PASSIVE PARALLEL AIRGAP SERIAL FLUX (PASF) MAGNETIC BEARINGS

W.K. Steve Khoo, Seamus D. Garvey, Karuna Kalita & Keith J. Bradley
School of Mechanical, Materials, Manufacturing Engineering & Management
The University of Nottingham, University Park
Nottingham NG9 2RD, United Kingdom
steve.khoo@nottingham.ac.uk, seamus.garvey@nottingham.ac.uk,
eaxkk@nottingham.ac.uk, keith.bradley@nottingham.ac.uk

ABSTRACT

Specific load capacity - defined as the ratio of maximum sustainable weight to the total self-weight, is one of the most desirable characteristics in magnetic bearing design. Published literature suggests that conventional radial magnetic bearings have a specific load capacity in the order of 40:1. This paper presents a new class of magnetic bearings that can deliver substantially higher than 100:1. The bearing force is achieved through the integration of magnetic shear stress contributions caused when lines of magnetic flux are driven across a number of parallel annular airgaps at an angle to the normal - hence the name "parallel airgap serial flux" or "PASF". The optimum designs that give the highest specific load capacity have been found via execution of thousands of finite element analyses across the whole design space. The mechanical stability of the PASF configurations has also been investigated.

INTRODUCTION

Conventional active radial magnetic bearings have long been used in many industries. The physical structure of these bearings is predominantly a hollow cylinder of back-iron with inwardly-protruding poles – a design that has not changed much since their existence. Most of them operate by making use of the tensile magnetic stress that exists when magnetic flux is caused to pass through air by exciting the coils. The net force exerting on the rotor is found by integrating the magnetic stress around the air gap. Since the airgap flux density is limited, it follows that bearings that utilise the tensile stress will also have a limited sustainable force.

For direct comparison of various magnetic bearings regardless of whether they are active or passive, we introduce the term 'specific load capacity (SLC)' – defined as the ratio between the maximum achievable force F_{brg_max} and the total self-weight of the bearing $m_{brg}g$.

$$SLC = \frac{F_{brg\_\max}}{m_{brg}.g} \tag{1}$$

Different views have been offered as to what this ratio can approach for conventional radial bearings and values of 25:1 and 40:1 [1, 2] have all been put forward. This paper describes a new class of magnetic bearings that can deliver specific load capacities substantially higher than 100:1. The motivation behind this project is the 'more-electric' movement particularly for oil-less aero-engines or applications where high specific load capacity is desired. We simulated bearings with different parameters using a commercial finite element analysis (FEA) package in order to determine the highest possible specific load capacity. The designs were then checked for mechanical stability using a finite element code written in MATLAB.

PASF MAGNETIC BEARINGS

One route to achieving high specific load capacity bearings may be through the use of Parallel Airgap Serial Flux (PASF) configurations [3]. An example is shown in figure 1 where parallel airgaps come about when rotor discs are successively interleaved between stator discs with a certain gap between them. An appropriate MMF source then drives a bundle of magnetic flux through the stack of discs at an angle to the normal plane producing virtually the same magnetic shear stress across the multiple airgaps. The integration of magnetic shear stress over the airgap area gives rise to force and the sum of force contributions from all parallel airgaps constitutes the overall bearing force.

In some classes of PASF bearing, the more parallel airgaps incorporated for a given volume the higher specific load capacity will be. This is a distinct advantage of the PASF configurations over conventional magnetic bearings that rely directly on the basis of tensile stress. However, in our pursuit of high specific load there is a limit to which the discs can be made very thin because of the negative magnetic stiffness. The PASF bearings are inherently unstable in the axial direction so the discs must be sufficiently thick to prevent mechanical buckling.

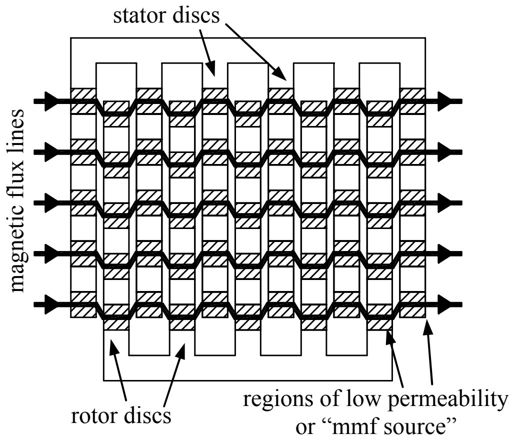

FIGURE 1: Parallel airgap serial flux concept

The PASF magnetic bearings can be passive, active or semi-active and there are many possible variants that can be constructed. We restrict this paper to two types of passive PASF bearing, namely: contra-magnetised and iron castellation disc configurations. Both are essentially reluctance bearings. The former is the preferred form in terms of specific load capacity. Hence, this paper emphasises the contra-magnetised PASF variant.

Figure 2 illustrates half of an axial cross-section of the contra-magnetised PASF design. Each disc has an even number of axially magnetised permanent magnet (PM) rings and each ring is magnetised in an alternate direction from its concentric neighbour as shown. The stator endplates are made of ferromagnetic material and their main function is to provide a low reluctance return path for the magnetic flux lines. If there is a relative eccentricity between the stator and rotor discs, flux lines tend to zigzag as they traverse across the airgaps. As a result a working shear stress is created in the airgaps and a restoring force pushes the rotor back to its concentric position.

The iron castellation disc design shown in figure 3 works on a similar principle. It has multiple concentric circular iron ridges sandwiching between non-permeable materials which form the grooves. These iron ridges offer a low reluctance path for magnetic flux from a current source or permanent magnet to pass through from one disc to another. The stator back-of-core provides a return path for the flux.

FINITE ELEMENT DESIGN PARAMETERS

The effective design of any PASF magnetic bearings inevitably requires a full 3D FEA because of the nature of the bearings concerned. A passive PASF bearing with zero radial displacement can be effectively modelled in 2D axis-symmetric environment but this modelling has no use since the net restoring force would be zero. There must be a relative displacement between the stator and rotor discs in order that a force is created which means that axis-symmetric models are strictly inappropriate.

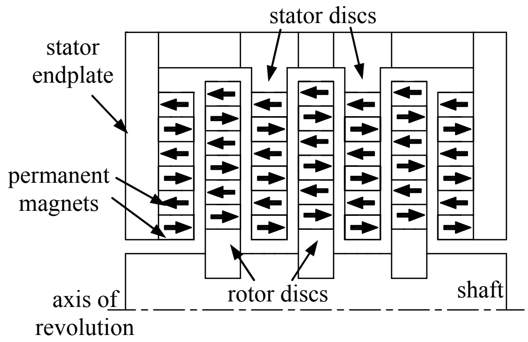

FIGURE 2: Contra-magnetised PASF magnetic bearing

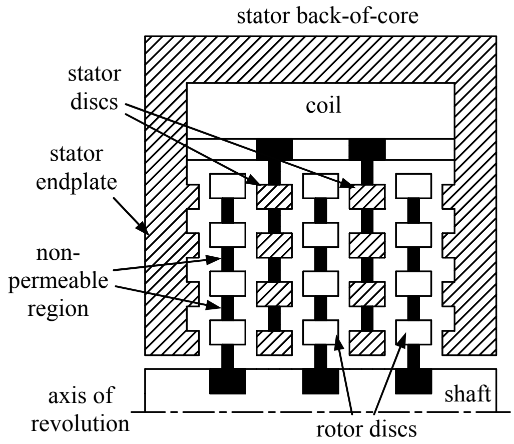

FIGURE 3: Iron castellation PASF magnetic bearing

Contra-magnetised PASF magnetic bearings	
Number of PM rings:	2-10 in steps of 2
Radial pitch, λ:	2-10mm in steps of 2mm
Air gap thickness, t_{gap}:	0.4-1mm in steps of 0.1mm
Disc thickness, t_d:	1.5-3mm in steps of 0.5mm
Mark space ratio, d/λ:	1 (best value)
(where d is the radial thickness of the PM)	
Horizontal displacement:	0-50% of t_{gap} in steps of 10%
Vertical displacement:	10-50% of λ in steps of 10%

Iron castellation PASF magnetic bearings	
Number of ridges:	5
Radial pitch, λ:	4-10 mm in steps of 2 mm
Air gap thickness, t_{gap}:	0.4-1 mm in steps of 0.2 mm
Central disc thickness, t_{cd}:	1-2 mm in steps of 1 mm
Mark space ratio, d/λ:	0.2 to 0.8 in steps of 0.2
(where d is the radial thickness of the ridge)	
Ridge height, d_{rh}:	1-2 mm in steps of 0.5 mm
Horizontal displacement:	0-50% of t_{gap} in steps of 25%
Vertical displacement:	10-40% of λ in steps of 10%

TABLE 1: Design parameters

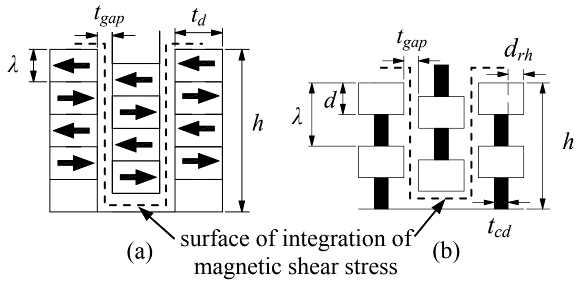

FIGURE 4: Geometry definitions (a) contra-magnetised design, (b) iron castellation design

Simulating a large design space in 3D would be prohibitively expensive in computation. To circumvent this problem we simulated the models in a 2D plane and adopted an approximate method to calculate the equivalent force (see section on mean effective magnetic shear stress). Axis-symmetric models can be used for estimating the negative stiffness where this value is at maximum when the stator and rotor discs are perfectly aligned.

A number of design parameters affect the specific load capacity of PASF magnetic bearings, namely: ridge number or number of PM rings, radial pitch of the rings, mark-space ratio of the rings, disc thickness, airgap thickness, degree of eccentricity, etc. Table 1 shows the list of parameters of the whole design space where tens of thousands finite element models have been executed automatically in nested loops. The corresponding geometry definitions are shown in figure 4.

MEAN EFFECTIVE MAGNETIC SHEAR STRESS (MEMSS)

A critical measure of goodness used pervasively in this paper as far as specific load capacity is concerned is the *Mean Effective Magnetic Shear Stress* (MEMSS). It is defined as the maximum force capability of a PASF bearing divided by the total area of airgap in the bearing. With reference to figure 4 the MEMSS can be evaluated as:

$$\mathrm{MEMSS} = \frac{F}{4 \times h}\ \mathrm{N/m^2} \qquad (2)$$

where F is the force per unit length per disc obtained from the surface integration of magnetic shear stress in the 2D plane FEA (N/m) and h is the difference between the outer r_{or} and inner r_{ir} radii of the rotor disc. The factor 4 in the denominator accounts for 2 times the height h and that the mean value of force in an actual displaced stator and rotor discs model is 2. The total force per disc and weight of one stator and one rotor disc are:

$$F_{disc} = 2 \times \mathrm{MEMSS} \times \pi\left(r_o^2 - r_i^2\right) \qquad (3)$$

$$W_{disc} = \rho t_d \pi\left[\left(r_{or}^2 - r_{ir}^2\right) + \left(r_{os}^2 - r_{is}^2\right)\right]g + w_o \qquad (4)$$

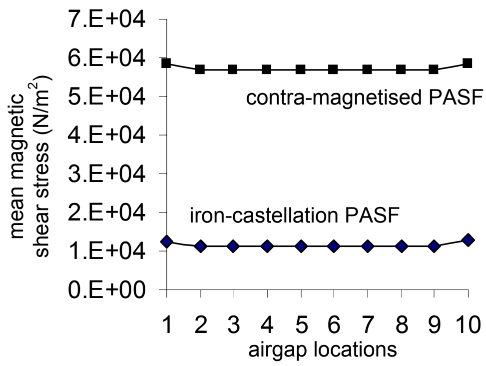

FIGURE 5: Mean magnetic shear stress at various airgap locations in 6 stator/5 rotor discs configurations

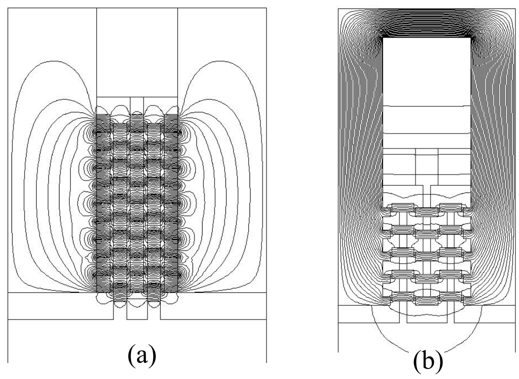

FIGURE 6: 2D flux plot (a) contra-magnetised design (b) iron castellation design

where r_{os} and r_{is} are the outer and inner radii of the stator disc and w_o is the averaged mass overhead contributed by the stator endplates, coil, non-permeable spacers, etc. Thus, the specific load capacity is then:

$$SLC = \frac{F_{disc}}{W_{disc}} \qquad (5)$$

It should be noted that for the iron castellation variant the weight of the disc W_{disc} must include the weight of the ridges.

Figure 5 shows a plot of MEMSS evaluated along 10 airgaps of the two PASF variants. It is unnecessary to simulate for more than a total of 5 stator/rotor discs because the magnitude of the magnetic shear stress stays tolerably constant after flux emerges from the first stator disc (an endplate). From the preliminary FEA trials it was found that a mark space ratio of 1 gives the best specific load capacity for the contra-magnetised variant so this value was kept constant for all subsequent FE models. The PM rings have a remanence flux of 1.2 T. A fixed current density of 5 A/mm^2 was applied to the coil region of the iron castellation models. In all cases the diameter of the shaft was taken as 200 mm. An example of 2D flux plot for both PASF variants is depicted in figure 6.

FINITE ELEMENT RESULTS – CONTRA-MAGNETISED PASF MAGNETIC BEARINGS

The first approach to achieving high specific load capacity in design of PASF magnetic bearings is to maximise the MEMSS. Although this is not absolutely true for all cases, the value of MEMSS does provide an early indicative idea of how well a PASF bearing performs. The contra-magnetised variant operates on a number of concentric PM ring pairs and it follows that the MEMSS increases with increasing number of PM ring pairs. Figure 7 shows a plot of MEMSS of one disc and airgap thickness (1.5 mm and 0.4 mm respectively) for a range of radial pitches and number of PM rings. The maximum of value of MEMSS predicted for this particular disc thickness is about 105 kPa at a radial pitch of 2 mm. Reducing the radial pitch of the PM has the effect of increasing the overall MEMSS.

The principle of operation of all passive magnetic bearings including the PASF designs is that a net restoring force comes to exist when there is a relative eccentricity between the stator and rotor. A maximum force can only be achieved when the rotor is displaced up to its maximum allowable distance with respect to the stator. For small relative displacements the net force is very low. Such a "softness" characteristic is typical for passive magnetic bearings and this one of the disadvantages compared to their active counterparts. Figure 8 shows the variation of MEMSS with radial displacement for a range of radial pitches. Accordingly, the rotor and stator discs of the PASF magnetic bearings must be displaced up to 50% of the radial pitch before a maximum force can be accomplished. Large radial pitches not only give rise to low MEMSS but also lead to a high degree of softness. Furthermore it is unpractical to allow a large radial displacement in order to gain a maximum force. While small radial pitches are favourable there is a limit to which the size of the pitches can be practically made. Moreover, radial pitch must not be much less than the airgap thickness.

The relation between the PM disc thickness and the MEMSS for a radial pitch of 2 mm can be seen from figure 9 where increasing the former leads to an increase in the latter. This trend appears to stay at a constant value of 123 kPa beyond a thickness of 3 mm. As such it is more beneficial in terms of performance and cost to construct discs with thickness of less than 3 mm. The variation of specific load capacity with disc thickness in the same figure further reinforces the fact that thin discs are favourable. Though the MEMSS is higher for thicker discs the specific load capacity does not fare so well because of the increase in self-weight. At a radial pitch of 2 mm and disc thickness of 1.5 mm the specific load capacity of the contra-magnetised variant is about 750:1. This is the optimum geometry for the range of parameters being investigated. The specific load capacity drops to 450:1 when the thickness of the disc is doubled.

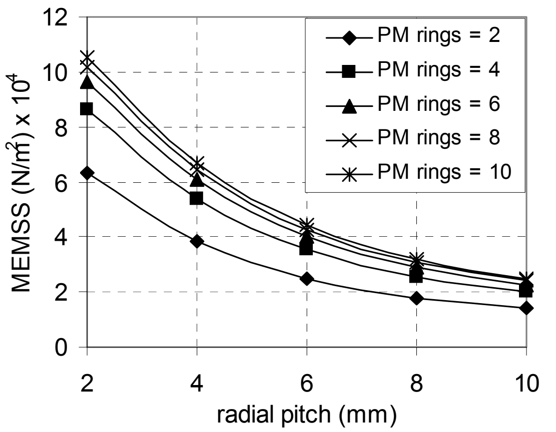

FIGURE 7: Variation of mean effective magnetic shear stress with the number of ridges and radial pitch (for disc thickness of 1.5 mm and 0.4 mm airgap thickness)

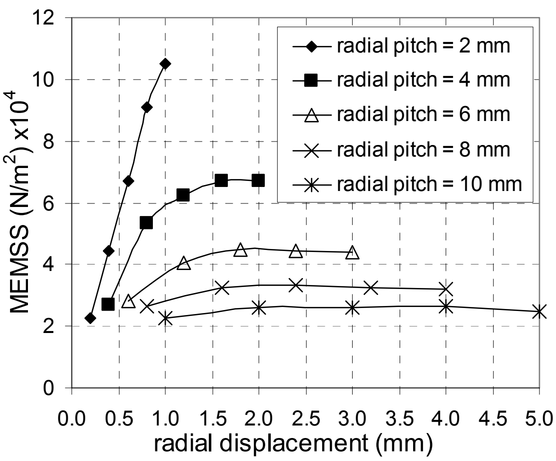

FIGURE 8: Variation of mean effective magnetic shear stress with radial pitch and displacement (for disc thickness of 1.5 mm and 0.4 mm airgap thickness)

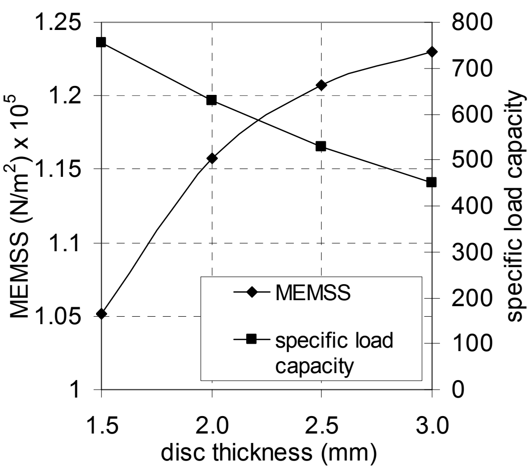

FIGURE 9: Variation of mean effective magnetic shear stress and specific load capacity with disc thickness (2 mm radial pitch and 0.4 mm airgap thickness)

The effect of increasing the radial pitch of the PM rings is represented in a surface plot as shown in figure 10. The airgap thickness is kept constant at 0.4 mm. It can be seen that increasing the radial pitch from 2 mm to 4 mm at a disc thickness of 3 mm does not affect the specific load capacity very much. However, further increase of radial pitch results in a reduction of specific load capacity at all disc thicknesses.

Thus far, the plots of MEMSS and specific load capacity for various geometric parameters are investigated by keeping the airgap thickness constant at 0.4 mm. It is known from the mechanics of magnetic force production that increasing the airgap decreases the force regardless of whether the force is as a result of integration of normal or shear stresses over the surface of interest. Figure 11 shows a plot of specific load capacity against the ratio of airgap thickness to radial pitch. As expected the specific load capacity reduces as the airgap thickness is increased. Increasing the ratio of airgap to radial pitch improves the specific load capacity for relatively small airgap in general. There appears to be an optimum for each airgap thickness. As the airgap thickness approaches 0.9 mm further increase of the ratio leads to a reduction of specific load capacity.

UNBALANCED MAGNETIC PULL AND MECHANICAL STABILITY

Although the PASF contra-magnetised bearings have a significantly higher specific load capacity than conventional magnetic bearings, one problem that plagues the former is the unbalanced magnetic pull due to the high negative magnetic stiffness. The consequence is that the interleaved PM rotor and stator discs tend to flex and pull over when there is a relative axial displacement between the discs. High specific load capacity and high load carrying capacity cannot be achieved by virtue of having thin discs and making the outer diameter of the discs large with respect to the inner diameter. It follows that the crucial mechanical stability associated with the design of the passive PASF bearings must be addressed so as to avoid failure due to buckling.

We investigated the mechanical stability via a dedicated 2D FEA coded in MATLAB [4] where the critical boundaries of stability for a range of disc and airgap thicknesses were established. This script presupposes that the disc is uniform and has orthotropic elastic properties. One example of assessing the stability of a disc is illustrated in figure 12 (disc thickness 1.5 mm and airgap thickness 0.4 mm). The negative magnetic stiffness for a range of radial pitches was evaluated per unit area from the 2D axis-symmetric FEA and plotted against the disc outer radius. The maximum allowable axial displacement is taken as 50% of the airgap thickness. The critical negative magnetic stiffness (denoted NMS in figure 12) obtained from the coded FEA was

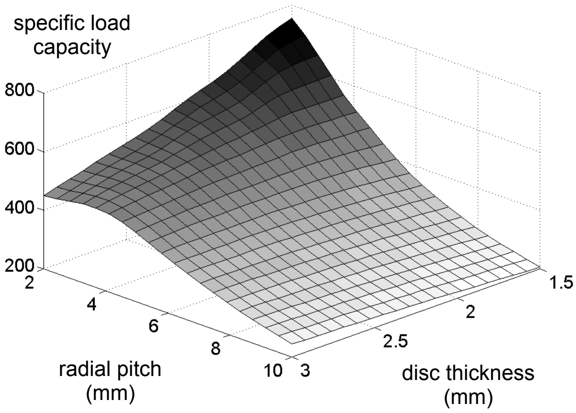

FIGURE 10: Surface plot of specific load capacity for a fixed airgap of 0.4mm

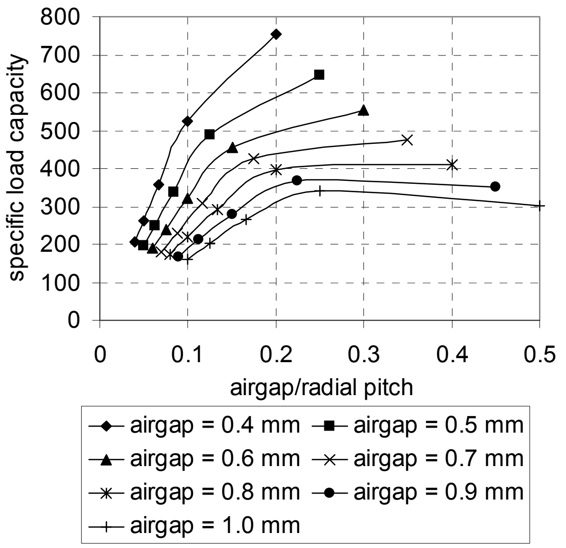

FIGURE 11: Variation of specific load capacity with the ratio of airgap thickness to radial pitch (for disc thickness of 1.5 mm)

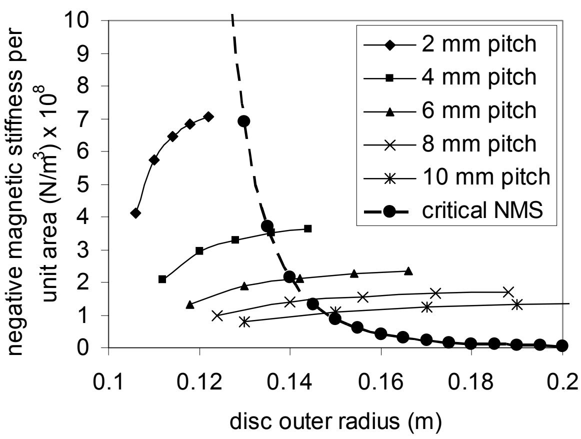

FIGURE 12: Mechanical stability of discs (for airgap of 0.4 mm and disc thickness of 1.5 mm)

then superimposed to reveal the stability limit. The region on the left of the critical NMS corresponds to mechanically stable designs. Any design points that fall beyond the critical NMS curve on the right are expected to be unstable.

It can be deduced that discs with smaller radial pitches are subjected to higher negative magnetic stiffness per unit area. Despite the high negative stiffness, the design with 2 mm radial pitch and disc outer diameter of 0.12 m (10 concentric PM rings) is well below the critical limit. Thus, the discs will not buckle due to unbalanced magnetic pull. However, the critical NMS curve cuts across the plots of radial pitches between 4-10 mm. This means that a total of 10 PM rings cannot be built stably as anticipated if a radial pitch of more than 2 mm is chosen. In order for the designs to be mechanically stable the number of PM rings or the outer diameter of the discs must be reduced until the design points fall below the critical limit.

With the stability determined a very fine mesh FEA was then conducted for the design with the highest specific load capacity. This is summarised in table 2 below.

Number of PM rings	10	
Radial pitch	2	mm
Mark space ratio	1	
Air gap thickness	0.4	mm
Disc thickness	1.5	mm
Max. radial displacement	1	mm
Discs weight	4.276	N
Force	7883.9	N/m
MEMSS	89590	N/m^2
Total force/disc	2749.2	N
Specific load capacity	643:1	

TABLE 2: Contra-magnetised PASF magnetic bearing

IRON CASTELLATION PASF MAGNETIC BEARINGS

The same 2D linear FEA and mechanical stability test procedures were run for the iron castellation variant. The geometric parameters that give the best specific load capacity are tabulated in table 3. Accordingly the MEMSS for the iron castellation variant is roughly half of what is achievable with the contra-magnetised variant. Coupled with the fact that discs are heavier because of the thick stator back-of-core and toroidal coil, the overall specific load capacity of this bearing is an order of magnitude lower than that of the contra-magnetised variant. It should be noted, however, that the iron castellation PASF bearings presented in this paper are not the true optimum. For example, since the flux density is lower towards the inner radius of the disc (figure 6(b)), the stator endplates can be tapered, thereby reducing the self-weight of the bearing. Nevertheless, it is very unlikely that the iron castellation design can have a comparable specific load capacity to the contra-magnetised variant even if the former has been fully optimised.

Number of PM rings	5	
Radial pitch	6	mm
Mark space ratio	0.4	
Air gap thickness	0.4	mm
Disc thickness	2	mm
Ridge height	2	mm
Max. radial displacement	1.2	mm
Discs weight	60.618	N
Force	7270.6	N/m
MEMSS	56101	N/m^2
Total force/disc	2654.2	N
Specific load capacity	43:1	

TABLE 3: Iron castellation PASF magnetic bearing

SUMMARY

The key finding from our finite element parametric studies is that bearing with a specific load capacity of 600:1 is achievable using the parallel airgap concept – a figure that an order of magnitude higher than conventional radial magnetic bearings. The contra-magnetised PASF bearings have at least 10 times higher specific load capacity compared to the iron castellation variant. The maximum achievable mean effective magnetic shear stress for the former is about 100 kPa.

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