Vibration analysis of metal–polymer sandwich structure incorporated in car bonnet

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Abstract. The weight of the car material plays an important role in its performance in terms of fuel efficiency, speed and smoke emission. Especially in hybrid electric vehicles, lightweight materials are important for balancing the weight of its electrical components and also to enhance vehicle speed on account of its comparative low mileage. Nowadays, aluminium, magnesium and Carbon Fibre Reinforced Composites (CFRP) were introduced in making of car body panels since those materials can improve its performance. Yet the material costs, fibre orientation and fabrication techniques in case of CFRP in mass production remains a question. Also, the interior of the car panels was filled with vibration barriers or isolators with considerable thickness which can also add weight to the car. Hence a sandwich structure is presented in this work where the polyurethane foam of higher thickness (core) is bonded between two thin metallic face sheets (skin). The polyurethane foam was chosen as the core because of its low density, porosity and viscoelastic behaviour where it can provide better damping capabilities. Hence, the foam can also help in reducing the volume of internal vibration isolators. Since the car bonnet receives the vibration from sources like aerodynamic exposure to air during travel and mechanical movements within the vehicle (engine operation), it was chosen as the subject for vibration analysis.

1. Introduction
In this current world, a large number of automobile industries started operating and rolled out number of variants of passenger and commercial vehicles which in turn posed challenges to the existing car manufacturers and also reached the larger population of most of the countries. Consequently, the demand for energy requirements for powering the vehicles has become increasingly high.

So nowadays the automobile and aeronautical industries took this challenge and explored intensively before centered towards the material system and then came out with new generation material—the composites against the conventional steel, aluminium and their alloys to achieve the object of fuel-efficient vehicle[1]. These new energy efficient vehicles reached the market with lot of queries regarding safety, comfort of the passenger and also their durability and life. Thus, the major concern of every automobile manufacturer took anew-turn and in addition to the energy efficient vehicles, to satisfy the needs of the customer they emphasized on safety and comforts of the
occupants, concentrated on improving the performance, and reducing the cost of the vehicles. The changed concern of the automobile industries gave avenue for lot of research in the field of automobile sector, further with the improved skill in computation and engineering sciences lot of newer materials came into the automotive applications owing to their unique properties.

1.1 Car bonnet
The car bonnet belongs to one of the automobile body panel that is located at the front which is hinged towards its chassis. The shape of the car bonnet is generally streamlined in order to reduce air drag during the vehicle travel. In other words, the bonnet should provide minimum hindrance to air flow. Yet its main purpose is to cover the engine along with its auxiliaries located inside and is assembled such that it should provide access to it for maintenance and repair. The car bonnet is composed of two parts. One is the outer car bonnet that serves its aesthetic purpose and confirms to the overall car body shape. The other part is the stiffener whose role is to prevent the outer panel from bending whose reason can be described by its presence of multiple beads, grooves and slots. Between these two components, reinforcing members are place for strengthening purpose. The reinforcing members consist of a structure with base material of plastic or aluminium and an insert material of hard metal [1, 2].

The external loading starts from the outer panel which gets distributed to the stiffener via the reinforcing members. The edges of both components are joined by hemming whereas strong adhesives are applied in the middle areas for bonding. Some High-performance cars like racing cars have opening in the bonnet to allow the engine breathe faster which is formerly known as “Hood Scoops” that channels the air directly to the air filter, which gives improved performance and efficiency[1,2].

1.2 Vibrations and constraints involved in car bonnet
The bonnet can be subjected to various factors like head impact of pedestrian during a car accident, dent formation or waviness on the surface called as oil canning and hood flutter due to continuous exposure to incoming air. Nowadays the engineers are looking for ways to improve the performance of a bonnet panel with respect to above conditions either modifying its design or varying some parameters. Yet the changes incorporated in the outer bonnet panel such as the curvature should not affect the entire physique of the vehicle [3]. Therefore, the most effective variable for the optimization of the vehicle is the panel thickness [4]. On the other hand an optimal thickness leads to lower fuel consumption and also helps in maintaining a good acceleration performance and hence leads to reducing the weight of a vehicle [5].

Since the surface area of the car bonnet is larger than the other components, it is considered as a target for weight reduction. However, by addressing all of the above-mentioned issues, the engine bonnet panel will be more susceptible to vibrations. Bonnet vibrations are mainly caused by [1, 2]
- Transmission of engine vibrations via structural mounting connected to car bonnet and it is located at immediate vicinity to the engine
- Fluid-structure interactions due to the aero elasticity also known as the flutter phenomenon that occurs when the car is moving at considerable speed and,
- Base excitations, resulting from the road surface irregularities.

1.3 Sandwich Structures
Sandwich structure is defined as special type of composites where a material of low density and higher thickness is sandwiched between two high density materials with smallest thickness (upto 1mm). The laminates are bonded by use of adhesives and later compressed [6, 7]. These composites exhibit higher specific modulus and are capable in reducing the weight of the components made out of it. These sandwich composites exhibit better properties like improved rigidity, low specific weight, superior isolating quality, excellent vibration damping ability and good fatigue strength [7]. These sandwiched composites are classic materials that find variety of applications for a passenger car in the form of roof, doors, floor, chassis, bumpers, laminated side glass, frames, luggage lids and bonnets.
The core material can either have a polymeric foam or high-density material (like metals) in a honeycomb, triangular, truss or any other profile [8]. The latter type is mostly used in aerospace applications because of its higher shear strength, high bending resistance and high stiffness to weight ratio [6]. Yet the bonding of these type of core towards face sheets can be challenging. In case of polymeric foam, it provides excellent damping capabilities and offers light weight but possess bending stiffness lesser than honeycomb core. Hence it is restricted to the application of housing panels and structures [6].

![Image of sandwich structure](image)

**Figure 1.** A typical sandwich structure having phenolic foam core bonded between two laminates with glue or adhesive [9].

### 1.4 Damping behaviour in polymers

Damping is a phenomenon that plays a role in reducing the amplitude of oscillations or prevents oscillations in a system by a mechanism that opposes the changes in it. When an external oscillating force is applied to a material/structure, damping occurs by dissipation of mechanical energy, viz., transformation of mechanical energy into other forms of energy such as heat. Hence, damping represents the capacity for energy absorption [10, 11].

Like, in case of polymers, hysteresis or structural damping is observed where heat dissipation is indicated by internal continuous deformation of viscoelastic solid. During dynamic loading of a polymeric solid, the relation between stress and strain is nonlinear and different. Metals are generally coupled with passive dampers i.e., polymers to reduce the vibration of the system since it can act like a combination of spring and dashpot system [10, 11].

Damping capability is influenced by many factors such as the nature of material, modulus, frequency, temperature and defects like dislocations and phase boundaries. In case of metals and ceramics, damping occurs mainly due to defects and cracks. Many mechanical properties such as fatigue life, toughness, wear and coefficient of friction are intimately related to damping. Several factors can influence damping in polymers, viz., viscoelasticity, glass transition phenomenon, presence of fillers (compounding ingredients), etc [10].

The following research papers and proceedings were explored widely that has motivated the formulation of sandwich structure type composite in our desired application and also methodologies adopted in evaluating its properties and performance.
Table 1. Literature review on research papers and proceedings.

| Authors | Subject for research | Parameters investigated | Methodology | Inferences |
|---------|-----------------------|-------------------------|-------------|------------|
| MP Arunkumar, Jeyaraj Pitchaimani, KVGangadharan and MC Lenin Babu[8] | Sandwich aircraft panels having | Sound transmission loss via frequency response | 1. Finite element analysis using 3D rectangular sandwich model and its equivalent 2D model in Ansys. 2. Vibration source was applied on both orthogonal and inclined direction. 3. Face sheet thickness, core height and cell size of honeycomb core were varied | 1. Increase in thickness of the face sheet resulted in increase in sound transmission loss at dense areas. 2. Variation in honeycomb cell size contributed no variations. 3. Sandwich employing triangular shaped core has better acoustic performance in low frequency application because of even stiffness across its whole surface area. |
| Ashish M Ganeshpur and DV Bhope[1] | Five different designs of stiffener panel in structural steel car bonnet with respect to location of slots at 1. center of stiffener 2. Front side 3. Rear side 4. Rear side along with inner rib. 5. All of the above four areas | Static stress, total deformation and free vibration | 1. Car bonnet was modelled in Pro-Engineer design software and the finite element analysis is done using Ansys. 2. Fixed boundary conditions were applied at the corners and uniform pressure is applied over the outer panel of car hood. | 1. The design having continuous slots at the rear side and at the inner rib has similar results on comparison with the original car hood design. The only difference is its weight. 2. The stiffener panel having large number of holes and slots (5th design) deforms more because of stress concentration. |
| Can Y, Yazici M and Guclu H[12] | Sandwich structure with polymer foam core and metal face sheet is introduced to outer side of car bonnet. 2. Different thickness of polymer foam core about 1. Sandwich structure with polymer foam core and metal face sheet is introduced to outer side of car bonnet. 2. Different thickness of polymer foam core about | Energy absorption level of outer car hood during pedestrian head impact via acceleration-time curve | 1. The surface model of car hood having sandwich structure are created in CATIA V5. 2. The adult head was modelled with assumption of aluminium sphere core with | 1. The waveform obtained in case of sandwich structure does not generate higher peaks as produced by the original steel car hood which indicates that HIC values are lower. 2. The work concludes that the energy absorption for sandwich |
5mm, 15mm and 25mm is used as research parameter. thick vinyl cover
3. Analysis is done using Abaqus FEA software by simulating the normal impact of human head.

structure is large due to the viscoelastic nature of polymer foam.

| Jun Zhang, Guoze Shen, Yu Du and Ping Hu[13] | Four different material configurations for outer car hood and stiffener were taken with different thickness |
|-----------------------------------------------|---------------------------------------------------------------------------------------------------|
| 1. St(0.8) – Al(1.5) | 1. Residual stress and strain; dimensional changes due to stamping process. |
| 2. Al(1.5) – St(0.8) | 2. Natural frequency with determined stamping effects. |
| 3. Al(1.5) – Al(1.5) | |
| 4. Al(0.8) – Al(1.8) | |
| Stamping effect on manufacturing of car hood were included for modal analysis. |

| Lucie Rouleau, Alain Guinault and Jean-Francois Deu[14] | 1st stage: For mechanical properties evaluations two polymer foam were taken |
|----------------------------------------------------------|----------------------------------------------------------------------------------|
| 1. Polyurethane foam | 1. Residual stress, strain and thickness change was observed by manufacturing simulation of car hood using DYNAFOAM. |
| 2. Melamine foam | 2. Natural frequency for four designs of car hood with and without stamping effects is determined using LS-DYNA (FEA software) |
| 2nd stage: For numerical simulation of vibration damping, three structural configurations were taken |
| 1. Bare aluminium panel | 1. Natural frequencies drop approximately 1-5 Hz while considering all three stamping effects together (stress, strain and thickness changes) |
| 2. Aluminium panel + foam (both alternatively) | 2. If considering the effect of thickness change alone, the change is finite. |
| 3 Aluminium panel + rubber + foam. | 3. But considering stress and strain effects separately, change is negligible |

1st stage: Storage and loss shear moduli as a function of angular frequency and applied temperature
2nd stage: Frequency response at the frequency range between 0 – 800 Hz under simply supported boundary conditions

1st stage: Mechanical parameters of both foams are determined experimentally using torsional rheometer MCR apparatus.
2nd stage: The above parameters obtained were used as material data for both foams under study in numerical simulation carried out in ANSYS

1. The third structural configuration suggested in second stage has greater damping capacity.
2. The damping property of polyurethane foam is greater than melamine foam.
Murat Sen, Orhan Çakar and İsmail Hakkı Şanlıtürk[15]

- Sandwich panel for vibration isolation having
  
  Face sheet: ST37 of dimensions 430 x 430 x 0.9mm
  
  Core: Polyurethane foam of density 28kg/m³ of thickness 16mm, 26mm and 36mm

- Natural frequencies at first five modes from both experimental and numerical

1. The fabricated sandwich samples were marked into different areas to which modal impact hammer is struck. The frequency data is obtained via accelerometer.
2. The obtained results then validated through FEA analysis.

- At first mode, the experimental and numerical values strongly agree with each other.
- The increase in foam thickness causes an increase in natural frequency as well as damping capacity.

Qi Liu, Yong Xia, Qing Zhou and Jenne-Tai Wang[16]

- Full aluminium sandwich structure incorporated in outer car bonnet. The core layer is of corrugated plate type glued between two metal skins.

- Energy absorption level of outer car hood during pedestrian head impact via acceleration-time curve

1. A finite element (FE) model of the sandwich hood with necessary front-end structures was created.
2. The glue strips were modelled in form of 3D solid and rigid body elements (RBE’s)
3. The outer car hood was divided into two areas for head impact i.e., child head section and adult hood section.

- Half the points on the car hood satisfy the head injury criterion less than 800 and remaining lies at the range between 800 and 1000.
- The manufacturability of the stated sandwich structure remains a question.

Vinay V Nesaragi, Maruthi BH, Chandru B T and Dileep Kumar[2]

- Metallic car bonnet with and without stiffener was taken as subject

- Natural frequency and mode shapes, Frequency response

1. A surface model of car hood is prepared using CATIA V5.
2. Pre-processing i.e., meshing of imported model is done in Hypermesh v11.0
3. Analysis was carried out under two boundary conditions i.e., free-free and free-fixed.
4. For frequency response analysis, a vibrating source is applied at the front end.

- Car bonnet having stiffener possess lower vibration levels with lower displacement and higher natural frequency.
Zhengqing Liu, Mohammad Fard and John Laurence Davy [17]

| Porous material made of woven fabric attached to the interior of car cabin model (simulated and prototype) | Sound pressure level (SPL) |
|------------------------------------------------------------------------------------------------------|---------------------------|
| 1. An experimental prototype of car cabin was created. An aluminium panel mounted in it was taken under study. |
| 2. Analysis was carried out with and without porous material by exciting the panel with electrodynamic shaker and SPL is recorded via microphones inside the car cabin. |
| 3. The results were obtained in waveform which is validated by use of FEA software. |
| The sound pressure level in the car cabin model is reduced when porous material is included. |
Various papers associated with the analysis of car body panels with respect to absorption of impact energy due to collision of head to car bonnet and NVH characteristic using different metals were explored. Also the contamination variables (say residual stress, strains and tolerance change due to stamping) that incurred in car bonnet due to manufacturing effects and its effects on mechanical properties were explored [13]. Hence, in order to optimize the panels in automotive applications, possible alternate materials to conventional metals were explored. That is where the composites come into play. Several research papers were explored that describes the property evaluations of sandwich type composite samples like strength, stiffness [8,9] resistance to external environmental effects, [18] vibration damping[15,17] etc. through experimental methods. However, its application in automobile subjected to vibrations were not covered in detail.

2. Selection of materials
In the production of conventional automobiles, steel was used in the body panels and chassis construction [19]because of its high hardness, strength, inherent capability to absorb impact energy in a crash situation. The steel used contains about 0.2 to 2.1 percent of carbon hence termed low carbon steel. But nowadays automotive engineers have steadily made use of aluminium because it can potentially reduce the weight of the vehicle body possessing similar strength with that of steel. Recent developments have shown that up to 50% weight saving for the body in white (BIW) can be achieved by the substitution of steel by aluminium. But the only limitation is that aluminium is more expensive than steel and hence it is restricted to expensive cars [19]. This limitation has directed the attraction towards the use of composites where different materials are arranged and bonded in layers which formally called as sandwich structures [6, 7]. This current work has made use of sandwich structure in the application of automobile body panels where the car bonnet is taken as the subject for our research.

| Table 2. Mechanical properties of materials adopted in sandwich structure. |
|--------------------------|--------------------------|--------------------------|--------------------------|--------------------------|
|                         | Low Carbon steel | Aluminium 2024 | Open cell polyurethane foam[17] | Epoxy adhesive |
| Young’s Modulus (MPa)   | 2 x 10^5         | 7 x 10^4       | 13.61                      | 3670                     |
| - longitudinal          |               |               |                           |                           |
| Young’s Modulus (MPa)   | 2 x 10^5         | 7 x 10^4       | 7.26                       | 3670                     |
| - Transverse            |               |               |                           |                           |
| Poisson’s ratio         | 0.3            | 0.33          | 0.11                       | 0.321                    |
| Density (kg/m^3)        | 7800           | 2700          | 32                         | 1460                     |

Polyurethane foam of open cell type was chosen as a core material because of its excellent damping properties at wide frequency range and offers light weight [10, 11]. Its damping nature is attributed to its viscoelasticity where the vibrational waves passing through pores or voids in it creates internal deformation and its mechanical energy developed gets dissipated as heat[11].

3. Sandwich structure arrangement
In this current work, five different kinds of material configurations were chosen for the outer bonnet of the car bonnet which is considered as major area of interest in vibration analysis. The following configurations were listed below.
- Steel (low carbon)
- Aluminium (Al 2024 T3)
- Steel – PU foam – Steel
- Aluminium – PU foam – Aluminium
- Aluminium – PU foam – Steel

The first two configurations are nothing but plain sheet metal of standard thickness gauge 20 (1.5 mm) [3, 20]. Those two configurations stated above were only considered as a base or reference model
for comparison purpose. Yet the material for the stiffener used for all is low carbon steel. In the last two configurations the metal is used as face sheets where a polyurethane (PU) foam core of thickness 5 mm is sandwiched between two face sheets as shown in below figure. The face sheets and the foam core were bonded together by an adhesive of epoxy type because it provides a rigid and stiff layer at its interface.

![Schematic diagram of sandwich structure](image)

**Figure 2.** Schematic diagram of sandwich structure where the polyurethane foam is bonded by epoxy adhesive between (a) two steel skins (b) two aluminium skins and (c) aluminium and steel skins.

### 4. 3D surface model of car bonnet

The entire 3D surface modelling of the car bonnet was done in CATIAV5R20 software because of its flexibility in creation of surface. The profile of the outer bonnet was taken from the blueprints of Volkswagen Jetta 2010 model which was taken from a site named ‘the-blueprints.com’ available in four different two-dimensional views (front view, top view, side view and rear view) in orthogonal projections within a single image. Only the front view, top view and the side view were cropped from the blueprint. The cropped images were placed on the three orthogonal design reference planes designated as x-y plane, y-z plane and z-x plane. The following were done in ‘sketch tracer’ workbench that allows the placement of two-dimensional sketch for profile extraction. In other words, the sketch tracer allows reverse engineering process. The scale size of the image imported can be defined as per the overall dimension of the car (length = 4739mm, width = 1778mm and height = 1482mm).
Figure 3. Profile extraction of outer car bonnet from 2D sketch of an existing car model.

(a)  
(b)

Figure 4. CAD surface model of (a) outer car bonnet (b) stiffener.

The profile of the car bonnet was extracted by manually defining three dimensional curves tracing the outline of the car bonnet. The position of defining points on the curve were adjusted by accordingly switching between three different views. The curves defined serves as a basement for defining the outer surface of car bonnet. Next, the stiffener (inner bonnet) was design by performing a rough extraction of the outer bonnet offset at a distance of 18-20mm in order to maintain the outer dimension same as the outer car bonnet. The detailing of the stiffener was done keeping in mind of the bending stiffness in outer car bonnet. The holes and small beads in stiffener were ignored for convenience during pre-processing phase in Hyper mesh software.

5. Pre-processing
The 3D model of car bonnet saved in iges file format was imported into Hyper Works FEA software. The first stage was meshing or simply put discretization of whole model into a number of small elements. The elements are where the set of equations can be solved. These equations approximately represent the governing equation of interest via a set of polynomial functions defined over each element. The shape of the elements for 2D type surface can be triangle (called as tria containing 3 nodes), rectangle (called as quad containing 4 nodes) [21].

However, in structural analysis, quad type elements are most preferable and tria elements are suited for computational fluid dynamics (CFD) and mold flow analysis [21]. Because of the complexity in its geometry, some tria elements have emerged in between quad elements. But most importantly, the elements have to satisfy some important criteria like aspect ratio (ratio of maximum to minimum length of an element), skew, warpage and jacobian (shape of the element represented in matrix form) such that it can deliver accurate solutions [21].

Hence before the meshing process, the imported iges model from CAD software was thoroughly examined in order to identify area resembling known shapes and also highly concentrated areas. These
identified areas were split and meshed separately in order to produce a high-quality mesh of suitable density. Later, the remaining areas were also meshed or discretized such that its nodes at the ends connect to the elements in the already meshed areas.

The meshing of both the parts were done using interactive mode available in ‘automesh’ option. The interactive mode allows the variation of number of nodes at the edges in order to control the number and shape of elements. The number of nodes is varied in such a way that maximum percentage of quad elements should appear. Also, frequent verification and quality checks during the stage of meshing process was done to identify elements that doesn’t comply with its standard criteria. The failed elements can be later optimized, remeshed or converted into equivalent tria elements using appropriate clean-up tools. It is also known that both the components (stiffener and outer car bonnet) of the car bonnet along with its elements were assigned separately in form of component collectors. The collectors are analogous to folders in windows explorer where files are organized.

![Figure 5. Discretized or meshed model of (a) outer car bonnet (b) stiffener.](image)

5.1 Assembly of stiffener with outer bonnet

The assembly of components can be achieved by use of connectors. Connectors are defined as geometric entities that define connections between other entities. The connectors are used to provide link between elements or nodes of two opposing surfaces or solids that can be represented in any forms like rigid link, rigid link with mass, welding, adhesives, bolted joints, riveted joints, spring elements etc [21]. In a broad sense, the connector elements mostly contain one dimensional beam or bar elements connected between nodes.

Generally, in car bonnet the ends of both the stiffener and outer car bonnet are joined together by means of edge hemming [1, 20]. The faces of the projecting portions in the middle of the stiffener is bonded with a strong heat resistant adhesive of epoxy type [16]. The bond can be represented by the use of ‘area connectors’ option.

![Figure 6. (a) The stiffener is assembled to the outer car bonnet by adhesive connectors (b) a detailed view of connectors having solid elements connected to both parts by rigid body elements (RBE’s).](image)
The above figure 6 shows the use of connectors (indicated in aqua blue colour) that is used to link both components. Initially the element on anyone component (say stiffener) where the bonding has to be applied were selected and the unrealized connectors between two components were created (connectors without finite elements). In order to define the adhesive property, the finite elements have to be defined which is possible by realizing the unrealized connectors keeping the type of connectors as ‘adhesive’ option.

The adhesive type connectors is mathematically represented by solid (PSOLID) elements having rigid body elements (RBE2 and RBE3) linked to the elements of outer car bonnet and stiffener as shown in figure 6 The rigid body elements are multi point constraint MPC elements that allows connection from one node to multiple number of nodes. The RBE2 elements represents a rigid link that allows transfer of motion from independent (master) node to dependent (slave) node and induces additional stiffness [21]. The RBE3 elements allows the transfer of loads between nodes without inducing additional stiffness. In case of RBE2 elements, there can be one independent node and multiple dependent nodes where as in case of RBE3 elements, it is reverse [21]. The PSOLID elements allows the definition of material properties of adhesive used. Hence its role is to induce some elasticity.

5.2 Creation or Simulation of sandwich structure in outer car bonnet

Hyper Works also provides features for incorporating composites in product design where it is laminate or fibre reinforced with definite orientation based. Like component collectors, the material and properties are also organized as collectors. In this work, separate plies were created each having the suitable material collector assigned to it. The laminate collector is used to define the order of the plies where it indirectly points towards the outer car bonnet by use of PCOMPP (composite-laminate of ply type) card available in property collector.

![Sandwich structure incorporated in outer car bonnet displayed in detailed view.](image)

Figure 7. Sandwich structure incorporated in outer car bonnet displayed in detailed view.

MAT1 card under material collectors is used for specifying material data for metals and adhesive since they are isotropic. In case of Polyurethane foam being orthotropic, MAT8 card is chosen. The MAT1 refers to material data of form 1 where temperature independent isotropic elastic properties can be defined where as MAT8 refers to material data of form 2 where linear orthotropic material properties that are assigned to the two-dimensional elements (material data in 1 and 2 direction). For pure steel and aluminium car bonnet, the PSHELL card (two-dimensional property) is assigned along with its respective material collector.
5.3 Boundary conditions
The boundary condition is the application of a force and/or constraint. In Hyper Mesh, boundary conditions are stored within what are called load collectors similar to component collectors. The boundary conditions applied in this current model was adopted from work done by Vinay M. Nesaragi et al [2].

The single point constraints were created with fixed degrees of freedom using the card image SPC available in ‘constraints’ option under ‘analysis’ panel. The fixed boundary conditions are indicated by the numbers 123456 where numbers 1, 2 and 3 refers to longitudinal direction in x, y and z direction and 4, 5 and 6 refers to rotational direction about x, y and z direction. To perform frequency response analysis, an additional constraint is required to define an external vibrating source otherwise called as frequency dependent load. The frequency dependent load requires two load collectors. One refers to unit load under the card image DAREA where location for vibrating sources defined and is applied at z direction indicated as 3. The other load collector requires a frequency range where 0-500 Hz is specified in it. The following is created using the TABLED1 card image that represent linear relation between two variables. For sandwich structure having the polyurethane foam core, an additional collector is specified to define the damping behaviour of the foam. The card image used for the following is TABDMP1 where the damping coefficient of value 0.06 [8-9] is associated along with the defined frequency range.

![Figure 8. Necessary boundary conditions required for solving vibrational frequency response.](image-url)

6. Modal analysis
Normal Modes Analysis, also called eigen value analysis or eigen value extraction, is a technique used to calculate the vibration shapes and associated frequencies that a structure will exhibit. These frequencies are important because if cyclic loads are applied at these frequencies, the structure can go into a resonance condition that will lead to catastrophic failure. The mode shapes are also important in order to make sure that loads are not applied at points that will cause the resonance condition. Normal modes analysis is also required for modal frequency response and modal transient analysis. The equilibrium equation for a structure performing free vibration appears as the eigen value problem:

\[
[K - \lambda M] \phi_i = 0
\]  

where K denotes the stiffness matrix of the structure and M is the mass matrix. Damping is neglected. The solution of the eigen value problem yields eigen values \( \lambda \), where n is the number of degrees of freedom. The vector \( \phi_i \) is the eigenvector corresponding to the eigen value \( \lambda \).
The eigen value problem can be solved using three numerical methods Lanczos, AMSES, or AMLS. But in most cases Lanczos method is preferred because it offers advantage of solving eigen values and associated mode shapes exactly and directly. In Hypermesh, the card EIGRL is used available under ‘load collector’ option. The natural frequency \( f_i \) follows directly from the eigen value \( \lambda_i \).

\[
f_i = \frac{\sqrt{\lambda_i}}{2\pi}
\]  

The modal analysis was performed in Radioss under bulk data format that can be set under user preference option while opening the Hyper Works window. The discretized model and associated collectors were imported where the single point constraints and number of mode shapes defined as separate load collector were taken as inputs for calculating normal modes through ‘loadsteps’ in analysis panel. The above data created were stored as .fem file in some local drive which can be read by Radioss solver and performs the necessary FEM calculation.

The output results file stored in the format .h3d file was loaded into Hyper View module which is actually a post-processing software used to display results in form of contoured plots. The format of the output file can be modified by using control card named ‘OUTPUT’. The file was loaded in Hyper View, a post processing module where the deformed mode shapes (eigen vector) for the first ten natural frequencies (eigen value) can be viewed in contoured plots. Yet in designing structures to respond towards vibrations, the first six modes were only considered as the main priority as shown in below figures.

![Figure 9. First six mode shapes of steel outer car bonnet with steel stiffener (a) 1st mode (b) 2nd mode (c) 3rd mode (d) 4th mode (e) 5th mode (f) 6th mode.](image-url)
Figure 10. First six mode shapes of aluminium outer car bonnet with steel stiffener (a) 1st mode (b) 2nd mode (c) 3rd mode (d) 4th mode (e) 5th mode (f) 6th mode.

Figure 11. First six mode shapes of steel – PU foam – steel sandwich structure in outer car bonnet with steel stiffener (a) 1st mode (b) 2nd mode (c) 3rd mode (d) 4th mode (e) 5th mode (f) 6th mode.
Figure 12. First six mode shapes of aluminium – PU foam – aluminium sandwich structure in outer car bonnet with steel stiffener (a) 1st mode (b) 2nd mode (c) 3rd mode (d) 4th mode (e) 5th mode (f) 6th mode.

Figure 13. First six mode shapes of aluminium – PU foam – steel sandwich structure in outer car bonnet with steel stiffener (a) 1st mode (b) 2nd mode (c) 3rd mode (d) 4th mode (e) 5th mode (f) 6th mode.
As shown in the contour plots, the deformation is maximum at the rear end which then decreases towards the central portion as seen in the first mode shape of all the material configurations. Similarly, in the second and third mode shapes, the maximum deformation is observed at the front end and sides respectively which is also common for all configuration. But in the last mode, the mode shapes differ between the first two (pure metallic car bonnet) and the last three configurations (sandwich structures). In metallic type, the deformation is maximum at center whereas in sandwich type, it is at the sides towards the rear end as seen in the contour plots.

7. Modal Frequency response analysis
Frequency response analysis or otherwise called as harmonic analysis’s used to calculate the response of a structure under a harmonic excitation. Typical applications are noise, vibration and harshness analysis of vehicles, rotating machinery, and transmissions. The analysis is to compute the response of the structure, which is actually transient, in a static frequency domain. The loading is sinusoidal. A simple case is a load that has amplitude at a specified frequency. The response occurs at the same frequency, and damping would lead to a phase shift.

The loads can be applied as forces or enforced motions(displacements, velocities, and accelerations). They are dependent on the excitation frequency (ω). All the loads are applied on the frequency where the response is evaluated. The results/responses from an FRF analysis are displacements, velocities, accelerations, forces, stresses, and strains. The responses are usually complex numbers that are either given as magnitude and phase angle or as real and imaginary part. The frequency response function can be solved in two different methods:

- The direct method solves the coupled equation of motion in terms of the excitation frequency.
- The modal method uses the mode shape of the structure to uncouple the equations of motion and the solution for a particular excitation frequency is obtained by summation of individual modal responses or modal superposition.

However, in this work the modal method was chosen in order to generate displacement-frequency plots based on the eigen values and eigenvectors calculated from normal modes analysis. As discussed in the subsection 6.3, in addition to single point constraints a frequency dependent load was created under the card image RLOAD2 which in turn includes the load collects for source placement and frequency range. The card image RLOAD2 allows specifying the load and frequency in magnitude and phase form.

The output results were produced inform of displacement – frequency waveforms by giving instructions to Hyper View through control card ‘OUTPUT’. The output results were stored as ‘xxx_freq.mvw’ format which was opened in Hyper View. The output values were displayed in terms of magnitude and phase. The frequency response was recorded at five different selected points on the outer car bonnet by including a set containing five nodes at the places where the maximum displacement along z direction in mode shapes were recorded.
Figure 14. Modal frequency response of steel car bonnet represented as displacement – vibrating frequency curves in magnitude and phase form recorded at (a) rear side (b) right side (c) center (d) front side (e) left side.
Figure 15. Modal frequency response of aluminium car bonnet represented as displacement – vibrating frequency curves in magnitude and phase form recorded at (a) rear side (b) right side (c) center (d) front side (e) left side.

Figure 16. Modal frequency response of steel – PU foam – steel car bonnet represented as displacement – vibrating frequency curves in magnitude and phase form recorded at (a) rear side (b) right side (c) center (d) front side (e) left side.
Figure 17. Modal frequency response of aluminium – PU foam – aluminium car bonnet represented as displacement – vibrating frequency curves in magnitude and phase form recorded at (a) rear side (b) right side (c) center (d) front side (e) left side.
Figure 18. Modal frequency response of aluminium – PU foam – steel car bonnet represented as displacement – vibrating frequency curves in magnitude and phase form recorded at (a) rear side (b) right side (c) center (d) front side (e) left side.

In all the displacement – frequency waveforms, the rise in vibration displacement is either gradual or none up to the frequency value of 40 Hz. Above 40 Hz, the displacement suddenly increases up to a maximum value at some particular frequency and then suddenly decreases. The sudden increase and decrease in the amplitude is what called as a peak. In steel car bonnet, a single higher peak is observed whereas in aluminium car bonnet, multiple higher peaks are observed. In case of sandwich structure, the peaks generated above 80 Hz varies in magnitude which depend upon the position on the car hood and type of metal skin employed in it. As shown in the waveform, the higher peak (maximum displacement) mostly occur at lower frequency about 60 – 80 Hz in all material configuration.

8. Result and discussions

Before evaluating the vibrational response of a particular structure, it is important to identify the possible vibrational frequencies exhibited in it without considering any external continuous disturbance. These frequencies are called as natural frequencies. Hence when the structure connected to the vibrating source operates at a frequency equal to the natural frequency, the structure oscillates at some random positions which is indicated by its deformations called as mode shapes. These data can be predicted by means of modal analysis which determines the possible natural frequencies (eigen values) and the mode shapes indicated by the contour plots as shown in figures 9 – 13. The natural frequencies of car bonnet for all the five material configurations are tabulated in a comparative manner. Also, the maximum displacement for all modes are also tabulated as shown in table 3 which was read from the displacement contour scale.

As shown in the graphical plot (figure 19), the difference in natural frequencies among all the configurations is negligible from the first five modes i.e., at a range of 50 – 140 Hz. The difference becomes significant starting from the next five modes from the frequency range 150 – 260 Hz. Hence
in other terms, the natural frequencies in case of car bonnet incorporated with foam core sandwich structure is greater than natural frequencies for pure metallic car bonnet [1][9]. This is because the sandwich structure has light weight properties on account of the polymer foam and it is known that the natural frequency is inversely proportional to mass of the structure under study [19]. It is also observed that the natural frequency curves for both pure metallic car bonnet and sandwich constructed car bonnet almost coincide with each other respectively.

**Table 3.** Tabulation of natural frequencies and maximum displacement for the first ten mode shapes.

| Modes | Steel N.F. | Aluminium N.F. | Steel – PU – Aluminium N.F. | Aluminium – PU – Steel N.F. |
|-------|------------|----------------|-----------------------------|--------------------------------|
|       | M.D. | N.F. | M.D. | N.F. | M.D. | N.F. | M.D. | N.F. | M.D. |
| 1     | 60.4  | 55.2 | 11.9 | 72   | 11.1 | 66.2 | 12.7 | 70.7 | 11.4 |
| 2     | 74.6  | 66.1 | 15.1 | 85.5 | 14   | 77.8 | 16.1 | 84.5 | 14.6 |
| 3     | 81.5  | 81.3 | 12.6 | 106.6| 8.6  | 103.4| 11.6 | 106.9| 9.6  |
| 4     | 111.8 | 102.5| 15   | 144.1| 12.5 | 131.7| 16.4 | 142.8| 13.4 |
| 5     | 117.9 | 118.7| 18.3 | 145.6| 11.6 | 141  | 18.1 | 146.2| 14.5 |
| 6     | 133.5 | 142.1| 90.6 | 200.9| 12.7 | 193.6| 18.8 | 197.1| 14.5 |
| 7     | 145   | 149.2| 75.2 | 214.4| 10.5 | 208.5| 15.8 | 214.5| 12.2 |
| 8     | 148.2 | 157.1| 94.1 | 223.3| 12   | 217.1| 24.3 | 221.6| 12.7 |
| 9     | 155.1 | 163.3| 59.6 | 247.8| 15.1 | 229  | 20.5 | 244.5| 15.3 |
| 10    | 160.1 | 167.5| 50.9 | 263.2| 9.1  | 252.6| 13.9 | 255.6| 10.9 |

*a N.F. – Natural frequency for each mode  
*b M.D. – Maximum displacement observed at each mode shape

Now looking towards the maximum displacement versus mode shapes graphical plot, the curve behaviour is almost inverse to the former graphical plot. It is observed that the metallic car bonnet gets heavily deformed at higher modes especially the car bonnet employing aluminium where the displacement reaches 90mm at mode shapes 6 and 8. The car bonnet having steel has maximum displacement value 50mm at mode shape 8. However, the car bonnet employing the sandwich structure contains displacement values restricted between 8 – 20 mm i.e., the curve is somewhat linear except with slight deviations. This obviously means that the sandwich structure possesses better resistance to bending than conventional metal while operating under service condition[8, 9]. Moreover, the higher stiffness and comparative lightweight exhibited by the sandwich structure is responsible for greater natural frequencies at higher mode shapes indicated in figure 19(a).

![Figure 19. Graphical plot of (a) natural frequency and (b) maximum displacement with respect to each mode shape showing a comparison among all material configurations.](https://example.com/figure19.png)
Most of the vibrations are transmitted from the engine to the car hood connected via the chassis operating at variable frequency. The operating frequency is a function of its rotations per minute (rpm). The vibrations are caused by various factors like fast oscillating movements in intake valve and sliding of pistons in cylinder, reactions produced by rotation of crankshaft and camshaft, pressure generated during fuel injection followed by its ignition etc. Hence a study is done to predict the level of oscillations over a wide range of frequencies which is achieved by frequency response numerical method. From the modal frequency response analysis, the vibrations in a form of displacements or amplitude are recorded over the specified frequency range $0 – 500\text{Hz}$ which is generated via displacement – frequency waveforms. The car bonnet vibrating at higher displacements are indicated by the peaks in the generated waveform. The maximum displacement from the highest peak are tabled along with its corresponding vibrating frequency as shown below. The maximum displacement mostly occurs at frequency range $60 – 80\text{Hz}$ except in aluminium car bonnet it occurs at the frequency $121\text{Hz}$ at left and right side and $151\text{Hz}$ at central position. Those frequencies match closer to the natural frequencies of first five modes determined through modal analysis.

**Table 4.** Tabulation of maximum displacement at a particular vibrating frequency at five different locations on car bonnet.

| Position on car bonnet | Steel | Aluminium | Steel – PU – Steel | Aluminium – PU – Aluminium | Aluminium – PU – Steel |
|------------------------|-------|-----------|---------------------|----------------------------|------------------------|
| Front                  | 71    | 0.21      | 71                  | 0.12                       | 90                     |
| Rear                   | 61    | 0.53      | 61                  | 0.25                       | 80                     |
| left                   | 61    | 0.3       | 121                 | 0.13                       | 70                     |
| right                  | 61    | 0.3       | 121                 | 0.13                       | 70                     |
| center                 | 61    | 0.36      | 151                 | 0.65                       | 70                     |

$^c$ V.F. – Vibrating frequency  
$^d$ M.A. – Maximum Displacement at the vibrating frequency

It is seen that the maximum vibration displacement obtained is less than 1mm which is practically normal that the car bonnet mounted on the chassis carrying the engine is running at low speed (hence lower frequency). As shown in the graphical chart (Figure 20), while looking towards the selected positions on the car bonnet, the vibrations are greater in front and rear ends compared to central, left end and right end portions. If we are focussing towards the material configuration aspect, the displacement generated by the sandwich car bonnet is comparatively lower than metallic car bonnet.

![Figure 20](image_url)  
*Figure 20. Graphical plot of maximum displacement generated by the car bonnet of all five material configurations at the vibrating frequency.*
Except in case of sandwich structure having only aluminium as skin, the vibration displacement at front end is almost at the same level as that of steel car bonnet as seen at left and right side (about 0.3 mm). But the sandwich structure having steel skins has lowest displacement at all the positions compared to the other four configurations. Hence it is obvious that in the fifth material configuration, the sandwich structure having steel as one of its ply generate less displacement while comparing with aluminium sandwich car bonnet.

9. Conclusion
While striving towards lightweight panel design that houses the system involving moving components it is important to consider the dynamic reaction forces absorbed by the structure that holds these components. The transmission of motion and forces towards the panel is translated to varying movements which is called as vibration. The modal analysis is performed for validation purpose to identify the frequencies where the structure is likely to deform which is represented via mode shapes. The mode shapes show the possible deformation or deviation from its original shape at the resonant condition. Hence during the frequency response analysis, if the applied load vibrates at the natural frequency, the structure gets deformed plastically and its service life diminishes drastically. Hence it is utmost important to properly redesign those critical areas or keep the vibrating source away from it.

In this work, the car bonnet having sandwich structure possess higher natural frequencies than pure metallic car bonnet especially at higher modes. Also, the deformations in this case is lower which is also proven in modal frequency response analysis due to the damping action offered by the viscoelastic polymer foam. Moreover, sandwich structure having steel has considerable stiffness and better resistance to deformation compared to aluminium sandwich when looking towards the result summary of both the analysis.

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