Comparative Study of Axial/Radial Magnetic Bearing Arrangements for Turbocompressor Applications

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***Abstract*—This article presents results of a comparative study of different radial/axial actuator solutions for an Active Magnetic Bearing (AMB) system in a real-life turbocompressor application.**

I. INTRODUCTION

Electromagnetic actuators are the largest components of Active Magnetic Bearing systems inside a machine and their design have significant impact on the machine size and performance. In this paper we analyze three types of actuator arrangements commonly used to control two radial and one axial degrees of freedom of a rotor using a real-life gas compressor example. These actuator arrangements include:

1. Heteropolar electrically-biased radial actuator and conventional electrically-biased axial actuator [1];
2. Heteropolar electrically-biased radial actuator and electrically-biased axial actuator with low target OD [1];
3. 'Side-By-Side' (SBS) homopolar permanent-magnet-biased combination axial/radial actuator [2].

We will investigate impact of the choice of the actuator solution on the following performance characteristics of an example gas compressor:

1. Rotor length;
2. Separation margin from the first forward-whirl bending mode frequency;
3. Axial actuator gain and load capacity loss with frequency;
4. Increase of the phase lag between the axial force and control current with frequency;
5. Aerodynamic drag;
6. Actuator axial negative stiffness;
7. Actuator radial negative stiffness;
8. Radial actuator performance loss with speed.

II. APPLICATION EXAMPLE

A machine that we use as an example for this study is a real life 300kW power range high-speed (≈40kRPM) turbocompressor equipped with a Surface Mounted Permanent Magnet (SMPM) synchronous motor. Figure 1 shows all the major machine components using the first actuator arrangement (heteropolar electrically-biased radial actuator and conventional electrically-biased axial actuator) as an example. The static load capacity requirements for the bearing arrangement on the impeller side are 3000N axially and 1200N radially. The impeller weighs approximately 4kg and is coupled to the shaft through a face tooth coupling (such as Hirth or Curvic) and Tie-Rod arrangement.

For the purpose of this comparative study we did not change any other component of the machine in the other configurations except for the radial/axial actuator arrangement located on the impeller side. Fig. 2 shows all three

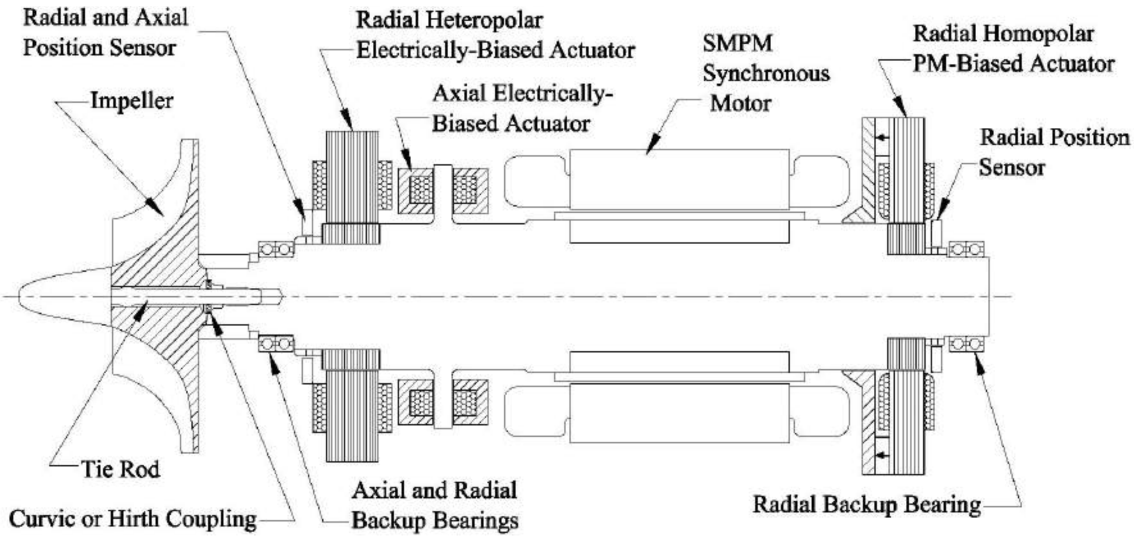

Figure 1. A turbocompressor example used to study effects of the radial/axial actuator arrangement. Shown with the heteropolar electrically-biased radial actuator and conventional electrically-biased axial actuator on the impeller side.

configurations side by side. In particular, the non-impeller side bearing is a homopolar permanent-magnet biased radial bearing in all three configuration. This is an effective solution for all configurations, but using a non-impeller side radial heteropolar bearing would not affect the results of the comparative study.

The comparison was based on the results of the numeric calculations, however, the configuration shown in Fig. 2c has

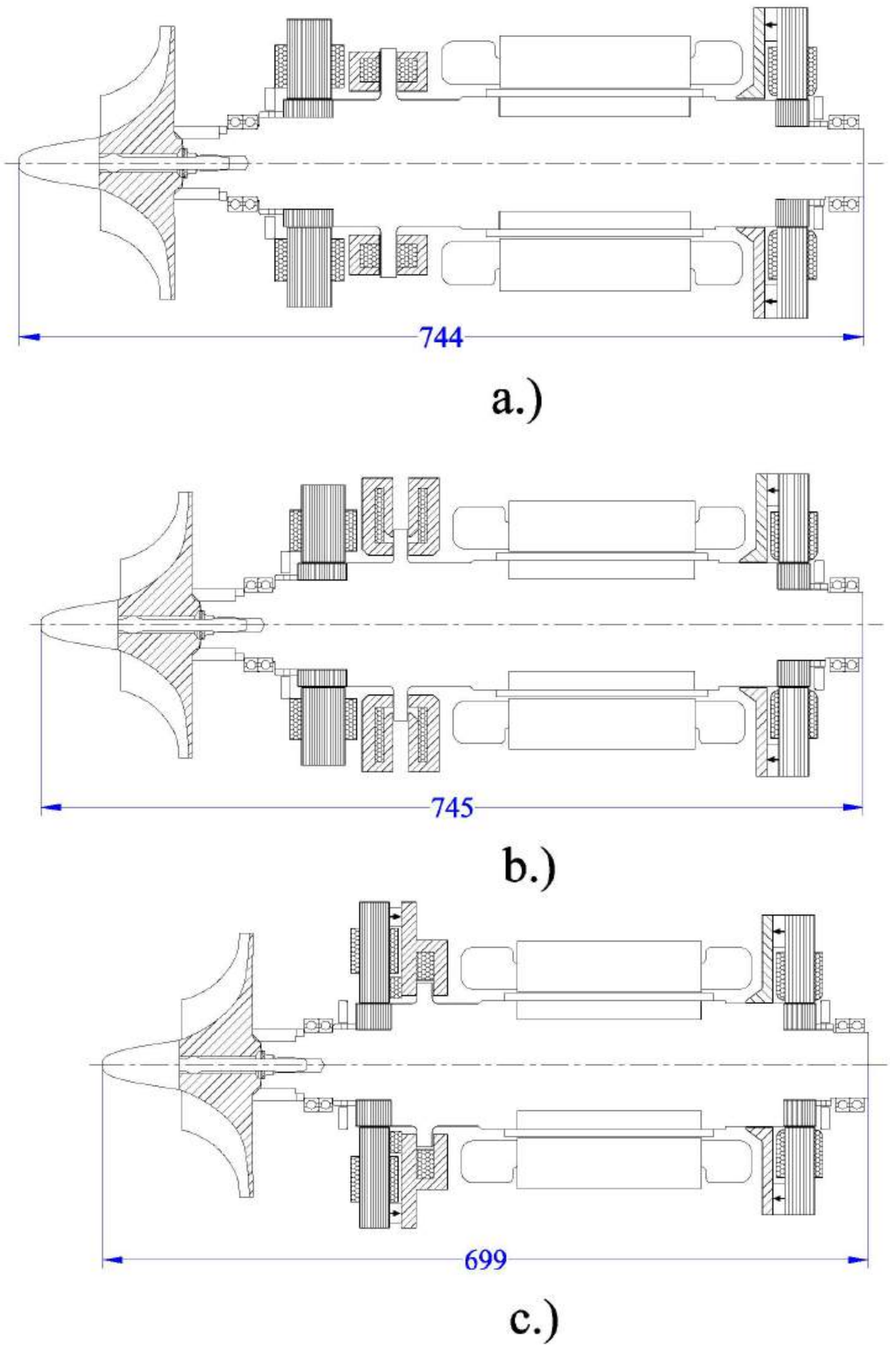

Figure 2. Turbocompressor configurations featuring different radial/axial actuator arrangements on the impeller side.

a) Heteropolar electrically-biased radial actuator and conventional electrically-biased axial actuator;
b) Heteropolar electrically-biased radial actuator and electrically-biased axial actuator with low target OD;
c) 'Side-By-Side' homopolar permanent-magnet-biased combination axial/radial actuator.

been built and tested. The measured performance characteristics in this case were found to be in a good agreement with theoretical predictions. Calnetix has also built several machines utilizing configuration similar to 2a with an exception of the radial bearing on the impeller side where a homopolar PM-biased radial magnetic bearing was used. In all of those machines the experimental results also matched well the analytical estimates.

III. COMPARISON METRICS

The following performance metrics were used to quantitatively characterize the machine performance for comparison purposes.

1. *Rotor length.*

The shorter rotor the more compact the machine would be.

2. *Rotordynamic margin at 37kRPM.*

This parameter defines separation (as a percentage of a rotational speed) between the rotor speed and the first forward-whirl bending mode frequency. In order to simplify controls it is preferable that the rotational speed stays below the first forward-whirl bending mode frequency and the separation is as large as possible.

3. *Axial actuator normalized gain at 40Hz.*

Bearing components defining the magnetic flux path in axial actuators are typically impractical to laminate. Therefore, there are significant eddy currents induced in those components whenever the magnetic flux changes in time. This parameter (dynamic load capacity as a percentage of the static load capacity) serves to compare actuator gain and load capacity loss with frequency between different designs.

4. *Axial actuator phase lag at 40Hz.*

In addition to the actuator gain loss, eddy currents also cause a phase lag between the forces and control currents in the axial actuators, which makes control design more difficult and at the end negatively impacts the machine performance. It is desirable to keep this phase lag to the minimum.

5. *Windage loss in actuator at 37kRPM.*

This is a portion of the total windage loss due to the axial/radial actuator arrangements shown in Fig. 3 only. Differences in overall windage losses between arrangements 2a, 2b and 2c show in Fig. 2 are caused exclusively by the differences between actuators 3a, 3b and 3c in Fig. 3.

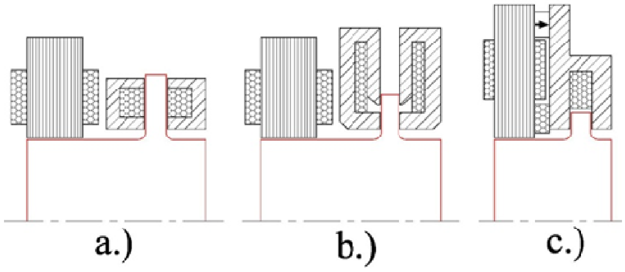

Figure 3. Portions of the systems 2a.), 2b.) and 2c.) shown in Fig. 2 in a respective order used for comparative study of windage losses. The rotor is assumed to be spinning in the air at 37kRPM.

6. *Axial negative stiffness.*

Negative stiffness effectively translates into a reduction of an actuator load capacity whenever the rotor position is off magnetic center due to manufacturing inaccuracies or a dynamic response to external loading. When comparing two magnetic bearings with otherwise similar performance metrics, the smaller this parameter - the better.

7. *Radial negative stiffness.*

Similar to the axial negative stiffness: when comparing two magnetic bearings with otherwise similar performance metrics, the smaller this parameter - the better.

8. *Skin-depth speed.*

Even though the radial actuator target is always made laminated in practical AMBs, at sufficiently high speeds this does not guarantee that there will be no expulsion of a magnetic flux from the laminations due to the skin effect, resulting in a loss of the bearing radial load capacity at speed, cross-coupling between the axes, and other (mainly negative) consequences. A numerical estimate of the effects of the eddy-currents in the rotor is a difficult task outside the scope of this paper. As a simple numerical parameter that can be used to carry out comparison between designs we used the rotor speed at which the skin depth associated with the fundamental electrical frequency seen by a spinning rotor becomes less than the lamination thickness. We considered two smallest standard off-the-shelf values of the silicon steel lamination thickness - 0.005in (0.127mm) and 0.007in (0.178mm).

TABLE I. Comparison of different axial/radial actuator arrangements on the impeller side.

	Heteropolar E-Biased Radial Conventional E-Biased Axial	Heteropolar E-Biased Radial Low Target OD E-Biased Axial	'Side-By-Side' Homopolar PM-Biased Combination
Rotor Length (mm)	744	745	699
Rotordynamic Margin at 37kRPM	42%	42%	42%
Normalized Axial Gain @ 40Hz	63%	38%	56%
Axial Phase Lag @ 40Hz (deg)	20	38	22
Windage Loss in Actuator at 37kRPM* (W)	3730	2050	1000
Axial Negative Stiffness (N/mm)	3100	3700	1800
Radial Negative Stiffness (N/mm)	9100	9100	7600
0.127mm Skin-Depth Rotor Speed (RPM)	42,000	42,000	84,000
0.178mm Skin-Depth Rotor Speed (RPM)	21,000	21,000	43,000

*In the air. See Fig.3 for definitions of the system portions to which these windage losses are attributed.

IV. COMPARISON RESULTS AND DISCUSSION

The results of the comparative study are summarized in Table 1. Each magnetic bearing arrangement analyzed here has its own advantages and disadvantages. In order to explain the origins of these advantages and disadvantages, Figs. 4, 5 and 6 illustrate structure, operating principles and key dimensions of a conventional heteropolar radial magnetic actuator, electrically biased thrust actuators and a homopolar permanent-biased 'Side-By-Side' combination radial/axial magnetic actuator shown in Fig. 2. More details of their operation can be found in [1] and [2].

As it can be seen from Fig.2 and Table 1, using 'Side-By-Side' combination actuator results in a shortest rotor because several components (particularly the permanent magnet) are used in both radial and axial parts of the actuator (see Fig. 6).

Another reason that the combo actuator is significantly more compact then alternatives is that a size of a modern rare-earth magnet needed to produce a certain magnetic field is only a small fraction of a size of a coil that would be needed to produce the same field. This is because reduction of a size of a coil with a given number of Ampere-turns inevitably leads to the increase of a total resistive power

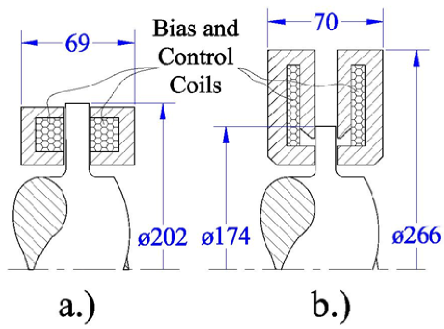

Figure 5. Structure and envelope dimensions of electrically-biased thrust bearings used in configurations shown in Figs. 2a and 2b respectively.

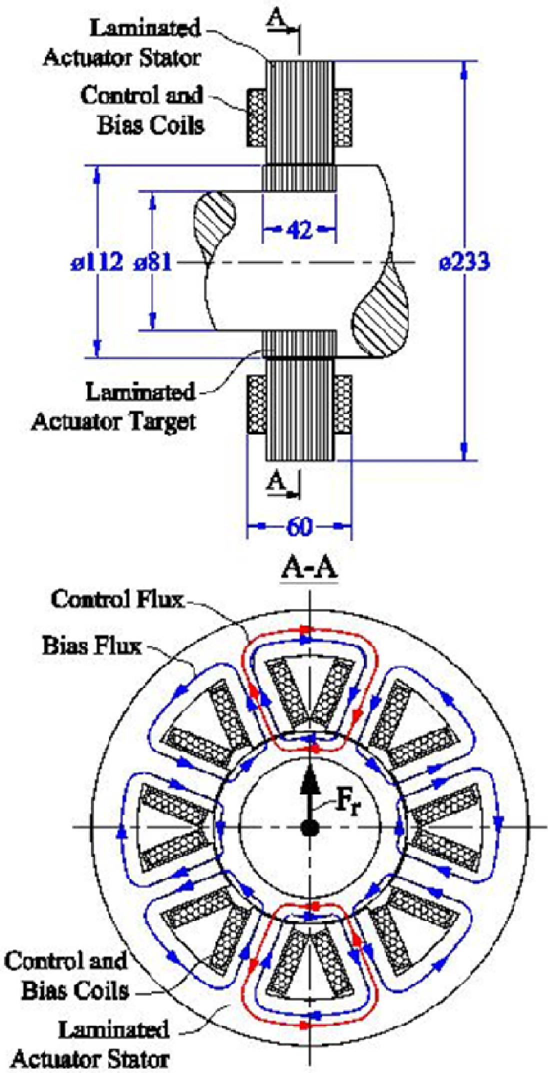

Figure 4. Structure, operating principle and key dimensions of a heteropolar radial magnetic actuator shown in Figs. 2a and 2b.

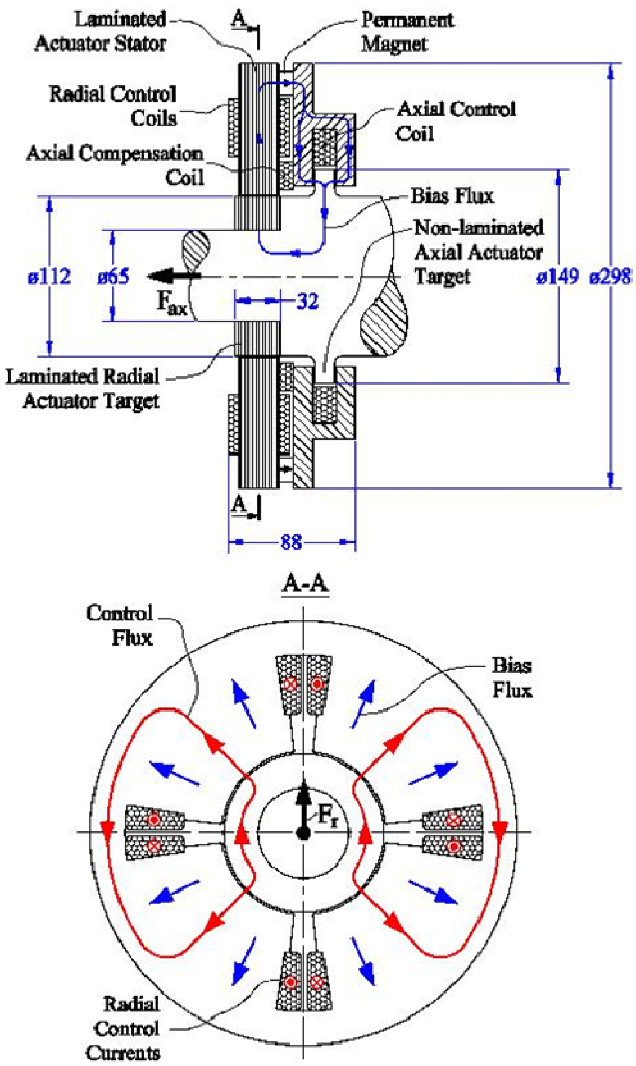

Figure 6. Structure, operating principle of a radial channel and key dimensions of a 'Side-By-Side' combination magnetic actuator shown in Fig. 2c.

dissipation in the coil, and even more to the increase of a resistive power dissipation in a unit volume of a coil, which may lead to the coil burn out.

A shorter rotor length in this example, however, does not translate into a rotordynamic advantage because heteropolar radial bearings used in the other arrangements require shallower lamination stacks on the rotor, leading to higher diameter of the rotor section under the stacks (compare Figs. 4 and 6). This diameter increase in arrangements 2a and 2b almost exactly compensates for the increases in rotor lengths in these arrangements (in this machine example) compared to arrangement 2c using a more compact combination bearing. See Fig. 7 for comparison of the first forward-whirl bending modes shapes and frequencies between variants 2a and 2c.

The reason for shallower actuator target laminations in heteropolar design is that both the bias and control fluxes are separated into numerous portions running short distances between neighboring poles (Fig. 4), whereas the conventional homopolar design utilizes four large poles with the entire control flux having to travel between the poles through the target laminations when the load is applied between the poles (Fig. 6). The price of this advantage of the heteropolar design is higher electrical frequency seen by the spinning rotor, leading to higher rotational losses, loss of the radial load capacity at speed and other complications which will be briefly discussed later herein. There is a known solution that

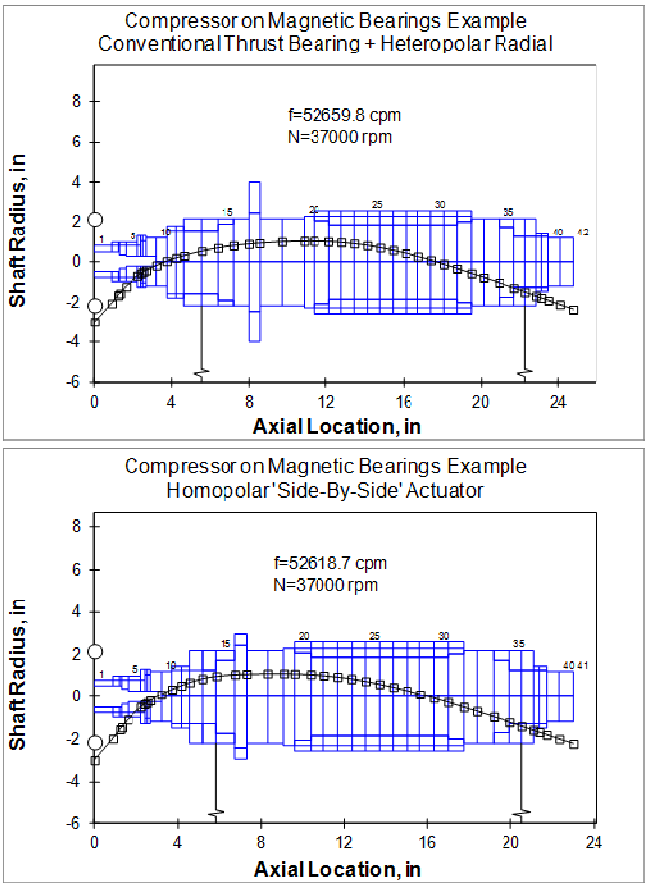

Figure 7. First forward-whirl bending mode shapes and frequencies at 37kRPM for the turbocompressor configurations shown in Fig. 2a (top) and 2c (bottom).

allows using shallow laminations in homopolar bearings as well [4], but it is more complicated to assemble and is not considered here.

The "classical" electrically-biased axial actuator shown in Fig. 2a and 5a offers the smallest losses of the axial gain and phase with frequency due to the shortest distance the control magnetic flux has to travel in a non-laminated iron from all the configurations. However, this actuator also requires a largest diameter target disk, resulting in much larger windage losses (windage losses for a spinning disk are increasing approximately as the fifth power of the disk diameter [3]). A modified version of this actuator shown in Fig. 2b needs a smaller disk diameter, resulting in almost two-fold reduction of the rotational loss (Table 1), but the gain and phase losses become much bigger due to a significant increase of the length of the control flux path in non-laminated iron.

The 'Side-By-Side' combination actuator offers the smallest target diameter from all the options (rotational losses are reduced by almost a factor of four compared to Fig. 2a) combined with a frequency response almost as good as in the version 2a.

FEA-calculated control magnetic flux distributions in different thrust bearings at 40Hz are shown in Fig. 8 and complete transfer functions are shown in Fig. 9. All calculations were done using FEMM [5].

The 'Side-By-Side' combination actuator also offers the lowest values of both radial and axial negative stiffness due to the bias flux being generated by a permanent magnet rather than a coil with a current. This is because permanent magnets have internal reluctances limiting bias fluxes to lower values than those produced by currents when either radial or axial targets deviate from the magnetic equilibriums.

The Skin-Depth Rotor Speed parameter comparison in Table 1 in effect simply illustrates that a better performance can be expected from homopolar radial bearings than heteropolar bearings at high speeds because of lower frequency of flux reversals seen by the rotor. However, Table 1 does not provide numerical estimates of the final parameters of interest affected by the eddy currents in the rotor such as loss of load capacity, amount of cross-coupling between the axes and rotational drag. The latter estimates are difficult to make and are not covered in this paper. It should be clarified that in case of the 8-pole heteropolar bearing (Fig. 4) we have chosen for the arrangements shown in Figs. 2a and 2b, the assumption was made that this bearing is used in NN-SS-NN-SS configuration to minimize electrical frequency seen by the rotor. Furthermore, we only considered the fundamental electrical frequency, which in this case is equal twice the rotational frequency of the rotor. There will be higher order frequencies caused by the gaps between the magnetic poles, which will get expelled from the rotor even faster. Therefore, the skin-depth frequency estimate for the heteropolar bearings listed in Table 1 is on the optimistic side.

In case of a homopolar bearing (Fig. 6) used in the machine arrangement shown in Figs. 2c, the fundamental frequency of the bias field will be zero in the absence of the radial loading and it won't be expelled from the rotor at any

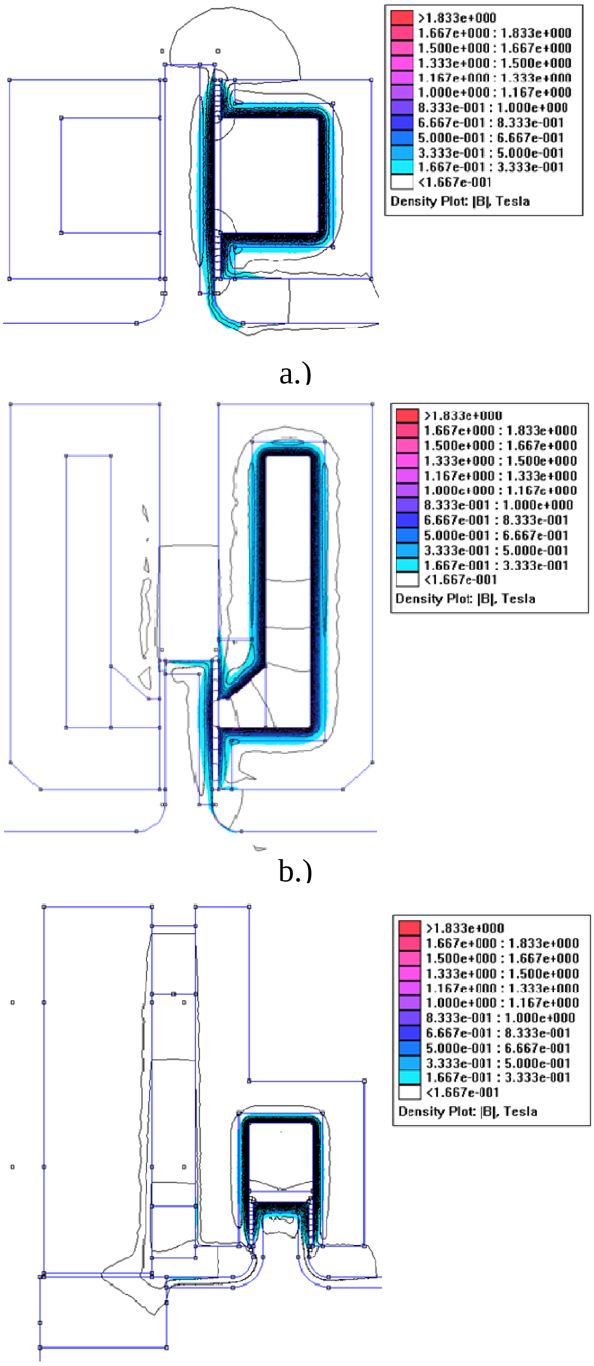

Figure 8. Axial control magnetic flux distributions in different configurations of electromagnetic actuators shown in Fig.2. The distribution shown in Fig. 8a corresponds to Fig. 2a, 8b corresponds to 2b, etc.

speed (there will be some field harmonics caused by having gaps between the poles which will be expelled similar to the heteropolar design). However, when a radial load is applied as illustrated in Fig. 6, a control magnetic field generated in response to that load will be non-uniform around the bearing resulting in a spinning rotor seen a time-varying magnetic

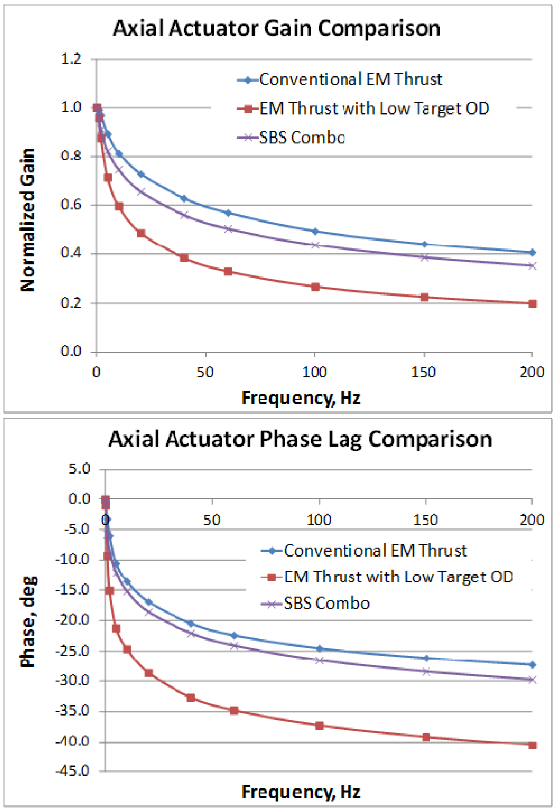

Figure 9. Comparison of the axial transfer functions for the different turbocompressor arrangements shown in Fig.2.

field with a fundamental frequency equal to the spinning frequency.

Based on a technical performance metric comparison in Table 1 it appears that the 'Side-By-Side' Homopolar Permanent would be a better choice for this particular application, but it has two additional drawbacks not reflected in the Table 1: higher cost and complexity, which may even the playfield in some applications.

V. CONCLUSIONS

A comparative study of three common radial/axial actuator arrangements for a real-life turbomachinery application has been carried out. While the answer to the question of which actuator would be the best clearly would depend on specifics of the application in hands, the results of the present study illustrate the importance of the actuator choice for the machine performance and typical trade-off associated with the most common actuator solutions.

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