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Structural Optimisation of Offshore Direct-Drive Wind Turbine Generators Including Static and Dynamic Analyses

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Abstract. One way to achieve increased wind capacity is by installing larger and more efficient wind turbines, which results in larger/heavier generators. Direct-drive, permanent magnet generators are favoured due to their increased efficiency, but the added weight is an issue, as this drives up the cost of the nacelle and turbine support structure, along with increasing the manufacturing and installation costs. Therefore, minimizing the mass, particularly the structural mass, of these low speed generators is becoming much more important. A vast amount of research has been done on trying to reduce the electromagnetically 'active' materials, but it is the supporting structure or 'inactive' materials, that makes up the biggest percentage of the generator's mass. Therefore, this paper studies the statics and dynamics of a large offshore direct-drive generator's supporting structure and the opportunities for light-weighting, as well as improvements to the generator's rotor structure through structural optimisation. The indicator for optimised design is system weight under each predefined scenario. These scenarios will cover different design considerations of the generator's rotor structure.

1. Introduction

An increasing amount of attention is shifting towards the use of renewable resources for energy production and supply, in order to reduce carbon dioxide emissions. In the first quarter of 2020, the International Energy Agency (IEA) [1] states that the global use of renewable energy increased by 1.5% on the previous year. One reason for this increase was due to the completion of 60GW of wind projects in 2019. The IEA also states that during the recent COVID-19 pandemic, although the global energy demand fell by 3.8%, demand for renewables, including wind, grew due to the larger installed capacity and the fact that their output is mainly unaffected by demand, whereas demand for all other energy sources fell. The IEA also predicts that during the rest of 2020, this pattern will continue, with the demand for renewables continuing to increase due to the low operating costs and the priority given to them by certain regulations.

As briefly touched upon above, the requirement to increase the amount of energy extracted from the wind is growing and is predicted to continue to grow for the foreseeable future. One way to achieve this requirement is to install larger wind turbines. WindEurope [2] states that since 2014, the capacity of the turbines have increased by 16% every year. Larger wind turbines and the generation of more power, typically equates to larger generators. This is because the generator is the key component that converts the mechanical power from the rotating shaft, to electrical power which is supplied to the grid.

The most commonly used generators in current wind turbines are either doubly-fed induction generators (DFIGs) or direct-drive generators, with the interest in direct-drive generators growing. Direct-drive generators are described as low speed generators because they connect directly to the rotor hub, thus eliminating the gearbox. By eliminating the gearbox and reducing the number of moving parts in the drive train, maintenance costs and the losses caused by the gearbox reduce, in addition to increasing the reliability of the system.

Low speed generators can either have a permanent magnet rotor or electrically excited rotor, with permanent magnet being the preferred type. This is due to the fact that they do not require an external power source for magnetization, so decreases the heat produced by the system and increases the efficiency and energy yield. This paper looks at a permanent magnet direct-drive generator. It specifically looks at the radial-flux topology, which means that the magnetic field runs radially with respect to the direction of the rotor shaft.

Although there are many advantages of permanent magnet low speed generators, the downside is that they are large and heavy. This is due to the fact that they operate at a low rotor speed, so in order to produce the required power output, the torque needs to be high, which can be seen from Equation 1, in which 'P' corresponds to the power output, 'T' is the required torque and 'ω' is the rotational speed. Torque can be determined from Equation 2, where 'R' is the radius, 'σ' is the shear stress and 'l' is the axial length of the machine. To achieve the required torque, the generator's dimensions such as radius, and/or length, need to increase (see Equation 2), thus confirming that low speed generators are larger and expectedly heavier than their high speed counterparts.

$$P(\uparrow) = T(\uparrow)\omega(\downarrow) \tag{1}$$

$$T(\uparrow) = 2\pi R^2(\uparrow)\sigma l(\uparrow) \tag{2}$$

The increase in weight will increase the cost of the nacelle and turbine support structure, along with the manufacturing and installation costs. Therefore, it is important to investigate ways of reducing the weight, especially with direct-drive generators becoming more popular. Generators are constructed of 'active' and 'inactive' materials. Permanent magnets, copper windings and back iron are classed as 'active' materials. Whereas the generator's supporting structure, that is required to maintain the airgap between the rotor and stator whilst various forces are acting upon the system, is classed as 'inactive' material. A wide range of research has been done on ways to reduce the 'active' material but not much research has looked at reducing the mass of 'inactive' materials and Hartkopf *et al.* [3] states that, approximately two-thirds of the mass in a direct-drive radial-flux generator is in the supporting structure.

This paper looks at a specific machine, the NREL 15MW reference wind turbine [4] and investigates the structural mass optimisation of the rotor's supporting structure, in order to reduce the total mass of the generator and improve the overall design. The ANSYS Design Explorer optimisation tool was used to allow various dimensions and parameters of the generator's rotor structure to be varied in a controlled manner, in order to reduce the structural mass of the rotor. Both the static and dynamic perspectives were analysed. The combined static and dynamic analyses were compared to the static only analyses, in order to see what effects including the dynamics has on the structural parameters.

The rest of this paper is structured as follows: Section 2 explains the previous research that has been carried out in the field. In Section 3, a detailed description of the wind turbine and electrical generator used is given. Section 4 explains the methods used to model and analyse the rotor structures and Section 5 describes the results obtained for each model. Section 6 concludes the results from the paper and suggests proposed future work.

2. Literature Review

In this section the most up to date investigations are introduced. Structural mass optimisation of direct-drive electrical generators has become an important object of study, as a substantial reduction in turbine's capital and maintenance costs can be achieved. In 2013, Zhang *et al.* [5] determined that by using a numerical method and genetic algorithm, most of the mass and cost of high power generators is in its supporting structure. In [6] McDonald, explained that polynomial modelling showed that the 'inactive' mass of the stator is greater, than that of the rotor and that by combining both the rotor and stator 'inactive' masses, it contributes to at least 82% of the total mass for a 5MW machine. The author determined that 'inactive' mass is dominant and that the major factor that influences the 'inactive' mass is normal stress. Bearing that in mind, Zavvos [7] investigated both the analytical and structural optimisation of permanent magnet, direct-drive generators for 5MW wind turbines. An optimal strategy to minimise the mass was proposed. Firstly, simple structures were modelled to then be used to estimate the deflection and fatigue levels caused by the structural forces. Then structural optimisation tools were used to decrease the mass, whilst ensuring the stiffness of the structure was not compromised. Finally, the stiffness of the lightweight structure was verified by using finite element tools. In 2017, Jaen-Sola [8] investigated a variety of ways of designing low speed, high torque radial-flux permanent magnet electrical generators, for direct-drive wind energy converters, with the aim of minimising the mass. Both disc and conical structural configurations were analyzed from static and dynamic perspectives using finite element techniques. Different strategies were attempted with the aim of minimising the overall mass, as well as modifying structure's natural frequencies, including the implementation of stiffeners. The use of composite materials was also studied in detail. It was determined that the conical structure required less material to withstand the same loads and natural frequencies, as opposed to the disc structure.

Other authors have proposed to employ distinct mathematical algorithms to come up with more efficient and intelligent optimal designs. In this respect, Zabihinejad *et al.* [9] discussed the use of an Ant Colony Optimisation (ACO) algorithm to design and optimise a radial-flux, direct-drive permanent magnet generator. The permanent magnet dimensions, as well as the diameters of the stator and rotor were optimised, using the ACO algorithm to meet the main objectives, which were to achieve high power density and efficiency. High efficiency was achieved by decreasing the core losses and cogging torque within the ironless structure. Their results showed that the optimised structure had a higher efficiency, when compared to other optimisation methods. A genetic algorithm was used by Oliveira *et al.* in [10]. The authors began the optimisation process of a permanent magnet, synchronous generator by modelling three processes that occur during the operation of the generator. A genetic algorithm was then used to determine the optimised generator solution. The results showed that the optimised solution had a smaller radius and smaller depth, thus this smaller volume meant a reduction in the generator weight.

Jaen-Sola *et al.* [11] proposed looking at the structural mass of radial-flux generators in the early phase of the design process. The paper looks at the machine's structure, the rotor and stator separately, from a dynamic perspective and uses the effective mass participation factor (EMPF) method. It focused on the supporting structures formed by disc (including the addition of stiffeners) and conical sub-structures. The authors concluded that the conical structures should be taken into account when designing the generator's structure, due to the fact they have good static and dynamic characteristics. They also concluded that the addition of stiffeners to conical structures should be investigated and that by removing certain areas that were carrying no load, helped to occasionally increase stiffness, as well as reduce mass. The use of composite materials was also mentioned as being an area that should be considered.

This paper looks at modifying the NREL 15MW generator rotor design by adding or removing material. A number of scenarios were investigated including the addition of holes and a combination of both holes and stiffeners, in order to alter the structure's natural frequencies, with the aim of maximizing the machine's operating range while reducing it's mass.

3. Turbine and Generator Specification

The wind turbine that was analysed was the IEA Wind 15-Megawatt Offshore Reference Wind Turbine [4]. This turbine is an IEC Class 1B, direct-drive machine, with a hub height of 150m and rotor diameter of 240m. This turbine contains a direct-drive, permanent magnet, synchronous, radial-flux, outer rotor generator and the layout of this generator is shown in Figure 1.

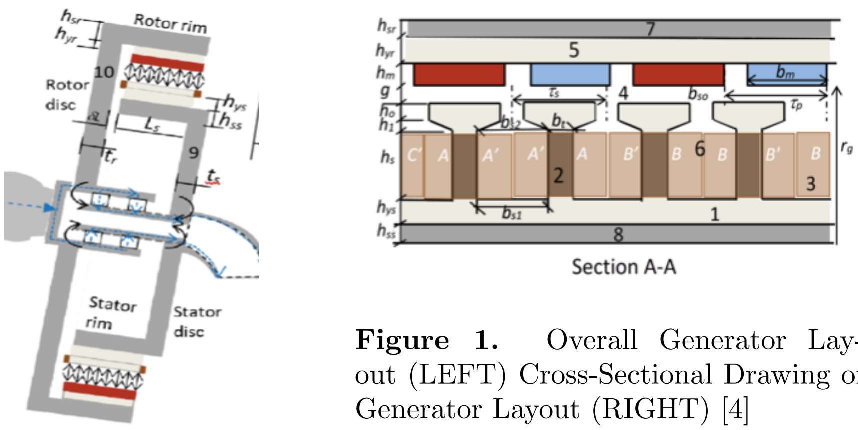

Figure 1. Overall Generator Layout (LEFT) Cross-Sectional Drawing of Generator Layout (RIGHT) [4]

This generator has an air gap radius of 5.08m and stack length of 2.17m. According to the specification the structural support constitutes 50% of the total mass.

4. Methodology

This paper explores the development of a method to optimise the design of a direct-drive generator rotor structure, subject to both static and dynamic constraints, in order to minimize mass. The modelling is carried out in ANSYS Workbench, using the Geometry module to produce and parametrise the solid models, Static Structural analysis and Modal analysis modules and the Direct Optimization module for the optimisation. Finite element models were analysed making use of the ANSYS adaptive sizing meshing tool. A high quality 3D tetrahedral mesh with 4241 elements and 9102 nodes was generated for the baseline rotor structure.

In terms of the static analysis, 4 different loadings were considered. The forces arising from the Maxwell's electromagnetic loads lead to a torque of 21MNm (applied uniformly to the inside of the rotor cylinder, tangential to the surface) and a nominal stress of 447kPa (towards the centre) applied normally to the rotor cylinder surface. Because the rotor is rotating at up to 0.79rad/s, there is a small centripetal force acting normal to the rotor cylinder radially outwards. In addition to this, gravity acts on the mass of the rotor. Because of the small tilt angle of 6 degrees, the effective component of gravity in plane is 9.42m/s^2.

The normal component of the Maxwell stress is actually dependent on the airgap flux density. This, in turn, depends on the airgap clearance. Because this clearance can vary with respect to the angle around the machine, we consider two cases: uniform loading (447kPa) and non-uniform loading (varying sinusoidally with a minimum of 400kPa and a maximum of 425kPa). As this is an outer rotor machine, the stress is applied radially-inwards onto the inner surface of the rotor rim. For the non-uniform loading, the rim is partitioned into discrete sections where different stress magnitudes can be applied to each section. Each section has a constant stress, calculated based on the step-wise approximation of the sinusoidally varying stress.

Additionally, the mass of the rotor yoke and magnets were calculated and this value (46,021kg) was applied uniformly to the inside of the rotor cylinder, as a distributed mass.

As well as considering the static loading on the generator, it is important to analyse the generator structure's dynamic. The normal wind turbine speed range is 5 to 7.56rpm. By adding a 30% contingency, this range extends to 3.5 to 9.8rpm. With $p = 100$ pole pairs, there is a forcing frequency of pP in the generator, which can excite the rotor vibrating modes.

The optimisation of the generator aimed to minimize the mass of the generator structure, subject to a number of constraints. In order to avoid the ultimate limit state of strength, the Von Mises stress should be kept below 200MPa and the deformation in the radial direction (i.e. into the airgap) should be limited to less than 20% of the airgap clearance (10.16mm), i.e. 2.03mm. To avoid resonance, the generator structure's natural modes should be lower than 6Hz or higher than 16Hz - the dynamic constraints.

A base case generator structure was developed in [4] (solid models can be downloaded at [12]). The model was parameterised so that the rim thickness, h_{sr} and disc thickness, t_r are design variables. Further scenarios included mass removal with holes added into the disc, addition of strengthening ribs and a reinforcement of the rim. These parameters describing these features were added to the optimisation.

5. Results

Figure 2 shows the design variables adjusted in each case during the optimisation process.

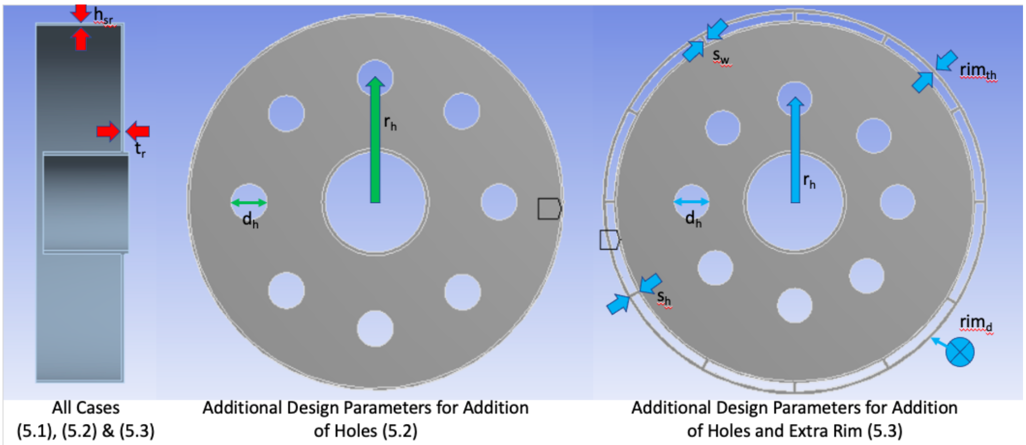

Figure 2. Design Parameters Adjusted For Each Scenario

5.1. Base Case

Table 1 shows the results of the base case for both uniform (Mode 0) and non-uniform loading (Mode 1), both before and after the static and dynamic analyses have been carried out. It shows that prior to running any optimisation, both modes exceed the targeted deformation value of +/-2.03mm (Figure 5). It also shows that the base case design demonstrates two paired frequency modes (6 and 7), shown in Figure 6 and the Campbell diagram (Figure 3), coinciding with pP frequency within the operating range, that the designers would like to avoid.

As previously mentioned, Table 1 also presents the results post optimisation. It shows that the rotor design can be modified in order to meet the required dynamic constraints for both uniform and non-uniform loading. For both loading cases (uniform and non-uniform), the thickness of the disc needs to increase approximately 2.5-fold and 1.6-fold respectively. The thickness of the rim can be decreased slightly for uniform loading and increased approximately 3-fold for

Table 1. Results for the Original Design Prior to and Post Optimisation

	Original		Optimised			
			Static Only		Static and Dynamic	
Description	Mode 0	Mode 1	Mode 0	Mode 1	Mode 0	Mode 1
Mass (T)	148.80	148.80	140.73	244.16	225.68	244.16
Stress (MPa)	48.83	45.63	110.71	76.50	102.58	76.50
Radial Deformation Min (mm)	-5.41	-5.49	-2.03	-2.03	-2.03	-2.03
Radial Deformation Max (mm)	3.80	3.87	2.03	0.99	0.42	0.99
Frequency (Mode 6) (Hz)	9.70	10.13	-	-	16.00	17.61
Frequency (Mode 7) (Hz)	9.83	10.25	-	-	17.83	33.97

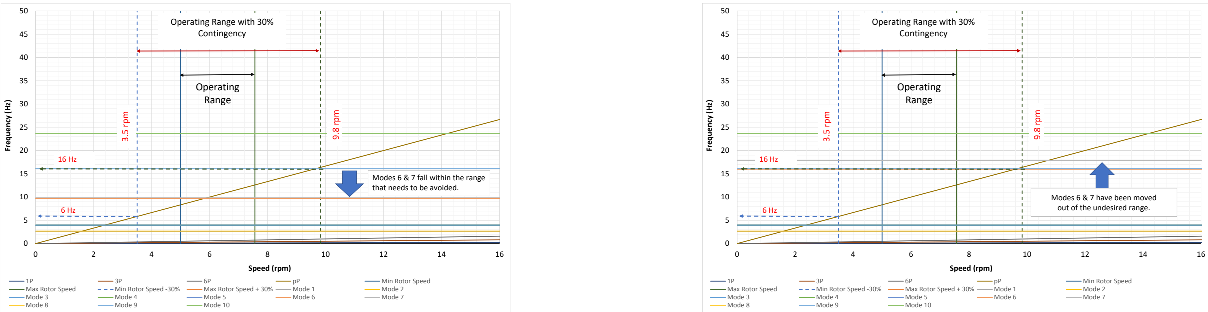

Figure 3. Campbell Diagram of the Base Case (Mode 0) for the Original (LEFT) and Optimised (RIGHT) Case

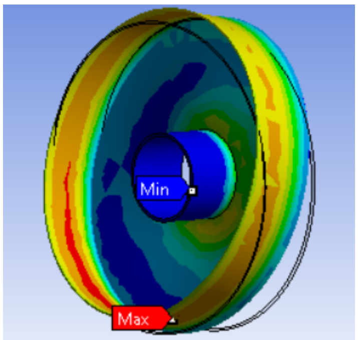

Figure 4. Maximum Equivalent Stress on the Base Case (Mode 0)

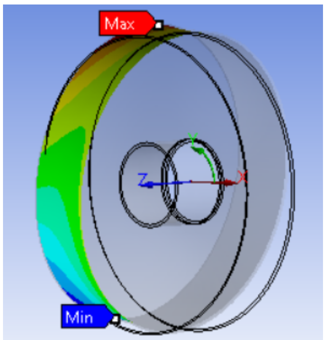

Figure 5. Maximum Radial Deformation on the Base Case (Mode 0)

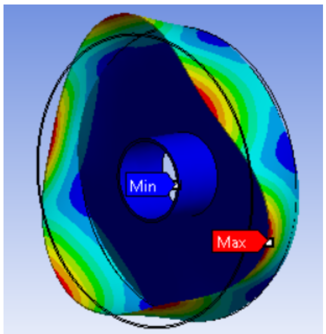

Figure 6. Mode Shape Deformation (Mode 6) on the Base Case

non-uniform loading (shown in Table 2) in order to meet the set constraints. This implies that in order to meet the design constraints, the mass ends up increasing by approximately 50%.

5.2. Addition of Holes

In order to try and reduce the mass of the rotor, the addition of holes in the disc were introduced. Twelve different scenarios were carried out with the addition of seven, eight and nine holes at a

Table 2. Dimensions for the Original Optimised Design

Description	Initial	Mode 0	Mode 1
Disc Thickness (mm)	81.75	206.13	131.94
Rim Thickness (mm)	63.69	60.99	168.49

distance of both 3m and 3.5m from the centre, for both uniform and non-uniform loading.

Table 3 shows the results for all holes at a distance of 3m from the centre for uniform (Mode 0) and non-uniform loading (Mode 1). The table shows that by adding holes, an additional two paired frequency modes (Modes 8 and 9) are shown to pass through the operating range that is to be avoided. These additional mode shapes are shown in Figure 9.

Table 3. Results for the Optimised Design with the Addition of Holes at 3m Distance for Uniform (Mode 0) and Non-Uniform Loading (Mode 1)

Description	7 Holes		8 Holes		9 Holes	
	Mode 0	Mode 1	Mode 0	Mode 1	Mode 0	Mode 1
Mass (T)	182.17	175.03	161.63	193.68	187.29	217.29
Stress (MPa)	99.78	-60.25	-21.18	195.23	-83.43	56.57
Radial Deformation Min (mm)	-1.22	-2.00	-2.03	-1.18	1.63	0.74
Radial Deformation Max (mm)	-1.58	0.14	0.46	1.01	-1.19	-0.93
Frequency (Mode 6) (Hz)	16.05	16.90	16.47	17.26	16.21	16.41
Frequency (Mode 7) (Hz)	17.85	17.64	16.28	18.83	18.63	21.60
Frequency (Mode 8) (Hz)	24.32	-	21.04	-	20.52	-
Frequency (Mode 9) (Hz)	26.95	-	21.58	-	25.33	-

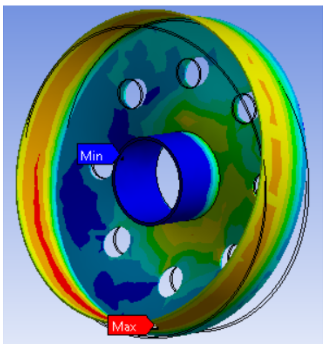

Figure 7. Maximum Equivalent Stress on the 8 Holes @ 3.5m (Mode 0)

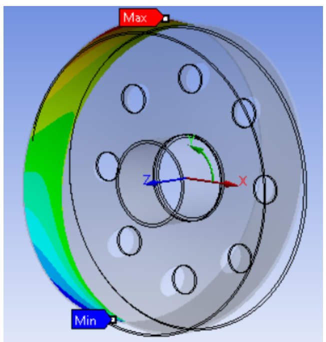

Figure 8. Maximum Radial Deformation on 8 Holes @ 3.5m (Mode 0)

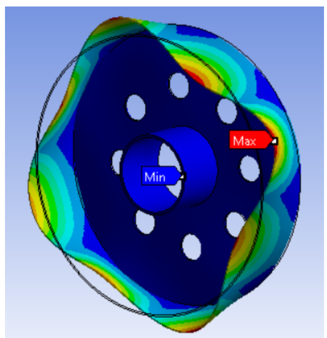

Figure 9. Mode Shape Deformation (Mode 8) for the Addition of Holes

Tables 4 shows the results for all holes at a distance of 3.5m from the centre for uniform and non-uniform loading.

Table 4. Results for the Optimised Design with the Addition of Holes at 3.5m Distance for Uniform (Mode 0) and Non-Uniform Loading (Mode 1)

Description	7 Holes		8 Holes		9 Holes	
	Mode 0	Mode 1	Mode 0	Mode 1	Mode 0	Mode 1
Mass (T)	207.43	180.14	179.67	161.69	168.04	214.95
Stress (MPa)	46.55	128.09	121.62	90.91	-10.21	122.41
Radial Deformation Min (mm)	-1.67	1.83	-1.37	-1.65	-1.98	-1.76
Radial Deformation Max (mm)	0.27	-1.15	-1.89	-1.22	-0.65	1.80
Frequency (Mode 6) (Hz)	16.00	16.44	16.25	16.64	16.29	17.74
Frequency (Mode 7) (Hz)	16.60	16.78	17.57	16.00	16.73	18.32
Frequency (Mode 8) (Hz)	24.90	25.82	23.24	-	28.39	-
Frequency (Mode 9) (Hz)	22.64	-	24.62	-	24.24	-

All scenarios met the set constraints but the scenario that provided the biggest average mass reduction when taking into account both loading cases, was 8 holes at a distance of 3.5m. Table 5 shows the dimensions that the rim and disc thicknesses need to be at to meet these requirements, along with the diameter of the holes.

Table 5. Dimensions for the Optimised Design for 8 Holes at 3.5m

Description	Initial	Mode 0	Mode 1
Disc Thickness (mm)	81.75	204.07	206.93
Rim Thickness (mm)	63.69	34.99	39.49
Hole Diameter (m)	1.00	1.70	1.60

Comparing these masses with the masses of the optimised base case (Table 1), shows a mass reduction of 20% and 34% for uniform and non-uniform loading respectively. Figures 10 and 11 show the modified rotor models.

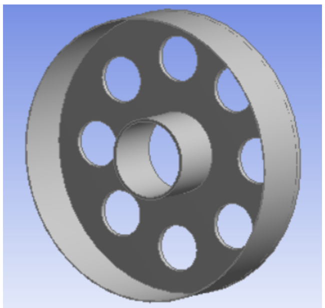

Figure 10. Optimised Design (Mode 0)

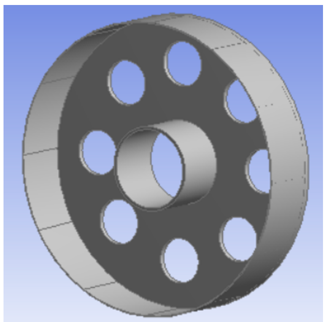

Figure 11. Optimised Design (Mode 1)

5.3. Addition of Holes and Strengthening Ribs with a Reinforcement of the Rim

It is normally observed that the rotor rim's local deformation contributes to the closing of the airgap (Figures 5 and 8). Therefore, adding a feature to increase the rim's local stiffness is desirable. Here, addition of the conventional triangular stiffeners are not feasible, as the design is an outer rotor. As such, one more scenario is considered as the addition of an outer rim with connecting flanges to seek the optimal solution in this scenario (third of Figure 2).

In section 5.2, the scenario which gave the best results were 8 holes at 3.5m radius, therefore, this scenario was selected and used as the design values for the starting point of optimisation in both these static and dynamic analyses.

Table 6 presents the results of the stress, deformation and natural frequency values post optimisation. Comparing the masses of optimised static and dynamics between Tables 6 and 1 (optimised baseline design), shows a mass increase of 14% and reduction of 19% for uniform and non-uniform loading respectively. However, the initially intended further mass reduction compared to those of Section 5.2 is not achievable through addition of this feature.

Table 6. Results for the Optimised Design with the Addition of 8 Holes at 3.5m Distance Plus Rim for Uniform (Mode 0) and Non-Uniform Loading (Mode 1)

Description	Section 5.2 Results		Static Only		Static and Dynamic	
	Mode 0	Mode 1	Mode 0	Mode 1	Mode 0	Mode 1
Mass (T)	179.67	161.69	253.88	198.02	257.65	196.92
Stress (MPa)	121.62	90.91	45.92	154.21	42.81	155.00
Radial Deformation Min (mm)	-1.37	-1.65	-2.01	-0.45	-2.03	-0.42
Radial Deformation Max (mm)	-1.89	-1.22	1.45	1.96	1.89	1.94
Frequency (Mode 6) (Hz)	16.25	16.64	-	-	20.51	19.40
Frequency (Mode 7) (Hz)	17.57	16.00	-	-	19.61	19.15
Frequency (Mode 8) (Hz)	23.24	-	-	-	-	-
Frequency (Mode 9) (Hz)	24.62	-	-	-	-	-

The optimised design parameters are shown in Table 7. It shows that disc thickness increases quite substantially in order to meet the set constraints. The modified rotor model is presented in Figure 12.

Table 7. Dimensions for the Optimised Design

Description	Initial Set Conditions	Mode 0	Mode 1
Disc Thickness (mm)	81.75	373.16	257.96
Rim Thickness (mm)	63.69	62.99	22.99
Hole Diameter (m)	1.00	2.17	1.92
Outer Rim Thickness (mm)	100.00	95.46	77.71
Outer Rim Length (mm)	200.00	336.38	186.47
Flange Height (mm)	259.30	1005.20	341.99
Flange Width (mm)	100.00	239.09	107.40

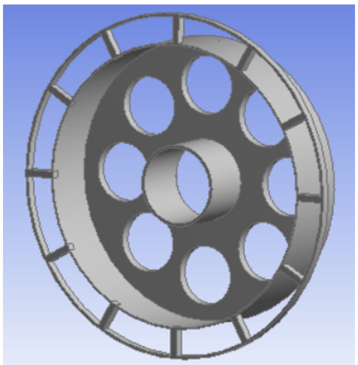

Figure 12. Optimised Design of the 8 Holes @ 3.5m Plus Rim Case (Mode 0)

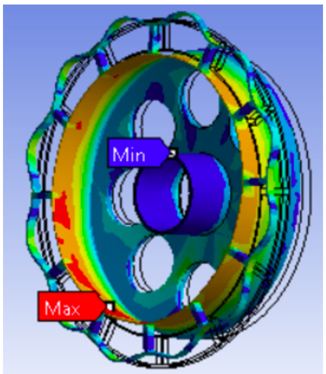

Figure 13. Maximum Stress on 8 Holes @ 3.5m Plus Rim (Mode 0)–Optimised

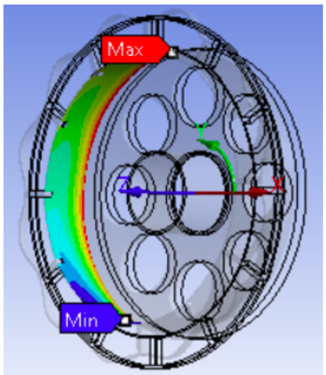

Figure 14. Maximum Deformation on 8 Holes @ 3.5m Plus Rim (Mode 0)–Optimised

6. Conclusion

In this piece of research, the NREL 15MW generator rotor structure has been optimised taking into consideration static and dynamic features. Finite element models with different loading conditions corresponding to Mode 0 and Mode 1 deflection have been analysed. The results obtained from the studies, reveal that it is possible to reduce the overall structural mass in a substantial manner.

It is necessary to study the structure from both static and dynamic viewpoints at the same time. If only the static features are used in the optimisation, then the resulting structure could be lighter but is at risk of resonance of some modes during operation.

It can also be concluded that it is important to consider both static loading conditions, uniform (Mode 0) and non-uniform (Mode 1) because the results obtained prior to optimisation showed that the static and dynamic features were slightly different for each condition, along with a difference in mass reduction post optimisation. Due to the fact that in real-world conditions no manufactured component and/or assembly is perfect, then it is important to take into consideration multiple loading conditions in order to cover all eventualities.

A number of modifications were made in the structure following the initial results. It's main dimensions were varied and a number of holes were positioned on the disc sub-structure. The number of holes, distance from the centre and size of holes were all altered and it was seen that they all had an effect on the end results. In all cases there was a mass reduction compared with the optimised base design but it was found that the optimal number of holes corresponded to 8 at a distance of 3.5m from the centre of the disc, for both loading conditions. The optimal diameter of the holes was approximately 2m. With the parametrisation of the problem, the effect caused by the independent variables in the mass optimisation process was evaluated. The common independent variables used in each case were the disc and rim thicknesses.

To improve the rim's local stiffness another scenario was defined, in which an outer stiffener rim was introduced to the optimal design with holes and the new scenario was optimised, subject to the static and dynamic constraints. This was done with the hope that further mass reduction can be obtained, however the result doesn't illustrate much success in this regard. The rotor design appears to rock about it's horizontal axis when all the forces and loading were applied, so by thickening up the disc sub-structure the rocking effect is slightly reduced, which in turn reduces the radial deformation. Therefore, as a future work, it is worth investigating the addition of some radial stiffeners to increase the disc's stiffness, enabling reduction of it's overall thickness.

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