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THE EFFECT OF A HOOD SHEET METAL THICKNESS AND FAN ASSEMBLY POSITION ON SYSTEM MODE FREQUENCIES

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ABSTRACT

Reducing sound propagation in hoods, which is a common requirement in electronic kitchen appliances, is essential for enhancing user comfort. One significant source of noise in hoods is the vibration of the product panels. As the motor serves as the primary excitation source, it is crucial to shift the modal frequencies of the panels as far away as possible from the operational frequency of the motor. This study aims to investigate the effects of panel sheet thickness and fan volute position on modal frequency and panel vibrations. Initially, the existing geometry is modelled using finite element method and validated through frequency response function (FRF) experiments. In the second part of the study, the thickness of the product panels, which is initially 0.5 mm in the reference geometry, is varied to 0.4, 0.45, and 0.55 mm. The impact of this thickness change on the modal frequency and panel vibrations is examined. In the final part, the influence of changes in fan volute mass and position on the modal frequency is investigated. The study reveals that increasing the panel thickness results in an increase in the first mode of the panel and a decrease in acceleration amplitude. On the other hand, it is observed that the fan volute mass affects only the frequency, while not significantly impacting the amplitude.

Keywords: *Hood, Modal analysis, Modal shape, Finite element*

1. INTRODUCTION

In contemporary kitchen design, the trend towards smaller homes has led to the reduction or integration of kitchens with other living spaces like living rooms or dining areas. Consequently, range hoods are required to serve not only their conventional purpose of extracting cooking vapors and moisture but also to encompass novel functionalities [1]. This, in turn, has led to a greater emphasis on aspects such as visual design, their role as odor extraction ventilation devices, energy efficiency, and noise emission levels. In essence, modern range hoods must address a broader set of requirements to harmoniously fit into the overall kitchen environment while excelling in air extraction performance and meeting aesthetic and functional demands.

Figure 1: Inclined type hood model.

Modal analysis is essential for evaluating the mode shapes generated by a component under vibrational excitation. These mode shapes provide valuable insights into the displacement and response of the component when subjected to real-life vibration scenarios [2]. In this study, an investigation was conducted on the modal parameters of a 60 cm inclined-type hood under different geometric conditions to observe the best fit design for minimum vibration induced noise. The initial phase of the study involved developing a finite element model to represent the existing hood. Computational analysis was conducted on this model to determine the system's modal frequencies. To validate the accuracy of these computed modal frequencies, experimental hammer tests were performed on the actual product, and the measured system modal frequencies were

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compared with the computational results. This comparison between the computational and experimental modal frequencies served as a validation step, ensuring that the finite element model accurately represented the dynamic behavior of the hood system. The second phase of the study focused on investigating the impact of various parameters on the system's modal frequencies. Specifically, the parameters examined included panel thickness, fan assembly mass, and fan assembly position.

To analyze the effect of panel thickness, the current sheet thickness of 0.5 mm was evaluated along with three alternative thickness values: 0.4 mm, 0.45 mm, and 0.55 mm. Modal analysis was performed for each thickness to assess how it influenced the system's modal frequencies.

Furthermore, the study explored the influence of fan assembly mass on the modal frequencies. Three different fan assembly masses were considered: the current mass of 1650 g, as well as two alternative masses of 1000 g and 2500 g. Modal analysis was conducted for each mass configuration to determine the resulting modal frequencies.

Lastly, the positional effect of the fan assembly group was investigated by shifting the center of mass in two axes. By altering the position of the fan assembly, the study aimed to assess how changes in its location affected the system's modal frequencies.

Through these investigations, the study aimed to gain insights into how parameters such as panel thickness, fan assembly mass, and position influenced the modal frequencies of the system. This analysis would provide valuable information for optimizing the design and performance of the hood system.

2. MODAL PARAMETERS PREDICTION

In hoods, the main source of excitation is the motor. To minimize system vibrations, it is essential for the modal frequencies to be well-separated from the motor's operating frequency. This section aims to determine the modal frequencies and shapes of the existing hood product using both numerical simulations and experimental techniques. The numerical methods involve finite element analysis, while experimental tests will be conducted to validate the computed results.

2.1 Modal frequency prediction with Finite Element Simulation

The Siemens NX software was utilized to create a finite element model to investigate the resonance conditions between the body modal frequencies of a 60 cm wide inclined type hood and the operating frequency of its motor. The hood model consisted of various materials, including metal, plastic, and glass, with their respective material properties defined as specified in Table-1.

The solid representation of the hood model was depicted using 196,154 shell and 104,897 solid cells, ensuring a comprehensive and accurate simulation. Maximum cell dimension of glass and side panels are 2 mm, body is 3 mm and electronic card is 4 mm. Additionally, a fan assembly group was incorporated into the model as a concentrated mass. This mass was physically linked to the body at specific locations denoted as RB2 connections, establishing a mechanical coupling to realistically simulate the system's response to external forces and vibrations. Furthermore, other components of the hood, such as the glass and sheet metal, were connected to each other using a gluing connection to ensure their proper mechanical interaction within the overall model.

Table 1: Material properties.

Parameters	Sheet metal	Plastic	Glass
Mass density (kg/m^3)	7850	6000	2450
Young's modulus (MPA)	200000	2000	75000
Poisson's ratio	0.29	0.4	0.23

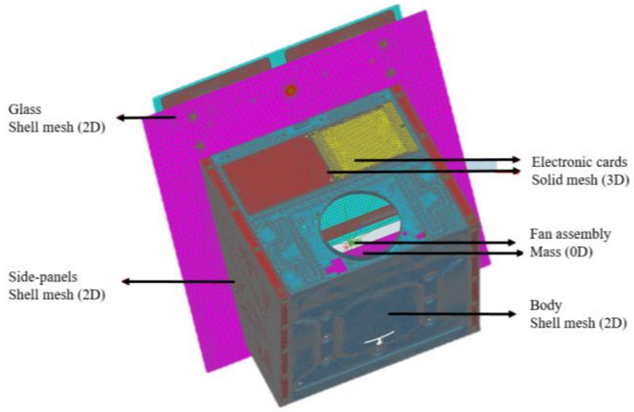
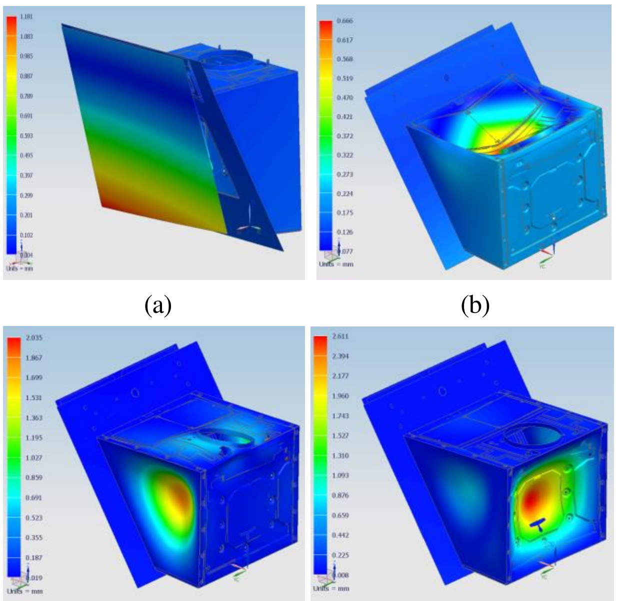


Figure 2: Cell types of system model.

Based on the results obtained from the finite element model, the first mode of the hood body was identified at 14 Hz, primarily localized in the glass region. Figure-3a illustrates the corresponding mode shape observed in the glass region. The second mode of the hood was observed at 22 Hz, predominantly in the upper panel region. Figure-3b displays the mode shape obtained from the numerical analysis for the upper panel. The side panels exhibited significant vibration amplitudes at 52 Hz, as illustrated in Figure-3c. Additionally, the back panel demonstrated notable vibration amplitudes at 67 Hz, as depicted in Figure-3d.



(a) (b)

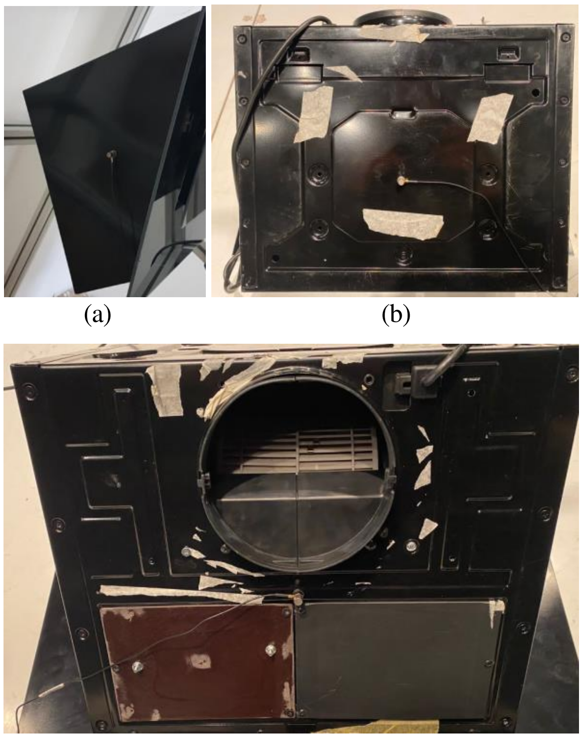
(c) (d)

Figure 3: Modal shapes at (a) 14 Hz, (b) 22 Hz, (c) 52 Hz, and (d) 67 Hz.

2.2 Computational model verification with impact hammer test

An impact hammer test was conducted on the side, back and top panels of the inclined type hood. This test involved using an impact hammer (Brüel & Kjaer 8206-002) to excite the panels and accelerometers (Brüel & Kjaer 4394) to measure their response. For the experimental study Brüel & Kjaer Pulse LabShop program was used with Pulse analyzer. Accelerometers were placed in the middle of the panels as shown in Figure 4. Compatible with the numerical model, the excitation was applied perpendicular to the relevant surfaces, and the response was read along the same axis.

Figure 5 presents the modal parameters of the side panel obtained through both computational analysis and impact hammer experiments. In the finite element analysis, two closely located modes were identified at 50 Hz and 52 Hz. Consistent with this, the first mode of the side panel was determined as 51 Hz through the hammer impact test. The subsequent modal frequencies obtained from the numerical solution ranged from 86 Hz to 94 Hz, whereas the experimental data exhibited frequencies in the range of 95 Hz to 98 Hz.



(a) (b)

(c)

Figure 4: Accelerometer positions on (a) side panel, (b) back panel, (c) top panel.

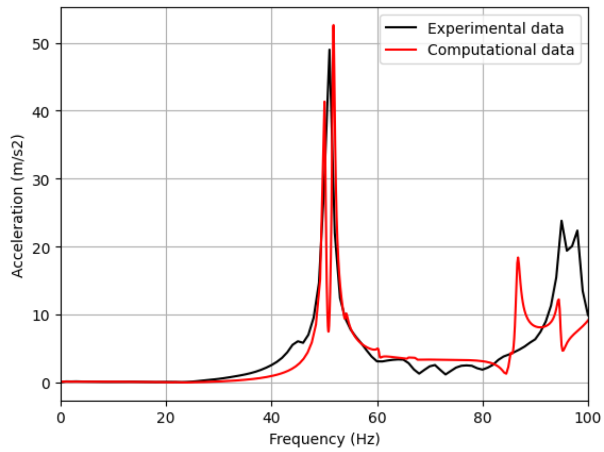


Figure 5: Side panel modal frequencies.

Figure 6 displays the modal frequencies of the back panel, comparing the numerical and experimental data. Both the computational analysis and the experimental measurements revealed surface modes at a frequency of 67 Hz.

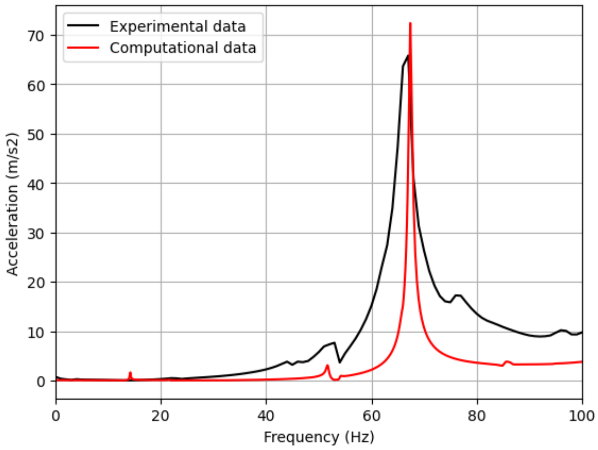


Figure 6: Back panel modal frequencies.

In the case of the upper panel, there is a slightly larger discrepancy between the numerical and experimental data compared to the other surfaces. However, despite this difference, it is noteworthy that the experimental mode frequency of 23 Hz closely aligned with the numerical solution, which reported a frequency of 21 Hz.

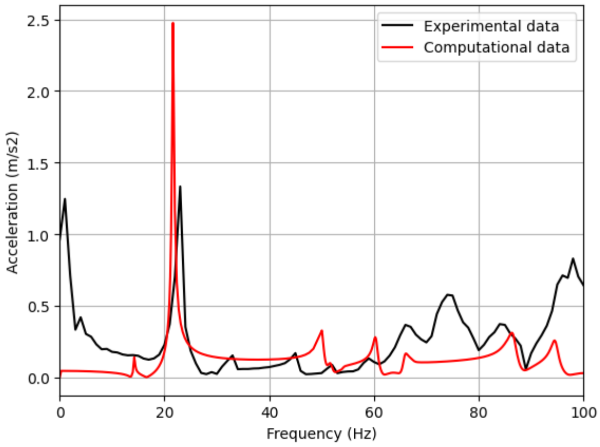


Figure 7: Top panel modal frequencies.

The numerical model has been successfully validated by comparing its results with those obtained from experimental modal analysis. With this validation, the study can now proceed to further investigate the effects of variations in panel thickness, fan assembly mass, and placement using the established and verified numerical model.

3. EFFECT OF SIDE PANEL AND FAN ASSEMBLY PROPERTIES

In this part of the study, it is aimed to investigate side panel thickness, fan assembly position and mass effect on system modal frequency so that an alternative design model deviating from motor excitation frequency can be detected.

3.1 Effect of side panel thickness

The original hood model had a panel thickness of 0.50 mm, resulting in a modal frequency of 52 Hz for the side panel. To investigate the effect of panel thickness on modal frequencies, updated finite element models were created with panel thicknesses of 0.40 mm, 0.45 mm, and 0.55 mm. Figure 8 illustrates the first modal frequencies obtained from these analyzed models. From the contour plot legends, it can be observed that as the panel thickness increases, the modal frequency also increases, ranging from 42 Hz to 56 Hz. Simultaneously, the corresponding amplitude at that frequency decreases. The maximum displacement, which is 2.05 mm for the panel with the original thickness, increases to 3.619 mm for the 0.4 mm panel, 3.480 mm for the 0.45 mm panel, and decreases to 1.096 mm for the 0.55 mm panel.

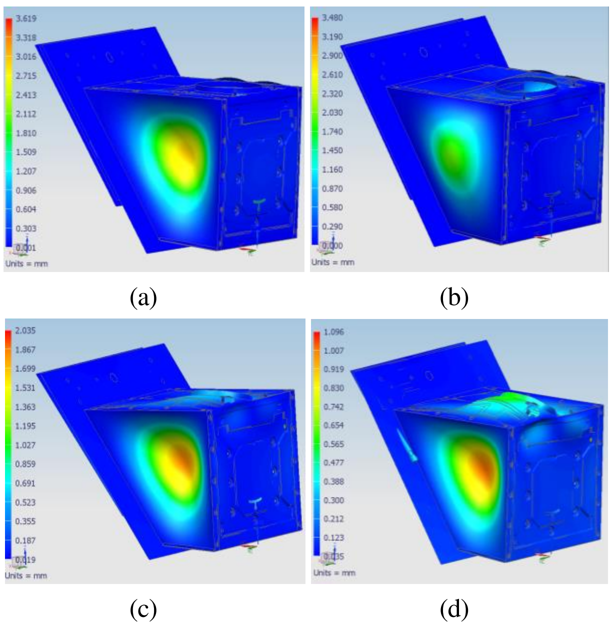


Figure 8: (a) 0.4 mm panel modal shape at 42 Hz, (b) 0.45 mm panel modal shape at 47 Hz, (c) 0.50 mm panel modal shape at 52 Hz, (d) 0.55 mm panel modal shape at 56 Hz.

3.2 Effect of fan assembly mass and position

In the current finite element model, the fan assembly, comprising a fan and a motor shaft, was represented as a concentrated mass of 1650 grams. In the initial modal analysis conducted using this mass, the frequency of the upper panel, to which the fan assembly is attached, was determined to be 22 Hz.

Further analyses were carried out to explore the effects of different fan assembly masses. These analyses included configurations with masses of 1000 grams and 2500 grams, positioned below and above the original 1650 gram mass, respectively.

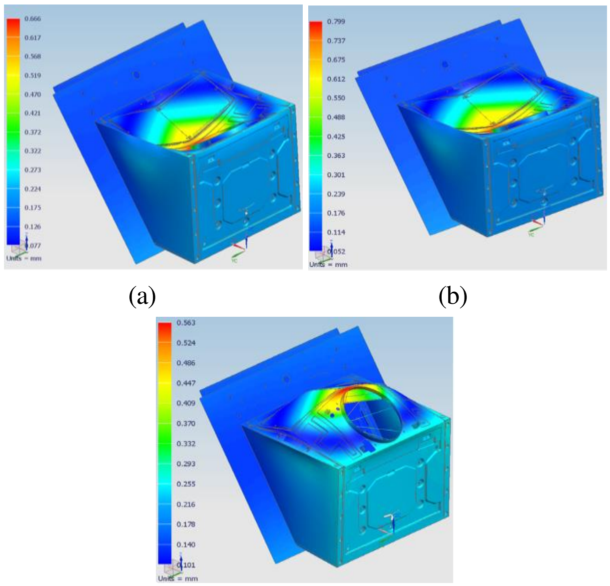
Figure 9 showcases the first modal frequencies and amplitudes of the system obtained from the analysis conducted for three different fan assembly masses. According to the results, as the weight of the fan assembly increases, the system modal frequency decreases from 22 Hz to 18 Hz. Moreover, at these respective frequencies, the amplitudes are highest for the 1650 gram mass with 0.799 mm.

In addition to the variation in fan assembly mass, the study also investigated the effect of fan assembly placement on the system modes. Analyses were conducted for three different positions, including the current position.

In the first design modification, the center of mass of the fan assembly was shifted by 40 mm in the positive x-axis direction, bringing it closer to the back panel. As a consequence of this change, the modal frequency of the upper panel increased from 22 Hz to 24 Hz. However, the displacement amplitudes at these respective frequencies did not exhibit significant changes.

In the second position modification, the fan assembly was shifted 50 mm in the negative z-axis direction, away from the upper panel. This change in position resulted in a decrease in the modal frequency from 22 Hz to 17 Hz compared to the original position. Additionally, the amplitude at this frequency decreased by 0.142 mm. The analysis results of the modified model are depicted in Figure 11.

In the final position modification examined, the center of mass of the fan assembly was shifted 30 mm in the positive z-direction, resulting in its closer proximity to the upper panel compared to the original model. With this change, the first modal frequency of the upper panel increased from 22 Hz to 24 Hz. Additionally, the displacement amplitude decreased by 0.072 mm.



(a) (b)

(c)

Figure 9: (a) 1000 g fan assembly at 25 Hz, (b) 1650 g fan assembly at 22 Hz, (c) 2500 g fan assembly at 18 Hz.

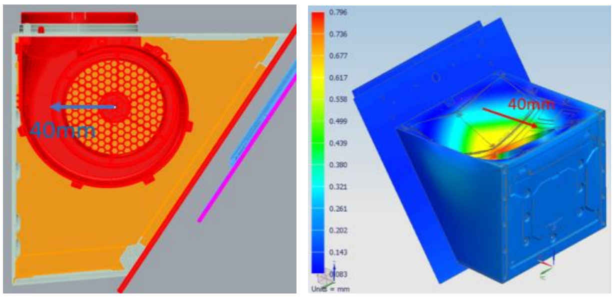


Figure 10: 40 mm removed fan assembly along +x direction.

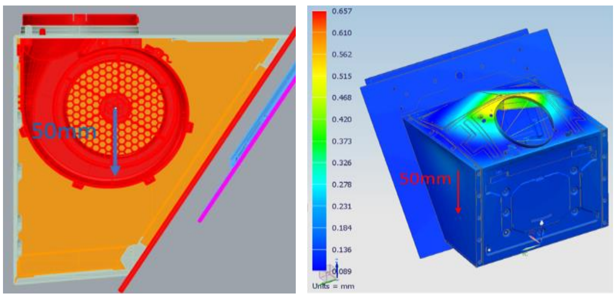


Figure 11: 50 mm removed fan assembly along -z direction.

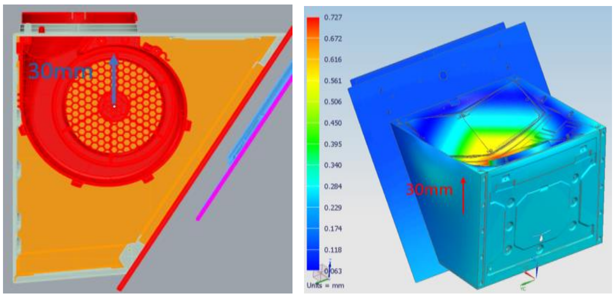


Figure 12: 30 mm removed fan assembly along +z direction.

4. CONCLUSION

In the present paper, the effect of panel thickness, fan assembly mass, and position on the modal behavior of a commercially available 60 cm inclined-type range hood has been investigated.

In the first section of the study, the existing product was modeled using the finite element method in the Siemens NX environment, and this model was validated using experimental impact hammer test results within the 0-100 Hz range.

In the second section, using the validated model, variations in panel thickness, fan assembly mass, and position were introduced, and the system modal frequencies and displacements at these frequencies were examined. The panel thickness, which is currently 0.50 mm, was investigated in the range of 0.40-0.55 mm. The analysis results showed that as the panel thickness increased, the modal frequencies of the side panels increased from 42 Hz to 56 Hz, while the displacement amplitude decreased from 3.6 mm to 1.1 mm.

In the current product, the fan assembly mass is defined as a concentrated mass of 1650 grams. Using the validated finite element model, the modal frequencies were examined by updating this mass to 1000 grams and 2500 grams. According to the computational results, as the mass increased from 1000 grams to 2500 grams, the modal frequency of the upper panel decreased from 25 Hz to 18 Hz. Additionally, the highest displacement, observed in the original case solution with a mass of 1650 grams, was 0.799 mm.

Lastly, the effect of fan assembly placement on the system was examined. Modal analyses were performed for three positions in addition to the current position. In the first model, the center of mass was moved 40 mm closer to the rear panel. According to the solution of this model, the modal frequency of the upper panel increased from 22 Hz to 24 Hz, but no significant change was observed in displacement. In the second model, the center of mass was moved 50 mm away from the upper panel. In this case, the modal frequency of the upper panel decreased from 22 Hz to 17 Hz, and the displacement decreased by approximately 0.142 mm. In the third and final model, the center of mass was moved 30 mm closer to the upper panel. As a result, the modal frequency of the upper panel increased from 22 Hz to 24 Hz. Additionally, there was only a 0.07 mm change in displacement at the respective frequency.

In the examined product, the operating frequency of the motor was 50 Hz. During the study, it was aimed to move modal frequencies of the system as far as from the driving frequency to decrease structural vibration and vibration caused noise. In the finite element analysis explained in this study, it was observed that the parts with model frequency closest to 50 Hz are the side panels and altering the side panel thickness specifically affected their modal frequencies and amplitudes. Based on all this information, it is considered that changing the side panel thickness is the most practical approach to reduce structural vibration of the system.

5. ACKNOWLEDGMENTS

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