07NVC-112

Noise Refinement Solutions for Vehicle HVAC Systems

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ABSTRACT

Noise and vibration have important influence on customer's perception of vehicle quality. Research and development have been conducted to investigate the vehicle Heating, Ventilation and Air Conditioning (HVAC) noise generation and transmission mechanism. Noise and vibration comparison tests have been completed for the proposed refinement solutions. Testing results are discussed paying special attentions to the air borne noise reduction.

One of interior noise major contributors is the HVAC system, as air handling unit of the HVAC system is located behind the vehicle instrument panel within the cabin. Modifications made to internal structural geometry of the system have been conducted to provide insight into the effect of each structural feature on the overall Sound Pressure Level (SPL) and frequency spectrum components. Acoustic felts have been applied onto some selected locations of the system, test data shows that both the overall noise level and the noise spectrum peaks at specific frequencies have been reduced.

INTRODUCTION

Current design of the HVAC systems relies on the circulation of liquid refrigerant and water in order to perform their respective functions of cooling, and heating. The design then exploits forced convection and conduction in the heating or cooling of ambient temperature air. This air is then expelled via ducting into the cabin of the vehicle to achieve the required temperature. The system is split into two sections, one dealing with liquids as the working fluid and the other with air. Both the liquid and air sides of the system produce noise from various components.

The majority of the liquid side of the system is located within the engine bay of the vehicle, with only two liquid-to-air heat exchangers positioned in the air handling unit. On the contrary, the air handling unit exists within the Instrument Panel (IP) of the vehicle that is located inside the cabin.

Although the air handling hardware is not responsible for the total noise level inside the cabin, its individual frequency contribution may be dominant within the confined cabin environment [1]. This is because the driver and occupant's sensitivity towards the noise is high due to their proximity to the HVAC unit. Suspected major contributors of the noise within the system are the centrifugal fan (both blower and motor) and aerodynamic flow through the ducting and vent geometry.

Objectives of this investigation are to reduce the noise output associated with the circulation of air in the current production HVAC units.

The previous investigations showed that the largest noise magnitude was generated with the unit set to the full face and full cold settings [2]. Structural vibration of the system did not contribute to the major frequency spectrum components of the cabin noise, the HVAC noise was air borne predominant [3].

For this investigation the liquid side of the system will be neglected. Testing will only be conducted on the air handling unit in a laboratory test rig. It is therefore a major objective of this investigation to determine through testing, the mechanism that is responsible for generating the predominant noise within the system and then to develop strategies for its reduction.

INSTRUMENTATION SET UP AND TESTING PROCEDURES

Figures 1(a) and 1(b) show the HVAC system layout where the number boxes represent the followings:

1 - Air Inlet – recirculate setting;
2 - De-mist and Front Vent – ducting connection port;
3 - Centrifugal Fan Housing;
4 - HVAC Integration Module (HIM) – dynamic vent controller;
5 - Rear Vent – ducting connection port;
6 - Main Heat Exchanger Housing;
7 - Air Inlet – fresh air / re-circulate;
8 - Refrigerant Lines – cooling;
9 - Water Lines – heating;
10 - Main Ducting – from fan to heat exchangers;

11 – Bungee cords;

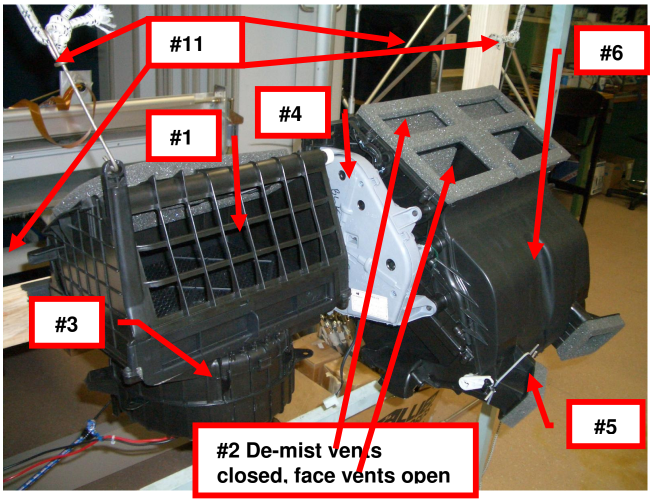

Figure 1(a) HVAC System Layout.

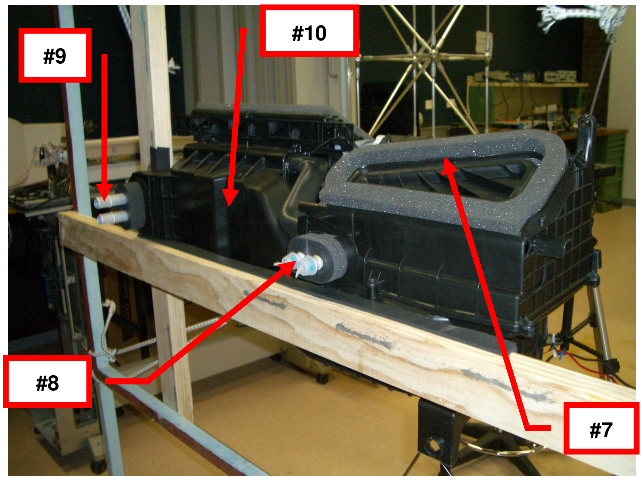

Figure 1(b) HVAC System Layout.

A microphone and its pre-amplifier were used to measure sound pressure data in this investigation. Variable voltage DC power supplier was used to power the HVAC system. Two digital multimeters were used to measure the voltage and current supplied to the HVAC system. A stroboscope was utilized to measure the fan speed.

A microphone was supported 1.35 meter above the floor and 1 meter away from the surface of the HVAC system by a tripod as shown in Figure 2(a). The microphone measurement points were selected at a 1 meter radius away from the HVAC system (free-field) to eliminate errors associated with near-field or reverberant-field measurements. Eight positional points were selected at 45 degree increments from a 0 degree datum directly in front of the system as shown in Figure 2(b).

B & K Pulse intelligent data acquisition and analysis system was used to record sound pressure spectra, sound pressure levels.

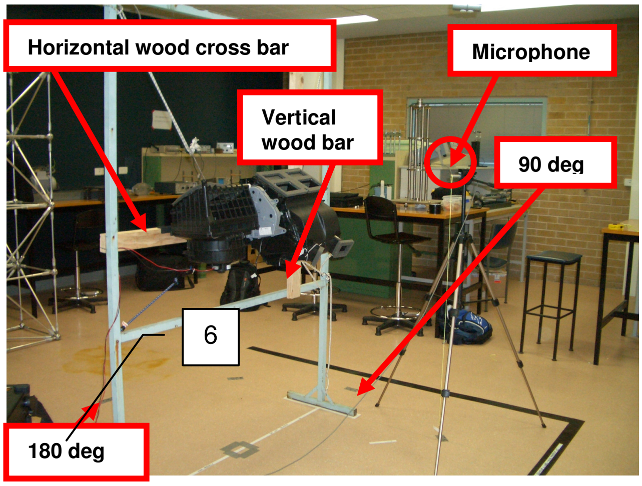

Figure 2(a) The HVAC system laboratory test rig and setup.

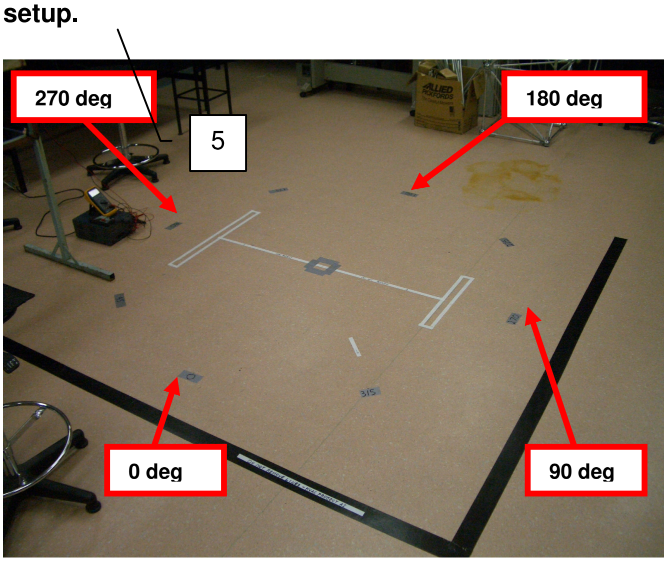

Figure 2(b) Microphone measurement points marked on the floor sequentially from 0 degrees to 315 degrees.

Speed of the centrifugal fan was measured with aid of a stroboscope. The fan wheel was marked with a black arrow for reference. The strobe flash frequency was read until the marking on the wheel looked stationary.

The settings of the HVAC system dictate the path of the air flow through the unit and hence have an impact on direction of propagating soundwaves. The unit was tested for acoustic emission under the worst case scenario which had the settings of "*Full Face – Full Cold – Re-circulate*" as shown in Figure 1(a). All of the valves and vents on the system were controlled by the HVAC Integration Module (HIM). To achieve the required settings the connecting rods from the HIM that physically operated the vents were removed. The vents were then fixed in place to simulate the correct settings.

The power consumption and performance of the system (air flow rate) were also controlled; the HVAC system specifications for in-vehicle voltage, current and volume flow rate of air are respectively, 12.5 volts, 22 amps and 140 Litres/second. The test unit was free of additional ducting that distributes the air throughout the vehicle cabin. The ducting would provide friction loss to the air flow which means that the fan motor must provide more power to achieve the required performance level. The absence of the ducting would result in a lower current flow to the motor than that of specified in-vehicle. Under the lab test conditions with normal operating voltage of 12.5 volts, the current reached at the specified voltage was approximately 14.9 amps. In order to control the system performance, air flow rate was monitored by measuring the pressure drop between the inlet and the air registers.

Four bungie cords were used to suspend the HVAC unit within test rig frame at each corner of the test unit as shown in Figure 1(a). As shown in Figure 2(a), within the test rig frame, a horizontal wood cross bar was used to provide the vertical positioning of the HVAC unit. A vertical wood support bar was installed to provide the horizontal positioning of the HVAC unit.

RESULTS AND DISCUSSIONS

There were a number of studies carried out:

- Baseline study for comparing the noise difference between the baseline and the development variations.
- Fan speed variation study.
- Modification of the existing geometry including the covered motor vent passage hole and bell mouth plastic weld gap.
- Application of acoustic felt on the inlet, main duct and register separately or in all.

The results will be discussed for each of the above studies as below.

FAN SPEED VARIATION STUDY

By conducting a number of tests at different fan speeds it was possible to isolate the frequency spectrum spike that was associated with the fundamental Blade Passing Frequency (BPF). The BPF is defined using Equation (1). The BPF was calculated for the baseline and three fan speed variations; the results are listed in Table 1.

$$\frac{BPF\ (Hz)}{Number\ of\ Blades} \times 60 = Fan\ Speed\ (RPM) \qquad (1)$$

Table 1 Calculation results of the blade passing frequency.

Fan Speed (RPM)	**Blade Passing Frequency (Hz)**
1880	1410
2120	1590
2360 **(Baseline)**	1762
2600	1950

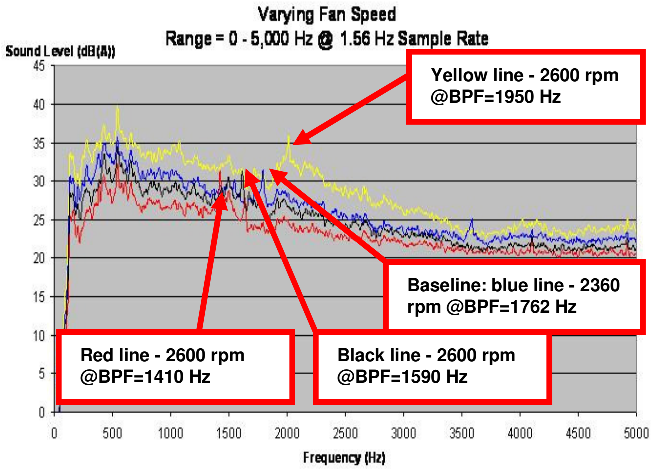

Figure 3 Noise spectral data for four different fan speeds.

The fan wheel used in the system had 45 blades. It is seen from Table 1 and Figure 3 that the fundamental BPF spikes vary in accordance with the calculated BPFs; The fundamental BPF is defined as 1 x BPF; The second order BPF is defined as 2 x BPF. The fundamental and the second order BPFs related spikes are able to be observed in Figure 4 which verifies that the spikes were caused by the blade passing fan rotation.

The polar plot in Figure 4 displays the total Sound Pressure Level (SPL) for each measurement position at an elevation of 1.35 meter. It can be seen that the SPL increased with fan RPM. This is due to the increased flow rate through the system. The SPL also decreased as the fan speed decreased.

MODIFICATION OF THE EXISTING GEOMETRY

The following results were generated by modifying the existing internal geometry of the HVAC system that was subject to air flow. The acoustic data generated with the modified geometry was compared to the baseline noise spectral data. This type of investigation was carried out to provide insight into the acoustic benefit from minor physical changes within the system.

The electric motor on the HVAC system was air cooled. The supply of cool air was directed by a vent system that provided a passage from the wall of the main duct to the base of the motor. Figure 5 shows modification of the motor vent passage hole by sealing it by adhesive electrical tape.

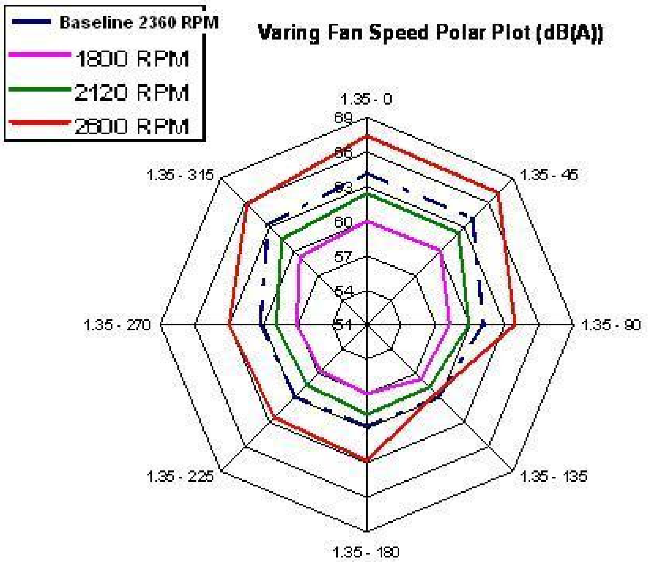

Figure 4 Polar plot for four different fan speeds.

It is noticed from Figure 6 that the fundamental Blade Passing Frequency (BPF) peak was not decreased at 1,780 Hz. This might be an error caused by a poor frequency resolution of the spectral spike. The second order BPF peak was shown to be reduced by 2 dB(A) in SPL at 3,560 Hz. Placing adhesive electrical tape over the hole rendered a smooth surface for exposure to the air flow, therefore, reduced the air flow friction loss and reduced aerodynamic noise at the BPFs. It was observed from Figure 6 that the noise spectrum achieved a 2.5 dB(A) reduction in magnitudes between 370 and 430 Hz for the same reason. In addition, cavity resonance can also contribute to low frequency sound pressure magnitudes. Covering the vent hole eliminated the possibility of air entering the cavity and hence exiting cavity resonance.

The scroll assembly consisted of two plastic injected halves that were welded together. The weld seam run through the bell mouth and down the main duct. On inspection of two test parts, approximately 2mm gaps were found where the injected halves should have been smoothly joined as shown in Figure 7(a) and 7(b). The imperfections were located in the same area of the both test parts which suggested that there was a quality issue with the assemblies manufacture process. Adhesive electrical tape was used to cover the imperfection. It is shown in Figure 8 that sealing the weld imperfection gap did not decrease the fundamental BPF noise peak at 1780 Hz. Again, this might be an error caused by a poor frequency resolution of the noise spectral spike. It is shown that sealing the weld imperfection gap decreased the second order BPF spectrum peak by 2 dB(A) at 3,560 Hz. The removal of the imperfection gap improved the air flow turbulence, thus reduced the BPF spectrum peaks. It is noticed From Figure 8 that the noise spectrum achieved an approximately 2 dB(A) increase in magnitudes between 790 and 840 Hz. The removal of the weld gap meant that air flow was then exposed to a smooth surface rather than a small cavity absorber which would absorb the noise at the frequencies.

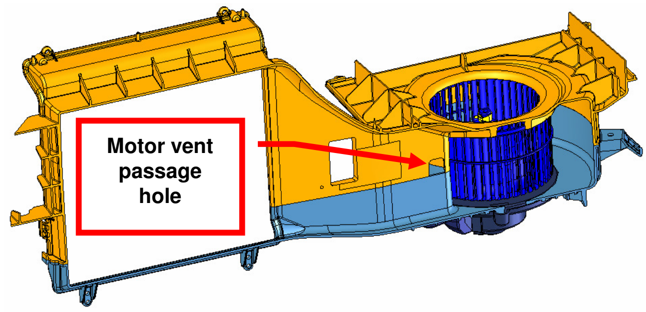

Figure 5 Covered motor vent passage hole.

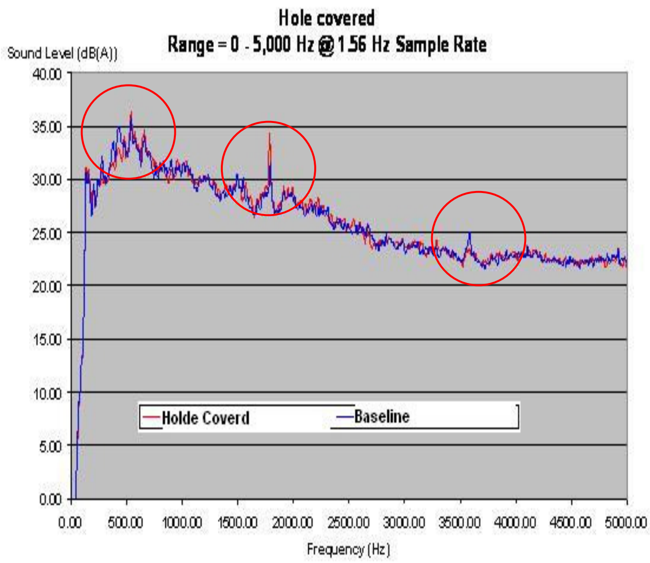

Figure 6 Noise spectral data with the hole covered.

APPLICATIONS OF ACOUSTIC FELT

Acoustic felt material was applied within the HVAC system to eliminate noise. The material “Polyfelt” in the form of 10 mm thick sheets and 1,000 grams per square meter, was cut to size and glued onto the system. Test data generated from the different applications of the material provides insight into which physical areas of the system contribute most to the total noise level and noise frequency spectrum components.

The inlet of the HVAC system in recirculation setting draws air directly from vehicle cabin. The inlet opening is very close to the fan and therefore acts as a path for noise into the cabin. The application of the acoustic felt material in the inlet is shown in Figure 9. It is noticed from Figure 10 that the application of the felt material in the inlet decreased 3 dB(A) in SPL from 2,160 to 2,300 Hz, 2 dB(A) in SPL from 2,860 to 5,000 Hz, and eliminated the second order BPF spectrum peak. The BPF spectrum peak at 1780 Hz did not look decreased from the felt application which might be an error caused by a poor frequency resolution of the peak spike.

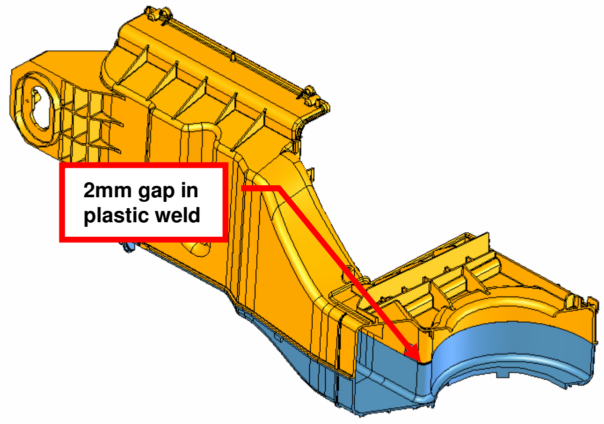

Figure 7(a) Scroll assembly, location of the weld imperfection.

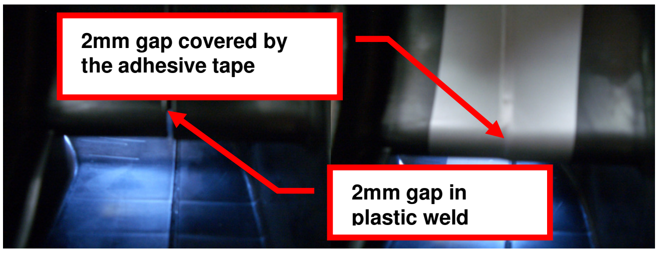

Figure 7(b) Scroll assembly, the weld imperfection on test part.

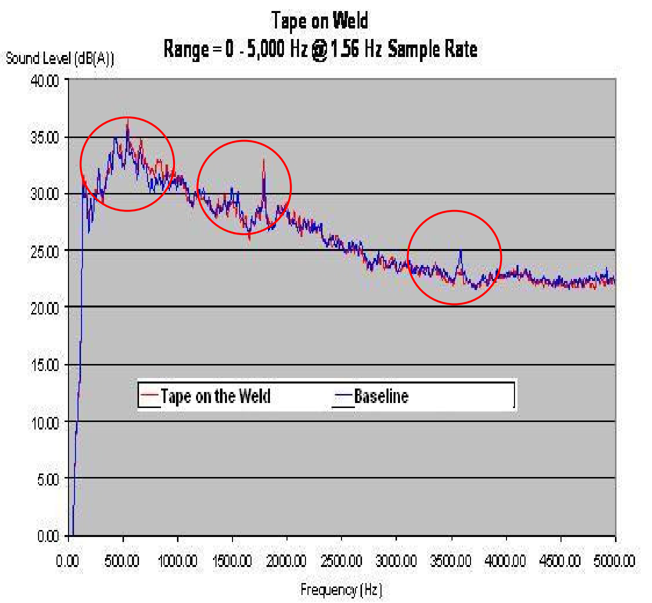

Figure 8 Noise spectral data, the weld gap eliminated.

Figure 9 Inlet assembly, underside showing application of the acoustic felt.

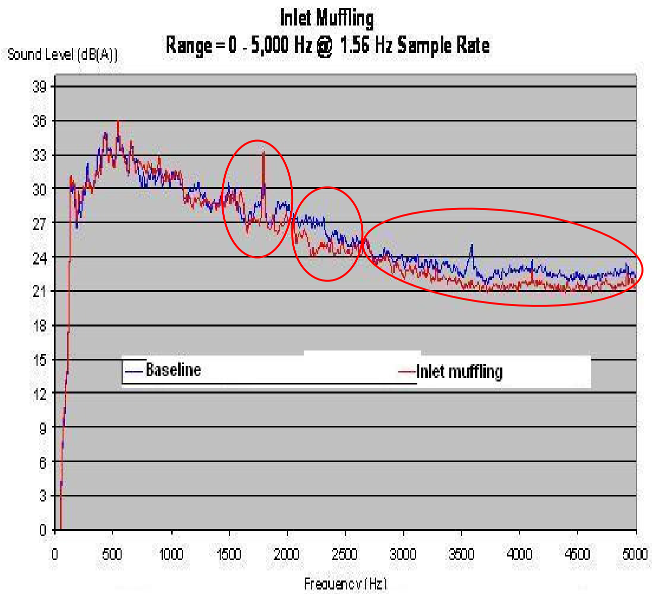

Figure 10 Noise spectral data, the felt material in the inlet.

It is noticed from Figure 10 that the application of the felt material in the inlet decreased 3 dB(A) in SPL from 2,160 to 2,300 Hz, 2 dB(A) in SPL from 2,860 to 5,000 Hz, and eliminated the second order BPF spectrum peak. The BPF spectrum peak at 1780 Hz did not look decreased from the felt application which might be an error caused by a poor frequency resolution of the peak spike.

It is shown from Figure 11 that the acoustic felt material in the inlet reduced the noise by approximately 1.5 dB(A) in SPL at the 45 degree incidence measurement point. This point was closest to the inlet which verified that the acoustic felt material did absorb noise. The polar plot also illustrates that the noise level reduction from the inlet felt application varied according to perceived locations and directions.

The main duct connects the centrifugal fan to the main body of the system. Air flow passes through the duct. The test data using the acoustic felt in this area would highlight effective noise reduction in the corresponding frequencies. Figure 12(a) and 12(b) show that the acoustic felt material was secured in place in the duct and the edges were covered with adhesive electrical tape to lessen the sharp transition for the flow.

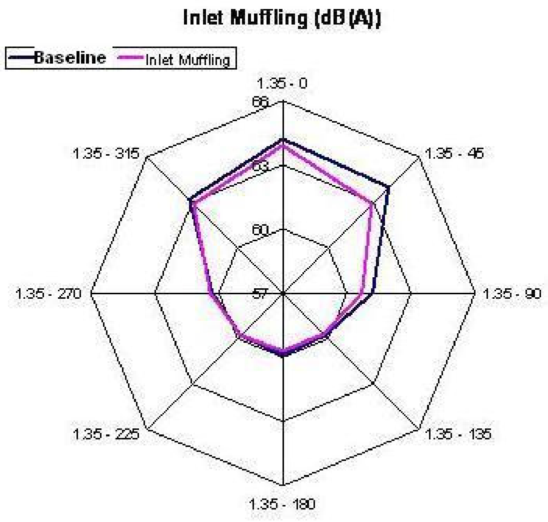

Figure 11 Polar plot, with and without adding the acoustic felt in the inlet.

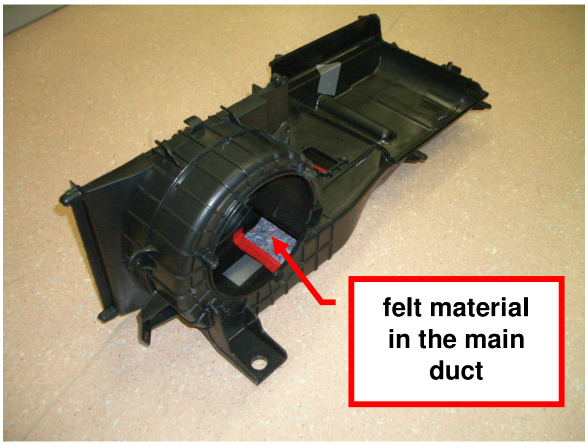

Figure 12(a) Scroll assembly test part, the felt material visible in the main duct.

It is seen from Figure 13 that a significant noise peak reduction was achieved for the second order blade passing frequency, 3560 Hz by applying the acoustic felt material in the main duct. The cross sectional area of the main duct had been decreased by the introduction of the material and caused the air flow rate reduction and more air flow turbulence. In order to maintain the same air flow rate or pressure drop as that in the baseline, the fan speed would have to increase.

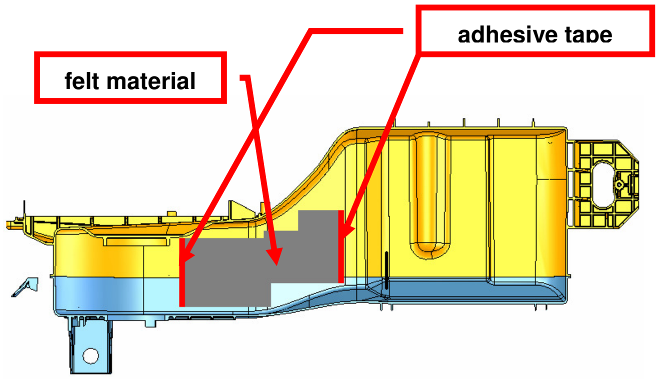

Figure 12(b) Scroll assembly, positioning of the felt material within the duct

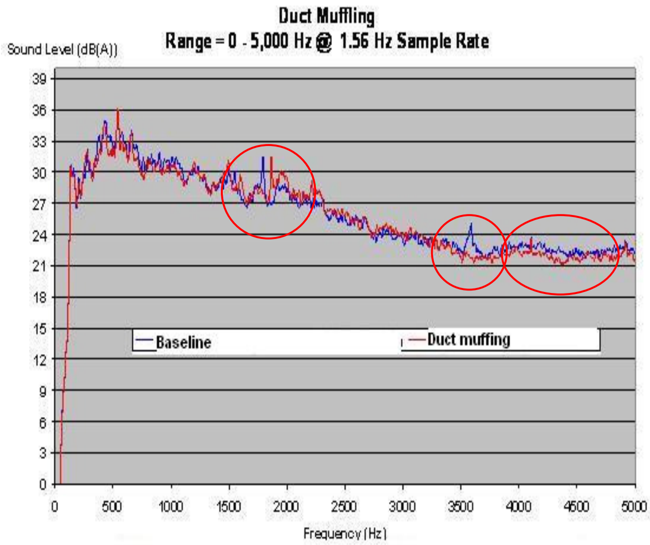

Figure 13 Noise spectral data, the felt material in the main duct

The fundamental BPF had increased by 75 Hz, from 1,781 to 1,856 Hz which suggested an increase in fan speed of 100 RPM as shown in Figure 13. The increase in the turbulence was responsible for the noise level increase between 1,856 Hz and 2300 Hz. The magnitude reduction of the noise spectrum above 3700 Hz was most likely achieved by sound absorption of the applied felt material.

Figure 14 demonstrates that the acoustic felt material in the main duct reduced the total SPL of the HVAC system at 270°, 315° and 0° incidence measurement points. The largest magnitude reduction occurred at the 315° measurement point which was closest to the air registers. This indicates a noise reduction in the exhausted air flow. Although the systems noise output was reduced, the magnitude of the reduction was not significant. The polar plot in Figure 14 also illustrates that the SPL increased at 90° and 135° incidence measurement points which might be caused by the fan speed increase.

To investigate the noise reduction effect due to the main body of the system, the acoustic felt material was applied to the main face air registers. The acoustic felt material was fixed in place but did not directly obstruct the flow from the air registers as shown in Figure 15.

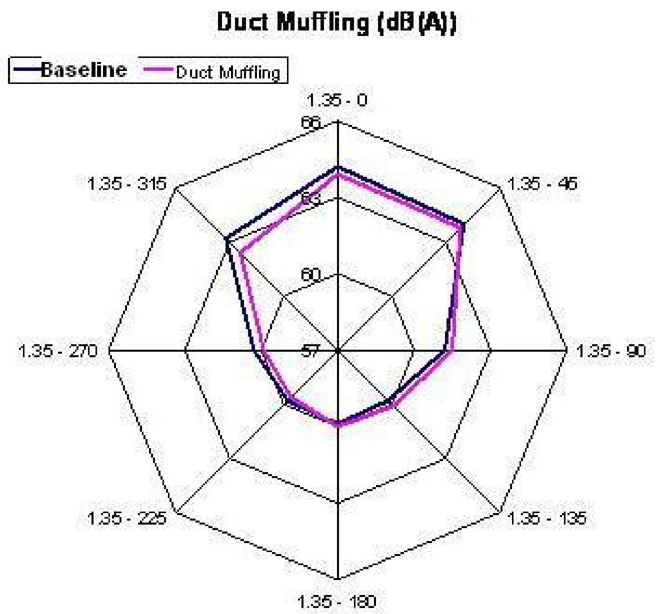

Figure 14 Polar plot, with and without adding the felt material in the main duct.

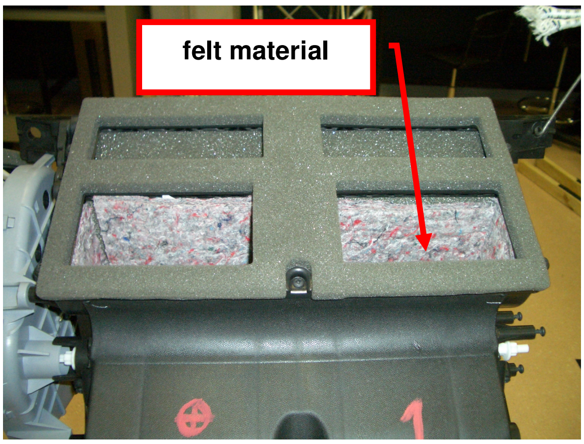

Figure 15 – Air Registers, face vents open and fitted with the felt material.

It is demonstrated from Figure 16 that the noise level was reduced by 1.5 dB(A) in SPL from 3581 Hz to 5000 Hz; the noise level was increased by 2 dB(A) in SPL from the fundamental blade passing frequency, 1781 Hz to the second order BPF, 3570 Hz. The felt material application increased the air flow resistance and turbulence, therefore increased the noise level in the frequency range.

It is seen from Figure 17 that the SPLs for the treatment were increased at all measurement points, except at 270° and 315° measurement points where the acoustic felt material was installed in the air register near by. This SPL increase might be caused by the increase in air flow resistance and turbulence. At the measurement points of 270° and 315°, the effect of the flow resistance and turbulence which increased the SPL was less than the effect of sound absorption from the applied felt material which reduced the SPL.

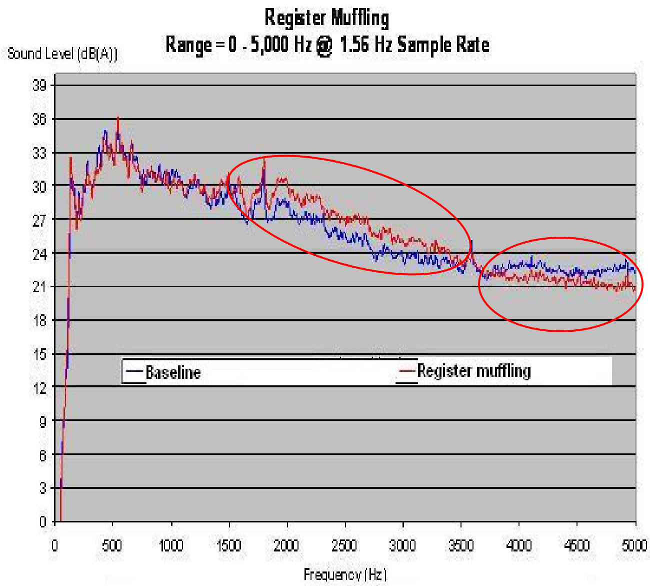

Figure 16 Noise spectral data, the felt material in face air registers.

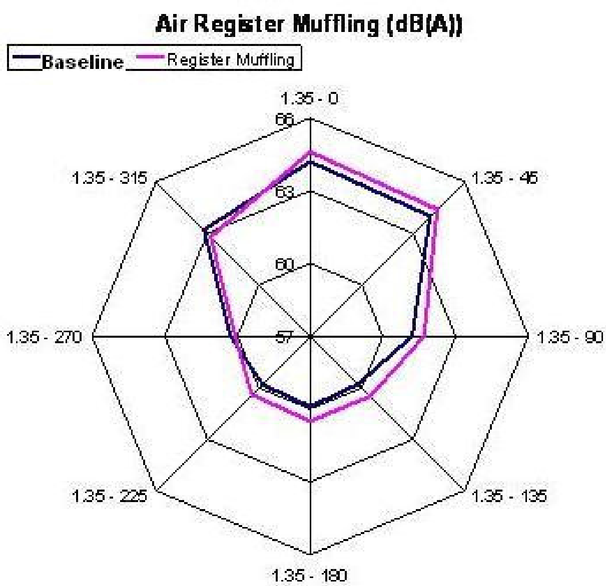

Figure 17 Polar plot, with and without adding the felt material in the air register.

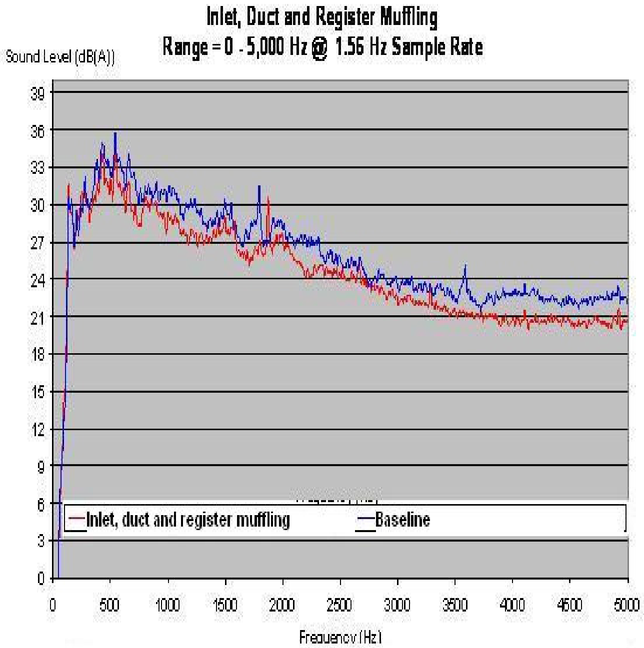

Figure 18 Noise spectral data, the felt material in inlet, duct and face air registers.

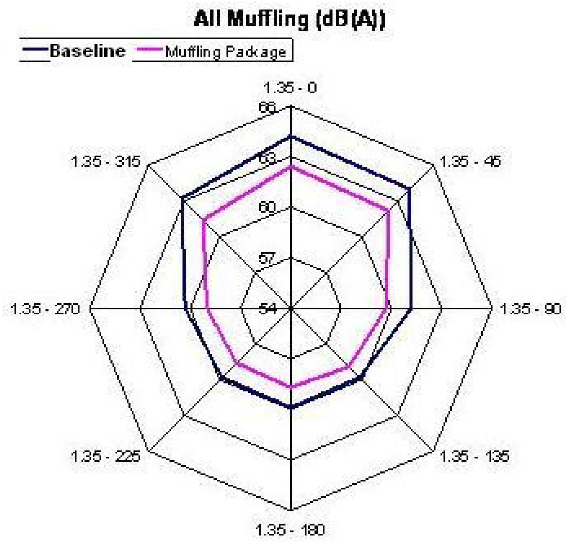

Figure 19 Polar plot, with and without adding the felt material in the three locations.

The decrease in SPL at 270° and 315° indicates that the noise was absorbed at the measurement points closest to the air register by the applied felt material. However, the application of the felt material in the air registers was least effective in the noise reduction comparing with the applications of the felt material in other locations as shown in Figures 10, 11, 13, 14, 16 and 17. The application of the felt material in the inlet was most effective in the noise reduction. This is because sound absorption close to the source is always most effective.

Figure 18 and 19 shows noise spectra and SPL position polar plot for applying the felt material in inlet, duct and face air registers. It is noticed from Figure 18 that a reduction of 2 dB(A) in SPL was achieved from 530 to 830 Hz, a reduction of 2.5 dB(A) in SPL from 850 to 1500 Hz, and a reduction of 2~2.5 dB(A) in SPL from 1,900 Hz to 5,000 Hz had been achieved. The installation of the felt material in the three locations restricted air flow and degraded the system performance which had to be compensated by increasing the fan speed by 110 rpm or up-shifting the fundamental BPF by 83 Hz to maintain the same pressure drop or air flow rate as that in the baseline. Figure 19 shows a decrease of 2 dB(A) in the SPL readings taken at all positions. This result illustrates that the application of the acoustic felt material to each location produced a net effect greater than the sum of individual effects. It indicates that the changes were not mutually exclusive in their ability to reduce noise.

CONCLUSION

The following conclusions have been drawn from the noise refinement studies through the acoustic measurements on the lab test rig.

- The fundamental and the second order blade passing frequencies (BPF) are identified and related to the fan speed. Increasing the fan speed not only increases the BPFs, but also increases the total noise level and the magnitudes of the predominant noise spectral peaks.

- Covering the motor vent passage hole and sealing the weld gap do not change the total noise level, but do reduce the noise spectral peaks at the blade passing frequencies and affect cavity resonance peaks at the low frequencies.

- Application of the acoustic felt is most effective at the inlet area and least effective at the face of air registers. The most noise reduction is achieved at the points closest to the felt application area. Application of the felt material in the HVAC system is able to reduce at least 2 dB(A) for the total noise level and the noise spectral peaks in the frequency range above 200 Hz.

- The poor frequency resolution of the spectral spike at the fundamental blade passing frequency is not able to reflect the magnitude change of the spectral spike from the geometry modifications and the felt material applications. The n^{th} octave analysis is recommended to replace the FFT analysis in a future investigation.

ACKNOWLEDGMENTS

Authors would like to thank final year students Adrian Dipierro, Jeremy Wee and Brian Gleeson from RMIT University for providing experimental data, Simon StHilland and David Christensen for providing the test parts, Cad data, supports and advices.

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