Additive Manufacturing: A New Paradigm for the Next Generation of High-Power-Density Direct-Drive Electric Generators

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ADDITIVE MANUFACTURING: A NEW PARADIGM FOR THE NEXT GENERATION OF HIGH-POWER-DENSITY DIRECT-DRIVE ELECTRIC GENERATORS

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ABSTRACT
In recent years, there has been a growing demand for high-power-density direct-drive generators in the wind industry owing to their high reliability, torque per unit volume, and conversion efficiencies. However, direct-drive wind turbine generators are very large, low-speed electric machines, which pose remarkable design and manufacturing issues that challenge their upscaling potential and cost of implementation. With air-gap tolerance as the main design driver, the need for high stiffness shifts the focus toward support-structure design that forms a significant portion of the generator’s total mass. Existing manufacturing processes allow the use of segmented-steel-weldment disk or spoke-arm assemblies that yield stiffer structures per unit mass but tend to be heavier and more expensive to build. As a result, there is a need for a transformative approach to realize lightweight designs that can also facilitate series production at competitive costs. Inspired by recent developments in metal additive manufacturing (AM), we explore a new freedom in the structural design space with a high potential for weight savings in direct-drive generators. This includes the feasibility of using nonconventional complex geometries, such as lattice-based structures as structurally efficient options. Powder-binder jetting of a sand-cast mold was identified as the most feasible AM technology to produce large-scale generator rotor structures with complex geometry. A parametric optimization study was performed and optimized results within deformation and mass constraints were found for each design. The response to the maximum Maxwell stress due to unbalanced magnetic pull was also explored for each design. Further, a topology optimization was applied for each parameter-optimized design to validate results and provide insights into further mass reduction. These novel designs catered for AM are compared in both deflection and mass to conventional rotor designs using NREL’s systems engineering design tool, GeneratorSE. The optimized lattice design with a U-beam truss resulted in a 24% reduction in structural mass of the rotor and 60% reduction in radial deflection. It is demonstrated that additive manufacturing shifts the focus from manufacturability constraints toward lower mass.

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NOMENCLATURE

\( r_s \) Air-gap radius  
\( \dot{B}_g \) Peak air-gap flux density  
\( B_r \) Remnant flux density  

— For additional symbols refer to relevant tables

INTRODUCTION

In recent years, wind power has become one of the more popular forms of renewable energy. Various technological innovations have led to building larger and taller wind turbines to drive down the cost of energy while presenting remarkable challenges in design, installation, transportation, and operation. Because wind turbine components are subject to complex dynamic loading, vibration, system fatigue, and joint stresses, numerous manufacturing issues remain to be addressed.

One of the issues is the gearbox used in the powertrain, in which the manufacturers strive for weight reduction and improved assembly of threaded joints with close tolerances to improve their service lives. The move toward offshore installations with demand for higher reliability and efficiency is encouraging more wind turbine manufacturers to integrate higher power generators into their designs. Direct-drive generators are one of the many technology options touted to realize these goals. The direct coupling of the generator to the turbine leads to low-speed high-torque machines with a high pole count and very large stator diameters that pose remarkable design and manufacturing challenges [1]. Particularly with permanent-magnet (PM) generators, there are high magnetic forces constantly acting between the rotor and stator; hence, accurately fixing the position of the stator teeth and the windings in the stator is immensely challenging [2]. Moreover, the air gap between the stator and rotor must be kept uniform and small for proper torque transfer. As a result, designers are constantly seeking measures to minimize eccentricity to prevent potentially fatal distortions. Most designs employ substantial spoke-and-rim wheel structures to maintain the necessary structural integrity required when welding overly thick steel elements [3]. The state of the art shows that direct-drive generators are difficult to scale up to 10 MW and beyond [4], with traditional designs weighing between 200 and 500 tons [5]. With greater emphasis on top-head mass reduction, lightweight designs are needed so the final system can be assembled more easily. This need is urging manufacturers to develop, design, and use manufacturing technologies that will enable them to produce machines on a large-scale basis and at a competitive cost. However, this oftentimes conflicts with the need to fulfill demanding performance specifications such as efficiency, torque quality, and fault tolerance [1].

Conventional direct-drive generator support structures are made of spoked-wheel structures that support rims of iron/magnetic core laminations. Typical geometries are disks, structural profiled spokes/tension rods, ribs, or support arms. Their structural stiffness and interfaces are controlled by manufacturing tolerances [3]. Because these machines operate at very low rotational speeds, their corresponding torques and diameters have to be made large enough to generate the required power output. The air gap separating the rotor and stator of the machine is typically \( 1/1000 \) of its diameter. Therefore, eccentricities or closing of the air gap are major concerns. These effects drive the structural designs to be extremely rigid and massive. The spokes have to be increased dimensionally and in quantity or discs have to be made stiffer by adding more steel. The spokes can be manufactured as either continuous cast-iron or segmented-steel-weldment assemblies that yield stiffer structures per unit mass; however, they are expensive to build.

Design for Lightweight Generators

Lightweighting for large, direct-drive wind turbine generators is not a new concept. In the past, several researchers have considered optimization of direct-drive generator designs [6,7,8], with a focus on structural mass reduction [9,10]. Spooner et al. [11] explored lightweight spoked structures to reduce the structural mass of the stator by 70%-80%. However, they used an ironless stator for low-power machines. This spoked design can be implemented for high-power machines, but new geometries are needed to maintain strength with decreased total mass. A few researchers have resorted to mathematical methods such as topology optimization (TO) algorithms for lightweighting for direct-drive generators. Topology optimization is a mathematical method that optimizes the material layout within a given design space, for a given set of loads, boundary conditions, and constraints. Zavvos [10] used finite-element analysis (FEA) and topology optimization techniques to identify lightweight structures for large transverse flux PM generators. Zavvos et al. [9] used shape optimization that offered an additional 26% reduction in mass. Kirschneck et al. [12] investigated possible weight reduction for the structure of the generator rotor with emphasis on dynamic behaviour under magnetic loading. However, the optimized results did not perform better than the original design. Lee et al. [13] used shape optimization techniques to reduce the structural weight of an axial flux PM generator by up to 21%. Gangl et al. [14] utilized a shape derivative during optimization of an interior rotor PM motor. Their approach was based on a shape-Lagrangian method adapted for magnetostatic problems involving nonlinear partial differential equations (PDEs). The resulting shape reduced the cost function by 27% and improved the electric rotor’s rotation pattern and the motor’s overall performance. Numerous algorithms for Topology Optimization exist, including an open-source model by Zuo and Xie [15], which generates an FEA mesh and topology solver using the bidirectional evolutionary structural optimization method. Advantages of this code include its compactness (100 lines of code), ability to extend to other applications (open source), and utility for me-
chanical applications (three-dimensional geometries). Figure 1 depicts the input and output of the optimization software. Commercial FEA software [16] facilitates shape optimization according to discretized load functions. However, the major limitation of the software is that the resulting complex geometries are often difficult to manufacture.

![Figure 1: Topology optimization of disk geometry [15]](image)

**Design for Manufacturability**

More recently, attempts have been made to investigate designs that can potentially reduce manufacturing costs of generators. Tessarolo et al. [1] proposed a highly modular design to segment the generator into independent units that can be connected in parallel and used to feed separate power converters. The generator design had intrinsic fault tolerance through simple part replacement and repairs in case of a fault. Skemkken [17] proposed slanted spoke structural wheel architectures, which were constructed using layered sheet-steel elements to form the spokes and rim of the wheel faces. In a wind turbine, direct-drive electric generators can be subjected to continual flexing and extreme temperatures that can be potentially catastrophic. Eliminating deep structural welds can significantly reduce manufacturing cost and avoid problems associated with fatigue cracking and failure of welded connections [17]. Data from a global wind report [18] suggest that the cost per mass (inactive structural cost and labor cost) of a 5-MW PM direct-drive (PMDD) generator is $6.04/kg. However, the validity of material cost, labor, and allocation assumptions, and costs of stator casing, active components, burden, or engineering costs are not available for reference. At the same time, as new manufacturing techniques evolve, it is important to gain greater understanding of their associated costs.

**Additive Manufacturing for Direct-Drive Generators**

In recent years, metal additive manufacturing has gained momentum and is emerging across a broad range of sectors, including automotive, medical, and aerospace. A vast range of metallic materials, including several types of steel, titanium, cobalt, and aluminum alloys, are printable. Newer materials for soft-magnetic cores are being introduced and show promising results for manufacturing laminations in electric machines [19,20]. AM technologies such as selective laser melting and direct metal laser sintering provide great potential for weight savings [21,22], thereby allowing the manufacturing of complex geometries arising from topology optimization. Binder-jetting for sand casting is also being considered as an attractive process for making metal parts [23]. With binder-jetting, there are no induced thermal stresses. Further, it is capable of producing very large-scale molds that can be used for casting direct-drive generators. The use of 3-D sand-cast printing allows for complex geometries and undercuts without the need for conventional mold cores. Because sand-cast printing is more complex and therefore more expensive, the ability to decrease related costs helps reduce the manufacturing difficulty, thereby making this method more economically feasible [24]. AM has the potential to realize designs that are structurally efficient, yet complex structures that enable high stiffness and reduced weight. However, there is relatively little information available in the literature about designing large-scale components, particularly concerning design tools, structural analysis, and postprocessing for functional metallic components [25].

Inspired by these recent developments and challenges, this study attempts to identify manufacturing solutions by designing an innovative, lightweight direct-drive PM generator for a wind turbine. The resulting design aims to provide better power scalability and weight reduction, as well as reduce series manufacturing costs while preserving its structural integrity. Specifically, powder-binder jetting of a sand-cast mold is selected as a potential process to manufacture complex rotor geometries. The geometries in this study include a hollow truss lattice and U-beam lattice truss design for a 5-MW direct-drive generator. These geometries were analyzed using FEA in an ANSYS environment and an optimization study was performed to determine optimal geometric parameters. The results were compared against reference geometries created using the National Renewable Energy Laboratory’s (NREL’s) GeneratorSE [26] to identify potential improvements.

This report is available at no cost from the National Renewable Energy Laboratory (NREL) at www.nrel.gov/publications.
Permanent-Magnet Direct-Drive Generator Rotor Design – An Overview

For this study, a 5-MW radial flux permanent-magnet synchronous generator with interior rotor arrangement was chosen as the baseline design. This design endures torque to the order of 4.14 MNm. High-strength permanent magnets are fixed circumferentially around the rotor in a wheel-type arrangement that is mounted onto a shaft coupled to the turbine hub. Common structures for the rotor geometry employ spoked-arm or disk construction that ensure adequate structural stiffness against forces that act to close the air gap. Figure 2 and Table 1 provide a visualization of the structural design variables. NREL’s rigorous structural analyses for these designs [26] provided the optimized design parameters for the baseline design with estimates for structural mass (Table 2). Note that both designs have significantly different magnitudes of deflections, however they are both considered negligible. Also, the lower deflections for the disk design is accomplished by the considerable increase in structural stiffness (solid disk weighs more than the spoked arms).

LARGE-SCALE METAL ADDITIVE MANUFACTURING

Metal additive manufacturing poses significant challenges compared to fused deposition modeling or stereo lithography, namely that the feedstock is a metal powder that must be fused. Popular powder bed fusion processes, such as direct metal laser sintering, use a laser beam to create a high-temperature melt pool in which powder fuses to form the part. This type of sintering is common for small, detailed parts; however, it lacks speed and requires thermal management and postprocessing [27]. Electron beam melting consists of an electron beam in a vacuum chamber that fuses the powder. This process has the advantage of higher deposition rates and power efficiency, yet it has high structural costs due to radiation concerns. Both electron beam melting and direct metal laser sintering machines use a powder bed, which, for a large part such as a direct-drive generator, results in unacceptable material costs to fill the powder bed. Also, developing process parameters for new materials in powder bed fusion processing is extremely difficult. Owing to these challenges, few companies [28, 29] are considering indirect AM as a feasible option to produce large, complicated geometries. The technique employs powder-binder jetting of sand to create a sand-cast mold. Printing a mold allows for unique geometries and undercuts that are achievable with AM while also allowing traditional casting for the finished part. Typical build platforms are 2.2 x 1.2 x 0.6 m in size [28]. First, during printing, a “green” part is created by alternating a layer of 100 µm sand with a layer of binder from an inkjet printer head. Next, the z-axis powder stage moves up 100 µm and a roller ensures an even coating over the work bed. An inkjet printer head then deposits a layer of aque-

<table>
<thead>
<tr>
<th>TABLE 1. DESIGN VARIABLES</th>
</tr>
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<tbody>
<tr>
<td>Symbol</td>
</tr>
<tr>
<td>t_r</td>
</tr>
<tr>
<td>n_r</td>
</tr>
<tr>
<td>h_yr</td>
</tr>
<tr>
<td>b_r</td>
</tr>
<tr>
<td>l</td>
</tr>
<tr>
<td>d</td>
</tr>
<tr>
<td>t_w</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>TABLE 2. BASELINE DESIGN: MASS AND DEFORMATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
</tr>
<tr>
<td>-----------------------------------------------</td>
</tr>
<tr>
<td>Radial Deformation</td>
</tr>
<tr>
<td>Axial Deformation</td>
</tr>
<tr>
<td>Torsional Deformation</td>
</tr>
<tr>
<td>Active Mass</td>
</tr>
<tr>
<td>Structural Mass</td>
</tr>
</tbody>
</table>
ous binder in the correct location to build up the part. The binder is cured under an ultra-violet light and another 100 µm layer of powder deposited on the build platform. This process is repeated layer by layer until the finished part is produced. Fig. 3 depicts the powder-binder-jetting printing process.

![FIGURE 3. Powder-binder printing schematic](image)

Through powder-binder printing, the sand-cast mold can be directly printed. In this way, the mold cavity, risers, and cope are all directly printed in one step. Next, molten steel can be poured to create the optimized rotor geometry such as a lattice. Figure 4 shows the top and bottom molds used to print a lattice design.

![FIGURE 4. Molds for printing a lattice design: (a) bottom mold, and (b) top mold](image)

A unique feature about sand-cast printing is that “complexity is free.” Unlike traditional casting in which the price of the mold dramatically increases with complexity, an AM-printed mold’s cost is solely based on materials; therefore, increasing the complexity while maintaining constant volume results in a constant cost, as shown in Fig. 5.

![FIGURE 5. Cost versus complexity of AM and conventional mold making](image)

**ROTOR REDESIGN AND STRUCTURAL ANALYSIS**

First, the baseline models from GeneratorSE [26] were evaluated for their potential for mass reduction. To make a reasonable comparison and avoid changes to the electromagnetic design, we chose to retain the thickness of the rotor cylinder to be the same as the baseline rotor. The rest of the existing design space (the region occupied by the disk and arms) was re-evaluated for optimal material layout for a given set of loads, boundary conditions, and constraints, with the goal of minimizing the weight. As a next step, two alternative designs involving complex lattice structures were evaluated as potential design solutions. Lattice structures are open cellular structures with a continuous network of truss-like members that can be arranged in various configurations to achieve different strengths suitable for specific applications [25]. The cross section of the trusses can be designed to be hollow, circular, square, or any desired shape that gives them high, compressive load-bearing capacity. Although lattice structures have the potential to serve as structural members, the final components often have a complex topology, which is difficult to fabricate using conventional subtractive manufacturing methods such as machining. However, powder-binder jetting offers unprecedented design and material freedom to manufacture such complex geometries made of lattice structures.

For the purpose of evaluating the different designs, the following loading scenarios were considered based on McDonald’s work [8] to simulate rotor deformations by static structural analyses in ANSYS (Fig. 6 depicts the coordinate system used in rotor analysis):

1. Gravitational loading. A gravitational force of $9.81\,m/s^2$ acting in the negative Z direction accounts for the worst-case
axial deflection due to ground transportation.

2. Maxwell stress. The normal component of the Maxwell force acting radially outward between the rotor magnets and stator coils determined by:

$$\sigma_{\text{maxwell}} = \frac{\hat{B}^2}{2\mu_0}$$  \hspace{1cm} (1)

3. Centripetal force. The high torque induces a centripetal force that acts along the circumference of the rotor in the +Y direction (a maximum shear stress of 40 kPa was assumed).

Other forces exist, such as the shear component of the air-gap force, thermal expansion, and forces and moments of the five other degrees of freedom for the turbine system. However, it is assumed that all nontorque forces are transferred via the bearings, to the bed plate, and into the tower. Therefore, these forces were assumed to be negligible. Additionally, adequate cooling was assumed so that deformation from thermal expansion was small enough to be ignored.

Topography Optimization Using ANSYS

ANSYS provides an FEA topology optimization approach that removes material under specified boundary conditions and a constraint function. Selecting the required percent mass reduction results in the solver removing structurally unnecessary material until the desired mass reduction is achieved. For each additive-manufacturing-catered design in this study, topology optimization was applied to determine further areas of mass reduction. The mesh was refined for topology optimization to give a clearer picture of where material is removed. Although this was computationally expensive, it aided in developing new geometries. Also, the optimizer was unable to resolve inertia loading as the mass changed each iteration. A first-order approximation of the weight-reduction potential is given through Topology Optimization of two conventional rotor designs (spoke arm and disk arm) for a 5-MW generator.

1. Spoke-arm design

Running the 5-spoke-arm design through topology optimization forcing a 60% mass reduction suggests greater opportunity for structural optimization of the rotor. 60% was chosen based on iteration until the most material could be removed while maintaining the base shape. Figures 7 and 8 depict the topology optimization results.

2. Disk design

Running the disk rotor design through ANSYS topology optimization forcing a 60% mass reduction indicates the optimization’s directional dependence. The optimized design spokes are offset from the normal position in the direction of the stress.

FIGURE 6. Coordinate system for loading criteria

FIGURE 7. Topology optimization results for spoked-arm design: (a) preoptimization, where red depicts areas to be removed, and (b) postoptimization showing removed material

FIGURE 8. Topology optimization results for disc design: (a) preoptimization, where red depicts area to be removed, and (b) postoptimization showing removed material
Designing for Rotor Eccentricity

Deflection of the rotor in the radial direction as a result of loading conditions can cause one side of the rotor to be closer to the stator coils than the other. As a result, there is an unbalanced magnetic pull, which is stronger in the location of decreased air gap. Variables influencing the maximum normal stresses experienced by the rotor for the 5-MW spoked-arm rotor design are given in Table 3.

### TABLE 3

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \tau_s )</td>
<td>Air-gap radius</td>
<td>m</td>
<td>3.376</td>
</tr>
<tr>
<td>( l_s )</td>
<td>Core length</td>
<td>m</td>
<td>1.489</td>
</tr>
<tr>
<td>( s )</td>
<td>Number stator slots</td>
<td>–</td>
<td>1128</td>
</tr>
<tr>
<td>( p )</td>
<td>Number of magnet pole pairs</td>
<td>–</td>
<td>188</td>
</tr>
<tr>
<td>( m )</td>
<td>Number phases</td>
<td>–</td>
<td>3</td>
</tr>
<tr>
<td>( q )</td>
<td>Number slots per pole phase</td>
<td>–</td>
<td>1</td>
</tr>
<tr>
<td>( B_r )</td>
<td>Remnant flux density</td>
<td>Tesla</td>
<td>1.2</td>
</tr>
<tr>
<td>( \tau_p )</td>
<td>Pole pitch</td>
<td>mm</td>
<td>58.00</td>
</tr>
<tr>
<td>( \tau_s )</td>
<td>Stator slot pitch</td>
<td>mm</td>
<td>19.30</td>
</tr>
<tr>
<td>( h_m )</td>
<td>Magnet height</td>
<td>mm</td>
<td>8.53</td>
</tr>
<tr>
<td>( h_{yr} )</td>
<td>Rotor yoke height</td>
<td>mm</td>
<td>45.00</td>
</tr>
<tr>
<td>( h_{ys} )</td>
<td>Stator yoke height</td>
<td>mm</td>
<td>71.36</td>
</tr>
<tr>
<td>( B_s )</td>
<td>Stator slot width</td>
<td>mm</td>
<td>8.70</td>
</tr>
<tr>
<td>( B_{so} )</td>
<td>Stator wedge width</td>
<td>mm</td>
<td>5</td>
</tr>
<tr>
<td>( \gamma_s )</td>
<td>Air-gap correction factor</td>
<td>–</td>
<td>-0.0409</td>
</tr>
<tr>
<td>( g )</td>
<td>Air-gap length</td>
<td>mm</td>
<td>6.90</td>
</tr>
<tr>
<td>( g_{eff} )</td>
<td>Effective air-gap length</td>
<td>mm</td>
<td>14.40</td>
</tr>
<tr>
<td>( \mu_{rm} )</td>
<td>Relative permeability</td>
<td>–</td>
<td>1.06</td>
</tr>
<tr>
<td>( \mu_0 )</td>
<td>Permeability of free space</td>
<td>–</td>
<td>4 ( \pi \times 10^{-7} )</td>
</tr>
<tr>
<td>( K_{cs} )</td>
<td>Carter factor for stator slots</td>
<td>–</td>
<td>0.9693</td>
</tr>
<tr>
<td>( \hat{B}_{pm} )</td>
<td>Peak magnet flux density</td>
<td>Tesla</td>
<td>Varies</td>
</tr>
<tr>
<td>( \hat{B}_g )</td>
<td>Peak air-gap flux density</td>
<td>Tesla</td>
<td>Varies</td>
</tr>
<tr>
<td>( \sigma_{\text{maximum}} )</td>
<td>Maximum normal Maxwell stress</td>
<td>MPa</td>
<td>Varies</td>
</tr>
</tbody>
</table>

An air-gap correction factor with the Carter factor for stator slots (to account for the slot geometry) was used to calculate an average effective air gap [6].

\[
g_{\text{eff}} = K_{cs}(g + \frac{h_m}{\mu_{rm}})\tag{2}\]

where the Carter factor, \( K_{cs} \), is given by

\[
K_{cs} = \frac{\tau_s}{\tau_s - \gamma_s(g + \frac{h_m}{\mu_{rm}})}\tag{3}\]

\[
\gamma_s = (\frac{4}{\pi}) \frac{0.5b_{so}}{g + \frac{h_m}{\mu_{rm}}} \tan^{-1}\left(\frac{0.5b_{so}}{g + \frac{h_m}{\mu_{rm}}} \right) - \ln\left(1 + \left(\frac{0.5b_{so}}{g + \frac{h_m}{\mu_{rm}}} \right)^2\right)\tag{4}\]

The effective air gap was used to calculate the peak permanent-magnet flux density:

\[
\hat{B}_{pm} = \frac{B_r h_m}{\mu_{rm} g_{\text{eff}}}\tag{5}\]

The peak air-gap flux density was calculated using:

\[
\hat{B}_g = B_r (\frac{h_m}{\mu_{rm} g_{\text{eff}}})(\frac{4}{\pi}) \sin\left(\frac{\pi B_m}{2 \tau_p}\right)\tag{6}\]

Solving for the peak air-gap flux density provided a Maxwell normal stress of 0.1976 MPa that matched data from NREL’s Generator SE [26]. However, the magnet remnant flux density is assumed constant in these calculations. In reality, the remnant flux density may range from 1.1 to 1.4 Tesla depending on the magnet manufacturer and quality control. Additionally, to account for rotor eccentricity and radial deformations, the maximum Maxwell stress is found for the case where the air gap reduces to 0 mm (i.e., the rotor and stator are touching) and plotted visually for various remnant flux densities shown in Fig. 9.

The maximum Maxwell stress as a function of air-gap distance is plotted in Figure 10. The maximum Maxwell stress possibly achieved in the rotor is in the case when \( g \) tends to 0; therefore, the worst-case loading scenario occurs with this maximum Maxwell stress.

\[
\sigma_{\text{minimum}}(B_r = 1.1) = 0.196 \text{ MPa}\tag{7}\]

\[
\sigma_{\text{maximum}}(B_r = 1.4) = 1.104 \text{ MPa}\tag{8}\]

Because this maximum Maxwell stress occurs in regions where the air gap is closer to 0, it must be distributed around the circumference of the rotor decaying toward zero on the opposite
side. This is because if the deflection causes a smaller air gap on one side of the rotor, it can cause a similar increase in air gap on the opposite side. Because there will be a large air gap increase directly opposite of this point load of maximum Maxwell stress, it is assumed the Maxwell stress is 0. To simulate this loading condition, the maximum stress with either $B_r=1.1$ or $B_r=1.4$ is computed. Next, the circumferential direction is divided into $N$ subcomponents, which yield the angular step in degrees:

$$\delta \theta = \frac{2 \pi}{N};$$

assuming 100 subdivisions yielded an angular step of 3.6 degrees. Next, the maximum stress was distributed over the circumference evenly on both sides using the cosine relationship:

$$\sigma(\theta) = \sigma_{\text{maximum}} \cos(\theta); \quad \frac{\pi}{2} < \theta < \frac{3\pi}{2}$$  \hspace{1cm} (9)

Note that because of coordinate system notation, this pressure is applied as a negative pressure in ANSYS to ensure proper outward orientation as the Maxwell stress is an attractive stress. This model is an inherent upper-bound estimation of the absolute maximum stress, as it is spread over a 50-degree range, or $5\pi/18$ range of angular values. In reality, the maximum stress will be localized to a much smaller angular range.

Both Maxwell conditions are applied for each subsequent design to determine the deflections associated with each case. It should be noted that the maximum deflection is computed in the case in which the maximum Maxwell stress is applied (i.e., if the rotor were to touch the stator). The deflection from this loading situation is computed and compared against allowable deflections. A successful design is one where the computed deflections from this maximum loading case is below the allowable deformation limits.

**ALLOWABLE DEFORMATION LIMITS**

The allowable deformation limits of the rotor provide the basis of the structural evaluation of each new design. Each limit shown below is given for a 5-MW machine based on disk-arm design from NREL’s GeneratorSE [26].

1. **Radial Deflection:** No more than 10% of the air-gap length

   $$(0.10)(0.002r_s) = 0.35 \text{ mm}$$

2. **Axial Deflection:** No more than 2% of the axial length

   $$0.02l_s = 30 \text{ mm}$$

3. **Torsional Deflection:** No more than 0.05° angle of twist

   $$\frac{0.05^\circ \pi r_s}{180^\circ} = 3 \text{ mm}$$

**ALTERNATIVE DESIGNS AND DESIGN PARAMETERS**

1. **Lattice design with hollow truss**

   To make a reasonable comparison and avoid changes to the electromagnetic design, we chose to retain the thickness of the rotor cylinder to be the same as the baseline rotor structural model. A high-strength structure in the form of a lattice design with hollow trusses (spokes) was created to fill the space originally occupied by the spoke-arm design. The lattice consisted of repetitions of support ribbon and trusses.
distributed radially around the rotor. For this study, the location of the ribbons was fixed and their thickness and width allowed to change. This decision was made from preliminary simulations that indicated that a symmetric design does indeed have the greatest structural strength. A parametric optimization was carried out to remove as much structural material while ensuring the lowest weight and maximum strength. The optimized solid design proved to be heavier than the original spoked-arm designs. To attempt to remediate this problem, the trusses in lattice were made hollow with a variable wall thickness. A schematic of the hollow truss design with optimization variables is shown in Fig. 11.

![Fig. 11. Optimization parameters of a 5-MW lattice support with hollow trusses showing a cross-sectional view](image)

Table 4 indicates the design variables of the hollow truss design as well as their optimized value.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Design Dimension</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$w_d$</td>
<td>Disk width</td>
<td>mm</td>
<td>181.5</td>
</tr>
<tr>
<td>$d_s$</td>
<td>Truss diameter</td>
<td>mm</td>
<td>92.41</td>
</tr>
<tr>
<td>$w_s$</td>
<td>Truss wall thickness</td>
<td>mm</td>
<td>44.51</td>
</tr>
<tr>
<td>$w_r$</td>
<td>Ribbon width</td>
<td>mm</td>
<td>209.00</td>
</tr>
<tr>
<td>$t_r$</td>
<td>Ribbon thickness</td>
<td>mm</td>
<td>72.95</td>
</tr>
<tr>
<td>$n_{trusses}$</td>
<td>Number of trusses</td>
<td>–</td>
<td>19</td>
</tr>
</tbody>
</table>

2. **Lattice design with U-beam truss**
   A mold analysis was carried out to examine the feasibility of printing the various designs. Initial results suggested that the lattice design with hollow trusses was impossible to make using AM powder-binder printing technology. This is because the spoke necessitated a solid core floating in the center of the spoke to ensure it casts hollow. However, this design is impossible to create as the core must be supported in the center. To solve this problem, a lattice design with a U-beam truss was created, thereby changing the cross section of the trusses from a circle to a U-beam.

![Fig. 12. Optimization parameters of a 5-MW lattice support with U-beam trusses](image)

Highlighted design parameters in Fig. 12 depict the optimization variables used in ANSYS to select the optimized design. Table 7 indicates the design variables as well as optimized values.

**RESULTS**

1. **Hollow truss lattice design**
   The lattice structure with a solid truss design exhibited strong reduction in radial and torsional deformation. Therefore, creating hollow trusses facilitates mass reduction without sacrificing the deformation advantages.

   (a) **Mesh convergence**
   For the hollow truss design, mesh independence was validated for the total deformation for three different mesh sizes. The change in total deformation was less than 5% indicating confidence in mesh sizing at the lowest bound of 294,000 elements. However, care should be taken in confirming this result for the torsional deflection as it was observed to vary by as much...
as 10% with refinements to the mesh. Therefore, we conducted parameter and topology optimization with a coarser mesh to save computational resources as well as performed static analysis using a fine mesh for more accurate deformations. The mesh independence study depicted a 7% change in total deformation with mesh sizes from 294,000 to 2,146,000 elements.

(b) **Parameter optimization**

Results from the parameter optimization study in the ANSYS workspace are presented in Table 5.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Design Dimension</th>
<th>Unit</th>
<th>Value&lt;sub&gt;optimized&lt;/sub&gt;</th>
</tr>
</thead>
<tbody>
<tr>
<td>w&lt;sub&gt;d&lt;/sub&gt;</td>
<td>Disk width</td>
<td>mm</td>
<td>181.5</td>
</tr>
<tr>
<td>t&lt;sub&gt;d&lt;/sub&gt;</td>
<td>Disk depth</td>
<td>mm</td>
<td>90</td>
</tr>
<tr>
<td>d&lt;sub&gt;t&lt;/sub&gt;</td>
<td>Truss diameter</td>
<td>mm</td>
<td>92.41</td>
</tr>
<tr>
<td>t&lt;sub&gt;ws&lt;/sub&gt;</td>
<td>Truss wall thickness</td>
<td>mm</td>
<td>44.51</td>
</tr>
<tr>
<td>w&lt;sub&gt;r&lt;/sub&gt;</td>
<td>Ribbon width</td>
<td>mm</td>
<td>209</td>
</tr>
<tr>
<td>t&lt;sub&gt;r&lt;/sub&gt;</td>
<td>Ribbon thickness</td>
<td>mm</td>
<td>72.95</td>
</tr>
<tr>
<td>n&lt;sub&gt;trusses&lt;/sub&gt;</td>
<td>Number of trusses</td>
<td>–</td>
<td>19</td>
</tr>
</tbody>
</table>

Radial deformation | mm | 0.212 |
Axial deformation | mm | 1.71x10<sup>-3</sup> |
Torsional deformation | mm | 5.404 |
Structural mass | mT | 18.27 |
Active mass | mT | 16.47 |
Total mass | mT | 34.74 |

Figures 13-15 present results for FEA deformations found with ANSYS for the hollow-truss-lattice design.

(c) **Deformation with varied magnetic loading**

Table 6 lists the deformation associated with the hollow-truss design under the two extreme cases of magnetic loading.

(d) **Topology optimization of the design**

Fig. 16 illustrates areas of further mass reduction of the hollow-truss geometry. Results were extracted from ANSYS after imposing 0.20 MPa of normal stress and 40 kPa of shear stress. Topology optimization results suggested further mass reduction of the 5-MW hollow-truss design by:

(i) Creating an hourglass shape for the trusses instead of a continuous circular cross section
(ii) Removing the two outermost ribbons.

The hollow-truss design resulted in a 13% structural mass reduction as compared to the solid-truss design. However, this is still 11% greater structural mass than
the conventional spoke-arm design, warranting further study for mass reduction.

**TABLE 6. MAXIMUM DEFORMATION OF THE HOLLOW-TRUSS DESIGN UNDER DIFFERENT MAGNETIC LOADING**

<table>
<thead>
<tr>
<th>Direction</th>
<th>Unit</th>
<th>$B_r = 1.1$ Tesla</th>
<th>$B_r = 1.4$ Tesla</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial deformation</td>
<td>mm</td>
<td>1.985</td>
<td>3.084</td>
</tr>
<tr>
<td>Torsional deformation</td>
<td>mm</td>
<td>6.305</td>
<td>6.803</td>
</tr>
<tr>
<td>Axial deformation</td>
<td>mm</td>
<td>$2.06 \times 10^{-3}$</td>
<td>$4.43 \times 10^{-3}$</td>
</tr>
</tbody>
</table>

**FIGURE 16.** Topology optimization results for hollow-truss-lattice design: (a) pretopology optimization, and (b) post-topology optimization

2. **Lattice design with U-beam truss**

Owing to the challenges in printing a hollow-truss design, a U-beam design was chosen for the cross section of the truss and optimized. The U-beam was oriented along the radial direction to provide the highest moment of inertia in the direction of twist to reduce torsional deformation.

(a) **Mesh convergence**

Because of a more complicated cross section, attention must be given to meshing parameters to determine mesh independence. Both the x and z directions are readily mesh-independent, with less than a 2.5% change in deformation with refinements from 135,000 to 4,308,000 elements. The torsional deformation was considered mesh-independent, with a change in mesh sizing of less than 5%. Therefore, we selected a middle ground of a 592,000 element mesh arising from web element sizes of 40 mm and general element sizes of 60 mm.

(b) **Parameter optimization**

Results from the parameter optimization study in the ANSYS workspace are summarized in Table 7. Figures 17-19 indicate FEA deformations found with ANSYS for the U-beam-truss lattice design.

**TABLE 7. U-BEAM-TRUSS DESIGN: PARAMETER OPTIMIZATION RESULTS (TORQUE LOADING)**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Design Dimension</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$w_d$</td>
<td>Disk width</td>
<td>mm</td>
<td>192.67</td>
</tr>
<tr>
<td>$t_d$</td>
<td>Disk depth</td>
<td>mm</td>
<td>90</td>
</tr>
<tr>
<td>$w_r$</td>
<td>Ribbon width</td>
<td>mm</td>
<td>227.23</td>
</tr>
<tr>
<td>$t_r$</td>
<td>Ribbon thickness</td>
<td>mm</td>
<td>78.73</td>
</tr>
<tr>
<td>$w_k$</td>
<td>Truss width</td>
<td>mm</td>
<td>167.55</td>
</tr>
<tr>
<td>$w_a$</td>
<td>Truss arm width</td>
<td>mm</td>
<td>42.68</td>
</tr>
<tr>
<td>$h_s$</td>
<td>Truss height</td>
<td>mm</td>
<td>94.28</td>
</tr>
<tr>
<td>$n_{trusses}$</td>
<td>Number of trusses</td>
<td>–</td>
<td>27</td>
</tr>
<tr>
<td>Radial deformation</td>
<td>mm</td>
<td>0.208</td>
<td></td>
</tr>
<tr>
<td>Axial deformation</td>
<td>mm</td>
<td>$3.29 \times 10^{-5}$</td>
<td></td>
</tr>
<tr>
<td>Torsional deformation</td>
<td>mm</td>
<td>2.87</td>
<td></td>
</tr>
<tr>
<td>Active mass</td>
<td>mT</td>
<td>16.47</td>
<td></td>
</tr>
<tr>
<td>Structural mass</td>
<td>mT</td>
<td>12.47</td>
<td></td>
</tr>
<tr>
<td>Total mass</td>
<td>mT</td>
<td>28.95</td>
<td></td>
</tr>
</tbody>
</table>

**FIGURE 17.** U-beam-truss design: radial deflection

This report is available at no cost from the National Renewable Energy Laboratory (NREL) at www.nrel.gov/publications.
(c) Maximum deformation associated with changes in magnetic loading

The maximum deformations due to worst-case scenario Maxwell loading conditions with the smallest air gap are shown in Table 8. It should be noted that the radial and torsional deformations increase above acceptable limits; however, as the worst-case scenario, actual rotor forces will be below these bounds. Low deflections seen in the torque loading case are representative of actual rotor operation.

<p>| TABLE 8. DEFORMATION OF U-BEAM-TRUSS LATTICE DESIGN UNDER DIFFERENT MAGNETIC LOADING |
|---------------------------------|--------------|--------------|--------------|</p>
<table>
<thead>
<tr>
<th>Direction</th>
<th>Unit</th>
<th>$B_r = 1.1$ Tesla</th>
<th>$B_r = 1.4$ Tesla</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial deformation</td>
<td>mm</td>
<td>2.61</td>
<td>4.22</td>
</tr>
<tr>
<td>Torsional deformation</td>
<td>mm</td>
<td>4.24</td>
<td>5.10</td>
</tr>
<tr>
<td>Axial deformation</td>
<td>mm</td>
<td>$5.1 \times 10^{-3}$</td>
<td>$1.64 \times 10^{-3}$</td>
</tr>
</tbody>
</table>

(d) Topology optimization of design

Following parameter optimization in ANSYS, topology optimization was performed on the U-beam-truss design to determine additional opportunities for mass reduction. Figure 20 presents the results of topology optimization.

Further mass reduction from the U-beam-truss design was possible by:
(i) Removing the outermost set of rings on the lattice
(ii) Decreasing the thickness of the inner disk.

The optimized U-beam-truss lattice design proved to be at least 40% lighter than the solid-truss lattice design and 32% lighter than the hollow-truss lattice design. Additionally, the U-beam-truss lattice utilized 24% less structural mass than the conventional spoked-arm design, resulting in a total rotor mass savings of 3.975 metric tons.

MASS AND STRUCTURAL DEFORMATIONS: A DESIGN COMPARISON

The ultimate goal of creating optimized rotor designs is to decrease the structural mass, thereby resulting in overall system cost savings due to lower engineering constraints for other wind turbine components such as the tower. Figure 21 compares the mass of each optimized design to conventional designs. A continual decrease in structural mass can be noted with each design iteration.

Additionally, the final U-beam-truss lattice was the only design with a lower structural mass than the hollow-truss design, with a 24% reduction. Therefore, it was identified as the optimal design from this study. The radial and torsional deflections for
these designs are given in Figure 22 and 23. The axial deformations were ignored as they were on the order of a few microns and the machine is not expected to reach critical axial deformation under given loading conditions. It should be noted that the radial deformations of the new designs are more than 60% lower than the conventional spoked-arm design. The torsional deformations of the U-beam-truss lattice and solid-truss lattice designs are under the critical limit calculated earlier from the maximum angle of twist of the rotor.

**FIGURE 22.** Radial deformation of 5-MW rotor designs

**FIGURE 23.** Torsional deformation of 5-MW rotor designs

to design for AM. These steps include mold optimization of minimizing unwanted volume as well as indirect tooling approaches to create a “mold of a mold” to achieve multiple casts with the same mold.

**Cost Evaluation of Additive-Manufacturing-Catered Geometries**

Using powder-binder printing technology, a new costing model must be developed that takes into account the substantially different manufacturing process. To accomplish this, we developed a basic costing scheme for a sand-cast-printed generator as shown in Eq. 11 and the constant bounds are given in Table 9. To make a comparison, the costs for conventional rotors made were obtained from a study by the Global Wind Network [18].

\[
Cost = V_{\text{mold}}C_1 + M_{\text{steel}}C_2 + SA_{\text{design}}C_3 + C_4
\]

where, \( M_{\text{steel}} \) = steel mass (mT)
\( C_1 \) = printed material cost ($/m^3)
\( C_2 \) = steel casting cost ($/mT)
\( SA_{\text{design}} \) = design surface area (m²)
\( C_3 \) = finishing cost per surface area ($/m^2)
\( C_4 \) = transportation cost ($)

**TABLE 9. COSTING MODEL PARAMETER BOUNDS**

<table>
<thead>
<tr>
<th>Cost Components</th>
<th>Unit</th>
<th>Low Bound</th>
<th>High Bound</th>
</tr>
</thead>
<tbody>
<tr>
<td>Printed material cost</td>
<td>$/m^3</td>
<td>8,000</td>
<td>14,000</td>
</tr>
<tr>
<td>Steel casting cost</td>
<td>$/mT</td>
<td>1,700</td>
<td>2,500</td>
</tr>
<tr>
<td>Finishing cost</td>
<td>$/m^2</td>
<td>100</td>
<td>500</td>
</tr>
<tr>
<td>Transportation cost</td>
<td>$</td>
<td>6,000</td>
<td>18,000</td>
</tr>
</tbody>
</table>

Costing schemes for powder-binder printing can be split into...
multiple cases, as represented in Fig. 24.

Case I: Pay for all sand/binder from manufacturer with no optimized mold
Case II: Pay for all sand/binder from manufacturer with an optimized mold
Case III: Amortized cost of optimized mold over 10 turbine rotors.

Results indicate that an optimized mold is necessary to achieve significant cost reductions in single-use molds. Mold optimization refers to removing all volume not needed for casting. However, only indirect tooling (Case III) results in additive rotor costs below that of conventional means. Indirect tooling refers to creating a mold of a mold to spread out the cost of using additive manufacturing to create a sand-cast mold. In this case, the additive manufactured mold is used to create a solid metallic mold that can be used to directly cast the rotor design. In this manner, the mold cost can be amortized over many rotors, resulting in economic viability and lower lead times.

**Costs for Printing Conventional Spoked-Arm Geometries**

In order to compare AM sand casting with traditional approaches, we must compare the cost of creating the hollow spoked-arm conventional design with the same design using AM. Mold optimization is key to the analysis. As shown in Figure 25, since there are few arms, it is unnecessary to create mold support in areas in between the arms. As a result, the volume of the mold is drastically reduced. Because of this reduced volume, the cost to produce the conventional hollow spoked-arm design through additive manufacturing greatly decreases. This cost model is dependent on assumptions of printing costs, labor costs, and limited knowledge of manufacturer costs and should be considered a first-order approximation of the rotor cost; however, the AM cost is significantly below the conventional cost shown in Figure 26.

**CONCLUSIONS**

Additive manufacturing offers unique potential in improving the design of conventional direct-drive generators by reducing structural weight with increased stiffness. By taking advantage of additive manufacturing’s ability to make complex parts as cost-effective as simple parts, manufacturers can focus better
on designing for functionality and performance. The structural and costing analyses in this study provide the following insights:

1. Design for additive manufacturing of generator rotors for a permanent-magnet direct-drive machine has the potential to reduce the structural mass by up to 24%.
2. Nonconventional lattice-based structures with U-beam truss members elicit high structural strength resulting in 60% less radial deformation with torsional deformation within critical limits.
3. The nonconventional lattice-based designs are expected to preserve their structural integrity under unbalanced magnetic pull arising from rotor misalignment, eccentricity, and deflections.
4. A first-order AM rotor costing model developed during this study suggests that the printed material volume is the greatest cost component. Using an optimized mold (removing unnecessary volume) has the potential to decrease sand-cast printing costs by 33%. However, printing single-use molds can cost up to 38% more than conventional methods.
5. Indirect tooling (printing a mold of a mold) greatly reduces the per-rotor cost well below that of conventional means. Cost estimates for indirect tooling suggest that the casting cost can be significantly reduced for an increased number of rotors cast from one mold.

To reduce the mold volume, we envisioned printing only a mold for the structural component of the lattice design and welding this to the rotor to allow for greater structural complexity with decreased cost. However, cost estimates need to be refined to include labor and assembly to weld the lattices structures to the rotor.

**FURTHER WORK**

Further analysis has been planned on using AM to:

1. Explore the structural design space for novel geometries and conduct more detailed analyses of the entire generator including the stator.
2. Perform experimental validation creating the U-beam lattice rotor using additive manufacturing and testing structural properties.
3. Refine the cost approximation of both conventional and additive manufacturing approaches.

**ACKNOWLEDGEMENTS**

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