Design of Very High Speed Friction Spindle using Passive Magnetic Bearings

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***Abstract* - Aiming at rotation speeds above 1 million rounds per minute, a two-stage motor device is hereby proposed, based on the concept of so-called false twisting aggregates, known from the yarn processing industry. The first stage, a synchronous machine, is designed around a passive magnetic bearing. The second stage, a small axis, spun by the first stage using a friction transmission is designed to find its physical limits around 7 million rounds per minute. This report presents the design of the experimental set-up. Based on finite element analysis, the performances of the future experimental set-up will be estimated.**

***Index Terms* – Passive magnetic bearings, high speed motor, friction drive, synchronous machine, false twisting aggregate.**

I. Introduction

Very high rotation speeds above one million rounds per minute (rpm) are difficult to achieve. The existing solutions to this problem feature a sub-millimetric steel ball, suspended by active magnetic bearings and spun by magnetic induction [1]. This research finds promising applications in the field of inertial instruments, but the dimensions and geometry of a suspended steel ball below 1 mm diameter make it technically difficult to use the rotor for applications such as optical scanners, and optical choppers.

The present paper describes the design steps that will lead to a high-speed drive based on a friction transmission. Using passive magnetic bearings and going to the limits of what is nowadays machineable and available, the authors are confident that the proposed device will allow rotation speeds in the 1Mrpm range.

II. Concept

Figure 1 depicts the concept of the proposed device, a concept adapted from a well known industrial application: The false twisting aggregate, used in the textile industry to give man-made fibres like acryl a cotton-like touch and feel. These devices were capable of spinning a small spindle at speeds above 300’000 rpms. Their technology was abandoned for a completely other approach in the 1980s because their rotation speed was not sufficient. The authors expect that, using magnetic bearings and high tenacity materials, this limit can today be brought beyond the one million rpm range. The high-speed rotor (the “spindle”) **1** is a cylindrical rod, pressed between four friction discs **2** by means of its ferromagnetic sleeve **3** and a permanent magnet **4**. The friction discs are mounted on axes **5** that are suspended each on two passive magnetic bearings **6**. The unstable degree of freedom, inherent to these kinds of bearings is passively controlled by means of a mechanical bearing **7**. Both friction discs are spun using synchronous motors, with the windings on the stator **8** and permanent magnets on the rotor **9**.

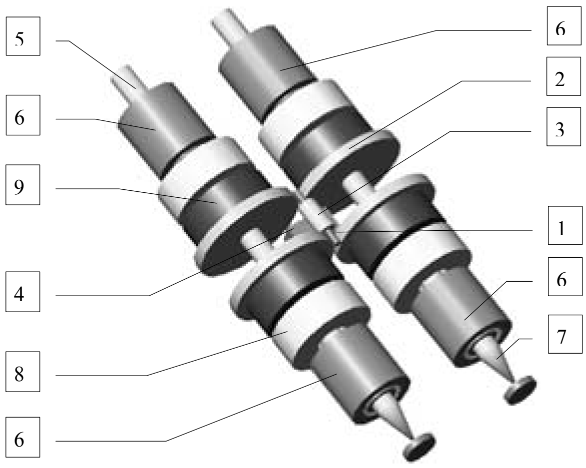

Fig. 1 Concept of friction spindle

III. Design

The specifications that were set for the spindle are somewhat simple: To reach rotation speeds above 1 Mrpm and to do so with a top-speed torque of 0.05 mNm.

A. High speed spindle

Looking at figure 1, it becomes clear that the spindle has to be pressed against the friction discs in order to reach the requested torque. Magnetic attraction is the only concept that is technically feasible. Hence, the spindle has to be ferromagnetic or it has to have some kind of ferromagnetic outer layer.

Whichever material will be used for the spindle, it has to resist to important centrifugal stress due to the high rotation speed.

The diameter of the spindle is set at the order of the millimetre. This choice is not so arbitrary as it may seem since a one millimetre rod is machineable and this is also the average diameter of the aforementioned false twisting aggregate spindles. At these dimensions, isotrope ductile materials (metallic alloys) are the best choice. Table 1 shows the relevant mechanical properties of some typical high-end metallic alloys used for rotary high-speed applications:

TABLE I
AVAILABLE HIGH PERFORMANCE DUCTILE MATERIALS

Material	E ($\times 10^9$)	σ_y ($\times 10^6$)	σ_T ($\times 10^6$)	ρ	ν	E / ρ ($\times 10^6$)	σ_y / ρ ($\times 10^6$)
Maraging steel Böhler 1.4548.4	117	2000	2300	7850	0.34	14.9	0.26
Al-Zn (7068)	72.5	683	710	2850	0.3	25.4	0.24
Mg-Al-Zn (AZ61)	44.8	230	310	1800	0.3	24.9	0.13
Amorphous Metal Vitrovac®	150	1900	2100	7700	0.3	19.5	0.25
Titan grade 5	110	1030	1100	4430	0.33	24.8	0.23

with

E	Young's modulus	[N/m^2]
σ_y	Yield strength	[N/ m^2]
σ_T	Tensile strength	[N/ m^2]
ρ	Density	[kg/ m^3]
v	Poisson's ration	[-]

The ratio σ_y / ρ indicates the materials' suitability for extreme rotary speeds.

The internal stress distribution for cylindrical objects made out of isotropic materials is described analytically by Larsonneur [2]. After analytical computing of the tangential and radial stresses, Maraging steel was chosen as spindle material (Böhler 1.4548.4). The computation showed that a 1 mm diameter spindle is theoretically capable of rotate at 9 Mrpm at its yield stress limit.

The Maraging steel spindle is not ferromagnetic enough for the required magnetic attraction force between spindle and friction discs: Due to the rotation of the spindle, its interaction with the static attraction magnet (figure 1, position 4) involves the skin effect

Table 2 shows the relevant magnetic and electric properties of possible ferromagnetic materials.

Based on these values, one can compute the flux penetration depth in function of the rotation speed (excitation frequency). Around the estimated maximal rotating speed of the spindle of 9 Mrpm, only high frequency ferrite (Siemens M33) features a penetration depth in the order of centimetres. The other materials are below 50µm, hence the high frequency ferrite is the material of choice. The material will be tube-shaped and fit around the spindle, as shown in figure 1.

However, the applied sleeve made out of M33 ferrite material is by no means capable to withstand the immense centripetal forces. The problem can be solved using a bandage made out of a fiber-epoxy compound.

TABLE II
MAGNETIC AND ELECTRICAL PROPERTIES OF AVAILABLE FERROMAGNETIC MATERIALS

Material	µ ($\times 4\pi\ 10^{-7}$)	σ
Maraging steel Böhler 1.4548.4	900	1'500'000
Sheeted amorphous metal Metglas-2714A (Magnaperm®)	80'000	705'000
Sheeted amorphous metal Metglas-2605SA1 (Microlite XP®)	250	705'000
High frequency ferrite Siemens M33	750	0.25
Sheeted FeSi alloy M530	7'000	3'000'000

with

σ	Conductivity	[S/m]
μ	Magnetic permeability	[Vs/Am]

For the calculation of the optimal bandage thickness, the internal pressure (exerted by the Maraging steel spindle and the ferrite sleeve) is maximized for a sound estimation: We considered the case of an internal hydrostatic pressure.

The mechanical properties of carbon fiber composites (e.g. TORAYCA® fibres, product of TORAY) show a tensile strength of up to 7 GPa for the T100 fibre. The T700 fibre has a much better availability and has a tensile strength of 5 GPa and is therefore the fibre of choice.

By varying the thickness of the ferrite sleeve and the thickness of the fibre bandage, one finds the optimal arrangement with respect to a maximal rotation speed.

It was found that with a ferrite sleeve of 0.5 mm wall thickness and a fibre bandage of 0.8mm wall thickness, the maximal speed of 7 Mrpm can theoretically be obtained. Above this speed there would be fatal failure of the inner bandage fibres. Figure 2 resumes the materials and dimensions of the high speed spindle:

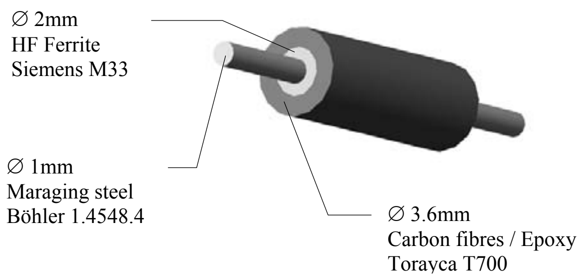

Fig. 2 Dimensions and materials of high speed spindle capable of rotating at 7 Mrpm.

B. *Friction drive*

The high speed spindle must be pressed against the friction discs (see figure 1) by magnetic attraction. The torque is function of this attraction force. This attraction force is limited by the M33 ferrite's saturation flux density of 400 mT.

Figure 3 shows the simulated set-up for the determination of the maximal attraction force. The maximal attraction force can be obtained by a Halbach array made out of NeFeB permanent magnets.

The size dimensions of the Halbach array were chosen to fit between the two motor axes (figure 1).

The attraction force is inverse proportional to the air-gap between spindle and Halbach array.

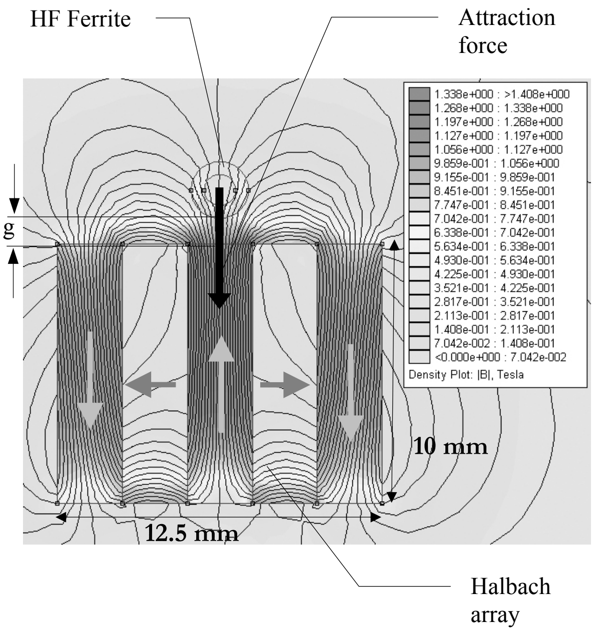

Fig. 3 FEM simulation of Halbach array for the magnetic attraction of the high speed spindle.

The simulation revealed that, with a clearance g of 1mm, a linear force density of 150 N/m can be reached.

By linear force density is described the fact that the longer ferrite and magnet array are, the bigger the attraction force will be.

With a 2 mm clearance, this force density equals 92 N/m, i.e. a 10 mm long ferrite sleeve at a 2mm distance over the Halbach array is attracted towards the latter with a force of 0.092 N.

The parameters of the proposed friction drive are shown in figure 4.

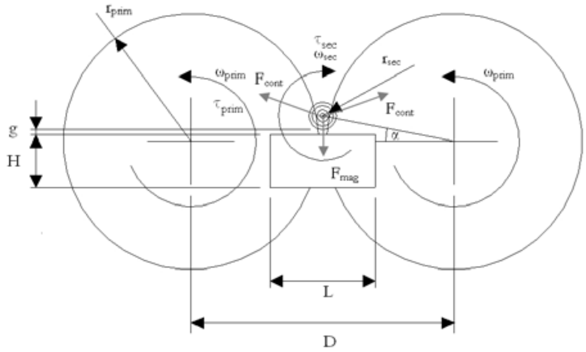

Fig. 4 Parameterisation of friction drive

with

τ_{sec}	Torque at spindle	[Nm]
τ_{prim}	Torque on friction-disc	[Nm]
r_{sec}	Radius of spindle	[m]
r_{prim}	Radius of friction-disc	[m]
μ	Friction coefficient	[-]
F_{mag}	Spindle attraction force	[N]
F_{cont}	reaction force spindle – frictiondisc	[N]
D	Inter-axis distance	[m]

Static and dynamic friction coefficient steel/steel is 0.4. With the hypothesis of small angles and trigonometry, one finds for the magnetic attraction force between spindle and Halbach array in function of the desired torque:

$$F_{mag} = \frac{2\tau_{\sec}}{\mu r_{\sec}} \arccos\left(\frac{D}{2\left(r_{prim} + r_{\sec}\right)}\right) \tag{1}$$

The spindle radius r_{sec} was previously set at 0.5mm. Aiming the 7 Mrpm rotation speed for the spindle, the radius r_{prim} is now set at 25 mm, corresponding to a maximal friction-disc rotation speed of 200'000 rpm.

Presuming a worst-case friction coefficient of 0.3 and according to (1), the necessary magnetic force for a 0.05 mNm torque is 0.08 N.

Hence the Halbach array of figure 3 is apt to deliver the necessary magnetic attraction force.

Figure 5 recapitulates the dimensions of the friction drive, resulting in 0.065 mNm transmittable torque.

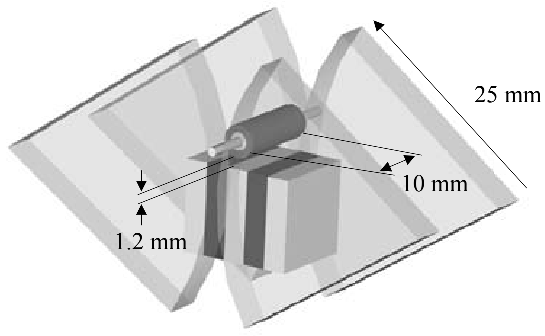

Fig. 5 Friction drive of spindle

C. Passive magnetic bearing

As presented in figure 1, the friction discs will be fitted on two axes, each equipped with a synchronous motor. These axes must spin with as little friction as possible, therefore passive magnetic bearings have been proposed.

The unstable degree of freedom must be stabilized by active (electromagnet) or passive (mechanical bearing) means. The forces against this additional bearing must be compensated by a so-called "weight compensation". Figure 6 plots the concept of the proposed bearing:

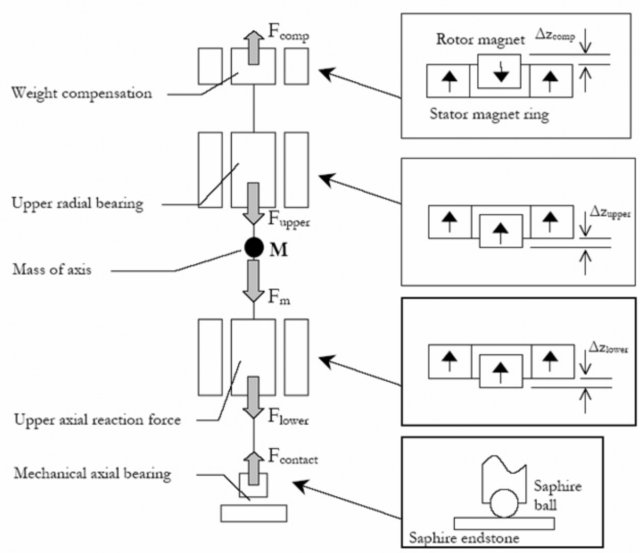

Fig. 6 Concept of passive magnetic bearing, showing the concepts of the radial bearings, the mechanical axial bearing and the weight compensation

The bearing reaction forces F_{upper} and F_{lower} can be quite important if one designs a radially stiff bearing. These forces, adding to the proper weight of the axis F_m, solicit the jewel single ball bearing. That is where the weight compensation comes in, taking up most of the axial force.

The sapphire bearing takes the contact force $F_{contact}$:

$$F_{contact} = F_{upper}\left(\Delta z_{upper}\right) + F_{lower}\left(\Delta z_{lower}\right) + Mg - F_{comp}\left(\Delta z_{comp}\right) \tag{2}$$

g being the gravity constant.

Prior to the dimensioning of the passive bearings, the limits of the jewel bearing have to be known (figure 7).

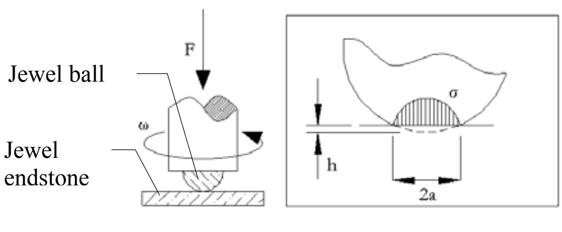

Fig. 7 Touchdown jewel bearing of passive magnetic bearing

Most common material for jewel bearing is ruby (Aluminium oxide plus some corundum impurities). Its approximated mechanical properties are given in table 3:

TABLE III
MECHANICAL PROPERTIES OF RUBY

Property	Symbol	Value	Unit
Maximal compressive stress	σ	2.0	GPa
Poisson ratio	ν	0.3	-
Young modulus	E	400	GPa
Fricion coefficient ruby / ruby	μ	0.12	-

Choice: A 5mm diameter ruby ball is chosen because of its availability and manageable size.

An approximation of the contact radius a between a sphere and a flat plane is given by

$$a = \sqrt[3]{\frac{1.5\left(1-\upsilon^2\right)Fr}{E}} \tag{3}$$

with r being the sphere's radius.The reduced height of the sphere, due to the pressure, is therefore approximated by

$$h = \sqrt[3]{\frac{2.25\left(1-\upsilon^2\right)F^2}{E^2 r}} \tag{4}$$

The pressure stress, known as Herzian pressure stress, is defined as

$$\sigma = \frac{1}{\pi}\sqrt[3]{\frac{1.5FE^2}{r^2\left(1-\upsilon^2\right)^2}} \tag{5}$$

It was found that a maximal axial load of about 6.5 N can be applied. To be on the safe side of things, a maximal load of 4 .5 N is imposed (30% safety factor). As for the heat dissipation (friction losses), the dissipated heat is around 0.5 W at the top speed of 200'000 rpm. Moreover, the torque necessary to spin the rotor was computed to be 17.5 μNm.

The passive magnetic bearing must assure the frictionless support of the primary rotors. Passive magnetic bearing do not have any inherent damping, so an additional radial damping is added by means of a inner copper tube. This provides an effective radial damping by Eddy current induction. Figure 8 sketches this principle and presents the parametrisation of the passive bearing.

The outer diameter of the axis D_{axis} is critical to the rotation speed. As decided previously, the primary axis is supposed to rotate with 200'000 rpm.

Material of choice for the axis is the EN AW 7075 AlZn5,5MgCu alloy with a yield strength of at least 420 GPa. This alloy is easily processable.

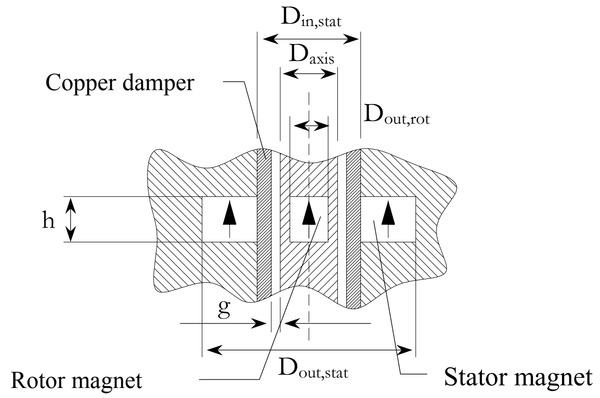

Fig. 8 Parametrisation of radial passive magnetic bearing

with

$D_{out,\ stat}$	Outer diameter of stator magnet	[m]
$D_{in,\ stat}$	Inner diameter of stator magnet	[m]
h	Height of bearing magnets	[m]
D_{axis}	Outer diameter of rotor	[m]
$D_{\mathrm{out,rot}}$	Outer diameter of rotor magnet	[m]
g	Airgap between rotor and stator	[m]

The airgap g has to be processable. 0.5mm is a reasonable small airgap, allowing high magnetic forces and yet some clearance between rotor and stator. The outer diameter of the axis D_{axis} is critical to the rotation speed.

Analytical analysis showed that with the following parameters, the system is stable, incuding a 15% security margin to yield stress: Rotor alloy: EN AW 7075, „Perunal" (AlZn5,5MgCu) with density ρ = 2660 kg/m^3, and a yield limit of σ_y = 400 GPa. Rotor diameters are for the rotor magnets $D_{out,ro\ t}$= 10mm and for the axis D_{axis}= 12mm.

The thickness of the additional copper damper is set to 1 mm. Therefore, the inner diameter of the stator magnet is $D_{in,\ stat}$ = 15mm.

The maximal outer diameter of a stator magnet equals the diameter of the friction disc, 50 mm that is. This diameter cannot be reached because of construction reasons, hence the outer diameter of the stator magnets is set to $D_{out,\ stat}$ = 46 mm. Furthermore, the height of the magnets, h, is set to an available dimension: h = 10 mm.

By finite element simulaton we find that such a combination has a radial stiffness of about 4500 N/m.

Now it is necessary to determine the desired radial stiffness of the passive magnetic bearing:

The magnetic attraction force between spindle and Halbach magnet array F_{mag} was found to be 0.9 N. Be the maximal allowable radial displacement of the friction discs 0.1mm (due to the magnetic attraction of the spindle that tends to separate the friction discs). If can be calculated that the required radial bearing stiffness equals 27.2·10^3 N/m.

This is the required radial stiffness of one axis, consisting of two passive bearings (upper and lower as seen in figure 6). It is evident that just one bearing element as proposed in figures 8 is not apt to deliver the desired stiffness. The solution is to stack multiple bearing elements as shown in figure 9:

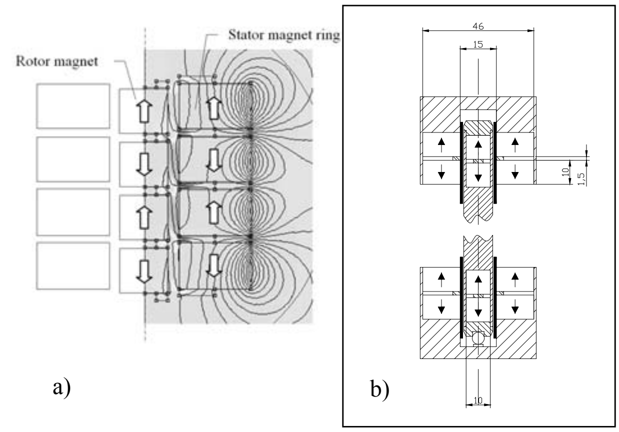

Fig. 9 a) Principle of stacked passive magnetic bearing. Shown are four layers. b) The final bearing design.

Finite element simulation revealed that two layers are apt to deliver 19'055 N/m radial stiffness, sufficient to satisfy the imposed conditions. Using two of such bearing per axis, the obtained radial stiffness is 38'910 N/m, using NeFeB grade 48 magnets.

Figure 10 depicts the concept of a permanent magnet weight compensation, as already suggested in figure 6:

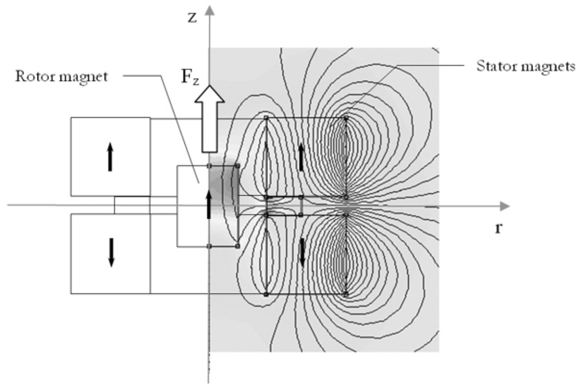

Fig. 10 Concept of weight compensation

The radial stiffness of one axis being 19'055 N/m, the axial stiffness is about the double: 38'110 N/m. If both, the upper and the lower bearing have an offset of 0.2mm, then these bearing develop a force of 7.62 N per bearing, that is 15.244 N per axis. This is the force supported by the jewel bearing. As seen previously, such a ruby bearing should not be loaded with more than 4.5N, therefore the weight compensation should block about 13.75 N.

By finite element simulation it can be seen that not even two stator magnet rings (figure 10) are necessary, the required force can already be obtained with one ring and an offset between rotor and stator weight compensation magnet of 1.7 mm.

IV. RESULTS

Figure 11 shows the final magnetic suspension system for the high speed spindle:

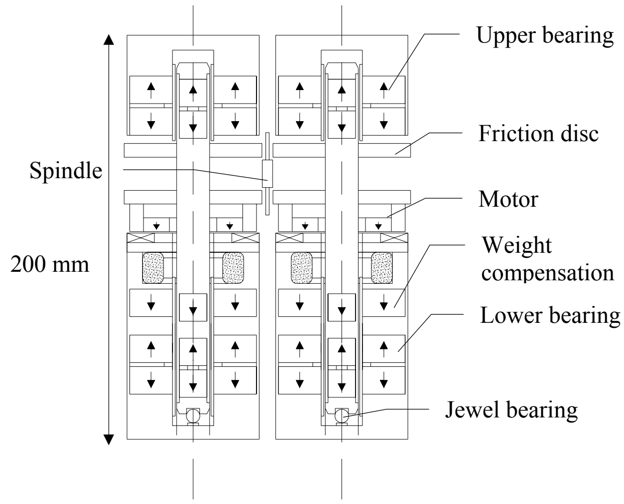

Fig. 11 Overview of magnetic suspension system

The system was modelled with quadratic elements including gyroscopic effects and it was shown, using Campell diagrams, that the four regions where it is safe to operate the systems are 120 – 140 krpm, 190-200 krpm, 240-250 krpm and 320-340 krpm respectively. The last two regions are not of interest due to the discussed material limitations. Hence, from also from a rotor-dynamic point of view, the friction discs can be driven at 200'000 rpm.

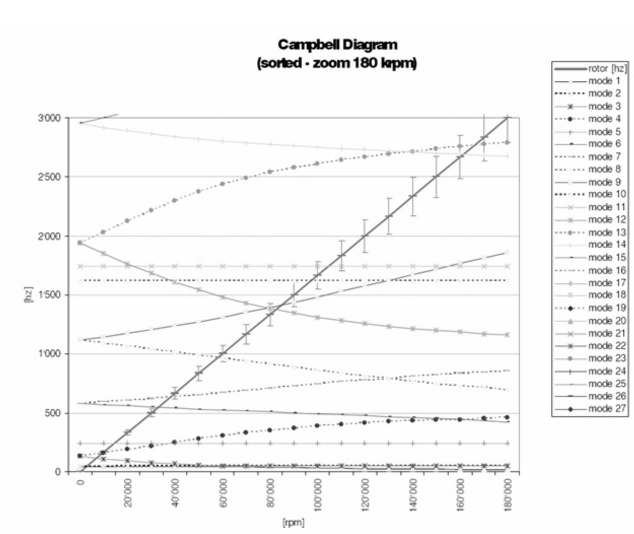

Fig. 12 Campell diagram from 0 to 190'000 rpm of the friction disc

The two speed ranges of interest are each bounded by modes whose shapes involve an important flexural deflexion. Between the target speed of 200'000 rpm (for the friction discs) respectively 7 Mrpm for the spindle, 12 different modes are exited, therefore it is important to swiftly accelerate through the critical rotation speeds.

Figure 12 shows the Campell diagram from 0 to 190'000 rpm of the friction disc.

V. CONCLULSION:

With the goal of reaching rotation speeds above 1 Million rounds per minute, a friction based motor on passive magnetic bearings was presented in this report.

The motor concept is not novel, it found a broad industrial application in the field of the textile yarn processing industry of the 1970ies. Instead of the here presented spindle, a hollow yarn guide (quite similar to a small tube) was spun at speeds above 300'000 rpm. The intention is to push this limit further by using modern materials and passive magnetic bearing technology.

The design and the material choices of the spindle and its magnetic suspension were reported. Also, the friction drive and the dimensioning of the passive magnetic bearings were presented.

It was shown by analytical and finite element analysis means that the proposed device is theoretically capable of rotating its spindle at speeds up to 7 million rounds per minute.

Future work consists of the fabrication of the device and its experimental validation.

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