Investigation on the Possible Use of Magnetic Bearings in Large Direct Drive Wind Turbines

G. Shrestha   **H. Polinder**   **D. Bang**   **A.K. Jassal**   **J.A. Ferreira**

g.shrestha@tudelft.nl, h.polinder@tudelft.nl, d.j.bang@tudelft.nl, a.k.jassal@tudelft.nl, j.a.ferreira@tudelft.nl

Electrical Energy Conversion Group, DUWIND, Delft University of Technology

Abstract

A direct drive generator used in wind turbine has high energy yield compared to other drivetrain topologies and low maintenance is expected as the technology matures. On the other hand direct drive generator weight and size increases rapidly when scaled up to larger units. This paper will investigate the possibility of weight reduction of large direct drive generator using magnetic bearings without compromising maintenance. Three different configurations using magnetic bearings i.e. single magnetic bearing concept, flexible rotor and hybrid concept have been analyzed. The results show that a combination of magnetic bearings and mechanical bearing with stiff radial structure and axial flexibility i.e. hybrid concept has the possibility to reduce weight of direct drive machine significantly compared to conventional design.

Keywords: Direct drive, magnetic bearing, generator, wind turbine

1 Introduction

Direct drive generator has been getting a lot of attention in wind turbine application. Some reasons for it are direct drive machine have the highest energy yield [13], and direct drive machine inherently have less components and lower speed. Thus less scheduled and unscheduled maintenance can be expected as the technology matures.

On the other hand direct drive generator is large and heavy compared to other generators. The weight increases rapidly as the size of the wind turbine increases. Figure 1 shows the expected weights of wind turbines for sizes till 20MW. The scaling is based on the design of generator in Zephyros Z72 direct drive wind turbine [10]. The weight will differ depending on the design used

and the aspect ratio. Similar results can be seen from data in other literature which is summarized in [7]. As the size of the wind turbine increases the weight and cost of the structural components dominates the cost of the direct drive machine. Apart from the component cost, the cost of logistics and the overall tower head mass is affected by this.

There is a lot of literature on possible ways to reduce the weight and size of direct drive machine. The authors have summarized the different possibilities [8], and their drawbacks. Few literatures have also reported on direct drive machine with magnetic bearings for slow rotating devices such as hydro turbines & ship propulsion [16], [17] & [18]. It is reported that magnetic bearing has also been used in large diameter rings of about 6m for certain applications [6]. But application of magnetic bearing for weight reduction has been reported in one patent [19]. This patent claim of weight reduction has been made for direct drive machine in wind turbines with a large diameter hollow hub.
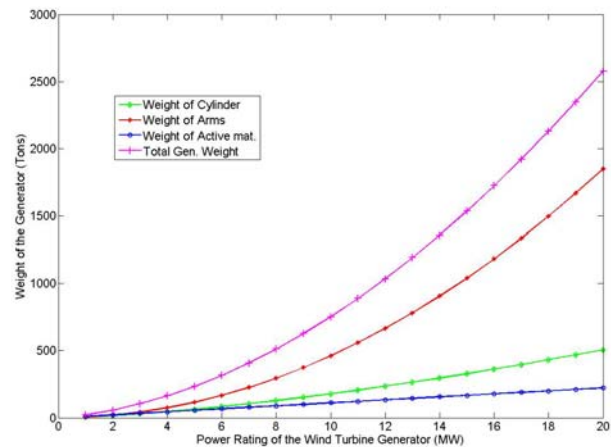

Figure 1: Estimated weight of DD generators in wind turbines (1-20MW)

The objective of the paper is to show some of the results of investigation into magnetic bearing technology as a solution for weight reduction in direct drive machine. Magnetic bearings are non contact type of bearing, so it is assumed that with a matured magnetic bearing technology the maintenance will remain low. Similarly it is assumed in this investigation that when considering the costs of material and logistics required for large direct drive machine, the cost of magnetic bearing can remain competitive.

The paper firstly introduces the magnetic bearing technology. Three different concepts using magnetic bearings that were investigated are listed. Motivation for each concept along with the analysis and results of the investigation are presented. The focus for investigation has been mainly on the scalability. Therefore the detail design models are not presented here but are referred to other papers by the author. Then a preliminary conclusion is derived with reference to future work in this field.

2 Magnetic Bearing Technology

Magnetic bearing in its simplest form in 1 degree of freedom can be described as a pair of controlled electromagnetic actuator as shown in figure 2. It is based on the principle of attraction between the iron and electromagnets. The currents in the electromagnets are controlled with a feedback of the gap sensor signal between the stator and rotor.
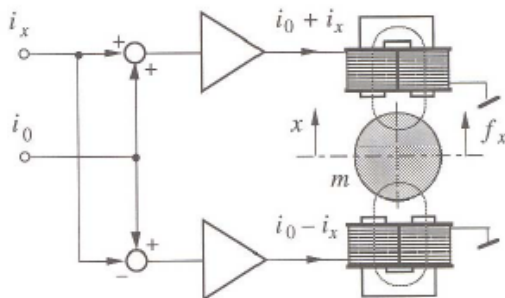

Figure 2: A magnetic actuator in differential mode [6].

There have been developments in magnetic bearings to use permanent magnet biasing for loss reduction, electrodynamic bearings, integrated machine and bearing i.e. bearingless machine. A basic overview can be found in [6] and [14].

Magnetic bearings have successfully been used in various applications such as large compressors to small heart pumps [6]. The motivation for the use has mainly been to achieve high speed, accuracy and/or low maintenance. Therefore the use of magnetic bearing for low speed system like the direct drive wind turbine is unconventional and new. For analysis in this paper the authors have relied on the use of conventional electromagnetic type magnetic bearings. Other aspects like integration and loss reduction are the likely steps after the proof of concept.

3 Concepts

Three different concepts using magnetic bearings were investigated. They are

    3.1 Single Magnetic Bearing Concept
    3.2 Flexible Rotor Ring Concept
    3.3 Hybrid Concept

3.1 Single Magnetic Bearing Concept

Description: The concept drawing is given in figure 4. The generator and the wind turbine hub are fully levitated using magnetic bearings. Basically the idea is to replace the mechanical bearing with a magnetic one.

Motivation: A hub of a wind turbine is of relatively large diameter and is stiffly constructed. A hub of a 5MW wind turbine can have a diameter of about 4m and it has high stiffness to carry the wind forces and moments. The direct drive generator rotor could be integrated to this stiff structure therefore reducing the amount of structural material. Also the shaft can be removed due to this integration. However this requires a large diameter bearing being used. Conventional mechanical bearing of such large diameters can be costly and heavy. Apart from that, reliability of such bearing can be a problem because of the high velocity of the mechanical bearing roller elements. Similarly mechanical bearing tolerance is in the micrometer range. Achieving such tolerance level can be expensive for a large diameter bearing. Therefore magnetic bearings being contactless could be used in such a case.
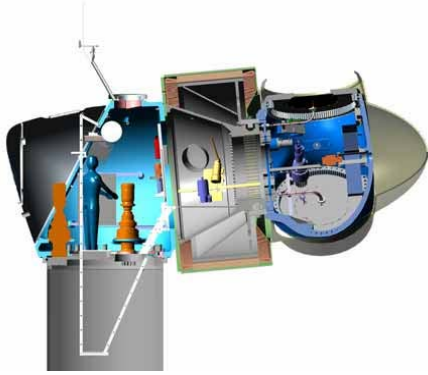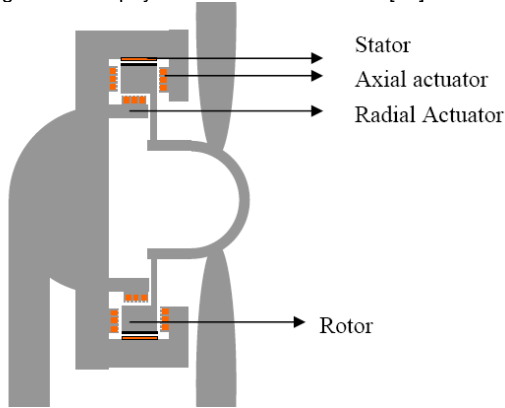
Figure 3: A Zephyros Z72 DD wind turbine [10]

Figure 4: A wind turbine concept with magnetic bearings.

Analysis and results: In order to compare the weight and losses in a magnetic bearing, an analytical model of the magnetic bearing for a magnetically levitated wind turbine was developed. A state of art Zephyros (now Harakosan) Z72 1.5MW direct drive wind turbine shown in figure 3 was chosen as a reference.

The r.m.s. value of force and moments at the hub due to wind in a 1.5MW wind turbine along with some other wind turbine parameters are given in table 1. The wind force and moments at the hub can be found using commercial software like Bladed.

Force Due to Wind	
Radial force (rms)	100kN
Axial Force (rms)	200kN
Bending moments (rms)	800kNm
Some parameters of Zephyros Z72 [10]	
Weight of the blades	4600*3 kg
Weight of the hub	19 tons
Weight of the generator stator	25 tons
Weight of generator rotor	12.5 tons
No. of poles	60
Axial length (Active)	1.2 m
Outer diameter	4 m
Weight of bearing	**4 tons**
Bearing support cone	**5 tons**

Table 1: Some parameters for magnetically levitated design

The values of the analytical design of the electromagnetic actuator are given in table 2. The design is based on the force calculated using the principle of virtual displacement. The magnetic actuator is designed as a U core as shown in figure 5. The details on the design can be found in [7].
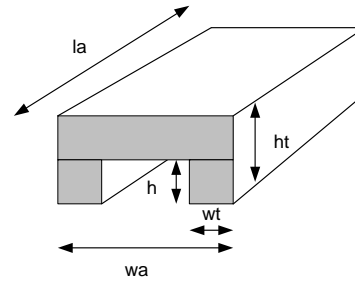

Figure 5: A U core magnetic actuator

This comparison for a 1.5MW machine looks promising. The weight of the magnetic bearing is about 22% more that the weight of mechanical bearing. The mechanical bearing weight includes the weight of the bearing cone is given in table 1 (9 tons). There is room for further weight reduction by using E core type actuators. The weight reduction due to integration has not been considered at this point. The losses for the magnetic bearing are less than 0.5% of the total power rating. This is also a typical loss for mechanical bearing.

However there are some problems with this concept when the wind turbine is scaled up to larger power levels. Therefore the scaling laws for such a wind turbine are analyzed. The power P and force F from wind have a quadratic relation with radius of the blade R_{blade} and generator radius R as given in equation 1 and 2 [15].

$$\frac{P_1}{P_2} = \frac{R_{blade1}^2}{R_{blade2}^2} = \frac{R_1^2}{R_2^2} \qquad (1)$$

$$\frac{F_1}{F_2} = \frac{R_1^2}{R_2^2} \qquad (2)$$

Design Parameters	
Flux density (B)	1.2T
Airgap length (s)	3mm
Current density (J)	4A/mm^2
Height of the tooth (h)	70mm
Flux density in the yoke (B$_{yoke}$)	1.2T
Copper fill factor (k$_{fill}$)	0.4 (round conductor used)

Number of turns (n)	100	
Axial length of the actuator (la)	1m	
Magnetic Actuator Design		
	Axial	Radial
Force required (F)	270kN	100kN
Mmf required (ni)	5.73kAT	5.73kAT
Current required (i)	57.3A	57.3A
Area of the pole shoe (A_l)	0.235m^2	0.0873m^2
Width of the actuator tooth (wt)	23.5cm	8.73cm
Width between two tooth (wa-wt)	8.3cm	8.3cm
Width of the yoke (ht-h)	23.2cm	8.73cm
Radius of the wire	2.1mm	2.1mm
Length of the conductor	247m	217m
Volume of iron	0.1635m^3	0.0347m^3
Volume of copper	0.0035m^3	0.0031m^3
Weight of the actuator	1.29 tons	295 kg
Loss in the actuator (Bias) (100^0C)	255W	224W
Maximum loss (100^0C)	1kW	897W
Total Magnetic Bearing		
No. of radial actuator		4
No. of axial actuator		8
Total weight of the Magnetic bearing		11.4 tons
Total maximum loss in the system		5.7kW

Table 2: Magnetic bearing design [7]

The force produced by an electromagnet F_{em} is given in equation 3, where n is the number of turns, i is the current, s is the airgap and A is the surface area of a tooth. Thus the force is proportional to the surface area of the tooth. Similarly the copper loss in the magnetic actuator is proportional to the resistance of the coil. This is because the current in the coil is nominal as the flux density is assumed nominal. The resistance R of the coil is given by equation 4 where l_{cu} is the length of the wire and A_{cu} is the cross section of the wire. Thus the loss in the bearing is proportional to the surface area of the tooth too. The scaling law is therefore given in equation 5 where m_{bear} is the weight of the magnetic actuator and $P_{lossbear}$ is the power loss of the magnetic bearing. The losses and the weight grow proportionally to the power of the wind turbine. So the scaling of the magnetic bearing doesn't seem to be the problem.

$$F_{em} = \mu_0 \frac{1}{4} A . \frac{n^2 i^2}{s^2} \qquad (3)$$

$$R = \rho \frac{l_{cu}}{A_{cu}} \qquad (4)$$

$$\frac{m_{bear1}}{m_{bear2}} = \frac{P_{lossbear1}}{P_{lossbear2}} = \frac{R_1^2}{R_2^2} = \frac{P_1}{P_2} \qquad (5)$$

Next the structural design of the machine should also be considered. A conventional direct drive machine stator and rotor can be modeled as a cylinder supported by a number of arms as shown in figure 6.
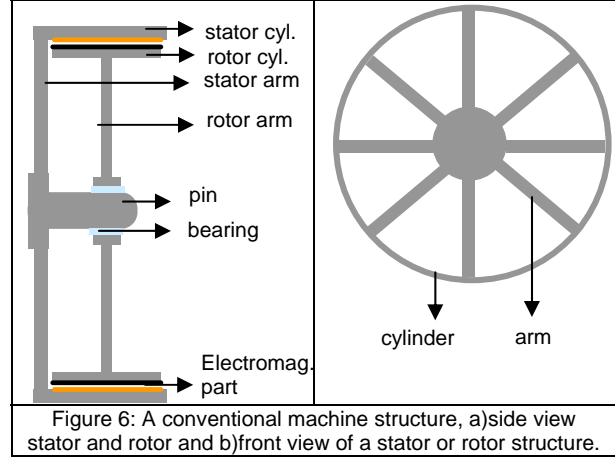

Figure 6: A conventional machine structure, a)side view stator and rotor and b)front view of a stator or rotor structure.

As the power rating of the machine increases the length of the support arms increases. For a magnetically levitated machine, the stator part will remain the same even though the rotor and hub are integrated. As the airgap of the machine is very small compared to the radius of the machine, a very stiff construction is required in order to keep the deflection within an allowable limit. The normally used airgap is about 1/1000th of the machine diameter and the allowable deflection of the rotor and stator is also very limited both on the axial and radial direction.

The magnetic bearings are kept near the airgap, so the wind forces are transferred from the rotor to the stator near the airgap. Therefore the stator arms of the stator can be modeled as a cantilever beam as shown in figure 7. The deflection of the beam is a superposition of the deflection due to self weight and a point force at the point where the magnetic actuator is acting. The point force deflection y_{pf} and distributed force deflection y_{df} of such cantilever beam is given in equation 6 and 7 [2] where F is the point force, q_g is the force per unit length, l is the length of the beam i.e. the radius of the machine, E is the Young's modulus and I is the area moment of inertia. The constant k_1 relates the proportionality between the force F and radius R of the generator support arm. Similarly k_2 relates the length of the beam and the point at which force is applied in the beam. The area moment of inertia of beam depends on the cross section of the material used as given in equation 8,

where k_{beam} is a constant that relates the different geometric shape used and d is the width of the beam. The mass of the arms m_{beam} can be found using equation 9 where n is the number of arms. It is assumed that there is a fixed ratio between the diameter of the machine $2R$ and the width of the beams or arms d. The scaling rule for such beam models can be derived using equation 6, 7, 8 and 9 and are given in equation 10 and 11 respectively. The scaling law for equation 10 can also be used to scale the stator and rotor cylinder of the machine shown in figure 6. The scaling law should be used by identifying the percentage contribution of each force component.

$$y_{pf} = \frac{-Fl^3}{3EI} = -\frac{k_1 R^2 k_2 R^3}{3EI} \tag{6}$$

$$y_{df} = \frac{q_g R^4}{8EI} \tag{7}$$

$$I = \frac{k_{beam} d^4}{12} \tag{8}$$

$$m_{beam} = nR k_{beam} d^2 \rho \tag{9}$$

$$\frac{m_{beam1}}{m_{beam2}} = \frac{P_1^{1.5}}{P_2^{1.5}} \tag{10}$$

$$\frac{m_{beam1}}{m_{beam2}} = \frac{P_1^2}{P_2^2} \tag{11}$$

It is seen that the weight of the beams increases more than linearly with the increase in power rating of the machine. A conventional machine shown in figure 6 can be modeled as a cantilever beam of the axial side and the deflection can be estimated using equation 7. So only the scaling law in equation 11 is valid in this case. This is because the wind force does not load the generator. Therefore a large amount of material is required to keep the deflection within the allowable limits as the size of the wind turbine increases for both cases. However the increase in weight will be more pronounced in the single magnetic bearing concept because the deflection is a superposition of equation 6 and 7 as said earlier.
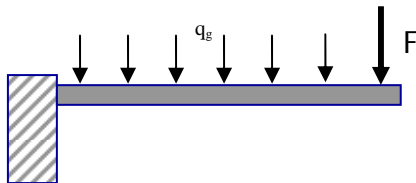

Figure 7: A model of a stator arm as a cantilever beam.

3.2 Flexible Rotor Ring Concept

Description: The idea for this concept is to have a rotor made of thin cylindrical ring. The cylindrical ring is supported by a number of controlled magnetic actuators. The actuators are supported on the stator support structure. The concept is shown in figure 8.

Figure 8: Concept with distributed magnetic actuators (front view)

Figure 9: A NewGen concept [5]
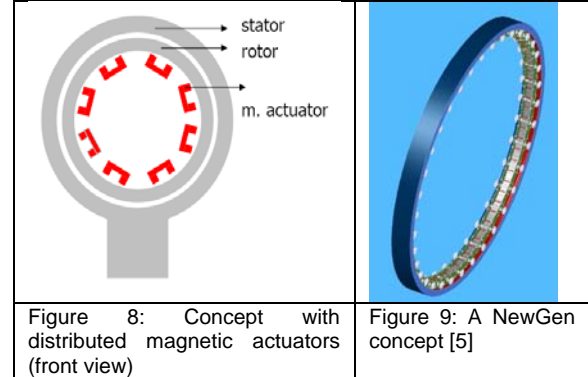
Motivation: Flexibility has been used to reduce weight of buildings i.e. skyscrapers and in other wind turbine components such as the blades and tower. A flexible structure in direct drive generator has also been proposed in literature [5] and is known as the NewGen concept as shown in figure 9. The authors claim the weight of such a generator to be about 30% of the conventional generator weight. In this concept the rotor is supported by the wheels revolving about a fixed rail, like in railway system. The main problem for such a concept is maintenance requirement. Similar railway components require regular reprofiling [12]. Distributed magnetic bearings could be used to replace the wheels like shown in figure 8.

Analysis and Results: A 5MW direct drive machine was designed analytically. A rotor was made flexible allowing the self weight deflection to be about 5% of the diameter of the machine. The number of magnetic actuators was chosen on the radial side based on the allowable deflection of 5% of the airgap. The weight of the rotor was about 20% of the rotor weight of the comparable rotor in literature [4]. The comparable rotor with an aspect ratio of 0.1 is given to be 50 tons. The details on the analytical design are not given here. More details on the analytical design using this concept can be found in [9].

However such flexible rings will have bending mode frequencies at very low frequencies. The

first two bending modes are shown in figure 10. The bending modes for the rotor as given in table 3 occur at 1.6 Hz and 4.3 Hz for the first and second bending mode respectively. The bending modes of such low frequency should be controlled. The second bending mode is a lateral buckling mode shown in figure 10b. Such bending cannot be controlled by 1 magnetic bearing as shown in figure 8 in a 2 dimensional plane. A pair of magnetic actuator is required on each point to control the radial side bending in the 3 dimension. The side view of the magnetic actuators configuration for such a case is shown in figure 11.

Airgap Radius	4.175m
Axial length	0.83m
Flux density at air gap (fundamental)	0.98T
Airgap length	8mm
Maximum deflection allowed	0.8mm
Thickness of ring	45mm
Maximum deflection due to gravity	3.7%
Maximum deflection due to normal force	0.35mm
Force due to eccentricity (0.8mm)	113kN
Number of actuator	12
Force per actuator	29kN
Maximum deflection	0.285mm
Weight of the rotor including magnet weight	10.2 tons

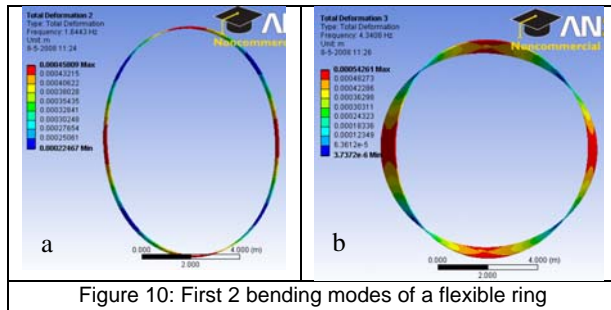Table 3: Some design values of a flexible rotor [9]

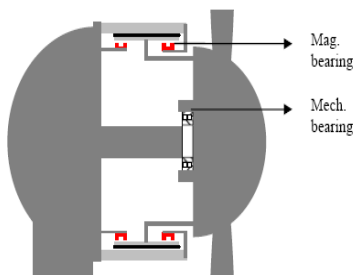Figure 10: First 2 bending modes of a flexible ring

Figure 11: A distributed magnetic bearing configuration.

This means that 24 controlled magnetic actuators are required in total to levitate the rotor. Controlling such large number of magnetic actuators in a coordinated fashion will be very complicated.

3.3 Hybrid Concept

Description: The hybrid concept has a mechanical bearing to take the wind loads and a stiff rotor ring that is magnetically levitated. This concept is shown in figure 12.
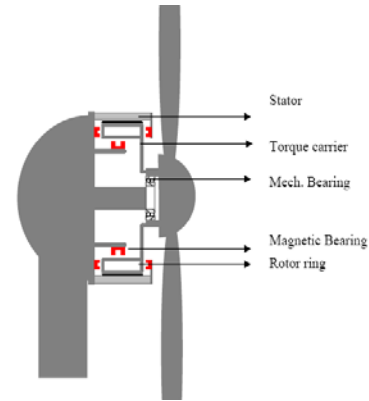

Figure 12: A hybrid magnetic bearing solution concept.

Motivation: From the previous analysis the totally magnetically levitated concept has the problem of structural weight and the flexible rotor concept has the problem of control complexity. Therefore a hybrid solution is investigated. The concept turbine is shown in figure 12. In this concept the mechanical bearings take the wind loads and transfer it to the tower. The rotor ring is made light and stiff using hollow structures. The magnetic bearings keep the airgap of the machine and controls forces in 5 degrees of freedom. The torque carrier carries the torque from the wind turbine to the generator. This torque carrier differs from the conventional arms of a direct drive machine. This is because the torque carrier carries only a single directional force, whereas an arm of a conventional machine is loaded in multiple directions.

The arms of a large direct drive machine are the major weight component as seen in figure 1. Eliminating them can reduce the weight of the direct drive machine substantially. Other advantages apart from weight reduction are as follows

i. gives more room for flexibility for the shaft/pin.
ii. No special mechanical bearing required.

Analysis and Results: A 5 MW direct drive machine was designed analytically. The analytical designs were done for both active and structural material of the machine including the magnetic bearing components. Then FEM method was used to verify some parts of the model. A summary of the component designs are given below. Some values are given in table 4. The details on such design concept can be found in [8].

a. Hollow ring: The rotor ring is made of hollow rectangular sections. Such rings are stiff in the radial direction but weak in the axial side. Therefore flat plate stiffeners were added to stiffen the rotor also in the axial direction. The first bending mode is kept at about 40Hz. The first bending mode frequency of the rotor ring is kept 30% above the generator frequency at the rated rotational speed. This is done to ensure that the rotor natural frequecy is not excited by other components of the wind turbine. The generator frequency is 25Hz. The natural frequency of other wind turbine components are much lower. The rotor ring structure is shown in figure 13. Some design values of the hollow ring is given in table 4.

b. Magnetic bearings: The magnetic bearings were designed to control a destabilizing force of 600kN, which is equal to the force due to eccentricity of the rotor of 4mm on the radial side. The force due to eccentricity of the rotor is given in figure 15. The magnetic bearings are designed as E-core type as shown in figure 14 in order to reduce the weight which leads to a bit higher losses. Some design values for magnetic bearings are given in table 4.
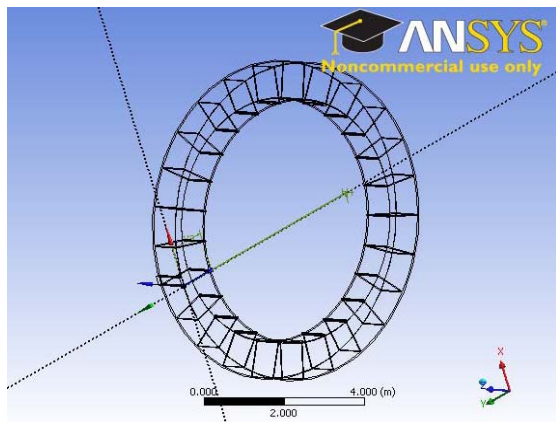

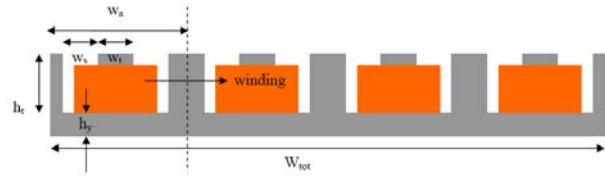Figure 13: A hollow ring skeleton view with stiffener.
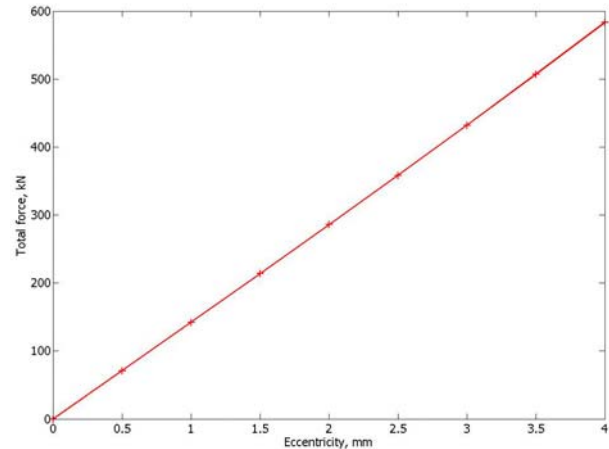

Figure 14: An E-core magnetic actuator

Figure 15: Normal force due to eccentricity of the rotor

c. Torque carrier: The torque carrier is designed using I beam structure because such structures are more efficient in carrying single directional force. Some design values of the torque carrier for a 5MW design is shown in table 4.
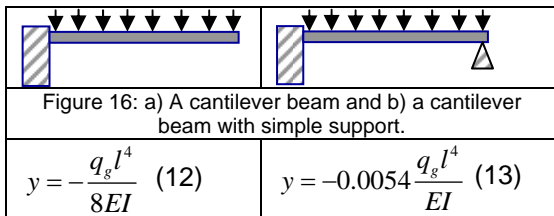
Wind turbine specification			
Rotational speed	12.5 rpm	*Torque carrier*	
Power rating	5.5MW	No. of torque carrier	6
Electromagnetic Design		Torque per arm	0.64MNm
Aspect Ratio	0.1	Structure type	I beam
Peak flux density at airgap	1T	Torque carrier weight	~1.75 tons
Airgap radius	4.175m	*Magnetic bearings*	
Airgap length	8mm	No. in radial side	4
No.of pole pairs	120	No. in axial side	8
Structural Design		Actuator type	E core
Rotor Ring		Nominal air gap change	2mm
Structure type	HRS	Maximum air gap change	4mm
Hollow ring thickness	25mm	Total actuator weight	~6 tons
Plate stiffener	24	Losses in the magnetic bearing	0.1-0.3%
Deflection of the ring	<10% of gap	**Total structural weight of the rotor**	**~28 tons**
Weight of the rotor ring	~20 tons		

Table 4: Design values of a 5MW generator design with hybrid concept [8]

The mass of the rotor using this concept is about 45% of the comparable rotor given in the

literature for the same power rating and aspect ratio [4]. The losses are less that 0.5% of the total power. The design could be optimized for further weight reduction. It should be noted that the analysis here has focused on the rotor only.

The result of a 5MW hybrid concept is promising but it is of interest to know the scalability of such concept. The hybrid type generator is not affected by the wind loads therefore it can be assumed that the generator is loaded on the axial side by its self weight only. This is also true for conventional machine design as shown in figure 6. Thus a conventional direct drive design can be modeled as a cantilever beam as shown in figure 16a. The magnetic bearing support of the hybrid concept can be modeled as a simple support on the other side of the beam as shown in figure 16b. The equation governing the deflection of cantilever beam for self weight is given in equation 12 and the beam with simple support at one end is given in equation 13 [2].

Figure 16: a) A cantilever beam and b) a cantilever beam with simple support.

$$y = -\frac{q_g l^4}{8EI} \quad (12)$$

$$y = -0.0054 \frac{q_g l^4}{EI} \quad (13)$$

It can be seen that the beam with a simple support at one end has 23 times less deflection that the cantilever beam. Thus even though the scaling law of equation 11 is still valid in this case the starting value for the scaling will be order of magnitude lower than the conventional machine.

4   Discussion

The analysis mostly focuses on that rotor side only. The analysis shows that the hybrid solution can be a new concept to reduce the weight of the generator. It gives axial flexibility to the rotor structure.

It should be noted that the scaling laws for all the designs remains the same but the constants used for the designs change. The constants of the equations can be changed by using efficient structures therefore changing the area moment of inertia of the structure or using other lighter and stiff material. Other method is to allow more flexibility in the design of the drive train.

Lighter materials like composites are not considered at this stage for cost reasons. Efficient structures are already used in large state of art direct drive machine like the Enercon E112. Hollow structures are used for weight reduction in those designs. But such direct drive machine still remains heavy because of very high stiffness.

Therefore the direction for allowing more flexibility in the drivetrain is necessary to reduce the weight further in direct drive designs. This is achieved in the hybrid concept without adding the control complexity. However the stator side is as important for the weight reduction of the direct drive machine. Therefore the next steps for reducing the weight further will be to reduce the stiffness of the structural design on the stator side.

5   Conclusion

The paper has investigated three concepts of direct drive machine using magnetic bearings. The single magnetic bearing concept has the scalability problem especially on the stator side when stiff construction is considered. The flexible rotor concept gives radial flexibility to the machine but the control of such rotor can be extremely complex. The hybrid solution gives the ease of control and at the same time provides flexibility on the axial side. So the hybrid solution with axial flexibility will be investigated further.

6   Acknowledgement

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