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Journal of Thermal Engineering
https://jten.yildiz.edu.tr/
https://doi.org/10.47481/jten.0052

RESEARCH ARTICLE

Identification and mitigation of resonance factors in heavy-duty gearboxes for marine offshore wind turbines

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Abstract

This experimental study aims to determine the resonant frequency in planetary and high-frequency gearboxes operating in offshore environments at 1150 rpm (corresponding to 740 Hz). This research aims to investigate resonance throughout the gearbox using an FFT test bench and Ansys software. The analysis methods include examining the entire gearbox and housing in ANSYS Workbench. Key findings from three load cases are as follows: in load case three, the gearbox housing exhibited resonance at the fourth mode, accompanied by significant noise, including whistling, reaching up to 100 decibels. To address this issue, a stiffener should be added to the gearbox housing to mitigate resonance. The practical value of this research lies in identifying the gearbox's resonance modes to prevent excessive vibrations that can lead to premature component wear. Recognizing resonance is essential for developing transmission systems that can withstand the potential for failure and extend their operating lifespan. By detecting and addressing these resonances, we can reduce disturbances, protect the gearbox, and allow seabirds and other wildlife to thrive.

Keywords: Wind turbine, gearbox design, resonance, calculations, vibration analysis

Cite this article as: Kumar, K. S., Senbagamalar, J., Sudha, S., & Jeha, J. (2026). Identification and mitigation of resonance factors in heavy-duty gearboxes for marine offshore wind turbines. Journal of Thermal Engineering, 12(5), 1611–1624. https://doi.org/10.47481/jten.0052

1. Introduction

Power generation in a wind turbine power plant depends on the rotational speed of the gearbox. Therefore, it is essential to manufacture a properly rated gearbox that can operate efficiently under both fluctuating and normal loads to ensure effective power generation [1,2]. Identifying potential faults is easier when separating high-repetition components from low-repetition ones than when analyzing the initial signals. To characterize the problematic frequencies of rotating gears and roller bearings, various homogenized phases, analyzed by spectroscopy, are introduced. The effectiveness of the proposed methods is demonstrated through tests on a synthetically deficient transmission and in a real-world scenario using an 800 kW wind power plant transmission system [3]. This results from poor design and analysis of the gearbox before it is manufactured and put into operation. [4-6]. It was observed that most wind power plants were negatively impacted by improper maintenance of the gearbox, primarily due to inadequate oil supply during operation at higher RPM [7]. The failure rate of gearboxes due to resonance has surged alarmingly to an average of 33% over the past two years. This significant increase underscores the urgent need for preventive measures and robust solutions to tackle this pressing issue.[8]. The issues at hand stem from the use of inferior materials that compromise endurance limits, the inadequate mechanical strength calculations, and the lack of a thorough vibration (modal) analysis prior to the gearbox's operational deployment. Addressing these concerns is crucial to ensuring reliability and performance. [9,10]. Implementing new algorithms to predict common faults using advanced numerical simulation codes can significantly reduce breakdown maintenance, as noted in

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Submitted: 26 March 2025; **Accepted:** 24 July 2025

This paper was recommended for publication in revised form by Editor-in-Chief Ahmet Selim Dalkılıç

sources [11, 12]. This approach extends the lifespan of the wind turbine generator (WTG) gearbox and its associated components. As a result, breakdown maintenance could be reduced by 10% compared to traditional technology systems focused on fault detection and diagnosis. From the studies [13], special techniques were employed to determine the resonance frequencies of the first-stage planetary gearbox using demodulator technology (DT) and spectrum technology (ST) to achieve the highest intensity levels. Additionally, the DPORE strategy was used to identify vibration levels in the gearbox, with intensities ranging from 50 to 90 decibels. DPORE, which stands for Dynamic Position Offshore Renewable Energy devices, is a key component in this analysis. A sophisticated mechanism employed in wind power plants to improve their performance and reliability. This technology combines adaptive navigational systems with alternative power sources, enabling offshore wind turbines to maintain their orientation and position at sea, even in hazardous conditions. The primary advantages of DPORE innovation for offshore wind turbines encompass improved equilibrium: innovative placement technologies enable offshore wind turbines to maintain stability and functionality in adverse marine conditions. Enhanced Economy: The procedure adjusts turbine positioning to maximize wind-energy extraction, thereby enhancing overall effectiveness. There are various types and sizes of maritime windmills, making them a flexible option for diverse applications. This flexibility helps reduce the substantial costs associated with gearbox breakdowns. Advanced neural network systems have been implemented at the gearbox stage to predict natural and harmonic frequencies at very high speeds ranging from 150 RPM to 800 RPM [14]. This technology allows identification of significant vibrations, thereby eliminating potential resonance issues. Additionally, a unique filter-based technology known as Vold-Kalman filters effectively manages high nonlinear signals caused by severe vibrations in the range of 500 Hz to 900 Hz [15]. Using this technology, predictions can be made, and these vibrations can be mitigated through the application of Kalman filters. As a result, this approach achieves 95% accuracy, significantly reducing breakdown costs and minimizing the risk of resonance [16]. The natural frequencies are compared with the spectral results. The recent studies from [17] imply the usage of recorded data from SCADA gives the oil sampling techniques to be replaced every six months, owing to the first stage of the planetary gearbox and associated parts. Higher torque causes severe oil fluctuations that vibrate the gearbox's main shaft and the bearings associated with it.

Determining the lower viscosity of the oil from the inlet shaft to the final high-speed shaft depends on data reliability and associated features governed by SCADA techniques. These techniques will increase the need for high-viscosity oil to achieve a longer lifetime [18]. The aerodynamic analysis of wind turbine blades, the gearbox, and all the rotating parts of the wind turbine are essentially conducted as part of research and development activities and is supported by well-established laboratories and the latest dynamic software, such as Ansys and Hypermesh. These techniques yield optimal results for identifying the preliminary stress involved and the secondary stresses engaged in them. The theoretical resonance calculations are done with the help of MBB software to develop the identification of fatigues and cracks [19]. From the literature [20], it is identified that the implementation of TEO, SSR, and MNSR gives an idea of the root cause of natural frequencies obtained during FFT analysis. From this analysis, it is evident that HSS exhibits higher frequencies because it is coupled with a generator, which causes the gearbox housing to vibrate more than in other stages. Hence, resonance associated with high-pitched whistling sounds causes major environmental problems and adverse effects on humans and the surrounding environment. Hence, it is required to design and analyze the gearbox with rated power output at lower, average, and higher velocities, which is suitable for operating at all operating conditions [21]. The researchers proved that Adding the stiffener has potential benefits in terms of suppressing the resonance and reducing the decibel sounds [22]. The impact of fatigue cracks that may develop in the housing during long-term operation on the resonance frequency Because Cracking can amplify vibrational magnitude, particularly at wavelengths, resulting in enhanced resonating consequences proved by [23]. The long-term environmental monitoring data demonstrates the benefits of reducing disturbances to aquatic organisms, notably those caused by wind-generating gear. Human-caused offshore vibration, including generator activity, has been shown in research to have significant detrimental effects on marine animals, including hearing impairment, stress symptoms, and shifts in behavior that can disrupt food intake, reproduction, and interactions [24,25]. Investigators employ life-cycle analyses and characterization variables to measure the effects of reduced noise levels on marine ecology. These techniques measure the harmful impacts of background noise by examining variations in aquatic animal behaviors and reproductive movements. For instance, research on whales and dolphins in the Gulf of Norway utilized LCA to assess the influence of background noise from offshore turbine-farm development on avoidance behaviors and ecological destruction. Furthermore, inexpensive, real-time detectors have been created to continuously monitor under-vibrations and their effects on aquatic life. These technologies supply vital information that can be utilized to set acoustic boundaries and launch successful reduction techniques [26]

The present study identifies resonance factors for gearboxes operating in offshore applications. The problem identified is that resonance, manifesting as high-pitched whistling at an operating speed of 1150 rpm, poses serious risks to birds and marine organisms. A high noise intensity of 100 decibels disturbed marine organisms in the ocean. The proposed approach for identifying the source of resonance and eliminating the resonant factor was developed by analyzing the gearbox using a three-division analysis.

- Entire gearbox
- Gearbox without the planetary stage
- Gearbox housing

These three iterative analyses using the first ten extraction modes, were conducted in ANSYS Workbench to identify and match the gearbox resonances to those in the test bench report, which helps eliminate resonance by increasing stiffness. A detailed report from the test bench and FFT analyzer observed that huge noises in 100 decibels occurred at the high-speed shaft (HSS), Which was mounted on the gearbox housing. The transmission of forces from 15 rpm to 1500 rpm was achieved at a high-speed stage, which serves as the input to the generator attached to the gearbox housing. The test bench report, as discussed in the results section, indicates that, with the aid of a vibrating pen and a transducer developed using specialized technology, a high-intensity vibration manifested as whistling noises that became permanently distorted at sound levels in the range of 100 decibels. Hence, the gearbox housing only was considered after reviewing the test bench report, and the Pro-E model of the gearbox housing was analyzed in ANSYS with a frequency range of 700 HZ to 800 HZ; mode extraction was limited to ten modes. The stiffener was added in perspective mode to avoid resonance. The following are the objectives of the study

- To enhance the dependability and extended operational lifespan of entire gearbox.
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- To prevent gearbox resonance, which improves the reliability of generator operations.
- To develop strategies to eliminate or mitigate resonance in the gearbox, explicitly focusing on the housing component.

The research gaps identified to overcome the resonance followed by absence of Simulations in Modeling, building precise models for simulation to forecast the performance of transmissions throughout diverse circumstances helps facilitate the creation and improvement of such elements. Another factor contributing to the research gap is the absence of vibration-tracking systems that create sophisticated vibration-detection capabilities to identify early indicators of resonance and other complications in clutches, which would help improve repairs and maximize availability. Examining the breakdown characteristics of the components used in gearbox sections subjected to repetitive stress is crucial. This can assist in forecasting transmission longevity and enhancing their resilience.

Research identified in offshore gearboxes has prompted numerous studies that focus on constant conditions; however, actual wind turbine transmissions frequently operate under unpredictable conditions due to fluctuating gusts and pressures. Further investigation is required to comprehend the behavior of reverberation under specific complex circumstances. Although approaches such as vibration-based defect diagnostics have been investigated, there remains a necessity for improved ways to process signals that can precisely identify and evaluate resonances in windmill transmissions [27-29]. Incorporating information gathered from different devices can yield a more thorough understanding of transmission conditions and vibrations. Nonetheless, investigations into efficient multi-sensor data extraction methodologies remain constrained. The irregular dynamics of wind-turbine transmissions, particularly super harmonic impacts, remain poorly understood. Additional research on nonlinear motions and their influence on transmission effectiveness is required. The effect of reverberation on the immediate surroundings, including nearby species and buildings, necessitates further examination to guarantee the environmental health of renewable power methods. Offshore wind farms are subjected to large, rapidly fluctuating loads from wind and wave interactions. Comprehending the impact of these factors on reverberation necessitates additional investigation. The study is significant for several reasons, including environmental impact: resonance-induced noise in offshore applications can have serious ecological repercussions, potentially causing lasting harm to marine life. The elimination of this noise is critical for the preservation of the environment. Resonance can also influence the structural integrity of the gearbox, perhaps leading to premature failure. Resolving resonance-related concerns improves the dependability and longevity of the equipment. The study's results may be applied to offshore wind turbine gearboxes and other heavy-duty applications, enhancing performance and reducing maintenance costs. The novelty in this work is the identification of resonance during the HSS stage using a test bench and a setup frequency range of 700 to 800 HZ in ANSYS Workbench. Among the first ten modes, the natural frequencies of the gearbox match the operational frequency. Comparison of the test bench report with the Ansys workbench showed that the gearbox's fourth natural-frequency mode matched; hence, a stiffener was added to ensure the design feasibility of the gearbox housing. An additional novelty of this work is that damage confined to the gearbox housing can critically damage the entire gearbox because the high-speed shaft runs through the housing and is connected to the generator. Hence, adding the stiffener to the housing reduces resonance. This will reduce noise from the gearbox, thereby lessening environmental impact. The motivation behind this research is to determine the occurrence of resonance in the entire gearbox using an FFT test bench and ANSYS software. Resonance may induce excessive vibrations, potentially resulting in failures of the transmission elements. Timely detection helps avert these issues and prolong the service life of the gearbox. When maintenance is minimized, malfunctions can cause considerable turbine outages, resulting in diminished energy output and income loss. Correctly detecting resonating concerns enables preventive repair scheduling and thereby minimizes unanticipated absences. Comprehending and alleviating resonance problems improves the overall endurance of the wind machine. This is especially critical for offshore wind turbines, where maintenance and repairs are more complex and costly.

2. Materials and methods

2.1. Materials

The materials used for the gearbox and its associated parts, and the properties of these materials are presented in Table 1. The materials and compositions for the torque arm, ring gear, cover of the gearbox, housing, sun gear, Intermediate gear and HSS gear, planetary gear, rotor, and bearings were clearly illustrated. The torque arm withstands substantial rotational shear forces. Hence, according to

AISI standards, the elemental composition is 85% nickel, 15% cast steel, and lower percentages of silicon and manganese. The choice of material selection is materials made up of Nickel and steel for the torque arm, cast steel for ring gears, Nickel cast steel for the gearbox housing cover, steel for planetary gears, cast steel for sun gears, Cast steel for IMS shaft, HSS shaft, and planetary career. The lifespan of the gearbox and its associated parts depend on the durability of the materials selected. The selection of materials, such as cast steel, for the planetary gear reduces the formation of cracks under dynamic loading compared with other materials. Cast steel was selected for the sun gears, IMS shaft, and HSS shaft because of its reliability and adaptability for casting compared with other materials. One more reason behind the selection of these materials is high reinforcements subjected to static and dynamic loading tend to result in less replacement of the parts owing to the higher span of the gearbox [30]. One additional advantage of selecting cast steel and stainless steel as materials is that their weight is significantly lower, which reduces manufacturing costs.

Table 1. Materials and properties of the gearbox

Part Name	Material	Youngs Modulus MPa	Density kg/m3	Poisson Ratio
Torque Arm	AISI85NI15CS	220000	7860	0.31
Ring gear	SAE 0022CS	170000	7260	0.30
Cover of Gearbox	AISI85NI15CS	104000	7350	0.32
Planetary Gears	AISI 52100SS	120000	7260	0.33
Sun gear	SAE-0055CS	110000	7321	0.28
IMS Shaft	AISI9310CS	109000	7240	0.29
HSS Shaft	AISI4140CS	110100	6240	0.30
Planetary Carrier	AISI1045CS	101010	6230	0.33
Rotor shaft	AISI52100SS	102000	6385	0.27
Bearings	AISI 52100SS	110100	6890	0.31 mm

2.2. Methods

Figure 1 represents the process involved in gearbox modelling and analysis. This stage includes the design of gearbox components using Pro-E and assembly. The entire housing is analyzed in ANSYS, and the results are compared for all three stages.

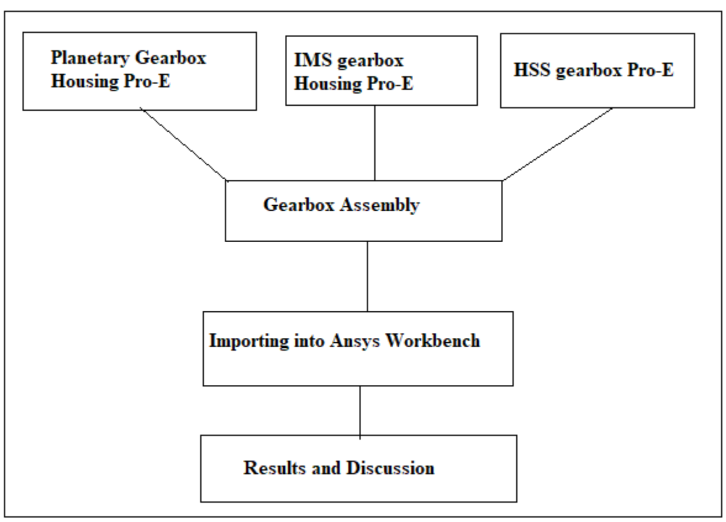

Figure 1. Flowchart for gearmodelling and analysis

The methods required for this analysis comprise two major classifications: the design part of the gearbox model, subjected to three different load conditions, including complete assembly of the gearbox and its associated parts, and neglecting the planetary gear. The final case is the gearbox housing, which is considered after the test bench report and FFT analyzer report for frequency calculation. The entire model is drawn in Pro-E software and imported into Ansys Workbench for this analysis. Table 2 represents the type of contacts used in this analysis.

Table 2. Contacts used in this analysis

S.NO	Type of contact	Tangential behavior	Normal behavior
1	Frictionless	Open Type	Open Type
2	Rough	Open Type	Closed Type
3	Bonded	Closed Type	Closed Type
4	No Separation	Closed Type	Open Type

3. Experimental set up

Figure 2 illustrates the general test bench system employed in this analysis; the complete gearbox will be mounted on the test bench, and a vibration sensor system will also be installed. All the data are keyed into the FFT graph to establish the natural frequencies of the gearbox and its components. The test bed has a first stage, a planetary gearbox, an intermediate stage, and a high-speed stage that records the vibrational frequencies. Nonetheless, load case III is of great significance, as several studies in literature have been conducted to forecast the resonance component of the gearbox housing.

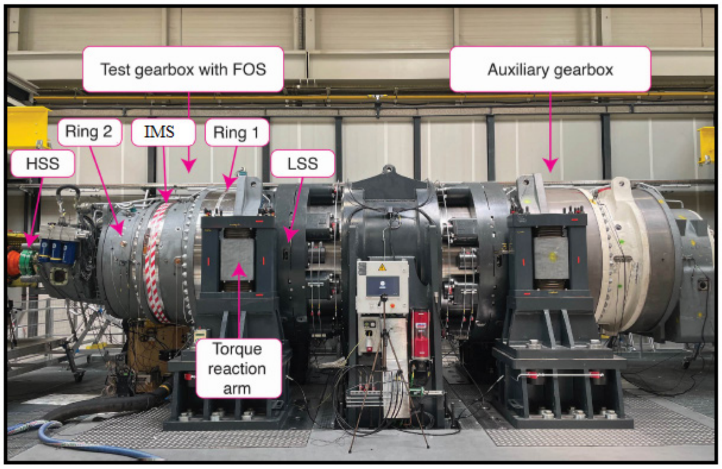

Figure 2. Test bench arrangement of the entire gearbox [31] permissions from wind energy science

3.1. Mathematical formulations

Figure 3 depicts the mass distribution, the kinetic energy and force distributions, and the gearbox's moments. For every stage, the basic kinetic energy equation that evaluates the reduction in torque of the planetary gears, intermediate gears, and high-speed gears is given by Equation (1), as referenced in [32].

$$KEE_1 = \frac{1}{2} I_{G_1} \omega_{G_1}^2 \quad (1)$$

Equation (2) refers to the kinetic energy and the relevant angular velocities that formed in the intermediate stage of the gearbox concerned with different rpm referred from [33].

$$KEE_2 = \frac{1}{2} I_{G_2} \omega_{G_2}^2 \quad (2)$$

Equation (3) refers to the kinetic energy and the relevant angular velocities that formed in the high-speed stage of the gearbox, which is concerned with different rpm referred from [34].

$$KEE_3 = \frac{1}{2} I_{G_3} \omega_{G_3}^2 \quad (3)$$

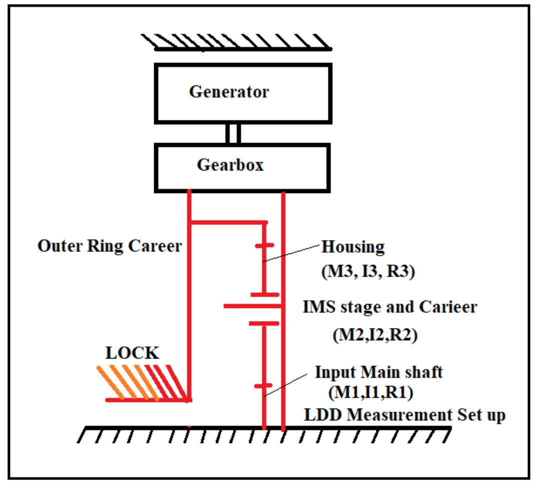

Figure 3. Mass distribution simplified setup

3.2. Validation study

The significance of the Validation study concerned with the different stages of the sun gear and its rotational speeds, the three important factors that affect the performance of the gearbox are determined by the a) Inertial factor, b) kinetic energy concerned with mass, c) Inertial or internal leverage mechanisms, of the sun and ring gears these can be correlated by the Equation (4) referred from [35]

$$I(C) = IC_1 + Mi_1 [I_i \frac{R_i}{3R^2}] + Mi_2 [\frac{R_i}{3R^2}] + Mi_3 [\frac{R_i}{(1 - I_W)}] \quad (4)$$

IC_1 = Inertial forces occured from all load cases

Mi_1 = Moment of inertia for all gear stages

I_W = Inertial forces caused for sun and ring gears

The restructuring of the forces in question that pertain to the appropriate modulation of the gear tooth transmissions is contingent on the transmission meshing frequencies and the time-dependent signals. The transmission frequencies are based on the variances of the excitation vibrational factors that result in heavy time-varying response signals that induce the disturbance motions of the time-travelling elements, as strongly exhibited in Equation (5) mentioned in [36].

$$(T)^g = \frac{a_i}{2} + \sum_{n=1}^{\infty} a \frac{\cos(2\Phi nt)}{T} + b \frac{\sin(2\Phi nt)}{T} \quad (5)$$

$$S(T)^g = Mi_1 [t_i - \frac{T_i}{(\in -1)}] + Mi_2 [t_i - \frac{T_i}{(\in -1)}] + Mi_3 [t_i - \frac{T_i}{(\in -1)}] \quad (6)$$

T_i = Time varying signals concerned to different intervals

$\in$ = Travel moments of the gear concerned to pitch

Equation (6) refers to the gear travel movements, which are considered [37], namely, on the meshing frequencies and length of transmission contacts and superposition principles during the periods. The various superposition principles that deal with definite transmission levels and the mode of travel period of the gears are subject to the modulation frequencies during the contract period. It will affect the functioning of the intermediate-stage gears and their vibrational characteristics. Equation (7) above, cited in [38], relates the overall gearbox performance to the various errors introduced by the transmission pitch.

$$e_T = e(t_o) + e(t_i)\sin\frac{(2\Phi nt)}{T} \quad (7)$$

e_T = Transmission dynamic error

$e(t_o)$ = Mean dynamic error

T = Meshing periodic elements of the gear

Φ = Modulation signals of the gears

4. PRO-E models

The gearbox design was modelled in Pro-E Software, as shown in Figure 4. The main parts include the design of the gearbox, comprising the main shaft of the rotor, the first-stage planetary gearbox, and the intermediate stage, which comprises a helical gear and assembly.

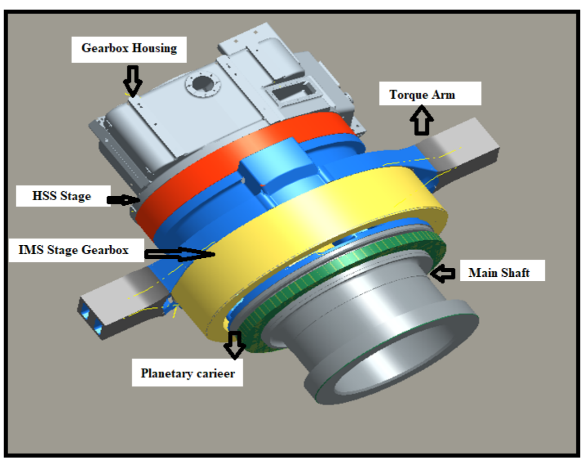

Figure 4. Pro-E Model of the Gearbox

The third section is a high-speed section that was run at the speed attained. All the other sections, including the torque arm, ring gears, main bearings, and axial bearings, were designed and assembled using Pro-E software. The volume of the planetary gearbox in relation to the sun, the planet and the ring gears is Equation (8). Referred from [39]

$$F_T = \sum_{i=n}^{n+1} f_{P*} \sum_{i=n}^{n+1} f_{R*} \sum_{i=n}^{n+1} f_S \quad (8)$$

where F_T= total forces acting on the gearbox planetary stage

where F_p= total forces acting on the pinion at planetary stage

where F_s= total forces acting on the sun gear at planetary

Figure 5 is the 3 D model of the gearbox housing with the torque arm. The torque arm is used as an element of integration to contain all degrees of freedom. It takes all the loads of the planetary carrier to the ring gears.

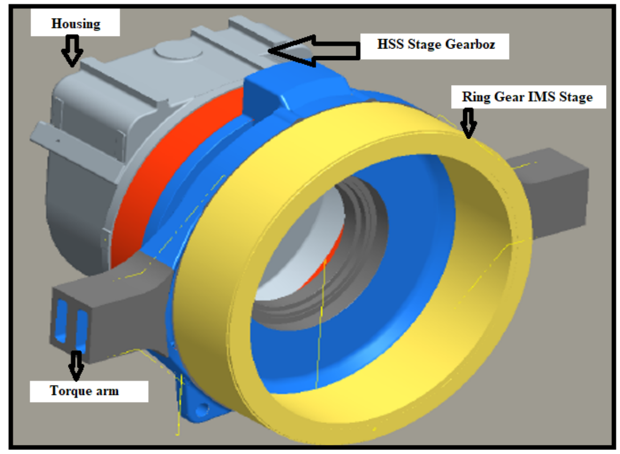

Figure 5. Pro-E Model of the Gearbox without planetary stage

Figure 6 is a pro-E model of gearbox-housing. The high-speed shaft is operating in the gearbox housing. Therefore, the deflections and deformations experienced during this stage are much higher than those under other loads.

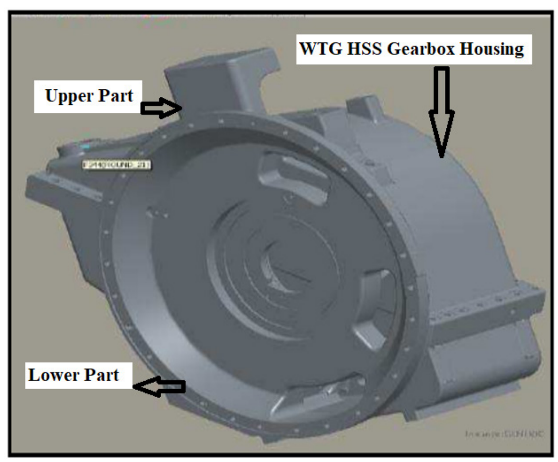

Figure 6. Pro-E Model of the Gearbox housing

5. Ansys analysis

5.1. Analysis of entire gearbox

The program, known as Ansys software, forecasts potential failures likely to occur during the operation of a given product prior to its production [40]. It lets the geometry preprocess the product to determine the sizes of the necessary elements and boundary conditions. The precision of any analysis report that is arrived at using ANSYS is determined by the nature of the element, the magnitude of the actual meshing strategy of the element and clear-cut conditions of the boundary [41]. The boundary conditions are clear to obtain precise outcomes commensurate with real analytical computations. Therefore, the choice of boundary conditions required a correct meshing strategy. The quantity and nature of elements and the mesh generated in the preprocessing step are determined by the gearbox's stiffness behavior and material strength [42]. The total number of nodes and elements for the entire gearbox was 609496 nodes and 201876 elements, as illustrated in Figure 7. Without planetary to form the nodes and elements to support load case II, it took 320952 nodes and 171088 elements, as depicted in Figure 8. In load case III, the number of nodes and elements was found to be 174438 and 153456, respectively, as illustrated in Figure 9. Therefore, the number of nodes and elements that are developed during the analysis depends on the geometry, material mass, and material shape. The mesh used in this case is a tetrahedral mesh to achieve high accuracy and improved results. In its bid to get better results, the element size has been maintained at 10 mm with the skewness factor.

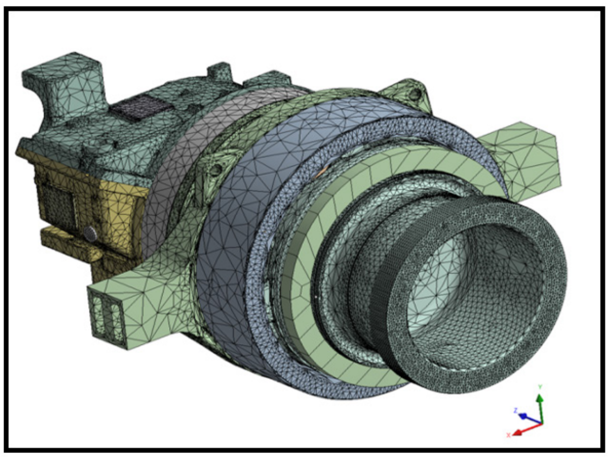

Figure 7. Ansys model of the entire gearbox

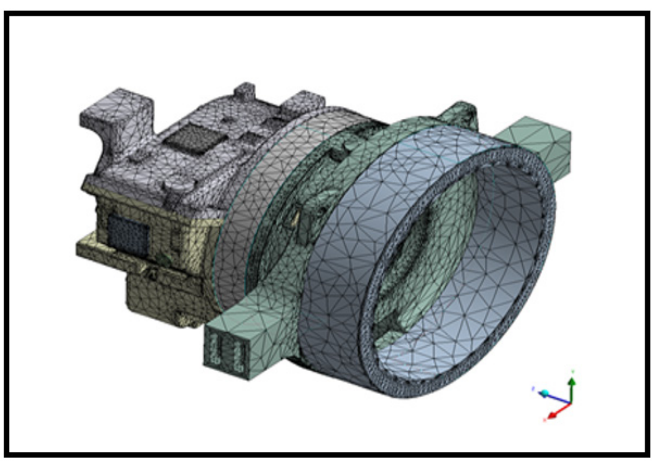

Figure 8. Ansys model of the gearbox without planetary

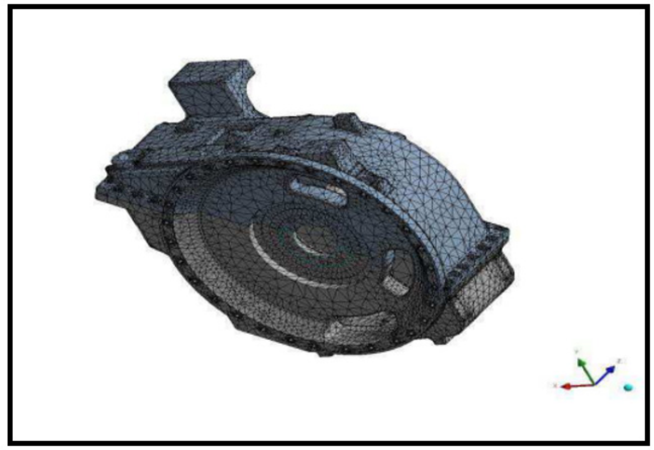

Figure 9. Ansys model of Gearbox Housing

5.2. Boundary conditions

Scientifically, the optimization of the design of any model to be analyzed in the analysis software, with the aid of necessary fixed supports that limit rotation in all directions, is referred to as a boundary condition [43]. That is, the word' boundary condition' is a critical limitation when conducting some analyses in the preprocessing phase. The accuracy of the analysis depends on the accurate boundary conditions and model geometry that were developed in the analysis part [44]. The boundary condition is presented at the top of the torque arm as illustrated in Figure 10. Each rotating force, such as shear stress, tensile stress, and compressive stress, was developed during the actual rotation of the entire gearbox, which was supported by two arms. Therefore, there was a need to provide constant support on both sides of the torque arm, which restrains the degree of freedom in all directions [45]. Figure 10 is a representation of the load cases 1 and 2, and Figure 11 is a representation of the case 3 boundary condition.

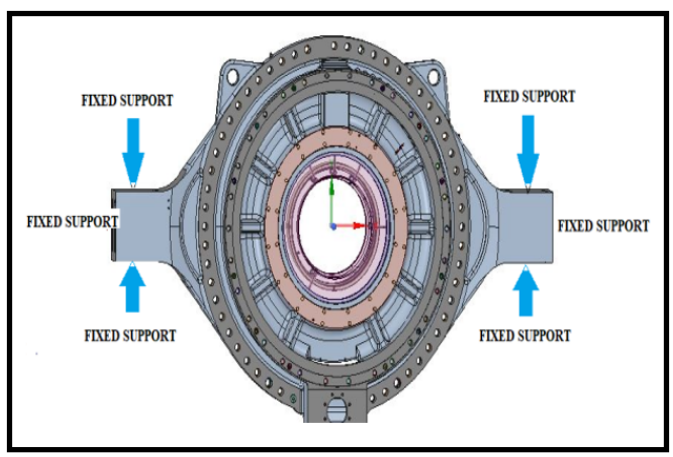

Figure 10. Boundary condition of torque arm for the load case 1 and 2

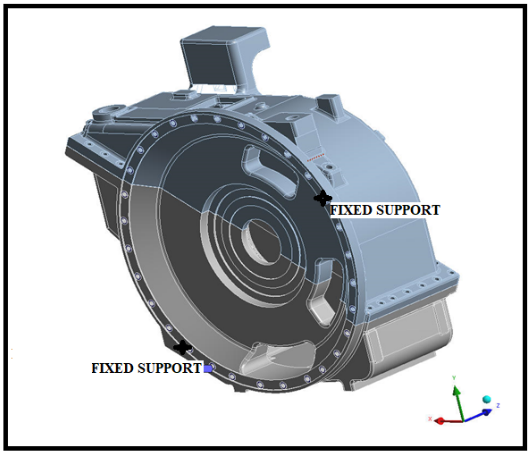

Figure 11. Boundary condition of housing for the load case 3

5.3. FFT Report

The extraction of natural frequencies through the determination of definite modes was a scientific calculation using a specialized discrete tool known as Ansys, and this analysis was called modal analysis [46,47]. Figure 12 is a representation of the FFT test bench report. All the gearboxes are in the first 10 modes in Ansys Workbench. Among the initial 10 extracted modes, the highest frequency was 272 Hz. It was found that the operational frequency of the gearbox housing was not in harmony with the first ten natural frequencies. This is because the mass, stiffness, and inertia forces are increased [48]. The thrust generated by rotating components between the main rotor and the planetary and IMS gears of the gearbox, such as bearings, produces lower frequencies of 57 Hz to 272 Hz. The obtained frequencies have no impact on the resonance of the complete gearbox [49]. Table 3. Denotes the Load case 1 Modal Frequencies output.

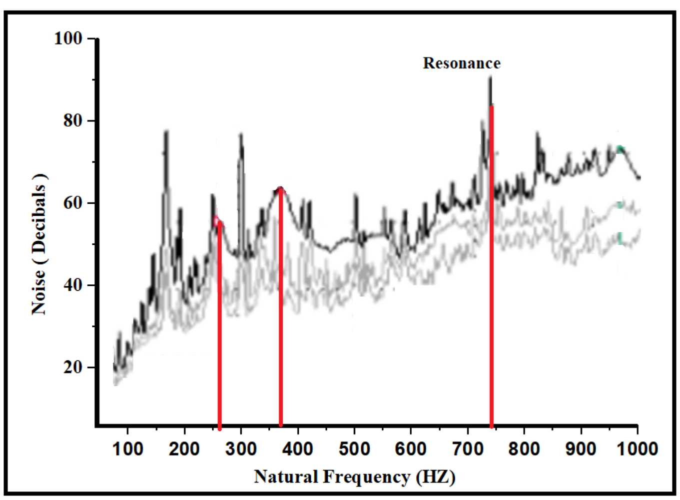

Figure 12. FFT report on the gearbox

6. Results and discussion

6.1. Case I: entire gearbox – modal analysis results

This computation of extracting specific modes to obtain natural frequencies was performed using a special discretized program, Ansys, and was referred to as modal analysis [50]. The tool is widely used to calculate the natural frequencies of any definite or indefinite structure and to obtain the model frequencies [51]. Figures 13 and 14 are the results of the extraction of the gearbox under vibration and the frequency of the first ten modes, respectively. The entire gearbox is obtained from the first 10 modes in Ansys Workbench. Of the initial 10 extracted modes, the highest frequency was 272Hz. It was determined that the first 10 natural frequencies did not coincide with the gearbox housing's operational frequency. This is because of the increased mass, stiffness and inertia forces [52]. The thrust generated by rotating components between the main rotor and the planetary and IMS stages of the gearbox, including the bearings, results in lower frequency levels between 57 Hz and 272 Hz. The resonance phenomenon of the entire gearbox is not dependent on these frequencies obtained [53].

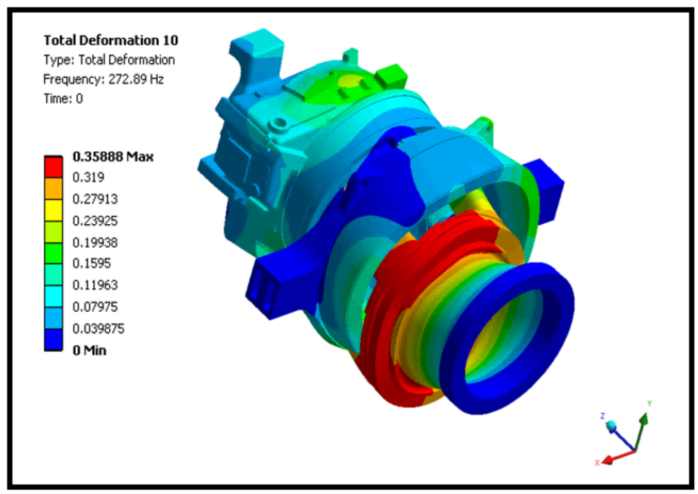

Figure 13. Modal results of gearbox

Table 3. Load Case 1 Modal Frequencies results

S.NO	Modes	Frequencies in HZ
1	Mode 1	57
2	Mode 2	104
3	Mode 3	154
4	Mode 4	163
5	Mode 5	175
6	Mode 6	202
7	Mode 7	216
8	Mode 8	240
9	Mode 9	260
10	Mode 10	272

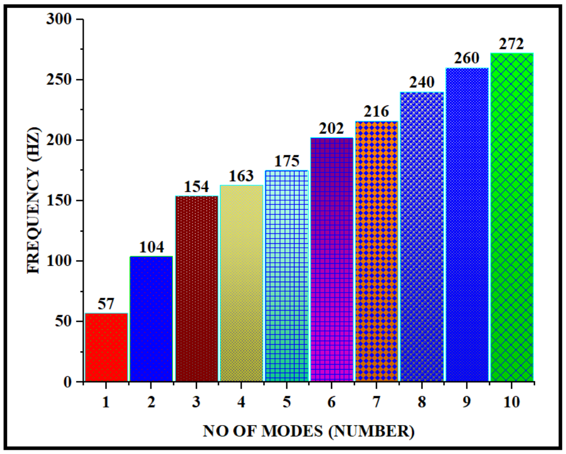

Figure 14. Entire gearbox modal results

6.2. Case II: gearbox without planetary – modal analysis results

In the obtained results of case I, the natural frequencies of the gearbox model did not coincide with the excited or operated frequency of the gearbox housing. This made us arrive at the specific decision not to consider the first stage planetary gearbox to minimize its mass and stiffness characteristics of the complete gearbox. Figure 15 shows the modal analysis of the gearbox without planetary stage. The results of the graphical analysis of load case II are presented in Figure 16. Table 4 gives indications of Load Case 2 Modal Frequencies findings. As shown in Figure 15 and Figure 16, Natural frequencies of the gearbox without the planetary stage could be obtained between the 47 HZ and 267 HZ, which were not the same as the operated frequency of the gearbox (740 HZ). This is because of higher weight ratio of bearings used in the intermediate stage and the mass of the torque arm, sun gear and ring gear and its related components would be optimal leading to lower frequency ranges compared to the excited frequency of the gearbox [54]

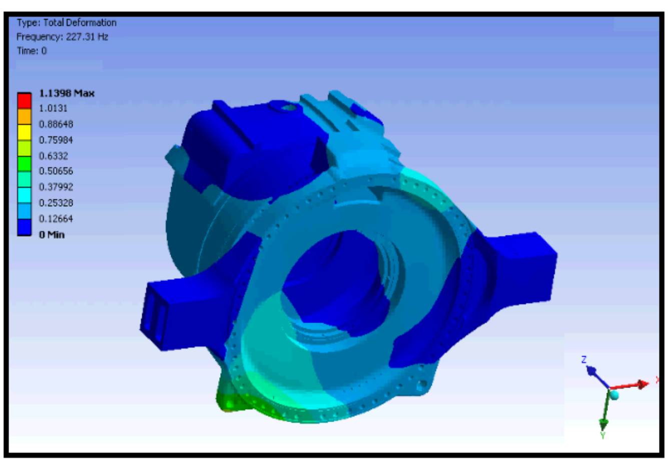

Figure 15. Modal results of gearbox without planetary

Table 4. Load case 2 modal frequencies results

Modes	Frequencies in HZ
Mode 1	47
Mode 2	75
Mode 3	134
Mode 4	153
Mode 5	173
Mode 6	198
Mode 7	226
Mode 8	248
Mode 9	252
Mode 10	267

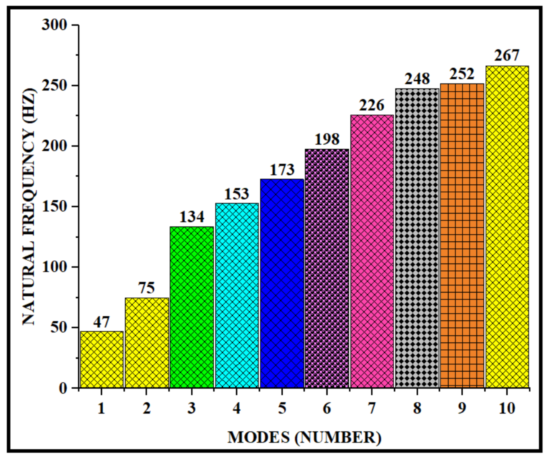

Figure 16. Gearbox without planetary modal

6.3. Case III. Gearbox housing- modal analysis results

Based on case I and II, it was found that the natural frequency of the first 10 modes did not coincide with the frequency of operation of the gear box. Therefore, it was resolved to overlook masses, minimize the preliminary phase of planetary gears and disregard the torque arm and Ims's stage. Figure 17 is the graphical illustration of the results obtained in the modes. It was obvious that the excited frequency of the gearbox corresponded to the natural frequency of the 4th mode (740HZ). This is attributed to the fact that in load case 3 the high-speed shaft is at 1150 rpm, and this is what leads to excessive resonance at the generator side [55,56]. Table 5 suggests Load Case III Modal Frequency Results.

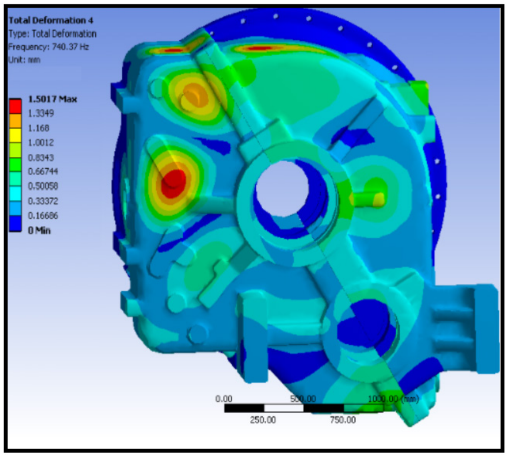

Figure 17. Modal results of the Gearbox housing

Table 5. Load Case III Modal Frequency Results.

S.NO	Modes	Frequencies in HZ
1	Mode 1	713
2	Mode 2	720
3	Mode 3	736
4	Mode 4	740
5	Mode 5	752
6	Mode 6	781
7	Mode 7	796
8	Mode 8	802
9	Mode 9	815
10	Mode 10	823

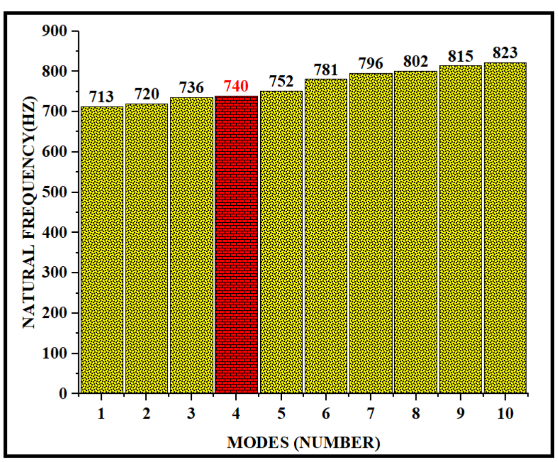

Figure 18. Gearbox Housing modal results

The modal analysis of the first ten modes is indicated in Figure 18. The graphical outcome of the modal results of load case 3 plays a significant role in representing the 4th mode that matches an operating frequency of the gearbox. Based on the analysis, it is perceived that the impact of excited frequency will be overcome using increased stiffener to reduce the resonance. This decreases the levels of intensity by 100 to 80 dba [57,58].

6.4. Implications and discussion

The insights gained during the analysis may be used as inputs to the design of gearbox components, including the addition of stiffeners to overcome resonance. Noise pollution will be reduced at offshore sites, benefiting marine life and human activities. Resonance and vibration dynamics are studied in a more connected world than in ordinary engineering applications. The main purpose of the study was to address resonance issues in weight-duty gearboxes operating in unfavorable offshore environments. Nevertheless, the concepts and methodologies acquired during the research can make a considerable difference in biomedical engineering, where precision and reliability are decisive factors. Although this seems to have little to do with the complexities of the biomedical arena, the heavy-duty gearboxes are in the same thread of resonance-related issues. The implications of resonance can be extended by both fields, not only the functioning of the apparatus, but also the health of people and in some cases even the environment. In the case of offshore gearboxes, vibration manifested as noise pollution that could reach levels threatening marine life. This resonance was identified and subsequently eliminated by engineering, and the achievement was a technical breakthrough and a credit to the company's broader efforts in the areas of responsibility. On the same note, the problems of resonance and vibration may pose a serious challenge in the biological sector. Total stability is required in precision medical equipment like MRI machines, CT scanners, and robotic surgical systems to ensure proper diagnosis and the delivery of sensitive treatment. Even minor resonances or vibrations can compromise the integrity of medical data or, in a worst-case scenario, endanger patients' health during surgery. Consequently, the way this work started, i.e. finding out and solving the problem of resonance in the offshore gearboxes, can be viewed as a transition to the problems associated with biomedical engineering. Although the apparatus and surroundings differ, the principles of vibration analysis and resonance abatement are quite similar. This paper has shown how knowledge can transcend distances and offer solutions that benefit many sectors, including biomedical engineering.

7. Conclusion

Analysis results can be fed into the design process for gearbox parts, for instance, to add stiffening of the gearbox to prevent resonance. Offshore noise will be minimized, supporting marine and human life. Resonance and vibration dynamics studies are not limited to normal engineering practice in an ever-connected world.

The aim of the study was to overcome the resonance problems associated with heavy duty gearboxes in harsh offshore applications.

However, the principles and techniques developed from this work could have significant implications for biomedical engineering, where accuracy and dependability are paramount. It might appear unconnected to the problems of complicated biomedical business, but heavy-duty gearboxes have a comparable resonance associated issue. In both fields, the effects of resonance can be far-reaching, affecting not only equipment performance, but also the health of people and in some instances, the environment.

Offshore gearboxes suffer from resonance like noise pollution that can be harmful to marine life. This finding of the cause and subsequent elimination of the resonance was an issue of engineering progress and a credit to the company's greater effort to be environmentally responsible.

Similarly, resonance and vibration can be very detrimental to the biological system. Medical equipment, such as MRI machines, CT scanners, and robotic surgical systems, needs utmost stability to ensure precise diagnoses and intricate surgeries. Medical information could be destroyed by resonances or vibrations, or worse, lives could be at risk during an operation.

Thus, the focus of this study, the discovery and treatment of resonance in offshore gearboxes, may be seen as a step towards solving problems in biomedical engineering. The equipment and operating conditions vary, but the concepts of vibration analysis and the resonance reduction do not differ very much.

This paper has highlighted the ability of knowledge to cross boundaries and provides answers that impact a number of sectors including biomedical engineering. We suggest the reader pursue the general problem of reverberation, which is applicable to the engineering professions, and we burrow into the specifics of resonance measurements, analysis, and elimination**.**

Author contributions statement

K. Sunil Kumar – conceptualization and writing - original draft. J. Senbagamalar– Formal analysis. S. Sudha- methodology, software. J. Jeha - Review & editing, supervision.

Declaration of interests

The authors have any competing financial or personal interests related to this work.

Data availability statement

The datasets used and or analyzed during the current study are available from the corresponding author on reasonable request.

Declaration of generative AI and AI-assisted technologies in the writing process

The authors would like to thank AI-powered writing tools, such as Grammarly, for their assistance in enhancing the linguistic precision, correctness of grammatical structures, and the overall coherence of this article. All content, analysis and conclusions contained in this work are the author's own and are not in any way the work of AI.All content, analytical discussion and conclusions presented in this work are the sole work of the author and have not been generated by the use of AI tools for any purpose other than to enhance the presentation and clarity of the text. The use of AI is aligned with the journal's standards of transparency and ethical guidelines in the authoring process.

Appendices

CS	Cast Steel
CT	Computed Tomography
Db	Decibels
DT	Demodulator Technology
DPORE	Dynamic Position Offshore Renewable Energy
IMS	Intermediate stage
FFT	Fast fourier transformer
Hz	Hertz
HSS	High-speed stage
MRI	Magnetic Resonance imaging
Rpm	Revolutions per minute
SAE	Society of automotive engineers
SS	Stainless Steel
TEO	Teager energy operator
SSR	Stochastic sensing reactance
MSNR	Modified signal numerical ratio
85NI15CS	85 percent nickel 15 percent Cast Steel

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