# A PASSIVE MAGNETIC BEARING

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## ABSTRACT

A novel design for a fully passive magnetic bearing is presented. In the proposed structure, non-contact suspension of a rotor is provided without additional electronic components or an external energy supply. Also, the bearing does not make use of superconducting materials and therefore can operate over a wide range of temperatures, including room temperature.

Stable suspension of the rotor in the axial direction is provided by the interaction between permanent magnets and soft-magnetic elements, installed on the rotor and the stator. Consistent with Earnshaw's theorem, since this system is stable in the axial direction, it cannot also be stable in the radial directions. To overcome this radial instability, a passive radial electromagnetic suspension system is proposed, which provides radial centering of the rotor when rotating above some critical speed. The system exploits the interaction of currents induced in shorted conducting loops installed on the rotor with the axial component of a magnetic field emanating from permanent magnets installed on the stator.

This type of bearing can provide adequate stiffness and load capacity for many applications. At the same time, the lack of mechanical contact between a rotor and a stator along with lack of external energy supplies and control systems will be significant advantages compared to other magnetic bearings. To give some numerical estimates of expected bearing performance: 60N/mm theoretical estimate of a radial stiffness and 120 N radial load capacity has been obtained for a radial suspension system with a outer diameter of 28 cm and a thickness of 3.3 cm. This suspension system will support a disk-shaped 2.5-kg rotor rotating above 9000rpm.

## INTRODUCTION

Magnetic bearings are very appealing for application to high rotating speeds and/or in severe environments such as low temperatures or vacuum due to the lack of mechanical contact between a rotor and a stator. The suspension of a rotor in a magnetic bearing is achieved by using interactions of an electromagnetic nature.

It is well known that, when designing a magnetic bearing, one needs to consider an important limitation known as Earnshaw's theorem. This theorem maintains that stable non-contact levitation of a body cannot be achieved by using only the interaction between permanent magnets or between permanent magnets and soft-magnetic elements. In particular, for bearings using the interaction between permanent magnets or between permanent magnets and soft-magnetic elements to suspend a rotor, Earnshaw's theorem implies that the sum of the principal stiffnesses of the suspension is zero or less than zero. Hence, if stable suspension is achieved in the axial direction, it will be unstable in the radial direction and vice versa.

In conventional active magnetic bearings, stable suspension in all directions is achieved by introducing an external control of the magnetic field supporting the rotor. Obviously, for the magnetic field to be controllable it cannot be generated only by permanent magnets but at least partially needs to be generated by electromagnets. This is the cause of some of the drawbacks to active magnetic bearings such as continuous external energy consumption and the requirement of complicated feedback control systems.

An alternative is provided by another type of magnetic bearing which exploits the interaction of superconducting materials with an external magnetic field to achieve non-contact suspension of a rotor. Such bearings are internally stable and external controls are not needed for their operation. However, the requirement of cooling down superconductors to cryogenic temperatures restricts significantly the area of their applications.

It was shown only recently that non-contact suspension of a body using interaction with a static magnetic field can be achieved without using superconductors [1,2,3]. This is possible because Earnshaw's theorem only prohibits stable levitation of stationary bodies in a static magnetic field. First, this theorem says nothing about time-varying magnetic fields and, in fact, it is well known that non-contact levitation of a conducting object can be achieved in an AC magnetic field [4]. However, generating an AC magnetic field always requires external energy consumption and such bearings are significantly less efficient than conventional active magnetic bearings.

Second, the theorem does not apply to rotating bodies. It was shown recently, that stable levitation of a rotating body can be achieved in a static magnetic field.

One way to do this is to utilize gyroscopic torques acting on a rotating body to overcome the destabilizing effects due to magnetic interaction [1].

Another approach is to include some conducting elements on the rotor and design the system in such a way that, when the rotor rotates, the conducting elements will experience a time varying magnetic field and, when interacting with it, will stabilize the system [2]. The present work attempts to show that the second approach may lead to designs with the performance level required by real applications.

## BASIC DESIGN CONCEPT

To understand the basic design concept, consider a planar rectangular conducting loop as shown in Figure 1. One side of the loop is exposed to magnetic field $B_1$ normal to the loop plane, and the opposite side is exposed to an oppositely directed magnetic field $B_2$.

Assume for the moment that there is no loop resistance and there is a loop position where no current flows in the loop. (A similar structure has been fruitfully employed in the design of superconducting bearings [5,6], where the loop resistance is zero.) If the loop is shifted from this position, a current will be induced in the loop with a magnitude and direction that exactly cancels the change of the magnetic flux through the loop interior.

If we assume for simplicity that, in Figure 1, $B_1$=$B_2$=B, then change of the external magnetic flux through the loop interior divided by the loop length in the y-direction will be $/l = 2 \times B \times X$, where $X$ is the loop displacement in the x-direction. The current needed to cancel this change of flux is

$$I = \frac{/l}{L/l} \qquad (1)$$

in which L is loop inductance. Assuming that size of the loop shown in Figure 1 is much bigger in the y-direction than in x-direction, the ratio *L/l* can be considered to be the inductance of unit length of the loop. Practical values of this inductance are about $10^{-6}$ H/m. For NdFeB magnets, the value of *B* corresponding to the most efficient use of the magnet energy (maximal ratio of magnetic energy in the air gap *vs* volume of magnetic material) is about 0.6 Tesla. Thus, if the conducting loop is displaced by 1 mm, the induced current is

$$I = \frac{2 \times 0.6T}{10^{\ 6} Henry/m} \times 10^{\ 3} m = 1200A \qquad (2)$$

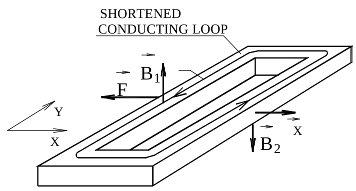


**FIGURE 1:** Schematic for the explanation of the design motivation

The lateral force acting on the loop over 10cm of its length will be

$$F = 2 \times 0.6T \times 1200A \times 0.1m = 144N \qquad (3)$$

in a direction which opposes the displacement.

If the loop resistance were zero (as for superconducting loops), there would be a restoring force without any external energy consumption. With finite coil resistance, the current in the loop and the restoring force will decay soon after loop stops its motion.

The key idea in the design of a fully passive magnetic bearing is to use the rotation of the rotor to continually excite these currents. The design places multiple conducting loops on the rotor. The stationary magnetic field is configured so that, whenever the rotor axis is displaced from the desired radial location, the rotation will produce a periodic variation in the flux through the rotating loops and corresponding periodic electromotive forces. The electromotive forces, in turn, will induce periodic currents which, interacting with magnetic field, will cause restoring forces acting on the rotor.

Energy losses due to the loop resistance in such a system are compensated by the work produced by the motor keeping the rotor rotating or are subtracted from the kinetic energy stored in the rotor itself. Because there is a direct energy transformation, this system can be expected to be very efficient.

Moreover, the proposed passive radial suspension system may be integrated into a 6-degree-of-freedom passive magnetic bearing, in which there will be almost no energy consumption at all as long as only an axial load is applied. This feature seems to be especially beneficial for stationary applications where the requirement of purely axial load is easy to satisfy.

## PRIOR DESIGNS

Probably the first attempt to design this type of passive magnetic bearing is reflected in [2]. The bearing presented there uses closed conductive loops moving through a series of alternating magnetic fields when the rotor rotates with respect to the stator. A drawback to this design is that the use of the permanent magnets and

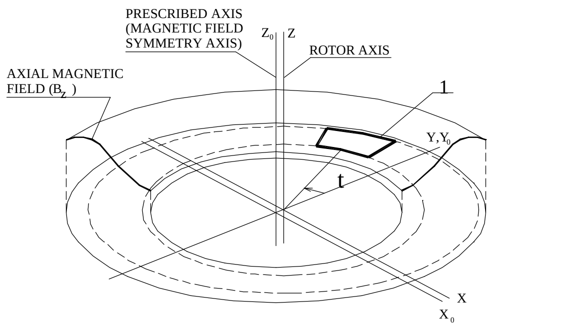


**FIGURE 2:** General schematic explaining operation principle of the radial suspension system.

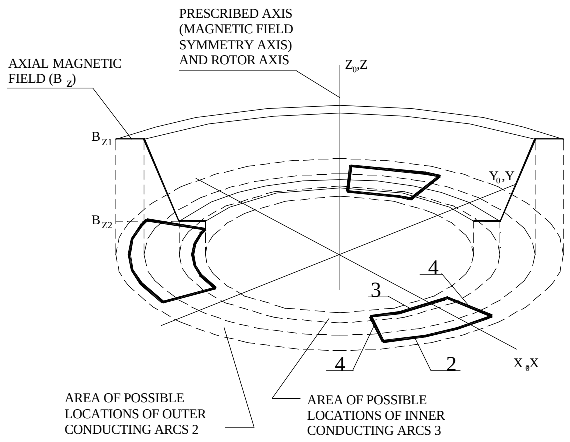


**FIGURE 3:** Operating principle of the radial bearing, desired shapes of magnetic field and conducting loops.

of the current carrying capacity of the conductors is somewhat inefficient. Further, the proposed bearing is extremely sensitive to variations in the geometry of the conducting loops and magnetic fields. If such variations occurred due to manufacturing inaccuracies or due to rotor expansion because of centrifugal forces, unintended currents would appear which would cause undesired torques and heating. Finally, the reaction forces produced by this bearing when subjected to a constant external load are pulsatile. These pulses may excite structural resonances, causing damage to the bearing or the rotor.
The present work attempts to address these shortcomings directly. The design makes more efficient use of permanent magnet energy and current-carrying capacity of conductors, is less sensitive to manufacturing accuracy and centrifugal growth, and generates a constant restoring force in reaction to a constant applied load. This bearing consists of a passive radial suspension system and a passive axial suspension system.

## RADIAL SUSPENSION SYSTEM

The radial suspension system provides centering of the disk-shaped rotor in the radial direction. The bearing becomes functional when the rotor rotates above some critical speed while exerting almost no force on the rotor in the axial direction.
The operation principle of the radial suspension system is explained in Figures 2 and 3. Figure 2 shows a shortened conducting loop 1 mounted on the disk-shaped rotor and exposed to a constant axial magnetic field $B_z$ generated by permanent magnets installed on the stator. The magnetic field is required to be circumferentially uniform about the prescribed axis $Z_0$ and non-uniform in the radial direction.
Coordinate frame $X_0Y_0Z_0$ is fixed to the stator. The rotor axis is designated as Z. The coordinate frame XYZ is chosen so that if rotor axis Z coincides with prescribed axis $Z_0$ and the disk is in the axial equilibrium, then axis X coincides with axis $X_0$ and axis Y coincides with axis $Y_0$.

If the rotor axis Z coincides with prescribed axis $Z_0$, then the magnetic field at every point of the loop and the magnetic flux through the loop interior will be constant. This is true whether the rotor rotates about its axis Z or not, because magnetic field is uniform circumferentially about axis $Z_0$. If this condition is satisfied, there will be no current flowing in the loop, no force and no torque acting on the rotor. This is true for any shape of the loop. No special requirements are imposed on the loop manufacturing accuracy. Radial expansion due to centrifugal forces will not cause undesired currents flowing in the loops as long as rotor axis Z coincides with prescribed axis $Z_0$.
When rotor axis Z is shifted from prescribed axis $Z_0$, the loop currents will no longer be zero. For example, Figure 3 shows the rotor being shifted in the radial Y direction. The loop path during the rotor rotation about its axis Z is represented by dashed lines. In this case, the magnetic flux through the loop interior will obviously vary as the rotor rotates, because the radial distribution of the axial magnetic field is required to be non-uniform. Therefore, a current will be induced in loop 1, which will interact with the magnetic field and cause a force exerted on the rotor.
The specific geometry of this design is presented in Figure 3. Each loop is formed by outer arc 2, inner arc 3 and two radial legs 4. The arcs are concentric with the rotor axis Z. The radial distribution of axial magnetic field is chosen so that the outer arc 2 is exposed to a virtually constant magnetic field $B_{z1}$, the inner arc 3 is exposed to virtually constant magnetic field $B_{z2}$, and these magnetic fields are different. These conditions should hold over the entire range of the rotor radial displacements. In this case, if rotor axis Z is shifted from prescribed axis $Z_0$, the current induced in the loop will be given by

$$I= \quad I_0 \cos(wt+q) \quad : \quad q = \tan^{1}\frac{R}{Lw} \qquad (4)$$

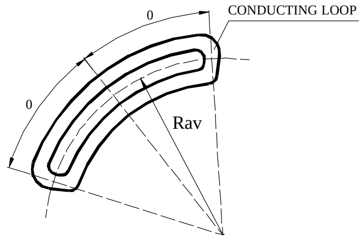


FIGURE 4: Geometrical parameters of conducting loops.

where $I_0$ is the current amplitude, $w$ is the rotor rotation frequency, R and L are the loop resistance and inductance respectively.
The force acting on the loop resolved on the displacement direction (Y direction) can be presented as:

$$F_y^1 = F_0^1(\cos q + \cos 2wt \cos q \quad \sin 2wt \sin q)$$

where

$$F_0^1 = \frac{(4BR_{av} \sin j_0)^2 r}{L\sqrt{1+(\tan q)^2}}$$

$_0$ is an angle characterizing the loop size and $R_{av}$ is average loop radius (see Figure 4) . Note that this force is proportional to $r$, the radial displacement.
The force acting on the loop resolved in the direction perpendicular to the displacement direction (X direction) can be presented as:

$$F_x^1 = F_0^1(\sin q \quad \sin 2wt \cos q \quad \cos 2wt \sin q)$$

However, if we install at least three loops as above, situated evenly around rotor axis Z, the oscillating force components will be cancelled and the net force will be constant in time and given by:

$$F_y = F_0 \cos q \,, \; F_x = F_0 \sin q$$

where $F_0 = n \times F_0^1$ and n is number of the loops. Noting that the force is proportional to the radial displacement, define a radial stiffness as

$$K = n \times \frac{(4BR_{av} \sin j_0)^2}{2 \times L\sqrt{1+(\tan q)^2}} \qquad (5)$$

If the magnetic field gradients above the conducting arcs 2 and 3 kept small, variations of the field in the conducting arcs due to rotor rotation will be small. This minimizes any eddy currents induced in the arcs and drag torque caused by the interaction of these currents with the magnetic field. Special measures must be taken to reduce eddy currents in legs 4 such as laminating the leg conductors with insulating interlayers.

## STABILITY

The fact that the restoring force component $F_y$ pushes the rotor towards the equilibrium position suggests that equilibrium may be stable. The equilibrium position can be made asymptotically stable by introducing some damping into the system. The required damping depends strongly on the angle $q$ between the direction of the restoring force and the rotor displacement.

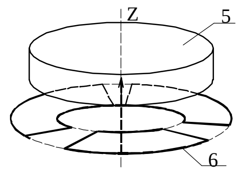


**FIGURE 5:** Operation principle of a possible damping system.

Assuming that there are no external forces acting on the rotor, the equations of the rotor motion in polar coordinates $r$ and $j$ can be written as follows:

$$m \times \ddot{r} = f_0 \times \cos q \times r \quad d_r \times \dot{r} + m \times \;^2 \times r \qquad (6)$$

$$m \times \dot{} = f_0 \times \sin q \times r \quad d_t \times \; \times r \quad 2 \; m \times \dot{r} \times \qquad (7)$$

The parameter $m$ is the rotor mass, $= \dot{j}$ is the rotation speed of the rotor mass center about the bearing axis, $d_r$ and $d_t$ are radial and circumferential damping coefficients. Note that the rotation speed of the rotor mass center about the bearing axis is unrelated to the rotor rotation speed about its axis $w$.
The system behavior is intuitively clear in some limiting cases. For example, if $q$ = 0, equations (6) and (7) become

$$m\ddot{r} = \quad Kr \quad d_r\dot{r} + m \;^2 r \qquad (8)$$

$$mr \dot{} = \quad d_t \quad r \quad 2m\dot{r} \qquad (9)$$

The second equation is automatically satisfied with 0, in which case (8) becomes $m\ddot{r} = \quad Kr \quad d_r\dot{r}$: damped radial oscillations. We may expect the origin to be stable in this case.
If = /2, equations (6) and (7) become

$$m\ddot{r} = \quad d_r\dot{r} + m \;^2 r \qquad (10)$$

$$mr \dot{} = \quad f_0 r \quad d_t \quad r \quad 2m\dot{r} \qquad (11)$$

If 0, then regardless of the sign of , r will eventually go to infinity because there is no force in the first equation able to counteract the centrifugal force $m \;^2 r$. Therefore, the origin is unstable.
In any case, to ensure assymptotic stability of the origin, some damping is required. Assuming that $d_r = d_t = d$, then it can be shown that the value of damping coefficient $d$ required for the system to be stable is

$$d > \sqrt{m \times K} \times \frac{\sin q}{\sqrt{\cos q}} \qquad (12)$$

This bound goes to infinity when $q$ approaches /2 and approaches zero when $q$ approaches 0.
Recognizing that $K$ and $q$ are both functions of $w$, (12) implies that, for a given damping coefficient, there exists a value of $w$ above which the rotor radial equilibrium is stable.

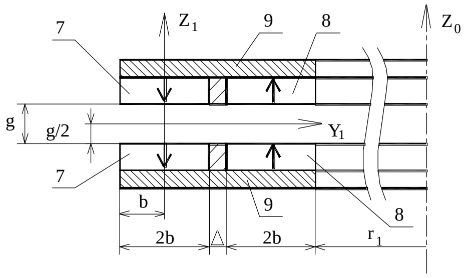


**FIGURE 6:** Possible structure to generate magnetic field in the radial bearing

## DAMPER DESIGN

A natural way to provide this damping is to employ a system similar to the one used for the radial positioning of the rotor except that the coils are now stationary and the magnets now rotate. A schematic of the damping system is shown in Figure 5. It includes a disk-shaped permanent magnet 5 mounted on the rotor concentrically with the rotor axis Z and at least three shorted conducting loops 6 mounted on the stator and exposed to the magnetic field emanating from magnet 5. If this magnetic field is circumferentially uniform about axis Z, rotation of the rotor about its axis will not cause any effect. However lateral motions of the rotor in the radial direction will result in currents induced in conducting loops and a damping force exerted on the rotor opposed to its velocity vector.

## STATOR DESIGN

In the system shown in Figure 3, the requirement of low magnetic field gradients is automatically satisfied in the areas adjacent to the extremes of the magnetic field distribution. Thus, an arrangement where current-carrying arcs 2 and 3 are located in the areas of magnetic field maximum and minimum, which are desired to be of opposite signs, seems to be especially advantageous, because it provides the maximal value of the Lorenz force acting on the conducting loops.

An example of a structure to generate a magnetic field satisfying these requirements is shown in Figure 6. It consists of two identical parts spaced axially, thus forming a magnetic gap between them in which the rotor and its conducting loops will be situated. Each part consists of an outer permanent magnet 7, an inner permanent magnet 8 and a soft magnetic disc 9 which provides a return path for the magnetic flux.

As a numerical example, consider the structure shown in Figure 6 with $r_1$=23cm, b=1.1cm, g=0.9cm, and =0.5cm. If NdFeB magnets are used, an 0.6 Tesla magnetic field can be generated in the magnetic gap with a magnet thickness of about 5mm [8]. The thickness of the components of the magnetic yoke 9 required to carry magnetic flux without saturation will be 7mm. Therefore, the total thickness of the assembly will be 3.3cm.

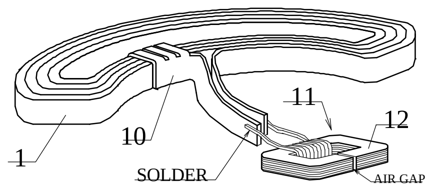


**FIGURE 7:** Multi-turn loop with additional inductance

Use four copper loops with cross section 5x11mm, $R_{av}$=14cm and $_0$=35°; the inductance and resistance of each loop will be about 0.22 H and 1.2×$10^{-4}$ Ohms.

In accordance with equation (5) the radial suspension stiffness $K$ at a rotation speed of 9000 RPM will be $K$=130N/mm. The smallest damping coefficient for the suspension to be stable as calculated using equation (12) with a rotor mass of 2.5kg is $d_{min}$=440N×s/m. This level of damping would be quite difficult to achieve in a passive system.

## A SIMPLE IMPROVEMENT

The source of this problem is the value of angle *q*, which is about 30°. This angle yields a high destabilizing circumferential component of the electromagnetic force. There is, however, a method to reduce amount of damping required for the system stability.

An inductor can be inserted in series with each rotor loop, as indicated in Figure 7. (Note that the loop shown is constructed of a concentric winding of multiple layers of thin wire. It is easily shown that such a multilayer coil behaves the same as a solid coil provided that the ends of the winding are connected together.) If this additional inductance includes a magnetic yoke 12, then overall ratio R/L can be reduced dramatically. An air gap in the yoke is required to avoid saturation of the yoke material. In addition, the yoke must be located far away from the stator magnetic field to prevent additional forces and torques. The bridge 10 in Figure 7 connects the interior wire end to the inductance 11.

An experimental loop was constructed with this structure using insulated flat ribbon wire of 0.5x5mm$^2$ cross-section (supplied by MWS Wire Industries). The loop includes 17 turns of wire. The added inductor consists of a magnetic yoke assembled of 0.35mm thick transformer steel laminations with total cross-section area of 5x9.5mm$^2$ and an air gap of about 0.2mm. The yoke is wound with 36 turns of 0.5x10mm$^2$ flat ribbon wire. The overall resistance and inductance of the assembly were measured to be 0.068 Ohm and 0.32 mH at 60Hz. The saturation current of the magnetic circuit is estimated to be 11A. This corresponds to the maximal net current flowing through the loop cross-section of 11A×17turns=187A. If the circuit resistance and inductance remain the same at 150Hz (9000 RPM), then the

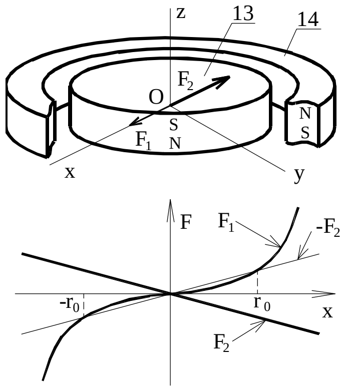


**FIGURE 8:** Schematic of an axial bearing.

suspension parameters will be: 13°, *K* = 60 N/mm, and $d_{min}$ = 88 Ns/m.

The design of the required damper can be examined by assuming that the damper has the same structure as the positioning system. The difference is that the conductive coils are mounted on the stator while a magnetic system as shown in Figure 6 is mounted on the rotor. Further, assume that all the sizes are scaled down by a factor of 2. Therefore, for each loop the resistance and average radius are $R$ = 0.24 m and $R_{av}$ = 7 cm. Assuming that the ratio *L/R* (*L* is the damping loop inductance) is much less than the period of the rotor natural oscillations, this system acts as a damper with damping coefficient

$$d = 2 \times \frac{(4\ B \times R_{av} \times \sin j_0)^2}{R} = 77 N \times s / m .$$

This value can be increased by thickening the magnets to 5mm, in which case the field density increases to about 0.8 Tesla and $d \quad 137 N \times s / m$.

## AXIAL SUSPENSION SYSTEM

One potential axial suspension is indicated in Figure 8 [9]. This consists of two permanent magnets, one of which (13) is shaped as a disc and enclosed within the other ring-shaped magnet 14. The magnetization directions of the magnets are opposite each other.

This arrangement is stable with respect to axial displacements and angular deflections about radial axes, but unstable with respect to radial displacements. Consequently, if magnet 13 is displaced from the central position then the resulting radial force $F_1$ will be in the same direction, thus moving the magnet 13 further away from the central position. To make the system stable in the radial direction we need to apply a radial force $F_2$, which would have negative radial gradient. In this case, as long as absolute value of the force $F_2$ is bigger than the absolute value of the force $F_1$ (as in region ($-r_0$;$r_0$) in Figure 8), then the net force acts toward the center. In the present work, force $F_2$ is provided by the proposed passive electromagnetic radial suspension system. If the absolute value of the radial gradient of the force $F_2$ is bigger than the absolute value of the radial gradient of the force $F_1$ within the region ($-r_0$;$r_0$), this condition will be sufficient for the system stability within this region.

With modern magnetic materials such as NdFeB, high values of bearing stiffness can be achieved easily. For example, assume that 10-mm thick NdFeB magnets are used. Let the diameter of the inner magnet be 34mm, the inner diameter of the outer magnet is 40mm, and the outer diameter is 60mm. With this, the positive axial stiffness of the assembly will be about 40N/mm, with radial destabilizing negative stiffness of about 20N/mm.

## CONCLUSION

The proposed structure of a passive magnetic bearing is shown to be able to provide non-contact stable suspension of a rotor rotating above certain critical speed. Among other parameters, the value of the critical speed depends strongly on the ratio between inductance and resistance of conducting loops and amount of damping in the system. Numerical estimates of the expected bearing characteristics show that they may be found acceptable for many applications. Lack of external control systems and power supply makes it an attractive alternative to active magnetic bearings for applications not requiring high accuracy of the rotor positioning. Essentially zero rotation resistance when only an axial load is applied is especially beneficial for stationary applications, where this condition is easy to satisfy.

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