Characteristics and suppression of NVH in twin screw refrigeration compressors

W Chen¹, Z Zhang², Z He², Z Xing²

¹ Suzhou Academy, Xi’an Jiaotong University, Suzhou, 215123, China
² School of Energy and Power Engineering, Xi’an Jiaotong University, Xi’an, 710049, China

Tjqchen@foxmail.com

Abstract. Characteristics of NVH (Noise, vibration and harshness) in twin screw refrigeration compressors, mainly caused by mechanical vibration and turbulent gas flow, are discussed in detail. According to the different generation mechanism, suppression methods of mechanical NVH and hydrodynamic NVH are proposed, respectively. Mechanical NVH is related to rotor profile, bearings, lubricating conditions and machining precision, while hydrodynamic NVH is inevitable and severe for all positive displacement compressors due to gas compression and turbulent flow. Concluded from three cases of NVH suppression for semi-hermetic twin screw refrigeration compressors, half-wave method based on acoustic interference theory, pulsation damper evolving from Helmholtz resonator and broadband perforated tube silencer are effective paths to alleviate hydrodynamic NVH in twin screw refrigeration compressors without bringing in extra energy losses.

1. Introduction

Compared with other types of refrigeration compressors, twin screw refrigeration compressor has advantages of high stability, tolerable with liquid and compact structure, making it widely employed in air conditioning, cold chain equipment and heat pump, etc. Twin screw refrigeration compressor has achieved remarkable improvements with both high volumetric efficiency and high adiabatic efficiency benefiting by precise processing equipment and proven technologies, such as profile design and oil injection. However, NVH (Noise, vibration and harshness) is gradually becoming prominent as noise pollution has been unacceptable in many areas, especially in applications closed to residential zones. Not only noise pollution, but also NVH can imperil the compressor’s efficiency and reliability [¹, ²]. With development of industrialization and urbanization, NVH standards for refrigeration equipment are more and more strict. Suppressing NVH at a low level has been the new challenge for twin screw refrigeration compressor and it is also a great highlight in market competition for manufactures.

2. Working principle of twin screw refrigeration compressor

Fig.1 shows a typical structure of the semi-hermetic twin-screw refrigeration compressor. It mainly consists of the casing, a pair of screw rotors (male rotor and female rotor), suction and discharge ports. The compressor and the motor are encapsulated in the same casing. The motor and the male rotor are coaxial. Driven by the motor, the male and female rotors engage and rotate like gears. And the
working volume enclosed by the rotors and the casing’s inner wall will expand and shrink periodically to achieve the process of gas suction, compression and exhaust \[3\]. According to the working principle, NVH in twin-screw refrigeration compressor can be divided into mechanical NVH caused by mechanical vibration and hydrodynamic NVH induced by gas flow pulsation.

![Fig.1 Typical structure of semi-hermetic twin-screw refrigeration compressor](image)

3. Mechanical NVH characteristics

Mechanical NVH is generated by structure vibration when the solid parts encounter impact, friction, alternating stresses or magnetic stresses. In twin screw refrigeration compressor, mechanical NVH mainly comes from the running rotors and bearings.

3.1. Vibration noise from rotor meshing

The male and female rotors are the key components of twin screw refrigeration compressor. During the working process, the rotors are not only subjected to the radial and axial forces from the gas, but also subjected to the force of the transmission mechanism and bearing support force. These forces change periodically, leading to mechanical vibration noise.

For twin screw refrigeration compressor, the male rotor directly drives the female rotor to rotate synchronously through the contact of the conjugated helical surfaces, inevitably producing mechanical vibration. The metal rotors are flexible due to material properties. Misalignment and imbalance because of machining or assembly errors can lead to radial vibration and abnormal noise when the rotors mesh with each other, which may also become the excitation source of mechanical NVH.

3.2. Vibration noise of bearings

Sliding bearings and rolling bearings are the commonly used bearings in twin screw refrigeration compressor. For the vibration of sliding bearings, the oil film will rupture if insufficient lubrication or abnormal friction happens, and then the “stick-slip” excitation between metals causes vibration. The vibration of the rolling bearings is mainly caused by the periodic impact of the discrete rolling body on the ball track \[4\]. In contrast, the vibration noise of the rolling bearings is greater than that of the sliding bearings, but the rolling bearings can provide accurate running precision and bear higher speed. For this reason, the rolling bearings are most commonly applied to bear the axial and radial force in the twin screw refrigeration compressor, while the sliding bearings are generally used in some large twin screw refrigeration compressors.

Under the premise that the machining accuracy and assembly error of mechanical parts are effectively controlled, the mechanical vibration noise of twin-screw refrigeration compressor can be effectively suppressed. Then the fluid dynamic noise is obviously exposed and becomes the dominating vibration noise source.
4. Hydrodynamic NVH characteristics

Hydrodynamic NVH refers to noise generated by the fluid vibration due to fluid flow or solid movement in the fluid. According to its generated location and characteristics, hydrodynamic NVH can be divided into suction noise, working chamber noise and exhaust noise.

4.1. Fluid noise inside working chamber

During the compression process, the enclosed working chamber is neither exposed to suction port nor discharge port. Gas in the working chamber will be compressed as the volume is decreasing and fluid noise will come up because the interior gas pressure can’t be completely homogeneous. Mass exchange through leakage paths also induces fluid noise, especially leakage between adjacent chambers through the blow hole. The blow hole, with relatively bigger flow area, connects two adjacent chambers under different pressure, similarly to two independent acoustic elements. Resonance may come up when the independent acoustic elements encounter unbalanced excitation, such as pressure pulsation, resulting in greater hydrodynamic noise. Besides, fluid noise induced by oil injection, liquid injection or superfeed may be also considerable because of drastic gas/liquid flow or flash.

4.2. Exhaust noise

Exhaust noise refers to discharge pressure pulsation. In general, gas pressure at the end of compression can’t be always equal to pressure in exhaust cavity, over-compression or under-compression is unavoidable. At the initial stage when the working chamber is exposed to discharge port, severe flow impact happens between gas in the working chamber and that in the exhaust cavity to reach pressure balance. As discharge area increases and volume of working chamber continuously decreases, gas inside working chamber is exhausted until working chamber disappears. Twin screw refrigeration compressor is rotating positive displacement compressor. At the same time, there are several working chambers in suction, compression and discharge process, respectively. Number of working chamber in discharge process is periodically varying, as well as the total discharge area and transient discharge speed, also resulting in discharge pressure pulsation.

4.3. Suction noise

Mechanism of suction noise is similar to exhaust noise. Characteristics of suction flow periodically change as a result of variation of suction area and suction speed, hence suction pressure pulsation, lower than discharge pressure pulsation, is induced. Sangfors et al. conducted a lot of research to identify main NVH sources in twin screw compressor [5]. The results all showed that NVH values at fundamental frequency of gas pressure pulsation and its harmonic frequencies were relatively high. In addition, hydrodynamic NVH induced by exhaust gas pulsation is more serious than that caused by suction gas pulsation as the gas density in the exhaust cavity is much higher.

5. Methods for mechanical NVH control

Mechanical vibration from running rotors and bearings transmits to compressor casing and radiates noise to surroundings. Control of mechanical NVH in twin screw refrigeration compressor is mostly carried out by manufactures and few research reports about mechanical NVH control of twin screw compressor can be found.

5.1. Improving structure precision

In order to control mechanical NVH at source, high machining precision of running parts and their fitting surfaces is necessary as well as the assembly process. High profile accuracy, surface finish and shaft system coaxiality of the rotors can effectively make mechanical NVH at a low level, accomplishing smooth operating. Based on experimental results, Jin et al. pointed out that improving
rotor precision by changing machining methods from milling to grinding can significantly suppress noises at medium-high frequencies \cite{6}.

5.2. Minimizing bearing clearance
In the premise of enough reliability, little bearing clearance is helpful to control the meshing rotor eccentricity and improve rotating accuracy. Therefore NVH induced by unbalanced mass can be suppressed at high speed operation. Yin et al. concluded through theoretical and experimental research that radial clearance of rolling bearings has the most significant influence on bearing NVH. Amplitude of NVH linearly rises as the radial clearance increases \cite{7}.

5.3. Optimized casing structure
Structure vibration of running solid parts finally transmits to compressor casing, exciting vibration of the casing and radiating noise to surroundings from the casing surface. If the casing has enough high stiffness to resist vibration excitation and small surface to reduce noise radiation, mechanical NVH must be well controlled. Therefore, compact structure is an important goal at design stage of twin screw refrigeration compressor and some strengthening ribs are necessary to strengthen the casing structure. Besides, double-wall casing is able to impede transmission of NVH and is usually employed in semi-hermetic twin screw compressor.

5.4. Shock absorber
Shock absorber is often used to damp vibration transmitting to pedestals from running machines. For twin screw refrigeration compressor, shock absorber should be designed according to rotor profile, rotating speed and compressor weight, thus it can be effective to damp vibration at fundamental frequency and harmonic frequencies.

6. Methods for hydrodynamic NVH control
As one type of rotating positive displacement compressors, periodic suction, compression and exhaust processes in twin screw refrigeration compressor inevitably produce gas pulsation, resulting in hydrodynamic NVH and even further structural resonance excitation which is destructive to system reliability. Consequently, managing to suppress hydrodynamic NVH by attenuating gas pulsation amplitude at its source and weakening its transmission is very important to achieve a reliable, effective and quiet twin screw refrigeration compressor.

6.1. Attenuating pulsation at discharge end
Based on the principle of acoustic interference, a half-wave bypass flow channel can be designed to produce a secondary flow pulsation which has the same amplitude but antiphase with the primary flow pulsation at discharge port. The two fluid flow will superimpose and offset each other to achieve attenuation of exhaust pulsation. Schematic of half-wave bypass flow channel is shown in Fig. 2.

![Fig.2 Schematic of half-wave bypass flow channel](image)

Excitation frequency and amplitude of the primary flow pulsation is $f_0$ and $p_0$, respectively. Path difference between the bypass channel and the primary channel is $s$. Fluid pressure $p_1$ in the primary channel:
Fluid pressure $p_2$ in the bypass channel:

$$p_2 = p_0 \cos(2\pi f_0 t - k(x + s))$$

where $k = \frac{2\pi f_0}{C_0}$ is the wave number of the primary pulsation and $C_0$ is fluid sound velocity.

When the secondary flow in the bypass channel mixes with the primary flow, mixed fluid pressure $p_3$:

$$p_3 = p_1 + p_2$$

$$p_3 = 2p_0 \cos(2\pi f_0 t - k(2x + s)/2) \cos(ks/2)$$

It can be found that the pressure $p_3$ comprises two parts: one is $\cos(ks/2)$, only concerned with the path difference $s$, defined as amplitude factor; the other is $2p_0 \cos(2\pi f_0 t - k(2x + s)/2)$, relating to time $t$, displacement $x$ and path difference $s$, defined as simple harmonic vibration factor.

In order to counteract the mixed flow pulsation completely, the amplitude factor should be zero:

$$\cos(ks/2) = 0$$

Solving the above equation 5, the result is:

$$s = (2n + 1)\frac{\lambda}{2} \quad n = 0, 1, 2...$$

Consequently, when path difference between the bypass channel and the primary channel is odd times of the fluid half-wave length, pulsation amplitude can be minimized as shown in Fig.3.

According to the actual periodic gas pulsation characteristics and oil-gas mixture sound velocity in twin screw refrigeration compressor, Zhou et al. [8] and Tan et al. [9] both designed half-wave bypass channel at discharge end as shown in Fig.4. Both experimental results showed that amplitude of exhaust gas pulsation was greatly attenuated at targeted frequencies and hydrodynamic NVH was effectively suppressed.
6.2. Helmholtz pulsation damper

Helmholtz resonator is a common noise reduction device in acoustics, which is mainly composed of a neck pipe and a chamber, as shown in Fig. 5. Under specific conditions, it can be used to attenuate gas pulsation amplitude in the exhaust cavity of twin screw refrigeration compressor. In Fig. 5, $V$ is volume of the chamber, $L$ is length of the neck pipe and $S$ is sectional area of the neck pipe. When length of the neck pipe $L$ is far smaller than wavelength of the incident sound wave, gas in the neck pipe can be regarded as a lumped element, and its mass can be expressed as $\rho_0 SL$. Displacement of the lumped element in the neck pipe will bring change of pressure inside the chamber. Therefore, if the lumped element intrudes into the chamber space, volume of gas inside the chamber will be compressed and pressure will be lifted, then the compressed gas will try to dislodge the incursive lumped element out. On the contrary, when the lumped element leaves the chamber of Helmholtz resonator, pressure of gas inside the chamber will decline and the lumped element will be tried to pushed back into the chamber by the outside high pressure gas. In this way, the chamber acts like a spring while gas in the neck pipe acts like a mass element.

![Fig. 5 Schematic of Helmholtz resonator and its acoustics model](image)

Similarly to the elastic coefficient of a spring, compressibility of gas reflects its pressure increasing degree of increase during compression process. The compressibility can be expressed by volumetric modulus of elasticity $K$:

$$
K = -\frac{\delta p}{\delta V} = \frac{\rho \delta p}{\rho \delta V} = \rho c^2
$$

(7)

For the Helmholtz resonator, displacement of gas in the neck pipe is represented by $x$, volume of the gas entering the chamber is $xS$, then pressure increment $\delta p$ inside the chamber, force $F$ acting on lumped element in the neck pipe and resistance $F_f$ acting on it during movement can be obtained from:
\[
\begin{aligned}
\delta p &= \rho c^2 \frac{Sx}{V} \\
F &= -\rho c^2 \frac{S^2x}{V} \\
F_I &= -R \frac{dx}{dt}
\end{aligned}
\]  

(8)

where \( R \) is damping coefficient.

The motion differential equation of the lumped element in the neck pipe can be written as:

\[
\rho SL \frac{d^2x}{dt^2} + R \frac{dx}{dt} + \rho \frac{pc^2S^2}{V} x = Sp(t)
\]  

(9)

and it can be solved:

\[
x = \frac{pS}{\frac{pc^2A^2}{V} - \omega^2 \rho LS}
\]  

(10)

When disturbance frequency is equal to natural frequency of gas inside the Helmholtz resonator, resonance of the lumped element in the neck pipe will be excited and its displacement \( x \) will reach maximum value, indicating:

\[
\frac{\rho c^2S^2}{V} - \omega^2 \rho LS = 0
\]  

(11)

Natural frequency \( f_r \) of gas in the Helmholtz resonator can be calculated by

\[
\begin{aligned}
\omega &= c \sqrt{\frac{S}{VL}} \\
\frac{f_r}{2\pi} &= \frac{c}{2\pi} \sqrt{\frac{S}{VL}}
\end{aligned}
\]  

(12)

When frequency of the incident sound wave \( p_i \) is close to this specific frequency \( f_r \), violent vibration will be induced in the neck pipe of Helmholtz resonator. Then sound energy will be consumed by overcoming the frictional resistance because of gas displacement in the neck pipe, and amplitude of the downstream sound wave will be attenuated.
Based on the principle of Helmholtz resonator, Wu et al. \cite{14} presented a Helmholtz pulsation damper mounting on the exhaust path of twin screw refrigeration compressor. Amplitude of discharge gas pulsation was successfully attenuated by more than 30% while vibration acceleration of the compressor footing was reduced by 36.2%-40.9% at the fundamental frequency of gas pulsation.

6.3. Broadband perforated silencer

Fig. 7 is illustration of the broadband perforated silencer. Similarly to Helmholtz pulsation damper, each hole on the perforated element combines a corresponding cavity, composing a Helmholtz resonator. Broadband perforated silencer can be regarded as a parallel unit of multiplied Helmholtz pulsation dampers.

\[
 f_{\text{MPA}} = \frac{c}{2\pi} \sqrt{\frac{P}{(t + 0.8d)D}}
\]

where \( c \) is sound velocity of the fluid, \( t \) is the thickness of the perforated element, \( d \) is diameter of the hole on the perforated element, \( D \) is the cavity depth of perforated element and \( P \) is the perforation rate.
Fig. 8 shows a broadband perforated tube silencer presented by Liu et al.\textsuperscript{[13]}. It was applied in variable speed twin screw refrigeration compressor, consisting of four cavities. With the four-cavity broadband perforated tube silencer, exhaust noise value at gas pulsation fundamental frequency declined by more than 3.0 dBA in the speed range of 3000rpm~4500rpm, while exhaust noise was suppressed by 5.0 dBA ~ 7.5 dBA in the high speed range of 4500rpm~5100rpm, achieving effective hydrodynamic NVH suppression in the full frequency band of the variable speed twin screw refrigeration compressor.

7. Conclusion
Twin screw refrigeration compressor is one type of rotating positive displacement compressors. Mechanical NVH is excited by rotating solid parts, mainly including rotors and bearings. The dominant hydrodynamic NVH is induced by gas pulsation due to periodic working process, inhomogeneous pressure distribution and mismatching of pressure ratio. Mechanism, generation location and characteristics of each NVH are discussed in detail. Mechanical NVH is sensitive to structure precision and bearing clearance. Compact double-wall casing with strengthening ribs and shock absorber are also effective to control mechanical NVH. Pulsation damping devices, including half-wave bypass channel based on acoustic interference theory, Helmholtz pulsation damper and perforated silencer, have been proved to be adequate for hydrodynamic NVH suppression.

References
[1] Jia X, Liu B, Feng J, et al. Influence of an Orifice Plate on Gas Pulsation in a Reciprocating Compressor Piping System. ARCHIVE Proceedings of the Institution of Mechanical Engineers Part E Journal of Process Mechanical Engineering 1989-1996 (vols, 2015, 229(1).
[2] Wu R, Tran V. Dynamic response prediction of a twin-screw compressor with gas-induced cyclic loads based on multi-body dynamics. International Journal of Refrigeration. 2016, 65(3) : 111-128.
[3] Xing Z. Twin-screw compressors —— theory, design and application. Beijing: China Machine Press, 2008.
[4] Norton M. Foundation of engineering noise and vibration analysis. translated by Y. Shen, et al. Beijing: Aviation industry press, 1993.
[5] Sangfors B. Modeling, measurement and analysis of gas-flow generated noise from twin screw compressors. International Compressor Engineering Conference at Purdue: West Lafayette, USA, 2000.
[6] Jin C, Fan L, Qiu Q, et al. Noise reduction research of semi-hermetic screw refrigeration. Noise and Vibration Control, 2003(4): 37-38.
[7] Yin Y, and Zhang J. Effect of structural parameters on vibration noise of rolling bearings. Noise and Vibration Control, 2014, 34(1):76-81.
[8] Zhou M, Chen W, et al. Active gas pulsation damping device in twin screw compressors. Chinese Patent, 201610965311.0.,2016-11-04.
[9] Tan J, Li R, et al. Screw compressor. Chinese Patent, 201810133021 .9. 2018-02-09.
[10] Wu X, Xing Z, Chen W, et al. Performance investigation of a pressure pulsation dampener applied
in the discharge chamber of a twin screw refrigeration compressor. International Journal of Refrigeration, 2018, 85.

[11] Ermilov M, Kryuchkov A, Balyaba M, et al. Development of a pressure pulsation damper for gas pressure regulators with account of operation parameters. Procedia Engineering, 2015, 106:277-283.

[12] Wu X, Chen W, Zhou M, et al. Development of gas pulsation damping device for twin-screw refrigeration compressor. Journal of Xi’an Jiaotong University, 2017, 51(4):23-29.

[13] Liu H, Zhang T, Chen W, et al. Experimental investigation of the perforated plate gas pulsation damping device with wide frequency bandwidth applied in variable rotational speed twin screw refrigeration compressor. Refrigeration and Air-Conditioning, 2018, 07(18):59-64.