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Kaburga Tasarımı Eklenen Kompozit Bir Kapağın Doğal Frekans Değerlerinin Belirlenmesi ve İncelenmesi

Determination and Investigation of Natural Frequency Values of A Composite Cover With Rib Design Added

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Öz

Bu çalışmada, Unigraphics NX programının CAD modülü kullanılarak uzun, ince ve esnek karbon fiber takviyeli kompozit bir kapak tasarlanmıştır. Bu tasarıma ANSYS ortamında sonlu elemanlar yöntemi ile modal analiz uygulanarak farklı modlar için analiz sonuçları elde edilmiştir. Modal analiz süreci ayrıntılı olarak anlatılmış ve temel yöntemler ve matematiksel denklemlerden bahsedilmiştir. Analiz sonuçları detaylı bir şekilde incelenmiş, eğilme ve burulma koşulları değerlendirilmiştir. Analiz sonuçlarına göre kompozit kapağın direngenliğini iyileştirmek için bir kaburga tasarımı yapılmış ve ilk tasarıma eklenmiştir. Nervürlü kapak tasarımı için bir modal analiz süreci gerçekleştirilmiştir. Modal analiz sonucunda farklı mod parametrelerindeki doğal frekans değerleri elde edilmiş ve değerlendirilmiştir. İlk kapak tasarımına kaburga eklenmesi ile mod 1, mod 2, mod 3, mod 4, mod 5 ve mod 6 değerlerinde sırasıyla %35,52, % 38,79,% 58,07,% 129,11,% 100,35 ve % 106,65'lik bir iyileşme gözlenmiştir.

Anahtar Kelimeler: *Kompozit, Doğal frekans, Modal analiz, Tasarım, Kaburga*

Abstract

In this study, a long, thin and flexible carbon fiber reinforced composite cover was designed using the CAD module of Unigraphics NX program. Analysis results for different modes were obtained by applying modal analysis to this design with finite element method in ANSYS environment. The modal analysis process is explained in detail, and basic methods and mathematical equations are mentioned. Analysis results were examined in detail, and bending and torsion conditions were evaluated. Based on the analysis results, in order to improve the stiffness of the composite cover, a rib design was made and added to the initial design. A modal analysis process was carried out for ribbed cover design. As a result of the modal analysis, natural frequency values in different mode parameters were obtained and evaluated. With the addition of ribs to the initial valve design, an improvement of %35.52, % 38.79,% 58.07,% 129.11,% 100.35 and % 106.65 was observed in mode 1, mode 2, mode 3, mode 4, mode 5 and mode 6 values, respectively.

Keywords: *Composite, Natural frequency, Modal analysis, Design, Rib*

1. Giriş

Every structure has a natural frequency. These frequencies, defined as modes, are used to determine the dynamic characteristics of the structure [1-5]. Natural frequency is seen as an important criterion in sectors with high safety requirements such as aviation and defense industry [6-8]. Resonance occurs when the natural frequency of the structure coincides with an effect of the same frequency [9]. All materials have resonances due to their physical properties [10, 11]. It is possible to obtain high vibrations by applying a very small driving force to a material at resonance frequency [12-14].

Whether a structure is damped or undamped affects the vibration that occurs. If there is no effect to cause friction, energy loss or damping in the system, vibration is called undamped [15-17]. If there is damping in a mechanical system, it is called damped system. In the examination of vibration and natural frequency values, neglecting the damping on the system may facilitate the analysis, but damping effects are very important, especially in the formation of resonance [18-20]. The vibration it has when it does not force the mechanical system is called free or natural vibrations. If vibration occurs when the mechanical system is exposed to an external force, the resulting vibrations are called forced vibration [21-26]. In Figure 1, the large-scale vibration movement of the composite cover is represented.

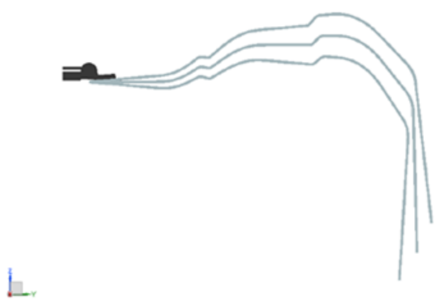

Figure 1. Large-scale vibration action of the composite cover.

It is important to determine the natural frequency of a design before creating a prototype and to determine whether it meets the desired boundary conditions. It is possible to use modal analysis to determine the natural frequency. T. Özben et al. conducted both experimental and theoretical modal analysis of composite structures with symmetrical layer structures of carbon and glass fiber and epoxy resin. They concluded that carbon fiber samples with higher modulus of elasticity than samples produced with glass fiber had higher natural frequencies [27].

Şakar et al., observed the effect of orientation angles and number of layers on the dynamic behavior of the composite structure in the sandwich panels they prepared. The mode shapes and natural frequencies of the produced panels were determined both experimentally and numerically with the ANSYS program [28]. Soni et al., made the most appropriate design study by changing the orientation angles for low displacement and high strength in laminated composite plates. They examined the layering as an angled layer, asymmetrical and symmetrical angled layer [29]. Eryiğit, investigated the effects of hole location and dimensions on lateral critical buckling load in layered composite samples with circular holes, taking into account different fiber orientation angles. He used ANSYS software in his studies [30].

It has been observed in the literature that many studies have been and are still being done for the proper design of laminated composite materials. In most of the studies on composite materials, it has been determined that the modal analysis method is used to examine the behavior of the piece. Composite materials are produced in order to create a more durable component by combining the superior properties of two or more different materials [31, 32]. Composite materials have advantages such as high strength, high rigidity, low weight, high wear resistance, high corrosion resistance, and thermal properties in the desired direction. On the other hand, there are disadvantages such as higher cost, processing difficulties, generally not being recycled, and production difficulty for some composites compared to metals [33-36]. With the development of technology, the usage areas of composite materials are increasing. Composite usage areas can be listed as aviation and defense industry, maritime transport, land transport, space programs, energy sector, infrastructure products, building / construction, sports products, household products, tanks and pressure vessels [37-43].

Composite materials mainly consist of resin and reinforcement components. Continuous or chopped fibers are generally used as reinforcement components. These components do not dissolve or mix in each other. Fiber in the composite materials provides, hardness and durability. The resin material, on the other hand, enables the fiber to form structural integrity, distribute the load between the fibers and protect the fiber from chemical effects and atmospheric conditions [44-46]. The factors determining the properties of a composite product can be listed as the feature of the reinforcing element, the property of the matrix material, the adhesion ability at the fiber-matrix interface, the fiber/matrix ratio, the reinforcement element's geometry and its orientation in the matrix [47-49].

In this study, a long, thin and flexible carbon fiber reinforced composite cover was designed using Unigraphics NX program CAD module. In order to improve the stiffness of this cover, a rib design was made and added to the initial design. Modal analysis of the initial cover design and ribbed cover designs were carried out using the finite element method in the ANSYS program environment. As a result of the work done, natural frequency values were obtained in different mode parameters. As a result of adding rib design to the composite cover, the changes in the natural frequency value of the part were determined and evaluated.

2. Materyal ve Metot

Composite cover design criteria and material information, derivation of natural frequency equations and modal analysis processes with ANSYS software are explained in detail.

2.1. Design Criteria and Material Properties

In this study, piece size is taken as a design constraint. The area to be covered by the cover in the system is required to be limited to at least 354x1710x234 mm (width, length, height). This criterion has been taken into account when designing ribs. The general visual of the composite cover is seen in Figure 2.

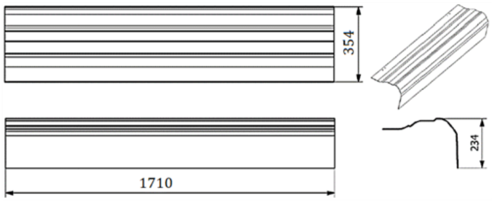

Figure 2. Ribless cover design.

It is aimed to improve the strength and natural frequency values by adding ribs to the composite cover design, which has a long and flexible structure. The cover, which weighed 4.8 kg with its initial design, had a weight of 6.4 kg with the addition of the rib. The visual of the initial and ribbed composite cover design is given in Figure 3.

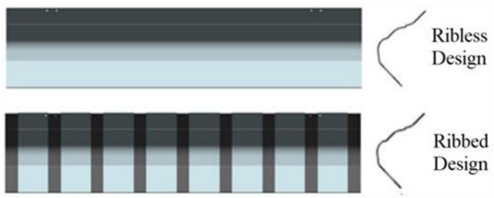

Figure 3. Initial and rib design geometries.

According to the usage conditions of the part, resin-impregnated one-way carbon fiber (prepreg) materials are used to reduce the effect of vibration loads on the part. Different paving angles have been determined in each pavement layer [50]. Composite materials with different characteristics at different laying angles are used. Mechanical properties of the composite material are given in Table 1.

Table 1. Carbon / epoxy prepreg material properties.

Parameter	Unit	Carbon Prepreg
Elasticity Module (0°)	GPa	121
Elasticity Module (90°)	GPa	8.6
Slip modulus	GPa	4.7
Poisson's ratio	-	0.27
Density	g/cc	1.49

In this way, it is aimed to increase the strength. The mechanical properties of the material are given in Table 2. A total of 7 layers were laid, with each layer's thickness as 0.48 mm for ribless design. A total of 14 layers were laid, with each layer's thickness as 0.48 mm for ribbed design. The number of layers and laying directions are shown in Table 2.

Table 2. Number of composite material layers and laying angles.

Layer number	Laying angles
Layer 1 / Layer 8 (0.48 mm)	90°
Layer 2 / Layer 9 (0.48 mm)	-90°
Layer 3 / Layer 10 (0.48 mm)	0°
Layer 4 / Layer 11 (0.48 mm)	45°
Layer 5 / Layer 12 (0.48 mm)	0°
Layer 6 / Layer 13 (0.48 mm)	-90°
Layer 7 / Layer 14 (0.48 mm)	90°

2.1. Vibration and Energy Equations

The operating system of the long and flexible cover is similar to the pendulum system shown in Figure 4 [51]. Therefore, the cover is an inertial element that stores kinetic energy. Since the system can only rotate in the x-axis, it is considered to be one degree of freedom, undamped since there is no damping element, and free vibration since it does not vibrate with external stress.

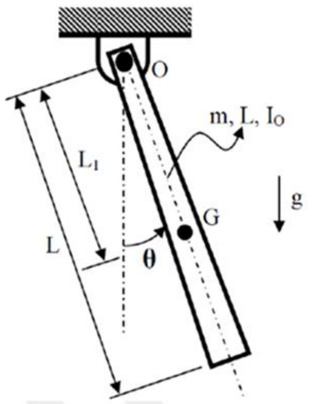

Figure 4. Compound pendulum system.

The equation of motion of this system is expressed by the Lagrange Method. In this method, potential and kinetic energies of the system examined are taken into account. Lagrange expression of the system is equal to the difference between kinetic energy and potential energy [51].

$$L = E_k - E_p \tag{1}$$

By placing the kinetic energy and potential energy difference into the Lagrange equation below, the equation of motion of the system can be obtained.

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \theta}\right) - \frac{\partial L}{\partial \theta} = Q_\theta \tag{2}$$

If the Lagrange equation is elaborated, it is transformed into the form below.

$$\frac{d}{dt}\left(\frac{\partial E_k}{\partial \theta} - \frac{\partial E_p}{\partial \theta}\right) - \frac{\partial E_k}{\partial \theta} - \frac{\partial E_p}{\partial \theta} = Q_\theta \tag{3}$$

Here θ refers to the general coordinate of a system, and Q_θ refers to the sum of forces affecting this coordinate [51]. The equation of motion of the system;

$$E_k = \frac{1}{2} I_o \theta^2 \tag{4}$$

$$E_p = m.g.L_1(1 - \cos\theta) \tag{5}$$

$$\frac{d}{dt}\left(\frac{\partial E_k}{\partial \theta}\right) + \frac{\partial E_p}{\partial \theta} = Q_\theta \tag{6}$$

$$I_o \ddot{\theta} + m.g.L_1.\sin\theta = 0 \tag{7}$$

I_O is the mass moment of inertia with respect to the rotation point of the pendulum. $sin\theta \cong \theta$ can be accepted for small angular displacements,

$$\ddot{\theta} + \frac{m.g.L_1}{I_o}.\theta = 0 \tag{8}$$

The form of free vibrations is pure harmonic.

$$\theta(t) = \theta_o \sin(\omega nt) \tag{9}$$

$$\theta(t) = -\omega 2\theta_o \sin(\omega nt) \tag{10}$$

If the 9th and 10th equations are placed in the equation of motion, the angular acceleration value is found as in equation 11 and the natural frequency value in equation 12.

$$\omega_n = \sqrt{\frac{m.g.L_1}{I_o}}\left(\frac{rad}{s}\right) \tag{11}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{m.g.L_1}{I_o}}(Hz) \tag{12}$$

This formula is used to support the theoretical modal analysis results of the cover with long and flexible design. As seen in the formula, the natural frequency and the mass moment of inertia are inversely proportional. The biggest factor affecting the natural frequency change is seen as mass.

2.3. Modal Analysis

Modal analysis is a method used to reveal the natural frequency, damping and mode shape values of materials. The operating system of the long and flexible cover is shown in Figure 5. Thanks to the hinge attached to the bolt holes at the two ends of the part, it can be opened at a

90° angle with the user force on the x axis and returns to its starting position with the user force. The weapon protective composite cover is mostly used in the starting position in the armored vehicle, and the cover is opened when necessary.

Figure 5. Top view of the composite cover.

The finite element model of the composite cover was created in the ANSYS program. The protective cover designed with the NX CAD program was transferred to the ANSYS program with STEP extension. In the next step, mesh is applied to the model. As the mesh type applied, surface and element shape are predominantly quadrilateral. Figure 6 shows tiling directions and tiling angles on the shell structure of the part.

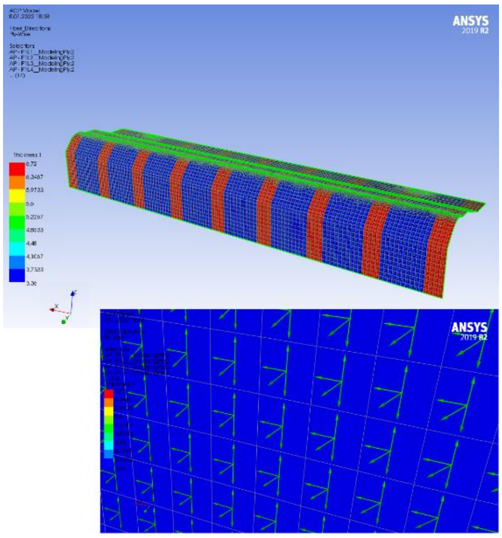

Figure 6. Finite element model of a long and flexible composite cover.

Network information in the model is given in Table 3.

Table 3. Number of elements and nodes of the model.

	Modal Analysis
Elements Number	55156
Nodes Number	113919

Figure 7 shows mesh quality graph (element quality and aspect ratio) of the part.

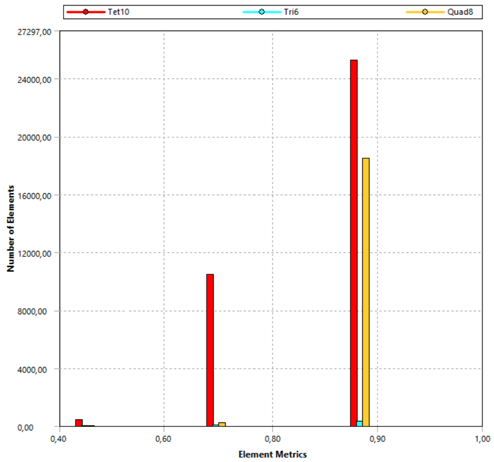

Figure 7. Mesh quality graph.

The cover and the hinge movable lug are defined as fixed joint and can be seen in Figure 8.

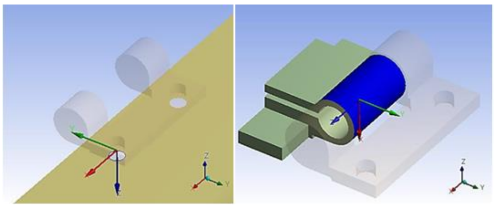

Figure 8. Fixed surface, hinge and cover connection.

It is connected as a fixed joint between the movable hinge part and the fixed hinge part. Analysis boundary conditions are defined by fixing two fixed hinge lug bases (fixed support). After these processes, a modal analysis was performed to give the part the first six modes. No force is applied to the part and it is considered to be vibrating freely.

3. Result and Discussion

Modal analyzes of the initial and rib design were carried out. The effects of the cover on different mode natural frequencies were observed in the composite structure with 7 layers in the non-ribbed and a total of 14 layers in the ribbed section and the analysis results were evaluated. Graphs were created with the analysis results obtained for different modes. The analysis results of the initial design and the rib design were examined in detail separately. The effect of adding ribs to the initial design on modal behavior was evaluated.

The first 6 mode values obtained as a result of the modal analysis using ANSYS software for the initial cover design with a ribless structure are given in Figure 9. According to the analysis results, mode 1 natural frequency was found to be 4.603 Hz, mode 2 natural frequency was 7.478 Hz, mode 3 natural frequency was 24.65 Hz, mode 4 natural frequency was 31.067 Hz, mode 5 natural frequency was 39.467 Hz and mode 6 natural frequency was 50.002 Hz.

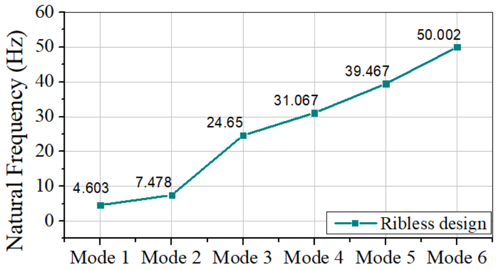

Figure 9. Modal analysis results of the ribless design.

The results of the modal analysis using ANSYS software for the long and flexible composite cover without ribs are shown in Figure 10. For this design, analysis for six different modes as 1st mode (a), 2nd mode (b), 3rd mode (c), 4th mode (d), 5th mode (e) and 6th mode (f) results have been obtained.

In Figure 10, in the 1st mode and 3rd mode parts, it gave the reaction that can be defined as bending. In this mode, the greatest deformation has been observed at the farthest point (colored in red on Fig. 10) of the part to the rotation axis. The part in the form of the 2nd mode gave the reaction that can be described as the first torsion. In this mode, the greatest deformation has been observed at the farthest point (colored in red on Fig. 10) of the rotation axis of the part. In the form of the 3rd mode, the part gave the reaction that can be described as the second bending. In this mode, the deformation is more unevenly distributed compared to the first bending case. In this mode, the largest deformation has been observed in the part that locates at the farthest from the rotation axis and in the middle (colored in red on Fig. 10). The part in the form of the 4th mode gave the reaction that can be described as the second torsion. In this mode, the deformation is more unevenly distributed than in the first torsion case. In this mode, the greatest deformation has been observed in the farthest and end regions (red colored area) of the part. In the 5th mode shape, the part produced a mixture of bending and torsion. In this mode, the greatest deformation has been observed in the farthest and end regions (colored in red on Fig. 10) of the part. In the 6th mode, the part produced a mixture of bending and torsion. In this mode, the most distant to the rotation axis of the part and the largest deformation in the middle (colored in red on Fig. 10) has been observed.

Modal analysis results of ribbed cover design are given in the graphic in Figure 11. In the analysis, natural frequency values were obtained for the first 6 modes. According to the analysis results, mode 1 natural frequency is 6.236 Hz, mode 2 natural frequency is 10.379 Hz, mode 3 natural frequency is 38.964 Hz, mode 4 natural frequency is 71.177 Hz, mode 5 natural frequency is 79.072 Hz and mode 6 natural frequency is 103.33 Hz.

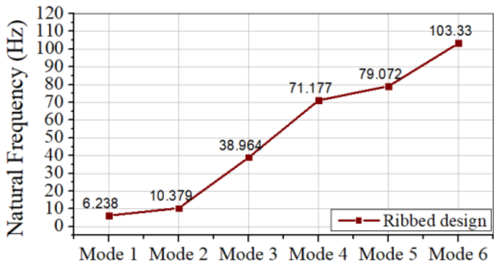

Figure 11. Ribbed composite cover geometry modal analysis results.

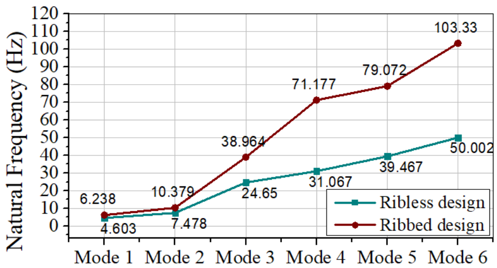

Figure 12. Modal analysis results for ribless and ribbed design.

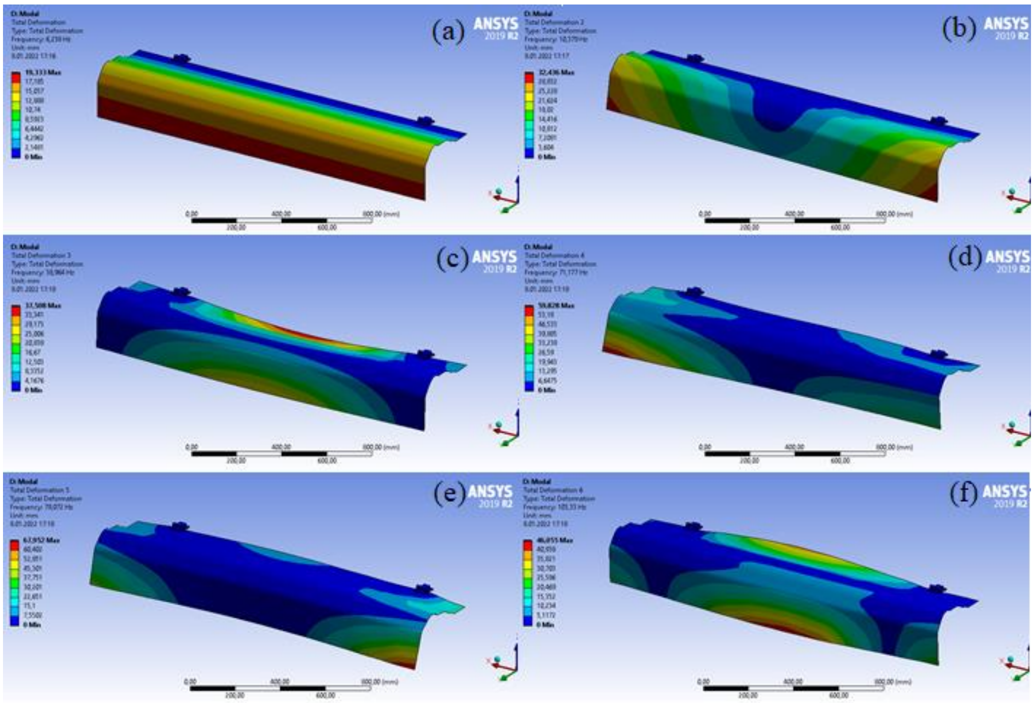

Figure 10. Modal analysis results of the ribless cover design in different modes.

The modal analysis results of the ribless and ribbed cover designs are shown in the graphic in Figure 12. It was observed that when ribs were added to the initial design, improvements were achieved by 39.61%, 20.87%, 39.99%, 22.96% and 26.5% in the natural frequency values of mode 2, mode 3, mode 4, mode 5 and mode 6, respectively.

According to the results of the modal analysis, participation values above 5% are indicated in bold and are shown in Table 4 below.

Table 4. Modal Analysis - Ratio of Effective Mass to Total Mass (Over 5% Participation Value).

Mod	Frequency	X	Y	Z	Rot X	Rot Y	Rot Z
1	6,238	0,30E-07	0,55E-02	**0,710315**	**0,299631**	**0,511746**	0,38E-02
2	10,379	**0,52E-01**	0,61E-04	0,95E-04	0,17E-04	**0,146166**	0,12E-07
3	38,964	0,54E-05	0,102008	**0,137992**	0,21E-01	**0,96E-01**	**0,76E-01**
4	71,177	0,85E-04	0,14E-01	0,26E-02	0,73E-02	0,45E-01	**0,112465**
5	79,072	0,26E-03	**0,91E-01**	0,39E-01	0,36E-01	0,11E-01	0,38E-01
6	103,33	0,11E-05	**0,78E-01**	0,10E-01	**0,73E-01**	0,78E-02	**0,51E-01**

4. Conclusions

In this study, the effect of adding ribs to a long and flexible composite cover on natural frequency values was investigated. Modal analyzes of ribbed and ribless designs were carried out in ANSYS environment. In the modal analysis process, the solution was realized for six different modes of ribbed and ribless designs. Following results were obtained:

•For the ribless design, mode 1 natural frequency is 4.603 Hz, mode 2 natural frequency is 7.478 Hz, mode 3 natural frequency is 24.65 Hz, mode 4 natural frequency is 31.067 Hz, mode 5 natural frequency is 39.467 Hz and mode 6 natural frequency is 50.002 Hz.

•The bending and torsion values for different modes of ribless composite cover design have been studied in detail and regionally. The obtained analysis results oriented the rib design, and it was aimed to improve the natural frequency values of the ribbed cover.

•For ribbed design, mode 1 natural frequency is 6.238 Hz, mode 2 natural frequency is 10.379 Hz, mode 3 natural frequency is 38.964 Hz, mode 4 natural frequency is 71.177 Hz, mode 5 natural frequency is 79.072 Hz and mode 6 natural frequency is 103.33 Hz.

•When ribs were added to the initial design, it was observed that the natural frequency values of mode 1, mode 2, mode 3, mode 4, mode 5 and mode 6 improved by %35.52, % 38.79,% 58.07,% 129.11,% 100.35 and % 106.65, respectively.

•It is seen in the derived equations that as the value of mass, which is the most effective parameter of natural frequency, increases, the natural frequency value should also increase. With the increase in the mass value of the cover design handled in the study, an increase was observed in the natural frequency value, too.

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