Exploring Optimization Opportunities in Four-Point Suspension Wind Turbine Drivetrains Through Integrated Design Approaches

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Exploring Optimization Opportunities in Four-Point Suspension Wind Turbine Drivetrains Through Integrated Design Approaches

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Drivetrain design has significant influence on the costs of wind power generation. Current industry practices usually approach the drivetrain design with loads and system requirements defined by the turbine manufacturer. Several different manufacturers are contracted to supply individual components from the low-speed shaft to the generator – each receiving separate design specifications from the turbine manufacturer. Increasingly, more integrated approaches to turbine design have shown promise for blades and towers. Yet, integrated drivetrain design is a challenging task owing to the complex physical behavior of the important load-bearing components, namely the main bearings, gearbox, and the generator. In this paper we combine two of NREL's systems engineering design tools, DriveSE and GeneratorSE, to enable a comprehensive system-level drivetrain optimization for the IEA Wind reference turbine for land-based applications. We compare to a more traditional design with integrated approaches employing decoupled and coupled design optimization. It is demonstrated that both approaches have the potential to realize notable mass savings with opportunities to lower the costs of energy.The coupled approach, particularly is effective in realizing up to 15% weight reduction compared to the traditional design.

I. Nomenclature

Sh_{ratio}	=	Ratio of shaft inner diameter to outer diameter
r_s	=	Air -gap radius
l_s	=	Core length
h_s	=	Stator slot height
b_s	=	Stator slot width
$\hat{B}_g$	=	Peak air gap flux density
$\hat{B}_{sy}$	=	Peak stator yoke flux density
$\hat{B}_{ry}$	=	Peak rotor yoke flux density
$\hat{B}_{st}$	=	Peak stator tooth flux density
$\hat{B}_{rt}$	=	Peak rotor tooth flux density
E_p	=	Generator terminal voltage
T	=	Rated torque
J_s	=	Stator winding current density
J_r	=	Rotor winding current density
A_1	=	Specific current loading
λ	=	Aspect ratio
σ	=	Shear stress
p	=	Pole pairs
D_{out}/D	=	Outer to inner stator diameter ratio
η_{target}	=	Target drivetrain efficiency
F_x	=	Axial load

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F_y	=	Horizontal load
F_z	=	Vertical load
M_x	=	Torque
M_y	=	Overturning moment
M_z	=	Yaw moment

II. Introduction

The drivetrain is the "heart" of a wind turbine and constitutes a system of subcomponents, including main bearing, low-speed shaft, gearbox, generator, power electronics, bedplate, and mechanical couplings. These subcomponents are sourced from different suppliers or manufactured by the turbine original equipment manufacturers (OEMs). As innovations in wind turbine designs continue to emerge and mature, the cost of wind energy is dropping such that wind energy is more competitive with other electricity generation. For drivetrain and gearbox design, the main requirements leading to a low cost of energy are high reliability, high availability, low capital cost, ease of manufacture, ease of maintenance, and high efficiency. To achieve greater performance at lower costs, the wind industry is transitioning to system-level engineering with a focus on a variety of components, including the rotor, nacelle, hub, tower, main bearings, operation and maintenance, safety systems, loads, turbine control, pitch, and yaw systems. Yet, integrated drivetrain engineering and design is particularly challenging because, aside from their intended physics of operation, the participating components are performing structural, mechanical, or electrical functions (in the case of generator) at the same time [1]. The confluence of such physical behavior makes it difficult to simulate the actual loads properly at the design stage. This is especially critical in components as complex as a gearbox and generator, which are primarily designed to withstand torque moments [2]; the ability to scale such designs with power rating is not well established. In turn, the need for high efficiency in turn can influence the generator type, power electronics and variable ratio gearboxes [2] and therefore the overall cost of energy (COE). The electromagnetic design of the generator itself is related to the COE through several aspects, including losses, material costs, and operational costs [3, 4].

Optimization of wind generator systems focused on drivetrain systems was reported in early 2000. These studies were focused on direct-drive, permanent magnet generators, electrically excited synchronous generators, and traditional multistage geared drive induction generators [5–8], but the generator size was roughly estimated and scaling laws were used for different power designs. In 2006, the National Renewable Energy Laboratory (NREL) released [9] the wind turbine design Cost and Scaling Model that better reflected the turbine technology of that time as validated by empirical data. Some of the later studies [10–12] that compared different drivetrain designs continued to use empirical models for constituent components, while physics-based models for sizing the generator were developed and used in the optimization process. Gear ratio was used as an important design variable, yet the mass of the gearbox was empirically estimated and resulted in inaccurate estimates. In 2015, NREL released DriveSE [13] to size wind-turbine components from the hub system, drivetrain, and overall nacelle (including bedplate and yaw system) using system configuration parameters as well as the aerodynamic loads from the rotor. DriveSE employs analytical models for sizing the major load-bearing components (the low-speed shaft, main bearing[s], gearbox, and bedplate), yet the high-speed side of the drivetrain, including mechanical brake, generator, and other auxiliary components, were not modeled in greater detail. In 2017, NREL released, GeneratorSE [14], a set of analytical tools exclusively meant for sizing generators adaptable to direct-drive, medium-speed, and high-speed geared drivetrain architectures. GeneratorSE integrates electromagnetic, structural, and basic thermal models and provides a more accurate and optimal design by trading off active and inactive materials to satisfy certain fundamental and interdependent factors, such as weight, costs, or efficiency. The availability of these two design tools provides the opportunity to improve the component weight trade-offs and better assess sensitivities to material costs (especially to expensive metal such as copper/magnets that find extensive use in electric machines).

In this study, we couple NREL's two analytical design tools to enable a comprehensive drivetrain design and optimization for the IEA Wind reference turbine. We investigate two integrated design approaches for optimal drivetrain architectures; the first approach decouples the optimization of the generator from the rest of the elements of the drivetrain. The second approach performs a simultaneous optimization of the generator and the rest of the elements of the drivetrain. In both approaches minimizing the overall nacelle mass is the objective. For the drivetrain, the overall gear ratio and main shaft ratio are chosen as key design variables. For the generator, air-gap radius, core length, maximum slip, magnetic loading, and excitation are allowed to vary. For a given power rating, the high-speed shaft speed (determined by the gear ratio) is the fundamental element linking the drivetrain and the generator design. The optimization itself is carried out using a multi-start procedure because generator design is highly influenced by starting points chosen within its space. The lightest drivetrain designs resulting from both design approaches are identified and compared. The remainder of this paper is organized as follows: Section III describes the tools and briefly explains the optimization approaches examined, Section IV presents details of the reference turbine model and initial mass estimates using DriveSE, Section V and VI discuss results from the decoupled and coupled optimization approaches, and Section IV concludes the paper with closing remarks.

III. Approach: Optimization Arrangement with DriveSE and GeneratorSE

In this study, we specifically studied high-speed drive train architectures employing doubly fed induction generators. We chose to use turbine specifications of the IEA Wind reference turbine developed under IEA Wind Task 37 [15], representative of land-based application. The extreme aerodynamic rotor forces and bending moments at the hub, gravity loads, rotor overhang distance and tower-top diameter provided by Techniche Universität Münichen [16] were the main design inputs.

A. DriveSE

DriveSE consists of a series of interacting mathematical models of drivetrain subcomponents with analytical formulations for sizing low-speed shaft, main bearings, gearbox, bedplate, and yaw system. The transformer and high-speed side of the drivetrain, including the high speed shaft (HSS), generator, and generator coupling are sized using empirical scaling law models. Figure 1 illustrates a modified version of DriveSE to allow for the coupling with GeneratorSE to implement a four-point suspension drivetrain optimization, with which two bearings support the main shaft. Such an arrangement will enable a component level optimization. DriveSE accepts turbine loads and gearbox design inputs for main bearings, transformer, and gearbox location. For the given input loads, the main shaft and bearing are sized first by determining the length from deflection limitations imposed by main bearings, which are selected based on shaft geometry. Each component within DriveSE is designed based on a set of assumptions, design variables, and constraints for allowable stress (using industry-recommended safety factors), deflection (to ensure proper geometric alignment with bearing and gear-tooth meshing) and center of mass (CM) of nacelle to optimize each sub-component for the minimum weight. For instance, the gearbox design is optimized for the minimum weight by optimizing the speed ratio of each stage up to three stages with different combinations of planetary and parallel stages. The bedplate size is approximated as two parallel I-beams carrying weights and centre of mass of the drivetrain components. The yaw system is modelled with a friction plate bearing at the nacelle tower and several yaw motors. For more information on the DriveSE model formulation, refer to the DriveSE model report [13]. In this study, the existing empirical module of the generator within DriveSE is replaced with GeneratorSE where detailed analysis is carried out.

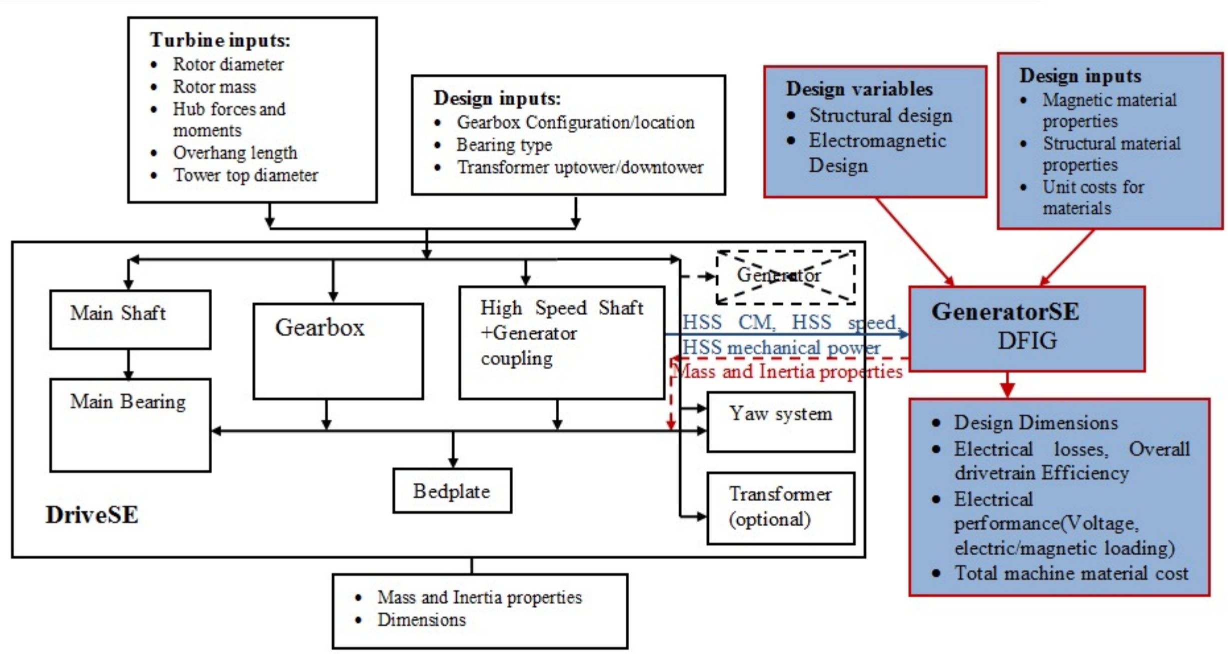

Figure 1. Layout for DriveSE and GeneratorSE (adapted from [17])

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B. GeneratorSE

GeneratorSE contains modules for sizing squirrel-cage induction generators (SCIG) and doubly fed induction generators (DFIG). The design dimensions of the machine have structural and electromagnetic components. Figure 2 shows the electromagnetic design dimensions. The tool accepts inputs for the center of mass of the HSS, shaft mechanical power available (after considering losses in the gearbox) and the HSS rotational speed from DriveSE. Shear stress, specific costs (i.e., unit costs per kilogram of material), and properties of materials (e.g., material density, magnetic field strength, resistivity) are used for basic design of the generator satisfying certain constraints and performance criteria (such as electric and magnetic loading). The magnetically required minimum generator dimensions, together with mass and inertia properties, are derived from initial design variables in compliance with the user-specified constraints on generator terminal voltage and constraints imposed on the dimensions and electromagnetic performance. For more detailed documentation of analytic models, refer to GeneratorSE [14].

The layout of tools described above allows for two approaches to optimization: 1) decoupled design optimization of the gearbox (and mechanical elements) and the generator, and 2) coupled optimization of the gearbox (and the other mechanical elements) and the generator. Decoupled optimization accepts design variables to size the gearbox and other mechanical elements, and the generator through a nested approach where the main components are sized with their own sub-optimization routines. This approach represents a more traditional design process that best reflects the current industry practice where gearbox and generator design is treated independently for a given set of turbine specifications. On the other hand, a coupled optimization uses a single global optimizer to minimize the total mass of both the generator and the gearbox (and the rest of the mechanical elements). The goal of this study was to quantify the benefit of creating systems-engineering architecture for a fully-coupled wind turbine drivetrain design. Figure 3 illustrates the optimization flowcharts for the two approaches. We anticipate that the decoupled optimization approach would result in sub-optimal mass whereas a coupled optimization approach would enable the optimizer to identify and exploit trade-offs between the design elements of all of the drivetrain components.

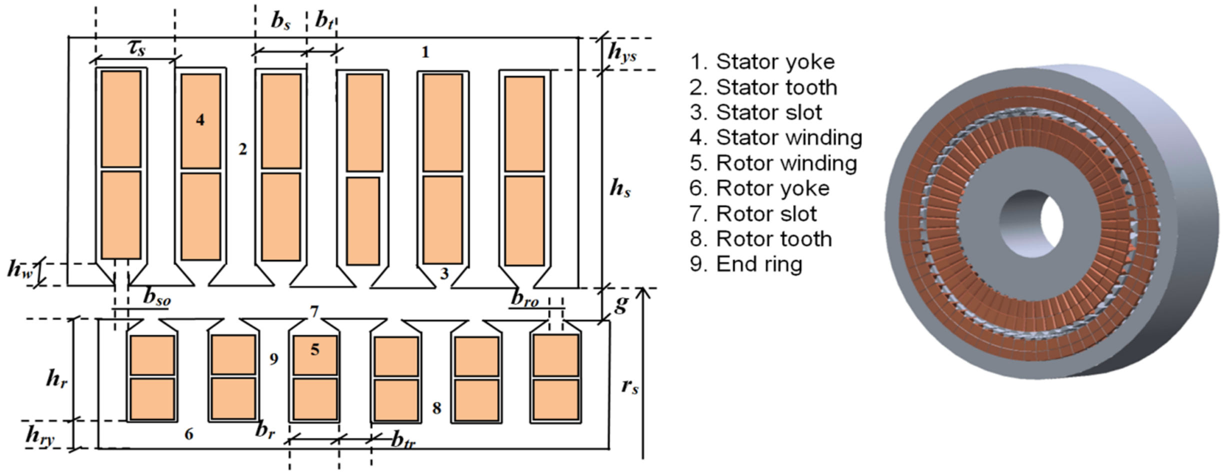

Figure 2. DFIG module within GeneratorSE; design dimensions (a) and CAD illustration (b). Illustration reproduced from [14].

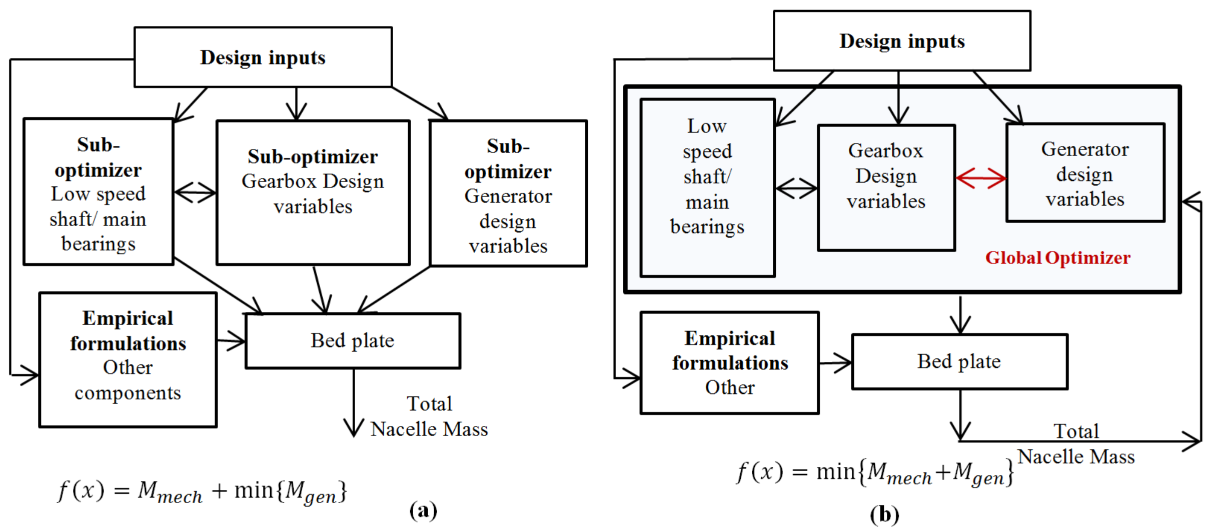

Figure 3. Integrated drivetrain design using: (a) Decoupled Optimization (b) Coupled Optimization

IV. The Reference Wind Turbine Model

In order to identify what positive impact the proposed design approach for drivetrain will have for the rest of the wind turbine system, we decided to modify a reference turbine system, its behavior has been well characterized and recently developed and co-ordinated under the IEA Wind Task 37 [16]. The reference wind turbine has an aerodynamic rating of 3.6 MW and is designed for onshore locations with low wind speed (IEC Class III). The turbine is a 3-bladed, upwind configuration characterized by a large rotor for a low specific rating/high capacity factor and a geared drivetrain. Table 1 presents the detailed specifications for the turbine together with details on the reference gear train and loads used in the design.

We chose to replace the single-stage gearbox drive of this reference turbine with a three-stage gearbox of epicyclic-epicyclic-parallel configuration. The design is consistent with the four point suspension described in DriveSE with two spherical roller bearings supporting the main shaft. The generator is a doubly fed induction machine with excitation power fed via slip rings. Table 2 specifies the key design variables used for the design optimization. The designed drivetrain is expected to have an overall efficiency of at least 93% and the generator terminal voltage was assumed not to exceed 5kV. Other performance criteria, specifically with regards to electromagnetic loading, are verified by constraints as listed in Table 4. Based on the rated rotor speed of the turbine and some of the commercially available designs for the doubly fed induction generator [18], the gear ratio was chosen to vary from 97-197. This allowed spanning a range of rated design speeds of 1140- 2315 rpm. One of the main benefits with a variable gear ratio is that it allows adjustments to the speed and torque input to the generator. OEMs tend to purchase gearboxes from suppliers who offer a wide range of reduction ratios, which provides more opportunities to select an accurate speed reduction or torque. Depending on the required gear reduction value and available space at the nacelle, different gear arrangements are possible. Specific gear arrangements allow high reduction values to be transmitted at a certain drivetrain weight. The shaft ratio that defines the ratio of the main shaft inner and outer diameters is also chosen as another design variable.

1. Baseline drivetrain designs

The baseline scenario typically represents a case in which the gear ratio is known and generator mass is empirically estimated; the existing design set-up in DriveSE was used as it is. This implied that only the gear sub-ratios were optimized within DriveSE for a given design specification. The key inputs for the design that include the rotor loads at the hub (defined using IEC coordinate system) are presented in Table 1. Figure 5 shows an incremental change in the gearbox mass with change in gear ratio. This reflected on the overall nacelle mass, which increased from 204 metric tons to 207 metric tons. More than 50% of this mass came from the gearbox and bedplate. The generator mass remained unchanged while the bedplate design had to accommodate the minimal increase in weight from the gearbox. This provided the initial rough estimate for the overall nacelle mass for the modified drivetrain to be close to 200 metric tons.

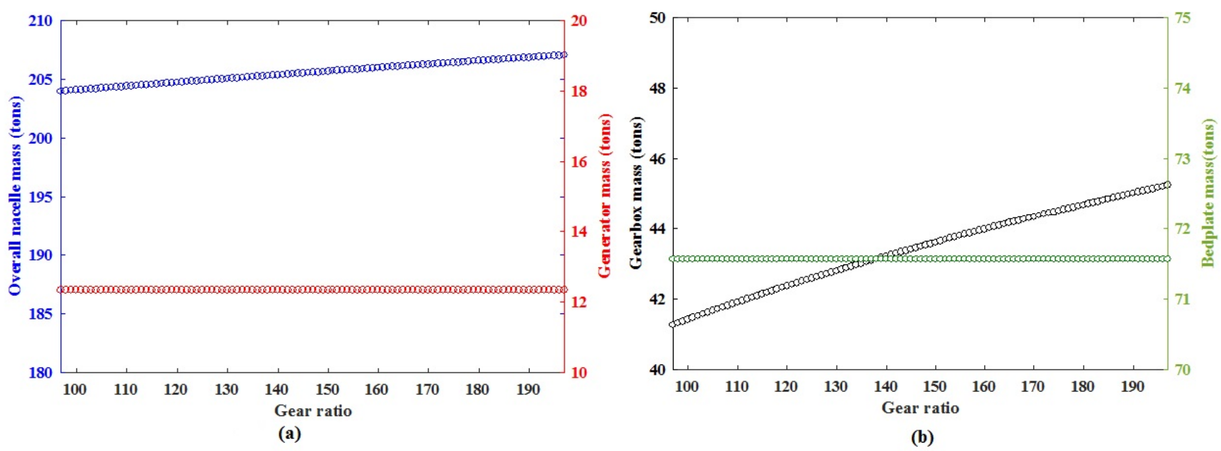

Figure 4. Mass estimates for the baseline models with different gear ratios

Table 1, IEA reference wind turbine specification

Specification	Value	Units
Rated Mechanical Power	3.6	MW
Class and Category	IEC Class 3A	
Rotor Rotation	Clockwise	
Rotor Orientation	Upwind	
Number of Blades	3	
Rotor Diameter	130	m
Hub Height	110	m
Rated Wind Speed	9.8	m/s
Rated Speed	11.753	rpm
Rated Torque	2.925	MNm
Max Tip Speed	80	m/s
Cut-In Wind Speed	3	m/s
Cut-Out Wind Speed	25	m/s

Specification	Value	Units
Control Variable speed	Collective-pitch	-
Drivetrain	Single-stage geared	-
Hub Mass	55	tons
Nacelle Mass	46.5	tons
Generator Mass	80.6	tons
Hub loads[1]		
M_x	3.83	MNm
M_y	11.1	MNm
M_z	11.7	MNm
F_x	1.1206×10^6	N
F_y	2.51×10^5	N
F_z	-1.03×10^6	N

[1] per IEC coordinates

Table 2, GeneratorSE main design variables and ranges.

Design parameters	Units	Lower Limit	Upper Limit
Air-gap radius	m	0.2	1
Core length	m	0.4	2.5
Stator slot height	mm	45	100
Rotor slot height	mm	45	100
Peak Stator yoke flux density	Tesla	1	2
No-load magnetization current	A	5	100
Maximum slip	-	-0.3	-0.1

Table 3, DriveSE main design variables and associated bounds.

Gearbox Configuration	3-stages
Gear ratio	97-198
Shaft ratio	0.1-0.4

Table 4, Main constraints driving the generator design

Design Constraints		
$0.7T < \hat{B}_g < 1.2T$	$A_1 < 60kA/m$	$500V < E_p < 5000V$
$\hat{B}_{ry} < 2$	$J_s < 6A/mm^2$	$4 <= h_s/b_s <= 10$
$\hat{B}_{st} < 2T$	$J_r < 6A/mm^2$	$0.2 <= \lambda <= 1.5$
$\hat{B}_{rt} < 2T$	$\eta_{target} > 93\%$	$2\pi r_s^2 l_s \sigma > T$

Pole count versus diameter ratio					
p	2	4	6	8	>=10
D_{out}/D	1.65-1.9	1.46-1.49	1.37-1.4	1.27-1.30	1.24-1.20

V. Decoupled Optimization

The decoupled optimization represents a scenario where the designs for the drivetrain and the generator are treated independently and the main element linking the two elements is the gear-ratio. The optimization is based on the premise that the lightest generator design will result in the lightest nacelle mass. The objective function that was optimised is given by;

$$Obj = M_{mech} + min(M_{gen}) \tag{1}$$

$$\begin{aligned} M_{mech} = & M_{LSS} + M_{main\,bearing\,1} + M_{main\,bearing\,2} + M_{Gearbox} + M_{HSS} \\ & + M_{mainframe} + M_{transformer} + M_{electrical} + M_{electronics} \\ & + M_{HVAC} + M_{Nacelle\,cover} \end{aligned} \tag{2}$$

where, LSS refers to the low speed shaft. For details on how each component masses were estimated, refer to [13, 14]. The main variables affecting the generator mass include air gap radius, core length, stator and rotor yoke heights, stator and rotor slot heights. While the entire range of possible gear ratios was passed to DriveSE, GeneratorSE required initial starting points for the designs for each gear ratio. It is emphasized that the generator design required at least seven variables to be initialized appropriately. Considering multi-dimensional searching with bounds for the design variables, the generator design space is expected to be expansive and several local optimal solutions are possible. The objective function was an estimate of the generator mass, as a proxy for generator cost for which an adequate model was not available, which is constrained by performance requirements of minimum efficiency and maximum magnetic loading. This helped narrow the design space of feasible solutions. For the optimizer, we chose to use Constrained Optimization By Linear Approximation (COBYLA) that uses a trust–region algorithm with linear approximations to the objective formed by interpolation at n + 1 points in the space of the variables [19]. In addition, a multi-start approach was used for choosing the starting points for the design variables and to address the presence of local minima in the design space [20].

1. Impact on generator design

We first tested generator optimization for selected gear-ratios of 97 and 147. For each case, at least 100 feasible solutions were identified and examined as to how the designs performed. Figure 5 shows the plots for the optimized generator masses compared against the main machine dimensions (i.e , the air gap radius and core length). For a rated speed of 1,140 rpm, the most feasible regions are identified to be bounded by 0.39m < r_s < 0.8m and 0.4m < l_s <1.5m. The optimized masses varied between 13.5 tons to 45 tons. Within these feasible regions, the lightest designs (i.e , the designs that weighed less than 20 tons, highlighted in orange) were found to have core lengths smaller than 1.2 m and radii less than 0.72 m. A close look at the histograms of the main dimensions of designs (Figures 6a-b) reveals the trade-offs within the generator design itself that lead to an overall optimized design. The best performing generator designs had a mass close to 13.95 tons and these designs had air gap radii in the range (0.48-0.5 m) and core lengths in the range (0.75-0.8 m). The impact of these lightweight designs on the bedplate and overall nacelle can be seen in Figures 6d and 6e. On increasing the gear ratio to 147, there was a tendency for the machines to become smaller in size, and minimum mass of the generator reduced from 13.95 tons, to 11.16 tons. This was intuitive because at higher gear ratio (higher speed of operation) a lower torque is required to overcome the tangential stress. This pushed the design envelope to smaller air-gap diameters but at similar core lengths (Figures 5c and 5d).

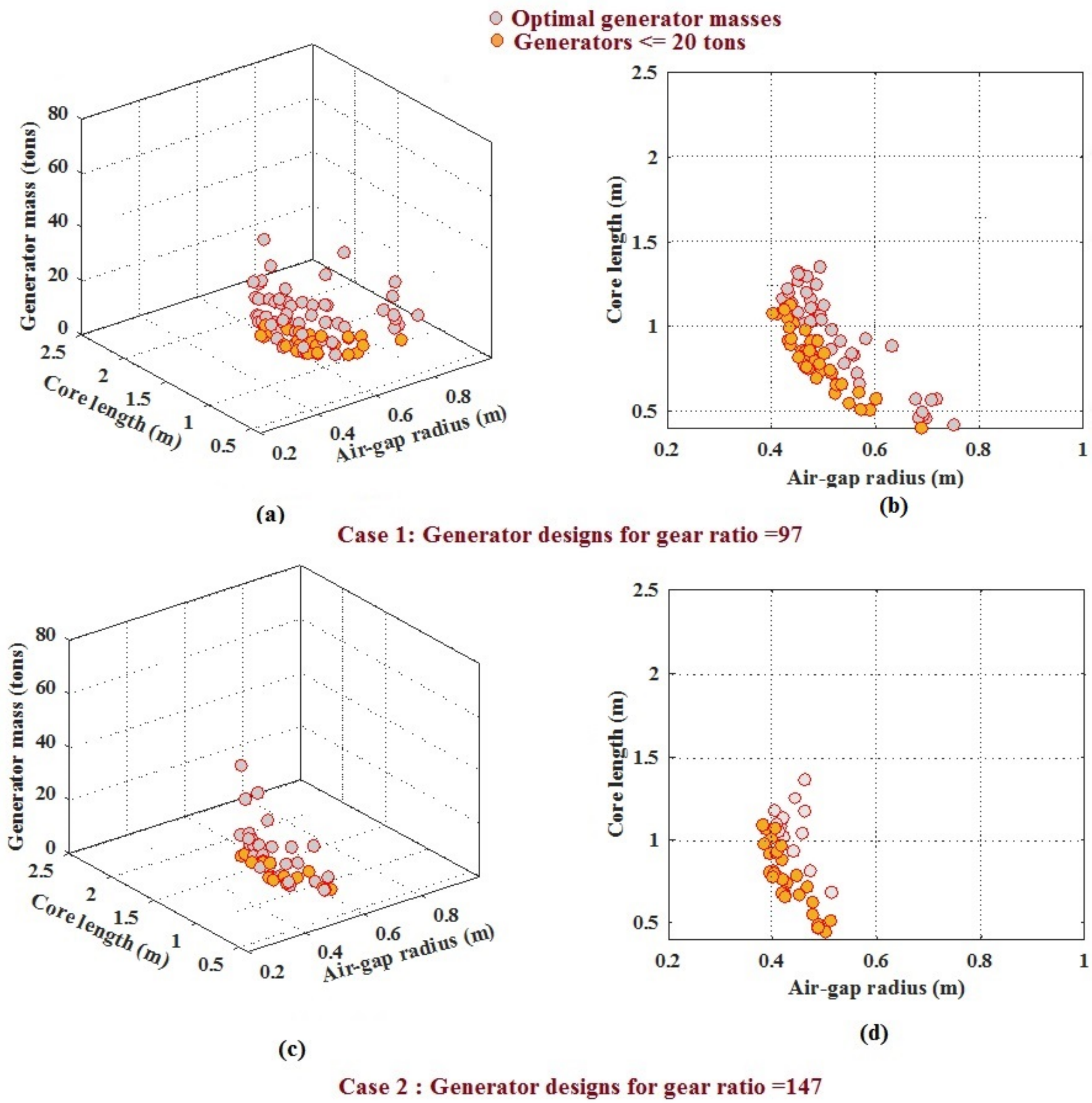

Figure 5. Solutions within feasible region using multi-start approach for gear ratios 97 and 147.

2. Impact on drivetrain/nacelle mass

The multi-start approach was repeated for the remaining gear ratios from 97 to 197 in increments of 5 for each case and the overall nacelle mass was determined for the lightest generator design identified from at least 100 feasible solutions. Figure 7 presents the results for the optimized mass of the main elements of the drivetrain.

The lightest generator weighing 10.47 tons was realized for the highest reduction ratio of 197 (Figure 7a). The corresponding nacelle/drivetrain design weighed 183 tons. This machine had a smaller air-gap radius (0.4m). Notice that as the rated speed increases, air gap radii and lengths mirror each other to achieve the optimal generator mass (Figure 8). Since the electromagnetic torque is a function of air gap radius and the core length (T $\propto r_s^2 l_s$) an increase in radius is complemented by a reduction in core length or vice-versa . Therefore, it is not necessary that the machines be smaller in diameter as torque reduces with the gear ratio. DriveSE optimized the internal stage ratios for the gearbox while determining the lightest possible design to meet transmitted power and strength properties requirements. The trend showed a linear increase in gearbox mass. Starting at 59.35 tons for a gear ratio of 97, the bedplate mass decreased with increase in gear ratio and stabilized at around 58 tons for gear ratios from 152-197. This result corresponded well

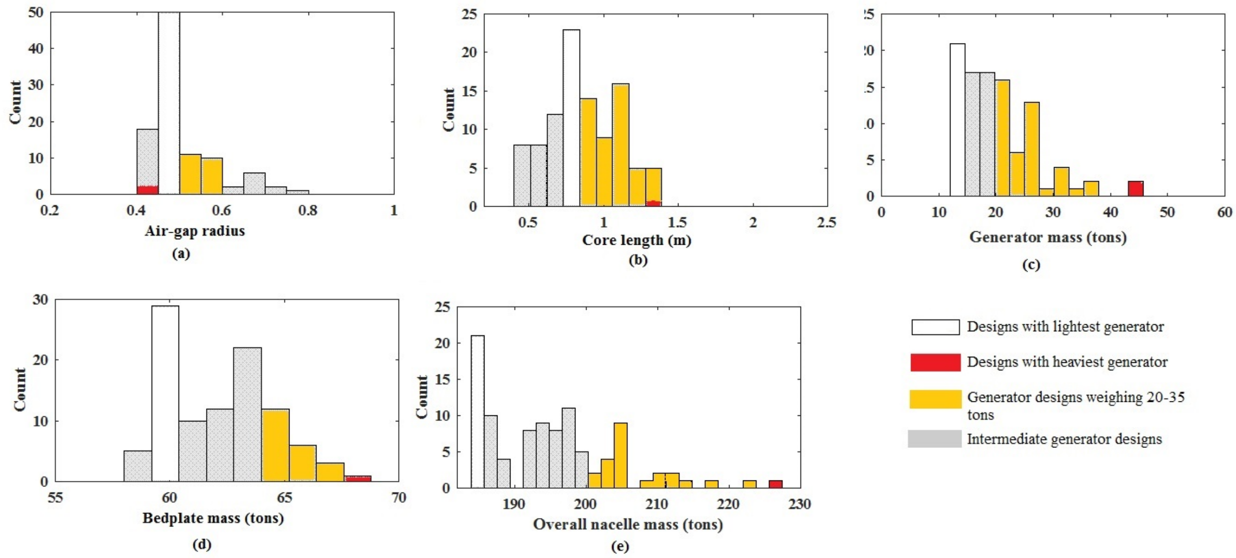

Figure 6. Histograms for (a) air-gap radius, (b) core length and (c) generator mass (d) bedplate mass, and (e) overall nacelle mass

with the generator mass reduction observed for that range suggesting that generator mass reduction was partly helpful. It is also noted that the lightest generator design did not result in the lightest nacelle design. A gear ratio of 152 showed a marginal improvement in the nacelle mass (182.5 tons) with a slightly heavier generator (at 11.26 tons). For the decoupled approach,the nacelle mass corresponding to the lightest generator design was deemed to be most optimal.

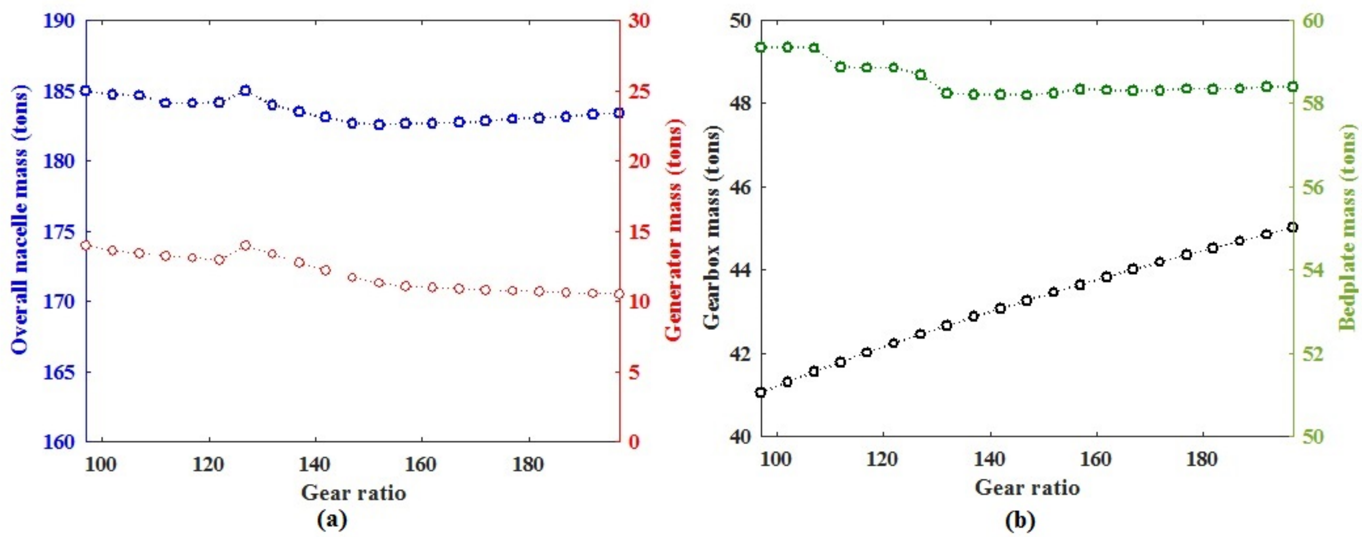

Figure 7. Optimized masses using decoupled approach. (a) nacelle and generator mass, (b) gearbox and bedplate mass

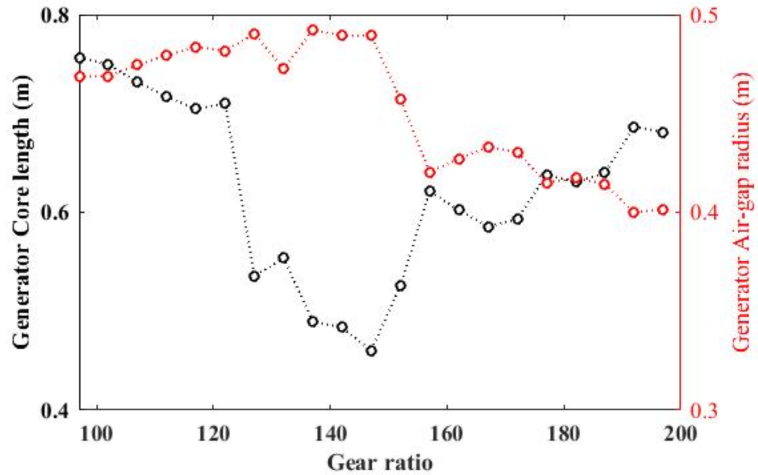

Figure 8. Trends in air-gap and core lengths for optimized generator designs at different gear ratio

3. Variable shaft ratio

Yet another optimization opportunity that was explored was a variable shaft ratio (Sh_{ratio}) for the low-speed shaft (LSS). This quantifies the ratio of inner diameter to the outer shaft diameter that is designed to meet the deflections and rigidity criteria. This parameter allowed optimization of the main shaft and helped to realize further reductions in the overall nacelle mass. The generator mass remained consistent with that obtained by varying the gear ratio. The lightest nacelle, weighing 179 tons for a gear ratio of 152 (Figure 9a), was lighter than the lightest design obtained from the decoupled approach by at least 4 tons (the corresponding generator mass was 11.26 tons). This was mainly due to reduction in the main shaft mass (Figure 10). Table 5 lists the optimized parameters for the generator and drivetrain obtained using a fixed and variable shaft ratio. It should be noted that only the main components expected to be affected by the optimization are listed. Masses for the HVAC (Heating ,ventilation and air conditioning) system, high-speed shaft, brakes, and yaw system remain unaffected at 0.87 tons, 0.28 tons, and 4.46 tons respectively. Nacelle cover mass is reported here, as it reflects any change in dimensions of the constituent elements of the drivetrain. There is a marginal improvement noted in drivetrain efficiency because of a more efficient generator.

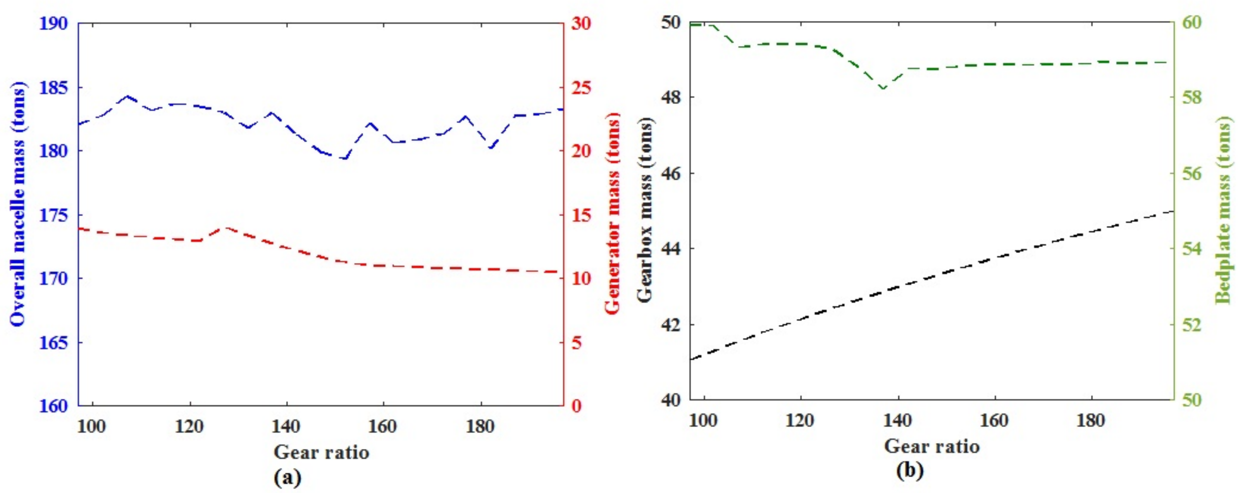

Figure 9. Optimized masses with a variable shaft ratio (a) nacelle and generator mass and (b) gearbox and bedplate mass

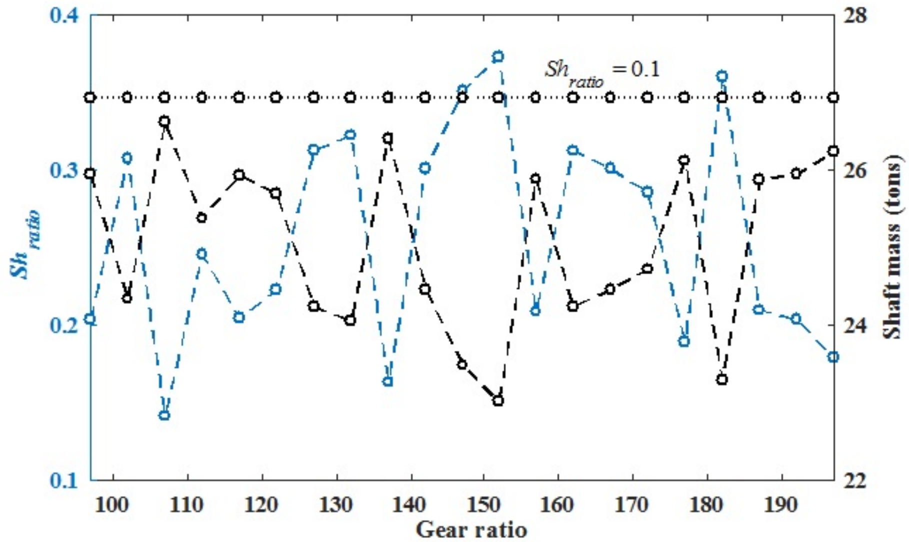

Figure 10. Variation of main-shaft mass with Gear ratio

VI. Coupled Optimization Approach

It is anticipated that simultaneous optimization of the generator design and the mechanical elements of the drivetrain would result in more optimal results. For this purpose, a new top-level assembly was created within DriveSE. The assembly included drivetrain and generator as two components, and the overall nacelle mass was assigned as the new objective function that was to be minimized.

$$Obj = min(M_{mech} + M_{gen}) \tag{3}$$

where, M_{mech} is given by equation 2. The main variables affecting the generator mass include air gap radius, core length, stator and rotor yoke heights, stator and rotor slot heights. Notice, that since this is a simultaneous optimization problem, the generator length and mass are key variables that can alter the bedplate sizing and mass. The generator design variables were retained similar to the decoupled optimization case, together with the same performance and design constraints. The gear ratio was allowed to vary from 97-197. This setup is expected to minimize the overall nacelle mass by simultaneously optimizing the mass of the generator and the bedplate. Similar to the decoupled approach, we chose to use COBYLA for optimizer [19], and multi-start runs by facilitating optimal generator search under the same umbrella for the optimal drivetrain search.

1. Optimized generator designs

For the coupled optimization, the generator designs were picked from nacelle designs weighing lighter than 200 tons. Figure 11 shows the plots for the generator mass compared to the generator dimensions optimized for rated speeds of 1,140 rpm and 1,728 rpm, respectively. The most feasible regions for generator designs corresponding to nacelle mass < 200 tons are highlighted in orange. For a gear ratio of 97 the optimal regions appeared to be widespread and bounded by 0.38 m < r_s <0.7m and 0.45 m < l_s < 1.75 m. The lightest nacelle was 178 tons with a generator design weighing 18.9 tons. This generator was at least 5 tons heavier than the lightest generator design found using a decoupled approach. A gear ratio of 147 resulted in a smaller envelope of feasible regions. The designs corresponding to lighter nacelles had a smaller range for air-gap radius (0.38 m-0.52 m), but the core lengths were marginally smaller (0.7m -1.25 m). Figure 12 shows the histograms for the nacelle and generator masses for a gear ratio of 97. Although several generator designs weighed less than 12 tons, these lightweight generator designs did not result in lighter nacelles. This is mainly due to the trade-off with masses of the rest of the components in the drivetrain. A close look at the histograms revealed five significant regions. The lightest generator designs (< 12 tons) tend to be associated with a slightly heavier gearbox designs (43 tons) and bedplate designs weighing 58 tons. There were also several generator designs between

Table 5, Optimized Generator and drivetrain properties obtained using decoupled approach.

Description	Sh_{ratio} =0.1	Sh_{ratio} = (0.1-0.4)	Units
Machine Rating	3.6	3.6	MW
Air-gap diameter	0.803	0.91	m
Core length	0.68	0.52	m
Pole pitch	631	718	mm
Stator slot height	100	100	mm
Rotor slot height	84	100	mm
Stator yoke height	80	105	mm
Rotor yoke height	80	105	mm
Peak air gap flux density	0.79	0.86	Tesla
Peak stator yoke flux density	1.99	1.86	Tesla
Pole pairs	2	2	-
Optimal maximum Slip	-0.3	-0.3	-
Magnetization current	38.65	41.8	A
Generator Efficiency	98.15	98.36	%
Total Mass	10.47	11.25	tons

Optimal gear ratio	197	152	

Component masses	Sh_{ratio} =0.1	Sh_{ratio} = (0.1-0.4)	Units
LSS[1] Mass	26.9	24.2	tons
Main Bearing1-SRB[2]	4.07	4.11	tons
Main Bearing2-SRB	4.07	4.11	tons
Gearbox Mass	45	43.1	tons
Overall Mainframe Mass	68.6	69.1	tons
Bedplate Mass	58.4	58.8	tons
Nacelle cover	8.07	7.9	tons
Yaw system	4.46	4.46	tons
Overall Nacelle Mass	183.3	179.3	tons
Drivetrain efficiency	93.73	93.9	%

[1] Low-speed shaft
[2] Spherical roller Bearing

12-20 tons and designs up to 30 tons that were associated with heavier bedplate designs up to 63 tons. The heaviest nacelle was also associated with the heaviest generator weighing 36 tons.

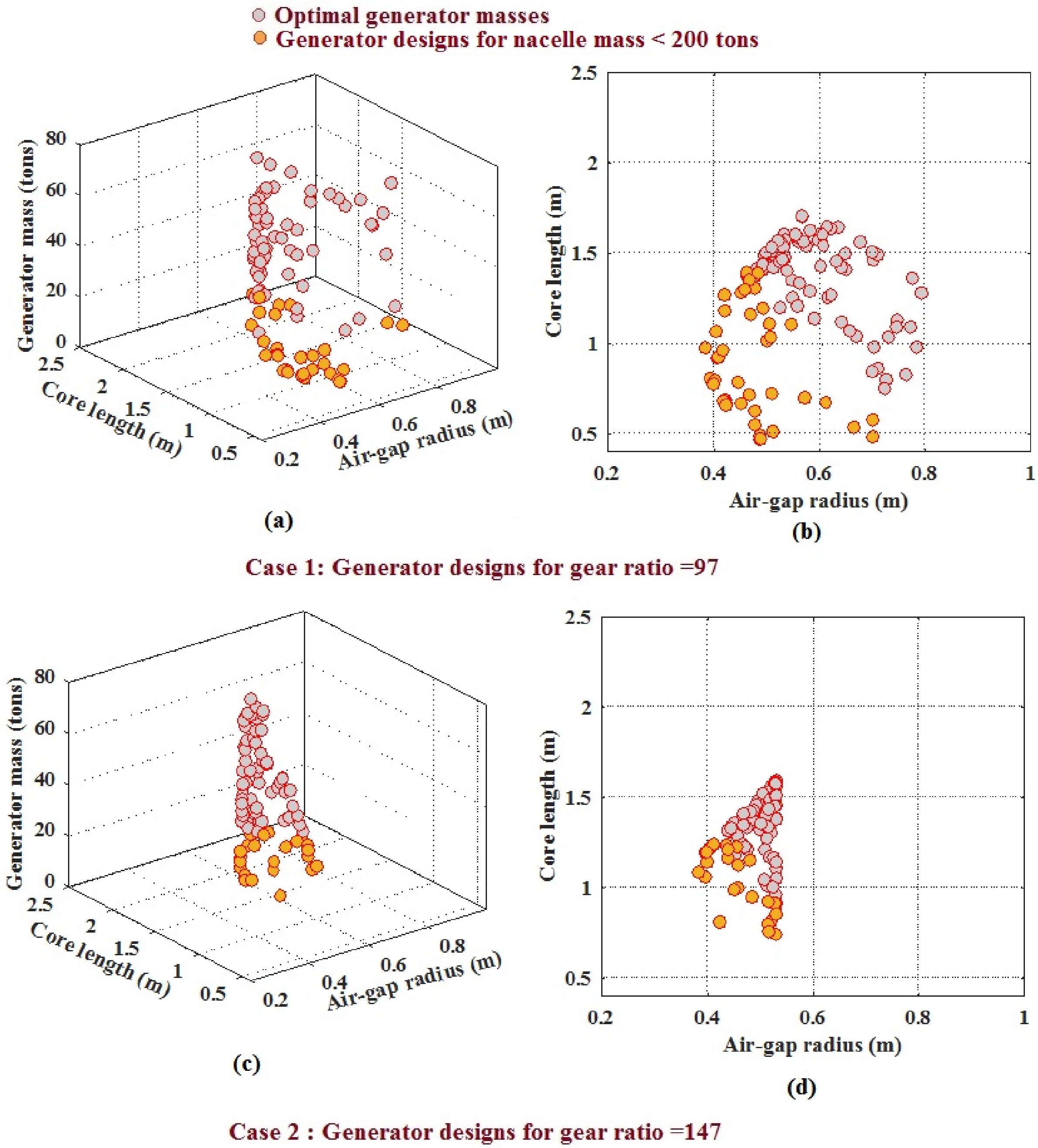

Figure 11. Solutions within feasible region using multi-start approach.

2. Optimized drivetrain/nacelle design

The multi-start procedure was repeated for the remaining gear ratios, and the optimal nacelle designs were identified from at least 100 feasible solutions. Figure 12 shows the results of optimized drivetrain designs for each gear ratio. The most optimal nacelle and drivetrain design was realized for a gear ratio of 197 with a generator that was slightly heavier than the design obtained from the decoupled approach (the generator was 0.7 tons heavier). The nacelle weighing 172.8 tons was lighter than the lightest design from the decoupled approach by at least 11 tons. This was realized despite having a slightly heavier generator. It should be noted that this machine had a larger air-gap diameter but a smaller core length. The trade-off appears with the simultaneous optimization of rest of the elements in the drivetrain, most notably,

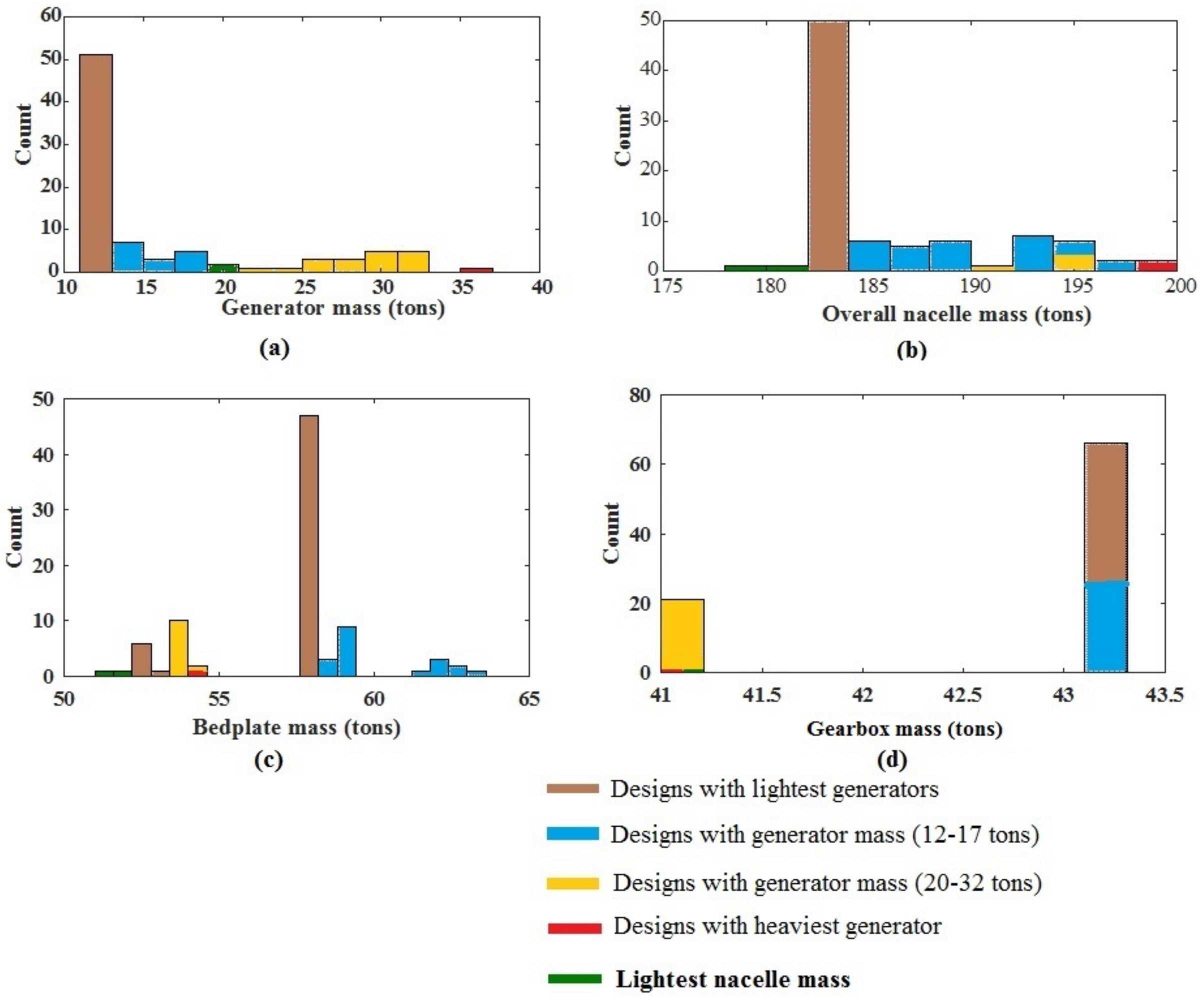

Figure 12. Histograms for optimal generator and nacelle masses

the bedplate mass that dropped from 58 tons to 50 tons. Other reductions were observed in main shaft mass (by atleast 1 ton) and nacelle cover mass (by 2.7 tons). Note the scatter in the plots for the nacelle mass that can also be traced in the estimates for generator mass. Nevertheless, for the first iteration with 100 feasible solutions, these results were useful to ascertain potential improvements with the model. Table 6 lists the optimized parameters for the generator and drivetrain designed for a rated speed of 2,315 rpm (gear ratio 197). It was noted that the drivetrain design also had a lighter main shaft mass as compared the optimized drivetrain from decoupled approach.

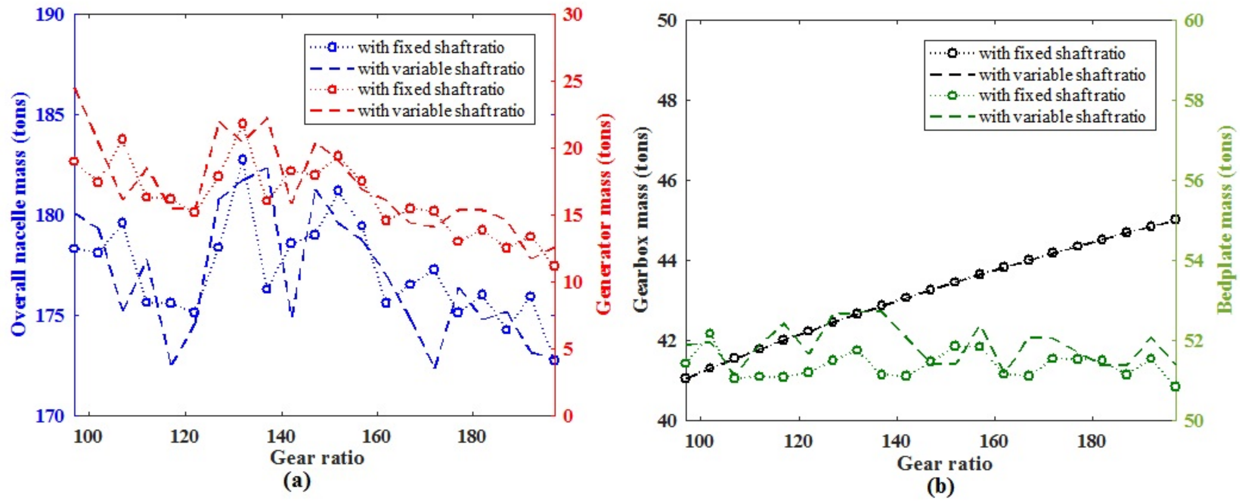

Figure 13. Optimized masses using coupled approach.

Table 6, Optimized Generator and drivetrain properties.

Description	Values	Units
Machine rating	3.6	[MW]
Air-gap diameter	0.93	m
Core length	0.5	m
Pole pitch	734	mm
Stator, rotor slot height	100, 92.6	mm
Stator, rotor yoke height	110, 110	mm
Peak air gap flux density	0.88	Tesla
Peak stator yoke flux density	1.87	Tesla
Pole pairs	2	
Optimal maximum slip	-0.3	-
Magnetization current	42.8	A
Generator Eeficiency	98.18	%
Total mass	11.18	tons

Optimal gear ratio	197	
Component Masses	Values	Units
LSS Mass	25.7	tons
Main Bearing1-SRB	4.07	tons
Main Bearing2-SRB	4.07	tons
Gearbox Mass	42.4	tons
Overall Mainframe Mass	60.2	tons
Bedplate Mass	50.8	tons
Nacelle Cover	5.26	tons
Overall Nacelle Mass	172.8	tons
Drivetrain efficiency	93.7	%

3. Variable Shaft ratio

A shaft ratio close to 0.4 can help realize a lightweight drivetrain design optimized for a gear ratio of 172. This design weighed 172.4 tons with the main shaft mass at least 5 tons lighter than that obtained with a fixed shaft ratio (Figure 13). The plots with variable shaft ratio do not show a consistent reduction in the masses suggesting that the found minima could be less than optimal. This is notably evident in the bedplate masses which were slightly heavier. If more feasible solutions were chosen, the global minima for the nacelle mass were expected to drop further. Table 7 summarizes the results from all of the integrated design optimization approaches.

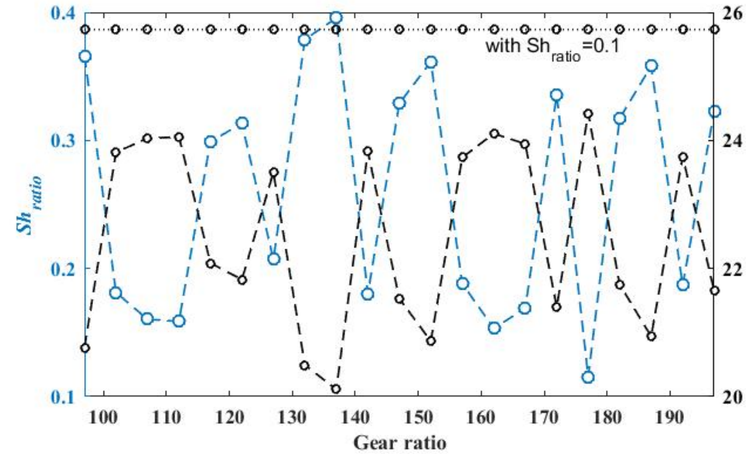

Figure 14. Shaft masses obtained using coupled approach.

Table 7, Summary of baseline and optimized masses using different approaches

Parameter	Units	Baseline	Decoupled	Decoupled	Coupled	Coupled
Gear ratio	-	97-197	97-197	97-197	97-197	97-197
Sh_{ratio}	-	0.1	0.1	0.1-0.4	0.1	0.1-0.4
Optimized properties						
Gear ratio	-	97	197	152	197	172
Generator mass	tons	**12.35**	**10.47**	**11.25**	**11.18**	**14.11**
Gearbox mass	tons	41.35	42.45	42.6	42.435	42.86
Bedplate mass	tons	71.6	58.21	58.7	50.56	50.84
Overall nacelle mass	tons	**204**	**183.3**	**179.8**	**172.8**	**172.4**
Main shaft mass	tons	25.7	25.7	24.2	25.7	20.2

VII. Conclusion

Design of wind turbine drivetrains is often done for a known set of specifications and loads for any given power rating. Usually generator design (in terms of rated speed and rated terminal voltage) is known ahead, and the optimal gear ratio is selected or vice-versa. Even better results can be achieved if both the gearbox design and generator design are integrated because many design parameters can be chosen to be variables. In this study we examined atleast two approaches to integrate NREL's two design tools, namely DriveSE and GeneratorSE, to investigate optimal drivetrain configuration for the IEA 3.6MW reference wind turbine. A baseline model in DriveSE was used to make initial estimates for a high-speed doubly fed induction generator drivetrain. The reference model was limited by empirical estimation of generator mass, suggesting that optimal gear ratio must be at the lowest to realize the lightest drivetrain design. In order to realize improvements in the design, we first coupled the DFIG module from GeneratorSE with DriveSE and carried out sub-optimization of the generator design. The main design variable linking the two models was the gear ratio. Because generator design is highly influenced by the starting point of the search, a multi-start procedure was performed to check for multiple minima. The optimal generator designs were found to have core lengths smaller than 1.2 m and air-gap radius smaller than 0.7m. Increase in gear ratio provided smaller torques, which made the design envelope smaller with either a smaller gap diameter or a smaller core length.

The decoupled approach was based on the premise that the lightest generator design will result in the lightest nacelle mass. The lightest generator was realized at the highest gear ratio of 197, and the optimal drivetrain/nacelle design weighed 183 tons. The design was at least 22 tons lighter than the lightest baseline design suggested by DriveSE and comparable to the total nacelle mass of the IEA reference drivetrain system. This was attributed to the optimization of the bedplate design that accommodated the new lighter generator. In the second approach we carried out a coupled optimization of the generator design and the mechanical elements of the drivetrain. The coupling demonstrated that it was possible to achieve lower nacelle designs despite having heavier generators. This was largely attributed to the trade-off in other component masses in the drivetrain, especially to the bedplate. The coupling also confirmed the presence of lightest nacelle/drive design at the highest gear ratio. The overall nacelle mass was found to be at least 11 tons lighter than that obtained using the decoupled approach. Additionally, flexibility in shaft geometry via shaft ratio may provide opportunities to realize further weight reductions. Further work will be pursued to validate the coupled approach using more feasible solutions and more detailed investigation of the optimization of the drivetrain subcomponents. Furthermore, this work will be leveraged to investigate more significant system trade-offs related to overall turbine design where the design of the rotor, drivetrain and tower are fully integrated. Through fully-coupled investigation of wind turbine design, significant potential exists for improving wind energy production while lowering system costs.

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