LOAD SENSITIVITY ANALYSIS OF A LARGE DIAMETER PERMANENT MAGNET GENERATOR FOR WIND TURBINES

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Abstract: Permanent magnet (PM) and electrically-excited direct-drive generators are usually described by their high mass and production costs. Studies have proved that when increasing generator’s capacity, inactive mass increases faster than the active mass. In this paper, a load component sensitivity study for the inactive mass is performed on a direct-drive PM generator via Finite Element Analysis (FEA). Taguchi method is utilized for describing the relations between design parameters and the resulting stresses and deformations. Most influential load components are determined and general countermeasures are offered.

Key words: sensitivity analysis, permanent magnet generator, finite element analysis, Taguchi method.

1. INTRODUCTION

During past decades a high number of research papers have been presented on the electromagnetic layout of an electrical generator in a wind turbine [1-2]. Despite the numerous solutions that have been offered, the geared drive train with an induction generator, still remains the dominant technology [3]. Alongside it, various alternative drive train solutions, like the PM synchronous generator, have emerged. Although, these machines offer higher reliability and power density, they are usually described also by their high mass and production costs [4]. With machines near megawatt scale, the final cost and mass of the generator has been largely associated with the active electrical parts. However, electrical machines also contain material that fulfills a structural role and works against a number of high forces. Studies have proved that when increasing the generator capacity, inactive mass increases faster than the active mass and begins to dominate [3, 5]. This, alongside the fact that price of the active materials has steadily been dropping, indicates that when dealing with higher capacity generators, more refined methods to reduce the inactive mass are necessary. This paper aims to provide insight to the relative importance of the acting forces on the generator, in order to provide information for the initial selection of design parameter values.

2. ANALYSIS DESCRIPTION

2.1 Generator

The machine subjected to the analysis is a PM ultra large direct-drive generator with an air-gap diameter of 12 meters and rated capacity of 3 MW. The machine consists of an inner rotor with permanent magnets and an air-cooled outer stator with electrical windings. Active elements are supported by a lattice structure consisting mainly of tubular steel beams. While the rotor is supported equally from both sides, the stator has one sided unsymmetrical support, like shown in Figure 1a. This is of interest when considering the acting gravity forces, which in this case result in an air-gap deforming rolling moment.

Electromagnetic setup of the generator is similar to the one that is described in the previous work [6]. The stator coils are located in the air-gap to reduce the normal component of Maxwell stress.
2.2 Acting forces

Forces present in an electrical machine are similar for most types of generators. In addition to those that can be found in regular electrical generators, effects from wind and tower top accelerations must be considered for a machine that is used in a wind turbine. As a result, in the presented analysis, following loads are considered: deadweight resulting from gravity, rotor rotational velocity, wind load on the structure, tower front-to-back acceleration and side-to-side acceleration, normal stress and generator torque.

Torque \( \tau \) and rotational speed \( \omega \) are proportional to the generator’s electrical power output \( P_{el} \) and efficiency \( \eta \) (1). Therefore, they both cannot be subjected to minimization but their relation can be manipulated in a certain range if it turns out to be beneficial. The limits of this manipulation are set by maximum allowed rotational speed for the blades (tip speed limit \( \approx 80 \) m/s). Meaning that for larger turbines and rotors, the rotational velocity is smaller.

\[
P_{el} = \tau \cdot \omega \cdot \eta \quad (1)
\]

From an electromagnetic point of view, torque is a product of the tangential component of Maxwell stress, which is also designated as shear stress \( \sigma \).

Therefore, torque of a rotating electrical machine can also be given as

\[
\tau = 2\pi \sigma R^2 l \quad (2)
\]

where \( R \) is the machine air-gap radius and \( l \) axial length of the generator \(^{[4]} \). Torque is the useful force in an electrical machine and does not serve to close the air-gap. In most generator topologies Maxwell stress has also a normal component (normal stress), which is caused by the attraction of the ferromagnetic surfaces of rotor and stator. Normal component can be a magnitude greater than the shear stress and directly acts to close the air-gap. Designers have long looked for ways how to reduce the normal stress in relation to the shear stress and as a result air-cored machines have been developed \(^{[7-8]} \). Presented machine is a hybrid solution, as the windings are located in the air-gap, like shown in Figure 1b.

Wind turbine towers have become highly optimized and in many cases extremely soft and flexible towers are used \(^{[9]} \). For geared drive trains this has a low impact but direct-drives with their high mass and material distribution on the outer perimeter are vulnerable to tower top accelerations. Depending on the topology of the support structure, considerable extra stresses and deformations can occur.

From (2) it can be seen that one option to increase the generator torque and therefore the power output, is to increase the air-gap radius. Many designers have taken advantage of this, which has led to generators with ultra large diameters. As the perimeters of the machines grow, so does the force that is resulting from wind acting on the generator’s structure \(^{[10]} \). This can be found by

\[
F_{\text{wind}} = \rho / 2 \cdot v_{\text{wind}}^2 \cdot A \cdot c_p \quad (3)
\]

where \( \rho \) is the air density, \( v_{\text{wind}} \) the wind speed, \( A \) surface area of the generator that is perpendicular to the wind direction and \( c_p \) the shape parameter of the surface.
Therefore, depending on the actual surface that is exposed to wind, the force \( F_{\text{wind}} \) could become a significant factor that influences the generator design.

2.3 FEA model
Framework of the generator is modeled in ANSYS Workbench environment with two-node beam elements (BEAM188) having six degrees of freedom at each node. For the beam-to-beam connections it is assumed that they have the same carrying capacity as the profiles. Therefore, different beams are connected by shared nodes and automatically transmit all moments and forces.

![Fig. 2. FEA model of the generator](image)

Only the framework of the generator is directly modeled while rest of the elements like magnets and coils are approximated by point masses. It is assumed that they do not add significantly to the structural stiffness of the machine. In addition, rotor shaft and stator king-pin are not modeled, as for direct-drive turbines they are mostly subjected to forces and moments from the rotor blades [7]. Therefore, their cost and end mass is not greatly influenced by the generator design parameters.

The setup and in particular the stiffness of the FEA model have been validated with an experimental study on the full scale generator structure. The dimensions of rotor and stator rings were measured with and without magnetic forces and calculated deformations were found to be in good correlation with the FEA results.

3. SENSITIVITY STUDY

3.1 Initial parameter selection
In the first part of the analysis all the seven acting load components are applied to the structure one-by-one while resulting stresses and deformations are extracted. Results are compared with values from the nominal working situation where all the components are acting with the same values simultaneously. Overall stress and deformation values in the nominal working situation are within allowed limits to obtain reasonable results.

![Fig. 3. Principles of the study](image)

A cylindrical coordinate system is used to record the results and only rotor positive and stator negative radial deformations are investigated, as they serve to close the air-gap. The results of the FEA analysis are presented in Table 1, where the contribution of each component to the overall stress level is presented.

The initial study is performed to determine the most influential load components for further analysis. This enables to reduce the required number of experiments when investigating parameter interactions. From the comparison between the sum of components and actual results it is clear, that further analysis is required, as deformations differ about 20% and stresses 50%.
Only the parameters that contribute more than 5% to at least one of the outputs are selected for further investigation. These include: deadweight, front-to-back acceleration, normal stress and torque (marked green in Table 1).

<table>
<thead>
<tr>
<th>Load</th>
<th>Rotor def</th>
<th>Stator def</th>
<th>Rotor stress</th>
<th>Stator stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deadweight</td>
<td>38%</td>
<td>64%</td>
<td>18%</td>
<td>42%</td>
</tr>
<tr>
<td>Rot speed</td>
<td>5%</td>
<td>0%</td>
<td>4%</td>
<td>0%</td>
</tr>
<tr>
<td>Side accel</td>
<td>4%</td>
<td>5%</td>
<td>2%</td>
<td>4%</td>
</tr>
<tr>
<td>Front accel</td>
<td>3%</td>
<td>2%</td>
<td>7%</td>
<td>5%</td>
</tr>
<tr>
<td>Norm stress</td>
<td>44%</td>
<td>23%</td>
<td>27%</td>
<td>21%</td>
</tr>
<tr>
<td>Torque</td>
<td>5%</td>
<td>4%</td>
<td>39%</td>
<td>26%</td>
</tr>
<tr>
<td>Wind</td>
<td>1%</td>
<td>2%</td>
<td>7%</td>
<td>5%</td>
</tr>
<tr>
<td>Sum</td>
<td>1,3</td>
<td>4,8</td>
<td>109</td>
<td>185</td>
</tr>
<tr>
<td>Actual total</td>
<td>1,1</td>
<td>4,1</td>
<td>74</td>
<td>128</td>
</tr>
</tbody>
</table>

Table 1. Load component contribution to overall stress level.

3.2 Taguchi analysis

Taguchi method for design of experiments is utilized to compose the mathematical model describing the relations between design parameters and the resulting stresses and deformations. Taguchi is chosen as it enables to reduce the number of required experiments even further. Five value levels are chosen for each input parameter: 100%, 60%, 80%, 120% and 140% of their rated value. The levels are chosen based on an assumption that for a defined output power the design characteristics would vary not more than ±40%. This is important since conclusions drawn from small scale experiments are valid only over the particular experimental region [11].

Variation is applied to the rated value of the parameter, without considering the safety factor that needs to be applied to the loads [10]. The value of the parameter is changed according to the required percentage and only then multiplied with the safety factor. In case of the deadweight, only the mass of active material together with needed fixtures is varied, as only this can directly be influenced by designer’s initial decisions.

Since there are four parameters and five levels, L25 orthogonal array is used. For each experiment, the stresses and deformations of rotor and stator are extracted as performance characteristics. The obtained FEM results are transformed into signal-to-noise (SN) ratios. The SN ratios are used as a measure of deviation from or nearness to the desired value. Goal of the analysis is minimization of performance characteristics. The needed SN ratio are found with

\[ SN_i = -10 \log \left( \sum_{u=1}^{N_i} \frac{y_u^2}{N_i} \right) \]  

where \( u \) is the trial number, \( N_i \) the number of trials for experiment \( i \) and \( y_u \) is the value of the performance characteristic for a given experiment [11]. After calculating the SN ratio for each experiment, the average SN value is calculated for each factor and level. In order to determine the effect of the variable on the process, \( \Delta \) value (high SN – low SN) is found for each parameter.

Response of the SN ratios for the rotor deformation can be found in Table 2. The rakings of all four parameters with their corresponding \( \Delta \) values for all performance characteristics are presented in Table 3.

<table>
<thead>
<tr>
<th>Level</th>
<th>Deadweight</th>
<th>Front accel</th>
<th>Norm stress</th>
<th>Torque</th>
</tr>
</thead>
<tbody>
<tr>
<td>100%</td>
<td>0,00</td>
<td>0,12</td>
<td>-0,08</td>
<td>-0,11</td>
</tr>
<tr>
<td>60%</td>
<td>1,19</td>
<td>0,13</td>
<td>1,82</td>
<td>0,09</td>
</tr>
<tr>
<td>80%</td>
<td>0,50</td>
<td>-0,04</td>
<td>0,81</td>
<td>0,15</td>
</tr>
<tr>
<td>120%</td>
<td>-0,59</td>
<td>-0,14</td>
<td>-0,89</td>
<td>0,03</td>
</tr>
<tr>
<td>140%</td>
<td>-1,11</td>
<td>-0,07</td>
<td>-1,66</td>
<td>-0,17</td>
</tr>
<tr>
<td>( \Delta )</td>
<td>2,30</td>
<td>0,28</td>
<td>3,48</td>
<td>0,32</td>
</tr>
<tr>
<td>Rank</td>
<td>II</td>
<td>IV</td>
<td>I</td>
<td>III</td>
</tr>
</tbody>
</table>

Table 2. Response table for SN ratios for rotor deformation.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Deadweight</th>
<th>Front accel</th>
<th>Norm stress</th>
<th>Torque</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor def</td>
<td>( \Delta )</td>
<td>2,30</td>
<td>0,28</td>
<td>3,48</td>
</tr>
<tr>
<td>rank</td>
<td>II</td>
<td>IV</td>
<td>I</td>
<td>III</td>
</tr>
<tr>
<td>Stator def</td>
<td>( \Delta )</td>
<td>3,65</td>
<td>0,19</td>
<td>1,72</td>
</tr>
<tr>
<td>rank</td>
<td>I</td>
<td>III</td>
<td>II</td>
<td>IV</td>
</tr>
<tr>
<td>Rotor stress</td>
<td>( \Delta )</td>
<td>0,98</td>
<td>0,39</td>
<td>0,82</td>
</tr>
<tr>
<td>rank</td>
<td>II</td>
<td>IV</td>
<td>III</td>
<td>I</td>
</tr>
<tr>
<td>Stator stress</td>
<td>( \Delta )</td>
<td>2,49</td>
<td>0,06</td>
<td>0,62</td>
</tr>
<tr>
<td>rank</td>
<td>II</td>
<td>IV</td>
<td>III</td>
<td>I</td>
</tr>
</tbody>
</table>

Table 3. Rankings of all parameters for all performance characteristics.
4. DISCUSSION

According to results presented in Table 3, deadweight and normal stress are the most influential factors for the air-gap deformations. For rotor, normal stress prevails but for stator, deadweight has by far the biggest influence. This can be explained by cantilever nature of the stator structure (Figure 1a). It could be compensated by adding a frontal support structure. However, it would also make the mechanical construction and specially the assembly process more complex. For the stator the deadweight is not an alternating force. Therefore, the resulting deformations could be already factored into the shape of the structure prior to the assembly and later the machine would deform itself into right dimensions. Alternatively, after the stator assembly the windings could be adjusted in order to compensate for the static deformations after stator has been lifted into the vertical position. Another option could be also utilization of materials with better stiffness to weight ratio i.e. performing structural optimization [12-15].

Frontal acceleration has a minor significance for all of the performance characteristics. This can be explained by the mechanical setup of the frame structure. As it requires space to use material effectively, it also contributes to the axial stiffness of the machine and therefore reduces the acceleration influence. But it must be noted that for disc like sheet metal structures with their material located mostly on the outer rim, this most likely does not apply.

From Table 1 and 3 it can be summarized that for this type of generator, the key influential factor determining the levels of stress and deformation, is the mass of the material located at the air-gap radius.

When taking actions to reduce the mass on the air-gap radius, care has to be taken that these actions do not increase the normal stress, as any benefits would be then cancelled out. For example one of the possible solutions could be using higher grade PM’s with better mass to field strength ratio. At the same time the air-gap could be extended to keep normal stress at acceptable level. The reduced mass would also be beneficial for the other turbine parts, as it would mean for example lower loads for the yaw system.

This conclusion indicates that solutions offered in literature to use heavy but cheap magnetic materials like ferrites [16] are most likely cost wise not beneficial. The savings on magnet material costs would be lost in the extra investment that is required for the structure reinforcement.

On the stress side, the designer’s aim, in addition to reducing mass of the active material, should be also limitation of the torque value. Due to the torque’s direct dependency on the output power (1) and constraints laid on the rotational speed, there is usually not much room left to play. However, as a rule of thumb, the blades should be always utilized at their maximum allowed rotational speed.

Finally, electromagnetic setups that offer either considerably lower mass or normal stresses could be considered. For example solution of an ironless design presented by Gordon and Spooner [7] would fulfill these criteria.

5. CONCLUSION

Experimentally validated FEA model of a large diameter PM generator for a wind turbine has been built in ANSYS environment. The main loads acting on the generator structure have been described. Their individual contribution to the overall stress and deformation levels has been determined. Taguchi method of design of experiments has been used to study interactions of the individual load components and their relation to structural stresses and air-gap deformations. The mass of material located on the air-gap radius appears to be the most influential component when looking at both stresses...
and deformations. When regarding only the deformations, normal stress has to be strongly considered and when focusing on stresses, the generator torque is the dominant parameter. The obtained results imply that all efforts must be focused on the reduction of mass at the air-gap radius as it would bring about further reduction of structural mass.

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6. REFERENCES

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