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Noise evaluation of automotive A/C compressor

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Abstract
Passenger compartment's interior noise and thermal performance are essential criteria for the driving comfort of vehicles. The air-conditioning system influences both field of comfort. It creates comfortable thermal conditions. On the other hand, the noise radiation of the air-condition system's components can be annoying. The blower, the air distribution ducts and the registers affect air rush noise. In some cases, the refrigerant flow creates hissing noise. Such noise has a great influence on vehicle acoustical comfort and on overall quality perception of a vehicle Therefore, the acoustic performance of air-condition compressors become more important for passenger comfort. At engine idling and at extreme temperatures the air-condition compressor can be audible as the significant sound source. However, the aim of this paper is to quantify air-borne noise characteristics of vehicle air-condition compressor. A simulated experimental model comprises a small wooden box with dimensions of 0.5 x 0.5 x 0.5 m represented the principle of hemi-anechoic room was designed and acoustic characteristics of the sound field inside the box were determined. The air-condition compressor characteristics parameters considered in this paper are fan position and electric motor speed. In addition, a single number of the air column natural frequency is calculated. The results indicate that significant information can be obtained in order to investigate the vehicle air-condition compressor and consequently improve the vehicle interior quietness.

Keywords: Air column natural frequency, Air conditioning compressor, Airborne noise, Noise generation, Sound pressure level.

1. Introduction

The refrigerant compressor excites pressure pulsations of the refrigerant by every compression event. These vibrations are transmitted through the air-condition-line to the passenger compartment where they can be radiated by the air-condition housing. In addition, the compressor propagates air-borne noise as well as structure-borne noise through its' mounts. Therefore air-condition -system development is making effects to design silent compressors [1-4].
For an accurate determination of the sound propagated along a duct system, a complete wave analysis is needed, based on a set of sound wave generation and transmission properties for each of the elements of the system; these properties are complex functions of frequency, different for each acoustic wave mode, and dependent on geometry and airflow. However, the noise through air circuits for ventilating or air-conditioning ducts is usually estimated by means of simpler sound power methods, that the data of the generation or attenuation of the sound energy for each element and frequency band, such as the formulas and tables indicated by [5-6] It is well-known that if a secondary sound source operates inside a

reverberation chamber, a diffuse sound field is generated because of the sound-reflecting non-parallel interior surfaces.
This diffusivity of the sound field is described by the undetermined phase shift between sound pressure and sound particle velocity. In an ideal reverberation chamber the sound pressure level is a constant value at any space coordinate of the room. In this case, the sound pressure level is proportional to the sound power level emitted by the sound source inside [7-10]. The identification of alternator noise mechanisms of mechanical, aerodynamic and electromagnetic are presented in [9, 10], where the alternator was reconfigured in many different ways, so as to separate each individual noise source. Noise generated by each configuration running at different speeds was measured, and its characteristics were analyzed. To accomplish these goals, the diagnosis of the alternator noise is conducted in two cases, i.e., with load and without load (unloaded). In the case of unloaded, the alternator mechanical and aerodynamic noises are identified, while in the case of with load, focus is made on electromagnetic noise under various loading condition and at different speeds. It is indicated that at speed range used in this study, the mechanical noise is predominant followed by aerodynamic noise with the electromagnetic noise creates the lowest of the total alternator noise level.
Recently the contribution pattern of noise sources to the overall interior noise in a vehicle has changed. A large number of secondary sound sources, previously masked, have become perceptible to passengers with the reduction principally of engine noise levels and must be considered by the acoustical engineer. These noises, nowadays, play an important role in the passenger comfort and have a considerable influence on the quality perception not only of the source but of the whole automobile. Some of these sources are active only occasionally, like the electric seat adjustment or electric power windows. Others, like the air-condition system, are likely to be operating as long as the automobile is in use. This system can operate with alterations to the air temperature and velocity if it is in manual mode, or can change these parameters over a larger time window if it is in automatic mode, where these parameters and the state of the compressor (turned on or off) are adjusted according to the thermal situation inside the vehicle. All these working conditions introduce a large variability in the air-condition noise. In hot climate conditions, it operates at high ventilation speeds, especially when no air-conditioning unit, which is very usual in hot climate conditions, is installed in the vehicle. Under these conditions, the contribution of the air-condition noise to the acoustical comfort of the passenger is very significant. In addition to the influence that air-condition noise has on the comfort and quality perception it can modify the thermal comfort inside the car due to the interaction of thermal and acoustical perception [11-13].
However, the aim of this paper is to quantify air-borne noise characteristics of vehicle air-condition compressor. A simulated experimental model comprises a small box represented the principle of anechoic room was designed and acoustic characteristics of the sound field inside the box were determined. The air-condition compressor characteristics parameters considered in this paper are fan position and electric motor speed. In addition, a single number of the air column natural frequency is calculated.

2. Vehicle air-condition system

2.1 General

The early history of transportation systems starts mainly with the horse drawn carriage. This was eventually surpassed by the invention of the vehicle. Early automobiles had cabin spaces that were open to the outside vehicle. This means that the occupants had to adjust there clothing to allow for different weather conditions. Closed cabin spaces were eventually introduced which required heating, cooling and ventilating to meet customer expectations. Early heating systems included heating clay bricks and placing them inside the vehicle or using simple fuel burners to add heat to the vehicle's interior.
Ventilation inside the vehicle was achieved through opening or tilting windows or the windscreen; vents were added to doors and bulkhead to improve air circulation and louvered panels were the equivalent to our modern air ducts. Air flow was difficult to control because it was dependent upon the vehicle speed and sometimes would allow dirty, humid air which contained fumes to enter the interior from the engine compartment. Cooling could be as simple as having a block of ice inside the vehicle and allowing it to melt! Eventually a number of design problems were overcome, these included air vents at the base of the windscreen for natural flow ventilation and electric motors to increase the flow at low speeds. Eventually heat exchangers were introduced which used either the heat from the exhaust system or water from the cooling system as a source, to heat the inside of the vehicle cabin. Early cabin cooling systems were aftermarket sourced and worked on evaporative cooling. They consisted of a box or cylinder fitted to the

window of the vehicle. The intake of the unit would allow air to enter from outside and travel through a water soaked wire mesh grille and excelsior cone inside the unit. The water would evaporate due to absorbing the heat in the air and travel through the outlet of the unit which acted as a feed to the inside of the vehicle. The water was held in a reservoir inside the unit and had to be topped up to keep the cone wet otherwise the unit would not operate. The air entering the vehicle would be cool if the relative humidity of the air entering the unit was low. If the relative humidity of the air was high then the water could not evaporate. When the unit was working effectively it would deliver cool saturated water vapor to the inside of the vehicle which raised the humidity levels. These units were only really effective in countries with very low humidity.
The air-conditioning system works on a continuous cycle (Figure 2). A compressor receives low pressure heat laden refrigerant vapor from the evaporator. This increases the temperature from approximately. The increase of temperature and pressure let the refrigerant to be above its boiling point. The compressor discharges superheated refrigerant vapor to the condenser [14].

2.2 Air-conditioning components

- The compressor

The function of the compressor is to compress and circulate superheated refrigerant vapor around a closed loop system (any liquid or dirt will damage the compressor). Compressors vary in design, size, weight, rotational speed and direction and displacement. Also compressors can be mechanically or electrically driven. Some compressors are variable displacement and some are fixed .The compressor uses 80% of the energy required to operate an air-conditioning system.

- Handling unit
    - The handling unit comprises the following:
    - The refrigerant fluid
    - The receiving cooling fan
    - The direr/ accumulator
    - The expansion valve
    - The fixed orifice valve
    - The evaporator
    - Basic control switches

3. Enclosur column of air natural frequency

Consider the sound pressure response in a very small enclosure. Frequently, in the noise control problems, a noise source is enclosed in a very small box to prevent it from radiating noise to the exterior, as conceptually represented in Figure 1. When the nose source has a frequency low enough so that the wavelength of the sound is long compared to the largest dimension of the box, the sound pressure produced by the source will be uniform throughout the interior cavity. Frequently, the walls of the enclosure will be damped by applying panel-damping treatment or absorption material over the walls.
The uniform, steady-state sound pressure in such enclosures can be shown to be $P(t) = p \cos(\omega t + \theta)$, where [15]

$$p = (\rho c2q) / V[(\omega + 2\delta j)2 + (2 \delta r)2]^{1/2} \quad N/m2 \qquad (1)$$
$$\theta = \Phi - \tan-1 (\omega + 2\delta j) / 2\delta r \quad rad$$

Here ρ is the air density, c is the speed of sound, V is the enclosure volume, $Q(t) = q \cos(\omega t + \Phi)$ is the volume velocity of the noise source operating at the forcing frequency $f = \omega / 2\pi$ (Hz) with the amplitude q and phase Φ, and $\delta = \delta r + I\,\delta i$ is a complex damping factor ($I = \sqrt{-1}$) that accounts for the acoustical impedance $Z = R + iX$ of the wall area A, where

$$\delta r = cA/ 2V \, Re \, \rho c/Z \quad , \quad \delta i = cA/ 2V \, Im \, \rho c/Z \qquad (2)$$

For the interior noise source in Figure 1, Q(t) is taken as positive for outward volume flow (m3/s).
From Eq. (1) the magnitude of p of the sound pressure in the enclosure depends not only on the noise source amplitude q and forcing frequency $f = \omega / 2\pi$ but also on the enclosure volume V and total wall impedance in δ. For a rigid-wall impedance, $(|Z|\to\infty)$, so that $\delta r , \delta i = 0$, and

$$p = (\rho c2q) / \omega V \quad N/m2, \qquad (3)$$
$$\theta = \Phi - \tfrac{1}{2} \pi r \quad rad$$

The mass of the column of air is $M = \rho V$; for a mass-spring model of the box walls, and $M\ddot{Y} = PA$ where, $\dot{Y}$ is the velocity of the air column. Since $\ddot{Y} = \omega \dot{Y}$, $q = A \dot{Y} = A^2p/ \omega M$ can be obtained. Substituting this for q in Eq. (3), gives $(1- \rho c^2A^2/ \omega^2 MV)\ p = 0$ or $(1- K_{air}/ \omega^2 M)\ p = 0$, where $K_{air} = \rho c^2A^2/V$ is the air stiffness, the column air natural frequency (ω_o) is then

$$\omega_o = \sqrt{K_{air}/ M} = c \sqrt{A / LV} = c/ L \qquad (4)$$

where L is the air column length, in m.

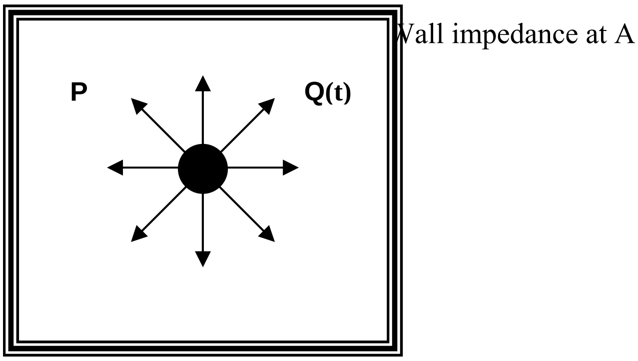


Figure 1. Sound in small enclosure with impedance boundaries generated by interior noise source

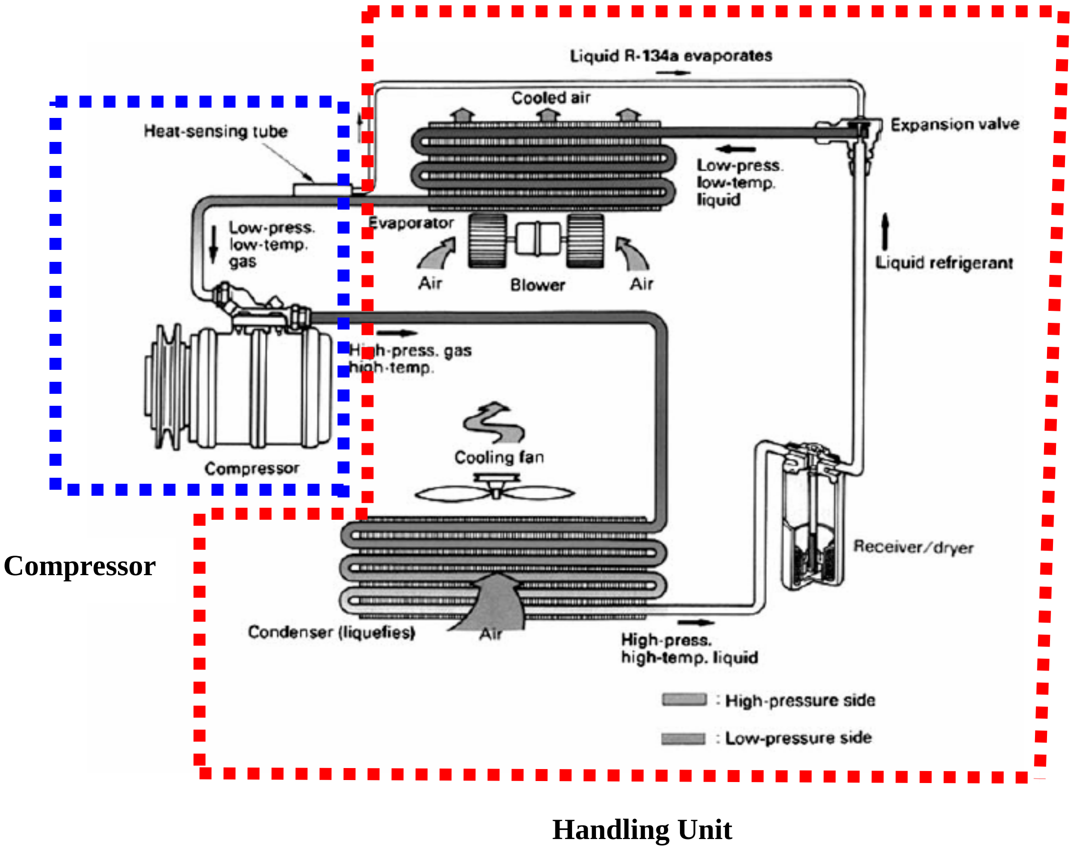


Figure 2. Vehicle air-condition system

4. Test stand and measuring system

4.1 Test stand setup

Figure 2 shows the test stand setup for measuring noise spectra of the piston compressor considered. The compressor was driven by 1.5 hp electric motor (its maximum speed is 1440 rpm). An inverter was used to run the compressor at the desired speed. The principles of hemi-anechoic room for the fabricated wooden box are considered, the hemi-anechoic room is used to test sources that normally are mounted on or operate in the presence of a reflecting surface, where the surfaces of hemi-anechoic room are made highly absorptive by lining them with deep sound-absorptive materials. The lining typically consists of wedges made of glass fiber with thickness of 20 mm. The lowest frequency at which hemi-anechoic room can be used depends on the room volume and the depth of the wedges. The compressor was then enclosed by a wooden box with dimensions of 0.5 x 0.5 x 0.5 m treated with highly sound absorptive materials on the interior and exterior surfaces. Also, the floor on which the piston compressor was mounted as well as the surrounding areas was treated with sound absorptive materials, while the ceiling is made from glass. By this way, sound radiation from the electric motor and reflections from nearby surface were suppressed to the minimum levels. The box has flat, reflecting and sound-absorptive walls except ceiling. Such wooden box represented a hemi-anechoic room mentioned above where it is necessary to measure accurately the unperturbed sound radiated by the compressor.

4.2 Measuring system

Measurements of sound pressure levels (SPLs) were carried out in the in the box as shown in Figure 3. Bruel & Kjaer (B&K) portable and multi-channel PULSE type 3560-B-X05 with microphone and preamplifier type 4189A-021 were used. The B&K PULSE labshop with measurement software type 7700 were used to analyze the measurements. Figures 4, 5, 6 and 7 illustrate handling unit, the tested compressor, the isolated test box and the measuring and analysis equipment respectively. The motor speed range is limited to be up to 3000 rpm and a constant discharge pressure in the refrigerant cycle of 20 bar. A slight variation of discharge pressure (± 0.3 bar) during measurements is existed.

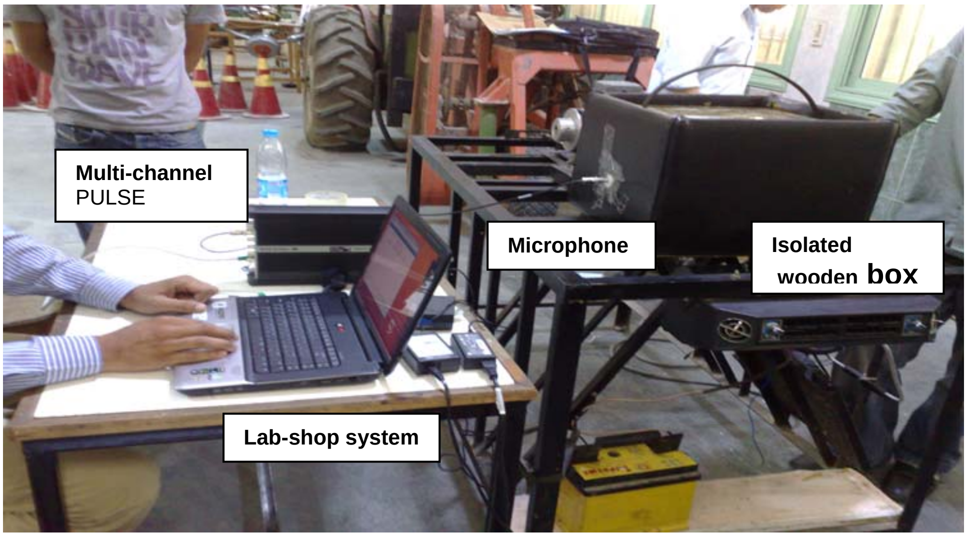


Figure 3. Test stand setup

Figure 4. Handling unit

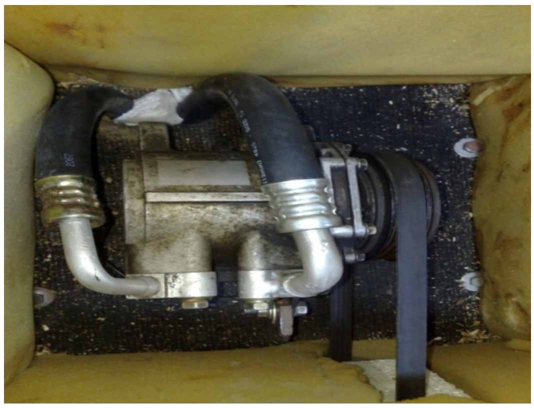


Figure 5. Tested compressor

Figure 6. The isolated wooden box

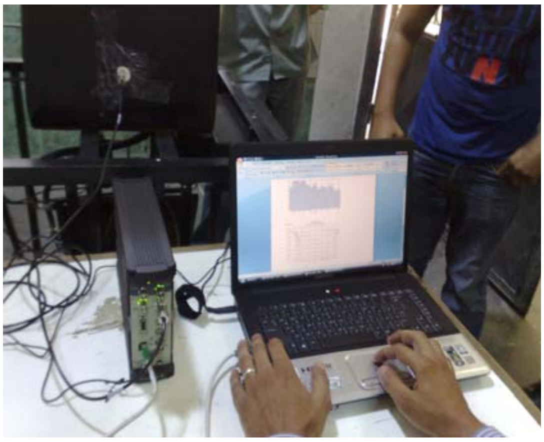


Figure 7. Measuring and analysis system

5. Results and discussions

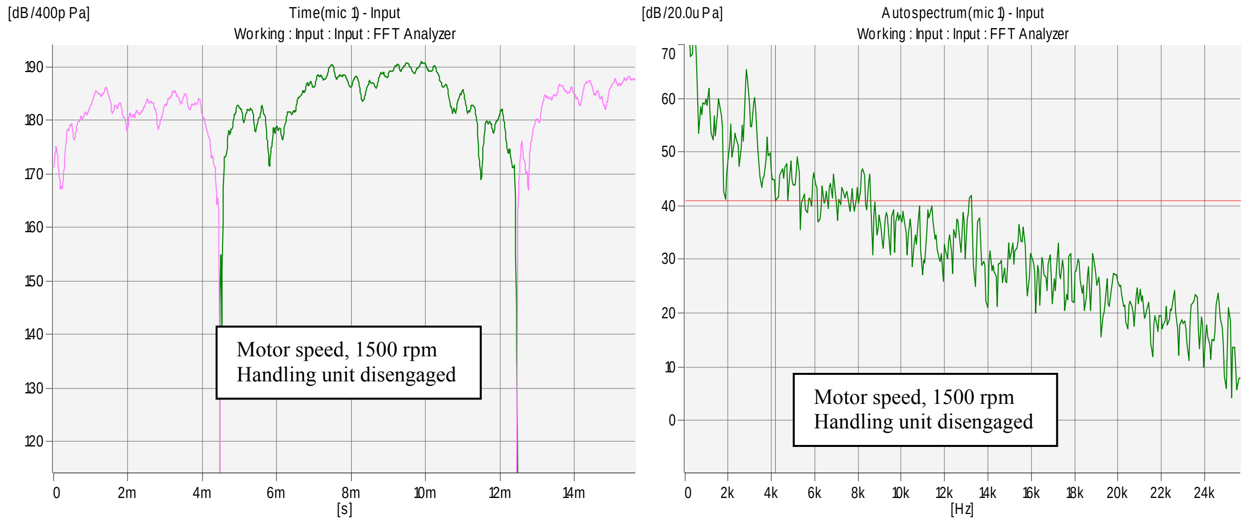
In Figure 8, where the motor speed is 1500 rpm and the handling unit is disengaged, the sound pressure level (SPL) measured within the box in time domain (Figure 8a) and in frequency domain (Figure 5b). This indicates high levels in the frequency ranges up to 6 kHz, while the levels of the remaining frequency are lower and decreases as the frequency is increased. The influence of the motor speed in the case of disengaged handling unit on the measured SPLs at speed of 1500 rpm is presented in Figure 9, where the 1/3-octave SPL is increased with the increase of the motor speed dispite some small discrepancies exsited in the 1/3-octaves range between 500 Hz to 2 kHz (Figure 9b). This may be attributed to the influnce of air column natural frequency and compressor structure rigidity resonance frequencies.

In Figure 10, where the motor speed is 1500 rpm, and load the handling unit is engaged at position 2, the sound pressure level (SPL) measured within the box in time domain (Figure 10a) and in frequency domain (Figure 10b). The whole spectrum levels are slightly higher when compared with those present in (Figure 8b), particularly around the air column natural frequency. The same discussion stated for (Figure 8b) can applied here. The influence of of the handling unit position on the measured SPLs at speed of 1500 rpm is presented in Figure 11, where the 1/3-octave SPL is slightly increased with the increase of the handling unit position dispite some small discrepancies exsited allover the entire spectrum (Figure 11b). This may be attributed to the compressor structure rigidity resonance frequencies.

The influence of the motor speed in the case of engaged handling unit of pistion 2 on the measured SPLs within the box is presented in Figure 12, where the 1/3-octave SPL is increased with the increase of the motor speed (Figure 12b). The increase of SPLs in this case is much higher than those measured for the engaged handling unit (Figure 9b) without any discrepancy.

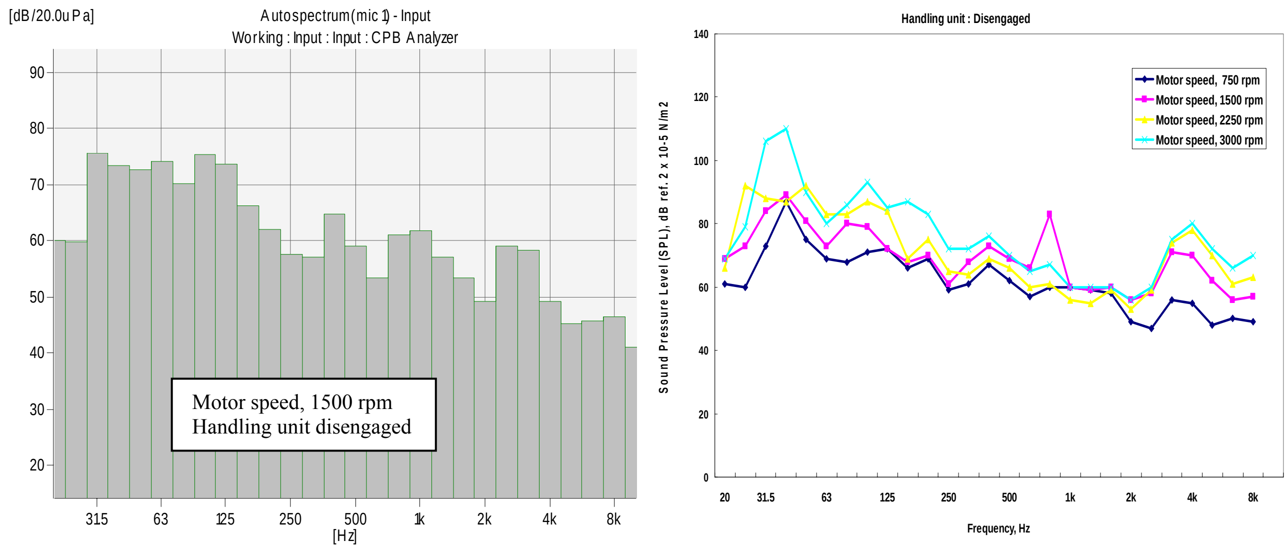
The RMS averages are applied at speed of 1500 rpm where their results are shown in Fig. 13,a and indicate that almost the RMS averages are almost the same either the case of disengaged or engaged case. A comparison between the broadband SPL averages measure in the cases of engaged and disengaged is shown in Figure 13,b, where the broadband SPLs for the engaged case are higher than those in disengaged case particularly for the positions 2 and 3 .

In order to apply aforementioned equation (4), the air column length within the box is considered to be 0.5 m and c is the speed of sound in the air and equal to 347.3 m/s. The resultant natural frequency is nearly 695 Hz. It can be seen that around this frequency, the sound pressure level are high for all the cases measured. This is clearly shown by the arrows drawn in the Figure 14, where Figure 14a for the case of motor speed of 750 rpm and at handling unit engaged at position 2, and Figure 14b for the case of motor speed of 2250 rpm and at handling unit engaged at position 2.



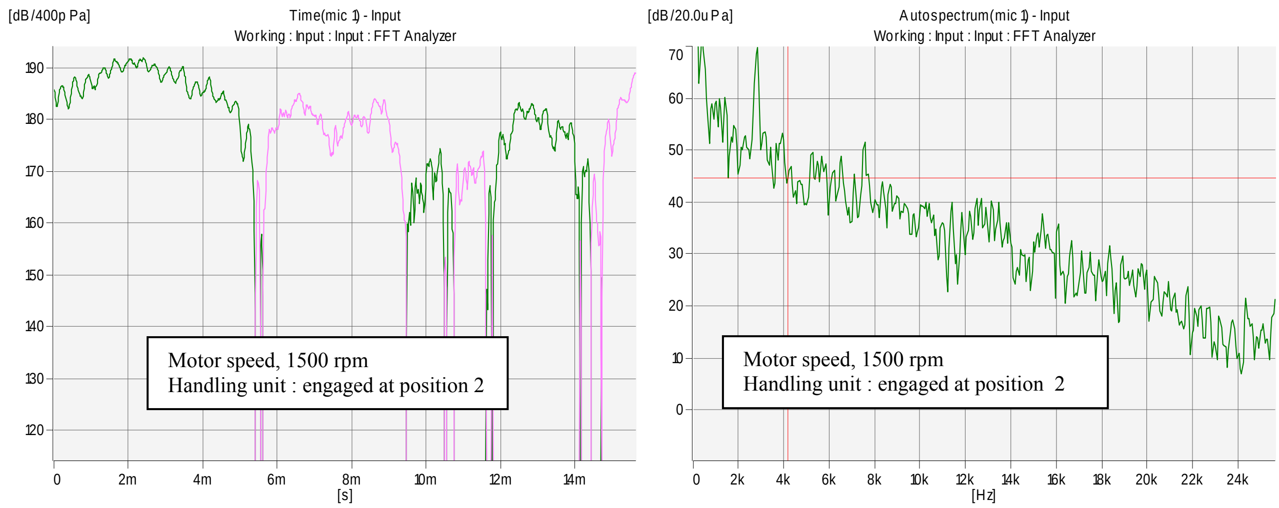
(a) (b)

Figure 8. (a) Time history of sound pressure level; and (b) frequency domain of sound pressure level



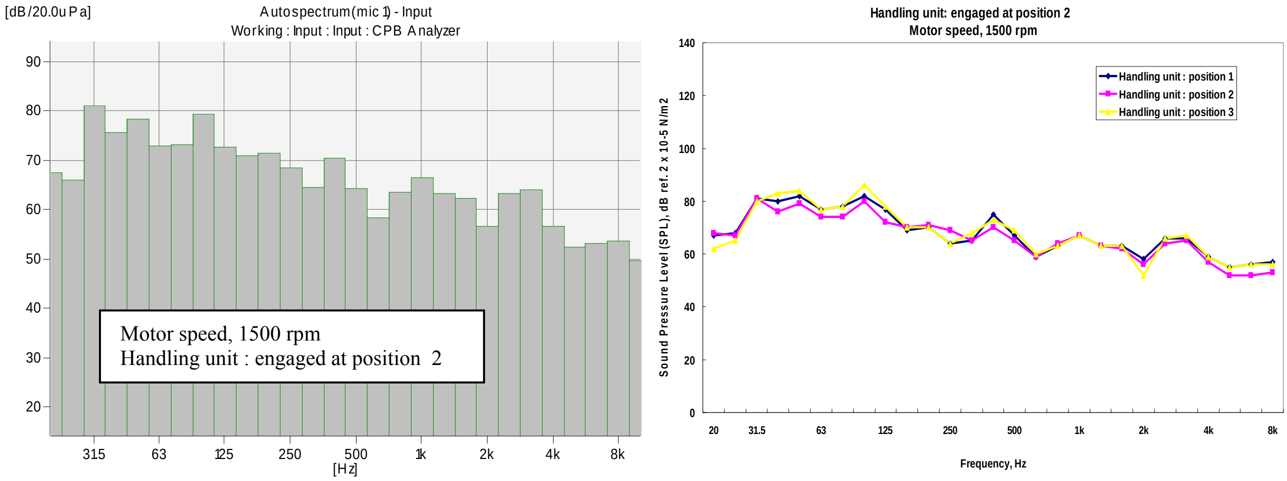
(a) (b)

Figure 9. (a) 1/3-octave sound pressure level; and (b) 1/3-octave sound pressure level



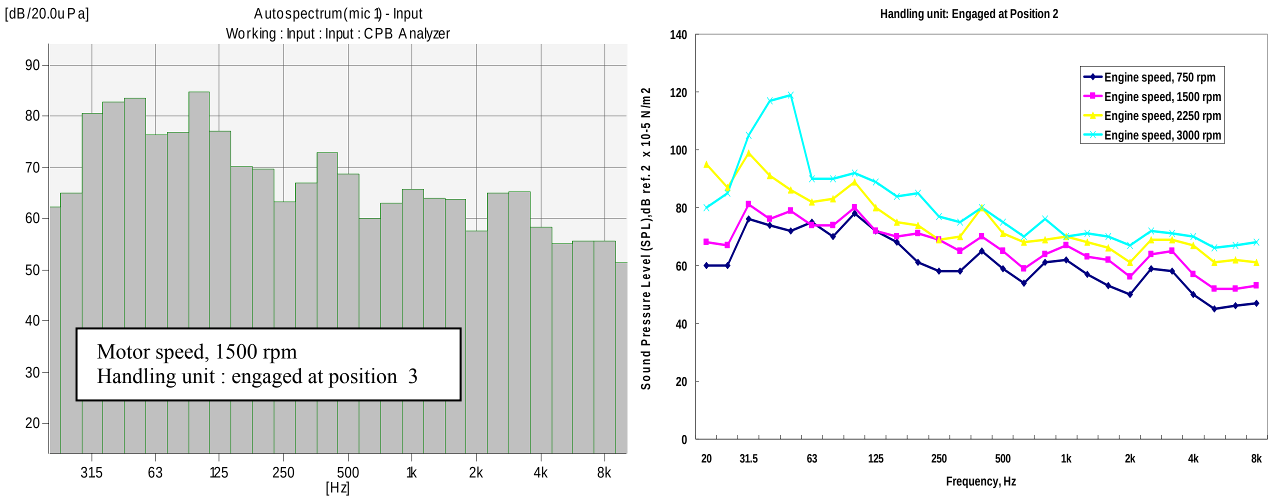
(a) (b)

Figure 10. (a) Time history of sound pressure level; and (b) frequency domain of sound pressure level



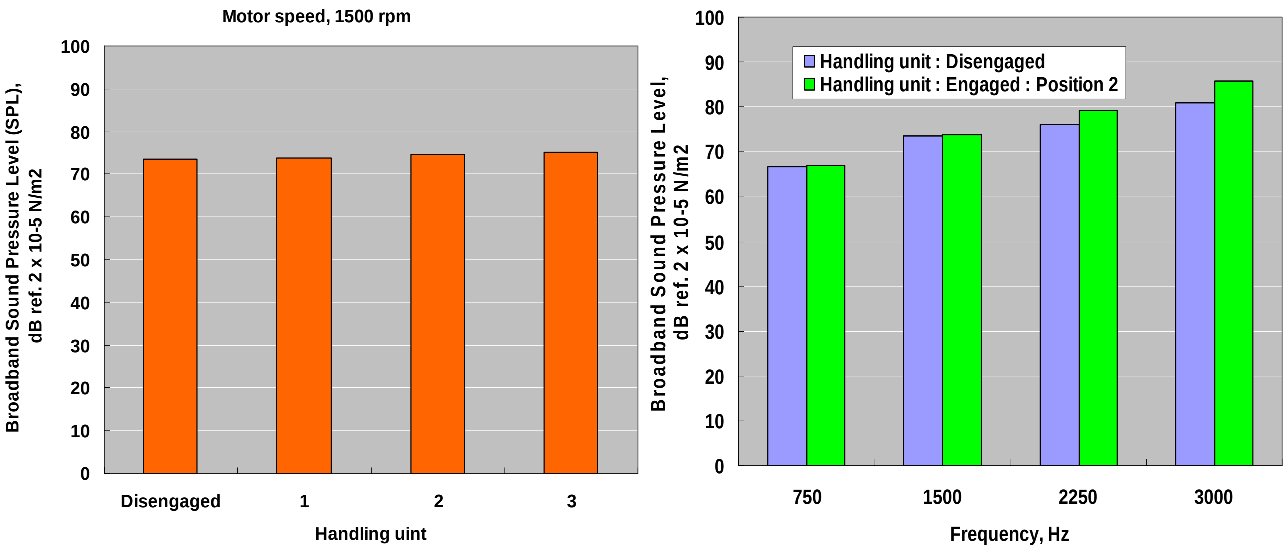
(a) (b)

Figure 11. (a) 1/3-octave sound pressure level; and (b) 1/3-octave sound pressure level



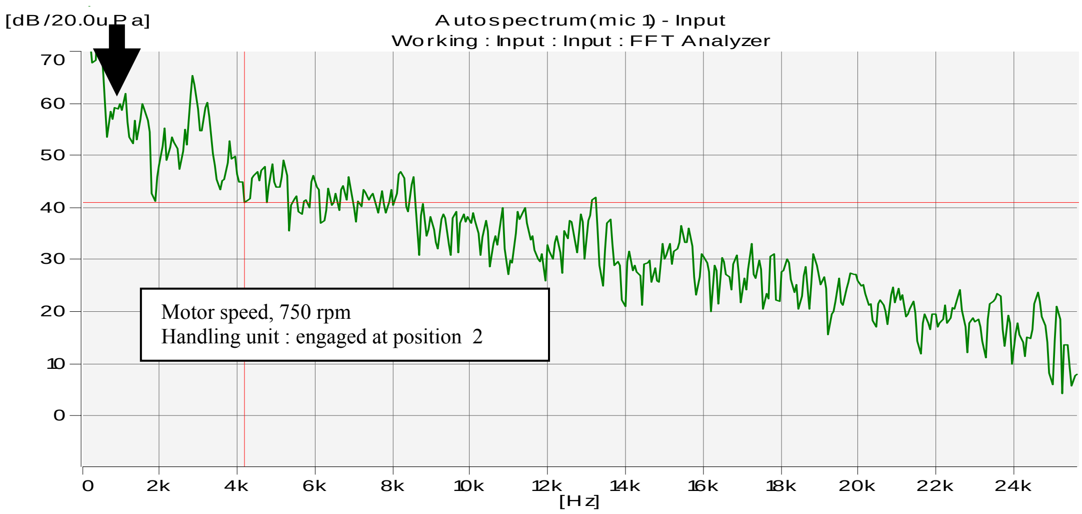
(a) (b)

Figure 12. (a) 1/3-octave sound pressure level; and (b) 1/3-octave sound pressure level

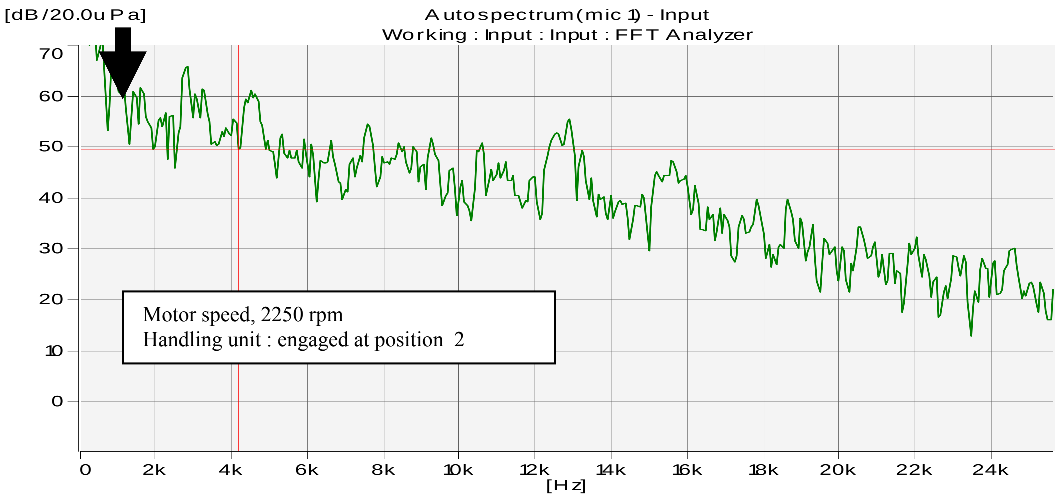


(a) (b)

Figure 13. (a) Influence of load on broadband sound pressure level; and (b) influence of crack dimension on broadband sound pressure level



(a)



(b)

Figure 14. (a) Locations of the air column natural frequency for healthy case; and (b) locations of the air column natural frequency for faulty case

6. Conclusions

1. It can be seen that the sound pressure exhabits high levels in the frequency ranges up to 6 kHz, while the levels of the remaining frequency are lower and decreases, . This is attributed to the air column natural frequency and the compressor structure rigidity resonance frequencies. However, if any reduction technique for the compressor noise is required, both the air natural frequency and and the compressor structure rigidity resonance frequencies must be considered.
2. Based on the broadband noise in terms of sound pressure level (SPL) measured within the box, the influence of changing the handling unit position and motor speed on the broadband SPL, which indicate that the increase in either handling unit or motor speed is accomplished by increase in sound pressure level. Based on that, the noise characteristics of the compressor and handling unit play an important role for determining the noise quality of the whole air-condition system.
3. The air column natural frequency is found to nearly 695 Hz. It can be seen that around this frequency, the sound pressure level are high for all the cases measured. Furthermore, the comparison between the broadband SPL averages measure in the cases of engaged and disengaged is made, where the broadband SPLs for the engaged case are higher than those in disengaged case particularly for the positions 2 and 3 .

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