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Dynamic response characteristics of an automotive lamp assembly

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Abstract

The objective of this paper is to model and analyse the dynamic response of an automotive lamp assembly. Modern automotive lamp assemblies have complex geometry and are composed of different parts made of polymer materials. As part of design verification, automotive lamp assemblies are subjected to accelerated random vibration tests to assess their integrity over a life-time exposure to mechanical vibration loading. Understanding the dynamic behaviour of the lamp is crucial in the numerical evaluation of the fatigue life. Dynamic analysis involves characterising the modal and harmonic behaviours. In this work, numerical modal properties and harmonic responses were validated using experimental testing. The numerical analysis was carried out using the ANSYS finite element analysis (FEA) software. Experimental modal properties including mode shapes and corresponding frequencies were determined using Polytec PSV-500 Xtra laser scanning head at a frequency range of 10 to 1000Hz. The experimental harmonic transmissibility responses of all the components of the lamp assembly were determined using a vibration shaker. The experimental and numerical mode shapes and responding frequencies obtained in the analyses compared well thus validating the numerical modal model. Furthermore, the mode shapes showed that the lamp assembly was mostly vibrating in bending, therefore subsequent analysis should take this into account. Harmonic response validation showed that the first few numerical resonant frequencies, that dominate the response, compared well with experimental results.

Keywords: Dynamic; Modal; Mode shape; Transmissibility; Automotive; Lamp assembly; Resonance; Laser Scanning; Component

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1. Introduction

As part of design verification, automotive lamp assemblies are subjected to accelerated random vibration tests during design to assess their integrity over a life-time exposure to mechanical vibration loading. Often, the lamp assemblies experience fatigue failure during testing. This results in a costly and time-consuming design cycle as design changes and modification of the injection moulding tool are needed, Okeke et al. (2019). Numerically predicting the fatigue life of an assembly prior to producing a prototype can help to address this shortcoming. However, reliable numerical fatigue life prediction requires to capture the actual dynamic behaviour of the lamp assembly. Accurate determination of the dynamic response of a structure is very essential to avoid undesirable vibration resonances, Presas et al. (2017), Hiremath et al. (2016). Hence, understanding the dynamic behaviour of the lamp is crucial in the numerical evaluation of the fatigue life. The objective of this paper is therefore to validate the dynamic response of an automotive lamp assembly for subsequent fatigue analysis. Raviprasad et al. (2015) pointed out that a dynamic analysis of a structure is crucial in early design phase to determine the natural frequencies and the corresponding modes of vibration and make design changes to move them away from the danger zone. The modern automotive lamp assembly has complex geometry and is constructed with different parts of polymer materials. The assembly is normally designed with the gap between parts being very small. With each part of the assembly having different geometry and material, they are expected to respond differently under dynamic loading. Roucoules et al. (2010) evaluated the frequency response function and durability of a headlamp, they noted that the results of frequency prediction, FRF and stress correlations are dependent on the gaps in the assembly, material properties, and boundary conditions.

Dynamic analysis involves characterising the modal and harmonic behaviours. Marzuki et al. (2015), numerically investigated the dynamic behaviour of a chassis structure using modal analysis and harmonic analysis. They noted that the analysis result can be used as a reference in improving the chassis design and dynamic performance. The modal analysis provides a good understanding of the structural resonance and the deformational modes. Harmonic analysis reveals the system's transmissibility response. The modal evaluation of automotive lamps has previously been performed by Kharche et al. (2016) and Molina-Viedma et al. (2018). In this study, the numerically obtained modal properties and harmonic response were validated with the experimental data. The numerical analysis was performed with ANSYS software. The experimental modal properties (mode shapes and corresponding frequencies) were determined with Polytec PSV-500 Xtra laser scanning head at frequency range of 10 to 1000Hz. The harmonic response experimental test was performed under room temperature using LDS V721 vibration shaker. The numerical results are validated with the experimental results.

2. Modal and Harmonic Response analysis

2.1. Modal analysis

The modal analysis method is known to be the most elemental of all dynamic analysis types, ANSYS-Module 03 (2017). Structural resonance and the deformational modes are important parameters in structure design, and they are estimated using modal analysis. Modal analysis gives an idea of how a structure will respond to a given dynamic load and it forms a prerequisite in the frequency based random vibration analysis as it helps in determining solution controls. A detailed overview of modal analysis is given by McConnell (1995). In modal analysis, it assumed that the structure is linear (the stiffness matrix [K] is constant) and no external loading. The modal governing equation is given in equation (1):

$$[M]\{\ddot{X}\} + [K]\{X\} = 0 \tag{1}$$

Normally, the solution of equation (1) has the form:

$$\{X\} = \psi_i \sin(\omega_i t + \theta_i) \tag{2}$$

where ψ is the amplitude vector, ω is the harmonic response frequency and θ is the phase angle and i is the mode number. If we differentiate equation (2) twice, we have equation (3):

$$\{\ddot{X}\} = -\omega_i^2\{\psi\}_i \sin(\omega_i t + \theta_i) \tag{3}$$

Substituting equation (2) and equation (3) in equation (1) leads to equation (4):

$$([K] - \omega_i^2[M])\{\psi_i\} = \{0\} \tag{4}$$

$\{\psi\}_i$ and ω_i are eigenvectors (mode shapes) and eigenvalues (natural frequencies)

More information on the theory of modal analysis can be obtained from many vibration textbooks including Ewins (1995).

2.2. Harmonic response analysis

Harmonic dynamic response analysis is significant in understanding the mechanical vibration performance of a system. In order to design a robust structure, engineers seek to have a good understanding of the performance of the materials involved in the design, Okeke et al. (2019). Harmonic analysis provides the steady state response of a structure to harmonic loads. The analysis reveals the system's resonance frequency which is an important parameter in the system design. The characterisation of the harmonic response of a system is normally done by sine sweep – from low to high frequency, covering frequency of interest. The theory governing sinusoidal harmonic response is given in equation (5):

$$[M]\{\ddot{X}\} + [C]\{\dot{X}\} + [K]\{X\} = \{F\} = \{f sin\Omega\} \tag{5}$$

where $\{X\}$, $\{\dot{X}\}$ and $\{\ddot{X}\}$ are nodal displacement, nodal velocity, and nodal acceleration vectors, respectively. $[M]$, $[C]$ and $[K]$ are mass, damping and stiffness matrices and $\{F\}$ is the applied force. The load $\{f sin\Omega t\}$ represents the applied sinusoidal harmonic load. More information on the harmonic response analysis can be found in ANSYS-Module 06 (2017) and in many mechanical vibration textbooks including Rao (2011).

3. Numerical simulation

Numerical modal analysis and harmonic response analysis were performed using ANSYS finite element software. Mode shapes and corresponding modal frequencies of the lamp assembly were obtained from modal analysis while transmissibility response of individual parts was obtained from harmonic response analysis.

3.1. Model geometry and its materials

A unique simplified lamp assembly solid model geometry was designed using Catia V5 software. The design is such that extraneous complexity does not influence the assessment of the dynamic performance of the assembly. All the key components are designed with polymer materials, including housing, bezel, outer lens, and optical lens, and the heatsink is represented with structural steel to add weight to the assembly. The designed lamp assembly solid model and the cut-out geometry showing all the key components are given in Fig. 1, and the components details including the materials and the weights are shown in Table 1.

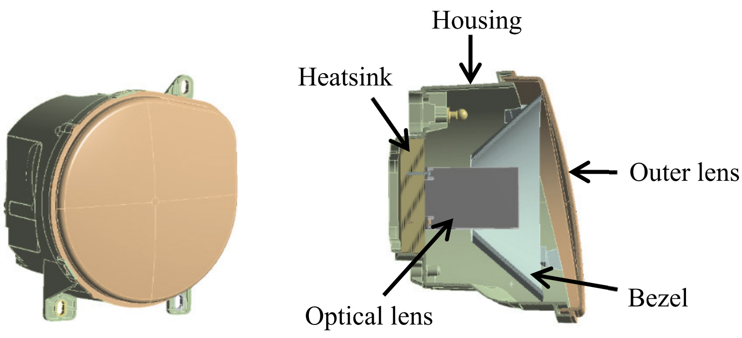
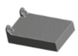
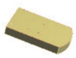


Fig. 1: Lamp assembly solid model and the cut-out model geometry

Table 1 Lamp assembly components information

Component	Component name	Material	Weight (g)
	Housing	PPT40	354.7
	Bezel	PC	112.8
	Outer lens	PC	167.2
	Optical lens	PMMA	72.9
	Heatsink	Steel	1953

3.2. Boundary conditions (restraints and loading)

The lamp CAD model was mounted on the fixture. The fixture was constrained on the eight crew holes on the base plate (fixed support) and 1m/s^2 acceleration was used to base excite the assembly at the fixture fixed support from 10 to 1000Hz frequency, see Fig. 2.

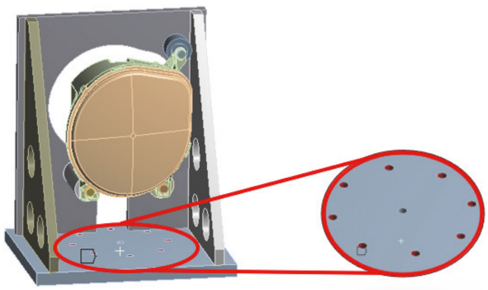


Fig. 2: Lamp assembly mounted on the fixture – eight fixing holes on the base plate highlighted

4. Experiments

4.1. Physical model of the lamp assembly

The manufactured lamp components are shown in Fig. 3. The housing and outer lens were manufactured at Wipac using injection moulding process. The bezel and optical blade were made by machining, and the steel representing heatsink was cut-out of a steel sheet. The lamp components were assembled using different techniques. The steel mass representing the heatsink was bonded to the housing using automotive based two parts epoxy resin. Optical blade was mounted to the heatsink using two screws with 0.5Nm tightening torque. The bezel was fixed to the housing using 4 screws with 0.7Nm tightening torque. The outer lens was bonded to the housing using hot melt glue. Fig. 4 shows the assembled lamp without outer lens (A) and assembled lamp with outer lens (B).

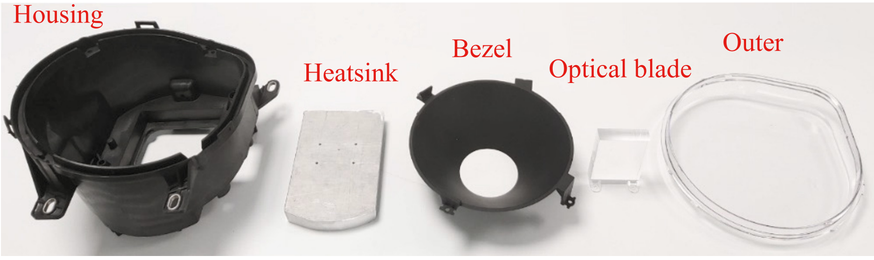


Fig. 3: Physical components of the lamp assembly

Fig. 4: Assembled lamp, A: without outer lens, B: with outer lens

4.2. Modal testing

The modal testing was carried out at Polytec Ltd, UK. The lamp assembly was suspended using an elastic cord to form a free-free arrangement. The cord was tied to both of the two lower fixings of the lamp and allowed to hang down from the ceiling. A vibration shaker LDS V201 was used to excite the lamp assembly through a force gauge PCB 208C01 connected to the upper lamp fixing which also measures the exciting force going into lamp. The force gauge was fixed to the upper fixing of the lamp and connected to the vibration shaker. The lamp was excited from 10 to 1000Hz frequency and the response characteristics (velocity and mode shapes) were measured with Polytec PSV-500 Xtra scanning head. Three scanning heads were used to capture the complex three-dimensional velocity of the lamp assembly. Mode shapes from different views of the head array positions were stitched together to form a single file. The complete test set-up is shown in Fig. 5.

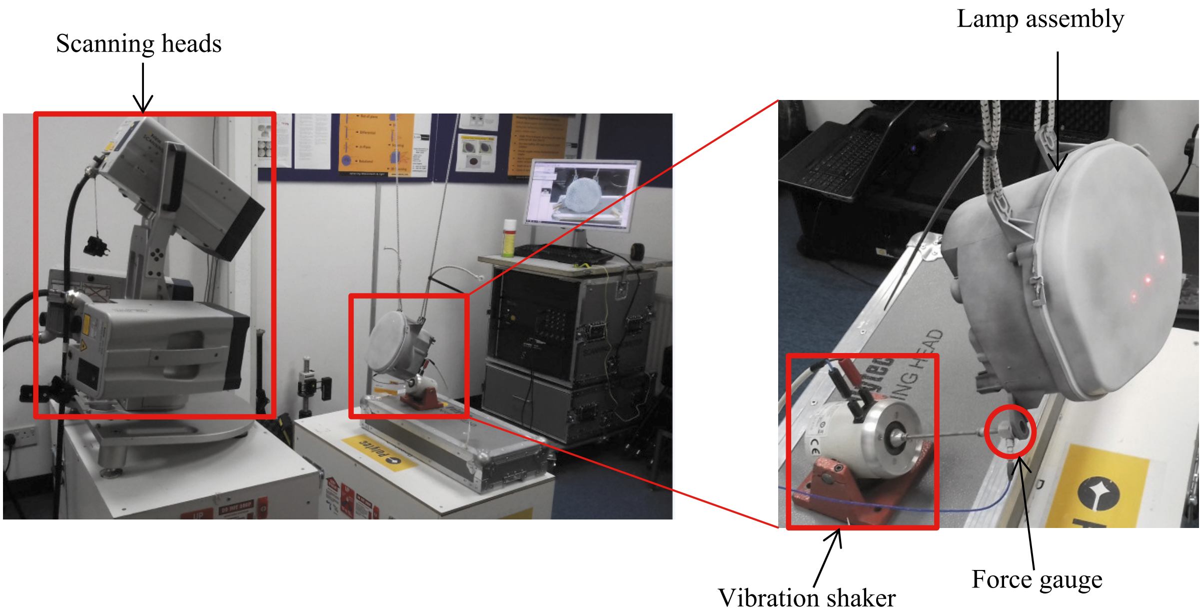


Fig. 5: Laser vibrometer set-up for experimental modal testing

4.3. Harmonic response testing

The harmonic response experimental test was performed under room temperature using LDS V721 vibration shaker. The shaker was driven by LDS 5KVA Spak Power Amplifier, and controlled with LDS laser USB controller. The lamp assembly mounted on a test fixture with 5Nm torque was placed on a shaker and excited from 10 to 1000Hz frequency along the lateral direction (x-axis). The shaker was driven with a constant sine sweep input acceleration of 1g, and the response was measured with a PCB Piezotronics 352C22 miniature accelerometer. The test set-up is shown in Fig. 6.

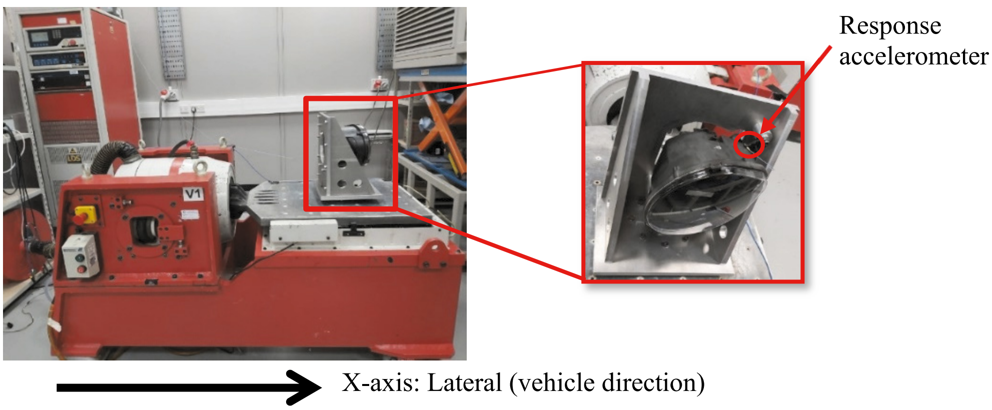


Fig. 6: Harmonic and random vibration shaker test set-up

5. Results and discussion

5.1. Modal validation

Figure. 7 shows each of the four mode shapes and corresponding frequencies obtained from the experimental testing and numerical simulation, respectively. Row A is the simulation results and row B is the experimental results. As both methods have different boundary condition, the presented mode shapes are those that are independent of boundary condition – they are not likely to be affected by the support. It can be seen that the mode shapes obtained with finite element are in good agreement with the experimental mode shapes at every frequency, and the modal frequencies of both methods show very little error as can be seen in Table 2. The percentage error between the frequencies of both methods ranges from 0.002 to 0.95%. Mode shapes are normally determined by material properties and the boundary conditions and they are very crucial for engineers in understanding the natural vibrational behavior of a structure. Furthermore, validating the numerically obtained mode shapes with the experimental mode shapes is essential prior to proceeding with harmonic analysis, Ewins, 1995, Gadwal, et al 2019.

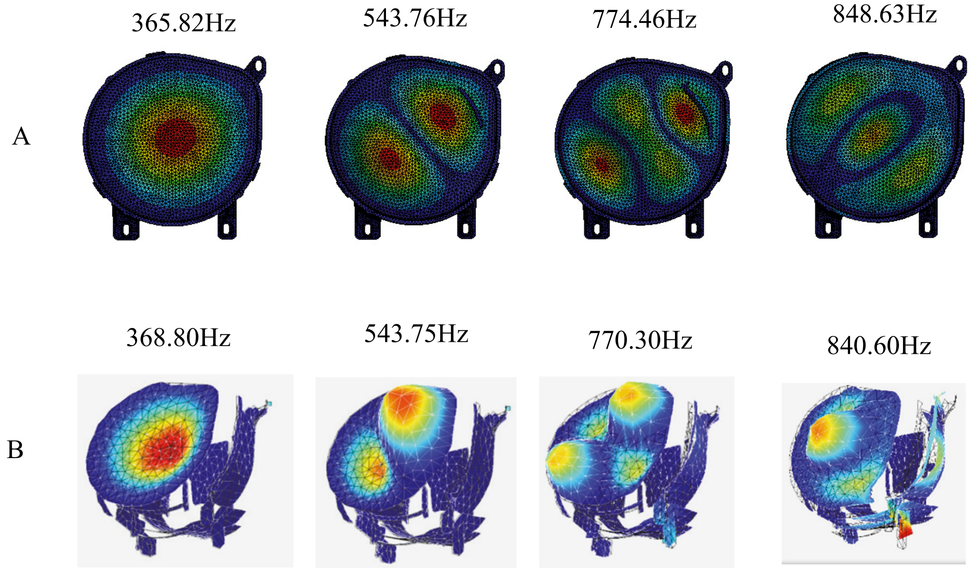


Fig. 7: Mode shapes and corresponding frequencies, A: Simulation; B: Experiment

Table 2: Percentage error between numerical and experimental modal frequencies

	Mode 1	Mode 2	Mode 3	Mode 4
% Error	0.815	0.002	0.537	0.946

5.2. Harmonic response validation

In this section, the finite element based frequency response was validated with the experimentally obtained frequency response. The numerical transmissibility response curve of each of the key components, housing, bezel, optical lens, and external lens of the lamp assembly was plotted together with that obtained experimentally, see Fig. 8. It can be seen that for housing and bezel components, the first four resonant frequencies of the simulation and experiment are in good agreement. These resonant frequencies fall within 10 to 240Hz frequency range. After 240Hz frequency, the transmissibility curves of both methods show discrepancy. This is also the same for the optical lens and outer lens, however, the frequency of which the discrepancy starts is different. For optical lens, the discrepancy starts from 203Hz while it starts from 180Hz for the outer lens. This discrepancy in transmissibility response of the components is attributed to the fixing of the components. When the tightening torque is applied in bolt tightening of the components and mounting of the assembly to the test fixture, the components / the assembly will deform. This tightening torque induced deformation cannot be captured in simulation. The effect is basically seen at higher frequency which is where there is increase in deformation which results in change in stiffness. The test fixture dynamic response will also play a role in the transmissibility response discrepancy.

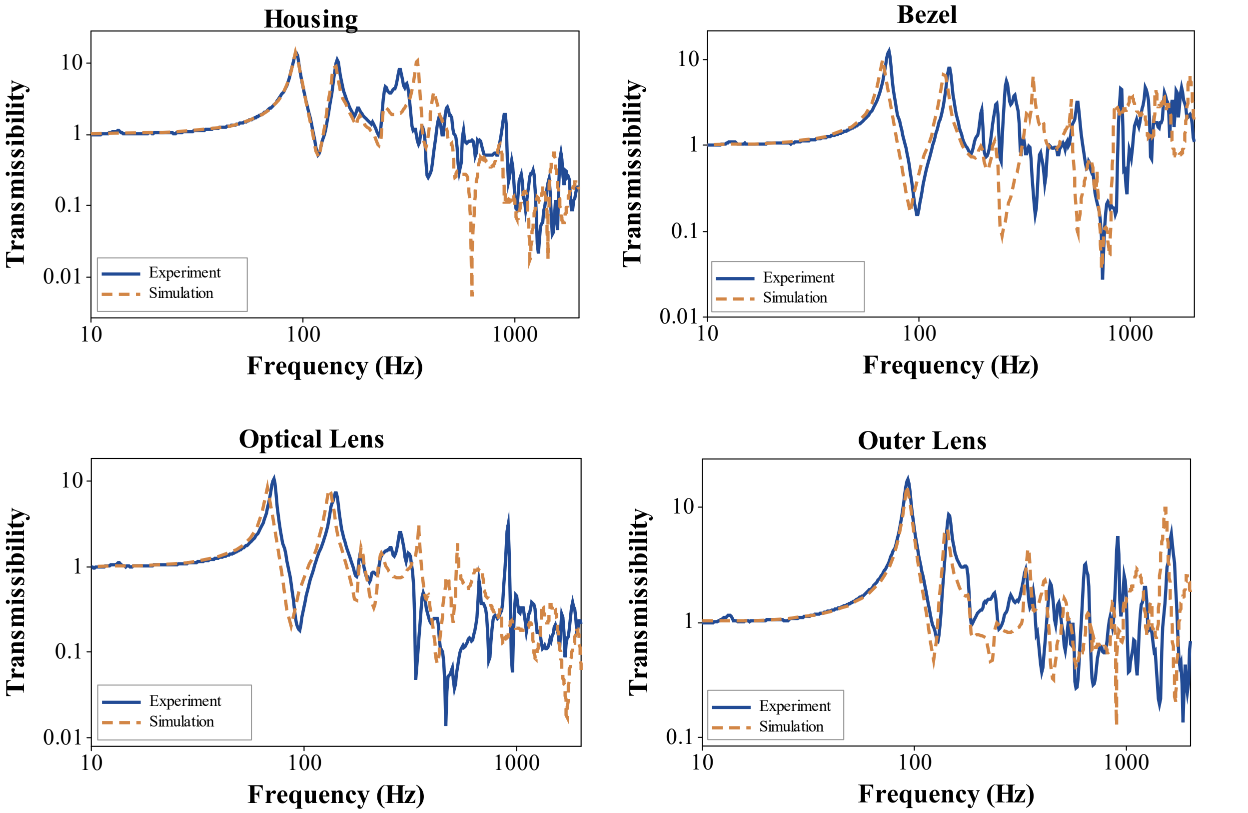


Fig. 8: Vibration Transmissibility of the LAMP assembly components

The lamp's mode shapes corresponding to the first three resonances of Transmissibility are shown in Fig 9. The vibrational mode shape defines deformation pattern at a particular resonance frequency. The first mode shape shows that the whole assembly is under bending condition – vibrating about the upper and lower left lamp mounting brackets at 92.9Hz frequency. The second mode shape also shows that the assembly is under bending mode, vibrating along lateral direction (X-axis) at 140.29Hz frequency, however, the maximum deformation is located at the bezel. For the third mode shape, there is combination of tensile/compression and bending modes, the assembly is vibrating at 181.99Hz frequency with respect to longitudinal direction (Y-axis). The mode shapes have clearly shown that we are dealing with bending related problem, therefore subsequent analysis should take this into account.

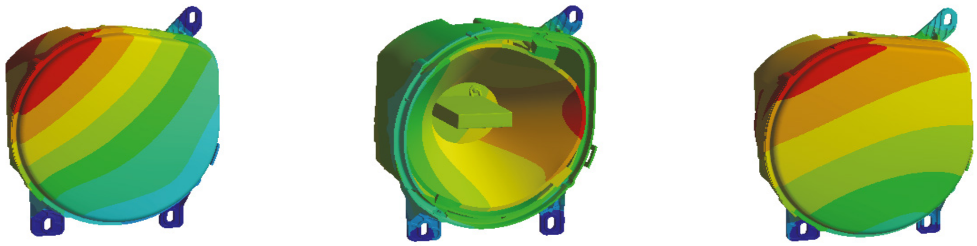


Fig. 9: First three numerical mode shapes in-situ boundary condition and corresponding resonant frequencies

6. Conclusion

Validation of the numerical dynamic behaviour of an automotive lamp assembly provides confidence on the subsequent numerical analysis such as vibration fatigue. Dynamic response of an automotive lamp assembly was validated using modal response properties (mode shapes and corresponding frequencies) and harmonic transmissibility response. The validation of the mode shapes and corresponding frequencies provided high degree of accuracy. Furthermore, the mode shapes showed that the lamp assembly was mostly vibrating in bending, therefore subsequent analysis should take this into account. Harmonic response validation showed that the first few numerical resonant frequencies that dominate the response compared well with experimental data, this was the case for all the components of the lamp assembly.

Acknowledgements

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