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ABSTRACT

Since the accumulator is one of the most important component with the largest sound radiation surface area in rotary compressor,its noise contribution may be substantial.Noise generation and transfer mechanism of the accumulator are so complicated that it is difficult to identify the acoustic characteristics,because both structural and cavity modal are possible to be excited by many sources such as structural vibration,aero-acoustics,pressure pulsation etc.,in addition coupling between them cannot be ignored either.In this paper,both of the noise generation and transfer mechanism are studied systematically,also standing wave and mechanical vibration theory are applied to build the mathematical model. To clarify the natural characteristics of the accumulator,both structural modal experiment and cavity modal simulation approach are applied by using commercial software Test.Lab and SYSNOISE respectively.Finally the structural modification experiment are carried out and proved that the model and simulation in this paper are reliable enough,it shows that both overall noise level and sound quality are improved,especially noise level was reduced about 10 dB(A) at few special frequency band.

1. INTRODUCTION

Usually accumulator of rotary compressor is connected with main case by a holder and suction pipe B,as is shown in Figure 1,both vibro- and aero- acoustics can be excited by many sources such as main case vibration,pressure pulsation,gas flow and evaporation of liquid refrigerant,finally it is radiated out via the vibration of the accumulator shell no matter whatever the noise source is.Many studies were done on accumulator noise by foreign experts,Kim *et al.*(1998) used structural modal analysis approach to eliminate the abnormal noise peak near 2kHz[1],Zhou *et al.*(1998) simulated the noise level radiated from accumulator shell due to internal pressure pulsation by FEM and BEM numerical approach[2], Wang *et al.*(2002) modified the accumulator rigid modes to improve the compressor noise level by topology optimization techniques[3].While up to now we still haven't seen any paper about the accumulator noise due to coupling of structural and cavity modes yet,its mechanism and acoustic characteristics was discussed systematically in this paper combining theory,modal experiment and simulation approach, afterwards it's validated by noise experiments.

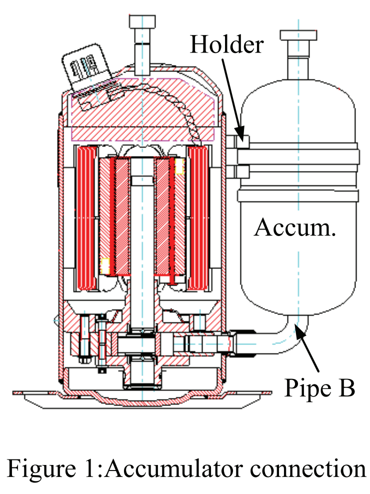

Figure 1:Accumulator connection

2. MATHEMATIC MODEL

While building the mathematic model of accumulator, we simplified it is a rigid long pipe with two closed end as is shown in Figure 2,where pipe length is l,left end $x=0$ and right end $x=l$,also we assume that dimension of pipe diameter φd is much more smaller than sound wavelength λ what we are concerned about,so only plane waves parallel to the pipe axis can be propagated in the pipe.If incident sound wave $p_i(x,t)$ which transmits from left to right equal to $p_0\sin(2\pi ft-k(l-x))$,where p_0 is sound pressure amplitude, f is sound pressure frequency, k is wavenumber,and $k=2\pi/\lambda$,corresponding particle velocity of incident wave $u_i= p_i(x,t)/\rho_0 c$,where ρ_0 is medium density,C is sound velocity. According to transfer characteristics of sound wave,incidence wave will cause a reflection wave $p_r(x,t) =p_0\sin(2\pi ft+k(l-x))$ which transmits from right to left with the same amplitude after it reaches the right rigid end,corresponding particle velocity of reflection wave $u_r=-p_r(x,t)/\rho_0 c$,according to the theory of sound wave superposition,the sound pressure $p(x,t)$ anywhere in the pipe is given as below,

$$\begin{aligned} p(x,t) &= p_i(x,t)+p_r(x,t) = p_0\sin(2\pi ft-k(l-x))+p_0\sin(2\pi ft+k(l-x)) \\ &= 2p_0\cos k(l-x)\sin 2\pi ft \end{aligned} \tag{1}$$

Equation(1) shows that its amplitude $2p_0\cos k(l-x)$ is a fuction of single variable x,independent of variable t,it means that a “standing wave” is formed in the pipe due to the superposition of incident and reflection wave.When $\cos k(l-x)=0$, amplitude is always equal to *zero*,normally we call this position wave notch,when $\cos k(l-x)=1$, amplitude is always equal to $2p_0$, we call this position wave crest，obviously sound wave “stay” in the pipe into a stable state. In the same way particle velocity $u(x,t)$ can be given as below,

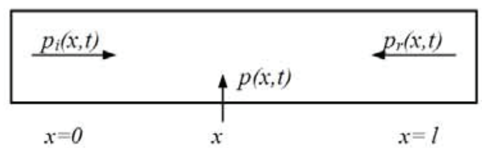

Figure 2: Accumulator mathematic model

$$\begin{aligned} u(x,t) &= u_i(x,t)+u_r(x,t) = \frac{p_i(x,t)-p_r(x,t)}{\rho_0 c} \\ &= \frac{p_0\sin(2\pi ft-k(l-x))-p_0\sin(2\pi ft+k(l-x))}{\rho_0 c} \\ &= \frac{2p_0\sin k(l-x)\cos 2\pi ft}{\rho_0 c} \end{aligned} \tag{2}$$

There is no doubt particle velocity $u(x,t)\equiv 0$ while $x=0$ and $x=l$ due to the assumption of rigid boundary condition in both end,if we want to let $\sin kl\equiv 0$,k must be some specific value given as below,

$$k=\frac{n\pi}{l} \quad or \quad \lambda=\frac{2l}{n} \qquad n=1,2,3\ldots \tag{3}$$

Furthermore we can get the standing wave frequency $f_n=c/\lambda=nc/2l$,that is also the modal Frequency which modal shape is axial standing wave.Discussion above is based on the assumption of dimension of pipe diameter φd is much more smaller than sound wavelength λ and only plane waves parallel to the pipe axis can be propagated in the pipe,actually there are high-order cavity modes exist due to dimension of pipe diameter is not always small enough.At the same time,accumulator also has a series of part and rigid structural modes because it is a thin-walled structure attached to main case,noise peak is possible to occure because of resonance if cavity modes couple with some structural modes and its frequency are closed to the vibration or noise frequency of the main case.

3. SOUND SOURCE IDENTIFICATION

We choose the simplest separation method to identify the noise contribution of the accumulator,pipe B is extended and accumulator is placed outside the semianechoic room,basically we can achieve the purpose of removal of accumulator noise by this way.The compressor sound power spectrum before and after the separation of the accumulator are compared as is shown in Figure 3,from the results it is found that the noise contribution frequency of the accumulator in this case mainly focus on band 400,500Hz and 1250Hz,especially at band 400,500Hz more than 10dB(A) exceeded,so the studies below we focus on cavity and structural modes which frequency is close to band 400,500Hz and 1250Hz.

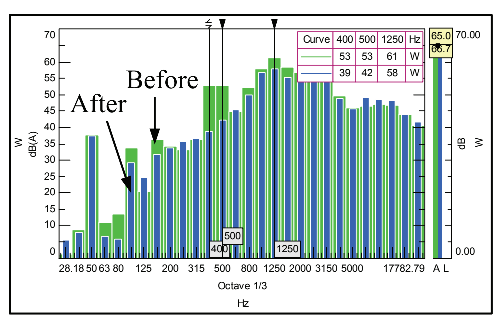

Figure 3: Noise spectrum comparison before and after separation of the accumulator

4. CAVITY MODAL ANALYSIS

We simulate accumulator cavity modes using acoustic FEM approach by commercial software SYSNOISE, parameters density ρ and sound velocity C are chosed according actual situation, both ends of the accumulator are set to rigid boundary condition, the first 10 simulated modal frequency are shown in table 1, it shows that the 2nd, 7th

Table 1: Simulated cavity modal frequency

Modal number	1st	**2nd**	3rd	4th	5th	6th	**7th**	**8th**	9th	10th
Modal frequency(Hz)	188	**427**	566	824	940	1147	**1259**	**1260**	1376	1376

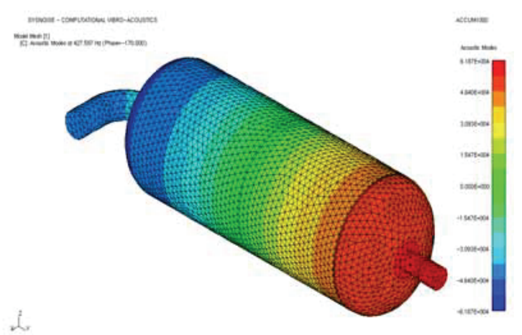

Figure 4: The 2nd cavity modal shape

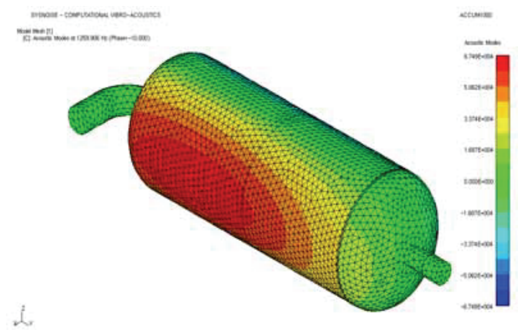

Figure 5: The 7th cavity modal shape

and 8th modal frequency are very closed to band 400,500Hz and 1250Hz respectively which we are concerned about.Their modal shapes are shown in Figure 4,5,6 respectively, obviously these three modal shapes are all main body vibration of the accumulator cavity,and the 2nd is low-order axial mode,the 7th and 8th are high-order radial modes,the result matches the former theory analysis,also the 7th and 8th modal shapes are similar and frequency are nearly equal,only the vibration direction space quadrature ,it is typical double root.

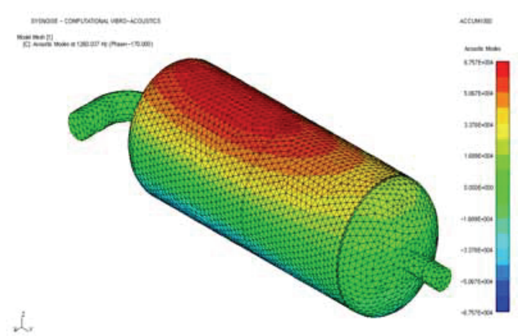

Figure 6: The 8th cavity modal shape

5. STRUCTURAL MODAL ANALYSIS

Structural modes of accumulator include part mode which means sub-structure mode of the detached accumulator and rigid mode which means whole body mode of the attached accumulator as is discussed before,we identify both part and rigid modal parameters by commercial software Test.Lab in the studies below,

5.1 Part Modal Analysis

By modal impact SIMO approach in Free-Free suspension,we get the first 7 experimental modal frequency as is shown in table 2,from Table 2 we can find that the part modal frequency are much more higher than the band 400,500Hz and 1250Hz what we are concerned about due to its high stiffness,basically we can judge its contribution to noise is not high enough,the first two modal shapes are shown in Figure 7,8 for reference.

Table 2: Experimental part modal frequency

Modal number	1^{st}	2^{nd}	3^{rd}	4^{th}	5^{th}	6^{th}	7^{th}
Modal frequency(Hz)	2242	2244	2280	2332	2558	2982	3010

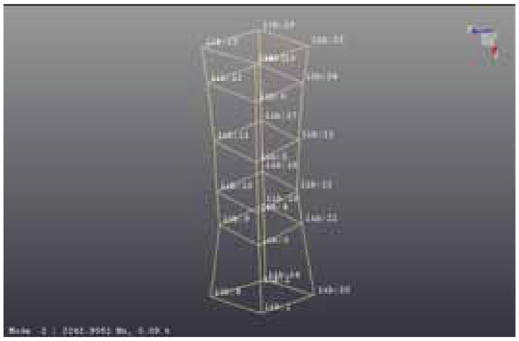

Figure 7: The 1st part structural modal shape

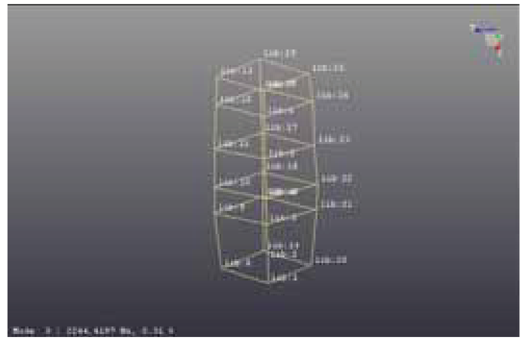

Figure 8: The 2nd part structural modal shape

5.2 Rigid Modal Analysis

Also by modal impact SIMO approach in Free-Free suspension while attached to main case,we get the first 5 experimental modal frequency as is shown in table 3,from Table 3 we can find that the 3^{rd} and 4^{th} modal frequency are very closed to the band 400,500Hz and 1250Hz respectively what we are concerned about, obviously these two modal shapes are all accumulator rigid structural modes as are shown in Figure 9,10 respectively, and the 3rd is vertical mode,the 4^{th} is horizontal mode. Comparing with 2^{nd}, 7^{th} cavity modes simulated before, we found that their

Table 3: Experimental rigid modal frequency

Modal number	1^{st}	2^{nd}	$\mathbf{3^{rd}}$	$\mathbf{4^{th}}$	5^{th}
Modal frequency(Hz)	98	214	**429**	**1326**	2005

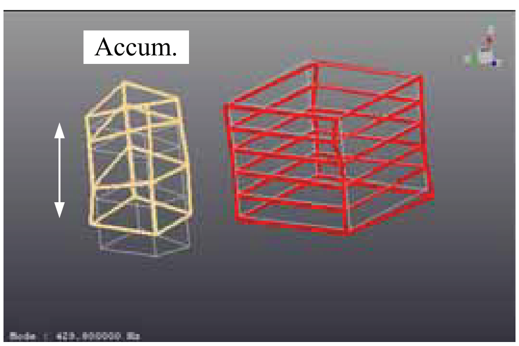

Figure 9: The 3^{rd} rigid structural modal shape

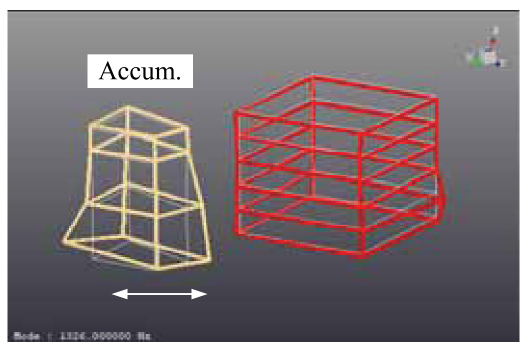

Figure 10: The 4^{th} rigid structural modal shape

modal shapes are similar and frequency are adjacent respectively, combining the sound source identification results discussed before basically we can judge its contribution to noise is high possible due to coupling of rigid structural and cavity modes.

6. NOISE EXPERIMENTAL VALIDATION

According to the analysis and judgement before, mainly noise contribution of the accumulator on band 400,500Hz and 1250Hz in this case is because of the coupling of rigid structural and cavity modes,in order to validate it by experiment,we modify the cavity modes and structural rigid modes respectively to decouple it by structural modification,than comparing the noise spectrum before and after the modification.

6.1 Cavity Modal Modification

According the former analysis that the 2nd cavity mode is low-order axial mode, so we design a ring thin plate in the lower part of the accumulator as is shown in Figue 11 in order to modify this order modal frequency and try not to influence the rigid structural modes.Simulated modal frequency are shown in table 4 by the same parameters and boundary conditions,it shows that the 2nd mode frequency moved from 427 to 321Hz while the 7th and 8th mode frequency change few,also the 2nd mode is still axial mode as is shown in Figure 12.Noise spectrum comparison before and after the modification is shown in Figure 13, from it we

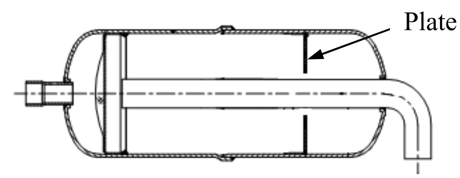

Figure 11: Accumulator with plate

Table 4: Simulated cavity modal frequency with plate

Modal number	1st	**2nd**	3rd	4th	5th	6th	**7th**	**8th**	9th	10th
Modal frequency(Hz)	187	**321**	564	823	940	981	**1256**	**1257**	1303	1303

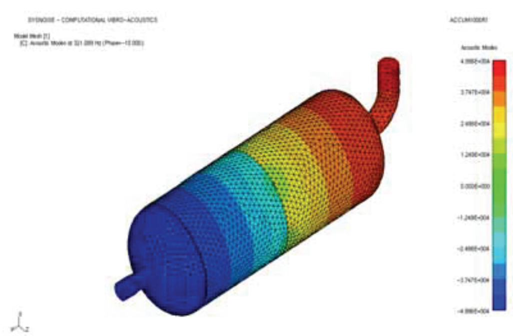

Figure 12: The 2nd cavity modal shape with plate

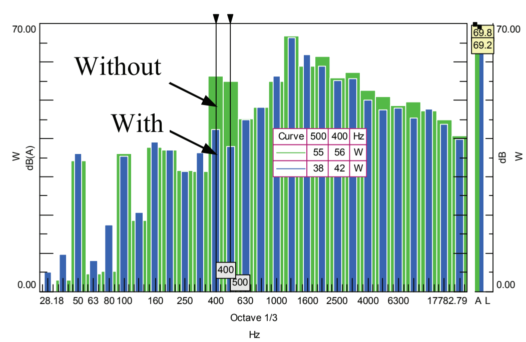

Figure 13: Noise spectrum comparison with and without plate

can find that noise reduction is 14,17dB(A) on band 400,500Hz respectively,few change on band 1250Hz,it matches the analysis discussed before,also overall level slightly decrease about 0.6 dB(A).

6.2 Rigid Structural Modal Modification

According the former analysis that the 3rd and 4th rigid structural modes coupled with the cavity mode,connection stiffness with the main case need to be changed if we want to move the rigid mode frequency,we modified its connection by shorten the suction pipe B,cancelled the holder and welded it directly with the main case as is shown in Figue 14. By the same testing conditions of rigid modes like before,we get the first 5 experimental modal frequency after welding as is shown in table 5,from the result we can find that the 3rd modal frequency moved sharply from 429 to 920Hz,the 4th modal frequency moved slightly from 1326 to 1412Hz.Noise spectrum comparison before and after the modification is shown in Figure 15,it shows that noise reduction is 13,14,8dB(A) on band 400,500,1250Hz respectively,that matches the analysis discussed before,also overall level slightly decrease about 1.1 dB(A).

Table 5: Experimental rigid modal frequency after connection modification

Modal number	1st	2nd	**3rd**	**4th**	5th
Modal frequency(Hz)	316	343	**920**	**1412**	2125

Figure 14: Accumulator connection modification

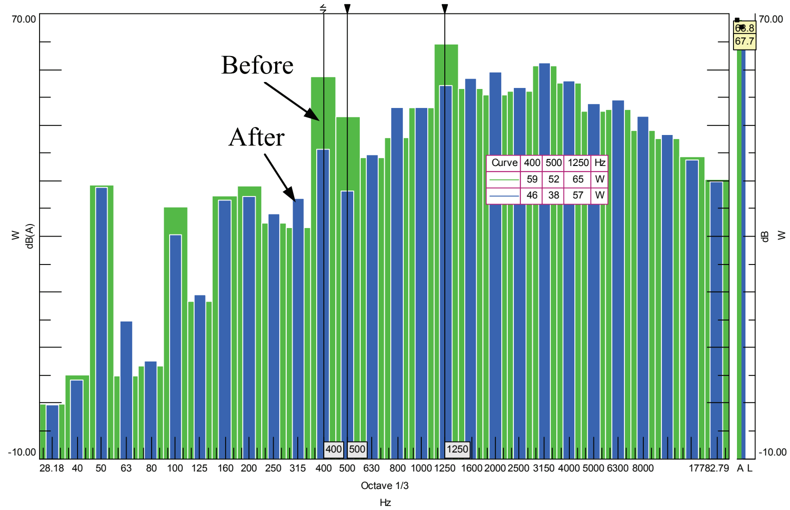

Figure 15: Noise spectrum comparison before and after connection modification

7. CONCLUSIONS

Accumulator has a series of part and rigid structural modes because it is a thin-walled structure attached to main case,noise peak is possible to happen because of resonance if cavity modes couple with some structural modes and its frequency are closed to the vibration or noise frequency of the main case,it is possible to decrease its noise contribution and eliminate the noise peak by modification of the cavity modes or structural rigid modes to decouple it.In this paper,noise mechanism and acoustic characteristics of accumulator due to coupling of cavity and structural modes is studied systematically combining theory,experiment and simulation,it is proved that the mathematical model,cavity modal simulation and structural modal experiment are effective and reliable approach to analyze the acoustic characteristics of accumulator.

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