Research Article

design and optimization of automotive cardan shafts for industrial applications with enhanced performance and durability

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ARTICLE INFO

ABSTRACT

A crucial part of a locomotive is the cardan shaft. A rear-wheel-drive vehicle's differential gear receives power from the engine through a spinning shaft known as a cardan shaft. The cardan shaft must turn at ever-changing angles between the gearbox and axle in order to function. High vibrations are experienced on it as a result of the constant rotation. This issue causes shafts to flex or distort. In this study, a cardan shaft's design and development are carried out. Although it is extensively beneficial in the locomotive industry, vibration is a significant issue with cardan shafts. Creo parametric software was used for developing the design, while Ansys was used to analyze it. Using Ansys software's modal analysis, verify the natural frequency. It is then contrasted with the different shaft cut-out geometries.

1. INTRODUCTION

The systems that allow for changes in the relative translations and angles between the shafts' axes, which are used in power transmission, are known as cardan shafts. The most fundamental and prevalent kind is the Cardan joint, also known as the Hooke joint. Figure 1 illustrates how the cross shaft, an intermediary component with a cross form, connects the two yokes, one on each shaft. The operating angle, which is the angle formed by the two shafts, is often constant throughout operation but is not always. Excellent design practice demands modest operating angles, often less than 25°, depending on the application. Despite this requirement, mechanical interference during the construction of the Cardan joint imposes a maximum working angle that is determined by the joint's proportions. The adaptability of the rotation speed ratio is a key characteristic of Cardan mechanisms. The magnitude of the rotation speed ratio's periodic change is controlled by the angle between the input and output shafts [1, 2, 3]. Depending on their spatial proximity, the input and output shafts' rotation angles will vary.
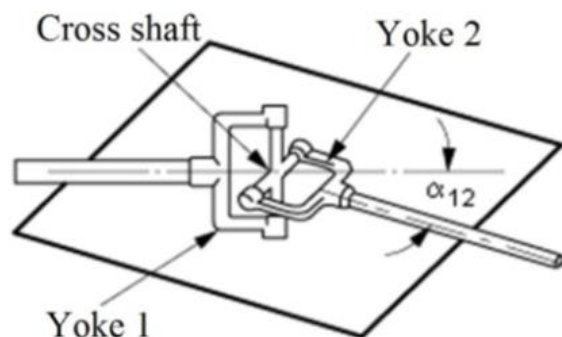

Fig. 1 Single Cardan joint [1]

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gearbox in a rear-wheel-drive vehicle. The angle between the gearbox and axle, at which the driving shaft must work, must change continually. Steel, often 35C8, 45C8, 55C8, etc., is the traditional material used for ordinary shafts. When an enormous amount of strength is required, an alloy of steel, such as nickel, nickel-chromium, or chrome vanadium steel, tends to be used [4]. A typical building material is high-quality steel (Steel SM45). Because the fundamental natural frequency of a shaft is inversely proportional to the square of the specific modulus and directly proportional to the square of the beam length, steel drive shafts are commonly built in two halves. The two-piece steel drive shaft increases the weight of a car and reduces fuel economy because of its design, which comprises three universal joints, a bearing in the center, and a bracket. It would be ideal to lower the weight of the drive shaft in order to reduce the overall weight of the vehicle without having to raise prices or compromise on reliability and quality. The weight of the drive shaft may be greatly lowered by optimizing the design parameters and following all limitations [5]. Advanced composite materials can significantly lower the weight of the driving shaft assembly. As a result, it is possible to transmit power to the assembly's differential components using a single drive shaft rather than a two-piece drive shaft. In addition to being lightweight, composites ensure reduced vibration and noise [6]. To make this economical transportation into more economical, there are many research works going on every year on several aspect like design modification and using composite materials. The propeller shaft of Maruthi Omni is taken and designed with it minimum dimensions to satisfy current problem specification and then replaced conventional steel material with composite materials. Torsional buckling and modal analysis on propeller shafts show carbon/epoxy composite material as a good substitute for conventional steel, using NX NASTRAN to compare computational models and empirical findings [7]. The researchers, Atul Kumar R. et al. [8], came to the conclusion that epoxy carbon UD, a thermoplastic polyimide containing 30% carbon fiber, might be employed as a substitute material for the propeller shaft. and shown that thermoplastic polyimide with 30% carbon fiber is the most advantageous material as a replacement for conventional material because its natural frequency is very similar to conventional material and its maximum stress generated is the same as that of conventional propeller shaft material. Comparing the optimized weight to traditional propeller shaft material, it is reduced by up to 82.04%. Basically, the shafts for propellers were made of steel, glass epoxy/HS carbon, and HM carbon/epoxy. The Bernoulli-Euler theory calculates a high recurrence because it ignores the effects of transverse shear and rotating idleness. The two-piece normal steel drive shaft of a vehicle has been proposed to be replaced by a single piece high-strength carbon/epoxy composite drive shaft going forward. The FEA evaluation was really helpful in getting the research approved scientifically [9,10]. The work's outcomes include advocating and promoting the replacement of standard propeller shafts with composite shafts, which have an extra preferred viewpoint. In addition, composite shafts are chosen because they are lighter and have a greater strength-to-weight ratio. The maximum static torsion might be increased by adding more layers, which would result in a mass reduction of 42% compared to a steel drive shaft and a maximum static torsion that is greater than pure aluminium by around 66% for +45/-45 3s laminates. Finite Element Analysis software was used to analyses a one-piece hybrid composite complete drive shaft to its optimum level. The results were then compared to those of a steel drive shaft that was already in use [11]. The propeller shaft near the universal coupling joint is experiencing frequent breaks due to high frequency vibration. To address this issue, the shaft thickness has been increased, and design changes have been made to the universal coupling. Vibrational analysis, structural analysis, stress and displacement values, and buckling analysis were conducted. The redesign offers resistance to damage to the propeller shaft and universal coupling assembly, resulting in a more robust design [12]. Insignificant weight savings of up to 28% have been achieved thanks to the use of composite material when compared to a standard steel shaft. It is obvious that of the two materials under consideration, the E-Glass/Epoxy composite has the most comforting features to go about as a substitute for steel when taking into consideration the weight sparing, twisting, shear stretch incited, and booming frequencies. In general, this is performed by using lightweight composites like E-Glass resin. The new work was intended to reduce the fuel consumption of the automobile in the specific machine or any machine that uses drive shafts. The provided work also offers the opportunity for plan progression, such as the conversion of a two-piece drive shaft into a single-piece, lightweight composite drive shaft [13]. Performance and fuel economy may be enhanced as a result. According to calculations made by Swapnil S. et al. [14], an aluminium shaft with an outer diameter of 72 mm and a thickness of 2 mm may be filled with 4 mm of Kevlar epoxy and utilized in place of a steel shaft for the specified torque gearbox application. The typical steel shaft in the provided application may be effectively replaced by a composite shaft comprised of aluminium alloy and Kevlar epoxy material. When a composite shaft is employed as the propeller shaft of the supplied vehicle instead of a typical hollow steel shaft, the weight of the shaft is decreased by 55.87%. The composite shaft was able to withstand the torque required for the application. The large wheel bases of vehicles are supported by lengthy propeller shafts. The shaft will create critical spinning and strong vibrations at a particular speed [15]. Thus, the shaft length should be shortened in order to decrease the critical spin. The length can then be decreased by splitting the shaft's length in half and positioning the intermediate with a universal joint. Power transmission decreases as the number of universal joints rises. As a result, composite materials can be used to manage it. According to E. Desnica et al. [16], rotating agricultural double cardan shafts at angles greater than 20 degrees is not recommended. When dependability cardan shafts are examined, it is found that very high diagnostic parameter values and a lack of lubrication from the bearing assemblies occur during operation at predetermined rotation angles of joints

and with very low workloads. The condition analysis of roller bearing sets for farm equipment mounted on cardan shafts under the particular use conditions highlighted all system defects while in use and suggested adjustments for ongoing use

(through the service life). This was done to rise the durability of cardan shafts. The dynamic behaviour of a Hooke's joint and a fracture-induced parametric excitation have been included in the model. An extremely nonlinear 3-DOF system with four linked inertias was produced using a Lagrangian approach to construct the equation of motion by taking into account the torsional vibration and frequency variation of the propeller. The system's unpredictable response has since been identified by applying the wavelet approach. A realistic nonlinear model has been built [17, 18] that considers the potentially relevant moments that act on a ground vehicle's propeller shaft in order to anticipate the response and characteristics of the propeller shaft system. The shaft made of carbon fiber now weighs just 2.4 kg and has a maximum stress of 181 MPa. 2.68mm and 9.59mm, respectively, are the shared stress values in the upper layers. Because the ultimate torque gearbox is less than the torsional buckling load, it is safer. With a maximum stress of 181 MPa, the carbon fiber shaft's weight has been decreased to 2.4 kg. The share stress measurements for the first and last layers at the top are 2.68 mm and 9.59 mm, respectively [19]. At frequencies beyond the working frequency range, propeller shaft stresses and displacement amplitudes reach their maximum levels. Because the ultimate torque gearbox is less than the torsional buckling load, it is safer. Employing energy theories, the governing equation of transient lateral and torsional vibrations has been determined. The equation was governed by a highly nonlinear system, with the introduction of a parametric excitation that simulated a breathing fracture. According to the investigation, most super-harmonic resonance occurs when the rotor passes it at rotational speeds close to critical speeds. This resonance results in high vibratory frequencies and an orbit of entangled loops. The research findings are applicable to supercritical twin-rotor systems [20].

2. DESIGN OF CARDAN SHAFT

Shaft length - 1500 mm

Cutout length - 50x50
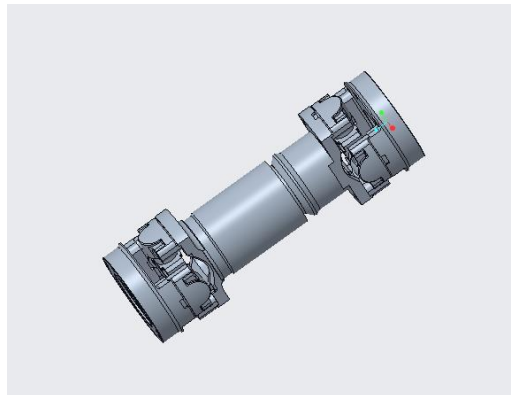
Cutout diameter - 150 mm

Fig.1. Existing Model in Creo Parametric Software
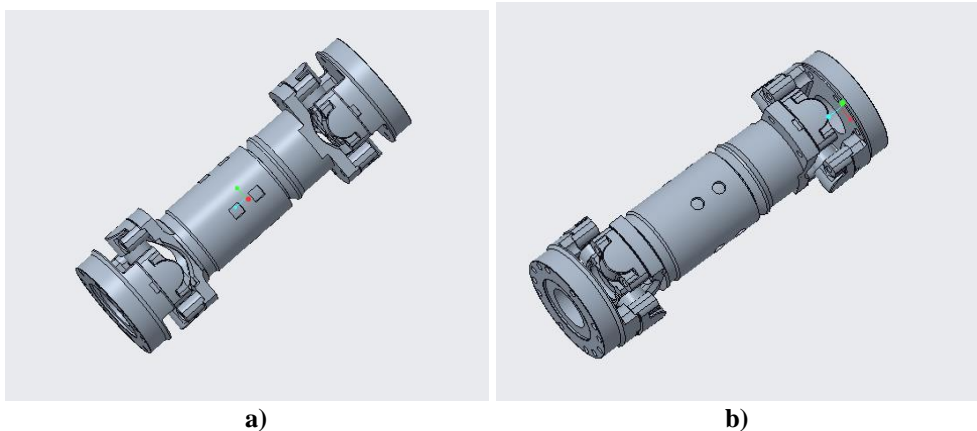

a)   b)

Fig. 2. Proposed a) Rectangular & b) Circular cut-out Model in Creo Parametric Software

3. MATERIAL SELECTION

A type of carbon steel known as "low carbon steel" is mild steel, which has a low carbon content. According to various sources, mild steel characteristically has between 0.05% and 0.25% carbon by weight, whereas higher carbon steels are sometimes characterized as having a carbon content of between 0.30% and 2.0%. Steel would fall under the category of cast iron if more carbon were added. Table 1 lists the material characteristics of mild steel, including its molecular density, Young's modulus, Poisson's ratio, and yield strength. Mild steel is widely used in various industries due to its affordability and versatility.

Table 1. **Properties of Mild Steel**

Properties	Units	Values
Density	Kg/m3	7850
Youngs Modulus	GPa	200
Poisson's Ratio	-	0.3
Yield strength	MPa	250

4. MODAL ANALYSIS

Using ANSYS FEA software, you can tackle challenging mechanical engineering issues and make wiser, quicker design choices. You may automate and customize solutions to your structural mechanics issues using the finite element analysis (FEA) solvers in the suite, then parameterize those solutions to analyses various design set-ups. For even higher fidelity, it is very simple to link to additional physics analysis tools. In the industry, engineers may optimize their product designs and save on physical testing expenses by using the ANSYS structural analysis tool.
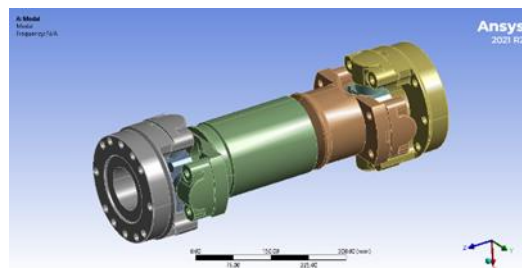

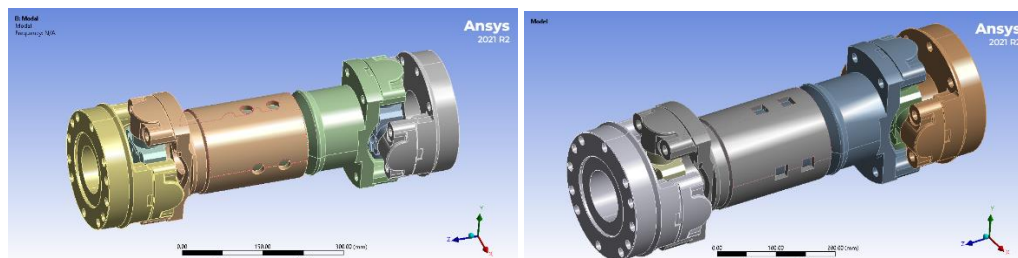FIGURE 3. Existing Cardan Shaft Model Imported into Ansys

FIGURE 4. Proposed Cardan Shaft Models Imported into Ansys

Ansys is the perfect answer for everyone, from designers and infrequent users searching for quick and reliable results to professionals looking to simulate complicated materials and non-linear behaviour. Engineers of various skill levels may

obtain prompt and accurate information thanks to ANSYS Mechanical's user-friendly interface. With ANSYS Mechanical, you can rapidly choose the best mesh for any model thanks to its intelligent mesh technology. It is straightforward to add controls for any necessary final modifications since automatic and clever algorithms ensure that high quality meshes are created.
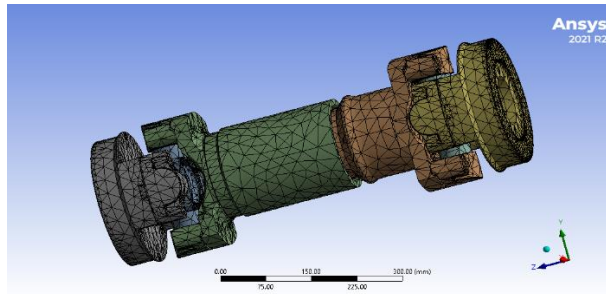

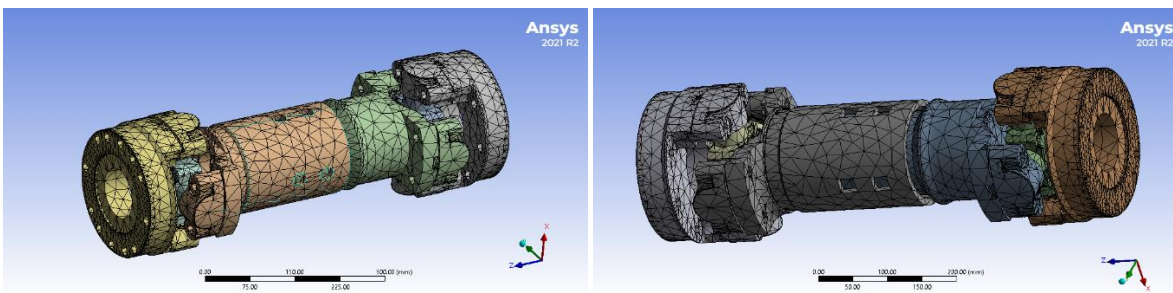FIGURE 5. Mesh Geometry of Existing Cardan Shaft Design

FIGURE 6. Mesh Geometry of Proposed Cardan Shaft Design

5. RESULTS AND DISCUSSION

The vibration properties, such as natural frequencies and mode shapes, are determined using modal analysis when designing a structure or a machine component. Modal analysis is used to identify the total deformation with respect to the vibrational frequency. The results are presented below for a brief comparison among them.
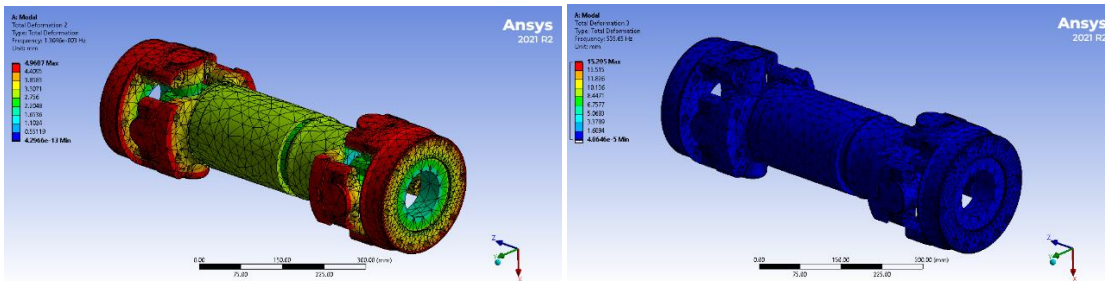
5.1 CONVENTIONAL MODEL

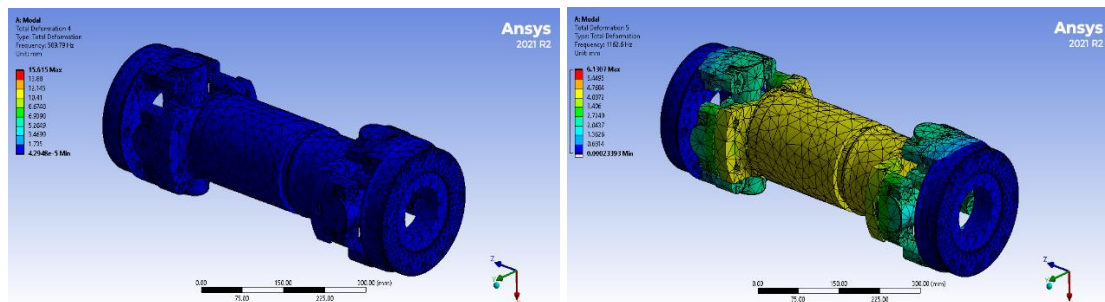FIGURE 7. 1st & 2nd Mode of Frequency Analysis for Model Without Cut-outs

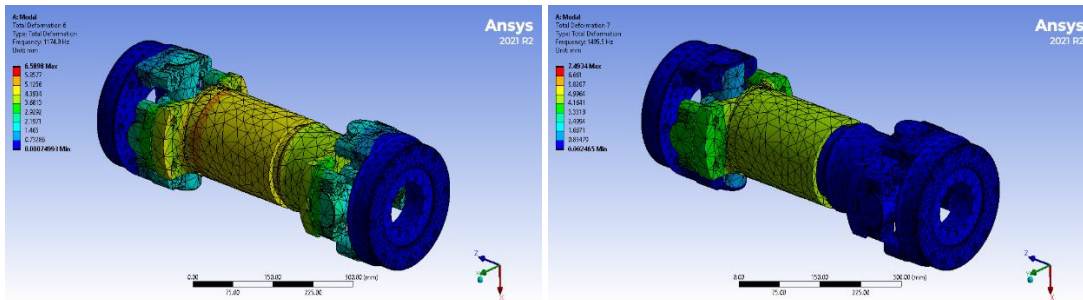**FIGURE 8. 3rd & 4th Mode of Frequency Analysis for Model Without Cut-outs**

FIGURE 9. **5th & 6th Mode of Frequency Analysis for Model Without Cut-outs**

All six modes of vibration findings from the modal analysis are shown in figures 7 to 9. There are deformations of 4.9607 mm at the least and 7.4934 mm at the greatest. The smallest and largest values of natural frequency are, respectively, 0.0013046 and 1495.5 Hz. The corresponding graphs of the results of the current model's deformation and frequency calculations are shown in Figure 10.
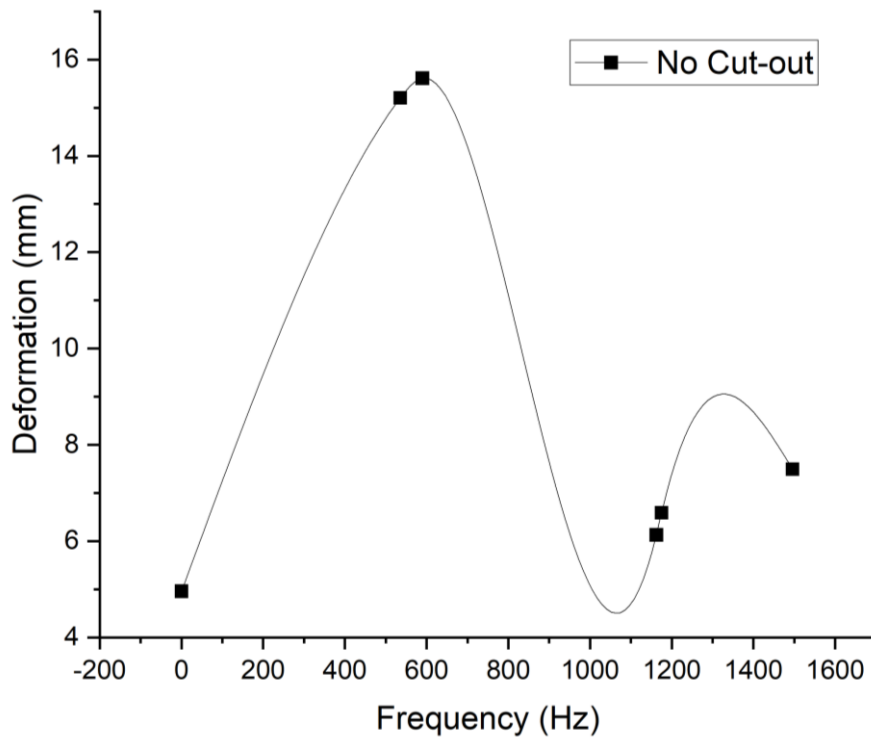

FIGURE 10. **Frequency Vs Deformation Results for Model with No Cut-outs**
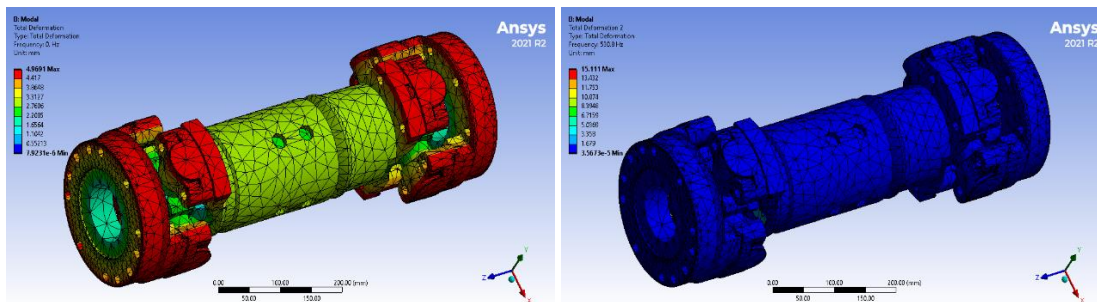
5.2 CIRCULAR CUT-OUT

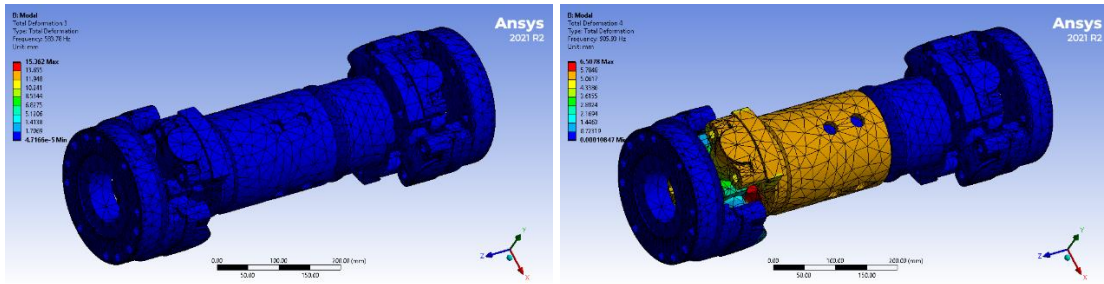**FIGURE 11.** 1ˢᵗ & 2ⁿᵈ Mode of Frequency Analysis for Model Circular Cut-outs

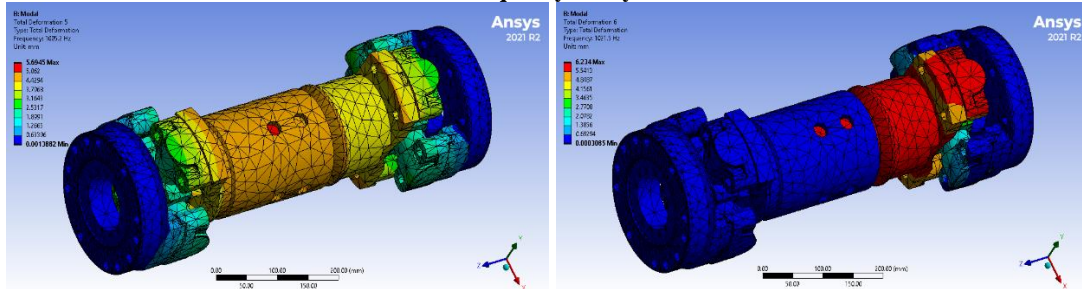**FIGURE 12.** 3ʳᵈ & 4ᵗʰ Mode of Frequency Analysis for Model Circular Cut-outs

FIGURE 13. 5ᵗʰ & 6ᵗʰ Mode of Frequency Analysis for Model Circular Cut-outs

All six modes of vibration findings from the modal analysis are shown in figures 11 to 13. There are deformations of 4.9691 mm at the least and 6.234 mm at the greatest. The smallest and largest values of natural frequency are, respectively, 0.00 and 1021.5 Hz. The corresponding graphs of the results of the circular cut-out model's deformation and frequency calculations are shown in Figure 14.

FIGURE 14. Frequency Vs Deformation Results for Model with Circular Cut-outs

5.3 RECTANGULAR CUT-OUT

All six modes of vibration findings from the modal analysis are shown in figures 11 to 13. There are deformations of 4.9691 mm at the least and 6.234 mm at the greatest. The smallest and largest values of natural frequency are, respectively, 0.00 and 1021.5 Hz. The corresponding graphs of the results of the circular cut-out model's deformation and frequency calculations are shown in Figure 14.
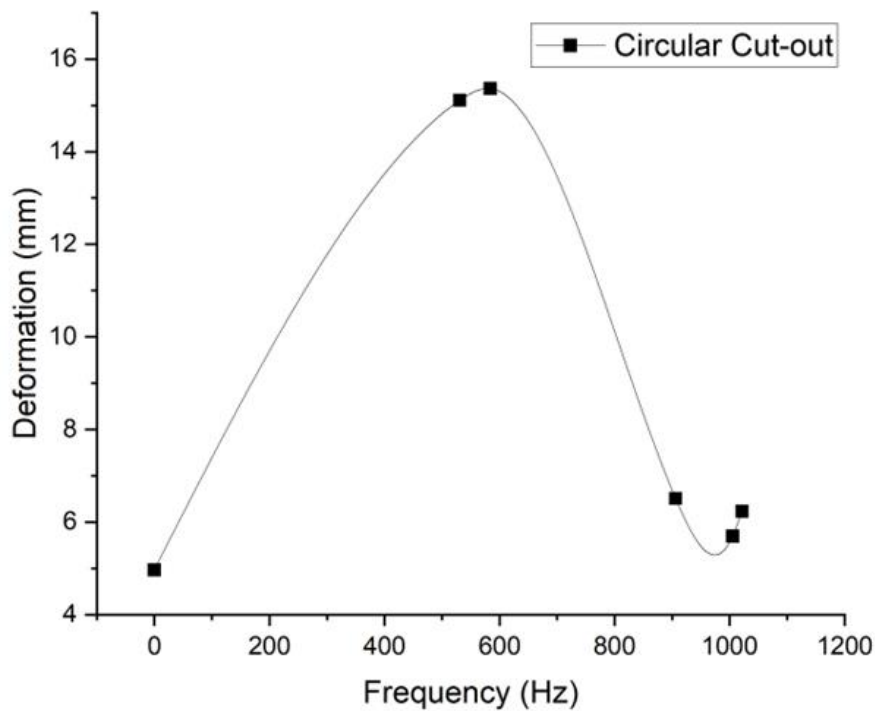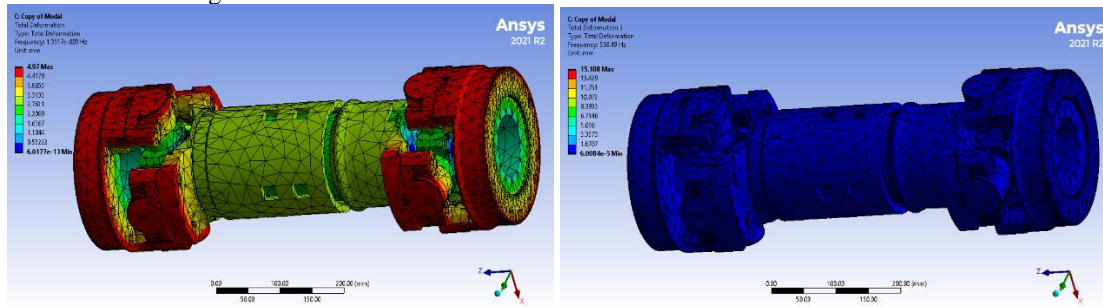

FIGURE 15. 1st & 2nd Mode of Frequency Analysis for Model Rectangular Cut-outs
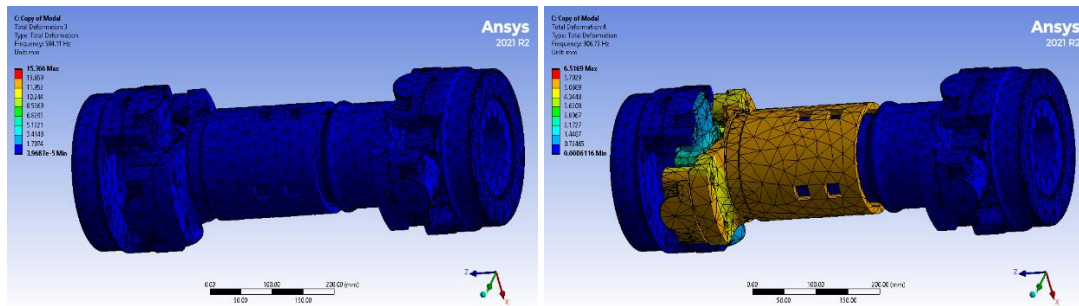
FIGURE 16. 3rd & 4th Mode of Frequency Analysis for Model Rectangular Cut-outs
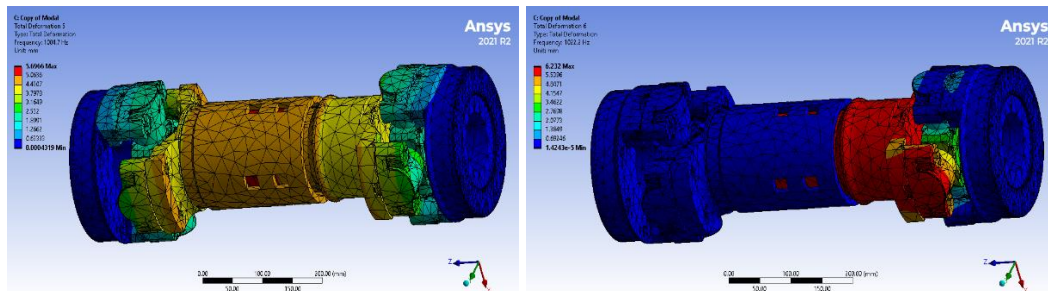
FIGURE 17. 5th & 6th Mode of Frequency Analysis for Model Rectangular Cut-outs

FIGURE 18. Frequency Vs Deformation Results for Model with Rectangular Cut-outs

All six modes of vibration findings from the modal analysis are shown in figures 15 to 17. There are deformations of 4.97 mm at the least and 6.232 mm at the greatest. The smallest and largest values of natural frequency are, respectively, 0.001352 and 1022.2 Hz. The corresponding graphs of the results of the circular cut-out model's deformation and frequency calculations are shown in Figure 18.
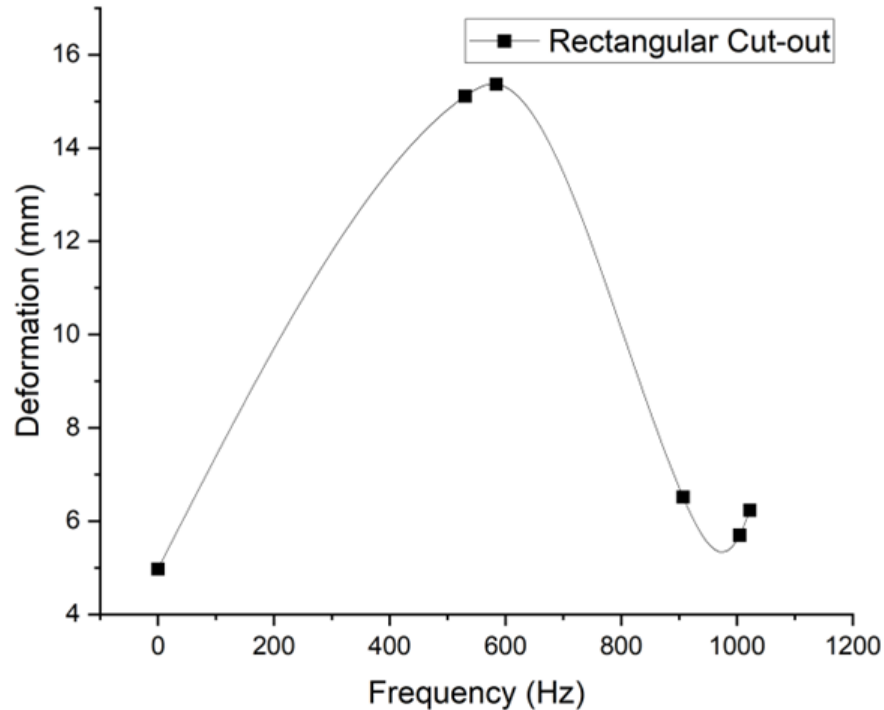
Table 2. **Natural Frequency Result Comparison of All Designs**

Modes	No-cut	Circular-cut	Rectangle-cut
1	0.0013046	0	0.001352
2	535.65	530.8	530.49
3	589.79	583.78	584.11
4	1162.6	905.93	906.72
5	1174.9	1005.2	1004.7
6	1495.5	1021.5	1022.2

Table 3. **Total Deformation Result Comparison of All Designs**

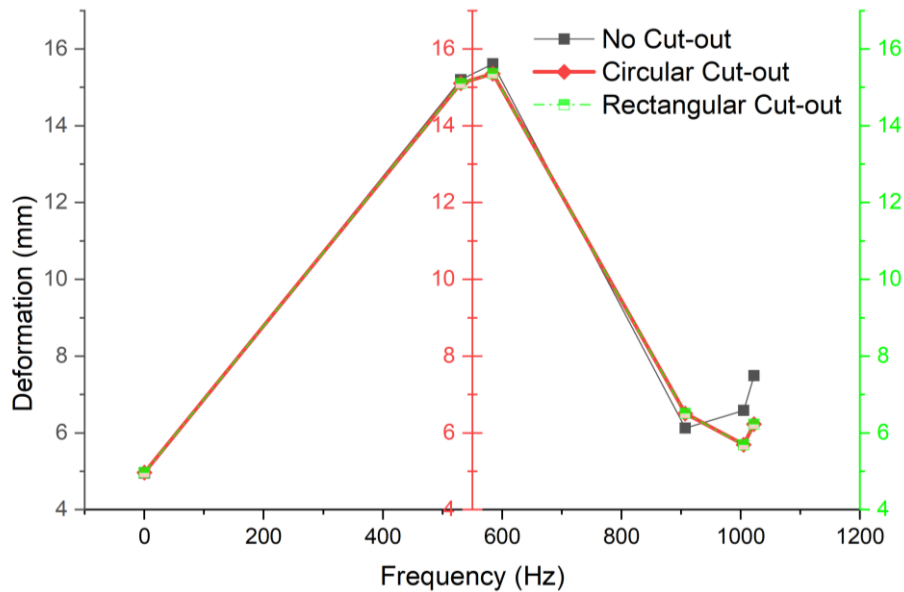| Modes	No-cut	Circular-cut	Rectangle-cut
1	4.9607	4.9691	4.97
2	15.205	15.111	15.108
3	15.615	15.362	15.366
4	6.1307	6.5078	6.5169
5	6.5898	5.6945	5.6966
6	7.4934	6.234	6.232

FIGURE 19. **Overall Comparison of Frequency Vs Deformation Results**

6. CONCLUSION

The modes of a cardan shaft's natural frequencies may be determined using a modal analysis for several cut geometries. In this study, several cut design forms are produced according to the greatest frequencies. Following the model analysis, Tables 2 and 3 list all modes in accordance with their deformation and natural frequencies. According to these findings, the circular cut geometry model exhibits less deformation and better vibration outcomes at the same boundary conditions. This suggests that the circular cut geometry is more efficient in reducing vibrations compared to other cut geometries. Additionally, these results highlight the importance of considering the shape and design of the cut geometry to be able to optimize the performance of cardan shafts.

Conflicts Of Interest

The authors declare no conflicts of interest.

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