

Acoustical contribution calculation and analysis of compressor shell based on acoustic transfer vector method

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Abstract. Nowadays, people are paying more and more attention to the noise reduction of household refrigerator compressor. This paper established a sound field bounded by compressor shell and ISO3744 standard field points. The Acoustic Transfer Vector (ATV) in the sound field radiated by a refrigerator compressor shell were calculated which fits the test result preferably. Then the compressor shell surface is divided into several parts. Based on Acoustic Transfer Vector approach, the sound pressure contribution to the field points and the sound power contribution to the sound field of each part were calculated. To obtain the noise radiation in the sound field, the sound pressure cloud charts were analyzed, and the contribution curves in different frequency of each part were acquired. Meanwhile, the sound power contribution of each part in different frequency was analyzed, to ensure those parts where contributes larger sound power. Through the analysis of acoustic contribution, those parts where radiate larger noise on the compressor shell were determined. This paper provides a credible and effective approach on the structure optimal design of refrigerator compressor shell, which is meaningful in the noise and vibration reduction.

1. Introduction

Refrigerator is one of the widely-used household appliances. The hermetic reciprocating compressor is the main noise source so that vibration and noise reduction of compressor is the key of refrigerator noise reduction.

As so far, many researchers have studied many different methods to reduce the refrigerator noise. Zhou and Jiang^[1] used fast multipole BEM to find the weak part of compressor shell where some optimal methods were used to reduce the radiation noise of compressor shell; Zhong and Jiang^[2] found two biggest noise source in compressor through the block test method and determined that the piston mechanism is main cause of low-frequency noise radiation; Li and Huang^[3] used theoretical method to establish the vibration velocity formulas on compressor shell which can be used to describe the relationship between vibration velocity on shell and the pressure on field point; Philippe Dugast^[4] introduced an interior source finite element method to simulate the compressor noise which has been used as an optimal method to reduce compressor noise about 2dB.

Compared with these methods used in compressor noise analysis mentioned above, the Acoustic Transfer Vector approach used in this paper has some advantages: firstly, it merely depends on the inherent attribute of compressor structural, the position of field points and the medium in the sound field while doesn't depend on the load fluctuation on compressor and the structural response of compressor; then, it just need be calculated once when obtaining the response of one certain compressor in different operating condition; therefore, it saves large time on calculation and increases the research efficiency.



In this paper, the compressor shell is divided into several parts, and the acoustic contribution of each part is obtained basing on Acoustic Transfer Vector approach. After analysing the acoustic contribution of each part, some parts on compressor shell where radiating biggest noise were determined. With the help of this method, we can design the compressor shell reasonably to reduce noise.

2. Acoustic theory basis

2.1 The Acoustic Transfer Vector approach

Acoustic Transfer Vector is the inherent attribute of acoustic system. If the surface shape of vibrating body, the acoustic medium and the field points are fixed, the ATV in this sound field is fixed. So for a sound field, the ATV is calculated one time that it can be used to predict sound pressure causing by vibrating surface on different loadings.

The surface of radiating body is discretized into several units which can be regarded as a vibration source. The vibration on these units cause the pressure fluctuation in medium then the sound pressure on one field point is linear superposed by sound pressure caused by each unit on this point. This relationship in one frequency can be described as formula (2-1) [5]:

$$p(\omega) = ATV(\omega) \cdot v_n(\omega) \quad (2-1)$$

In this formula: $p(\omega)$ is the sound pressure; $ATV(\omega)$ is the Acoustic Transfer Vector; $v_n(\omega)$ is the vibration velocity in normal direction of structure surface; ω is the angular frequency.

When using the boundary element to obtain discretization, the value of grid node can be obtained as the matrix following:

$$[A]\{P_b\} = [B]\{v_b\} \quad (2-2)$$

Corner mark b stands for the boundary.

The sound pressure on field point can be calculated by formula (2-3):

$$p = \{d\}^T \{P_b\} + \{m\}^T \{v_b\} \quad (2-3)$$

Combining formula (2-2) and formula (2-3) obtains this formula:

$$p = \{ATV\}^T \{v_b\} \quad (2-4)$$

Above, $\{ATV\}^T$ can be expressed as this formula:

$$\{ATV\}^T = \{d\}^T [A]^{-1} [B] + \{m\}^T \quad (2-5)$$

It is obvious that the ATV is the transfer function between the vibration velocity of vibrating surface in normal direction and the sound pressure on filed point. When the sound pressure is calculated in several positions, formula (2-4) can be replaced by formula (2-6)

$$p = \{ATM\}^T \{v_b\} \quad (2-6)$$

Above, $\{ATM\}^T$ is consist of different Acoustic Transfer Vectors.

2.2 Acoustic contribution

2.2.1 Sound pressure contribution

Basing on ATV method, the sound pressure contribution of each grid unit can be expressed as formula (2-9):

$$p = [ATV(\omega)]^T [v_e(\omega)] = \sum_{i=1}^N ATV_i(\omega) \cdot v_{e,i}(\omega) = \sum_{i=1}^N p_{e,i}(\omega) \quad (2-7)$$

In this formula: N is the total number of units; $ATV_i(\omega)$ is the element of matrix $[ATV_i(\omega)]$; $v_{e,i}(\omega)$ is the velocity in normal direction of unit i ; $p_{e,i}(\omega)$ is the sound pressure contribution, which can be described as this formula:

$$p_{e,i}(\omega) = ATV_i(\omega) \cdot v_{e,i}(\omega) \quad (2-8)$$

In order to analyse the sound pressure contribution of each unit, the value of $v_{e,i}(\omega)$ and $ATV_i(\omega)$ must be known advanced.

Panel is consist of a set of units, so that the sound pressure contribution of one panel is the summation of the sound pressure contribution of each unit on this panel, which can be described as this formula [6]:

$$p = \sum ATV_i(\omega) \cdot v_{e,i}(\omega) = \sum p_{e,i}(\omega) \quad (2-9)$$

The sound pressure on field point is the vector addition of the sound pressure caused by each panel, shown as this formula:

$$\vec{p} = \sum_{i=1}^n \vec{p}_i \quad (2-10)$$

Above: \vec{p}_i is the sound pressure contribution to the field point of panel i ; n is the total number of panel.

2.2.2 Acoustical power contribution

Panel sound pressure contribution can only describe the contribution to one field point, so in order to describe the noise radiation of one panel contribute to whole sound field, the acoustical power contribution is introduced as following:

$$W_c(\omega) = \frac{S}{\rho_0 c} \left(\sum_{i=1}^{n_p} \{ATV_i(\omega)\}^T \{v_i^n(\omega)\} \right)^2 \quad (2-11)$$

$$W = W_{c1} + W_{c2} + \dots + W_{cn} \quad , \quad n = 1, 2, 3 \dots \quad (2-12)$$

Above: W_{ci} is the acoustical power contribution of one panel; $W_c(\omega)$ is the panel acoustical power contribution in frequency ω ; W is the total acoustical power in the sound field; n is the total number of panel.

With the help of panel acoustical power contribution, the larger panel acoustical power contribution will be found quickly and some optimal methods will be used in these panels.

3. Sound field establishment and compressor shell division

In noise experiment, the test points are arranged in the place where is 300mm distance to each surface of compressor. In order to verify the accuracy of calculation, five field points on the same position to text experiment are established in LMS Virtual.Lab software (the distance to ground is 150mm and the distance to compressor centre is 300mm of S1、S2、S3、S4; S5 is right above compressor and its distance to ground is 300mm), which shown as figure 3-1.

For explaining ATV calculation of sound field further more, ISO3744 standard field points was established in this paper. This paper used LMS Virtual.Lab software to establish ISO3744 standard field points on Fine Sphere Mesh type, which refines the standard field points. Under the situation of symmetry panel, there is a hemisphere surface field points grid consist of 19 nodes and 18 units around compressor shell when establishing the standard field points. The standard field points model is shown as figure 3-2.

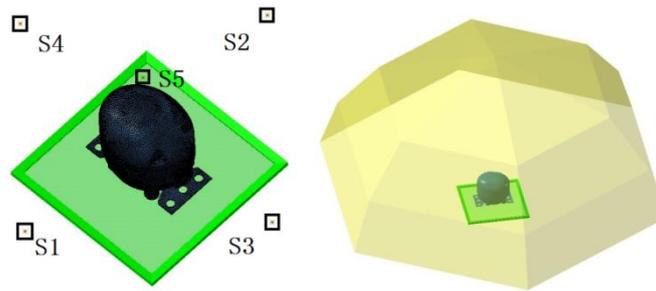


Figure 3-1.Field points arrangement **Figure 3-2.** ISO 3744 standard field points

After establishing the sound field, the compressor shell is divided into several parts which can be regarded as vibrating panel according to different structure in compressor. So the compressor shell is divided into upper shell, below shell and engine base. The upper shell is divided into four symmetrical parts; the below shell is divided into five parts where four parts above engine base are symmetrical parts same to upper shell and below the engine base is one part; the engine base is divided into two parts: left engine base and right engine base. Above all, there are eleven parts on the compressor shell surface shown as figure 3-3.

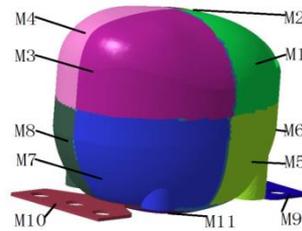


Figure 3-3.The division of compressor shell surface

4. The correctness verification of ATV calculation

After ATV calculation by LMS Virtual.Lab software, these points: ISO:1、ISO:6、ISO:14 and ISO:19 were selected to analyze. The following figures show the ATV cloud picture of these four points in 630Hz:

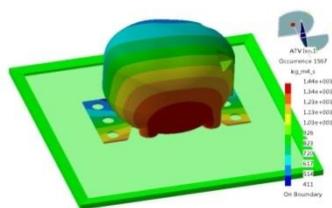


Figure 4-1. ATV cloud picture of point ISO:1 in 630Hz

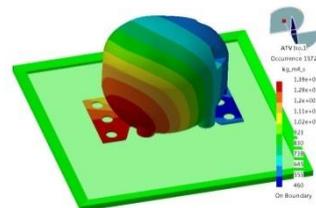


Figure 4-2. ATV cloud picture of point ISO:6 in 630Hz

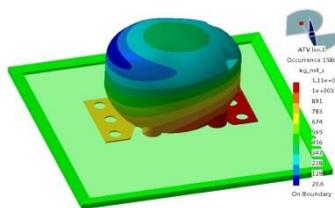


Figure 4-3. ATV cloud picture of point ISO:14 in 630Hz

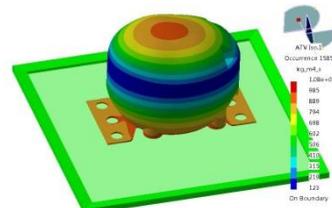


Figure 4-4. ATV cloud picture of point ISO:19 in 630Hz

Figure 4-1 to Figure 4-4 show the sound transfer of compressor shell to sound filed points as well as the sound contribution of compressor shell to sound filed points. Point ISO:1 is head against nameplate and close to the bottom where the nameplate on compressor shell contributes larger sound transfer shown as Figure 4-1; point ISO:6 is close to the opposite of junction box where the compressor engine base and spring pedestal make lager sound contribution shown as Figure 4-2; point ISO:14 is close to compressor junction box where the compressor engine base and junction box make lager sound contribution shown as Figure 4-3; point ISO:19 is right above compressor where the top surface of compressor shell and the ground reflection make larger sound contribution to shown as Figure 4-4.

In order to verify the correctness of ATV calculation on compressor shell, 5 field points same to those points in noise test experiment were defined in LMS Virtual.Lab software when the standard sound field was established. Figure 4-5 to Figure 4-9 show the 1/3 octave sound pressure level comparison of simulation value to experiment value of these 5 field points.

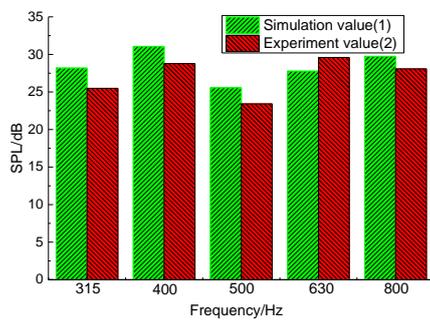


Figure 4-5.The 1/3 octave sound pressure level comparison of point S1

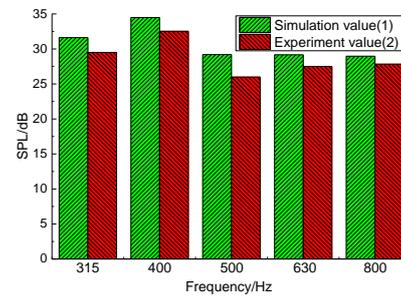


Figure 4-6.The 1/3 octave sound pressure level comparison of point S2

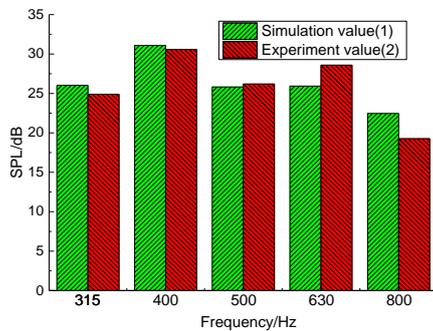


Figure 4-7.The 1/3 octave sound pressure level comparison of point S3

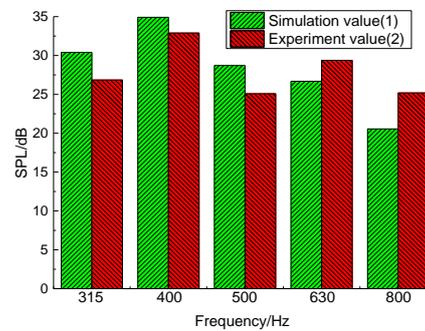


Figure 4-8.The 1/3 octave sound pressure level comparison of point S4

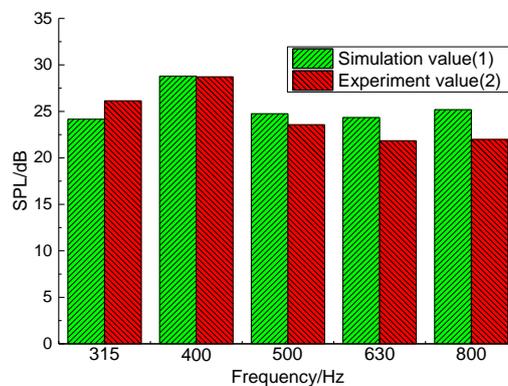


Figure 4-9. The 1/3 octave sound pressure level comparison of point S5

Shown as figures above, points S1、S2、S3 and S4 have larger 1/3 octave sound pressure level in both 400 Hz and 630 Hz, and point S5 is right above compressor which has larger 1/3 octave sound pressure level in 400 Hz. Comparing the simulation value to the experiment value of these points, their errors are almost within 12%, but there are still a few errors are beyond 12%. These errors are produced by the measurement errors in experiment, calculation errors in data processing and the error caused by different situation set in simulation and experiment while the reflection of ground is total reflection in simulation but the reflection of ground is not total reflection in experiment which makes the values in simulation are larger than experiment. In spite of the errors exist in simulation, we can also easily obtain the sound pressure variation tendency in different frequency and find the larger sound pressure in some frequencies. So the ATV method can be used in prediction of compressor shell radiation noise.

5. Acoustic contribution analysis

5.1 Panel sound pressure contribution analysis in ISO 3744 standard sound field

The distance of all points in ISO 3744 standard sound field is 1m to the shell which is beneficial to determine the larger noise resource on the shell. Because of the symmetry of compressor shell, the sound pressure cloud pictures of panel 1、4、6、7、9、11 in 400Hz were selected to analyse, shown as following figures:

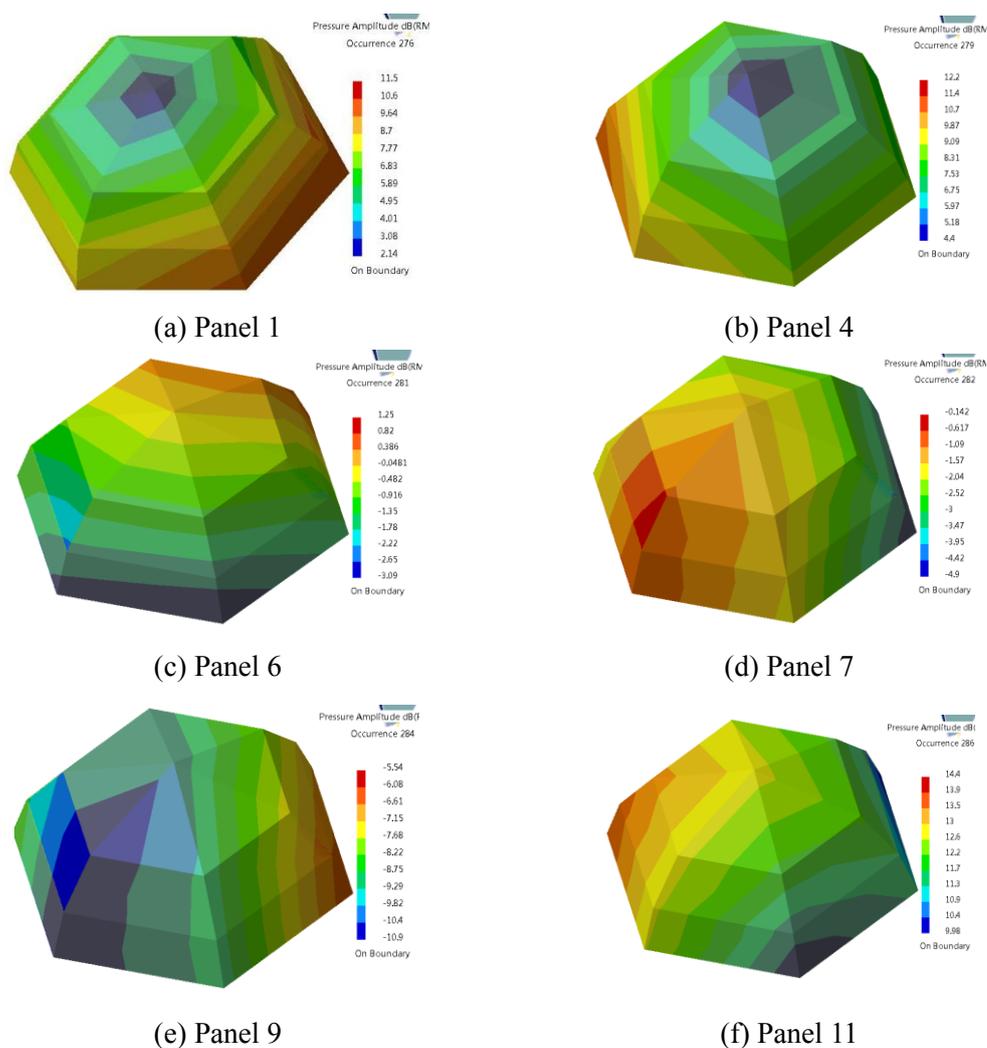
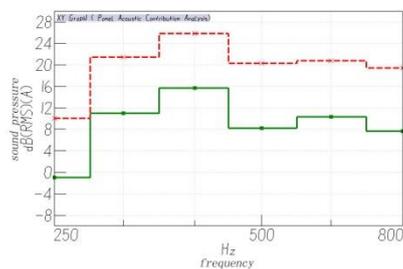


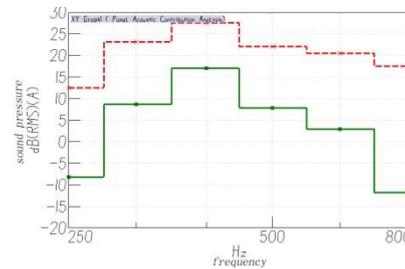
Figure 5-1. Panel sound pressure contribution cloud pictures in 400Hz

The sound pressure contribution made by different panel areas can be straight seen in the sound pressure contribution cloud pictures: the radiated noise in 400Hz of Panel 1 and Panel 4 make larger influence to nearby areas shown as Figure (a) and Figure (b); the contribution of Panel 6, Panel 7 and Panel 9 made to the sound field is negative shown as Figure (c), Figure (d) and Figure (e) which means that these panels weaken the sound level in the sound field; Panel 11 is the bottom of compressor shell and it radiates larger noise level to the sound field shown as Figure (f) which is relative to the ground reflection. From sound pressure contribution cloud pictures, we can tentatively determine that the suction panel, discharge panel and the bottom panel on the shell make larger sound pressure contribution to sound field in 400Hz.

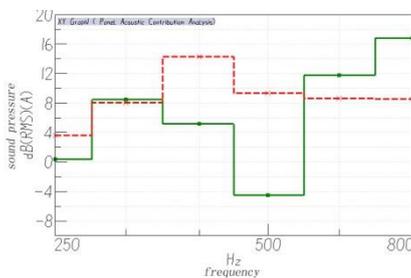
Researchers may easily acquire the sound radiation of each panel and find the larger contribution among these panels from sound pressure contribution cloud pictures. However, the frequency in which the radiated noise is larger may be focused on in compressor noise control. Shown as Figure 5-2, the contribution curves in different frequency of each panel are listed where the red imaginary line stands for the contribution of whole panels made to the field point and the green full line stands for the contribution of one panel made to the field point. In Figure 5-2, the marks in frequency axis are the value of target frequency we want to study which are 250 Hz, 315 Hz, 400 Hz, 500 Hz, 630 Hz and 800 Hz. The frequency around the target frequency hasn't been studied in this paper hence we think the sound pressure value in these frequencies is equal to the target frequency. Then the frequency axis is divided into equal parts where those target frequency are the midpoint in these parts. So the Picture (a) to Picture (f) in Figure 5-2 is shown as step plots that researchers can find the larger contribution made by each panel in the target frequency.



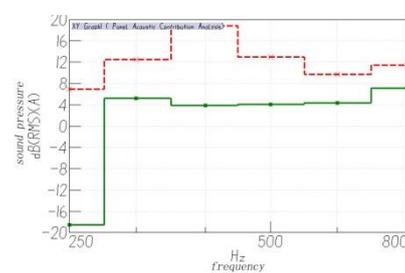
(a) The sound pressure contribution curve of Panel 1 made to point ISO:2



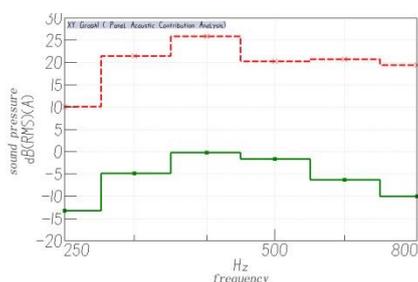
(b) The sound pressure contribution curve of Panel 4 made to point ISO:5



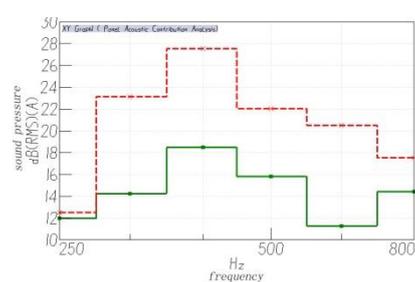
(c) The sound pressure contribution curve of Panel 6 made to point ISO:3



(d) The sound pressure contribution curve of Panel 7 made to point ISO:6



(e) The sound pressure contribution curve of Panel 9 made to point ISO:2



(f) The sound pressure contribution curve of Panel 11 made to point ISO:5

Figure 5-2. The sound pressure contribution curves of Panels made to standard points

Compared to cloud pictures, the sound pressure contribution curves will help researchers understand the frequency characteristic of sound pressure more clearly. The 6 figures in Figure 5-2 show that all these 6 panels radiate larger noise in 400 Hz which is related to the structure of compressor shell. The main radiated frequency of Panel 1 is in 400 Hz and 630 Hz shown as Figure (a); the main radiated frequency of Panel 4 is in 400 Hz shown as Figure (b); the main radiated frequency of Panel 6 is in 630 Hz and 800 Hz shown as Figure (b); Panel 7 and Panel 9 make little influence to the field point and even weaken the sound pressure on the field point shown as Figure (d) and Figure (e); the main radiated frequency of Panel 11 is in 400 Hz shown as Figure (f).

5.2 Panel acoustical power contribution analysis in ISO 3744 standard sound field

Mentioned above, panel sound pressure contribution can only account for the contribution made to one field point. But in compressor noise control, the contribution made by compressor shell to whole sound field need to be studied, so that the panel acoustical power contribution was selected to be analysed which can reflect the radiation characteristic of compressor shell as a whole.

In LMS Virtual.Lab software, there are several modes to analyse the acoustical power contribution, including: the acoustical power contribution curves, the contribution histogram of each panel in one frequency and so on. There are different advantages of different modes: the peak frequency may be seen clearly in curve graph which help researchers find the larger contribution in some frequency of different panels; the contribution level of different panels in one frequency may be seen clearly in the contribution histogram. The acoustical power contribution curves made by each panel to the sound field are listed in Figure 5-3.

In Figure 5-3, the curve SUM stands for the acoustical power contribution made by whole compressor shell to the sound field and it has largest contribution in 400 Hz which is fit with the peak noise tested in noise experiment preferably. As a whole, each panel has larger contribution in 400 Hz. However, the acoustical power contribution of Panel 4, Panel 8 and Panel 11 are larger than other panels obviously in 400 Hz because of suction panel, discharge panel and bottom panel on compressor. The larger acoustical power contribution of Panel 5, Panel 8 and Panel 11 are in 630 Hz which is due to the motor junction box, compressor suction and discharge line and bottom panel of compressor. Aiming at these two frequencies which have larger acoustical power contribution, the contribution histograms in these two frequencies are acquired in LMS Virtual.Lab which describe the contribution level of different panels in these two frequencies shown as Figure 5-4 and Figure 5-5.

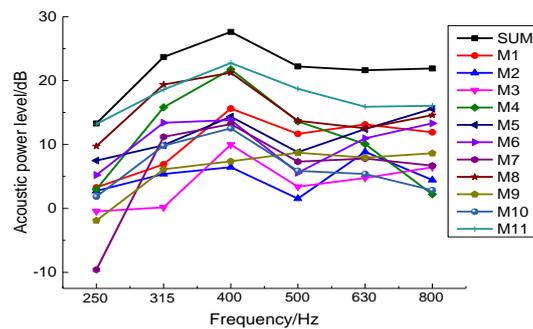


Figure 5-3. The 1/3 octave acoustical power curves of each panel

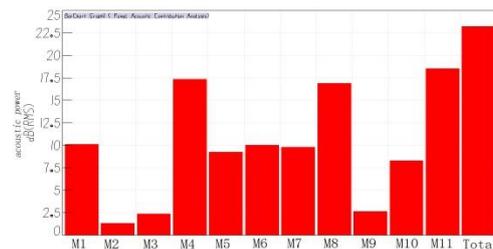


Figure 5-4. Panel acoustical power contribution in 400Hz

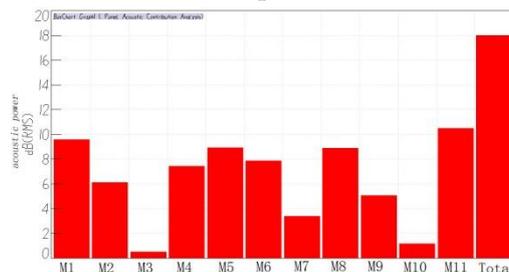


Figure 5-5. Panel acoustical power contribution in 630Hz

The result of contribution histograms analysis is fit with the analysis conclusion that Panel 4, Panel 8 and Panel 11 make larger acoustical power contribution in 400Hz and Panel 1, Panel 4, Panel 5, Panel 8 and Panel 11 make larger acoustical power contribution in 630Hz. Above all, these panel areas may be emphatic focused on in freezer compressor noise control, meanwhile, the analysis of panel acoustic contribution provide a direction to determine the noise source inside the compressor.

6. Conclusion

The ISO 3744 standard sound field surrounding compressor shell surface was established in this paper and the ATV on the sound field was calculated. The ATV method was used to predict the sound pressure on field points which was compared with the sound pressure on the same field points tested in noise experiment. After ATV calculation, the area on compressor shell was divided into several panels on which the panel sound pressure contribution and acoustical power contribution was analysed. Some conclusions were acquired following:

- (1) The acoustical contribution of compressor shell was predicted through ATV calculation. The sound pressure predicted was compared with the sound pressure tested in experiment and the result fit preferably. The result of comparison indicates that ATV method can be used in prediction of compressor shell radiation noise.
- (2) The panel acoustical contribution of different parts on compressor shell was analyzed which indicates that some panels can enhance the sound pressure on field points and others weaken the sound pressure. The acoustical contribution spectrogram of each panel was acquired from which some conclusions were obtained: the discharge and suction line of compressor and the

bottom panel of compressor shell made larger noise contribution to the sound field in 400Hz; the discharge and suction line of compressor, the panel closed to motor junction box and the bottom panel of compressor shell made larger noise contribution to the sound field in 630Hz; the engine base make negative contribution to the sound field both in 400Hz and 630Hz. The contribution characteristic of compressor shell made to sound field was acquired through panel acoustical power contribution analysis which confirms the conclusions above.

- (3) The ATV method can predict the radiation characteristic of compressor shell accurately. The analysis of panel sound pressure contribution and panel acoustical contribution based on ATV method may help researchers acquire the acoustical contribution made by compressor shell to whole sound field and the areas where made the larger contribution. A new method was presented in this paper to confirm the noise source inside the compressor which can enhance the efficiency of noise research on freezer compressor as well.

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