DESIGN OF WAVEGUIDE VIBRATION ABSORBER WITH GROOVED CIRCULAR DISK FOR REDUCING REFRIGERATOR VIBRATION AND NOISE

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Based on the theory of vibration and the principle of impedance, a design method is proposed for a new type of waveguide vibration absorber consisting of an energy guide rod and a grooved circular disk. Compared with traditional dynamic vibration absorbers, this type of absorber can effectively absorb broadband vibration energy in the middle-low frequency range, and ultimately reduce the noise radiated by a vibrating body. This method is applied to design and develop the vibration absorbers for a beverage refrigerator. Finite element simulation results of the compressor casing with and without installing the designed absorbers show that the vibration level of the casing in the frequency range below 1 kHz is significantly reduced. The results of vibration and noise measurement conducted in a semi-anechoic chamber also show that after the installation of vibration absorber, the vibration of the compressor is reduced by 1.0 dB ~ 4.3 dB in the frequency range of 125 Hz ~ 1 kHz, and the maximum noise reduction reaches 4.0 dB at 500 Hz.

Keywords: beverage refrigerator, reciprocating compressor, vibration, noise, dynamic vibration absorber

1. Introduction

Dynamic vibration absorber was first proposed by Ormondroyd and Den.Hartog; it absorbs the vibration energy of a body based on the principle of resonance [1]. The simplest dynamic absorber is a mass-spring system, i.e. the oscillator-type vibration absorber, which attenuates the vibration level of a vibrating body through the reaction force at the resonance of the oscillator [1]. Later, because classical oscillator absorbers normally have immutable dynamic parameters and narrow frequency range of vibration absorption in the vicinity of the resonant frequency of the controlled system, dynamic vibration absorbers with distributed parameters were proposed. Michel.A introduced these vibration absorbers in detail [2]. Beam type vibration absorber is the most simple distributed parameter dynamic vibration absorber, which including uniform beam and variable cross-section beam. Not only its effective frequency range can be increased [3, 4], but also the basic parameters can be adjusted [5]. Based on the beam type vibration absorber, ring beam vibration absorber is developed [6, 7]. They also absorb vibration with a relative wide frequency range and can be used in different occasions. These absorbers with multi-order resonance frequencies can absorb the vibration energy in multiple frequency bands in the vicinity of its resonant frequencies; however, they have no obvious effect on the vibration in other frequency bands.

Aiming at improving the limitation of dynamic vibration absorber, in the 80s of the last century, Ungar and Kurzwell [8, 9] put forward a waveguide vibration absorber, which has a characteristic distinct from the dynamic vibration absorber: it is not only able to absorb the vibration energy near its resonance frequency, but also has very good vibration absorbing effect in a wide frequency range. The waveguide vibration absorber is composed of two components, the energy guide rod and the energy dissipation device. The beam type energy dissipation device was the subject of initial research [10], and later the new idea of a circular disk with narrow grooves was proposed as a means of great potential to improve the energy dissipation rate of waveguide absorber because the grooved disk can be regarded as the combination of several beam-type energy dissipation devices [11, 12].

In this paper, a design method is proposed for developing such a new type of waveguide absorber with grooved circular disk. The effective frequency range of vibration absorption is oriented to the inherent vibration characteristics of the target vibration source, and reasonable design parameters are obtained by optimizing the energy-dissipation rate of the energy-dissipation device and the energy reflectivity of the guide rod. Hence the maximum attenuation of source vibration level can be realized in the effective frequency range of vibration absorption. Finally, the proposed design method is used to develop vibration absorbers for suppressing the vibration of the compressor in a beverage refrigerator, and both finite-element simulation and experiments are conducted to test the performance of the designed absorbers in vibration and noise reduction.

2. Theory of the proposed design method

To achieve high-level absorption of vibration energy in a specified frequency range, the natural vibration characteristics of the vibration source is first analyzed to determine the target frequency range of vibration absorption. Next, the new waveguide absorber consists of an energy guide rod and a grooved circular viscoelastic disk, and six parameters need to be determined: the disk diameter, disk thickness, number of grooves, grooving angle, rod length and rod diameter. The optimal values of these six parameters are determined by making the vibration-absorber system satisfy the following conditions at the same time: 1) the absorber system is in the resonant state; 2) the energy reflectivity of the energy guide rod should be small enough; 3) the energy dissipation rate of the grooved disk should be as large as possible in the frequency range of vibration absorption.

The grooved disk is composed of “n” numbers of parallel connected beams with varying cross-sectional area. Therefore, the energy dissipation rate of the disk is equal to the energy dissipation rate of a single non-uniform beam. Figure 1 is the schematic diagram of a single beam with varying cross-sectional area, and the thickness of the beam is “*h*”.

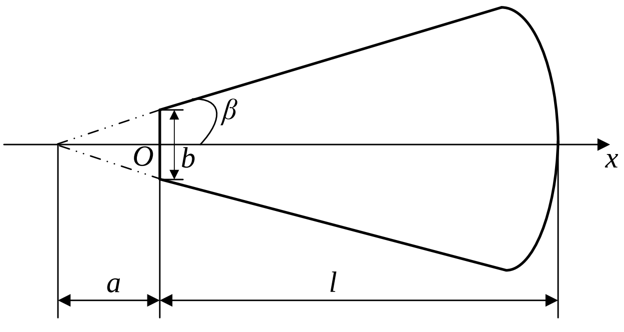


Figure 1: Schematic diagram of the beam with varying cross-sectional area.

Based on the Euler beam theory, the differential equation of the motion of the beam is:

$$A(x)G^*S''(x,t)+K_0G^*S'(x,t)-\rho A(x)\ddot{S}(x,t)=F(x,t), \tag{1}$$

where ρ is the density of beam material; $A(x)=h(2x\mathrm{tg}\beta+b)$; β is related to the number of grooves and groove angle; $S(x,t)$ is the displacement of the beam in the x direction; $S'(x,t)=\partial S/\partial x$, $S''(x,t)=\partial^2 S/\partial x^2$, and $\ddot{S}(x,t)=\partial^2 S/\partial t^2$ are the strain of the beam, the 1st-order differential of strain, and the acceleration, respectively; $F(x,t)$ is the inertial force exerted by the environment; $G^*=G_1+jG_2$, in which G^* is the complex shear modulus, G_1 is the storage modulus, and G_2 is the viscous modulus closely related to the energy dissipation of the non-uniform beam.

The fixed end of the beam (x=0) is subjected to a displacement disturbance: $S(0,t)=S_0 e^{jt}$, and assume that the displacement of the beam has the following form:

$$S(x,t) = ce^{j(\omega t - \lambda x)}, \tag{2}$$

where ω is the angular frequency. Inserting Eq. (2) into Eq. (1) and applying the boundary conditions $S(0,t)=S_0 e^{jt}$ and $S(\infty,t)=0$ yields the general solution:

$$S(x,t) = S_0 e^{-(\lambda_2 + j\lambda_1)x + j\omega t}, \tag{3}$$

where

$$\begin{cases} \lambda_1 = \{1/8[(4\rho\omega^2 J_1 - K^2) + \sqrt{(K^2 - 4\rho\omega^2 J_1)^2 + 16\rho^2\omega^4 J_2^2}\,]\}^{1/2} \\ \lambda_2 = 1/2K - \{1/8[(K^2 - 4\rho\omega^2 J_1) + \sqrt{(K^2 - 4\rho\omega^2 J_1)^2 + 16\rho^2\omega^4 J_2^2}\,]\}^{1/2} \end{cases}, \tag{4}$$

where J_1 and J_2 are the real part and the imaginary part of the reciprocal of the complex shear modulus respectively and $K = 2htg\beta / A(x)$.

According to the mechanics of vibration, the dissipated energy, stored potential energy and kinetic energy of a non-uniform beam in one cycle are derived respectively as:

$$W_D(l) = G_2 \pi S_0^2 \int_0^l A(x)(\lambda_1^2 + \lambda_2^2) e^{-2\lambda_2 x} dx, \tag{5}$$

$$W_s(l) = \omega / 4\pi \int_0^l EI(x) S''(x)^2 dx, \tag{6}$$

$$W_s(l) = \omega / 4\pi \int_0^l \rho A(x) \dot{S}(x)^2 dx. \tag{7}$$

Then, the energy dissipation rate (denoted by C) is the ratio of the energy dissipated in one cycle over the total energy:

$$C = \frac{W_D(l)}{W_D(l) + W_S(l) + W_E(l)}. \tag{8}$$

Equation (8) indicates that the energy dissipation rate is always less than 1. When designing a grooved waveguide absorber, the energy dissipation rate is required to be as large as possible.

Structural impedance can represent the transfer of energy between structures, so the energy reflectivity of the energy guide rod is analyzed by means of impedance. When one end of the rod is connected with the grooved disk, the disk is a load of the rod, and because the grooved disk is formed by "n" numbers of parallel connected beams with varying cross-sectional area, the impedance of the grooved disk is the sum of "n" numbers of z_l, the impedance of a single beam:

$$Z_L = n \int_0^l z_l(x) dx = R_L + jX_L, \tag{9}$$

where

$$z_l(x) = \frac{G}{j\omega} \frac{A(x) S'(x,t)}{S(x,t)}. \tag{10}$$

The energy reflectivity of the guide rod is the ratio of the energy reflected by the grooved disk over the energy transmitted from the rod to the disk, and it can be expressed in the form of impedance:

$$R = \frac{Z_L - Z_0}{Z_L + Z_0}. \tag{11}$$

The impedance of the guide rod $Z_0 = r^2\pi\sqrt{E_s\rho_s}$, where r, E_s, and ρ_s are the radius, Young's modulus, and density, respectively.

Equation (11) indicates that the energy reflectivity satisfies $0\leq|R|\leq 1$. In the design of vibration absorber, the size and the material of the grooved disk and guide rod are adjusted to make the energy reflectivity be less than 0.5, so as to ensure that most of the energy transmitted from the vibration source to the absorber will not return to the source.

3. Application of the design method - a case study

The proposed method is used to design and develop a vibration absorber for a beverage refrigerator, and the performance of the developed absorber is tested by experiment. The refrigerator has only one power component, a Matsushita ASF51X (50 Hz) reciprocating compressor with the maximum size of 142 mm×170 mm×162 mm. The finite-element simulation is used to obtain the natural vibration characteristics of the compressor and then the required frequency range of vibration reduction. Next, the proposed design method is used to design a suitable vibration absorber. Finally, the effect of the vibration absorber is verified by the result of finite-element simulation and validated by the experimental result.

3.1 Design process

Figure 2 shows the spectrum of the noise radiated by the refrigerator located in a semi-anechoic room. It can be seen that the noise energy is mainly concentrated in the middle-low frequency range below 1kHz, suggesting that the mechanical noise radiated from the compressor housing is dominant. The mechanical noise mainly comes from two sources: the vibration caused by the unbalanced force in the rotor is transmitted to the compressor housing via the base spring, and the friction between various connecting parts in the compressor also generates noise, which is also transmitted to the housing via the air inside.

Additionally, besides the mechanical noise, the electromagnetic noise and fluid pulsation noise in the compressor is ultimately radiated to the outer space through the compression casing. Therefore, suppressing the vibration of the compressor casing using a vibration absorber is expected to reduce the noise radiated by the refrigerator.

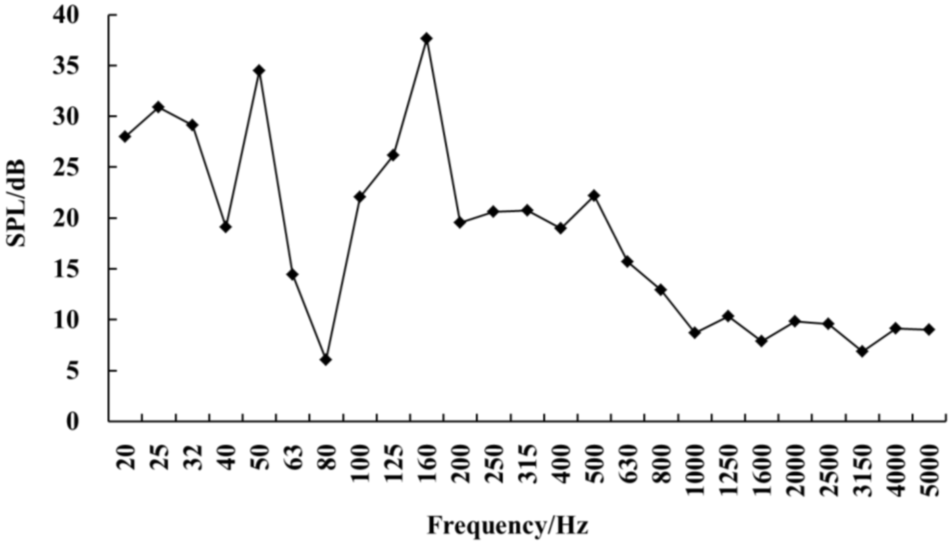
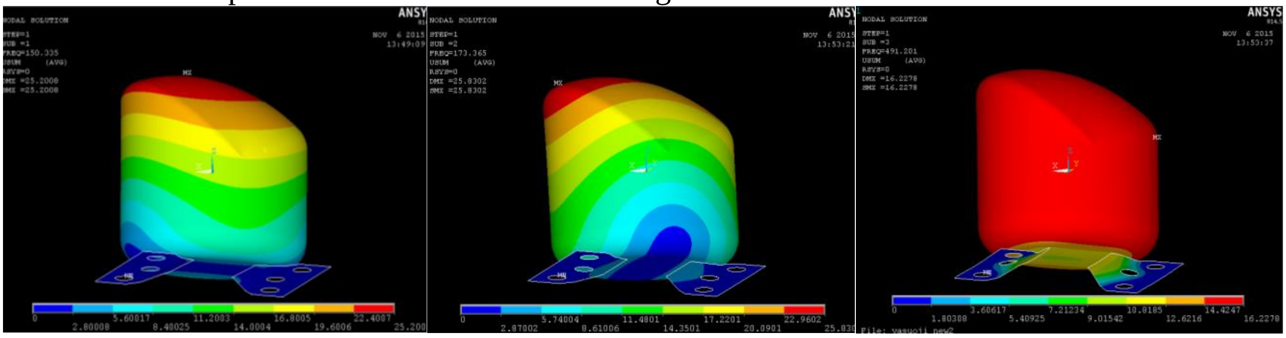


Figure 2: Spectrum in one-third octave bands of the noise radiated by the beverage refrigerator.

According to the proposed design method, the natural vibration characteristics of the compressor casing are studied first. Figure 3 shows the simulated vibration profiles (with ANSYS) of the first three modes of the casing at 150 Hz, 173 Hz, and 491 Hz. These frequencies are also located in the frequency range with maximum noise radiation shown in Fig. 2, indicating that the frequency range of vibration absorption should be located in the range of 125 Hz ~ 500 Hz.



(a) : the first eigen-mode (b) : the second eigen-mode (c) : the third eigen-mode

Figure 3: Eigen-mode profiles of the compressor.

According to Eqs. (8) and (11) in Section 2, two vibration absorbers with grooved circular disk are designed through optimization: absorber #1 is devoted for absorbing the vibration of compressor casing in the range of 125 Hz ~ 250 Hz and absorber #2 devoted for absorbing in 250 Hz ~ 500 Hz. The shape of the vibration absorber is shown in Fig. 4. The grooved disk is made of rubber, and the guide rod is made of aluminium. The difference between absorber #1 and #2 mainly lies in the disk diameter and rod length: the disk diameters of absorber #1 and #2 are 8 cm and 7 cm, respectively, and the rod lengths of absorber #1 and #2 are 3 cm and 5 cm, respectively.

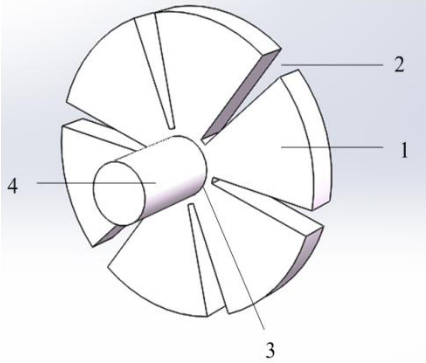
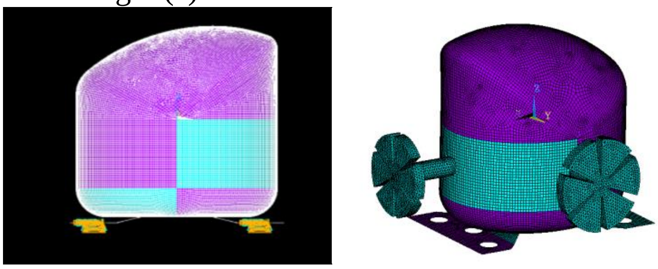


Figure 4: The vibration absorber with grooved circular disk (1. grooved circular disk, 2. groove, 3. hole, and 4. metal rod).

3.2 Finite-element simulation based verification

A finite element model of the compressor is established in ANSYS. It consists of the casing and the machine feet, but not including the motor rotor, crank, connecting rod, piston, and other internal mechanisms. The cell type is defined to be “Shell181”, and the thicknesses of the casing and foot are 0.54 cm and 0.38 cm, respectively. According to the actual working condition of the compressor, all degrees of freedom of the round holes in both machine feet are fixed. The model is divided into 34209 elements, as shown in Fig. 5(a).



(a): without absorber (b): with absorber

Figure 5: Finite-element grid model of the compressor

On the basis of the model in Fig. 5(a), models of absorber #1 and #2 are added to the surface of the compressor casing, as shown in Fig. 5(b). Both absorber models use the Solid182 type of element, and they are connected to the casing model through rigid region in ANSYS, to realize force and torque transfer.

A sinusoidal excitation force with amplitude of 10 N is applied to the casing with and without the installation of absorbers. The excitation frequency is from 0 Hz ~ 2 kHz and with an interval of 1 Hz. Figure 6 shows the computed acceleration levels at the point of vibration detection in the experiment. The curves indicate that after the installation of vibration absorbers, the vibration level of the casing in the frequency range below 1 kHz is significantly reduced, so the vibration absorber is initially verified to be effective.

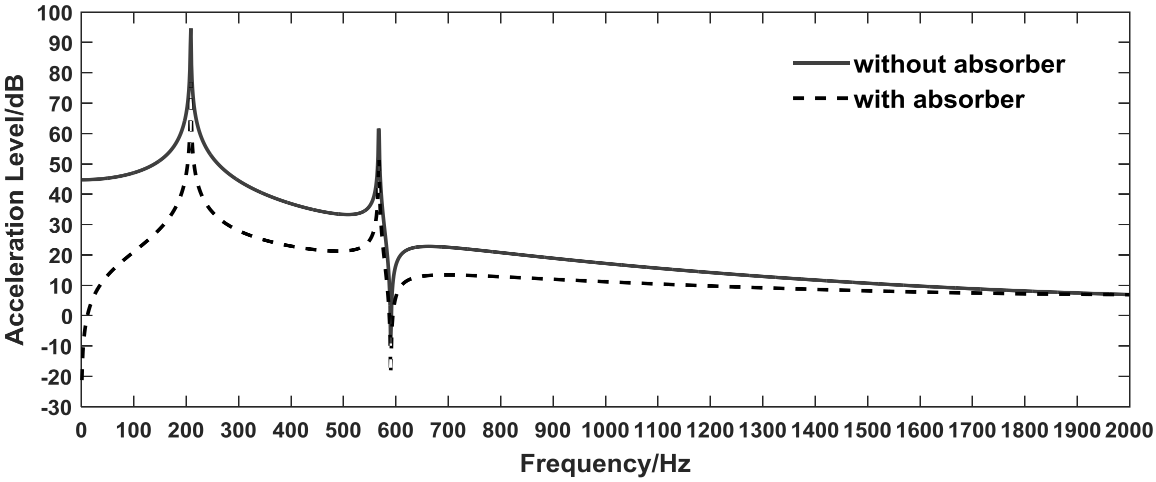


Figure 6: Simulated vibration level of the compressor casing before and after the vibration absorbers are installed.

3.3 Experimental validation

In a semi-anechoic chamber, the vibration of the compressor and the noise level radiated from the refrigerator are measured with and with the vibration absorbers. When installing the vibration absorbers, it is necessary to prevent any possible contact of the absorber with any adjacent objects because the compressor is installed in a narrow space in the refrigerator. In the experiment, an accelerometer is mounted on the top of the compressor casing; and according to the standard of “BS EN ISO 3744-2010” [13], in front of each, a BSW-MA201 microphone is placed 1 m away from each of the four side surfaces of the refrigerator and all the four microphone are at the height of 97 cm. The B&K3560C data analyzer is used for data collection and processing.

The vibration levels of the compressor before and after the installation of absorber are compared in Fig. 7, and the reference acceleration is 10^{-6} m/s^2. It can be seen that the vibration absorber can effectively absorb the vibration energy of the compressor casing. In the whole analysis frequency range (20 Hz ~ 5 kHz), the total vibration level is reduced by 1.5 dB; particularly, in the range of the eigen-frequencies of the first three modes (125 Hz ~ 500 Hz), the vibration level is reduced by 1.7 dB ~ 4.3 dB.

Figure 8 shows the noise level radiated by the refrigerator. It can be seen that after the installation of vibration absorber, the noise level is also been reduced: in the analyzed frequency range, the total A-weighted noise level is decreased by 2.0 dB(A); particularly, in the frequency range of 125 Hz ~ 1 kHz, the noise level is reduced by 1.1 dB ~ 4.0 dB, and the maximum noise reduction of 4.0 dB is reached at 500 Hz.

Both the results of vibration and noise radiation measurement indicate that the designed vibration absorber with grooved circular disk can significantly suppress the vibration of the compressor casing and reduce the noise radiated by the refrigerator. Hence the proposed method could be applied to design and develop vibration absorber for beverage refrigerator and other types of refrigerator for making quieter products.

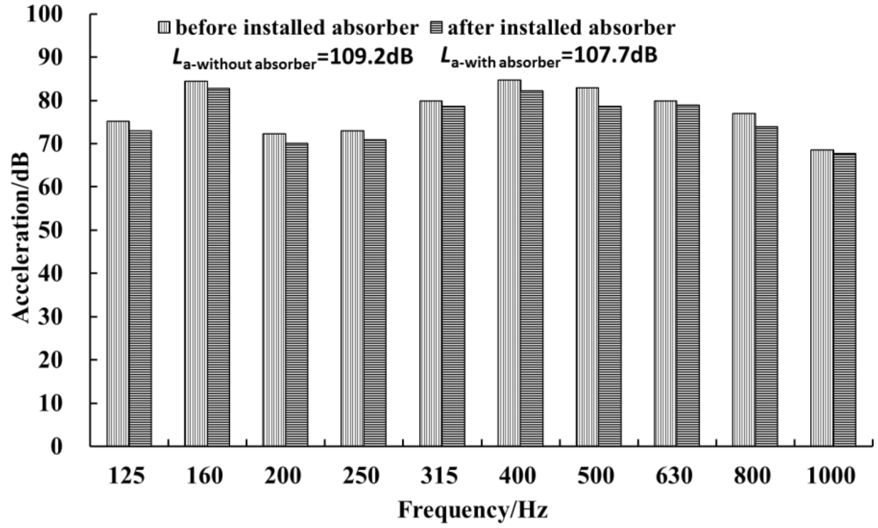


Figure 7: Measured vibration level of the compressor casing before and after the vibration absorbers are installed. ($L_{a\text{-without absorber}}$ and $L_{a\text{-with absorber}}$ denote the total acceleration level from 20Hz to 5kHz before and after installed the absorber respectively)

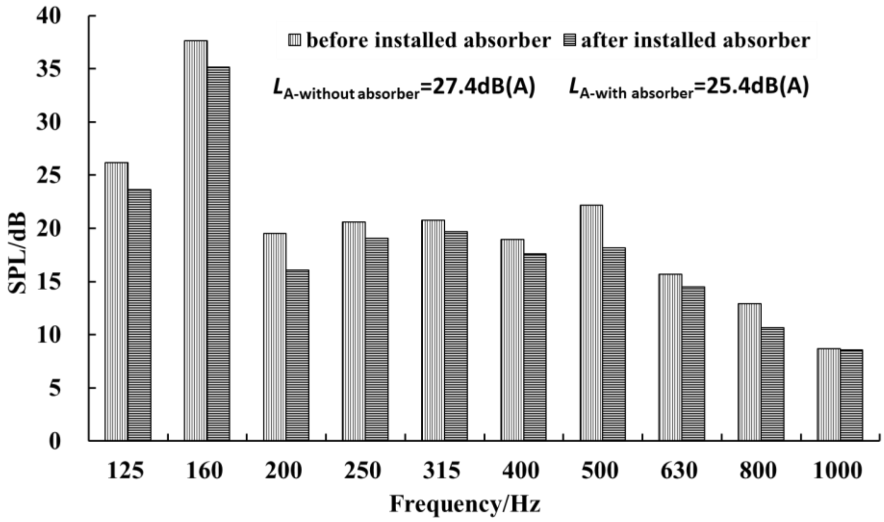


Figure 8: Measured vibration level of the compressor casing before and after the vibration absorbers are installed. ($L_{p\text{-without absorber}}$ and $L_{p\text{-with absorber}}$ denote the total sound pressure level from 20Hz to 5kHz before and after installed the absorber respectively)

4. Conclusions and acknowledgements

In this study, a design method is proposed to derive the optimized parameters of a new type of waveguide absorber with grooved circular disk for a vibrating body whose natural vibration characteristics are available. This method is then used to design and develop vibration absorbers for suppressing the vibration of a beverage refrigerator and reducing the noise radiation. Finite element simulation results of the compressor casing with and without installing the designed absorbers show that the vibration level of the casing in the frequency range below 1 kHz is significantly reduced. The results of vibration and noise measurement conducted in a semi-anechoic chamber also show that after the installation of vibration absorber, the vibration of the compressor is reduced by 4.3 dB in the frequency range of 125 Hz ~ 1 kHz; the total A-weighted noise level is decreased by 2.0 dB(A), and particularly, in the frequency range of 125 Hz ~ 1 kHz, the noise level is reduced by 1.1 dB ~ 4.0 dB, with the maximum noise reduction of 4.0 dB at 500 Hz. Both the results of vibration and noise radiation measurement indicate that the designed vibration absorber with grooved circular disk can significantly suppress the vibration of the compressor casing and reduce the noise radiated by the refrigerator.

This study is supported by the National Science Foundation of China (No. 51609202). I also would like to express my gratitude to all colleagues who helped me during this research.

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