Pulsation damping of the reciprocating compressor with Helmholtz resonator

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Abstract. Research presented in this paper investigated the mounting of a Helmholtz resonator near the valve chamber of a reciprocating compressor to attenuate the gas pulsation in the valve chamber as well as the pipeline downstream. Its attenuation characteristics were simulated with the plane wave theory together with the transfer matrix method, and the damping effect was checked by comparing the pressure pulsation levels before and after mounting the resonator. The results show that the Helmholtz resonator was effective in attenuating the gas pulsation in the valve chamber and piping downstream, and the pulsation level was decreased by 40% in the valve chamber and 30% at maximum in the piping downstream. The damping effect of the resonator was sensitive to its resonant frequency, and various resonators working simultaneously didn’t interfere with each other. When two resonators were mounted in parallel, with resonant frequencies equal to the second and fourth harmonic frequencies, the pressure pulsation components corresponding to the resonant frequencies were remarkably decreased at the same time, while the pulsation levels at other harmonic frequencies kept almost unchanged. After a series of simulations and experiments a design criterion of chock tube and volume parameter has been proposed for the targeted frequencies to be damped. Furthermore, the frequency-adjustable Helmholtz resonator which was applied to the variable speed compressor was investigated.

Keywords: Helmholtz resonator; gas pulsation; frequency-adjustable; parameters investigation

1. Introduction
The intermittent suction and discharge of reciprocating compressors would lead the gas pulsation and piping vibration in the system, which degrades the compressor performance, impede the working conditions of the valves and excites the piping vibration [1]. Therefore, it is very important to prediction and control the gas pulsation and piping vibration in a compressor piping system [2]. Currently, installing an orifice plate at the suction and discharge cylinder flange connections [3] is widely used to attenuate the pulsation in the cylinder nozzle as well as the piping system thanks to its simple structure and convenient installation. However, the orifice plate would generate an obvious pressure drop and consume a large amount of horsepower, which might significantly reduce compressor capacity and efficiency. Thus, controlling gas pulsations while maintaining acceptable compressor performance has been a key issue in reciprocating compressors for the last decades.

Helmholtz resonator had been proven to be effective for solving the problem of high amplitude compressor cylinder pulsations which is the source of the piping pulsation downstream. The SwRI (Southwest Research Institute) [4-6] studied many advanced pulsation control technologies including Helmholtz resonator, volume-choke-volume filter, infinite nozzle, and etc. Their research showed that the SBA (side branch absorber) could be installed on both low-speed and high-speed compressors to
replace any cylinder nozzle orifices that are consuming large amounts of horsepower. The location of installation is either the suction or discharge side of the cylinder or both side of the cylinder. Anderson [7] investigated a single side branch Helmholtz resonator in a circular duct and obtained the resonant frequency from the transfer function of the resonator. Broerman et al. [8] studied the VO (virtual orifice) to replace the compressor nozzle orifice. Their experiments indicated that the pulsation amplitudes in the cylinder nozzle were approximately 50% lower with the virtual orifice than they were with the normal orifice. For the double-acting compressor, Liu Boxiang [9] clarified the installation positions of Helmholtz resonator had a significantly influence on the attenuation effectiveness, and the best obtained when two resonators were installed at both the HE and the CE. Chen Zhicai [10] proposed an extended neck or a spiral neck to take the place of the traditional straight neck of a Helmholtz resonator when space was limited. Jean-Michel Coulon [11] investigated the role of distance between multiple Helmholtz resonator side branch openings on the whole array attenuation which could form a wide band silencer.

The analysis of the gas pulsation was based on the plan wave theory which has been proved accurate in the low-frequency pulsation analysis [12-13]. In this article, a model, based on the plan wave theory, was built to simulate the gas pulsation which contained reciprocating compressor, nozzle, chamber bottle, scrubber and heat exchanger. To verify the simulation, series of experiments were conducted with a reciprocating compressor. Based on the experimental results, the non-interference characteristics of multiple Helmholtz resonators was analysed, as well as the transmission properties of Helmholtz resonator with/without the piping system. For a wider absorption bandwidth, a frequency-adjustable Helmholtz resonator was proposed and investigated. Meanwhile, for the Helmholtz resonator parameters diversity of the same characteristic frequency, the further studies focused on the choke tube and volume's parameters analysis had been carried out, and a design criterion in practice application was recommended.

2. Theoretical studies on the frequency characteristic

As shown in figure 1, the Helmholtz resonator is composed of a choke tube and a volume, where V is the volume of the container, S is the cross-section area of the choke tube, L is the length of choke tube, d is the diameter of the choke tube.

![Figure 1. Illustration of Helmholtz resonator.](image1)

The characteristic frequency of the Helmholtz resonator could be calculated by following Equation,  

\[
f_0 = \frac{a}{2\pi \sqrt{LV}}
\]

(1)

where \(c\) is the local speed of sound, \(L_b\) is the acoustic length of the choke tube, which could be obtained from Eq. (2),  

\[
L_b = l + 0.7d
\]

(2)

where \(l\) is the physical length of the choke tube.
To get the frequency characteristic of the Helmholtz resonator in the piping system, the study of acoustic propagation property in branch piping is necessary. Figure 2 shows the branch pipe system, where $p_i$ is the main piping incoming wave pressure, $p_r$ is the reflected wave pressure, $p_t$ is the transmitted wave pressure and $p_2$ is the side branch wave pressure.

![Figure 2. Acoustic propagation in branch piping.](image)

Selected the intersection of the branch pipe and the main pipe as the origin of coordinates, based on the plane wave theory, the pressures and velocities could be obtained from Eq. (3):

$$
\begin{align*}
    p_i &= p_a e^{i\omega t}, \quad v_i = \frac{p_i}{\rho_0 c_0} \\
    p_r &= p_a e^{i\omega t}, \quad v_r = -\frac{p_r}{\rho_0 c_0} \\
    p_t &= p_a e^{i\omega t}, \quad v_t = \frac{p_t}{\rho_0 c_0} \\
    p_2 &= p_a e^{i\omega t}, \quad v_2 = \frac{p_2}{S_z z_2}
\end{align*}
$$

(3)

Considering the continuity of the pressure and velocity at the intersection point, Eq. (3) could be deduced to Eq. (4) and (5):

$$
\begin{align*}
    p_i + p_r &= p_t = p_2 \\
    S v_i + S v_r &= S v_t + S_z v_2
\end{align*}
$$

(4)
(5)

And the reflection coefficient of pressure $|r_r|$ could be derived from Eq. (3) and (5):

$$
|r_r| = \left| \frac{p_r}{p_a} \right| = \frac{-\rho_0 c_0 / 2S}{\rho_0 c_0 / 2S + z_2}
$$

(6)

The transmission coefficient of pressure could be obtained from Eq. (4) and (6):

$$
\alpha_t = \frac{R_{a2}^2 + X_{a2}^2}{(\rho_0 c_0 / 2S + R_{a2})^2 + X_{a2}^2}
$$

(7)

The impedance of Helmholtz resonator could be calculated by following equation:

$$
Z_{a1} = j(\alpha M_a - \frac{1}{\alpha C_a}) = jX_a
$$

(8)

where $M_a$ is the sound mass, $C_a$ is the acoustic capacitance.

Generally, the transmission coefficient is used to denote the frequency characteristic of the Helmholtz resonator, and the transmission coefficient of the branch pipe with the Helmholtz resonator could be derived from Eq. (7) and (8):
\[
\alpha_t = \frac{1}{1 + \frac{\rho_0^2 c_0^2}{4S^2(\omega M_a - \frac{1}{\omega C_a})^2}} = \frac{1}{1 + \frac{c_0^2}{4S^2(\frac{\omega L_b}{S_b} - \frac{\omega N^2}{S_b})^2}}
\tag{9}
\]

In order to study the frequency characteristic of a Helmholtz resonator, a series of simulations and experiments were conducted with an air reciprocating compressor and piping system. The compressor rated speed was 750 rpm, and a frequency transformer was applied to adjust rotational speed. Considering the fundamental frequency of the compressor was 12.5 Hz, two Helmholtz resonators with characteristic frequency of 25 Hz and 50 Hz were designed and processed, which were named 2x resonator and 4x resonator. The parameters of resonators were shown in Table 1.

| Parameter of the resonators. | 2x resonator | 4x resonator |
|-----------------------------|-------------|-------------|
| Diameter of choke tube/m    | 0.051       | 0.051       |
| Length of choke tube /m     | 0.95        | 0.4         |
| Diameter of volume/m        | 0.152       | 0.152       |
| Length of volume/m          | 0.6         | 0.35        |
| Volume/m³                   | 0.011       | 0.007       |

The transmission coefficient of the 2x resonator can be obtained by Eq. (9), as shown in figure 3. The transmission coefficient decreased sharply to 0 around 25 Hz. Beyond this narrow bandwidth, the transmission coefficient remained at 1.0. It indicated that the 2x resonator merely worked efficiently at its characteristic frequency. When the frequency to be suppressed coincides with the characteristic frequency, the Helmholtz resonator could attenuate gas pulsation effectively. Otherwise, the resonator had no effect.

![Figure 3. Theoretical transmission coefficient of 2x Helmholtz resonator.](image)

The 2x Helmholtz resonator connected with the main piping was also simulated, as shown in figure 4. A constant velocity was given as a motivation at the left end of the main piping. And the pressure at the right end, which was an anechoic end, could be obtained by the frequency-swept method. The simulation result of the 2x resonator transmission coefficient was shown in figure 5. It was noted that there was an obvious difference between the transmission coefficients in figures 3 and 5. The analysis would be discussed after completed the experimental investigation.
In order to study the pulsation attenuation of a Helmholtz resonator, an entire interstage piping system of the double acting air reciprocating compressor adopted for the experiments was built. As shown in figure 6, the model contained the 1st discharge nozzle, 1st discharge chamber, heat exchanger, scrubber, 2nd suction tank and 2nd suction nozzle. The 2x Helmholtz resonator was mounted near the 1st discharge valve chamber. The boundary conditions contained the reciprocating compressor motivation and the closed end. A series of simulations were carried out at various compressor rotational speed. The rotational speed was changed from 650 rpm to 800 rpm, and the interval was set as 50 rpm. Nine nodes were monitored to record the pulsation pressure. Node 1 to 3 were near the 1st discharge cylinder, including the Head End and Crank End. Node 4 and 5 were on the 1st discharge nozzle. Node 6 and 7 were on the 1st discharge tank. Node 8 and 9 were on the pipe after the buffer tank.
Figure 7 illustrated the pressure pulsation amplitude of the nine nodes at 2’ order under different compressor rotational speed. As shown in figure 7, thanks to the attenuation effect of the 1st discharge tank, the pressure pulsation amplitude of nodes 5 to 9 were much less than that of nodes 1 to 4. Figure 8 showed the pressure pulsation attenuation rate of the nine nodes at 2’ order under different rotational speed. It was obvious that the 2x Helmholtz resonator had no influence on the pulsation under 650 rpm and 700 rpm, but the maximal attenuation rate was reached 40% while the compressor rotational speed was at 750 rpm. The 2’ order frequency of 750 rpm was equal to the characteristic frequency of 2x Helmholtz resonator, so the transmission coefficient of the resonator reached the minimum value, which led to a maximal pulsation attenuation. Meanwhile, the pressure pulsation attenuation rate under 750 rpm declined from 40% to 20% with the nodes changed from 1 to 9. It indicated that a Helmholtz resonator could attenuate pressure pulsation at not only the mounting position but also the pipeline downstream, but the farther away from the mounting position, the less attenuation effect of the resonator.

![Figure 7](image1.png)

**Figure 7.** Pressure pulsation amplitude at different rotational speed.

![Figure 8](image2.png)

**Figure 8.** Pressure pulsation attenuation rate at different rotational speed.
3. Experimental study
To verify the attenuation effect and the frequency sensibility of Helmholtz resonator, the 2x resonator and 4x resonator whose characteristic frequency were 25 Hz and 50 Hz were mounted on the cylinder valve gap, as shown in figure 9. A gate valve was employed as a switch to control the resonator quickly and easily. Two piezoresistive pressure sensors were used to collect the pressure signals at different position, and the signals were transferred into the data acquisition system for further processing and recording. The sensor 1 was installed in the valve chamber and the sensor 2 was installed in the pipeline after the 1st discharge tank.

![Schematic diagram of the test rig.](image)

Figures 10 and 11 illustrated the first six order’s pressure pulsation amplitude of the sensor 1 and 2 while the compressor rotational speed kept at 750 rpm. It can be seen that as the 2x resonator connected to the valve chamber, the pressure pulsation of 2’ order declined from 3.75 kPa to 2.25 kPa at the sensor 1 position, and at sensor 2 position, the amplitude declined from 3.40 kPa to 2.40 kPa. However, the pressure pulsations of other orders were nearly unchanged at all sensor positions no matter the 2x resonator was connected or not. When the 4x resonator was connected to the valve chamber, the pressure pulsations of 4’ order were decreased from 1.65 and 1.20 kPa to 0.95 and 0.80 kPa at two sensor positions, respectively. Similarly, connecting the 4x resonator also had no influence on the pressure pulsation to the other orders. It was worth noted that while the two resonators were connected to the valve chamber in parallel, the pressure pulsation amplitudes of 2’ and 4’ orders were both decreased simultaneously, and the attenuation rates were as same as the resonators worked separately. Meanwhile, the pressure pulsations of the other orders were still almost unchanged.

Based on the experimental results, it can be draw that employing a Helmholtz resonator could attenuate the pressure pulsation effectively in the valve chamber and pipeline downstream. However, the pulsation attenuation rates of 2’ and 4’ order were 40.0% and 42.4% at valve chamber, and 29.4% and 33.3% at pipeline downstream. Furthermore, employing multiple resonators could attenuate the multi-frequency pulsation synchronously, which proposed an effective way to attenuate the pressure pulsation which was resulted from more than one high-amplitude orders.
Adjusting the compressor rotational speed from 600 rpm to 810 rpm and connecting the 2x resonator to the valve chamber. The pressure pulsation attenuation rates of the 2’ order at sensor 1 position were shown in figure 12. It can be seen that attenuation rate increased obviously as the rotational speed changed from 600 rpm to 750 rpm, then decreased from 40.0% to 33.8% when the speed was increased continuously. The maximum attenuation effect occurred at the speed of 750 rpm, whose corresponding frequency was equal to the characteristic frequency of the 2x resonator. It was indicated that employing a Helmholtz resonator could attenuate the pressure pulsation in a bandwidth, but the best effect occurred at its characteristic frequency.

4. Frequency-adjustable Helmholtz resonator
As discussed above, the pressure pulsation attenuation effect of the Helmholtz resonator was not satisfied when its characteristic frequency was not equal to the targeted attenuation frequency, which was a common phenomenon for the variable speed compressor. Therefore, a kind of frequency-
adjustable Helmholtz resonator was designed and manufactured as shown in figure 13(a). The choke tube was composed of the fixed choke and adjustable choke. By rotating the adjustable lever, the length of the choke tube was changed, so as the characteristic frequency. Thus, the adjustable Helmholtz resonator would have a maximal attenuation effect within a range in the frequency.

![Physical model](image1)

![Experiment test](image2)

**Figure 13.** Frequency-adjustable Helmholtz resonator.

By rotating the adjustable lever, the characteristic frequency of this adjustable Helmholtz resonator could be changed from 20 Hz to 26.5 Hz. To verify the attenuation effect of the adjustable resonator, the compressor rotational speed was changed from 600 rpm to 795 rpm, and the characteristic frequency of the adjustable resonator was adapted to motivation frequency at each rotational speed. Figure 14 illustrated the comparison of the transmission coefficient between the 2x resonator and the adjustable Helmholtz resonator. It can be seen that the transmission coefficient of the 2x resonator decreased obviously with the motivation frequency increased from 20 Hz to 25 Hz (compressor rotational speed increased from 600 rpm to 750 rpm), and then increased with continuing to increase the motivation frequency. However, the transmission coefficient of the adjustable Helmholtz resonator was around 0.63 as the motivation frequency was changed for 20 Hz to 26.5 Hz. It indicated that a satisfied attenuation effect could be achieved in a range of the compressor rotational speed by using the adjustable Helmholtz resonator.

![Transmission coefficient comparison](image3)

**Figure 14.** Comparison of the transmission coefficient between 2x resonator and adjustable resonator.

5. Parameter Analysis

According to Eq. (1), there were several combinations of the Helmholtz resonator parameters for one specific characteristic frequency, including the volume, the length and the diameter of the choke tube. However, there was few studies of the Helmholtz resonator parameters’ influence on the attenuation effect. Thus, the further simulation and experiment investigations were carried out to analyse the influence of the resonator parameters. In the simulation investigations, the interstage piping model was the same model as shown in figure 6, except the Helmholtz resonator. The compressor rotational speed was set at 750 rpm. The characteristic frequency of the Helmholtz resonators was kept at 25 Hz, which
was equal to the characteristic frequency of the 2x resonator. The \( d \) (the diameter of the choke tube) was kept at 50 mm, the \( l \) (the length of the choke tube) was serialization from 0.2 mm to 1.4 mm. The other key non-dimensional parameter was \( d/D \) (\( D \) was the diameter of the volume), which indicated the diameter rate of the resonator. It was serialization from 0.15 to 0.45, and then the \( L \) (the length of the volume) could be calculated from the Eq. (1).

**Figure 15.** 2’ order pressure pulsations in the valve chamber.

Figure 15 illustrated the 2’ order pressure pulsations in the valve chamber where the Helmholtz resonator was mounted on. The 2’ order pressure pulsation in same position without the resonator was 3.8 kPa. It was worth noting that although the characteristic frequencies of the various resonators were same, the pressure pulsations had a remarkable difference, the maximum pressure pulsation was probably twice that of the minimum. It can be seen that for the same length of the choke tube, the pressure pulsation decreased at first and then raised up with increasing of the \( d/D \). And no matter how much the length of the choke tube was, the minimum pressure pulsations were always obtained when the \( d/D \) was 0.35. It indicated that compared to the diameter of the volume, the diameter of the choke tube could be neither too big nor too small when design a Helmholtz resonator. Meanwhile, when the diameters of the volume and the choke tube were confirmed, the pressure pulsation fell obviously with decreasing of the length of the choke tube. It meant that the bigger volume had a better pressure pulsation attenuation effect.

**Figure 16.** Three resonators with the different parameters.
In order to verify the simulation results, three different Helmholtz resonators were designed and processed. These resonators had the same characteristic frequency and the other different parameters, as listed in Table 2. The HR1 and HR3 had the same d/D (the diameter rate of the Helmholtz resonator) and the length of choke tube, but the volume of HR1 was larger than that of HR3. The volume of HR2 was equal to that of HR1, but the diameter rate of HR2 was only 0.17, which was much smaller than HR1. The compressor rotational speed was kept at 750 rpm, so the 2’ order of the compressor motivation frequency was 25 Hz, which coincided with the resonators characteristic frequencies. Figure 17 illustrated the 2’ order pressure pulsation amplitude in the valve chamber with different resonators. It can be seen that the pressure pulsation decreased from 3.8 kPa to 2.4 kPa when the HR1 was mounted, the attenuation rate of the HR1 achieved 36.8%. However, when the HR3 was mounted, the pressure pulsation was 3.1 kPa, and the attenuation rate was 18.4%. It indicated that a Helmholtz resonator with a larger volume had a better attenuation effect. Meanwhile, although the volume of the HR2 was equal to that of the HR1, the attenuation rate of the HR2 was only 5.3%. It suggested that the diameter rate (d/D) of the Helmholtz resonator had a significant influence on the attenuation effect. According to the experiment and simulation results, a design criterion of the Helmholtz resonator in practice application could be recommended, the diameter rate is the most important resonator parameter, which should be set as 0.3 to 0.35, meanwhile the volume of the resonator should be designed large enough to ensure a better attenuation effect.

![Figure 17](image.png)

**Figure 17.** 2’ order pressure pulsation amplitude with different resonators.

### 6. Conclusions

To investigate the attenuation effect of a Helmholtz resonator, the transmission coefficient in side branch piping system was studied on the basis of the plane wave theory with transfer matrix method. A series of simulations and experiments were carried out. The following conclusions can be drawn:

1. Mounting a Helmholtz resonator near the discharge valve chamber could significantly attenuate the pressure pulsation in the valve chamber and the pipeline downstream.

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**Table 2.** Parameter of three resonators.

| Parameter                          | HR1 | HR2 | HR3 |
|-----------------------------------|-----|-----|-----|
| Diameter of choke /m              | 0.051 | 0.034 | 0.034 |
| Diameter of volume/m              | 0.152 | 0.2 | 0.103 |
| Length of Choke tube/m            | 0.95 | 0.4 | 0.95 |
| d/D                               | 0.33 | 0.17 | 0.33 |
| Characteristic frequency/Hz       | 25 | 25 | 25 |
(2) Multiple Helmholtz resonators can be mounted in parallel to attenuate the multi-frequency pulsation synchronously and effectively, which proposes a solution to attenuate the pressure pulsation which is resulted from more than one high-amplitude orders.

(3) The frequency-adjustable Helmholtz resonator whose choke tube was composed of fixed choke and mobile choke proved valid in the variable speed compressor. It eliminated the inconvenience of the resonator on account of its frequency sensibility.

(4) For a specific characteristic frequency, the Helmholtz resonators with various parameters have different attenuation rates. To obtain the best attenuation effect, the diameter rate of the Helmholtz resonator should be set as 0.3 to 0.35, and the volume of the resonator should be designed as large as possible.

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