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Determination of Natural Frequency and Mode Shape for Hollow Thin Plates

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Abstract*: The determination of natural frequency and the mode shape of structures are important in characterizing the properties or behavior of the structures. The existing hollow in solid thin plate will change the vibration behavior in term of resonance frequency and mode behavior. A lot of structure consists of hollow space such as inner panel of car hood and bonnet. However, simplifying the model from hollow structure with solid thin plates only will resulting some difference in their vibration performance. This study begins with conducting modal analysis experiment using solid thin plate under free-free boundary condition. Next, the finite element methods were performing to compare with the experimental results. The average error of 3.96% was obtain when compared which provide a good confident level with the simulation result. Next, the study continued by finding the natural frequency and the mode shape for the thin plate under various position of hole, number of holes and shape of hole by using FEM software. Finally, finite element was also being carried out on the simplified design model of the inner panel car hood based from the real product and compared the resonance frequency from the model with some vibration reference sources. Through this study, it is found that the diversity of the hole position, the increasing number of holes and the change of the hole shape play a role in changing the natural frequencies of a thin plate. Since there is a lot of hollow thin plates in the car part, designing every parts are important to make sure the NVH quality is improve and not opposite of it.

***Keywords*: natural frequency, mode shape, Modal Analysis, finite element method**

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I. INTRODUCTION

Thin plates are commonly used materials in many areas of technology including in civil engineering, mechanical, chemical and marine fields. Thin plate has a combination of lightness and strength to withstand heavy loads when it is properly designed. N.V.Divya and Rizman [1] state that the plates can be categorized as thin plates when the thickness ratio to the width does not exceed 0.1 and the maximum deflection is less than one tenth of the thickness. In the automotive field, thin plates are among the major components needed in the production of a car body structure such as car hood. The car hood is a thin sheets of pieces to protect the essential components of the car, including engines. Its main function is to facilitate access to the engine, battery and important parts of the car for the components' conservation. The car's hood design involves the basic resistance to the structure, the comfort of the conveyor as well as the reduction of noise during the rigging. There are many car hood designs produced in this automotive industry. The design of the skeletal car hood has a significant change in the natural frequency and also the shape of the structure. This is because vibrations can cause premature wear and tear on the components of the vehicle.

Now days, beside the engine performance and body design, the automotive industry invest a lot in improving the noise, vibration and harshness (NVH) quality including vehicle acoustics. However, NVH involved a vast range of frequencies thus a lot of researchers and technique required to tackle the issues [2-7]. Nesaragi et al. [8] noted that noise and vibration problems are difficult to control especially at the frequency range below 200 Hz. One of conventional popular method in NVH was known as modal analysis.

According to Owunna et al. [9], modal analysis is a method used to obtain the mode shape when a vibration excitation is imposed on a component. Mode shape should be known to determine the displacement or response of the component when the occurrence of vibration in the daily application. Simulation such as finite element method also widely uses in determine the mode shape analysis [10-13]. To solve the problem by using modal analysis, eigenvalue problems need to be known first. In addition, natural frequencies, ω and mode shape, $\emptyset$, should also be determined. The frequency extraction procedure uses eigenvalue techniques to extract the frequencies of the current system.

The eigenvalue problem for the natural frequencies of an undamped finite element model is:

$$(-\omega^2 M + K)\emptyset = 0$$

Where M is the mass matrix (which is symmetric and positive definite), K is the stiffness matrix (which includes initial stiffness effects if the base state included the effects of nonlinear geometry) and $\emptyset$ is the eigenvector (the mode of vibration). Natural frequency is one of the important aspects in dynamic analysis. In general, when an object or structure is initially initiated, it will generate vibration. When the vibration is generated, it tends to vibrate at a specific frequency or set of frequencies and it is known as the natural frequency. According to Puranik et al. [14] the natural frequency of a structure is the frequency at which the structure tends to vibrate by its self. Therefore, each structure will vibrate at its own natural frequency because the vibrations of different types of structures occur at different natural frequency. When a structure vibrates at the same frequency or near the natural frequency of the structure, it will produce a high vibration, which can cause damaged to the structure. This condition is known as a phenomenon of resonance.

Mode shape is also inherent property in every structure. It is due to the interaction between elasticity and inertia in a material. During resonance, the mode shapes formed dominates the entire vibration shape (deflection) on the structure at its natural frequency. The mode shapes resulting from resonance vibrations can be used to determine the deflection or reaction of a component under the influence of vibration in engineering applications. In this project, the dynamic properties of thin plates were studied by conducting the modal experiments and verified through finite element analysis using appropriate software. In addition, this study also examined the effect of the presence of the hole and its position on the natural frequency as well as the form of thin plates for different boundary conditions. At the end of the study, the thin plates for the car hood in the car was designed to assess the effect of natural frequency and shape on the panel.

II. METHODOLOGY

Experimental Modal Analysis

The first part of the study was to determine the modal analysis from experiment. Modal analysis experiment was conducted using an impact hammer in semi-anechoic chamber laboratory and data acquisition systems (DAQ) was used to detect acceleration data obtained by the accelerometer. The setup of the experiment was shown in figure 1(a). The data was stored in the Signal Express software and the analysis was done by using the Matlab programming. The thin plate used was stainless steel with free-free boundary condition. Table 1 shows the thin plate specification used for modal analysis of experiment and simulation. The sampling frequency was 2000 Hz with averaging measurement of 3.

Table 1 Properties of the plate with the type of boundary condition.

Information	Detail
Geometry	450 mm x 200 mm x 5mm
Material	Stainless steel
Density	7750 kg/m^3
Boundary Condition, BC	Free-free condition along all edges
Poisson's ratio	0.31

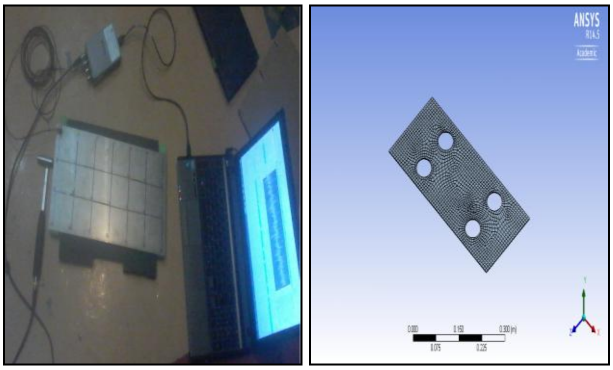


Figure 1 a) Left-Set-up for Modal Analysis experiment b) right- example of meshing element for the model

Finite Element Method

The second part of the study was to determine the modal mode shape and resonance frequency using ANSYS software as the finite element method. Figure 1(b) show example of the model with the meshing elements. All the properties and boundary condition was set up according to table 1 properties. The element type used was tetrahedron with element size of 5mm. There was three type of simulation done for finite element analysis

(i) Solid thin panel with free-free boundary condition

The first simulation was done using the solid thin panel with free-free boundary condition. The results from this simulation later will be compared with result from experimental modal analysis. This will become the validation stage to ensure the simulation planned is behave as expected as experimental result.

(ii) thin panel with various position of holes and hollow under free-free and clamped-clamped boundary condition

Four type of panel with different hole position (one in the middle, four in the middle symmetrical, one at the edge and two at the edge) were done using the solid thin panel. The simulation was run under free-free boundary condition and clamped-clamped boundary condition.

(ii) Two simplifying model of inner car hood panel under free-free and clamped-clamped boundary condition

The third simulation involved simplifying inner car hood panel based from a real part. Two model was built with Model A designed near to the pattern of the real part, and model B, designed with modification for the hole design as an alternative.

III. RESULTS AND DISCUSSION

After the experimental and simulation processes f have been conducted, the natural frequency values derived from the two methods of modal analysis are shown in Table 2.

Table 2 Natural frequency of experiment and simulation Modal Analysis

Mode No.	Natural /Resonance Frequency (Hz)		Error, %
	Experiment	Simulation	
1	37	38.2	3.29
2	63	52.7	16.30
3	110	106.2	3.41
4	197	194.7	1.16
5	217	216.7	0.10
6	227	226.4	0.22
7	291	300.4	3.26

Through Table 2, it was found that the natural frequencies obtained from the simulation more less same with the natural frequency obtained through the experiment. In addition, the percentage of errors that calculated by comparing the frequency values obtained through experiment and simulation as shown in Table 2 was only at 3.96% for the average percentage error. Therefore, the used of ANSYS software in solving the problem of modal analysis study was acceptable.

A. *The effect of the hole's presence under two different plate boundary condition.*

Table 3 shows the natural frequency of perforated thin plates for the first five modes when the thin plate boundary conditions change

Table 3 Natural frequency of perforated thin plate for first five mode

Boundary Condition	Position of hole	Mode shapes 1	2	3	4	5
Free condition (F-F-F-F)	Hole at centre	33.8	47.7	102.1	112.6	186.1
	Hole at one corner	39.2	55.5	107.3	121.2	194.4
	Hole at two corners	40.7	57.8	110.4	126.8	195.4
	Four hole at side	35.6	51.34	95.7	106.4	185.7
Clamp condition (C-C-C-C)	Hole at centre	243.7	257.2	417.9	460.1	557.5
	Hole at one corner	219.4	270.6	362.7	494.9	579.2
	Hole at two corners	220	273.3	369.0	506.6	580.1
	Four holes at side	216.2	265.7	350.1	464.3	580.9

Referring to Table 3, the natural frequency increases as the number of mode shapes increases for free and clamp boundary conditions. Thin plate with holes in two corners shows the highest natural frequency value for all mode shapes when compared with other hole positions whether the boundary plate boundary condition was free or constrained(clamp) at the whole boundary.

Figure 2 to Figure 6 shows the first five mode shape of thin plate of all the hole positions for boundary conditions are in free condition.

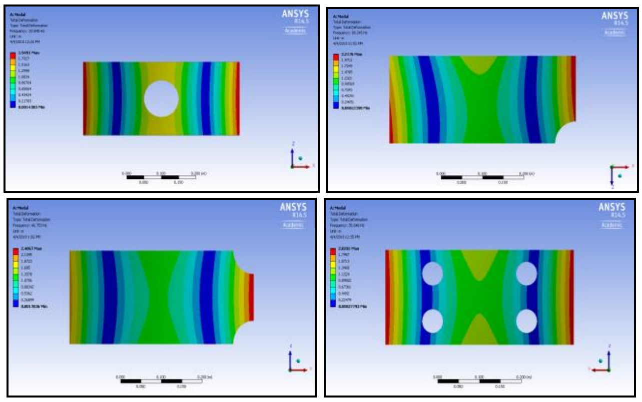


Figure 2 First mode shape for (a) hole at center, (b) hole at one corner, (c) hole at two corners (d) four holes at side

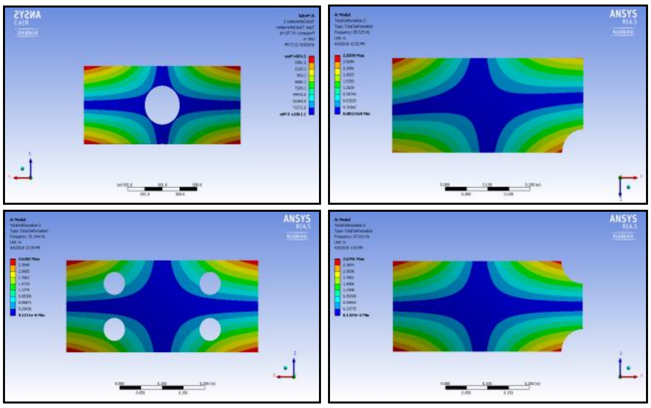


Figure 3 Second mode shape for (a) hole at center, (b) hole at one corner, (c) hole at two corners (d) four holes at side

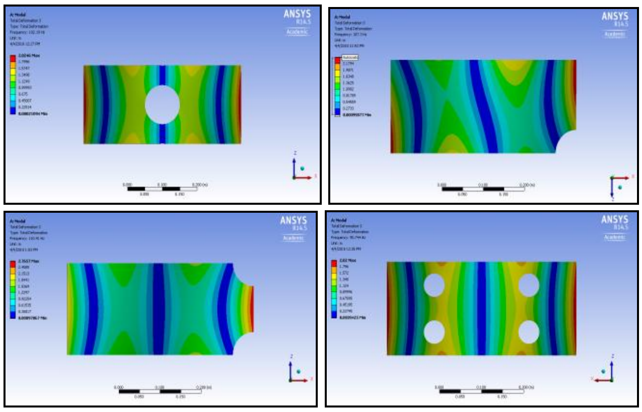
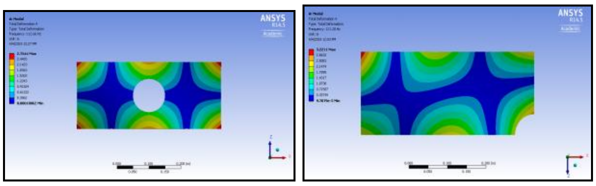


Figure 4 Third mode shape for (a) hole at center, (b) hole at one corner, (c) hole at two corners (d) four holes at side



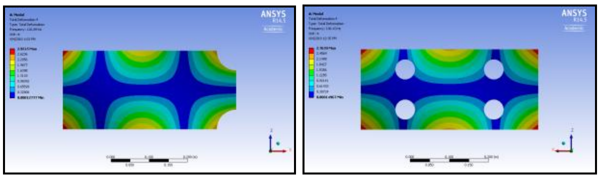


Figure 5 Fourth mode shape for (a) hole at center (b) hole at one corner, (c) hole at two corners (d) four holes at side

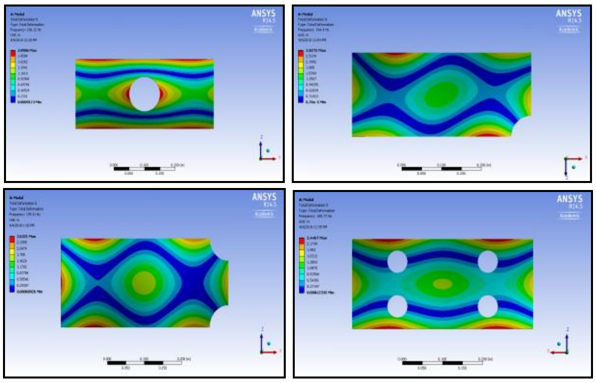


Figure 6 Fifth mode shape for (a) hole at center, (b) hole at one corner, (c) hole at two corners (d) four holes at side

As for overall, when the number of holes' increases at the centre of the plate, the natural frequency of the thin plate will decrease. However, when the boundary conditions change, from Table 3 it can be seen that the natural frequency value of a center hole in the middle is lower than the natural frequency of the four-holes thin plate in the mode shape of the second, fourth and fifth modes. Furthermore, it can be known that when the boundary conditions are free, the natural frequency of four holes at side thin plate show the lowest frequencies value of all mode shapes relative to the natural frequency of the other position of hole. Therefore, from this study, it is known that the presence of holes, changing in hole position and boundary condition affect the natural frequency of thin plates.

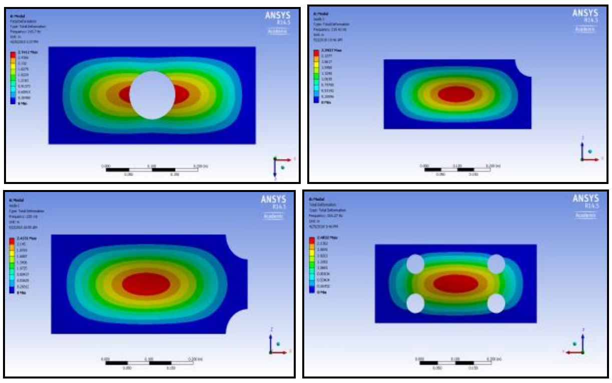


Figure 7 First mode shape for (a) hole at centre, (b) hole at one corner, (c) hole at two corners (d) four holes at side

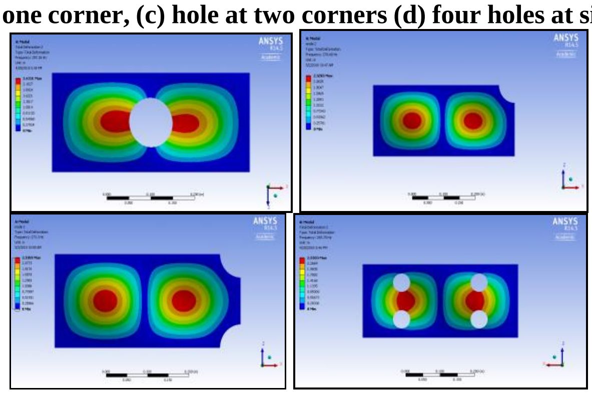


Figure 8 Second mode shape for (a) hole at center, (b) hole at one corner, (c) hole at two corners (d) four holes at side

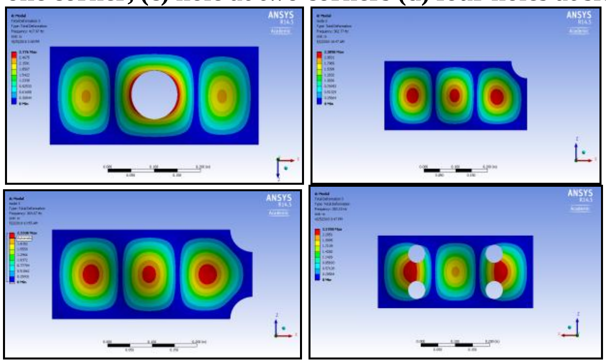


Figure 9 Third mode shape for (a) hole at center, (b) hole at one corner, (c) hole at two corners (d) four holes at side

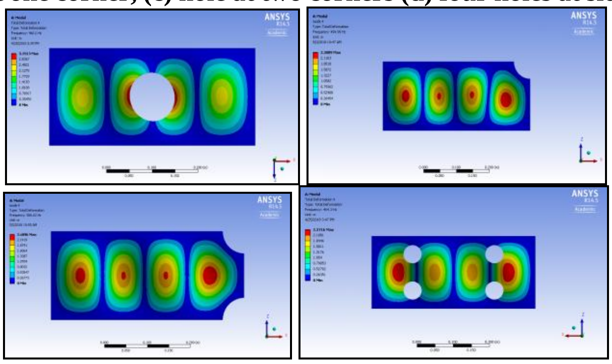
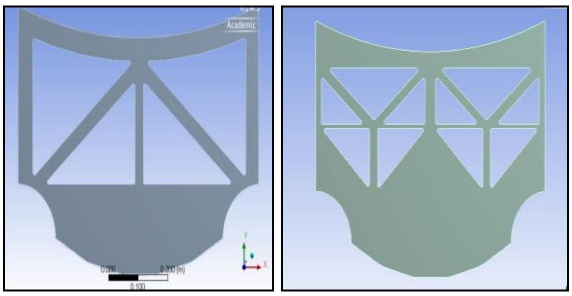


Figure 10 Fourth mode shape for (a) hole at center, (b) hole at one corner, (c) hole at two corners (d) four holes at side

Next, figure 7 to figure 10 shows the mode shape of thin plate when the boundary conditions are constrained entirely. When compared from figure 2-6 and 7-10, changing in the boundary condition will definitely gave a different mode shape and frequency. However, with the existing of hole and increasing number of holes, in general, the pattern of the mode shape was still look like the same. But, it will slightly change the deformation values and effect on the resonance frequency results.

B. *Analysis inner panel car hood*

After identifying the appropriate geometry of the hole to be modeled on the inner panel of car hood in the discussion above, 2 models have been designed. Figure 11 shows the two models of inner panel car hood used in this study. The model was develop based from the inner panel car hood available in market as shown in figure 12



Model A (b) Model B

Figure 11 Two example models of inner panel car hood

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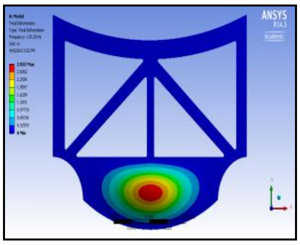
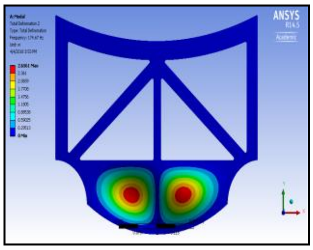
There are several factors to consider in selecting the panel design in the hood of the car is such as the vibration from critical car noise and vibration. Engine vibration (usually on engine excitation - frequency at cruise speed, 175 - 178 Hz and idle at 27 Hz), [4], foods' lunch shudder' and driveline boom phenomenon (10hz-22 hz), [3] and vibration due to wind noise (dominant at frequency 41 Hz, 83 Hz and 151 Hz), [16]. Table 5 shows the natural frequency of panels in the car hoods of models A and B with some of the source of the vibration in the car

Figure 12 Inner panel car hood, Ashley [15].

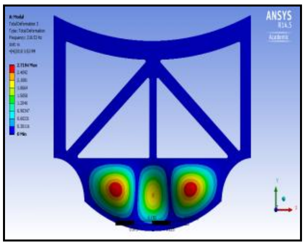
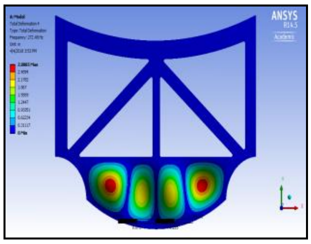
Table 5 Natural frequency of inner panel car hood for Model A, B

Mode shape	Model A	Model B	Engine vibration	Launch shudder	Driveline bloom	Wind noise
1	138.3	75.8				
2	174.7	127.6				
3	218.5	163.1				
4	272.5	189.8				
5	337.8	230.2				
6	365.2	268.8				
7	416.0	286.0				
8	428.0	305.6	175.0	10.0 15.0 22.0	53.0	41.0 83.0 151.0
9	494.3	366.7				
10	507.1	372.4				
11	569.8	405.8				
12	609.4	446.6				
13	651.8	458.6				
14	701.7	477.3				
15	721.0	522.1				

Referring to Table 5, there is no natural frequency for all two models A, B that are approaching the engine excitation frequency at 'idle speed'. However, for engine excitation frequency at cruise speed, 175 - 178 Hz, it was found that the panel design in the car hood of model A approaches this frequency value at second mode shape. When comparisons were made for the frequency of occurrence of phenomena on the driveline, there was a natural frequency of model B which was 75.4 Hz approaching the resonance frequency of 'driveline boom', 53 Hz. However, it was still considered far. When looked into other noise source, all the mode shape for both of the model is quite different and not matching. Hence, it was known that the hole design on the panel of model B is more suitable when designing the inner hood. Figure 13 and 14 shows the first four modes of the car hood panel for Models A and B.

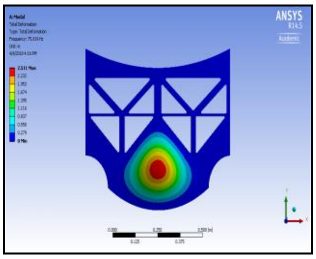
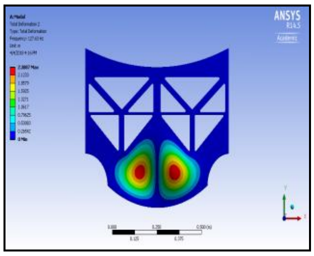



(a) First mode shape (b) Second mode shape

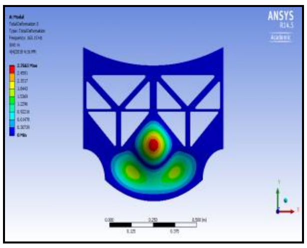
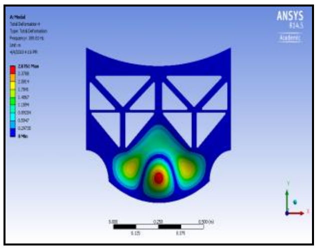



(c) Third mode shape (d) Fourth mode shape

Figure 13 First four mode shapes of Model A

(a) First mode shape (b) Second mode shape

(c) Third mode shape (d) Fourth mode shape

Figure 14 First four mode shapes of Model B

Based on Figure 13 and 14, the resulting mode shapes vary for these two models even though the boundary conditions are unchanged due to each model has its own design. The natural frequency for both of the model also noted a significant different value. However, this selection of the designed model only covers the scope of this study. This study shown, that designing the hollow location also need to considered to avoid experience any resonance frequency from other sources in the car.

IV. CONCLUSION

This research deal with modal analysis of isotropic thin stainless steel plate. The first part is about validation of numerical solution using the ANSYS program with experimental of modal analysis under free-free boundary condition.

With the small average percentage error of 3.96%, the results from the finite element analysis was accepted and the simulation process proceeded The next part is about comparison of a few hollow and edge condition under two different boundary condition, free-free and clamp-clamp for all the four edges. The final part is about simulate two simplify inner car hood panel based from a real inner panel. When comparing the resonance frequency of the inner car hood panel with some vibration sources (literature based reference), model A which more look like to the real inner panel noted some overlapping at 175 Hz. This overlapping frequency will affect the NVH for a car and possibility resulting damage to the structure when vibrate to its resonance. In the same time, model B doesn't demonstrate any overlapping with the vibration sources can provide some alternative design for the inner car hood panel. Thus, this research show that hollow in the thin plate will resulting changes in mode shape and resonance frequency. Furthermore, it is also important to determine their vibration characteristic others than the strength analysis in designing any structure or part such as the inner hood in car panel. It shows in model A design where the mode shape shows potential resonance clash with the engine vibration that will led to the material failure.

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