

Integrated Structural and Electrical Design of the Linear Permanent Magnet Synchronous Machine

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Abstract

Linear permanent magnet generators are a potentially useful technology for wave power applications. Typically, optimisation and comparison of these generators is based on an electromagnetic analysis with limited regard for the structural analysis. This paper presents a comparison of two alternative designs of the double-sided linear permanent magnet synchronous machine which includes structural and bearing requirements for a more accurate assessment of cost and feasibility. It is shown that both cost and feasibility depend heavily on these issues due to the large internal and external forces acting on the machine.

Keywords: Direct-drive generator, linear synchronous machine, permanent magnet machine, wave energy converter.

Nomenclature

A	= area
B	= magnetic flux density
b	= width
d	= depth
E	= Young's Modulus
g	= airgap
h	= height
H	= power coefficient
I	= 2nd moment of area
k	= stiffness
K	= stress concentration constant
l	= translator width
L	= length
M	= moment
P	= bearing load
p	= number of pole pairs
q	= flow coefficient
R	= reaction force
t	= thickness
w	= force per unit length
x	= bearing land
y	= deflection
γ	= boundary lubrication constant
θ	= angular displacement
ρ	= recess pressure
μ	= permeability

σ	= tensile stress
τ	= pole pitch

Subscripts

A	= relating to leftmost point of beam
a	= distance a
b	= bearing pad
CU	= copper wire turn
d	= sliding distance
e	= stress concentration
l	= distance l
I	= I beam width
m	= magnet
o	= free space
p	= pole
r	= lubrication factor
s	= slot
t	= tooth
T	= total
w	= web
y	= yoke
yp	= yield point

1 Introduction

Wave energy converters (WECs) have working surfaces that reciprocate at low speed and operate over a wide range of loadings making conventional off-the-shelf rotary generators less suitable than in other power generation technologies. Permanent magnet (PM) generators exhibit high part load efficiencies and, while they have been demonstrated at sea to a limited extent, designs are not yet fully optimised. Using direct-drive generators reduces the number of moving parts but, because of their low speeds and consequent high forces/torques, adopting conventional machine topologies leads to large and costly generators.

Optimisation of cost and performance requires integrated structural and electrical design techniques to establish the cost of energy produced. This work explores how the design of the prime-mover, drive-train, and generator may be fully integrated to define lighter, cheaper, reliable machines that will operate at slow speed with improved efficiency over a wide range of loads.

Linear Generator Topologies

A number of different electrical machine types have previously been compared for use in direct-drive WECs. For example, Polinder *et al.* investigated linear versions of induction generators with and without iron in the secondary, switched reluctance generators, PM synchronous generators (with air- and iron-cored windings) and transverse-flux PM machines [1]. Baker and Mueller examined Vernier hybrid machines and air-cored tubular machines [2].

These studies have tended to address the electrical and electromagnetic aspects of the generator design without really tackling the mechanical characteristics of the machines. Work on low speed electrical generators for direct-drive multi-MW wind turbines shows that over 80% of the generator mass is required for the purpose of structural robustness – to prevent the stationary and moving parts of the generator deflecting under the action of magnetic forces – rather than directly contributing to electrical power production [3].

In this paper, two designs are presented for a 0.5 MW linear iron-cored PM synchronous generator based on the electromagnetically optimised designs presented in [1] but expanded to include: (a) structural analysis and design (coupled to the electrical and magnetic models) and (b) analysis of bearing types that might be applicable for the generators. These integrated designs demonstrate how the optimal machine design changes significantly when these aspects are included.

2 Machine Models

Sections of the iron-cored machines are shown in Fig. 2 and Fig. 3. The electrical modelling for this machine is fairly well documented, for example [1]. However, some additional constraints were added to the machine designs in this analysis.

Electrical Design

It is assumed that the generator is attached to a power converter capable of supplying any amount of reactive power in order to keep the phase voltage and currents in phase. In practice, this could result in a large and expensive converter, the cost of which is not considered here. It is also assumed that this converter can present any input resistance at its terminals and that this can be used to limit the current amplitude. Any power converter is likely to require an rms emf from the generator of at least a few hundred volts for operation with acceptable losses. It will be seen that the expected phase voltages from the designs exceed this limit at the rated powers.

The rms current density is limited to a maximum of 5 A/mm² in order to prevent excessive losses and failure through overheating. If the machine design initially exceeds this limit, the current is reduced by the power converter by increasing its input impedance. In all designs 1 mm diameter conductors are used. The copper fill factor of 0.585 assumed in [1] is also used here.

Magnetic and Structural Models

To calculate the forces for the structural analysis, a method based on fitting a polynomial to data generated from finite element analysis (FEA) was used. Full details of this technique are found in [4].

In this case, the variable under consideration was the force per unit area acting between the stator and translator due to Maxwell stresses. This was calculated from the FEA by extracting the average of the square of the flux density along the surface of the iron secondary for the iron-cored machine and calculating the force according to the equation

$$F = \frac{B^2 A}{2\mu_0} \quad (1)$$

Following this, a least-squares fit was performed on the resulting data and a fit with R^2 values greater than 0.95 was achieved for both machines.

The primary goal of any electrical machine structure is to maintain clearance between the stator and translator. In low-speed permanent magnet machines, this presents unique difficulties due to the extremely high flux densities required to achieve a useful power output. In the machine in question, the net force on the armature is theoretically zero provided there is an identical airgap on both sides, and the machine surfaces are perfectly homogeneous. In reality, these exact conditions are unlikely to be true initially due to manufacturing tolerances and later due to bearing surface wear in operation. However, although the net forces on the armature will be zero, the lateral forces acting on each side of the translator will still be large and must be withstood by any structural design.

The increase in force on both the stator and translator with any lateral displacement can be very large due to the normal forces in the machine being proportional to the square of the flux density in the airgap. The flux density in the airgap can also increase non-linearly as the airgap closes (with the rate of increase dependent on the specific geometry). This requires that any structural analysis consider not only the initial forces on components, but any increase in forces due to deflection in those components. An analysis of this type is presented here.

It was decided to provide support for the translator with a series of beams running transverse to the direction of travel (as shown in Fig. 2). There is an extremely wide choice of possible cross-sections for the beams but the basic I-beam cross-section, also shown in Fig. 2 with the pertinent dimensions, was chosen for its efficient use of material and the wide variety of standard cross-sections commonly available [5]. The I-beams are stacked flush to one another with no regard for division by electrical pole. This assumes that the forces acting on the beam are distributed uniformly in the direction of travel.

A practical value of 5 mm is used as the initial airgap. For the purposes of this analysis, it is assumed that the maximum allowable deflection is 10% of this value, due to the uncertainty in the variation in force with displacement. Although a reasonable attempt is

made to estimate the increase in forces due to the closure of the airgap, the degree of uncertainty in the calculation makes it prudent to include a wide margin of safety.

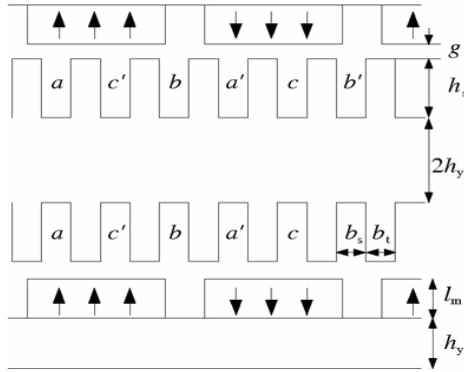


Figure 1: Cross section of linear synchronous permanent magnet machine.

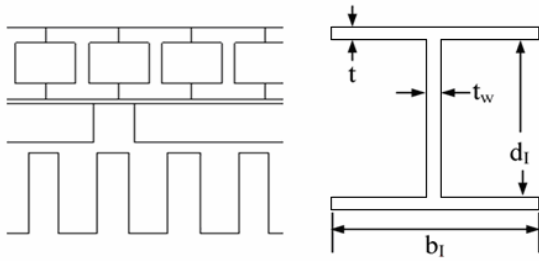


Figure 2: I-beam use in the iron-cored machine, and I-Beam dimensions.

Calculation of Deflection

This following analysis assumes that the bearing system between the stator and translator is infinitely stiff, and that the beams are “built-in” at both ends. This is a reasonable approximation, as in practice the beams are likely to be clamped in place or attached to stiff webs as in Fig. 3. This hypothetical clamping and support mechanism is not included in the cost estimations presented later. Using these assumptions we calculate the deflection due to the loading using standard beam theory [6].

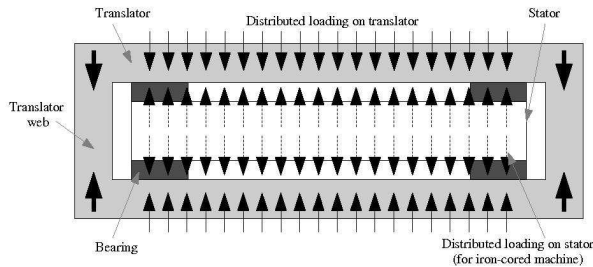


Figure 3: Cross-section of structural design showing translator webs with infinite stiffness.

$$y(x) = \begin{cases} y_A + \theta_A x + \frac{M_A x^2}{2EI} + \frac{R_A x^3}{6EI} & x < a \\ y_A + \theta_A x + \frac{M_A x^2}{2EI} + \frac{R_A x^3}{6EI} - \frac{w_a}{24EI} \cdot (x-a)^4 - \frac{w_l - w_a}{120EI \cdot (l-a)} \cdot (x-a)^5 & x \geq a \end{cases} \quad (6)$$

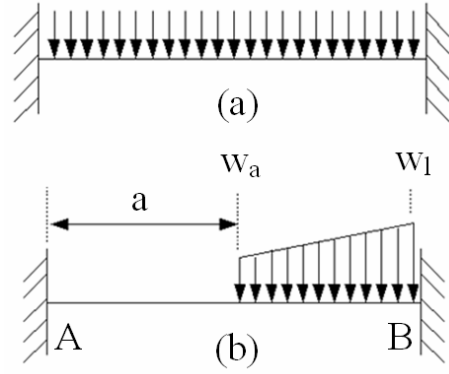


Figure 4: (a) Uniformly distributed force on beam, (b) general case used to evaluate this state.

Standard beam theory allows the superposition of multiple simple cases in order to generate more complex loadings, provided the beam remains in the elastic deformation region. In both machines, the beams will initially undergo a uniformly distributed loading as shown in Fig. 4(a). From [6] the equation for the lateral deflection at a position x length units from the end support of a beam fixed at both ends undergoing a linearly distributed loading (Fig. 4(b)), is given by (6).

For a beam fixed at both ends, θ_A and y_A (the initial angular and lateral deflection at end A) are zero and the reaction forces and moments are given by (2) and (3):

$$R_A = \frac{w_a}{2l^3} (l-a)^2 (l+a) + \frac{w_l - w_a}{20l^3} (l-a)^3 (3l+2a) \quad (2)$$

$$M_A = \frac{-w_a}{12l^2} (l-a)^3 (l+3a) - \frac{w_l - w_a}{60l^2} (l-a)^3 (2l+3a) \quad (3)$$

The second moment of area of an I-beam is given by (4) for the dimensions shown in Fig. 2.

$$I_1 = \frac{b_l^3 t}{6} + \frac{t_w^3 d_l}{12} \quad (4)$$

The 2nd moment of area in the iron secondary is given by (5). This was developed by treating the solid core and tooth separately, then combining them, taking account of the tooth displacement from the neutral axis.

$$I_{ICA} = \frac{h_y \tau_p^3}{12} + 6 \left(\frac{h_s b_t^3}{12} + h_s b_t \left(\frac{h_y + h_s}{2} \right)^2 \right) \quad (5)$$

To approximate the effects of the airgap forces, the beam is split evenly into sections of uniform force. The initial, pre-deflection, forces in each section are calculated by superposition of multiple cases of the type shown in Fig. 4(b).

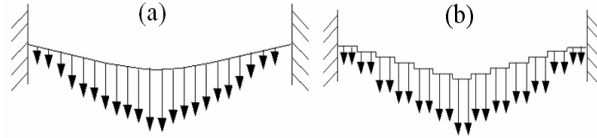


Figure 5: Continuous variation in forces due to deflection in beam (a) and the approximation using linear sections used to evaluate the resulting effects (b).

The deflection due to the initial uniform force (Fig. 5 (a)) results in increased loading due to the reduced airgap along the beam's length. These additional forces are evaluated by treating each section as a different machine of the same dimensions, but with an airgap of the new reduced size, as depicted in Fig. 5 (b). This method assumes the 3D flux path is negligible along the stack length, and 2D FEA was used to obtain the flux density.

The force per unit area from each simulation is multiplied by the area of that section, and it is this force which is assumed to act on that part of the deflected beam. The force previously calculated for that section is then subtracted from the new value, to give only the additional force on the beam due to the deflection. The deflection on a beam due to these additional forces is then calculated and added to the initial deflection. This method is applied to all of the sections, and repeated until the deflection due to additional forces falls below a threshold, or until the airgap is reduced to less than 90% of its initial value, meaning the beam must be rejected.

The choice of possible I-Beams is limited to the standard sections in [5]. These sections are tested in order of increasing cross-sectional area, and hence cost, until a suitable beam is found.

3 Design Comparison

To demonstrate the effect of the beam selection on the cost and feasibility of the machine, a single machine dimension is varied, the stack length l_{st} orthogonal to the pole pitch and direction of motion of the translator. To get the full beam length required, the additional width necessary to accommodate the bearings is added to this. Otherwise, the dimensions of each pole of the machines will be identical, having the values shown in Table 1. These dimensions are similar to those found to be optimal for this type of machine without consideration of the structural aspects in [1].

Stack lengths of 0.5 m and 1.5 m, excluding the bearing width, were chosen to highlight the difference in structural needs between each design. These stack lengths were chosen to be 50% and 150% of the 1 m width chosen arbitrarily in [1]. Although also quite arbitrary, there are some additional reasons for these selections. The lower value is sensible as narrower coil windings produce a voltage that is impractically low for any power converter, and would therefore require some further design alteration resulting in a less clear comparison between machines. The larger width of 1.5 m was chosen to demonstrate that the inclusion of structural requirements is essential in any future

optimisation of the machines. A design of this size could conceivably be built but may have significant disadvantages not included in the electromagnetic design. A stack length of this size could encounter thermal problems not yet accounted for by the simple current density restriction included here. Future work will also take heat dissipation into account for a globally optimised design.

Generator Parameter	Value
Pole Pitch τ_p (mm)	100
Slot Height h_s (mm)	100
Number Of Slots Per Pole Per Phase	1
Stator Slot Width b_s (mm)	17
Tooth Width b_t (mm)	17
Stator Yoke Height h_y (mm)	30
Length Of The Magnet l_m (mm)	15
Airgap (mm)	5

Table 1: Common Generator Dimensions

Calculation of Machine Stiffness

As stated in the previous section, the effect of the bearing needs on the structure must also be included. This can be assessed by determining the allowable deflection as a result of loading when the airgap is unbalanced, and the reaction of the structure to the load put upon it, the structural stiffness. Whether the bearing plays a role in enforcing this property or whether it is affected by the deflection and experiences compression is significant in lifespan prediction. A method for estimating an effective bearing width based on machine geometry is given in section 4.

The structural model assumes the translator to be elastic with fixed ends and undergoing a uniform distributed load due to the Maxwell stresses between the PMs and the iron core as shown in Fig. 3. The magnets on either side of the translator are attracted to the centre of the machine, while the central part is attracted outward in both directions. If there is an identical airgap on both sides, the net force on the stator is zero and the translator webs react all of the internal forces. However, any eccentricity of the stator (such as a manufacturing defects, defects due to bearing wear, corrosion etc.) will lead to the bearings on one side being constantly loaded by the net difference between the unbalanced normal forces. The total forces and net forces for unbalanced airgaps are shown in Fig.7.

Structural Assumptions For Stiffness Model

An electromechanical study to assess the structural stiffness of a lightweight radial flux rotary direct drive machine was undertaken in [7] in order to reduce the total machine mass. Based on this, a similar technique was applied to the linear machine to assess the combined effect of airgap distortion and structural deflection on the bearings.

In [7] the airgap forces are modeled as an equivalent magnetic stiffness, k_m . This stiffness is negative as the restoring force is inversely related to the extension. If this force is greater than the structures' stiffness it will reduce the airgap, increasing deflection in the translator. For the linear PM synchronous machine this stiffness can therefore be described by (7), determined from eqn(10) in [7].

$$k_m = \frac{B_p^2 l_{st}}{4 g \mu_o} 2 p \tau_p \quad (7)$$

This stiffness can then be divided by beam width to give the stiffness under each beam section to compare with the stiffness model developed below.

The structure proposed from the magnetic and structural model is assumed to be stiff enough to handle these loads, however this does not tell us how they are transferred to the bearings. The stiffness model can be used to investigate the extent of this compression.

The initial assumption, as shown in Fig. 4(a), is simplistic as it assumes the ends are rigidly supported and there will be no deflection in the translator web. In reality there will be deflection in both parts. This can be modeled as a distributed load on a simply supported beam with two vertical members, as illustrated in Fig. 6.

The stiffness derivation for the plates and bearing surfaces is the same for all loading conditions, as given in (8 & 9). The contribution of the plate to the overall rigidity is minimal but it provides additional structure to align and connect the beams and to allow for the placement of the magnets on the translator face, shown in Fig. 2.

$$k_{plate} = \frac{F}{\delta} = \frac{EA \delta}{L \delta} = \frac{EA}{L} \quad (8)$$

$$k_{bearing} = \frac{b_l \cdot b_b \cdot E_{sxl}}{t_b} + \frac{b_l \cdot b_b \cdot E_{bronze}}{t_{rm}} \quad (9)$$

The stiffness derivation for the beams with distributed loads is based on beam theory [8] taking the second moment from (4). The translator stiffness, k_{trans} was found from the sum of the individual stiffnesses of the beam and plates added in series and then used to find the overall structural stiffness, $k_{structural}$ made up of $k_{bearing}$ and k_{trans}

$$k_{Structural} = \left(\frac{1}{k_{bearing}} + \frac{1}{k_{trans}} \right)^{-1} \quad (10)$$

Stiffness Reactions Under Load

k_{mag} is in parallel with $k_{structural}$. If eqn (11) is not satisfied the machine will distort during operation. This is to be avoided.

$$|k_{mag}| < k_{structural} \quad (11)$$

For a given magnetic stiffness, there is a trade off between the necessary stiffness of the beam and the bearings so that the inequality shown in eqn. (12) is satisfied.

$$\left| \frac{1}{k_{mag}} \right| > \frac{1}{k_{beam}} + \frac{1}{k_{bearing}} \quad (12)$$

Reducing loads on the bearings reduces wear and improves maintenance/ replacement intervals.

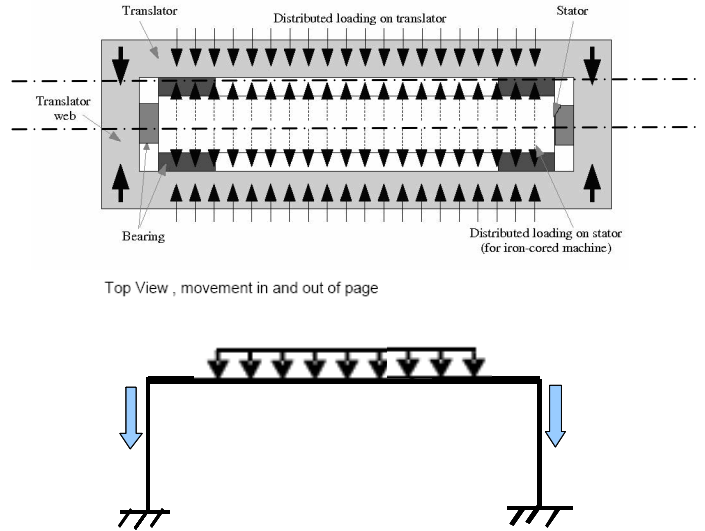


Figure 6: Machine Cross-section (top) and distributed load with vertical members model (bottom) .

For the two machines chosen, the structural analysis was used to determine the geometry of the machine. A number of bearing lengths were determined for each machine based on an empirical zero wear estimate, explained in section 4. Using the stiffness model the reaction of the frame can then be compared for each assumption to investigate whether the bearings need to contribute structurally. This is investigated for the net load due to a given deflection and the maximum load expected under full deflection to provide a prudent analysis. In order to choose these bearing pad sizes, a suitable bearing material must first be identified.

4 Bearing selection for linear generators

A reliable bearing system is required to allow the translator to move freely relative to the stator. The minimum acceptable lifetime between maintenance intervals for this application is chosen to be 12 months.

For a wave energy converter based in the North Atlantic travel is expected to be up to 5,000hrs and 9,444 km per year for a WEC with a 6m stroke [1]. A prediction of 6 active months (26 days), 3 months of medium activity (18 days) and 3 months of minimal activity (1 day) was used based on wave conditions off the coasts of Ireland [8] and Portugal [9]. The design aims for the bearing system are the following:

- System capable of constant operation for long periods
- Low friction contact between moving surfaces, minimal maintenance.
- High tolerances to maintain the machines effective airgap

- Ability to deal with and recover from shock loads
- A design allowing 0.5 mm of wear per year

It is assumed that all wave loading on the WEC is absorbed through bearings external to the linear generator and only the internal forces must be reacted.

Generator Loads

As described in Section 2, the net forces between each side of the stator and translator, and hence the forces reacted by the bearings, are zero, provided the surfaces are perfectly homogenous and the airgap is identical on both sides. However, any reduction in one airgap will result in forces acting between the translator and stator of the side with the reduced airgap. The total pressure acting between the stator and translator of any one side for a given airgap is shown in Fig. 7(a), while the Fig. 7 (b), shows the smaller net closing pressure for an increasing airgap imbalance.

A 10% reduction in the chosen 5 mm airgap, caused by wear or deflection gives a net stress of 22.5 kPa on the reduced airgap side, this will be the basis of the bearing design with comparisons made to the more extreme full closing load of 200 kPa. The resulting force on each strip of bearing material is then found from $(\sigma_n l_s) / (2b_b)$. With a net stress of 22.5 kPa, this corresponds to 16875 N ÷ bearing width and 5625 N ÷ bearing width for machines 1 and 2 respectively.

Bearing Materials

The main bearing materials under consideration are polymer sliding bearings, with water as a lubricant, against a bronze running mate or other non-ferrous corrosion resistant metal. Mating surfaces most suitable for a corrosive environment are bronzes, either Gunmetal or Copper Nickel [10]. The use of commercial sliding contact bearings and roller bearing systems had been ruled out primarily due to the re-lubrication intervals needed for reliable operation.

Collins [11] states that bearing systems involving metal to metal contact require significant lubrication to avoid a range of wear processes such as adhesive, abrasive and corrosive wear, surface fatigue and deformation. On this basis, wear in lubricated and unlubricated systems was assessed, indicating that a well lubricated high viscosity system is essential for reduction in surface wear between low speed metals in contact. Adhesive wear rates were investigated for the required bearing lifespan using materials found in marine applications i.e. graphite, bronze, molybdenum, and Nylon 6, shown as G, B, M and N6 respectively in Fig. 8.

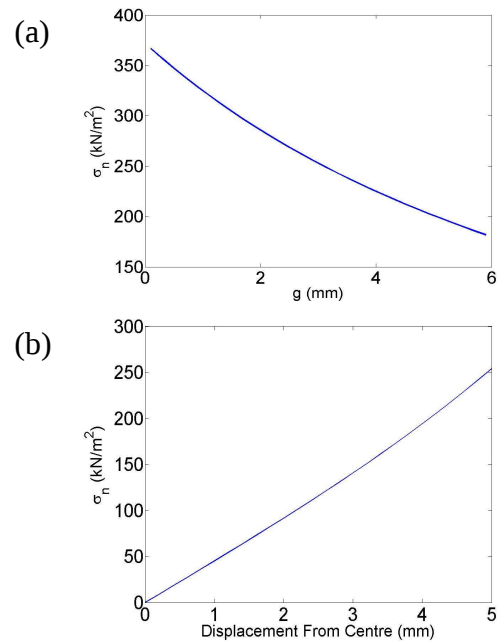


Figure 7: (a) The total airgap closing stress in an iron-cored machine with varying airgap clearance, g . (b) The net stress seen by the bearings on one side (the side with reduced airgap clearance) of a double sided machine with a displacement of 0.1 mm to 5 mm from centre.

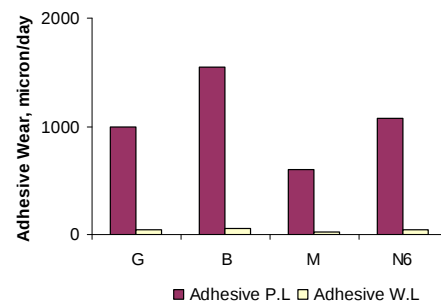


Figure 8: Wear rates of sliding bearings under 22.5 kPa of load.

As detailed in [11] polymers and graphite have lower adhesive wear rates so offer a better solution. When well lubricated they are acceptable for long term contact. However, it is evident from Fig 8. that a few days of poor lubrication will significantly reduce the lifespan and ability of the bearing to maintain the critical airgap with rates of wear much greater than the required 0.5 mm per year. From this it is evident that conventional contact bearings are unsuitable to be the sole bearings in the system if they cannot be kept well lubricated and clean.

This has led to the investigation of polymer bearings that claim to work in the marine industry in water lubricated heavily loaded rotary applications such as propeller shafts [10] and hydroelectric plants [12]. Whether these properties are maintained in a linear application has yet to be verified.

Water Lubricated Contact Bearings

Water is an attractive lubricant, supply is plentiful and sealing from the influence of corrosive environments or to prevent leakage to the environment

is not critical. Water can also enhance the heat transfer from the coils, therefore increasing efficiency. A demand for low maintenance dry bearings and corrosion resistant bearings for dirty water environments or for compliance with food safety standards has resulted in the availability of a range of polymer bearings offering low maintenance and low wear operation under boundary lubrication. For these reasons these materials have been chosen for further investigation.

When 25 mm *Thordon* SXL strips are run against smooth stainless steel they gave an annual wear rate estimate of 0.3 mm at an average load of 10 kPa [10]. They have a modulus of elasticity of 440 MPa, indicating a superior ability to regain shape after large loads. *TENMAT* T814 Feroform, is a durable abrasive resistant bearing used offshore and in hydro plants. It boasts a load rating of up to 40 MPa from a 20 mm wide pad and can also run dry or wet [12].

Both materials are polymer based, one composite, and the other woven fibre. The wear estimates are acceptable and the elastic nature of the material is such that it can handle shock loads and regain its shape. As the material is an elastomer it does not have a maximum compressive strength. All estimates assume constant loading.

5 Polymer Bearing Analysis

While the structural analysis assumes that the bearing interface is infinitely stiff and of constant dimensions, in reality this cannot be the case. The results of the structural stiffness model to assess the ability of the chosen structure to respond to loads are now discussed.

Polymer Bearing Stiffness

Using the chosen materials, the stiffness of the pads can now be determined for inclusion in the bearing stiffness model. A depth of 5 mm is chosen to keep the thickness to a minimum and reduce swelling allowances for thermal expansion and water absorption to 0.01 and 0.06 mm respectively per pad. The SXL material can withstand up to 5 MPa of pressure before deforming to a maximum limit that effects rotation. In a shaft and housing bearing, the pressure can exceed 14 MPa before rotation is impeded [10]. There were no linear test data available, but oscillating motion studies advised a maximum deflection of 4% before performance deteriorates. In a linear bearing of 5 mm thickness this equates to a 0.2 mm deflection, within the design limit.

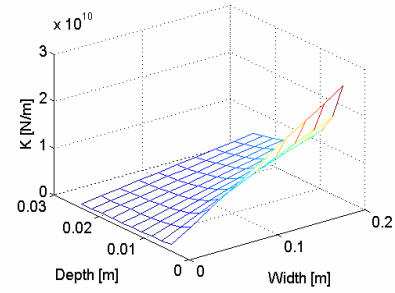


Figure 9: Stiffness of SXL pads with geometry changes.

It was found that shallow and wide pads were stiffest, as shown in Fig. 9. The deflection under constant load in the bearings also showed a linear decrease as pad width increased, as to be expected from the increase in area. *T814* is slightly stiffer than *SXL*, so the bearing sizes based on this shall be reduced. Experimental testing will determine which material has the lower wear rate under the expected conditions. The maximum width will be used in order to allow a margin of safety. Until experimental data is available, estimates based on wear models are used to determine the bearing widths.

Empirical zero-wear model for size estimation

There has been significant work done in developing empirical wear models for polymers [13], as they are very different from metals and the wear mechanisms differ, but as yet there are no fully resolved theoretical predictions of polymer wear. Hence estimates and manufacturer recommendations have been relied upon in the past. Experimental studies on SXL [14] showed the importance of misalignment in accelerating wear in journal bearings when both linear and non-linear wear was observed. The wear model derived in [14] for hydrodynamic journal bearings is based on the fundamental linear wear equation derived by Archard [15].

The empirical model for zero wear in sliding surfaces from [11] was used to estimate an effective bearing size. This is also based on Archard's work and is a suitable approximation for providing a design for zero wear based on the material information available. Two terms for maximum shearing stress in the vicinity of the surface are equated to determine a relationship for the bearing width, b_b .

$$b_b = \left[\frac{K_e P \sqrt{(\frac{1}{2})^2 + \mu^2}}{L_t \gamma_r \sigma_{yp} (\frac{2000}{L_s})^{1/9}} \right]^{1/2.11} \quad (12)$$

The results of the pad width calculations are shown in Table 2 for each material. The calculations are performed for a maximum of 0.5 mm wear in a well lubricated bearing, γ_r of 0.54, with sharp edges, K_e of 500, for net loads of 22.5 kPa, and annual travel of 9444 km. The bearing length is assumed to be the full length of the surface, point loads on discrete bearing pads are not included in this study.

Impact of Bearing Size on Machine Costs

As a smaller pad width is required when the polymer material is on the stator, less structural material will be required as the beams must also span the bearing space. Furthermore, it is logical to have the cheaper bearing material on the stator and the more expensive metal runners on the shorter moving surface of the translator.

It was found that the additional structural material required when a 0.1 m pad width was used was not significantly different from that required for a 0.05 m pad width. It did, however require the use of a slightly larger I-Beam in the narrower design.

The range of widths in Table 2 is indicative of the difference in material properties of each bearing material. SXL has a yield strength of 37.5 MPa half of T814's, while the wet coefficient of friction is also higher. Under the maximum expected load a zero wear bearing design for these loads then resulted in minimum b_b of 0.62 m and 0.5 m for machines 1 and 2 respectively, giving machine costs of €59,5k and €126k respectively. These are significantly higher than the costs using the net load only, as shown in Table 5.

Design	Pad Length L_b [m]	SXL Pad Width b_b [m]	T814 Pad Width b_b [m]
1 (Stator)	7	0.096	0.05
1 (Translator)	1	0.55	0.55
2 (Stator)	9.1	0.075	0.04
2 (Translator)	3.1	0.2	0.1

Table 2: Estimated bearing sizes for each machine design.

Overall Structural effects:

The additional bearing area is included in the machine stack length and modeled as unloaded structure different beams for each bearing size resulted for machine 2. Modeling the deflection with infinitely stiff webs had no significant effect on either machines bearings. However, in the case of simply supported ends it was found, using the stiffness model, that in all cases the bearings are subject to wear and compression due to the increased deflection in the translator. This is demonstrated for the net load and maximum expected load for each design in Tables 3 and 4.

It is easily seen that machine 1, the more expensive, is also the most structurally secure. When the fixed end condition is removed the deflection in the translator increases significantly to a value that will compromise the bearing performance in both machines. This is regardless of the bearing widths, therefore the side supports of the machine will need additional consideration in future designs as they play a critical role demanding significant structure. Fig. 9 illustrated how wider pads are stiffer and of more structural benefit making them the preferred choice. The analysis shows that even with a narrower pad, the machine 1 distributes the load over a larger area due to its extra translator length providing a better design.

Max stress [kPa]	25	200	25	200
b_b [m]	0.1		0.05	

Deflection With Infinitely Stiff Webs

y [μm]	0.06	0.54	0.15	0.12
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Deflection Including Stiffness Model

y_{trans} [m]	0.016	0.13	0.06	.46
y_{leg} [μm]	69	559	220	1800

Table 3: Summary of Machine 1 reactions under Load

Max stress [kPa]	25	200	25	200
b_b [m]	0.1		0.05	

Deflection With Infinitely Stiff Webs

y [μm]	0.01	0.087	0.01	0.087
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Deflection Including Stiffness Model

y_{trans} [μm]	500	1000	370	3000
y_{leg} [μm]	6.7	54	5.6	46

Table 4: Summary of Machine design 2 reactions under Load

6 Fluid Film effects

Contact operation is investigated in order to design for a bearing that can handle all expected loads while still guarantee low wear operation for the minimum maintenance interval of a year. A non-contact design which can operate for years without replacement is the ultimate aim.

For the reasons discussed in Section 4, water has been investigated in developing fluid film effects to reduce the running friction between the bearings. Previous work, [13] has shown that with water, neither squeeze film effects nor hydrodynamic bearings are suitable for this application due to the low prime mover speed and low viscosity of the fluid. Therefore, an active method of maintaining the fluid film must be considered.

Hydrostatic bearings operate by keeping two moving surfaces separated by a thin film of pressurised lubricant. Coefficients of friction between 10^{-3} and 10^{-6} can be expected using this method in comparison to coefficients of 10^{-2} or 0.12 for roller and contact systems respectively [16], and 0.25 to 0.09 for the polymers mentioned.

To implement a bearing of this type, a series of recesses, rectangular or circular can be machined into the translator bearing material. Fig.10 shows a plot of the flow and power coefficients. q_b and H_b , against the ratio of land to length, $2x/L$ for a rectangular pad. The left hand side of each plot shows that the smaller pads have higher coefficients and hence higher power demands for their optimum operating clearance. This optimum clearance, h , occurs when the viscous losses in the fluid equal the leakage losses from the pad. For a rectangular pad it was found that a large land and a length equal to the translator is the most suitable design. For circular step bearings, with an outer radius limit of 0.1 m, the r_i/r_o ratio with the best design

coefficients was found to be 0.53. All calculations were derived from those in Chapter 13 of [17] with further design details mentioned in [16].

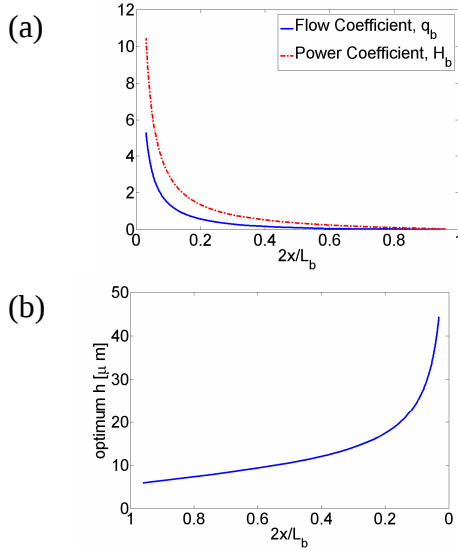


Figure 10: (a) Flow and Load coefficients for rectangular pads from 0.1m to the full translator. (b) Optimum clearance for each pad based on increase in recess area with pad length.

Pad Shape	Square	Rectangular	Circular
q_b [-]	0.026	5.49	4.74
H_b [-]	0.0426	10.68	9
a_b [-]	0.06	0.5	0.566
h [μm]	5.8	40	22
H_L [W]	620	44.9	2.5
ρ [MPa]	1070	4	0.058
When h is set at $50\mu\text{m}$			
H_T [W]	12302	123	28

Table 5: Estimated bearing sizes for each machine.

The aim of this initial design is to highlighting the main characteristics in developing suitable hydrostatic pads for water fed systems; size, load and power. As shown in Table 5, the circular pad offers the best design. It has significantly lower losses due to the elimination of sharp corners and a more uniform recess area. Using water reduces the clearance of fluid films to those of oil but they still allow the translator to operate above the limits of partial lubrication.

As these bearings are active, the cost will increase due to the pump requirements and machining of the compensating elements to control the recess pressure. The next stage of this work is to test this type of bearing and provide cost estimates for the magnetic and structural model.

7 Cost Analysis

The cost analysis used in this model is identical that used in [18] bar the following refinement. The length of a single turn of Copper wire, L_{CU} , is given by (13) and this is multiplied by the number of turns in the windings and the total number of windings to get the total length of wire used. From this, the total mass of

copper used can then be calculated.

$$L_{CU} = 2(l_s + 2\tau_p) \quad (13)$$

The following material costs were assumed in this analysis, based on the costs used in [1].

- Laminated Steel at 5 Euro/kg
- Structural Steel at 3 Euro/kg
- Copper at 10 Euros/kg
- Magnets at 30 Euros/kg

The laminated steel comprising the armature iron is assumed to be significantly more expensive than the structural steel comprising the standardised I-Beam sections. This is likely to be the case as its construction will be more complex and have considerably less economies of scale in comparison to the I-Beams.

The machines were designed to produce approximately 500 kW at their maximum rated speed of 2.2 m/s by multiplying the number of poles to the minimum number required to achieve at least this power output. The stator of both machines was extended to overlap the translator by 3 m at either end to allow for a maximum wave height of 3 m. In both cases, a space of 0.1 m either side of the stator was allowed to incorporate the bearings for the reasons described in the previous sections. The results of this analysis are shown in Table 5.

Stack Length (m)	0.5	1.5
Rated Power (kW)	516	505
Total Cost (kEuros)	55.4	118.7
Cost per Watt (Euro/W)	0.107	0.235
Copper Cost (kEuros)	28.8	54.8
Magnet Cost (kEuros)	7.6	7.3
Laminated Steel Cost (kEuros)	16.5	38.8
Structural Steel Cost (kEuros)	2.6	17.9

Table 6: Results of cost analysis for two machine designs of different stack lengths

These results show that the cost of the machine is sensitive to the required structure, although other factors are more important. For the shorter machine the structure accounts for only 4.6% of the total cost, while the wider machine's structure accounts for 15% of the much higher total cost, and is almost 7 times that of the narrow design. It is essential therefore that any future optimization routine takes the structural requirements into account in order to have full confidence in the results.

The bearing costs are not yet quantifiable due to the novel use of materials, however it is known that these materials will be cheaper than bronze, a common marine bearing alternative. More detailed and recent knowledge of the costs of all materials used in the machines is desirable for an accurate analysis.

8 Future Work

Enhancing the design models

The thermal performance of the generators is critical

to determining the generator efficiency. The main loss mechanism in the machine is copper loss in the windings, which is also temperature dependent. In future investigations, a lumped parameter thermal model will be incorporated so that the actual energy yield can be more accurately assessed for any optimisation process.

This work demonstrates the importance of including the empirical wear model into the structural model as the additional bearing area required can affect the size of beam required to provide the necessary structural rigidity. The stiffness assessment also highlighted the need to make the side supports extremely rigid to prevent the main load-carrying bearings from undergoing excessive compression in order to reduce the possibility of serious airgap imbalances and reduce fatigue in the beams and wear in the bearings. For these reasons, future design programs including these aspects will be developed.

9 Conclusions

During the early stages of designing a generator for minimal energy cost, most previous authors have focussed on the electromagnetic design. When structural and bearing models are also included, the optimal design changes and alternative generators become more attractive. It is therefore important to include as many aspects as possible early in the design process. This is illustrated by way of a comparison between two different iron-cored linear PM synchronous generators. It was found that a narrower stack length resulted in a lower cost per Watt of exported energy despite the resulting increase in total machine length to achieve the rated power. A significant part of the higher cost is attributable to the larger structure required to maintain the airgap.

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