DEVELOPMENT AND TESTING OF A FOUR POLE MAGNETIC BEARING

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SUMMARY

A 4 pole radial active magnetic bearing actuator has been developed, constructed, and tested. Among the anticipated characteristics of this configuration were: higher specific load capacity, lower power losses, and improved stability as compared to other heteropolar configurations. The theoretical basis of each of these aspects will be investigated, and some limitations of the design will be discussed. Testing of the new actuator verified that higher load capacities are achievable. Stability problems with the prototype, due to interaction with the power amplifiers, limited the ability of the testing to investigate actuator stability improvements.

INTRODUCTION

One of the major limitations of active magnetic bearing (AMB) systems is their relatively low specific load capacities. Specific load capacity, P, is generally defined as the load capacity of the bearing, F, divided by the bearing's projected area (rotor length, L, times rotor diameter, D).

$$P = \frac{F}{D*L} \tag{1}$$

While fluid film bearings typically have specific load capacities of more than 1.7 MN/mm^2 (250 psi), AMB's are more typically limited to 0.35 - 0.7 MN/mm^2 (50 - 100 psi). This limitation is due to two factors. First is the limitation of the flux carrying capacity of the magnetic material. The second factor is the efficiency with which the magnetic material can be used.

In the general case, the load capacity of an AMB in-line with an axis can be given by

$$F = \frac{B}{2\mu_o}(W_{pt}L) \tag{2}$$

where B is the nominal flux density in the magnetic gaps, W_{pt} is the total projected widths of the gap in the quadrant, and L is the length of the stator stack. An example geometry for an 8 pole bearing illustrating these concepts is shown in Figure 1.

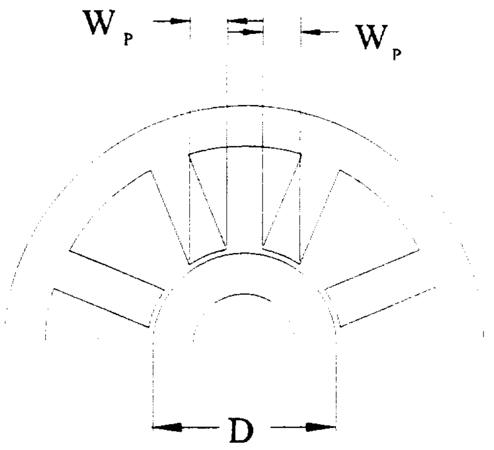

Figure 1: **Eight pole bearing projected area.**

By introducing a term for the bearing geometric efficiency, ξ, we can develop the following equations for this example:

$$\xi = \frac{2W_p}{D} \tag{3}$$

$$F = \xi \frac{B^2}{2\mu_0} DL \tag{4}$$

$$P = \xi \frac{B^2}{2\mu_0} \tag{5}$$

Equation 5 indicates clearly that the specific load capacity of an AMB is determined completely by the saturation flux density of the material and by the geometric efficiency of the bearing. Once the material is chosen, the saturation flux density is fixed. Selecting high flux materials (e.g. nickel and vanadium alloys) is usually associated with great increases in material costs. It is of interest, therefore, to determine ways to increase the specific load capacity, ξ.

From the example of figure 1 and equation 3, it is easy to see that the maximum value of geometric efficiency for a symmetric AMB is 0.707. Experience has shown that most real systems have values between 0.2 and 0.5. This limitation is due primarily to the space required for coils.

DEVELOPMENT OF A FOUR POLE BEARING

Again from figure 1, it can be seen that the slots between pole pieces, which are required for the placement of the coils, detracts from the geometric efficiency of the bearing. The wider the coil slots are, and the more coil slots there are, the less projected area is available for magnetic material adjacent the rotor. There are two basic ways to minimize the projected area required for the coils: the first is to minimize the coil width; and the second is to minimize the number of coils. Minimizing the coil width, if all other design parameters are held constant, results in a long thin coil and long pole pieces. This option is not always feasible due to space constraints and the increased flux leakage and dynamic problems associated with long pole pieces. The other alternative is to minimize the number of pole pieces.

Some work has been done (e.g. Grbesa [1]) that indicates that a minimum of three pole pieces is required for an AMB. This three pole configuration requires significant modifications to the control algorithms, and is not amenable to fault tolerant operation. It was postulated early in this project that a four pole bearing might not have either of these drawbacks, and would still allow a significant improvement to geometric efficiency. For this reason it was decided to investigate the performance characteristics of a four pole bearing. The basic geometry of this configuration is shown in figure 2. From this figure it is seen that one of the primary characteristics of this design is that the flux produced by one quadrant (i.e. the top pole) returns to

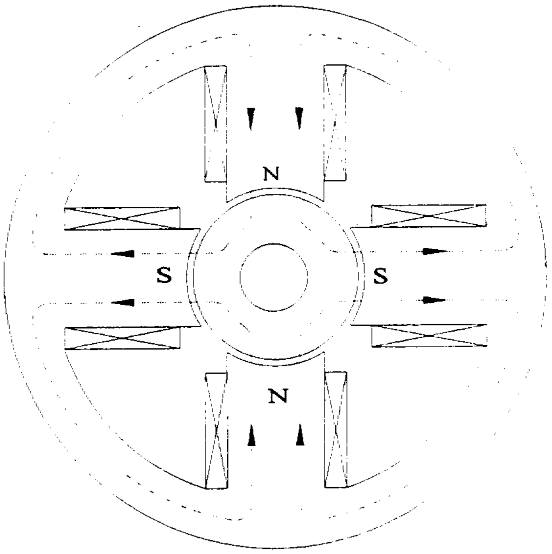

Figure 2: Basic four pole layout..

the stator through the two poles in the perpendicular axis (i.e. the horizontal axis). Another characteristic is that, assuming the total flux produced by the each axis remains constant, the control flux from the vertical axis will supply the bias flux to the horizontal axis, and vice versa. This is an important assumption, and in practice limits the performance of this configuration. This will be discussed further in the experimental results section.

Another characteristic of the bearing that can be developed from inspection is that the rotating power losses for this configuration should be lower than those of a corresponding eight pole bearing. Because the flux is split in the rotor, the rotor radial dimension needs to be only one half of the pole width. The configuration shown will, therefore, have approximately the same rotor volume as a corresponding eight pole bearing. There are, however, only half as many flux variations. Kasarda [2] has shown that the number of poles has a strong influence on rotor losses, while the orientation of the poles (NSNS vs NNSS) has a weak influence. The rotor losses for this configuration, therefore, may be comparable to a four pole homopolar bearing. This characteristic needs to be investigated experimentally.

The four pole bearing should also have a significantly different open loop stiffness behavior. In a standard heteropolar bearing, as the rotor is displaced towards the poles of one quadrant, all of the gaps in that quadrant decrease. Because the open loop stiffness, K_x, is inversely proportional to the sum of those gaps, its magnitude increases dramatically. This effect is due to the fact that the reluctance of the flux loops in the quadrant goes to zero as the gaps close (neglecting stator reluctance for illustrative purposes). In the four bearing, however, the flux loops all contain two perpendicular gaps. A vertical displacement, therefore, will close only one gap of the flux path, resulting in a smaller increase in the open loop stiffness magnitude. This

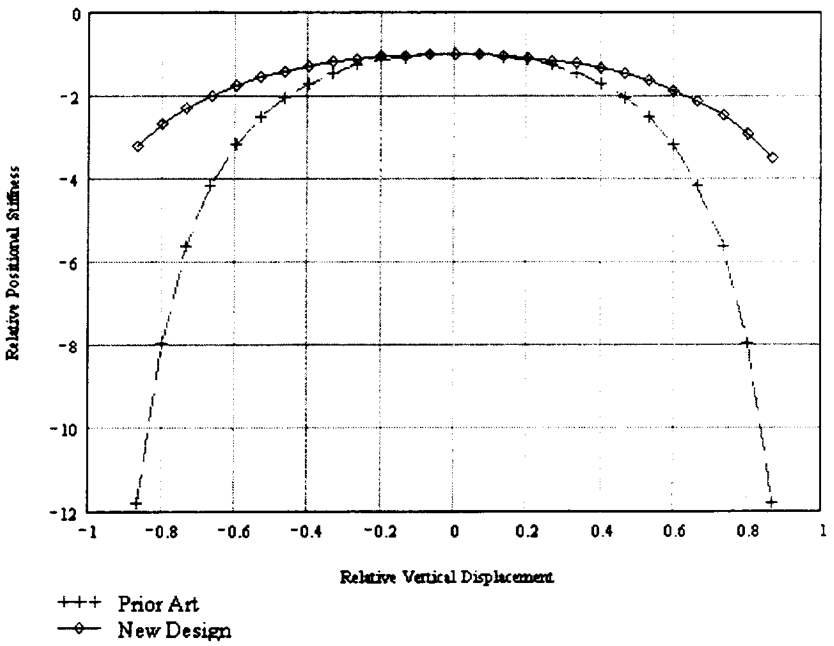

Figure 3: Comparison of open loop stiffness of 4 and 8 pole bearings

effect is illustrated in figure 3, which compares the normalized open loop stiffness of a four pole and an 8 pole bearing of the same basic size and load capacity.

Several things are evident from this plot. First is that the negative stiffness of a standard bearing increases dramatically with displacement. This increase, which must be compensated by the current stiffness and gain of the bearing, deceases the available stiffness of the bearing under large displacements. This contributes to bearing instability. Four pole bearings, however, should be tolerant of greater relative displacements without suffering adverse consequences, and should have a more uniform closed loop stiffness over the range of displacements.

Alternatively, the relative open loop stiffness of a four pole bearing at 50% relative displacement is comparable to that of an eight pole bearing at 33% relative displacement. If other design factors allow, this would allow a four pole bearing to operate with smaller gaps (e.g back-up bearing clearance of 66% magnetic gap, rather than 50% magnetic gap) than a similar eight pole bearing. This would in turn allow a reduced coil size and a further increase in geometric efficiency and specific load capacity. These benefits were deemed to be significant enough that a patent application for the technology was submitted [4]

One limitation that is evident in the four pole bearing is a limitation on it's size. Because the rotor radial dimension must be at least as wide as half of the pole width, larger bearings become impractical. For example, a bearing with a 150 cm pole width would require a 75 cm thick radial rotor lamination and a 75 cm thick backiron section. This geometry would result in a very large rotor diameter, and would probably be better served by a 12 or 16 pole bearing (38 cm or 25 cm thick rotor respectively). This factor should be addressed on a case by case design basis.

EXPERIMENTAL VERIFICATION

A prototype of a four pole bearing was developed for a particular application. Due to outside limitations on the allowable, rotor and the stator diameters complete design freedom was not available. This makes the comparison more representative of real world applications. An eight pole bearing was already in use in this application, and was available for comparison. The relevant design parameters for both bearings are shown in table 1.

It is noted that, while the geometric efficiency of the four pole bearing does not approach the maximum theoretical value (70.7%), it represents a 40% improvement over that achievable with the 8 pole bearing in this specific design. This translates directly into a 26% increase in theoretical load capacity, on a reduced rotor diameter. Testing indicated that the actual increase in load capacity was even greater than predicted. This is potentially due to a reduction in fringing effects due to the larger pole sizes.

PARAMETER	4 Pole	8 Pole
Stator OD (cm)	7.160	7.160
Stator ID (cm)	3.206	3.506
Gap (cm)	0.038	0.038
Rotor OD (cm)	3.130	3.430
Rotor ID (cm)	1.905	2.327
Stack Length (cm)	1.270	1.270
Coil Turns	120	120
Wire Gauge (AWG)	23	23
Bias Current (Amps)	3.3	3.3
Geometric Efficiency (%)	39.1	28.0
Nominal Load Capacity (N)	92.5	73.4
Experimental Load Capacity (N)	111.2	62.3

Table 1. Comparison Data

Some problems were encountered in the testing of this bearing. It was noted that the 4 pole bearing was significantly more "noisy" (audible noise) than the comparable eight pole bearing, and that there was an instability mode in which the bearing position would "jump" approximately 10 microns, and then drift back to center. This effect was predicted by Meeker [3], in bearings which utilize a separate power amplifier (PA) to drive each coil. The effect is due to a singularity in the inductance matrix and is excited by the switching nature of the power amplifier. Due to the randomness of the timing of the PA switching, there exist states in which currents in all of the coils are increasing or decreasing at the same time. Because of the series nature of the flux paths, this leads to a change in current with no corresponding change in flux density (inductance matrix singularity). To the PA this represents a temporary zero inductance, leading to errors in current output. As indicated in [3], a decoupling ring placed in series with the bearing coils eliminated the instability and significantly reduced the bearing noise.

As stated earlier, one of the assumptions in the original bearing configuration was that the total flux in the bearing would remain constant. Clearly, the problems associated with the switching PA indicate that the configuration in which each coil is driven independently by this type of amplifier violates this assumption. Several potential solutions to this problem exist. One is the use of linear amplifiers in conjunction with four pole bearings. Another is the use of separate bias and control coils, with the control coils for each axis connected in series and driven with a bi-polar amplifier. A third potential solution is the inclusion of permanent magnet biasing in conjunction with the series bi-directional control coils. These are subjects of future work.

CONCLUSION

The characteristics of a four pole AMB with a unique winding configuration have been developed. Theoretically, this type of bearing should offer a significant increase in specific load capacity, reduced rotor power losses, and improved stability characteristics. Experimental work has verified the increase in specific load capacity. Because of the specific coil configuration developed in the prototype, the bearing was susceptible to instabilities caused by an interaction with the coils and the switching power amplifiers. This instability resulted in an increase in bearing audible noise and a tendency for the bearing rotor to "jump." Potential solutions to this problem have been identified. One, the inclusion of a decoupling ring as suggested by Meeker [3], **resulted** in improved bearing performance. Other potential solutions, which do not require additional hardware, will be investigated in future research.

REFERENCES

1) Grbesa, B., "Low Loss and Low Cost Active Radial Homopolar Magnetic Bearing," 6th International Symposium on Magnetic Bearings, Boston, MA, 1998.

2) Kasarda, M.E.F., The Measurement and Characterization of Power Losses in High Speed Magnetic Bearings, Doctoral Dissertation, University of Virginia, 1997.

3) Meeker, D.C., Optimal Solutions to the Inverse Problem in Quadratic Magnetic Actuators, Doctoral Dissertation, University of Virginia, 1996.

4) Imlach, J., "High Specific Load Capacity Radial Magnetic Bearing Actuator," U.S. Patent #5,962,940, October 5, 1999.