Optimal Design Techniques of Composite Payload Adapter for a Typical Launch Vehicle

V. Pavithra and Gangadhar Ramtekkar

Abstract Composite materials made with the objective of getting a more desirable combination of properties are extensively used in weight-sensitive structures due to its very high strength to weight ratio and relatively high stiffness to weight ratio. A payload adapter forms an interface between the payload or the satellite and the launch vehicle core. The importance of weight savings in the payload adapter is that any reduction in its mass can help in a corresponding increase in the satellite mass since the sensitivity is 1:1. This is due to the positioning of the payload adapter near the satellite. This paper deals with the optimal design of a lightweight Composite Payload Adapter in three different configurations with a maximum mass advantage which at the same time should be able to withstand the loads acting on them during the flight. The design options studied were: monocoque, stringer-stiffened, and sandwich-structured construction. In monocoque construction, a metallic skinned structure made out of aluminium and also layered composite skinned structure made out of M55J/M18 prepreg laminates are considered for the study. The sandwich constructions are studied with metallic face sheets and layered composite face sheets in combination with a hexagonal aluminium honeycomb core. The study of stringer-stiffened construction was conducted by comparing the structure having the stringers and the shell made of aluminium with that made of M55J/M18 laminates. The optimum design out of these cases studied was arrived at. Static, buckling and free vibration analyses of all the cases were carried out using the general-purpose finite element software MSC. NASTRAN.

Keywords Composite · Laminate · Monocoque · Stringer-stiffened · Sandwich · Payload adapter

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1 Introduction

Any design which satisfies a set of design specifications while providing the best quality and performance is called an optimal design. It involves the determination of the design parameters which controls the shape, material properties and dimensions of the structure [1]. Thus, the theory of optimal design and its implications have a key role in the design of weight-sensitive applications like aerospace structures. The existing experience shows that direct substitution of carbon-epoxy composites for traditional ring-stiffened aluminium aerospace-framed structures results in usually 10–20% weight reduction accompanied by considerable cost savings [2]. There is a strong need in the aerospace industry to reduce the cost and improve the effectiveness of launching satellites into the orbit. The primary objective of a launch vehicle is to place as much as payload as possible into earth orbit at the lowest cost. Minimising the structural mass is crucial in many ways—it helps to increase payload mass and thus reduce the launch cost [3]. The objective of structural design is to achieve minimum weight with maximum safety so as to increase the economy of a space mission, at the same time, it should be able to withstand the loads acting on them during the flight. One of the ways to reduce the mass of the launch vehicle is to use composites in its components.

So, this paper deals with the design and analysis of a lightweight Composite Payload Adapter (CPLA). Figure 1 shows the exploded view of a launch vehicle showing a payload adapter (PLA). Payloads, such as spacecraft, which are mounted on launch vehicles, are subject to severe vibrations during flight [4]. The payload adaptor must have high stiffness and a high natural frequency in order to prevent dynamic coupling between the satellite and the launch vehicle dynamics. Detailed ground tests, static and dynamic, are performed to make sure its structural safety.

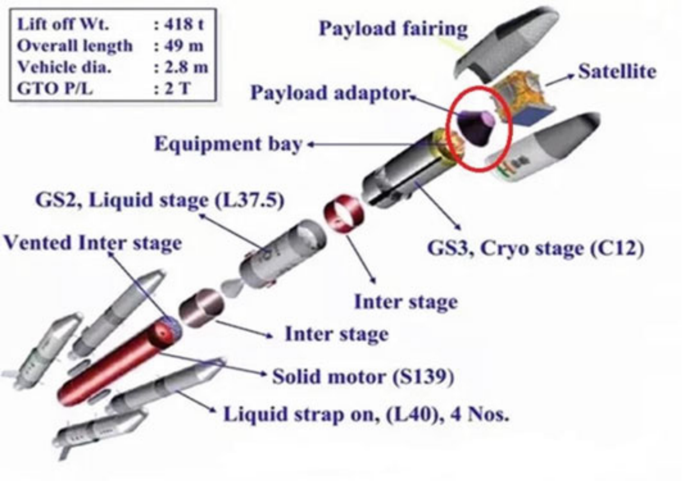


Fig. 1 Exploded view of a launch vehicle. *Source* https://www.quora.com

The PLA is a primary structure of a launcher providing the interface to the spacecraft [5, 6]. It is designed to support the payload during flight, transmit any loads induced by it back to the launch vehicle and facilitate power and signal connections between the spacecraft and the payload (United Launch Alliance, 2013). Optimization of this structure is of much importance as it is positioned towards the top of the launch vehicle because any mass saved in the PLA leads to a corresponding increase in the satellite mass in the ratio 1:1. In [7], the design of the monocoque and stringer-stiffened structure of PLA has been discussed. This paper extends the study by comparing it with the sandwich-structured PLA, which is a very lightweight design method.

2 Structural Configuration of PLA

The functional requirement of the PLA involves connecting the satellite and the launch vehicle of two different diameters. Hence, the PLA is to be designed in the shape of a truncated cone. The structure has a diameter of 937 mm at the fore end and 1370 mm at the aft end (Fig. 2). The total height of the structure is 1040 mm. Two metallic rings attached to the cone at the either ends act as interface attachment rings. The fore end joins with the satellite and the aft end flange with the launch vehicle core. The structure is to be designed in such a way that it is safe in strength, stiffness and buckling consideration also should have the least possible mass. Thus, optimization by investigating the best possible shape and material orientation of the three chosen configurations, namely, monocoque, stringer-stiffened and sandwich structure is performed.

3 Design Specifications

A structure is designed in such a way that it is able to take the design loads for its entire life span for which it is designed. In the case of a launch vehicle structure, it should be able to take the loads acting on them during the entire journey of the flight. Table 1 gives the design loads for PLA and Table 2 gives the satellite mass and inertia. A design safety factor of 1.25 shall be provided over the design load. The stiffness constraint given is that the lateral and axial frequency of PLA in base fixed configuration (Aft end of PLA has all D.O.F. arrested) with satellite mass and inertia simulated shall not be less than 15 Hz and 30 Hz, respectively [7].

This is done to avoid any chances of dynamic coupling because, in case of such a vibration, the dynamic response of the launch vehicle becomes the input to the satellite. So if their frequencies are not well-separated, their transmissibility values can couple resulting in rapid fatigue failures [8]. So the natural frequencies of both the satellite and the launch vehicle must be separated with an octave apart (means to double) to prevent coupling, i.e. the frequencies are separated by a ratio of more

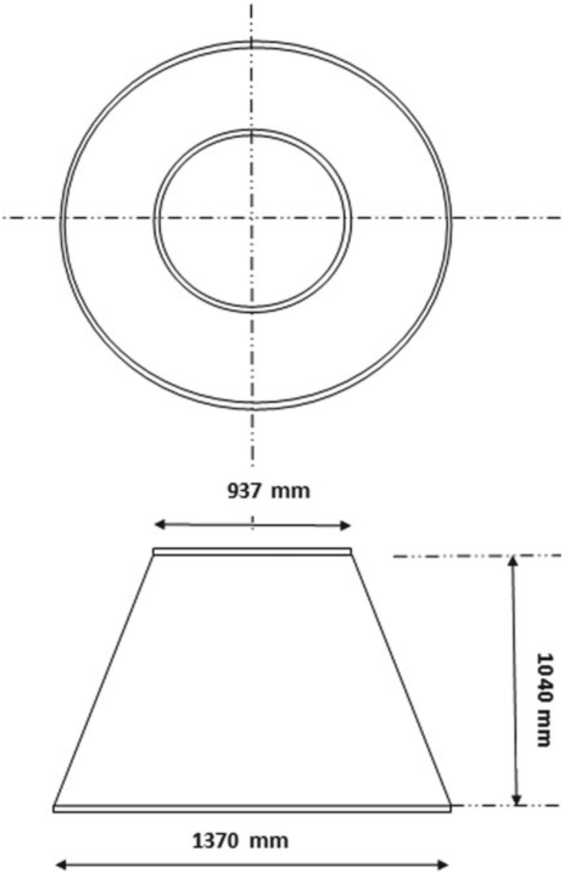


Fig. 2 Configuration of payload adapter

Table 1 Design loads on PLA

Mass of the payload (Satellite)	3500 kg
Axial load	300.43 kN
Lateral load	128.76 kN
Equivalent axial load (E.A.L.)	1255.3 kN
Bending Moment	327.04 kNm

Source Pavithra et al. [7]

Table 2 Satellite details

Mass of the satellite	3500 kg
Height of satellite C.G. above its base	1.5 m
Moment of inertia about axial direction	1500 kg m^2
Moment of inertia about lateral direction	3000 kg m^2

Source Pavithra et al. [7]

than 2:1. Here, the PLA is designed for a launch vehicle with a fundamental natural frequency of vibration as 7.5 Hz in the lateral mode and 15 Hz in the axial mode. Hence, the above frequency constraints must be satisfied to be safe in free vibration.

In the case of a buckling analysis, the buckling load factor with a knockdown factor (K.D.F) of 0.6 and a design safety factor of 1.25 should be greater than one [7]. The K.D.F. is applied to compensate for the material and fabrication imperfections. Here, the buckling load factor is the ratio of the critical buckling load to the applied load and when this ratio is greater than 1, the structure is safe in buckling.

The strength constraint is satisfied if the maximum stress generated in the structure is within the yield strength value of the material. A structure which satisfies all the three design constraints and has a mass within 25 kg can be taken as the optimum structure.

$$\text{E.A.L.} = \text{Axial Load} \pm \frac{2M}{R} \tag{1}$$

4 Construction Methodologies Adopted

The design strategy adopted here is design through analysis. It is basically an optimization method by mass minimization technique. Three configurations studied are monocoque, stringer-stiffened and sandwich structure. For each case, three types of analysis are carried out, i.e. static, buckling and free vibration. The best laminate sequence for structural stability is arrived at through an iterative process.

The stiffness of the structure is checked by conducting free vibration analysis after simulating the satellite mass and inertia at the fore end of the payload adapter as the frequency is directly related to the stiffness. The buckling strength is verified from the buckling analysis by applying the E.A.L. [Eq. (1)] at the fore end nodes of the PLA. The maximum stress and deformation in the structure is determined after carrying out static analysis.

4.1 Monocoque Construction

Different options of monocoque construction, which is basically a single shell structure, are attempted. The first option considered is with a metallic skin alone made out of aluminium, the thickness of which is fixed by an iterative process. The thickness is varied starting from 1 mm. The second option considered is with a layered composite made of M55J/M18 prepreg laminates which are a laminate made of carbon fibres and epoxy resin, each layer being 0.1 mm thick. The optimum number of layers and ply orientation for the layered composite is arrived by an iterative process. The material properties of aluminium and M55J/M18 prepreg are given in Tables 3 and 4.

Table 3 Material properties of aluminium sheet

Density of the material	2700 kg/m^3
Elastic modulus	68.9 GPa
Poisson's ratio	0.33
Shear modulus	25.89 GPa

Table 4 Material properties of M55J/M18 Prepreg laminate

Longitudinal elastic modulus (E_1)	264.78 GPa
Transverse elastic modulus (E_2)	5.952 GPa
Poisson's ratio	0.346
Shear modulus ($G_{12} = G_{13} = G_{23}$)	4.415 GPa
Density of the material	1750 kg/m^3
Longitudinal tensile strength	1326.9 MPa
Longitudinal compressive strength	724 MPa
Transverse tensile strength	21.78 MPa
Transverse compressive strength	117.43 MPa
In-plane shear strength	74.85 MPa

4.2 Stringer-Stiffened Construction

In stringer-stiffened structure which consists of a thin shell stiffened by uniformly spaced longitudinal stringers, both aluminium and laminate version was tried. In this first case, both the stringers and the shell were made of aluminium. Here, the number of stringers and the thickness of the shell were iteratively changed to reach the minimum mass structure. Later, it was replaced with laminate and different thickness and layup of the laminate layers were tried to reach the optimum design. The stringers are provided with a hat section whose cross-sectional details are as shown in Fig. 3.

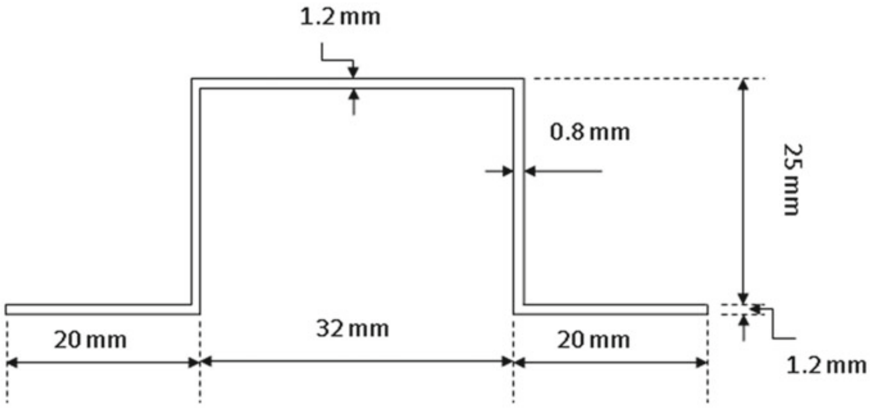


Fig. 3 Cross-sectional dimensions of the stringer. *Source* Pavithra et al. [7]

Table 5 Material properties of hexagonal aluminium honeycomb core

Density	36 kg/m^3
Transverse shear modulus	225.6 MPa
Compressive strength	2.1 MPa
Compressive modulus	540 MPa

4.3 Sandwich-Structured Construction

Two types of sandwich construction were studied—one with an aluminium sheet as the skin and the other with composite skin of M55J/M18 laminates. Hexagonal aluminium honeycomb was used as the core for both. The properties of aluminium honeycomb core are given in Table 5.

5 Finite Element Modelling

A finite element model of the PLA structure was created using MSC PATRAN. The geometry of PLA in the form of a truncated cone is obtained. 2-D shell elements and 1-D beam elements were used to model the structures. The discretized model of monocoque PLA with a rigid link attached on the top is given in Fig. 4 and the finite element models of stringer-stiffened and sandwich PLA are given in Figs. 5 and 6. MSC. PATRAN was used as the preprocessor and the post-processor and MSC. NASTRAN was used as the solver.

6 Finite Element Analysis

The boundary conditions for different analysis were given and the corresponding loads were also applied as in Table 1. Static, free vibration and buckling analyses were performed using MSC. NASTRAN [9].

6.1 Free Vibration Analysis

In the free vibration analysis, all the aft end nodes are fixed, i.e. all the degrees of freedom of the nodes are arrested and the fore end nodes are rigidly linked to the centre of gravity (C.G.) of the satellite as in Fig. 4. A mass of 3500 kg is mounted at the C.G. of the satellite. Here, mode shapes and the corresponding frequencies are obtained as the output.

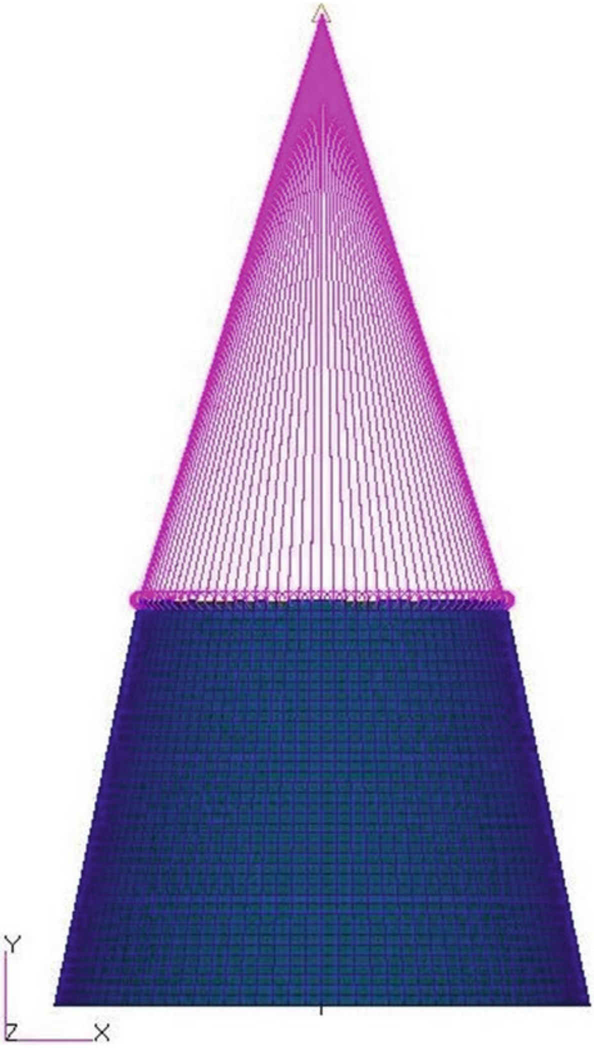


Fig. 4 Discretized model of monocoque PLA with rigid link mounted on the top. *Source* Pavithra et al. [7]

6.2 Static Analysis

Static analysis was carried out to estimate the maximum stress acting on the structure under the applied design loads. It also gives the maximum displacement in the structure under a set of given loads. In this analysis also, the aft end nodes are completely restrained and the fore end nodes are rigidly linked to a single master node at the satellite C.G. as in Fig. 4. The design loads are then applied at this master node. The stresses in each layer of the composite laminate were obtained.

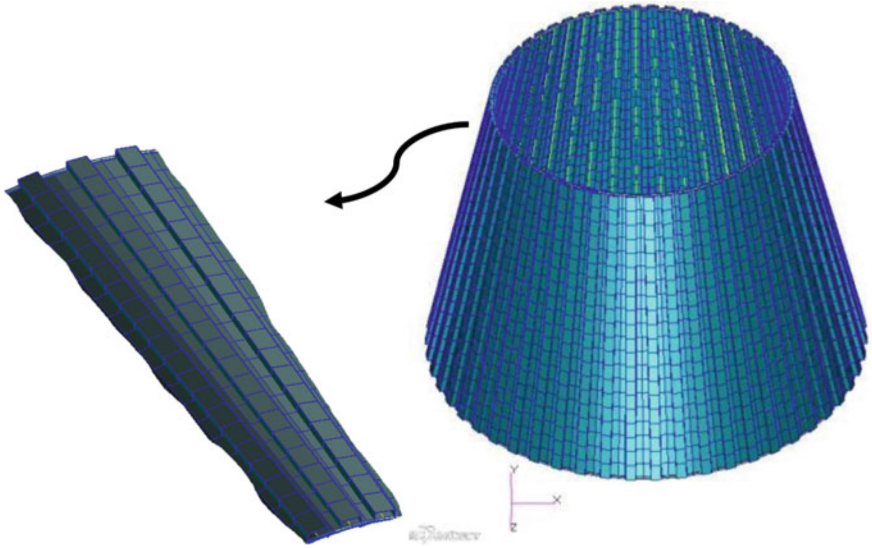


Fig. 5 Finite element model of stringer-stiffened PLA. *Source* Pavithra et al. [7]

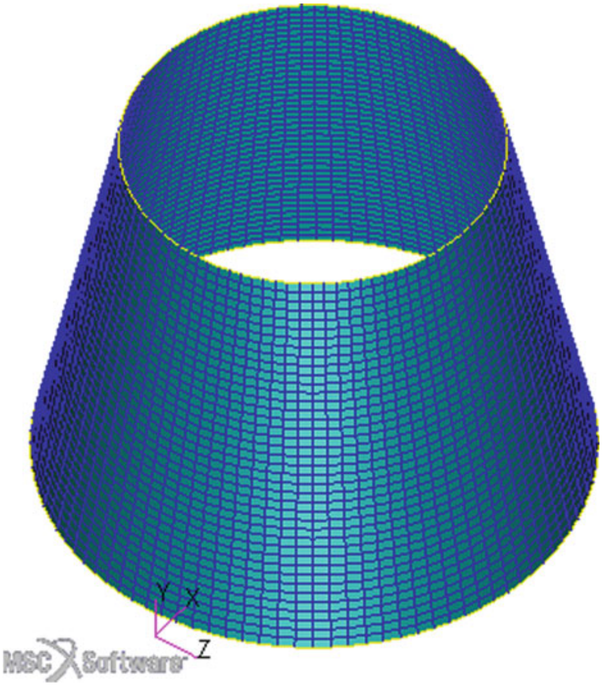


Fig. 6 Finite element model of sandwich-structured PLA

6.3 *Buckling Analysis*

Buckling analysis is done by applying E.A.L. uniformly on all the nodes at the fore end. In the analysis, the buckling load factor is obtained. These factors are the eigenvalues of the instability problem, and when multiplied by the external loads give the buckling condition for the system. To ensure that any imperfection in the laminated shell will have no essential influence on the buckling load, K.D.F. is applied.

7 Optimum Design and Results

The optimum structure of PLA in each of the three configurations studied was found out. It was found that composite structures weigh less compared to the metallic ones, which equally satisfy the design constraints. Among the three, sandwich- and stringer-stiffened weigh less compared to the monocoque construction. The free vibration and buckling analysis results which also satisfies the strength constraints are tabulated, and the contour plots are also shown in all the cases studied.

7.1 *Monocoque Design*

Metallic Monocoque Structure. In the monocoque structure made of aluminium sheet, the thickness of the skin is varied from 1 mm and it is found that a minimum of 6 mm-thick aluminium sheet is required for satisfying the design constraints. Then, the mass of the structure obtained is 62.35 kg. The results of free vibration and buckling analysis are shown in Tables 6 and 7. The free vibration modes and the first buckling mode of the structure are shown in Figs. 7 and 8.

Here, the minimum frequency in the lateral mode of free vibration is obtained as 18.15 Hz and in the axial mode is 98.48 Hz. The buckling load factor after applying

Table 6 Results of free vibration analysis of metallic monocoque structure

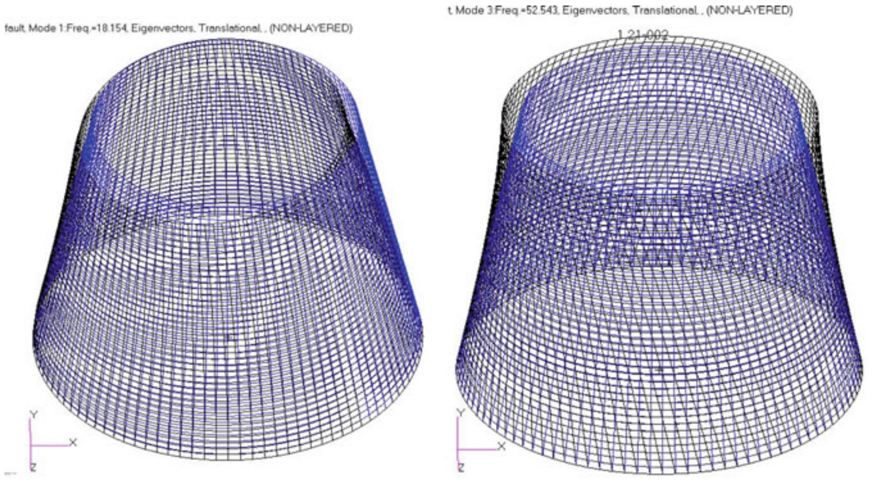
Modes	Natural frequency (Hz)	Remarks
Mode 1	18.15	Lateral mode
Mode 2	18.15	Lateral mode
Mode 3	52.54	Torsional mode
Mode 4	95.40	Lateral mode
Mode 5	95.40	Lateral mode
Mode 6	98.48	Axial mode
Mode 7	245.80	Shell mode
Mode 8	245.80	Shell mode
Mode 9	257.38	Shell mode
Mode 10	257.38	Shell mode

Table 7 Results of buckling analysis of metallic monocoque structure

Modes	Buckling load factor		
	Load factor	Applying K.D.F.	Applying F.O.S.
Mode 1	2.17	1.74	1.04
Mode 2	2.17	1.74	1.04
Mode 3	2.65	2.12	1.27
Mode 4	2.65	2.12	1.27
Mode 5	3.42	2.73	1.64
Mode 6	3.42	2.73	1.64
Mode 7	3.60	2.88	1.73
Mode 8	3.60	2.88	1.73
Mode 9	4.34	3.47	2.08
Mode 10	4.34	3.47	2.08

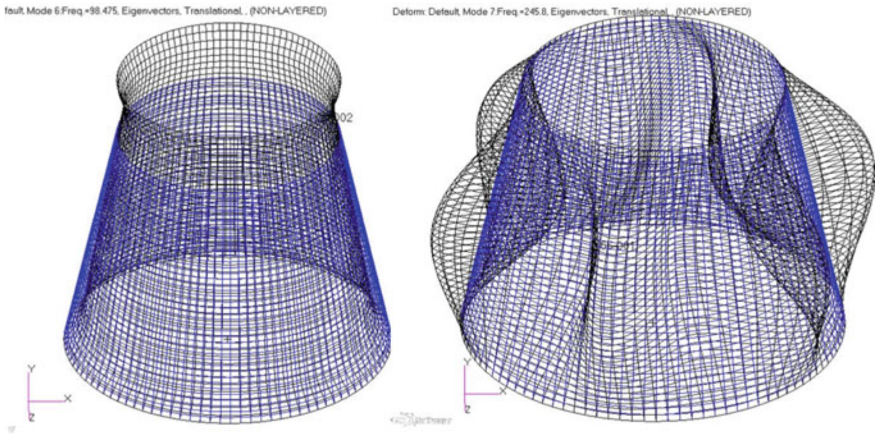
F.O.S. and K.D.F. is obtained as 1.04 and the stresses obtained are also less than the strength of the material. Here, the maximum stress (Fig. 9) acting on the structure is less than the strength of the material.

Composite Monocoque Structure. In the monocoque structure made of M55J/M18 prepreg laminates, the number of layers as well as the orientation was iterated to arrive at the optimum design. The optimum design was arrived at by using 45 laminate layers each 0.1 mm thick, in the following orientation: [0/0/90/90/0/90/0/-90/0/45/0/-45/0/90/0/-90/0/-45/45/0/90/0/45/0/90/0/45/-45/0/-90/0/90/0/-45/0/45/0/-90/0/90/0/90/90/0/0]. The mass of the PLA designed



(a) Lateral Mode (b) Torsional Mode

Fig. 7 Free Vibration modes of metallic monocoque structure



(c) Axial Mode (d) Shell Mode

Fig. 7 (continued)

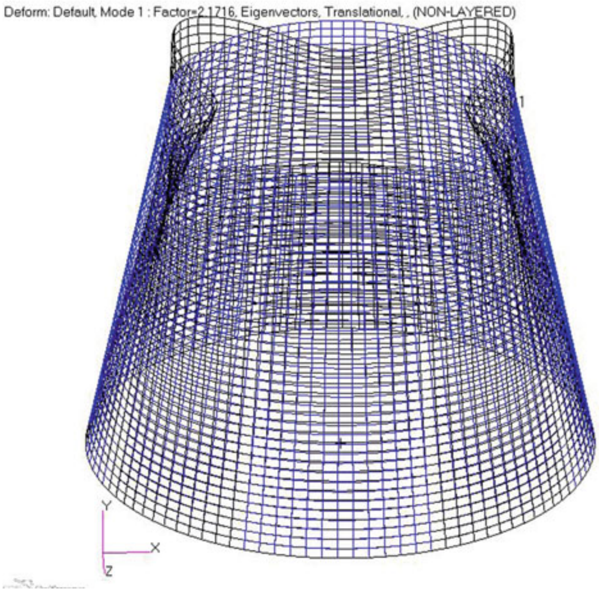


Fig. 8 First Buckling mode of metallic monocoque structure. *Source* Pavithra et al. [7]

was 30.31 kg. Even though the mass of composite monocoque PLA is almost 50% less compared to that of the metallic one, it couldn't meet the design mass constraints. The results of free vibration and buckling analysis are shown in Tables 8 and 9. The free vibration modes and the first buckling mode of the structure are shown in Figs. 10 and 11.

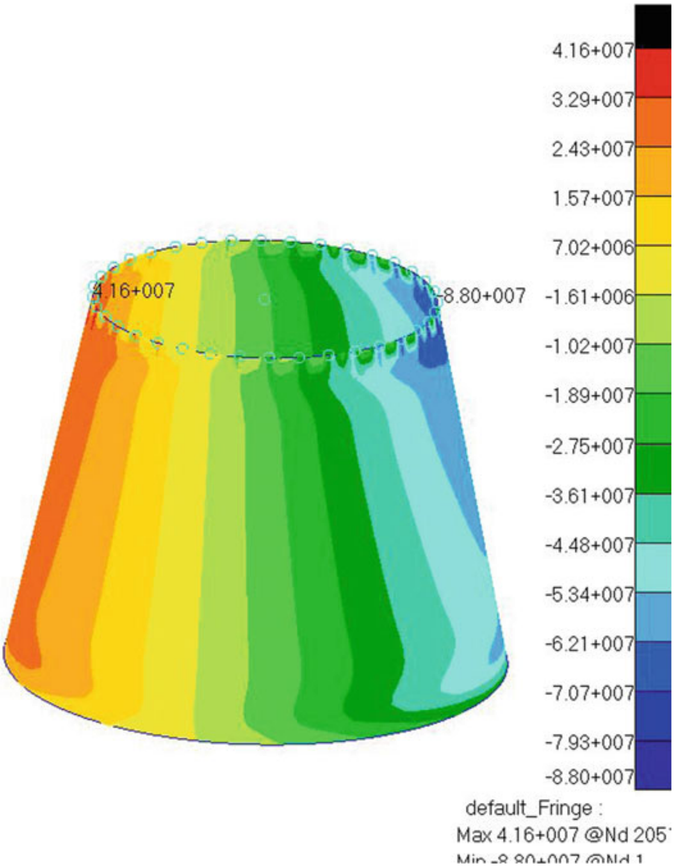


Fig. 9 Maximum stress distribution of the metallic monocoque PLA

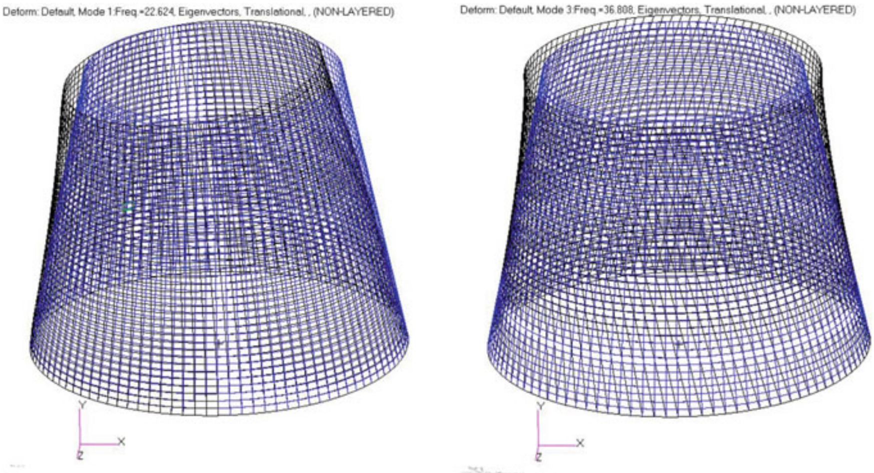
Table 8 Results of free vibration analysis of composite monocoque structure

Modes	Natural frequency (Hz)	Remarks
Mode 1	22.62	Lateral mode
Mode 2	22.62	Lateral mode
Mode 3	36.81	Torsional mode
Mode 4	69.43	Lateral mode
Mode 5	69.43	Lateral mode
Mode 6	123.04	Axial mode
Mode 7	278.70	Shell mode
Mode 8	278.70	Shell mode
Mode 9	291.26	Shell mode
Mode 10	291.26	Shell mode

Table 9 Results of buckling analysis of composite monocoque structure

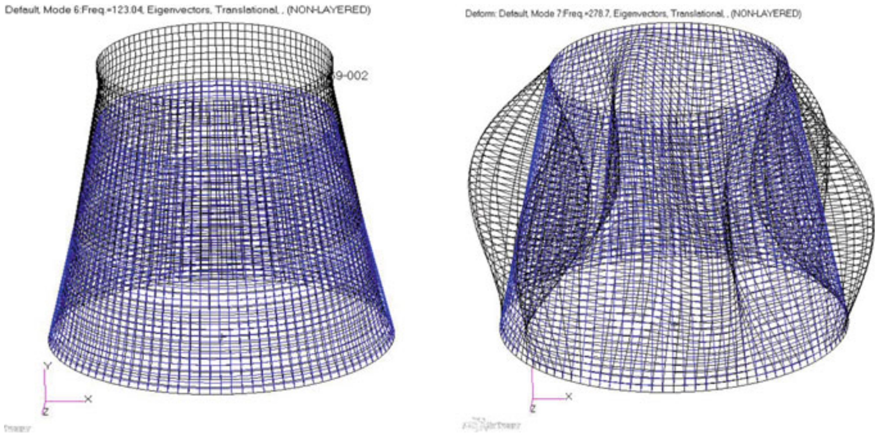
Modes	Buckling load factor		
	Load factor	Applying K.D.F.	Applying F.O.S.
Mode 1	2.08	1.67	1.01
Mode 2	2.08	1.67	1.01
Mode 3	2.39	1.91	1.15
Mode 4	2.39	1.91	1.15
Mode 5	3.03	2.43	1.46
Mode 6	3.03	2.43	1.46
Mode 7	3.70	2.96	1.77
Mode 8	3.70	2.96	1.77
Mode 9	3.70	2.96	1.78
Mode 10	3.70	2.96	1.78

Here also, the lateral and axial mode of free vibration have frequencies more than 15 Hz and 30 Hz, respectively, and the buckling load factor obtained is also greater than 1. The stresses obtained in all the 45 layers of the laminated shell are within the specified limit (Fig. 12). But, the mass of the structure is more than 25 kg.



(a) Lateral Mode (b)TorsionaMode

Fig. 10 Free vibration modes of composite monocoque structure



(c) Axial Mode (d) Shell Mode

Fig. 10 (continued)

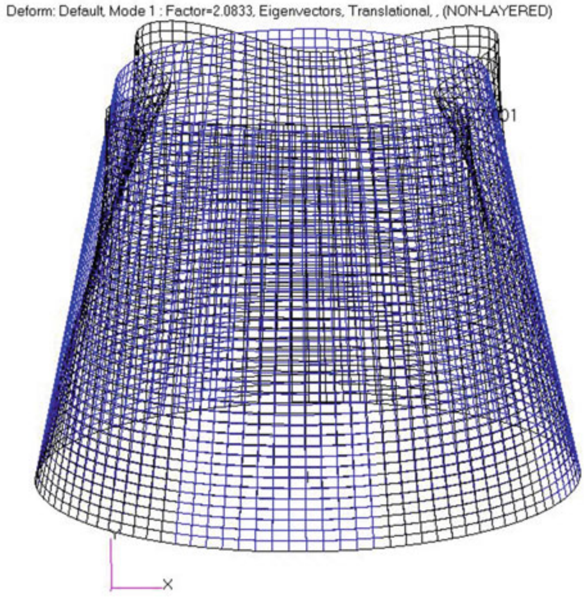


Fig. 11 First buckling mode of composite monocoque structure

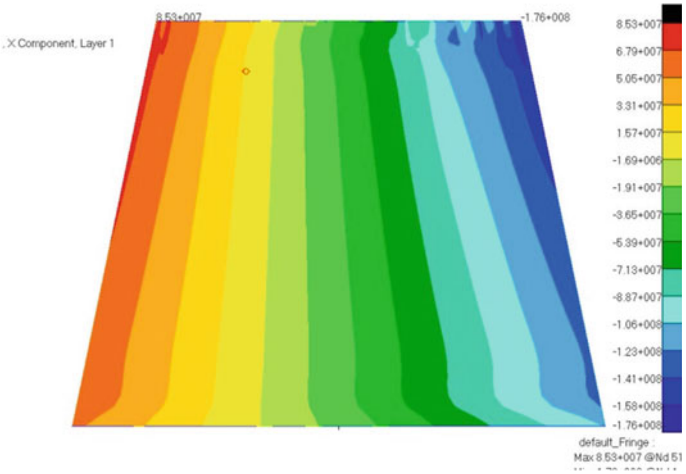


Fig. 12 Stress distribution of layer 1 of laminated monocoque PLA

7.2 Stringer-Stiffened Design

Metallic Stringer-Stiffened Structure. In the stringer-stiffened structure made of aluminium sheet, the thickness of the shell is varied from 1 mm and it is found that a minimum of 4 mm-thick aluminium sheet stiffened by 48 numbers of stringers are required for satisfying the design constraints. Then, the mass of the structure obtained is 55.10 kg. The results of free vibration and buckling analysis are shown in Tables 10 and 11. The free vibration modes and the first buckling mode of the structure are shown in Figs. 13 and 14.

Here, the minimum frequency in the lateral mode of free vibration is obtained as 16.77 Hz and in the axial mode is 93.11 Hz. The buckling load factor after applying F.O.S. and K.D.F. is obtained as 1.03 and the stresses obtained are also less than the

Table 10 Results of free vibration analysis of metallic stringer-stiffened structure

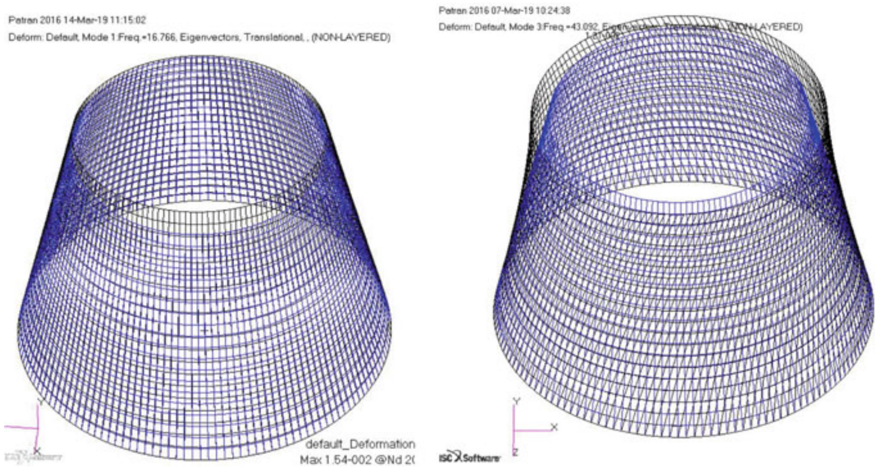
Modes	Natural frequency (Hz)	Remarks
Mode 1	16.77	Lateral mode
Mode 2	16.77	Lateral mode
Mode 3	43.09	Torsional mode
Mode 4	79.08	Lateral mode
Mode 5	79.08	Lateral mode
Mode 6	93.11	Axial mode
Mode 7	292.59	Shell mode
Mode 8	292.59	Shell mode
Mode 9	293.34	Shell mode
Mode 10	293.34	Shell mode

Table 11 Results of buckling analysis of metallic stringer-stiffened structure

Modes	Buckling load factor		
	Load factor	Applying K.D.F.	Applying F.O.S.
Mode 1	2.15	1.72	1.03
Mode 2	2.15	1.72	1.03
Mode 3	2.82	2.26	1.35
Mode 4	2.82	2.26	1.35
Mode 5	3.07	2.46	1.47
Mode 6	3.07	2.46	1.47
Mode 7	3.85	3.08	1.85
Mode 8	3.85	3.08	1.85
Mode 9	4.93	3.95	2.37
Mode 10	4.93	3.95	2.37

strength of the material. The maximum stress (Fig. 15) acting on the structure is also less than the strength of the material. But, the mass of the structure is much more than the limit.

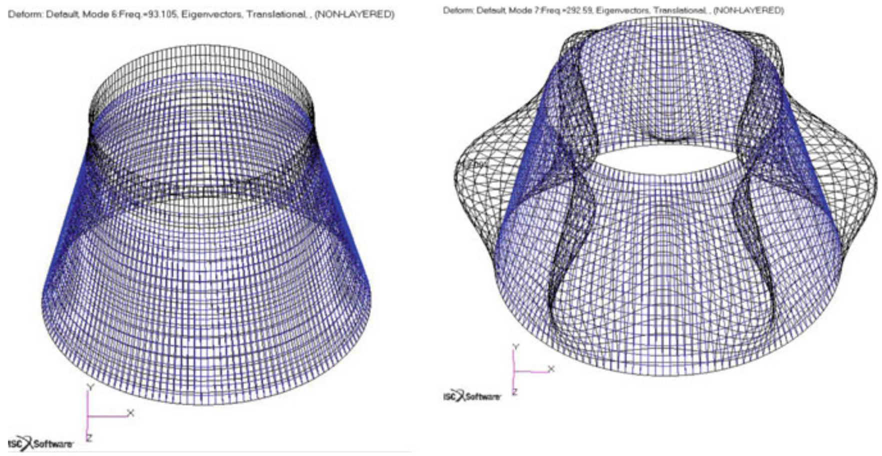
Composite Stringer-Stiffened Structure. In the stringer-stiffened structure made of M55J/M18 prepreg laminates, the number of layers as well as the orientation was iterated to arrive at the optimum design. The optimum design was arrived at by using 1.5 mm-thick laminated shell stiffened uniformly by using 64 numbers of laminated stringers. The optimum orientation of the laminated shell



(a) Lateral Mode

(b) Torsional Mode

Fig. 13 Free vibration modes of metallic stringer-stiffened structure



(c) Axial Mode

(d) Shell Mode

Fig. 13 (continued)

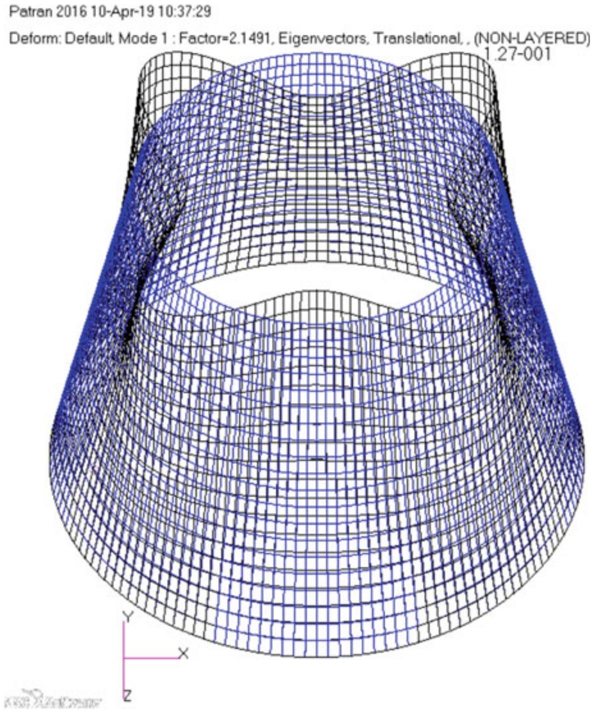


Fig. 14 First buckling mode of metallic stringer-stiffened structure. *Source* Pavithra et al. [7]

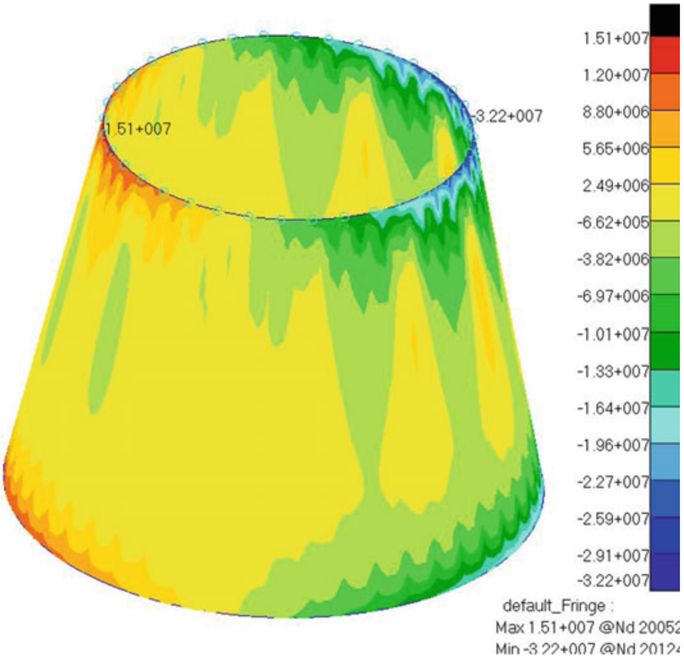


Fig. 15 Stress distribution of metallic stringer-stiffened PLA

is [0/-60/0/-90/-90/90/60/0/60/90/-90/-90/0/-60/0] and that of the stringers is [0/-45/0/45/0/90/90/0/45/0/-45/0]. The mass of the PLA designed was 21.79 kg. It is found that the mass of composite monocoque PLA is much less compared to that of the metallic one, still, it couldn't meet the mass constraints. The results of free vibration and buckling analysis are shown in Tables 12 and 13 and the contours are given in Figs. 16 and 17.

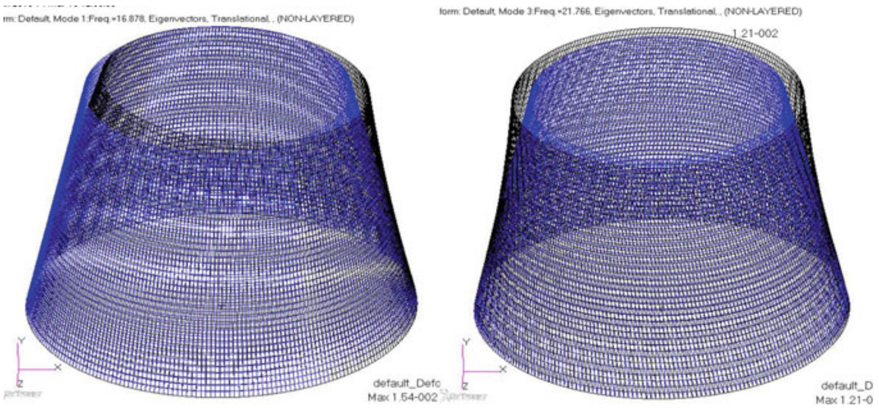
Table 12 Results of free vibration analysis of composite stringer-stiffened structure

Modes	Natural frequency (Hz)	Remarks
Mode 1	16.88	Lateral mode
Mode 2	16.88	Lateral mode
Mode 3	21.77	Torsional mode
Mode 4	42.36	Lateral mode
Mode 5	42.36	Lateral mode
Mode 6	97.73	Axial mode
Mode 7	393.20	Shell mode
Mode 8	393.20	Shell mode
Mode 9	393.26	Shell mode
Mode 10	393.26	Shell mode

Table 13 Results of buckling analysis of composite stringer-stiffened structure

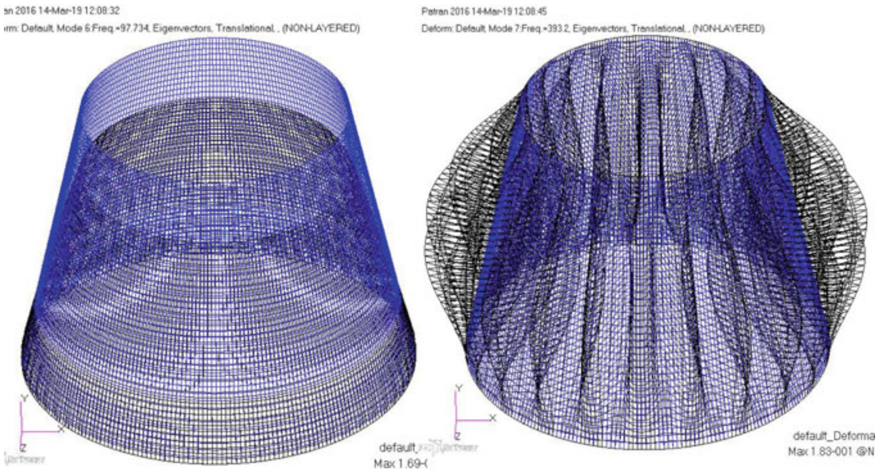
Modes	Buckling load factor		
	Load factor	Applying K.D.F.	Applying F.O.S.
Mode 1	2.15	1.72	1.03
Mode 2	2.15	1.72	1.03
Mode 3	2.82	2.26	1.35
Mode 4	2.82	2.26	1.35
Mode 5	3.07	2.46	1.47
Mode 6	3.07	2.46	1.47
Mode 7	3.85	3.08	1.85
Mode 8	3.85	3.08	1.85
Mode 9	4.93	3.95	2.37
Mode 10	4.93	3.95	2.37

In this case also, the lateral and axial mode of free vibration have frequencies more than 15 Hz and 30 Hz, respectively, and the buckling load factor obtained is also greater than 1. The stresses (Fig. 18) obtained in all the layers of the laminated shell and the stringers are within the specified limit. Here, the mass of the structure is less than 25 kg and hence meets all the design constraints including the mass criterion.



(a) Lateral Mode (b) Torsional Mode

Fig. 16 Free vibration modes of laminated stringer-stiffened structure



(c) Axial Mode (d) Torsional Mode

Fig. 16 (continued)

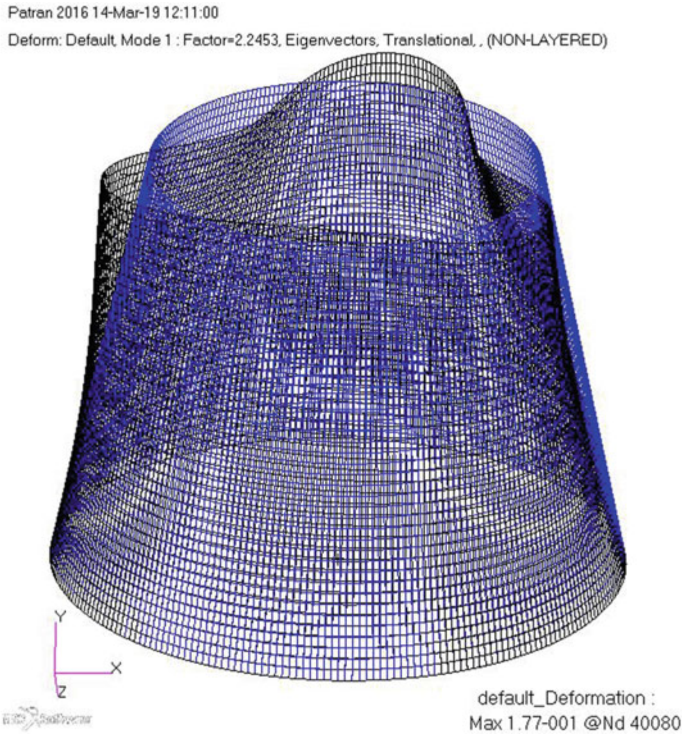


Fig. 17 First buckling mode of laminated stringer-stiffened structure

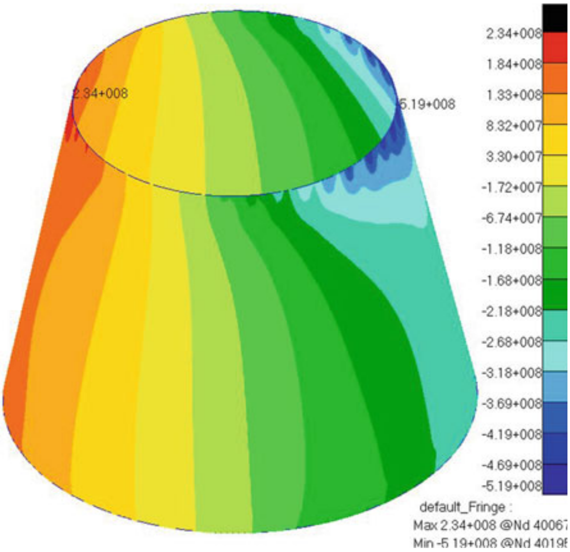


Fig. 18 Stress distribution of layer 1 of laminated stringer-stiffened PLA

7.3 Sandwich-Structured Design

Metallic-Skinned Sandwich Structure. In the sandwich-structured construction made of aluminium skin and aluminium honeycomb core, the optimum design was obtained when 2.5 mm-thick aluminium sheet was given on either side of 10 mm-thick core. The mass of the structure obtained is 53.35 kg. The results of free vibration and buckling analysis are shown in Tables 14 and 15. The free vibration modes and the first buckling mode of the structure are shown in Figs. 19 and 20.

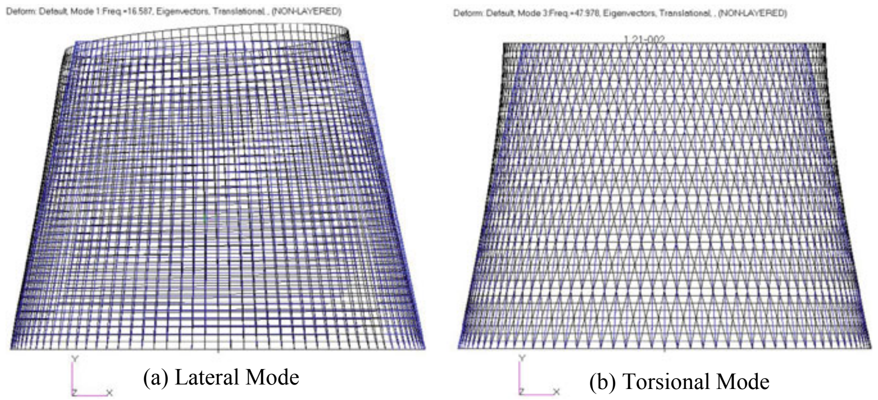
Here, the minimum frequency in the lateral mode of free vibration is obtained as 16.59 Hz and in the axial mode is 90 Hz. The buckling load factor after applying

Table 14 Results of free vibration analysis of metallic-skinned sandwich structure

Modes	Natural frequency (Hz)	Remarks
Mode 1	16.59	Lateral mode
Mode 2	16.59	Lateral mode
Mode 3	47.98	Torsional mode
Mode 4	87.36	Lateral mode
Mode 5	87.36	Lateral mode
Mode 6	90.00	Axial mode
Mode 7	393.36	Shell mode
Mode 8	393.36	Shell mode
Mode 9	430.18	Shell mode
Mode 10	430.18	Shell mode

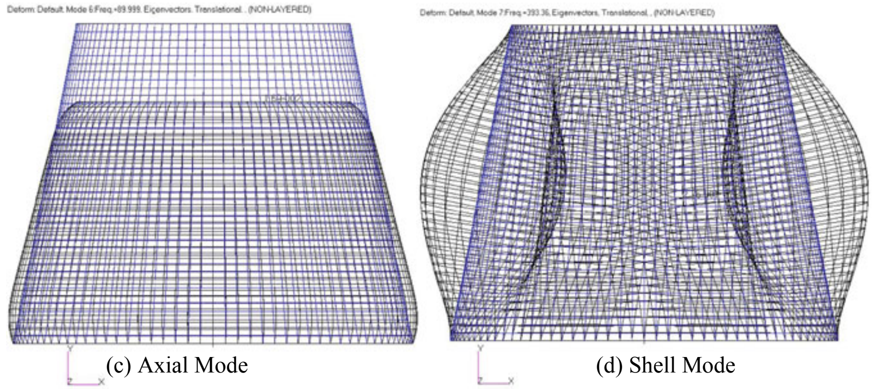
Table 15 Results of buckling analysis of metallic-skinned sandwich structure

Modes	Buckling load factor		
	Load factor	Applying K.D.F.	Applying F.O.S.
Mode 1	4.86	3.89	2.33
Mode 2	4.86	3.89	2.33
Mode 3	5.10	4.08	2.45
Mode 4	5.10	4.08	2.45
Mode 5	5.70	4.56	2.74
Mode 6	5.70	4.56	2.74
Mode 7	6.84	5.47	3.28
Mode 8	6.84	5.47	3.28
Mode 9	8.14	6.51	3.90
Mode 10	8.14	6.51	3.90



(a) Lateral Mode

(b) Torsional Mode



(c) Axial Mode

(d) Shell Mode

Fig. 19 Free vibration modes of metallic-skinned sandwich PLA

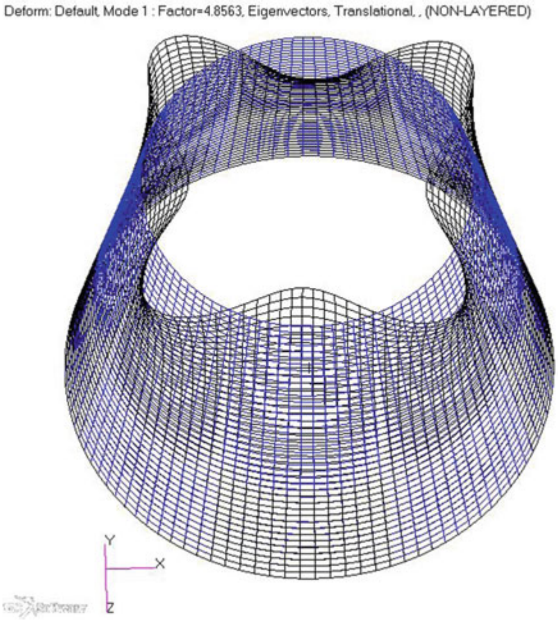


Fig. 20 First buckling mode of metallic-skinned sandwich PLA

F.O.S. and K.D.F. is obtained as 2.33. The maximum stress (Fig. 21) acting on the structure is also less than the strength of the material. But, the mass of the structure is much more than the limit.

Laminate-Skinned Sandwich Structure. In the sandwich-structured construction made of M55J/M18 laminate skin and aluminium honeycomb core, the optimum design was obtained when 15 laminate layers where used on either side of 10 mm-thick core. The optimum orientation of the shell is obtained as [0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0/90/0]. The mass of the designed structure is 21.59 kg. The results of free vibration and buckling analysis are shown in Tables 16 and 17. The free vibration modes and the first buckling mode of the structure are shown in Figs. 22 and 23.

In this case also, the lateral and axial mode of free vibration have frequencies more than 15 Hz and 30 Hz, respectively, and the buckling load factor obtained is also greater than 1. The stresses acting anywhere on the structure are within the specified limit (Fig. 24). Here, the mass of the structure is less than 25 kg and hence meets all the design constraints including the mass criterion.

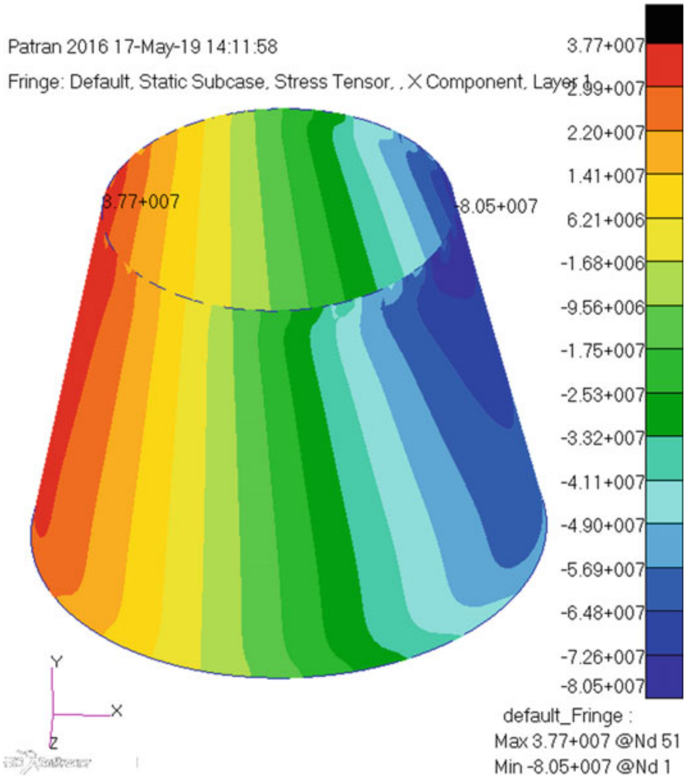


Fig. 21 Stress distribution on the skin of metallic-skinned sandwich PLA

Table 16 Results of free vibration analysis of laminate-skinned sandwich structure

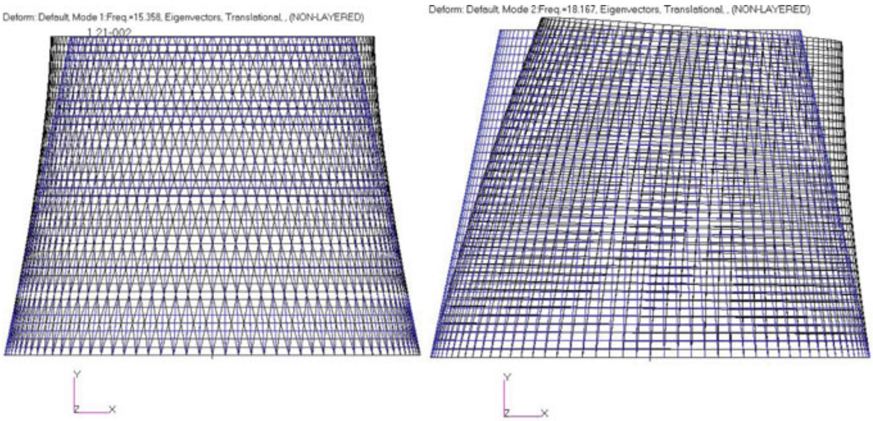
Modes	Natural frequency (Hz)	Remarks
Mode 1	15.36	Torsional mode
Mode 2	18.17	Lateral mode
Mode 3	18.17	Lateral mode
Mode 4	30.31	Lateral mode
Mode 5	30.31	Lateral mode
Mode 6	100.54	Axial mode
Mode 7	319.13	Shell mode
Mode 8	319.13	Shell mode
Mode 9	355.45	Shell mode
Mode 10	355.45	Shell mode

Table 17 Results of buckling analysis of laminate-skinned sandwich structure

Modes	Buckling load factor		
	Load factor	Applying K.D.F.	Applying F.O.S.
Mode 1	2.78	2.22	1.33
Mode 2	2.78	2.22	1.33
Mode 3	2.88	2.30	1.38
Mode 4	2.88	2.30	1.38
Mode 5	3.54	2.83	1.70
Mode 6	3.54	2.83	1.70
Mode 7	4.45	3.56	2.14
Mode 8	4.45	3.56	2.14
Mode 9	5.14	4.11	2.47
Mode 10	5.14	4.11	2.47

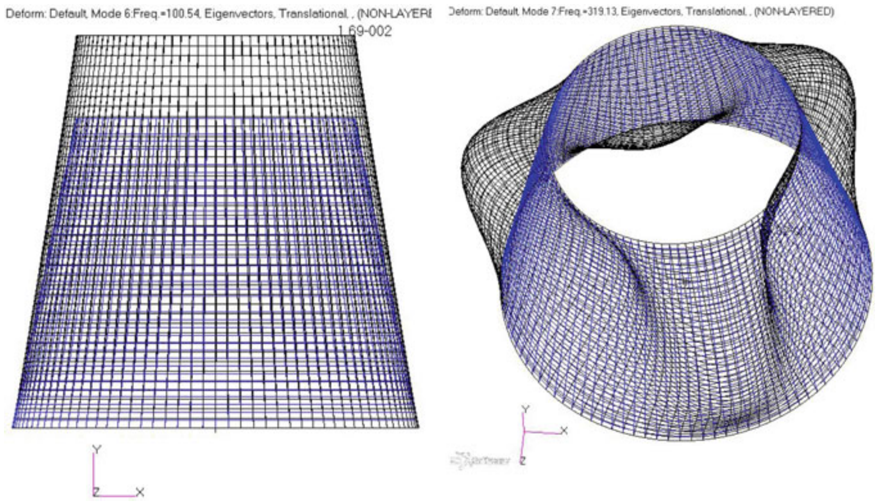
8 Conclusion

Three different configurations of PLA were studied. It was found that the use of composites resulted in more than 50% mass reduction in all three configurations compared to the metallic version. The sandwich structure and stringer-stiffened structures made of composites were found to save 29% and 28% weight, respectively, compared to the laminated monocoque structure. Even though both these structures were found to be effective in saving mass, sandwich structure is mostly preferred due to its ease of manufacturing compared to stringer-stiffened.



(a) Torsional Mode (b) Lateral Mode

Fig. 22 Free vibration modes of laminate-skinned sandwich structured PLA



(c) Axial Mode

(d) Shell Mode

Fig. 22 (continued)

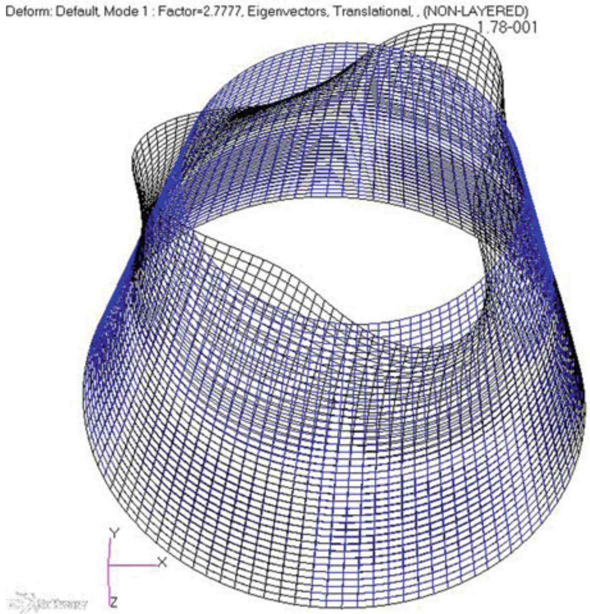


Fig. 23 First buckling mode of laminate-skinned sandwich structured PLA

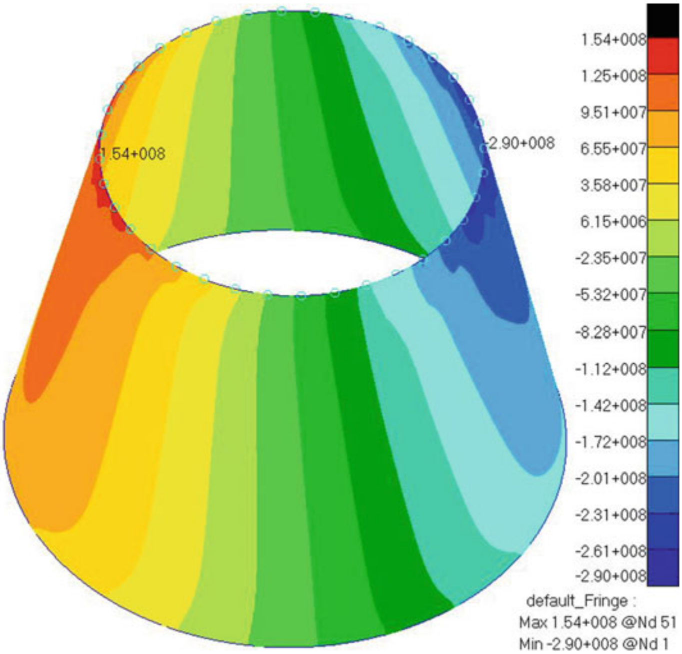


Fig. 24 Stress distribution in layer 1 of laminated sandwich structure

Thus, the directional property of composites is utilised here. The weight savings achieved by converting a sheet-like monocoque structure into a stiffened and a sandwich structure is shown. This establishes the role of composites in economizing space missions.

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