Application of the Structural Dynamic Modification Method to Reduce the Vibration of the Vehicle HVAC System

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Abstract. Heating, ventilation and air conditioning (HVAC) system is one of the major sources for the vehicle noise and vibration, which subsequently contribute to the bad acoustical environment. The components of the HVAC system can produce a significant level of vibration during the operation and contributed to an unwanted noise. As the vibration of HVAC components get worse, it will be transferred to other components and excites the natural frequencies of the HVAC system. Natural frequencies of the components are depending on the mass and stiffness of the HVAC components and the value can been modify using these two parameters. This study is focusing on the specific type of HVAC noise and vibration problem and the counter measure has been perform by implementing the structural dynamic modification (SDM) method to the air conditioning (AC) pipe. The lab-scale of the vehicle HVAC system is set up to represent the actual HVAC system with the real vehicle operation. Noise and vibration of the HVAC system in the system level are measured and compared. From this data, the Dynamic Vibration Absorber (DVA) is designed and applied at the AC pipe of the HVAC system. The result shows that, the natural frequency of the AC pipe can be shifted which resulting a wide effective frequency range of 100-500 Hz. The effect of DVA on the HVAC system is observed during operation whereby a significant vibration attenuation has been achieved for operating frequency range of 100-300 Hz.

1. Introduction

Heating, ventilation and air conditioning (HVAC) system is a crucial section in the vehicle system which can affected the passenger comfort. Interestingly, it has influences the psychological and physical perception of the passenger and vehicle quality [1]. However, it has drawback whereby it produced significant contribution to the interior noise level [2], [3]. The vibration is the source of an uncomfortable issue and might come from the components of HVAC system [4]. Therefore, many studies aim to reduce the vibration of the HVAC system to the lowest possible in ensuring the comfortable of the passenger.

Structural dynamic modification (SDM) is one of the techniques that can be used to reduce the vibration of the structure or system. Generally, the SDM is used to change the dynamic behaviour (i.e. natural frequencies, mode shapes and frequency responses) of the structure [5]. This technique can be done by predicting the modified behaviour of the structure such as lumped masses, rigid links, and dampers or by adjusting the configuration parameters [6]. In term of application, this technique has been applied to an active suspended handle to overcome the displacement limitation of a piezoelectric actuator by increase the stiffness of the lower beam of the handle. The finding from this study shows that the SDM able to alter the stiffness of the handle and shifted the modes of the structure beyond operating frequency range [7].
Dynamic vibration absorber (DVA) is one of the scheme under SDM that consist of a mass attached to the structure to eliminate the vibration of a harmonically excited system [8]. This scheme produced a significant reduction in force transmissibility on flexible plate structure with vibration reduction up to 92 % [9]. In an automotive application, the DVA has been applied to the suspension structure to reduce the vibration from the road disturbances. The finding shows the RMS of the body acceleration and suspension can be decrease by 4 % and 16 %, respectively [10]. This study investigates the application of DVA to the HVAC system to reduce the vibration and noise produced as it will increases the comfortability of the passenger in vehicle.

2. Methodology

2.1. Design and Implementation of the HVAC system

The HVAC components are assembled and mounted in position (similarly as a real vehicle assembly) as shown in Figure 1 and the real operational condition of the HVAC is implemented to provide the similar system condition for the reliability of the test results. Generally, this system is operated with two main power supplies, which is direct power supply to the motor and battery for the HVAC system. The system has an activation button to control the engagement of compressor, which also connected with the battery as power supply. The control panel has a blower speed controller to control the blower speed that flow the air throughout the air ducting.

![Figure 1. Overall view of the HVAC system](image-url)

2.2. Noise and vibration measurement

Figure 2 shows the flowchart of the noise and vibration measurement conducted on the rig of the HVAC system. Two types of sensors are used which is accelerometer for vibration measurement and tachometer for rotational acceleration measurement. The accelerometer is attached on the six components as highlighted in Figure 1 to record the vibration responses. The tachometer is mounted on the rotating motor belt to measure the motor (engine) speed.

All the accelerometers are connected through LMS SCADAS Mobile and the results were analysed with LMS Test.Lab. The software used will record the data in time and frequency domains. In this study, there are two condition of measurements, which is idle (850 rpm) and tracking (850-1800 rpm) conditions. The idle condition is used as a reference condition whereas the tracking condition to replicate the actual running vehicle.
Next, an Experimental Modal Analysis (EMA) is carried out to obtain the natural frequencies of the HVAC components. There are two patterns of test which is free-free boundary condition and incomplete system structures. For the free-free boundary condition, the all the HVAC components are hanged individually using the rubber band. An accelerometer is attached on each component and impact hammer is used to apply external force on the component. Natural frequencies and mode shapes for each component are determined using the LMS software. EMA is then performed with same equipment incomplete system conditions. The HVAC components for EMA is summarized in Table 1 including the detail component arrangement and geometry modelling.

Table 1. Test set up and geometry for the assembly and individual components of EMA.

| No. | Component                  | Test setup       | Geometry |
|-----|----------------------------|------------------|----------|
| 1   | Overall rig structure      | Sensor → Data Logger → Data Output & Analysis | ![Geometry Image] |

Figure 2. Experimental flow on vibration measurement
2.3. Modelling and Implementation of DVA.

The air conditioning (AC) pipe is selected among the four components due to the highest contribution of humming type noise that obtained in Table 2. Again, the EMA is conducted to obtain the natural frequency before and after implementation of SDM. The DVA is modelled as a second spring-mass system which added to the primary mass as illustrated in Figure 3. In this testing, the SDM is applied at point 2, which is selected as a point for SDM attachment as shown in Figure 4. The total mass prediction of the absorber with 0.132 kg is included in the modification.
Next step is the tuning of the DVA. The tuning is based on the theoretical calculation to determine the accurate distance between the two masses from the centre rod as shown in Figure 3(b). This distance becomes the first reference before the DVA is tuned into the targeted frequency. The theoretical calculation is shown as follows:

\[ F = k\ddot{\vartheta} \]

(1)

The deflection of cantilever beam,

\[ \vartheta = \frac{FL^3}{3EI} \]

(2)
\[ L^2 = \frac{3EI}{k} \]  \hspace{1cm} (3)

The natural frequency,
\[ \omega^2 = \frac{k}{m_a} \]  \hspace{1cm} (4)

\[ k = \frac{(2\pi f)^2}{m_a} \]  \hspace{1cm} (5)

Substitute Eq. (4) into Eq. (3), the distance required is obtained as follows:
\[ L^3 = \frac{3EI}{(2\pi f)^2 \times m_a} \]  \hspace{1cm} (5)

Second moment of inertia of rod is given by:
\[ I = \frac{1}{4\pi r^4} \]  \hspace{1cm} (6)

Where, \( m_a \) is the mass of each secondary mass (kg), \( r \) is the radius of the cantilever beam (m), \( L \) is the length of the cantilever beam (m), \( f \) is the natural frequency (Hz) and \( E \) is the Young’s modulus of elasticity of the cantilever beam (N/m²). By considering Eq. (1) to (5), the estimated length, \( L \) for the given secondary mass values can be calculated.

3. Results and Discussion

3.1. Noise and vibration measurement

Table 2 summarizes the noise and vibration measurement results in the system level. Both test patterns for idle and tracking with four type of noises are obtained. Hissing noise is an audible noise that involves a high frequency range of 4000 to 6000 Hz at idle and tracking conditions. From the measurement, the highest contribution is from the evaporator pipe inlet. The noise may induce by the action of refrigerant expansion at orifice tube, as the R134a gas is forces to flow through a fine restriction orifice.

Humming noise is produced at three different operating frequencies, which is at 300 to 400 Hz for idle condition, 400 to 500 Hz for tracking condition (AC off) and 100 to 300 Hz (AC on). Differently for the humming noise, two components show the highest contribution, which is compressor and AC pipe. This result gives a clear insight that the operating vibration of the system is transferred through the compressor to the AC pipe in sequential transmission. However, when AC is turned off, the vibration amplitude is detected within frequency range of 400-500 Hz, which transferred from the compressor up to the evaporator location.

The clicking noise is the effect of the compressor clutch impact. When the clutch is engaged, the compressor shaft will rotate, and refrigerant circulate the internal components which containing the field coil. Field coil is energized with the pressure plate towards it and causing the compressor internals to turn. Thus, the compressor is the highest contribution of this noise at frequency range of 200 to 300 Hz. The fourth noise found is air rush-type. The noise between both locations of outlets is induced due to the turbulence effect of airflow at the HVAC blower. As the velocity of fluid flow increases, the chance of turbulences is high. The noise is detected at frequency range of 1400 to 1700 Hz for both locations.
Table 2. HVAC noise indentation results in system level.

| No | Noise                  | Test method | Engine | Air condition (AC) | Blower speed | Accelerometer setup location | Noise Frequency (Hz) | Component with highest contribution |
|----|------------------------|-------------|--------|---------------------|--------------|------------------------------|----------------------|-------------------------------------|
| 1  | Hissing                | Idle        | On     | On                  | On           | 1                            |                      | 4000-6000 Evaporator pipe inlet    |
|    |                        | Rpm tracking| On     | On                  | On           | 1                            |                      |                                     |
| 2  | Humming                | Idle        | On     | On                  | On           | 1                            | 300-400              | Compressor and AC pipe            |
|    |                        | Rpm tracking| On     | Off                 | -            | -                            | 400-500              |                                     |
|    |                        |             | On     | On                  | On           | 1                            | 100-300              |                                     |
| 3  | Clicking (Compressor Clutch Engagement) | Idle | On | On | 1 | 1. Motor  | 200-300 | Compressor |
|    |                        |             | On | On | 1 | 2. Compressor | | |
|    |                        |             | On | On | 1 | 3. AC pipe | | |
|    |                        |             | On | On | 1 | 4. Evaporator pipe inlet | | |
|    |                        |             | On | On | 1 | 5. TXV | | |
| 4  | Air rush               | With Vents  | Off | Off | 1 | 1400-1700 | 1400-1700 HVAC Blower |
|    |                        | Without Vents | Off | Off | 1 | | |
3.2. Experimental Modal Analysis

Table 3 shows four components of HVAC system with different natural frequencies. The natural frequency of AC pipe is 148.9 Hz which is the highest natural frequency compared to other components. This is due to the structure is not mounted rigidly and can be easily deflected when the external force applied. Second highest is TXV with 128 Hz. The structure is a hollow aluminium pipe and linked with other components which significantly differ from AC pipe. Compressor and overall rig structure natural frequencies are at 99.78 Hz and 54.64 Hz, respectively. The compressor and rig are lower due to the structures have high mass.

| No | Component       | Geometry point | Frequency range, Hz | Mode number | First natural frequency, Hz |
|----|----------------|----------------|---------------------|-------------|-----------------------------|
| 1  | Overall rig structure | 19             | 0 - 1000            | 11          | 54.64                       |
| 2  | Compressor      | 4              |                     | 5           | 99.78                       |
| 3  | AC pipe         | 3              |                     | 5           | 148.9                       |
| 4  | TXV             | 2              |                     | 7           | 128                         |

3.3. Vibration measurement with application of DVA

Again, the EMA and vibration measurement is conducted for the second time after the implementation of DVA. Figure 5(a) and (b) shows the vibration spectrum and frequency response function (FRF) of the AC pipe before and after implementation of the DVA. Before implementation of DVA, the natural frequency of the AC pipe is obtained at 148.9 Hz while after the DVA implementation, the first natural frequency is shifted to frequency ranges of below 100 Hz and above 500 Hz as shown in Figure 5(b). In term of vibration spectrum in Figure 5(a), the vibration acceleration is changed and greatly reduced after implementation DVA. The figure clearly shows the vibration reduction and the effective frequency range for DVA is found from 100 to 500 Hz.
4. Conclusions
From this paper, the following conclusions can be made:

1. There are four type of vibration and noise found on the HVAC system which is hissing, humming, clicking and air-rush vibration and noises. Each noise has found at different operating frequency range and different highest component contribution.

2. The main component that has significant contribution for humming type noise and vibration in HVAC system is the compressor and AC pipe between which is between 100 to 300 Hz of operating frequency. The first natural frequency of the AC pipe is 148.9 Hz, which is within this noise range.

3. The DVA scheme of SDM method has been successfully applied to the AC pipe. With the implementation of DVA scheme, the resonance phenomenon can be avoided when the HVAC is running and subsequently reduced the vibration of the overall system. The effective frequency range of the AC pipe is found at 100 to 500 Hz after implementation of DVA.

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References
[1] H. B. Huang, X. R. Huang, M. L. Yang, T. C. Lim, and W. P. Ding, “Identification of vehicle interior noise sources based on wavelet transform and partial coherence analysis,” Mech. Syst. Signal Process., vol. 109, pp. 247–267, 2018.
[2] S. St. Hill, S. Watkins, S. P. Mavuri, X. Wang, and D. Weymouth, “An investigation of vehicle HVAC cabin noise,” in Proceedings of SAE World Congress, 2008, pp. 1–12.
[3] E. Canepa, A. Cattanei, and F. M. Zecchin, “Aerodynamic noise from cooling and HVAC systems,” in International Conference on Connected Vehicles and Expo, 2013, pp. 164–170.
[4] J. H. Yoon, I. H. Yang, J. E. Jeong, S. G. Park, and J. E. Oh, “Reliability improvement of a sound quality index for a vehicle HVAC system using a regression and neural network model,” Appl. Acoust., vol. 73, no. 11, pp. 1099–1103, 2012.
[5] A. Sestieri, “Structural dynamic modification,” Sadhana - Acad. Proc. Eng. Sci., vol. 25, no. 3, pp. 247–259, 2000.

[6] T. K. Kundra, “Structural dynamic modifications via models,” Sadhana - Acad. Proc. Eng. Sci., vol. 25, no. 3, pp. 261–276, 2000.

[7] A. Z. A. Ahmad Mazlan and Z. M. Ripin, “Structural dynamic modification of an active suspended handle with a parallel coupled piezo stack actuator,” Proc. Inst. Mech. Eng. Part I J. Syst. Control Eng., vol. 230, no. 2, pp. 130–144, 2016.

[8] H. L. Sun, H. B. Chen, P. Q. Zhang, X. L. Gong, and K. Zhang, “Application of dynamic vibration absorbers in structural vibration control under multi-frequency harmonic excitations,” Appl. Acoust., vol. 69, no. 12, pp. 1361–1367, 2007.

[9] Y. Du, R. A. Burdisso, and E. Nikolaidis, “Control of internal resonances in vibration isolators using passive and hybrid dynamic vibration absorbers,” J. Sound Vib., vol. 286, no. 4–5, pp. 697–727, 2005.

[10] Y. Shen, L. Chen, X. Yang, D. Shi, and J. Yang, “Improved design of dynamic vibration absorber by using the inerter and its application in vehicle suspension,” J. Sound Vib., vol. 361, pp. 148–158, 2016.