling the system apart, and similar forces are available along the axial directions, still keeping the axial force equilibrium point within the axial play of the rotor magnet.

A radial shear force PM bearing for support of a high-speed rotor system can be utilized advantageously by recognizing and taking into account both the intrinsic capabilities and the weaknesses of the bearing structure. Capabilities have already been mentioned. The axial unbalance stiffness and, in particular, the poor inherent damping of the radial shear force PM bearing are considered the main drawbacks of the structure. A suitable bearing design has been achieved by introducing the asymmetric support and damping concept to be outlined in the following.

The asymmetric support and damping concept

When considering high-speed rotor application a major item of the system design is the damping mechanism for the control of precessional motions. In view of their spatial characteristics these are often termed "conical" motions. In fact, during precession the rotor spin axis moves on a double cone surface with the node point being bound to the center of mass of the rotating assembly. A radial damper will be most efficient when it is located at some distance from the node or center-of-mass point. On the other hand, the efficiency of the support bearing with the primary purpose of balancing the rotor weight will be high when this is located close to the center of mass. These considerations have guided the development of a bearing concept according to Figure 2.

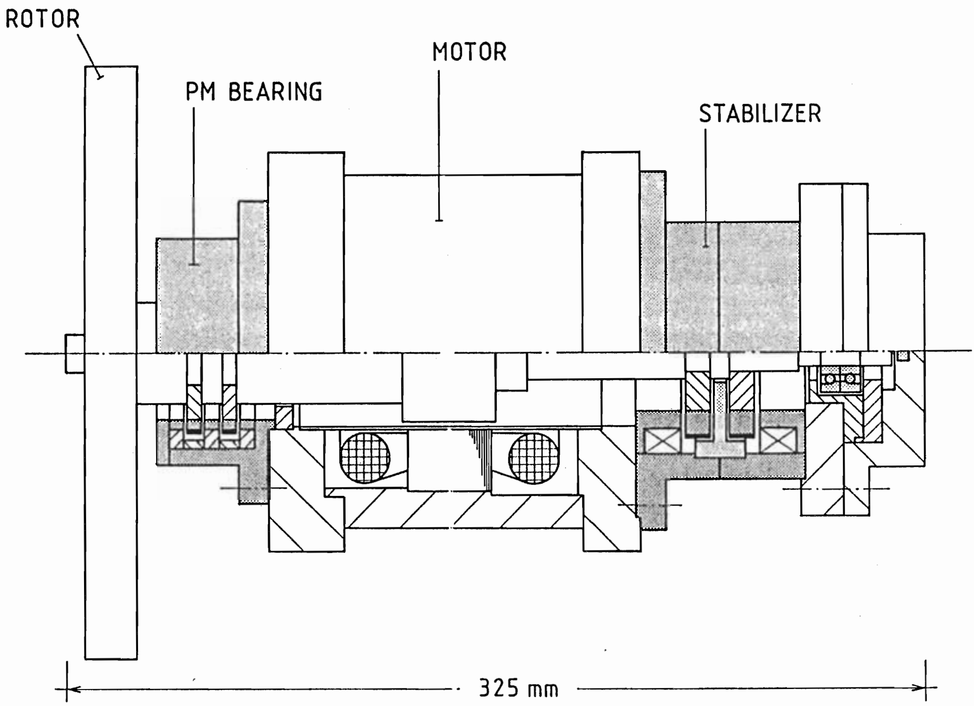

Figure 2. Passive radial bearing system with asymmetric support and damping

The main support is accomplished by a radial shear force PM bearing near the center of mass of the rotor system. Two rotating magnets are provided for higher bearing stiffness instead of one as described before.

A passive radial damping mechanism of the eddy current type is incorporated as an integral part in the stabilizer unit located near to the remote shaft end. The eddy current mechanism gets effective on radial vibration of the rotor shaft due to displacement currents generated within a massive copper disk. The disk is penetrated by the strong axial magnetic field extending between mutually attracting pole faces of rotating permanent magnets. No eddy currents are generated by the shaft rotation when the magnetic field is symmetric with respect to the spin axis. In fact, the overall eddy current losses observed on a 12-kg rotor system (Figure 3) running at 20,000 rpm was only about 3 watts including the losses generated within the support bearings and the motor.

Except for the damping effect the stabilizer maintains the non-contacting condition of the rotor system at the axial force equilibrium point of the support bearing, hence replaces the function of the mechanical pin stops as used in the original Mendelsohn concept. The axial tuning forces are generated by electrical coils acting on the damper magnets. The coils are driven by an electronic amplifier according to the output signal of an axial position sensor looking onto the end face of the rotor shaft. An automatic zero-power stabilization means drives and keeps the rotor system to the axial position where the static axial forces cancel. At that point the coil currents approach zero. The complete stabilization electronics is carried on a 70-cm^2-area printboard.

Besides the radial damping and axial control the stabilizer also serves for passive radial centering of the rotor shaft end. The centering is effected by radial shear-type restoring forces acting between the damper magnets and the ferromagnetic pole faces of the axial control magnets. The asymmetric support and damping concept provides a clear separation of the support and stabilization functions of the bearing system by locating either function in a bearing component of appropriate design. The functional separation is similar to the one applied for aircraft stabilization.

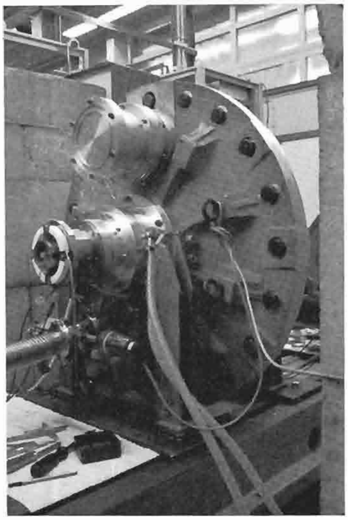

Figure 3. Neutron beam chopper prototype (1984)

Applications

The system displayed in Figure 2 was built in early 1986 as a test facility for high-speed chopper application [2] . Similar systems with support bearings on either side of the disk rotor have been realized at KFA-Jülich since 1981. Figure 3 shows a neutron beam chopper prototype under test before delivery to Institut Laue-Langevin, Grenoble, in 1984. A neutron monochromator consisting of four synchronized choppers was installed at ILL-Grenoble in 1985 for materials research. The system is operated in a 24-h service since that time.

Figure 4 shows an advanced test prototype of a 300-l/s turbomolecular pump which was developed on the basis of a cooperation project with Leybold AG. The pump is operated at a nominal speed of 51,600 rpm of the 1.5-kg turbo rotor. No adjustment is required when changing the pump axis orientation to any arbitrary direction. The bearing system has emerged by adaptation from the one outlined in Figure 2.

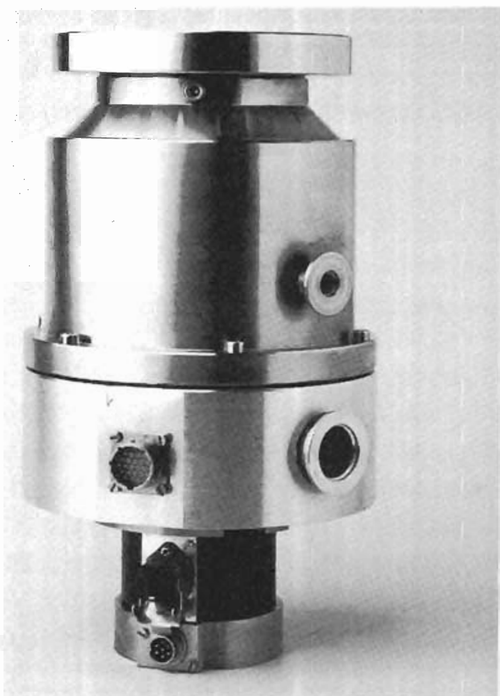

Figure 4. 300-l/s turbomolecular pump advanced prototype (photograph by courtesy of Leybold AG)

Acknowledgement

The author gratefully acknowledges continuous support of the magnetic bearing group by Professor George Comsa. As a research scientist in vacuum and surface physics he early recognized the potential benefits of the magnetic bearing technique for application in this field, and consequently incorporated the magnetic bearing activities in his institute.

References

1. US Patent 2,725,266
2. R. Hackenberg and W. Ebert (these proceedings)