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Experimental Modal Analysis of Radiator Fan Module to Predict its Influence of Structural Characteristics on Vibration and Noise Contribution

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A B S T R A C T

Among many accessories of a passenger car, engine cooling Radiator Fan Module (RFM) is an important source of noise and vibration. Modal parameters provide the basis for the noise generation mechanism and noise propagation phenomena. The objective of the current research work is to characterize RFM through its modal behavior to investigate the contribution of structural vibration to noise radiation during its operating state. Hence, the experimental modal analysis and the noise measurement of RFM were carried out. Additionally a finite element modal analysis of RFM was performed to validate the finite element model that can be used for vibro-acoustic and computational fluid dynamic studies. Below 500 Hz total of eight critical modal frequencies, three for fan and five for shroud are determined and between 500 Hz to 2000Hz seven modes for fan and fifteen modes for shroud were dedermined. From the correlation study, it is witnessed that the contribution of modal behaviour for noise radiation is observed with 20 dB(A) variation below 500 Hz, along the diagonal grid points whereas beyond 500 Hz the variation was negligible. Importantly, blade passing frequencies corresponding to the fan speeds between 500 to 3000 rpm were compared with resonance frequencies obtained and they do not conicide with modal frequencies.

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1. INTRODUCTION

Automotive Noise Vibration Harshness (NVH) is the main focus rendered by automotive industries due to tight market competition in attracting and retaining the coustomers. There are many sources of noise and vibration such as power train and transmission, tire road interaction and wind during high speeds. However, these are pronounced during mid or high speeds. During slow speed or waiting in the traffic signals, the important sources are accessories of the vehicle. Though these accessories are masked by other predominent sources one can not neglect during aforementioned situations. Among all the accessories of car, engine cooling RFM is one important system that takes care of preventing engine from over heating by pushing or pulling the air through the radiator. During its functioning, it creates tonal and braod band noise that increases with increasing speed of the fan. Air flow dynamics over the fan shroud and blades of the fan is the reason for vibro-acoustic phenomina and that forms the basis for the noise generation mechanism. Hence it is motivated to investigate structural energy transfer characteristics of RFM to identify its contribution considering it as an important noise source.

Zhu et al. [1] adopted hybrid procedure based on computational fluid dynamics and computational aero acoustics numerically to investigate automotive cooling fan sound. They considered the role of fan shroud for sound propagation followed by boundary element method to determine aerodynamic noise field of the cooling fan module. Modal behavior forms the basis to characterize structural vibrational behavior in terms of modal parameters which is essential for vibro-acoustic analysis. Liu et al. [2] investigated with the help of six

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degree of freedom model of an automotive condenser radiator fan module for its vibration and noise issues and concluded that the mount stiffness and natural frequencies influence the same. Baniasadi et al. [3] carried out computational aerodynamics study on cooling fan of an engine and concluded that the pressure loss depends on blockage distances and vehicle speed. The article published by Lim et al. [4] presents the importance of mass unbalance of a radiator cooling fan for its vibration during vehicle idling condition. Vibration signals of the fan blades were experimentally measured both with and without an added mass in their study. Kumar et al. [5] correlated the vehicle body vibration with radiator fan module vibration and have proved that the radiator fan module structural vibration and their modal behavior play an important role in vehicle NVH. Arefin and Islam [6] considered noise radiation characteristics as one of the validating factor for a proposed micro gas turbine for an electric truck. Rajabi et al. [7] examine the effects of blade design of an axial flow fan used for cooling electronic devices.

Wang et al. [8] and Suzuki et al. [9] proposed on noise control methods for cooling fan of a simple computer and automotive radiator fan respectively. There are many challenges to understand the noise radiation mechanisms and predication of structural behavior of the components. Jayachandran et al. [10] carried out component level simulations and testing to predict contact between fan motors mounted on fan shroud and radiator; then proposed as a requirement in designing a fan shroud to eliminate damage due to inertial movement of fan motors in low speed impacts. At certain frequencies the fan shroud assembly will resonate, and these resonating frequencies indicate the higher amplitude noise. From modal analysis, the modal frequencies, and mode shapes describe the dynamic behavior of the radiator fan and form the basis for noise generation mechanism. Hence there is a need to understand the mode shapes of the radiator fan and shroud. Authors of this paper motivated to read and understand the theory and application of modal testing from the classical text book written by Ewins [11]. Subsequently the experimenental modal analysis procedure is understood from the work published by Nijmeijer, et al. [12], Patil et al. [13], and Vervoort et al. [14] on tire modal anlaysis. There were no specific papers on experimental modal anslysis on RFM. The experimental procedure learnt from tire modal analysis is employed on RFM in the current work. Tare et al. [15] used computational fluid dynamics approach to propose an effective noise management technique for radiator cooling fan module of a passenger vehicle.

Three different setups of noise measurement were carried out, namely free space, wallmounted, and in vehicle measurement by Lu [16] in his very recent experimental research studies on installation effects on cooling fan noise. With the help of spectral decomposition technique the source strength and the energy propagation effect of the fan vibration is quantified and reported the modal behavior forms the basis of energy transfer. Application of analytical methods for the solution of free vibration study is possible theoritically. Bakhshi Khaniki and Hosseini Hashemi [17] carried out free vibration analysis of non uniform microbeams based on modifed couple stress theroy. Effect of material and geometrical parameters on natural vibration of sandwich beams were predicted for various boundary conditions by Bensahal and Nadir Amrane [18] by analytical approach. Euler-Bernoulli beam theory is used by Rezaiee-Pajand et al. [19] to predict the modal behaviour of a generalied frame for its variation in the structural properties, such as joint angles, springs' stiffness and flexural rigidity of members.

Though, this article study structural behaviour of RFM module that used in a vehicle to cool the engine on one hand, on the other hand for a whole vehicle is concern the comfort depands on not only with less noise and vibrations also depends on conditioning of the cabin. Particularly thermal energy management in electric vehicle batteries is vital and appropriate HVAC system is essential. Verma et al. [20] studied combined single effect Vapour compression system and tc-CO_2 compression refregertion system based on thermodynamic analysis. Enhanced occupant comfort and safety, lightweight materials with enhanced thermal insulation properties, and optimised vehicle energy management are important and their achivements through HVAC modelling for an electric vehicle investicated by Kapeller et al. [21]. Peng et al. [22] have applied modulation principles on both rotor and stator of the cooling fan module to control the tonal noise.

In the current project work, spectral test has been carried out on the radiator fan module to determine frequency response functions and further used to evaluate the mode shapes and corresponding natural frequencies using LMS Test Lab software. From the litereature it is understood carrying out analytical appraoch on a complex contour RFM is not viable. Hence, it is proposed to provide a finite element model of RFM for its computational fluid dynamic study. In the current work a finite element model of RFM is develpoed and validated through its modal behavior.

Numerical modal analysis has also been carried out on the RFM using ANSYS. An important objective of this work is to investigate the mode shapes of the RFM by both experimental and numerical method. The signature test has also been done on the radiator fan to predict the actual noise radiated.

2. METHODOLOGY

This work has been divided into two parts such as spectral test and the signature test. The first part is the spectral test used for experimental modal analysis and is performed to obtain the modal parameters of a radiator fan module. The mode shapes of the radiator fan obtained from FE modal analysis is validated by the experimental modal analysis. The second part is the signature test used for predicting the noise radiated from the radiator fan in near field. The methedology is presented in Figure 1. On one hand spectral test carried out to determine the modal parameters experimentally and the same has been used to validate the FE (finite element) model. On the other hand singature test carried out to measure near field noise measurement using microphone indexing procedure with rectanguler array field points. Finally, a correlation study has been carried out from inference of results to conclude the influence of force transfer charcteristics over noise radiation of RFM.

The following Figure 2a shows the fan and shroud assembly of the RFM for experimental modal analysis and its CAD model shown in Figure 2b is used for Finite Element (FE) modal analysis.

3. EXPERIMENTAL MODAL ANALYSIS

A dynamic characteristic refers to modal parameters such as natural frequencies, damping factors, and mode shapes that are required to formulate mathematical model for NVH studies of any mechanical components or components forming the system. With the help of experimental modal analysis, it is possible to convert the vibration signals of excitation given and responses measured at discrete points on a complex structure, into a set of modal parameters. This section describes the experimental set up and the procedure followed to obtain the modal parameters such as resonance frequencies and mode shapes of RFM. Figure 2 presents the measurement system that includes LMS Scadas Mobile data acquisition system, instrumented hammer and LMS Test Lab software for spectral test. There were four channels used for this experiment. First channel is reference signal channel for instrumented hammer and the other three channels used for measuring acceleration responses from various points marked on the RFM. Channel set up, Acquisition set up and Test run set up all were carefully defined. The software and the hardware communication were done with the help of LAN cord.

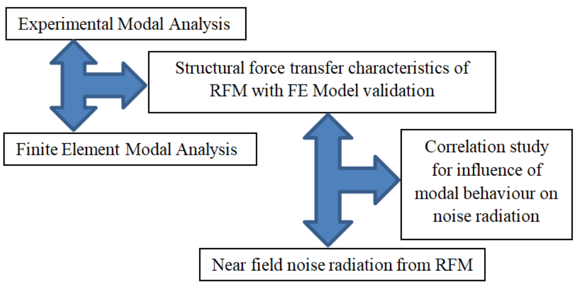
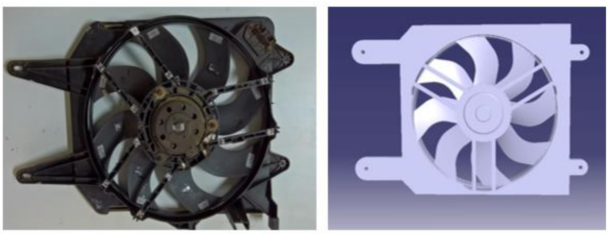


Figure 1. Methodology



(a) Actual RFM (b) RFM CAD model

Figure 2. RFM assembly and its CAD model

3. 1. Experimental Setup for Spectral Test

The primary objective of this spectral test is to do experimental modal analysis to determine the modal parameters of the radiator fan and shroud during free vibration and is tested in a free condition. This means that the RFM is not attached to the any structure or fixed support at any of its coordinates. In practice it is not possible to make a completely free support, but it is possible to provide a system which closely relates to the free condition. The Radiator fan and shroud assembly is hung with the help of light elastic bungee cords as shown in Figure 3. ISO coordinate system of the vehicle is followed. Hence x direction is considered to be coinciding with axis of the fan and other two directions are in transverse direction to the axis as per the right hand rule. There are 51 response points marked on the radiator fan and shroud which includes the 2 excitation points, one (point 51) is on the component fan and other is (point 1) on the shroud, this is because there is no rigid connection between the fan and the shroud and fan is freely rotating and the shroud which is stationary. For each and every response point of the fan and shroud the excitation is given with the help of instrumented hammer at one point (point 1) on the shroud and other point on the fan (point 51) in all three directions.

Figure 3. Experimental setup for spectral test

However, modal analysis is performed independently for fan and shroud considering respective FRFs of the components.

Figures 4 and 5 are the graphs that shows the variation of amplitude of frequency response function and the corresponding drive point coherence function as function of frequency of both excitation points. The peaks in the FRFs refer to the resonance frequencies of the component. It is observed the good coherence exists upto 800 Hz for axial direction drive points for both fan and shroud excitation compared to that of other two direction. Beyond 800 Hz the observation is that good coherence for the drive point exciation in transverse direction is observed for shroud point excitation compared to that of fan spindle exciation. Drive points of the shroud do have better coherence for the input compared to that of fan spindle drive points. However coherence corresponding to the peaks in frequency response function is good. This single point FRFs give the idea of resonance frequencies at just one excitation but are not sufficient to get mode shapes. All the response points FRFs are superposed to get overall FRF sum. The stabilization curve fit of the overall FRF sum is used to get modal frequencies and mode shapes using popular PolyMax algorithm. Figures 6 and 7 gives the corresponding stabilization curves.

The stabilisation diagram can be interpretated as that provides a set of poles and corresponding participation factors. Futher the mode shapes are determined from pole-redude model represented by Equation (1).

$$[H(\omega)] = \sum_{i=1}^{n}\left\{\frac{\{v_i\}\langle l_i^T\rangle}{j\omega-\lambda_i} + \frac{\{v_i^*\}\langle l_i^H\rangle}{j\omega-\lambda_i^*} - \frac{LR}{\omega^2} + UR\right\} \qquad \text{.......(1)}$$

where, n is the number of modes; v_i^* is the complex conjugate of a matrix; are the mode shapes; are the modal participation factors and are the poles, which are essentially a part of complex conjugated pairs and are relaated to the eigenfrequencies and damping ratios as given by Equation (2).

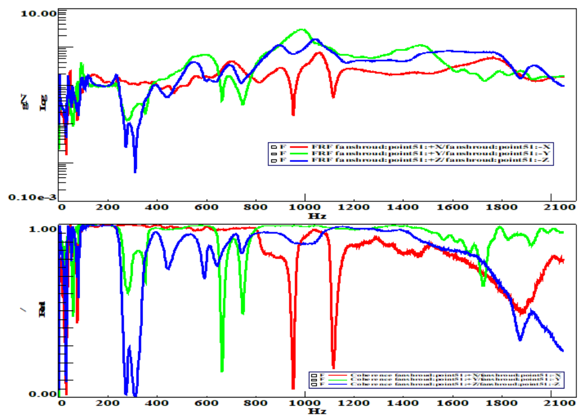


Figure 4. Variation of FRFs and Coherence Function for excitation at fan spindle

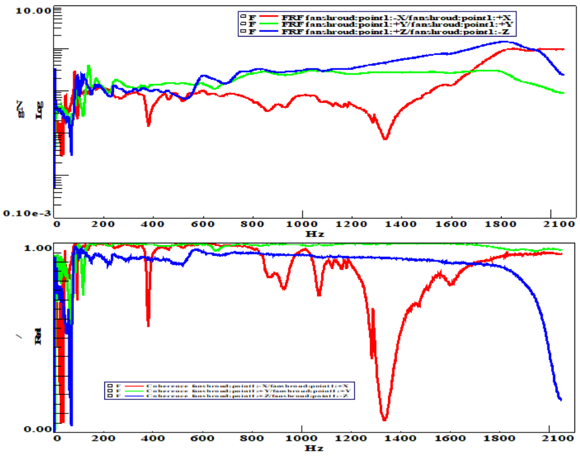


Figure 5. Variation of FRFs and Coherence Function for excitation at shroud

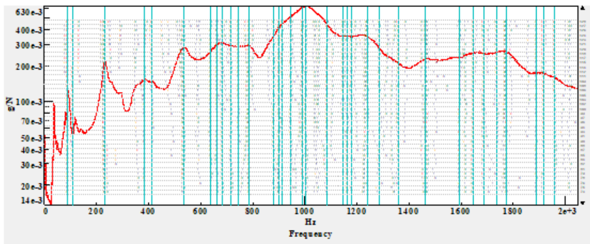


Figure 6. Stabilization curve for radiator fan for modal extraction

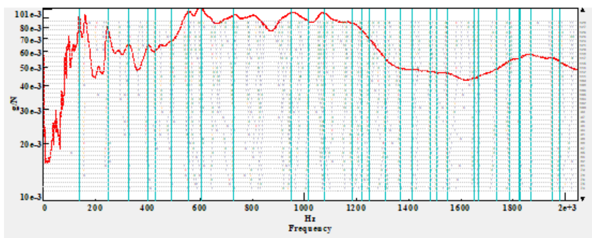


Figure 7. Stabilization curve for shroud for modal extraction

$$\lambda_i, \lambda_i^* = -\xi_i\omega_i \pm j\sqrt{1-\xi_i^2\omega_i} \qquad \text{........................(2)}$$

LR, UR are respectively the lower and upper residuals represents the out of band moeds in the considered frequecy band. A complete theoritical aspects of PolyMax is presented by Peeters et al. [23].

The stabilization curve is very important in modal analysis which brings in the complete characteristic nature of the component tested. The model size of the curve decides the number of poles. Due to the curve fit of all FRFs there are many mathematical poles along with actual physical poles. The PolyMax algorithm along with automatic modal parameter selection picks the physical poles alone which is further verified with modal assurance criteria and synthesized overall FRF sum. The vertical lines on the graphs are corresponding to the poles and each pole indicate the resonating frequencies of the fan and shroud respectively in the stabilization curve. Finally for all the stabilized poles the corresponding mode shapes are identified. The synthesized FRFs from the obtained modes are validating the complete experimental procedure followed. Figure 8 compares the estimated overall FRF sum obtained through impulse excitation test, with one which is synthesized from the modes obtained. This validate the experimental procedure.

3. 2. Experimental Setup for Signature Test

The signature test has been carried out in the night time when influence of environmental noise is almost negligible compared to the radiator fan noise. The fan along with the shroud is hung freely using bungee cords as shown in Figure 8. The minimum distance from the ground is maintained as 1m. A grid is prepared to position microphone to measure radiated noise in the near field. The microphone grid is placed 1.6 m from the ground and 1m in front of the fan. The motor is powered to rotate the fan and the measurement of noise is made for the rated rotational speed of the fan fixed at the speed of 3000 rpm. The noise measurements were made using single microphone method. LMS Test Lab software and the data acquition system is connected with the help of LAN cord and in turn the sensor indexed in the grid is connected to the data acquition system through BNC cables. Singature test module is used for measurement. Before the start of the measurement the ranging of sensor is carried out to decide the optimum power voltage to the active channel. The Free run option of measurement is chosen for 10 seconds and subsequently the sound pressure level time history has been recorded.

A rectangular array of grid size of 6 rows x 7 columns with 42 grid points is used to index the microphone using a tripod shown in Figure 9, which was positioned at 1 m distance from the radiator fan.

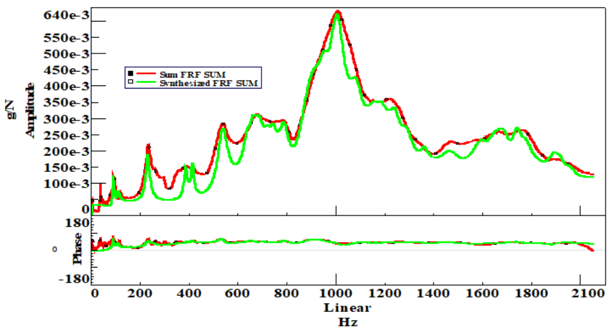


Figure 8. Comparison of synthesized overall FRF with estimated FRFs

Figure 9. Experimental setup for signature test for noise measurement

Sound field at the source plane was computed from the measured data. For 10 seconds for at all these grid point location the overall sound pressure level is obtained sequentially indexing the microphone and the Figure 10 presents the variation of the same for six diagonal points. The difference in sound pressure level between the extreme diagonal points is more compared to the of other points. The variation is observed to be between 60 to 95 dB sound pressure level. However, the overall sound pressure level obtained with A weighting is found out to be around 60 dB (A).

4. FINITE ELEMENT MODAL ANALYSIS

Numerical modal analysis is performed using ANSYS finite element software. The CAD model shown in Figure 2b is used for the same. The material definition and boundary condition were defined. Polymeric plastic material has been defined for shroud where as fan is made of aluminium. Table 1 below provide the material properties used to define the RFM. The shroud vertex are fixed, fan spindle is free to rotate in its axis. The FE model has been created with fine turning mesh size until the convergence criterion is met. Thus the prepared FE model is used further for modal analysis. The modal results were compared with experimental results.

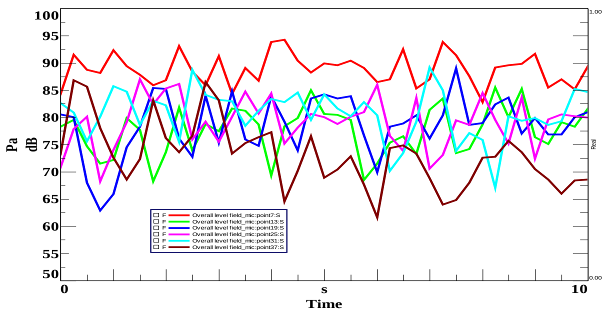


Figure 10. Variation of sound pressure level time histroy along the diagonal points

Figures 11 and 12 present and compare the modal frequencies and the corresponding mode shapes of the RFM obtained through numerically with experimental one for fan component.

Figures 13 and 14 present and compare the same for shroud. It is to be noted that though the mode shape compared in both the cases closer to the resonance

TABLE 1. Material properties

Component	Material	Density (kg/m^3)	Young's Modulus (N/m^2)	Poison ratio
Fan	Aluminium	2550	7×10^{10}	0.33
Shroud	Plastic	1200	2×10^{10}	0.15

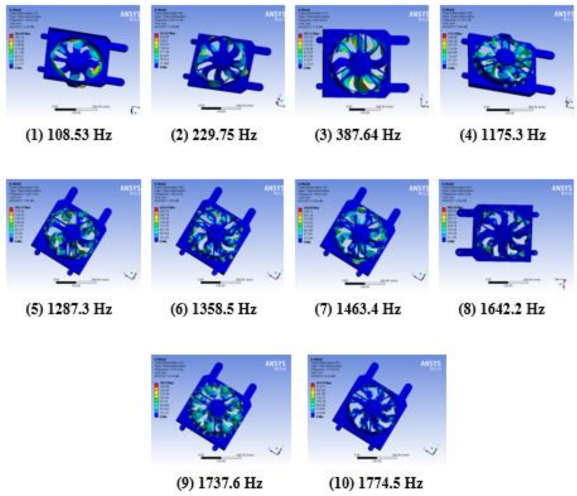


Figure 11. Modal parameters obtained from finite element modal analysis of radiator fan

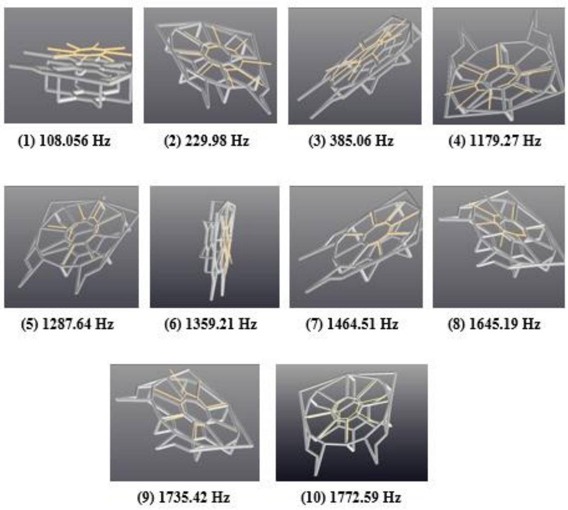


Figure 12. Modal parameters obtained from experimental modal analysis of radiator fan

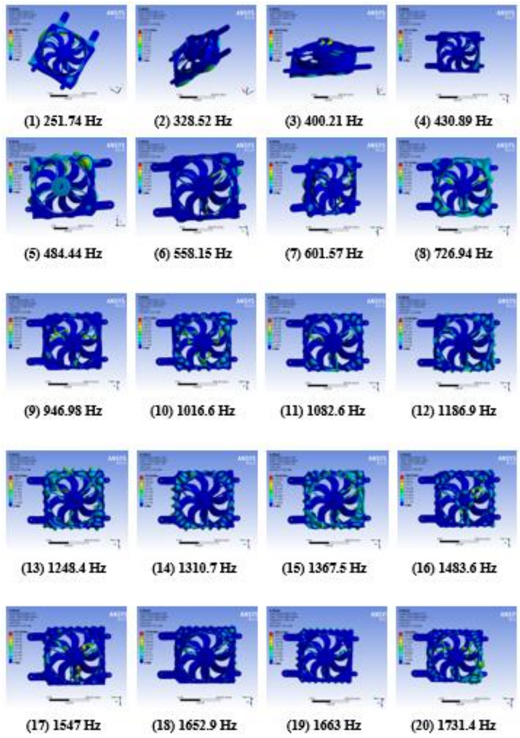


Figure 13. Modal parameters obtained from finite element analysis of shroud

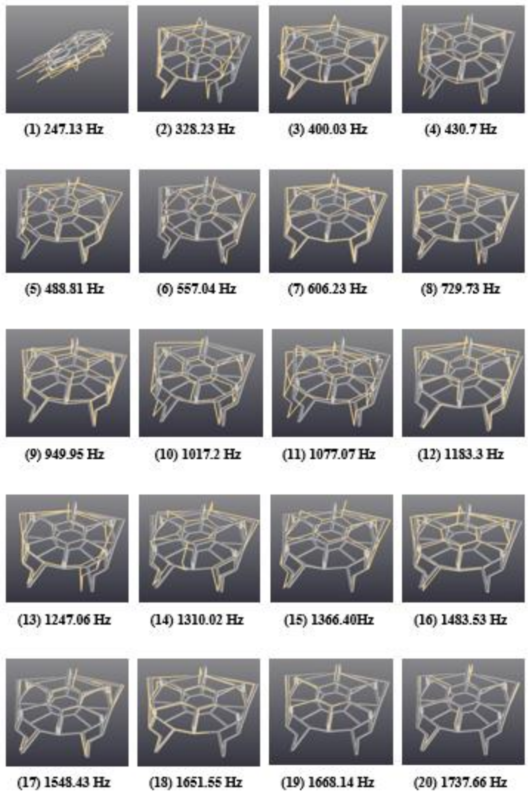


Figure 14. Modal parameters obtained from experimental modal analysis of shroud

frequencies the deflection shape obtained is not exact with numerical results. This is due to the reason that the number of measurement points in the experiment is limited. Finite number of discrete points are marked on the actual RFM as shown in Figure 2a for the response measurements during spectral test when impulse excitation is given with the help of instrumented hammer.

5. RESULTS AND DISCUSSION

Tables 2 and 3 summaries the natural frequencies of the fan and shroud components of RFM obtained from experimental and numerical modal analysis. The observed deviation is insignificant except at higher frequencies and the finite element model thus validated and proposed as an entension work to carry out vibro-acoustic and aero-acoustic simulation study. The resonance frequencies obtained and the speed of the fan need to confirm the design requirements. Figure 15 shows the variation of A weighted 1/3 rd octave band sound pressure level for the diagonal points (points 7, 13, 19, 25, 31 and 37) in the grid. Except grid point 7 rest of the points in the diagonal sound pressure level is below 50 dB(A) for the entire frequency range that include all the resonance frequencies. It is observed from the graph that the central frequencies between the bounds 561.2 to 2244.9 Hz for all the diagonal points negligible variation as compared to the frequency bound range up to 561.2 Hz.

Importantly,blade passing frequencies corresponding to the fan speeds between 500 to 3000 rpm are compared with resonance frequencies of listed in Table2 as well as in Table 3. They do not coincide with any of the modes. That ensures the optimum design consideration of structural noise contribution of the chosen automotive RFM.

TABLE 2. Resonance frequencies of radiator fan in Hz

Modes	Experiemntal Modal Analysis	Finite Element Modal Analysis
Mode 1	108.06	108.53
Mode 2	229.98	229.75
Mode 3	385.06	387.64
Mode 4	1179.27	1175.3
Mode 5	1287.64	1287.3
Mode 6	1359.21	1358.5
Mode 7	1464.51	1463.4
Mode 8	1645.19	1642.2
Mode 9	1735.42	1737.6
Mode 10	1772.59	1774.5

TABLE 3. Resonance frequencies of shroud in Hz

Modes	Experiemntal Modal Analysis	Finite Element Modal Analysis
Mode 1	247.13	251.74
Mode 2	328.23	328.52
Mode 3	400.03	400.21
Mode 4	430.70	430.89
Mode 5	488.81	484.44
Mode 6	557.04	558.15
Mode 7	606.23	601.57
Mode 8	729.73	726.94
Mode 9	949.95	946.98
Mode 10	1017.20	1016.6
Mode 11	1077.07	1082.60
Mode 12	1183.30	1186.90
Mode 13	1247.06	1248.40
Mode 14	1310.02	1310.70
Mode 15	1366.40	1367.50
Mode 16	1483.53	1483.60
Mode 17	1548.43	1547.00
Mode 18	1651.55	1652.90
Mode 19	1668.14	1663.00
Mode 20	1737.66	1731.40

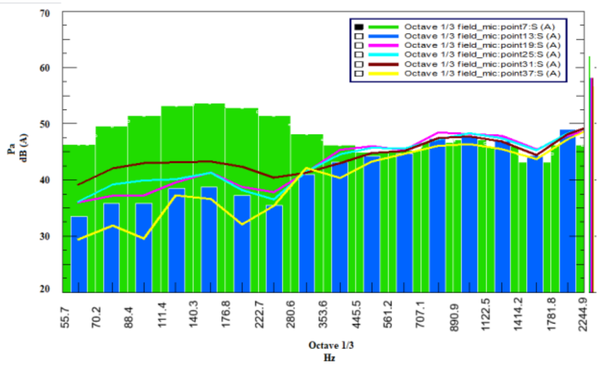


Figure 15. Variation of sound pressure level along the diagonal points

The difference in sound pressure level is significant upto 500 Hz bound frequency and the variation between the extreme diagonal points is arround 20 dB(A). This is becouse of structural resonance modes amplitudes are much pronounced till 500 Hz for both fan as well as shroud. The frequency corresponding to the speed of the fan is 314 Hz is the exciation and only the top right diagonal point captured the higher sound pressure level than the other points. At this frequency arround 8 dB(A) difference is observed between microphone location at point 7 and the rest of the diagonal points. Also it is evedent from the modal analysis that structural mode is not at 314 Hz hence it can be concluded it is due to speed of the fan the noise is contributed.

6. CONCLUSION

Modal parameters of radiator fan and shroud assembly was determined by carrying out spectral test. Sufficient number of FRFs is calculated to get appropriate overall curve fit of FRF sum. The drive point coherence and synthesized overall FRFs were obtained to ensure the quality and validity of the experimental modal analysis. Further the experimental results were used to compare the numerical modal results to validate the FE modal model and hence the FE model can be proposed for parametric study for product development which is not a scope of current work. The validated FE model is also proposed for extension work of vibro-acoustic and aero-acoustic simulation. There are only three critical modal frequencies for fan and five critical modal frequencies for shroud below 500 Hz are obtained. Whereas above 500 Hz to 2000Hz frequency range there were seven for fan and fifteen for shroud are obtained. And interestingly no coincident resonance frequencies of both fan and shroud as well as no conincidence of blade passing frequencies from 500 rpm to 3000 rpm with the incremental speed of 500 rpm are observed which an important design requirement. The signature test was also performed and the overall sound pressure level of 60 dB (A) is observed in the near field at a distance of 1 m from RFM which would be a permissible value. Important conclusion to correlate the modal behaviour with noise radiation can be observed in the two ranges of frequencies. Up to 500 Hz the sound pressure level has shown variation on the diagonal grid points whereas above 500 Hz the variation is negligible.

7. ACKNOWLEDGMENT

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8. REFERENCES

1. Zhu, M., Wang, K., Zhang, P. and Li, Z., "Numerical and experimental investigation of aerodynamic noise from automotive cooling fan module", ***Journal of Vibroengineering***, Vol. 17, No. 2, (2015), 967-977.
2. Liu, X.-A., Zhu, X., Shangguan, W., Wang, T., Wang, X.-L. and Chen, Y., "A study on design methods for condenser radiator fan module mounting system", ***International Journal of Heavy Vehicle Systems***, Vol. 26, No. 2, (2019), 119-135.
3. Baniasadi, E., Aydin, M., Dincer, I. and Naterer, G., "Computational aerodynamic study of automotive cooling fan in blocked conditions", ***Engineering Applications of Computational Fluid Mechanics***, Vol. 7, No. 1, (2013), 66-73, doi: org.10.1080/19942060.2013.11015454
4. Lim, J., Sim, W., Yun, S., Lee, D. and Chung, J., "Reduction of vibration forces transmitted from a radiator cooling fan to a vehicle body", ***Journal of Sound and Vibration***, Vol. 419, (2018), 183-199, doi: org/10.1016/j.jsv.2018.01.016
5. Kumar, S. and Talwar, H., *Optimization of radiator fan for nvh improvement*. 2017, SAE Technical Paper.
6. Wang, J., Huang, L. and Cheng, L., "A study of active tonal noise control for a small axial flow fan", ***The Journal of the Acoustical Society of America***, Vol. 117, No. 2, (2005), 734-743, https://asa.scitation.org/doi/10.1121/1.1848072
7. Arefin, A. and Islam, R., "Investigation of different validation parameters of micro gas turbine for range extender electric truck", ***International Journal of Engineering***, ***Transactions A: Basics,*** Vol. 31, No. 10, (2018), 1782-1788, doi: 10.5829/ije.2018.31.10a.22.
8. Rajabi, N., Rafee, R. and Farzam-Alipour, S., "Effect of blade design parameters on air flow through an axial fan", ***International Journal of Engineering***, ***Transactions A: Basics,*** Vol. 30, No. 10, (2017), 1583-1591, doi: 10.5829/ije.2017.30.10a.20.
9. Suzuki, A., Tominaga, T., Eguchi, T., Kudo, T. and Takada, T., "Study on the fun noise reduction for automotive radiator cooling fans", ***Mitsubishi Heavy Industries Technical Review***, Vol. 43, No. 3, (2006), 23-30.
10. Jayachandran, R., Alavandi, B., Niesluchowski, M., Low, E., Miao, Y. and Zhang, Y., *Fan shroud design for low speed damageability*. 2017.
11. Ewins, D.J., "Modal testing: Theory, practice and application, John Wiley & Sons, (2009).
12. Pieters, R., "Experimental modal analysis of an automobile tire under static load", ***DCT Rapporten***, Vol. 112, (2007), 1-24.
13. Patil, K., Baqersad, J. and Behroozi, M., "Experimental modal analysis on a tyre–lessons learned", ***International Journal of Vehicle Noise and Vibration***, Vol. 13, No. 3-4, (2017), 200-215, doi: 10.1504/IJVNV.2017.10010555.
14. Vervoort, J., "Experimental modal analysis of an automobile tire", Bachelor final project report No, DCT, (2007).
15. Tare, K., Mukherjee, U. and Vaidya, R.J., *Design optimization of automotive radiator cooling module fan of passenger vehicle for effective noise management using cfd technique*. 2017, SAE Technical Paper.
16. Lu, Y.-Y., *Experimental study of installation effects on cooling fan noise*. 2021.
17. Hosseini Hashemi, S. and Bakhshi Khaniki, H., "Free vibration analysis of nonuniform microbeams based on modified couple stress theory: An analytical solution", ***International Journal of Engineering***, ***Transactions B: Applications,*** Vol. 30, No. 2, (2017), 311-320, doi: 10.5829/idosi.ije.2017.30.02b.19.
18. Amrane, M.N. and Bensahal, D., "Effects of material and geometrical parameters on the free vibration of sandwich beams", ***International Journal of Engineering***, ***Transactions B: Applications,*** Vol. 29, No. 2, (2016), 222-228, doi: 10.5829/idosi.ije.2016.29.02b.11.
19. Rezaiee-Pajand, M., Aftabi Sani, A. and Hozhabrossadati, S.M., "Free vibration of a generalized plane frame", ***International Journal of Engineering***, ***Transactions A: Basics,*** Vol. 31, No. 4, (2018), 538-547, doi: 10.5829/ije.2018.31.04a.04.

20. Verma, A., Kaushik, S. and Tyagi, S., "Thermodynamic analysis of a combined single effect vapour absorption system and tc-CO_2 compression refrigeration system", ***HighTech and Innovation Journal***, Vol. 2, No. 2, (2021), 87-98.

21. Kapeller, H., Dvorak, D. and Šimić, D., "Improvement and investigation of the requirements for electric vehicles by the use of hvac modeling", ***HighTech and Innovation Journal***, Vol. 2, No. 1, (2021), 67-76.

22. Peng, Z., Ouyang, H., Wu, Y. and Tian, J., "Tonal noise control of cooling fan module by using modulation principles on both rotor and stator", in Turbo Expo: Power for Land, Sea, and Air, American Society of Mechanical Engineers. Vol. 50985, (2018), V001T009A007.

23. Peeters, B., Van der Auweraer, H., Guillaume, P. and Leuridan, J., "The polymax frequency-domain method: A new standard for modal parameter estimation?", ***Shock and Vibration***, Vol. 11, No. 3-4, (2004), 395-409.

Persian Abstract

چکیده

در میان بسیاری از لوازم جانبی خودروهای سواری، ماژول فن رادیاتور خنک کننده موتور (RFM)منبع مهم صدا و لرزش است. پارامترهای مودال اساس مکانیسم تولید نویز و پدیده انتشار نویز را فراهم می کنند. هدف کار تحقیقاتی فعلی، توصیف RFM از طریق رفتار مودال آن برای بررسی سهم ارتعاش ساختاری در تشعشعات نویز در طول حالت عملکرد آن است. از این رو، تجزیه و تحلیل مودال تجربی و اندازه‌گیری نویز RFM انجام شد. علاوه بر این، یک تحلیل مودال المان محدود از RFM برای اعتبارسنجی مدل المان محدود که می‌تواند برای مطالعات دینامیک سیالات ارتعاشی-آکوستیک و محاسباتی استفاده شود، انجام شد. زیر ۵۰۰ هرتز در مجموع هشت فرکانس مودال بحرانی، سه فرکانس برای فن و پنج فرکانس برای shroud تعیین شده است و بین ۵۰۰ هرتز تا ۲۰۰۰ هرتز هفت حالت برای فن و پانزده حالت برای shroud حذف شده است. از مطالعه همبستگی، مشاهده می‌شود که سهم رفتار مودال برای تابش نویز با تغییرات ۲۰ دسی‌بل (A)زیر ۵۰۰ هرتز، در امتداد نقاط شبکه مورب مشاهده می‌شود، در حالی که این تغییرات فراتر از ۵۰۰ هرتز ناچیز بود. نکته مهم این است که فرکانس‌های عبور تیغه مربوط به سرعت فن بین ۵۰۰ تا ۳۰۰۰ دور در دقیقه با فرکانس‌های تشدید به‌دست‌آمده مقایسه شد و با فرکانس‌های مدال همخوانی ندارند.