A study on pressure pulsations in the discharge piping of an oil-injected screw compressor

Untersuchung von Schwingungen im Druckrohr eines ölüberfluteten Schraubenverdichters

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Abstract

A problem of noise and vibration in the compressor discharge piping arises if the natural pulsations of the gas in the piping get in the resonance with the pulsations sent by the compressor. The frequency of the screw compressor generated impulses depends on the rotation speed and lobe number of the driven rotor. The natural frequency of the medium in the discharge pipe depends among other on the speed of sound in the gas-oil two-phase mixture. Using a mathematical procedure for the wave separation, the sound speed in the discharge medium was evaluated from the dynamic pressure measurements on an oil-injected screw compressor package. It was found that the influence of the oil on the sound speed in gas is rather weak and it is acceptable to use the sound speed of the dry gas in calculations of the natural frequency. For a real application a simulation model based on the wave equation was set up and solved in 3D using commercial FEM software. The results were used to solve the resonance problem in the discharge piping of a screw compressor package.

Kurzfassung

Das Problem von Lärm und Schwingungen im Verdichterdruckrohr entsteht, wenn die Eigenschwingungen des Gases im Druckrohr in Resonanz mit verdichterspezifischen Druckimpulsen geraten. Die vom Schraubenverdichter generierte Druckimpulsfrequenz hängt von der Drehzahl und von der Zähnezahl des angetriebenen Rotors ab. Die Eigenfrequenz des Fluids im Druckrohr ist unter anderem von der Schallgeschwindigkeit im Zweiphasengemisch von Gas und Öl abhängig. Die Schallgeschwindigkeit im Druckgasölgemisch wurde aus dynamischen Druckmessungen am Druckrohr eines ölüberfluteten Schraubenverdichteraggregates und durch ein mathematisches Verfahren für das Trennen von Wellen abgeleitet. Es wurde festgestellt, dass der Einfluss von Öl auf die Schallgeschwindigkeit im Gas ziemlich gering ist, und es ist akzeptabel, bei Berechnungen

der Eigenfrequenzen den Schallgeschwindigkeitswert für das trockene Gas zu nutzen. Für eine konkrete Anwendung wurde mit Hilfe einer kommerziellen FEM Software ein auf der Wellengleichung basiertes dreidimensionales Simulationsmodell aufgestellt und gelöst. Die Ergebnisse wurden verwendet, um das Resonanzproblem im Druckrohr eines Schraubenverdichteraggregates zu lösen.

1. Introduction

Pressure pulsations in the compressor pipelines result in troublesome vibrations and noise. In acute cases the span of these pulsations can become higher than the difference between the discharge and suction pressure leading to a failure of the measuring instruments or to a breakdown of mechanical components of the compressor package. Such components as valves and filters as well as welded and threaded connections in the pipelines and even the compressor bearings can collapse under extreme vibration conditions. Although it may seem from the measurements that the faulty compressor emits these pulsations, the real reason of the problem is the resonance phenomenon with so called standing waves in the piping.

The resonance occurs in the compressor suction or discharge piping if the frequency of the suction or discharge gas pulsations generated by the compressor becomes equal to the natural oscillation frequency of the gas in the piping. To avoid the resonance it is necessary to ensure that the frequencies are far enough from each other.

The pulsatory nature of the compressor generated flow is inherent to all types of the mechanical compression machines. The pulsation frequency depends on the machine type and rotation speed. For the screw compressor it is calculated as

$$f_c = n \cdot z \,, \tag{1}$$

where n is the rotation speed and z is the lobe number of the driven rotor.

The gas in the piping has certain natural oscillation frequencies because of the reflection of the pressure waves. The reflection occurs at the ends of the pipe and at any obstacle in the flow such as valves, pipe turns, changes in the diameter, etc. The reflected wave travels in the direction opposite to the incident wave. In that way the reflected wave reaches the origin and reflects again having now the same direction as the incident wave. If at this moment both waves have the same phase then they amplify each other and the process of natural oscillation goes on. If the waves have different phases the amplitude decays. Thus, the waves with certain oscillation frequencies (the natural frequencies) are supported by the medium, while other frequencies are damped. The natural frequencies depend on the piping geometry, wave reflection conditions on the boundaries (the phase shift at the moment of reflection) and wave travelling speed (the speed of sound). For example, the natural frequencies of the gas in a straight pipe with one end closed and the other end open to the infinite volume are given by

$$f = (2q+1)\frac{c}{4L}, \qquad (2)$$

where c is the sound speed, L is the pipe length, q = 0 for the fundamental frequency and q = 1, 2, 3, … for the 1st, 2nd, 3rd, … overtone.

The resonance condition for the screw compressor connected to such a pipe reads

$$m \cdot n \cdot z = (2q+1)\frac{c}{4L}. \qquad (3)$$

The harmonic number m = 1, 2, 3, … is included in Eq. (3) because the resonance can occur at one of the multiples of the compressor lobe frequency. The resonance condition should be checked for the first two harmonics at least. Higher harmonics are less critical because in the viscous flow the oscillations at higher frequencies quickly dissipate.

As shown, the natural frequency depends on the sound speed in the compressed medium. For dry gases the value of the sound speed can be found based on the thermodynamic properties of the gas. However, the medium in the discharge piping of the oil-injected screw compressors is not a dry gas but a two-phase gas-oil mixture. The theoretical prediction of the sound speed and, as a result, prediction of the natural frequencies becomes, therefore, a more difficult task. The classical Wood equation is widely used in the literature [1] - [3] to calculate the sound speed in two-phase gas-liquid mixtures. According to the equation the sound speed in a two-phase mixture is lower than in either of the pure components and even a small change in the phase fractions can lead to significant changes in the speed of sound. However, the Wood's equation has been derived for homogenous mixtures and although it has been experimentally proven for bubbly liquids [4] it still remains unclear how far is the theory applicable to the gas-oil two-phase flow in the discharge piping of a screw compressor.

A direct measurement of the sound speed in the compressor discharge pipe is problematic because of the pressure wave reflection at the oil separator end of the pipe. The incident and reflected waves form a standing wave as a result of their superposition. Only the resultant, standing wave can be measured with pressure sensors. Hence, to determine the sound speed the resultant wave has to be mathematically separated into the incident and reflected wave. The idea of the wave separation was presented by Poysat and Liegeois [5]. They published results of the pressure pulsation measurements and numerical wave separation for the discharge pipe of a scroll compressor. The obtained value of the sound speed was close to that of the pure refrigerant. This statement may differ for the oil-injected screw compressor because the oil fraction in the screw machine discharge flow is usually bigger than in the scroll.

This paper proposes an analytical method of the wave separation. The only numerical steps are the fast Fourier transformation (FFT) and the inverse FFT (iFFT) of the pressure signal. The well known advantages of analytical methods are the minimal programming effort, short computation time and solution stability. Based on the presented method the sound speed in the discharge pipe of the oil-injected screw compressor is evaluated from the dynamic pressure measurements. Afterwards, as an application example shows, it becomes possible to find the natural frequencies of complex pipe geometries and solve the problem of resonance in the discharge piping of screw compressor packages.

2. Analytical evaluation of the sound speed by separation of the incident and reflected waves

For a straight pipe with a diameter smaller than the half of the wave length the problem can be treated as one-dimensional. If a wave has the amplitude A and initial phase φ then the complex amplitude of this wave is defined as

$$\mathbf{A} = A \cdot \exp(i\varphi), \tag{4}$$

where $i = \sqrt{-1}$ is the imaginary unity.

If the harmonic pressure wave with the cycling frequency $\omega = 2\pi f$ propagates in the x-direction with the speed of sound c then the pressure in point x is described as

$$p = \mathrm{Re}\{\mathbf{A} \cdot \exp(i(\omega t - kx))\}, \tag{5}$$

where $\mathrm{Re}\{\ \}$ states for the real part of a complex number, t is the time, $k = \omega/c$ is the wave number.

At every location along the discharge pipe the resultant pressure wave is the superposition of the waves travelling from the compressor to the oil separator and back to the compressor:

$$p = \mathrm{Re}\{\mathbf{A}\cdot\exp(i(\omega t - kx)) + \mathbf{B}\cdot\exp(i(\omega t + kx))\} = \mathrm{Re}\{[\mathbf{A}\cdot\exp(-ikx) + \mathbf{B}\cdot\exp(ikx)]\cdot\exp(i\omega t)\}, \tag{6}$$

where $\mathbf{A}$ and $\mathbf{B}$ are the complex amplitudes of the incident and reflected waves respectively.

The complex amplitude of the resulting wave in Eq. (6) is

$$\mathbf{C}(x) = \mathbf{A}\cdot\exp(-ikx) + \mathbf{B}\cdot\exp(ikx). \tag{7}$$

The module of amplitude $\mathbf{C}$ depends on the coordinate, therefore the resulting wave is a standing wave. There are three unknowns in Eq. (7): $\mathbf{A}$, $\mathbf{B}$ and k. The complex amplitude $\mathbf{C}$ of the resultant wave can be obtained by the FFT of the pressure signal measured over the time at point x since the FFT delivers the amplitude and phase distribution in the frequency domain

$$\mathbf{C}(x,\omega) = \mathrm{FFT}[p(x,t)]. \tag{8}$$

To find the three unknowns in Eq. (7) the pressure pulsations have to be measured in three points, say in x_1, x_2 and x_3. Next, the system

$$\begin{cases} \mathbf{C}(x_1,\omega) = \mathbf{A}(\omega)\cdot\exp(-ikx_1) + \mathbf{B}(\omega)\cdot\exp(ikx_1) \\ \mathbf{C}(x_2,\omega) = \mathbf{A}(\omega)\cdot\exp(-ikx_2) + \mathbf{B}(\omega)\cdot\exp(ikx_2) \\ \mathbf{C}(x_3,\omega) = \mathbf{A}(\omega)\cdot\exp(-ikx_3) + \mathbf{B}(\omega)\cdot\exp(ikx_3) \end{cases} \tag{9}$$

has to be solved for each frequency in the domain.

The amplitude of the incident wave is obtained from the first equation in (9) as

$$\mathbf{A}(\omega) = \mathbf{C}(x_1,\omega)\cdot\exp(ikx_1) - \mathbf{B}(\omega)\cdot\exp(2ikx_1). \tag{10}$$

The substitution of Eq. (10) into the second equation in (9) yields the amplitude of the reflected wave

$$\mathbf{B}(\omega) = \frac{\mathbf{C}(x_2,\omega) - \mathbf{C}(x_1,\omega)\cdot\exp(ik(x_1 - x_2))}{\exp(ikx_2) - \exp(ik(2x_1 - x_2))}. \tag{11}$$

Introducing the expressions for $\mathbf{A}$ and $\mathbf{B}$ into the third equation in (9) and performing algebraic transformations we obtain

$$\mathbf{C}(x_1,\omega)\cdot\sin(k(x_2 - x_3)) + \mathbf{C}(x_2,\omega)\cdot\sin(k(x_3 - x_1)) + \mathbf{C}(x_3,\omega)\cdot\sin(k(x_1 - x_2)) = 0. \tag{12}$$

Equation (12) can be numerically solved for k. But in the case of equally spaced points x_1, x_2 and x_3 there exists an analytical solution, which greatly simplifies the computations! The solution reads

$$k = \frac{1}{\Delta x}\arccos\left(\frac{\mathbf{C}(x_1,\omega)+\mathbf{C}(x_3,\omega)}{2\mathbf{C}(x_2,\omega)}\right), \tag{13}$$

where Δx is the spacing between the points x_1, x_2 and x_3.

Equations (10), (11) and (13) form the analytical solution of system (9). The sound speed is obtained from the wave number as

$$c = \frac{\omega}{k}. \tag{14}$$

The incident and reflected waves are reconstructed in the time domain with the inverse Fourier transformation:

$$p_i(x,t) = \mathrm{iFFT}[\mathbf{A}(\omega)\cdot\exp(-ikx)], \tag{15}$$

$$p_r(x,t) = \mathrm{iFFT}[\mathbf{B}(\omega)\cdot\exp(ikx)]. \tag{16}$$

The resultant pressure wave at any point x at time t can be calculated as

$$p(x,t) = p_i(x,t) + p_r(x,t). \tag{17}$$

With the presented method it is possible to calculate the pressure distributions in the incident, reflected and resultant waves as functions of the coordinate and time. Another important result obtained with the method is the sound speed. All calculations are based on the pressure pulsation measurements in three equally spaced points. Except the Fourier transformations all calculations are analytical.

3. Results of the experiments

The experimental installation consists of an oil-injected screw compressor connected with a straight discharge pipe to the oil separator. The theoretical volume displacement of the compressor is 2748 m^3/h at 2940 rpm; the rotor lobe combination is 5/6. Five equally spaced piezo-resistive pressure sensors are flush mounted on the discharge pipe. Three of them are used to collect the input data for the model presented in Section 2 and the rest two are used to validate the solution. The spacing between the sensors is 100 mm, the distance between the compressor rotors discharge end face and the oil separator end of the discharge pipe is 1575 mm and the pipe diameter is DN150. The installation is filled with ammonia. Figure 1 represents a view of the experimental set-up.

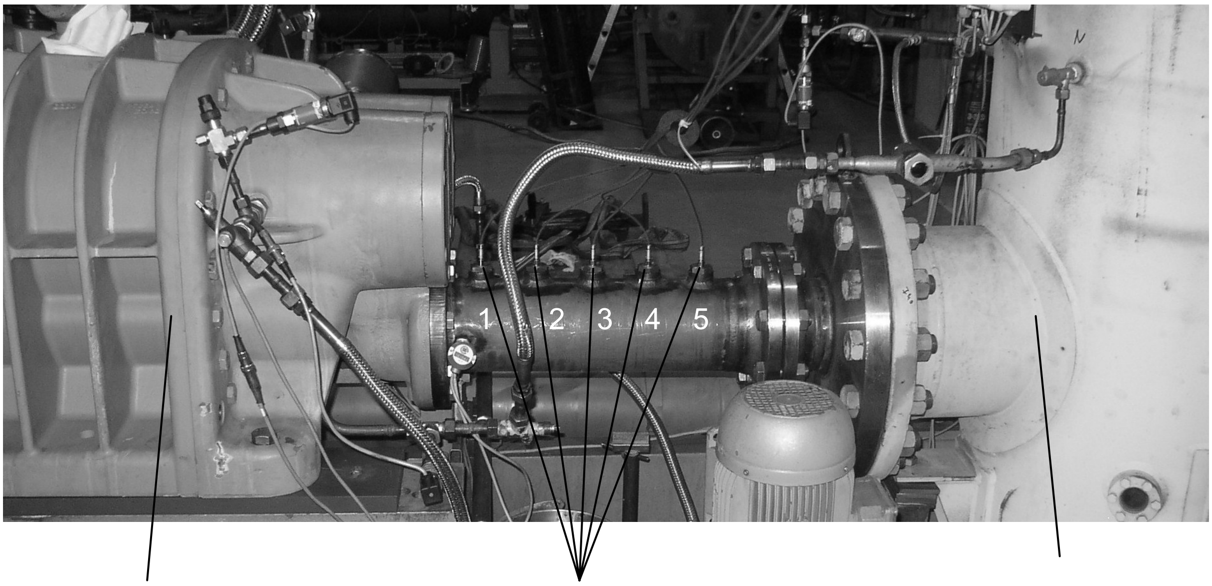


Fig. 1: Experimental installation

Figure 2 shows an example of the measured pressure oscillations. The sensors are numbered as illustrated in Fig. 1. The results were obtained with the compressor rotation speed of 2400 rpm, suction pressure of 0.4 bar(a), discharge pressure of 1.9 bar(a) and with the discharge temperature of 60°C. The sensors show different pressure amplitudes. This confirms qualitatively Eq. (7): caused by the wave reflections the amplitude of the resultant wave is coordinate dependent.

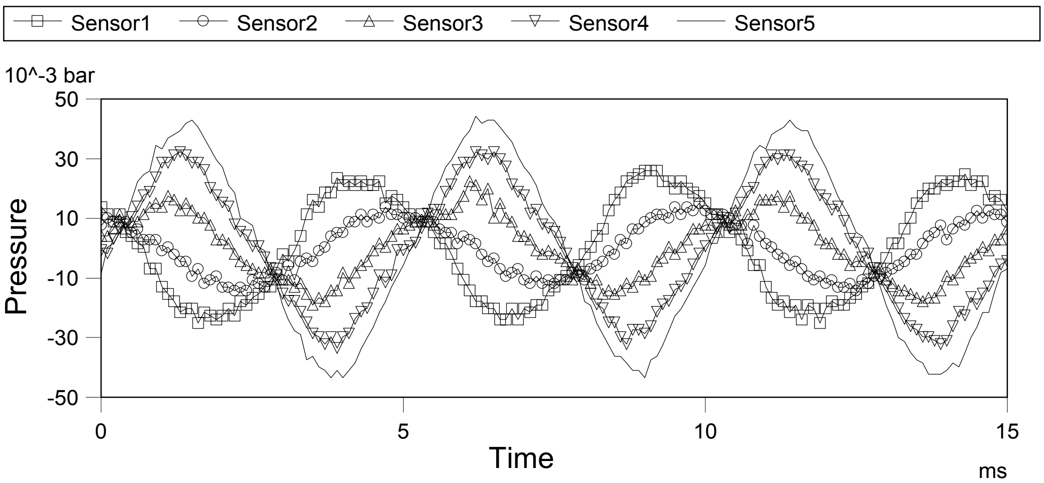


Fig. 2: Pressure pulsations measured with five sensors

The readings of Sensors 1, 3 and 5 are used as the input for the model. The model's output is the sound speed as well as the pressure in the incident, reflected and resultant waves as a function of the coordinate and time. To validate the solution the model predictions are compared with the readings of two additional sensors, namely Sensors 2 and 4. Figure 3 shows the pressure measured with Sensor 2 and the pressure in the resultant wave calculated with the model for the coordinate of Sensor 2. The agreement between the measured and calculated pressure is rather good. Some deviations in the high frequency range can be caused by transversal pressure waves, which are registered by the sensor but are not considered in the one-dimensional model.

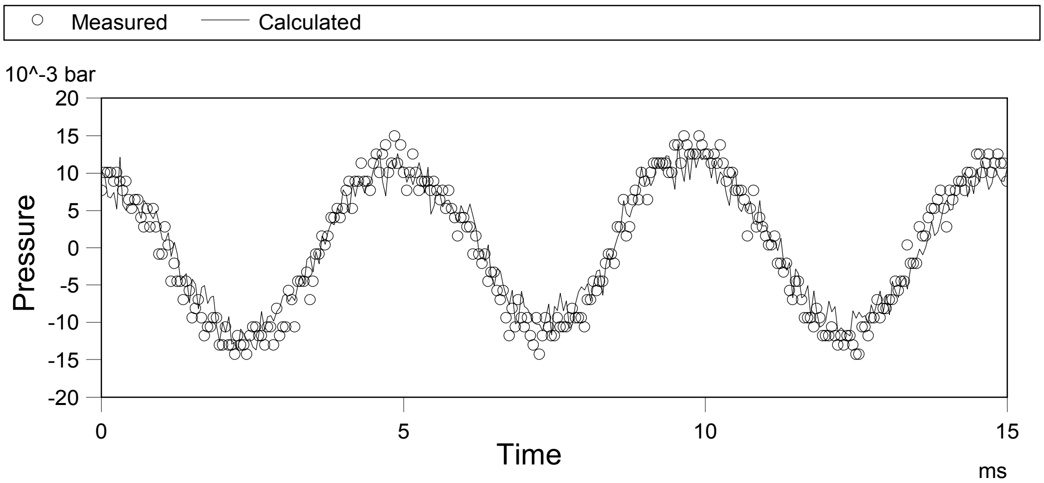


Fig. 3: Comparison between the measured and calculated pressure, Sensor 2

An extensive experimental program has been carried out to evaluate the sound speed in the two phase mixture in the compressor discharge pipe. The compressor has run with different rotation speeds, pressures, temperatures, oil flows and capacity slide valve positions. The sound speed in the discharge flow has been evaluated from the measurements and compared with that of dry ammonia gas. The last has been calculated for the given pressure and temperature with Refrigerant Calculator [6]. Figure 4 gives a comparison between the sound speed obtained from the measurements and that calculated for the dry NH_3. The results have been obtained with the compressor rotation speed ranging from 2000 to 3600 rpm and with the capacity slide valve position varying between 0 and 100%.

The results indicate that the influence of the oil on the sound speed in screw compressor discharge flow is rather weak. The sound speed obtained from the measurements deviates from that in the dry gas less than on 10%.

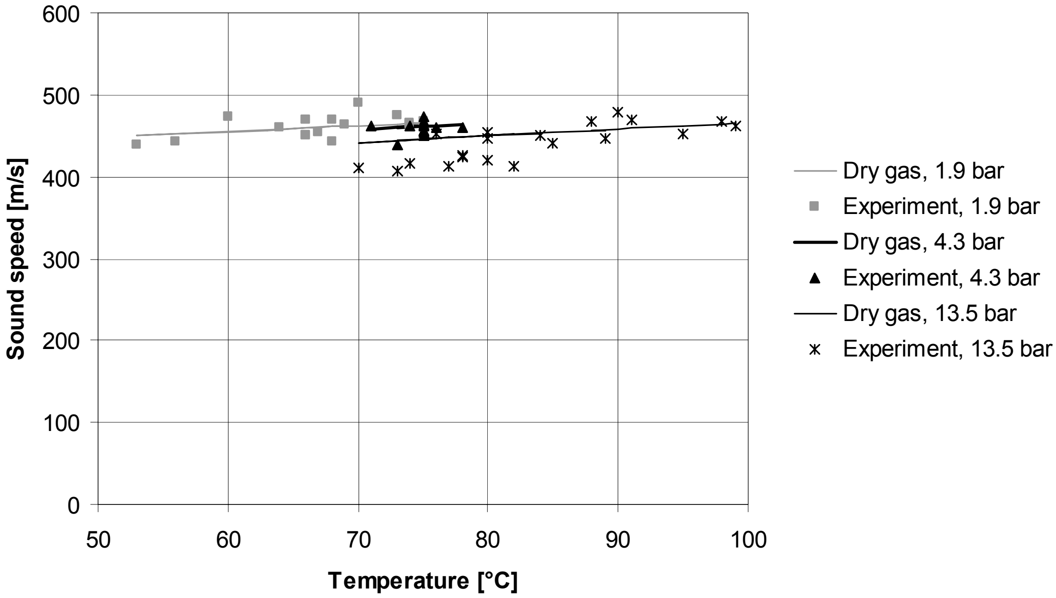


Fig. 4: Sound speed as a function of the discharge pressure and temperature

The amplitudes of the incident, reflected and resultant waves at two different rotation speeds, 2400 and 3000 rpm, are given in Fig. 5. Other compressor operation parameters are similar as in the example in Fig 2. The coordinate axis starts at the compressor flange and is directed to the separator. It can be seen that the amplitudes of the incident and reflected waves are practically constant and the amplitude of the resultant wave varies with the coordinate. The position of the pressure node (the point on a standing wave with the minimal amplitude) depends on the rotation speed of the compressor, i.e. on the frequency.

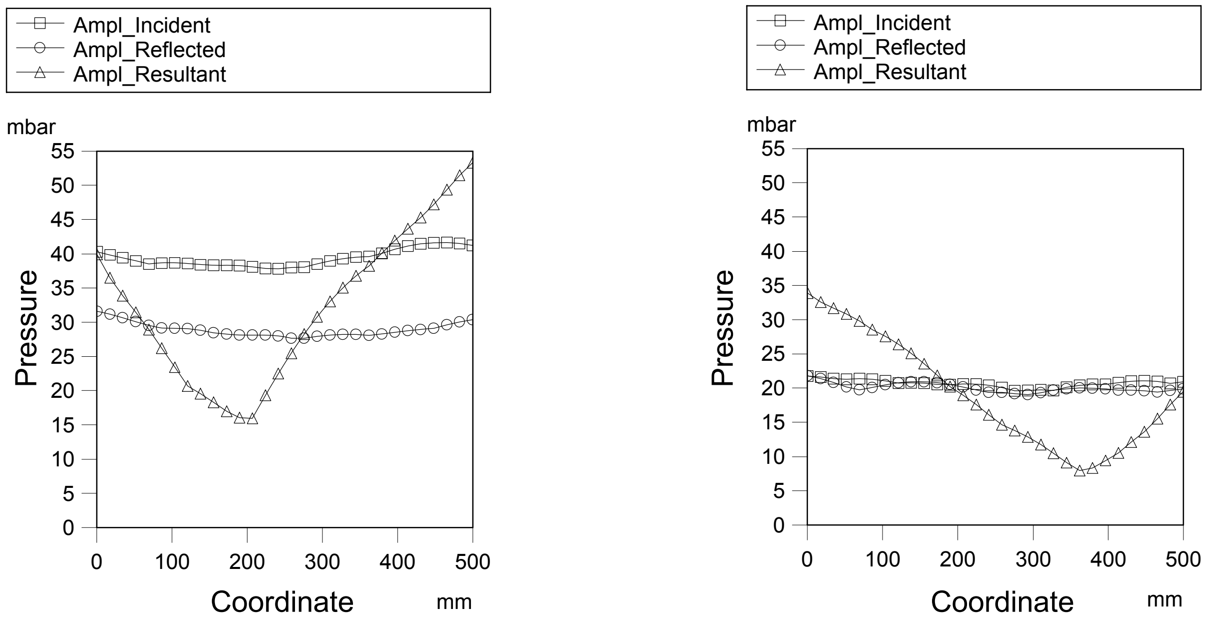


Fig. 5: Amplitude of the incident, reflected and resultant waves

Left: compressor speed 2400 rpm Right: compressor speed 3000 rpm

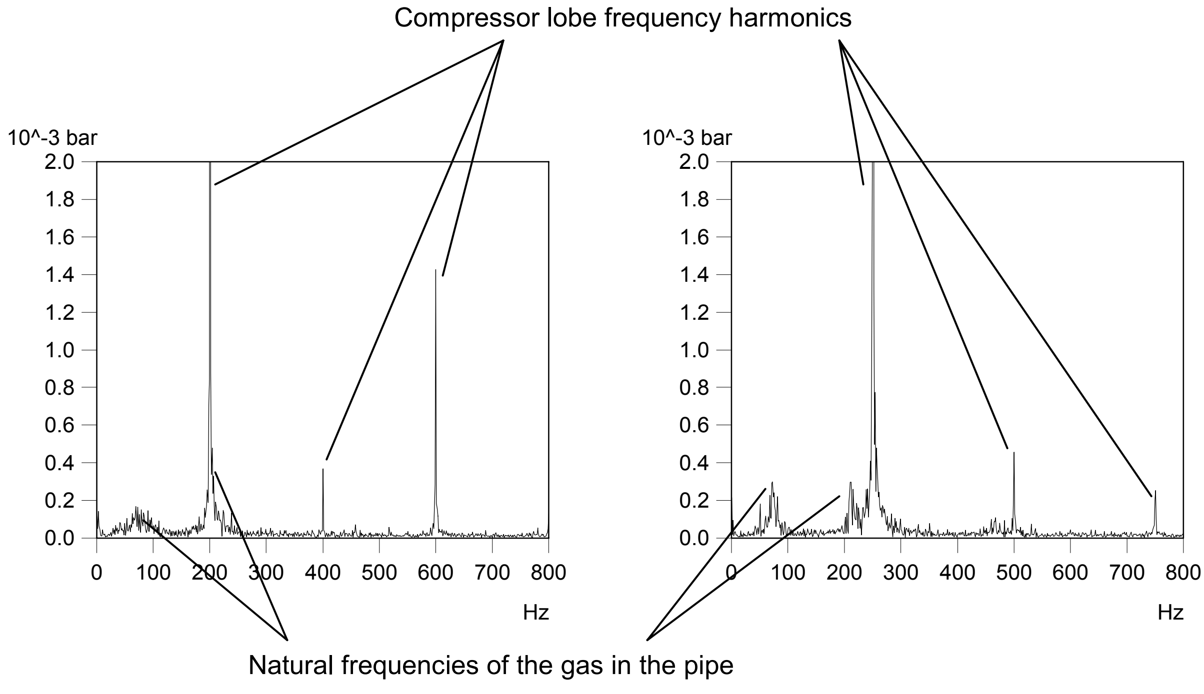


Fig. 6: Spectrum of the resultant wave at coordinate x=0

Horizontal axis – frequency; vertical axis – half amplitude

Left: compressor speed 2400 rpm; Right: compressor speed 3000 rpm

The spectrum of the resultant wave at the compressor discharge flange is illustrated in Fig. 6. There are two kinds of the amplitude peaks in the spectrum: the peaks from the pulsations excited by the compressor and the peaks form the natural pulsations of the gas in the discharge pipe. The frequencies of the compressor stimulated pulsations depend on the rotation speed. They can be calculated with the left hand side of Eq. (3). For the rotation speed of 2400 rpm (40 s^{-1}) and the male rotor lobe number z=5 the exiting frequencies are 200, 400, 600, … Hz. For 3000 rpm they are 250, 500, 750, … Hz. The natural frequencies in the discharge pipe according to Fig. 6 are at 70 and 210 Hz. They do not depend on the compressor speed. At 2400 rpm the compressor lobe frequency and the second natural frequency are close to each other causing the resonance. This explains the fact that in spite of the lower rotation speed the amplitudes of the incident, reflected and resultant waves at 2400 rpm are larger than at 3000 rpm (Fig. 5). The natural frequencies can be predicted with Eq. (2). For the length of 1.575 m and the sound speed of the dry ammonia gas of 460 m/s the first two frequencies are 73 and 219 Hz. This is in a good agreement with Fig. 6 and confirms that the sound speed value for the dry gas can be used in computations of the natural frequencies for the discharge piping of oil-injected screw compressors.

4. Elimination of the resonance in field applications

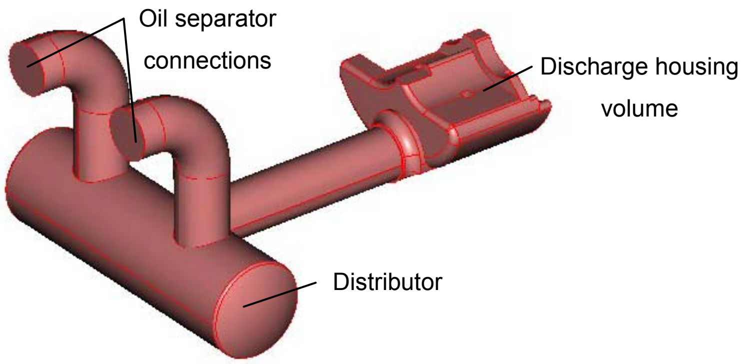
If the sound speed is known, the natural frequencies can be found by solving for the eigenvalues associated with the wave differential equation with corresponding boundary conditions. Under the assumptions of a constant density (small pressure oscillations) and absence of damping the wave equation can be written as

$$\frac{\partial^2 p}{\partial t^2} - c^2 \nabla^2 p = 0 . \qquad (18)$$

The associated eigenvalue equation is

$$\lambda p + c^2 \nabla^2 p = 0 . \qquad (19)$$

The eigenvalue λ is related to the natural frequency f as $\lambda = (2\pi f)^2$. For simple geometries an analytical solution for the natural frequencies can be found, for example, Eq. (2). For more complicated cases Eq. (19) has to be solved numerically. In this way the piping of real compressor packages can be checked on resonance frequencies at the design stage. Also in the case of resonance problem with an existing package the piping modifications can be pre-calculated. As an example, Fig 7 illustrates the geometry of the discharge piping in a screw compressor package. The natural frequencies were computed for this geometry using commercial FEM software. The applied boundary conditions were the zero pressure oscillations at the oil separator connections and the zero velocity (zero pressure gradient) on the walls. The spectrum of the calculated natural frequencies is shown in the left column in Fig. 7. If the compressor operates with 3000 rpm, the second harmonic of the lobe frequency (500 Hz) gets close to the natural frequency of 502 Hz. To avoid the resonance the length of the distributor and the distance between the compressor and distributor were reduced. The spectrum of the natural frequencies for the modified geometry is shown in the right column in Fig. 7. There is no match anymore between the exciting and natural frequencies.



Original piping	Modified piping
50	56
97	118
316	400
502	441
584	592
615	639
733	715
905	893

Fig. 7: Discharge piping geometry with computed natural frequencies, Hz

5. Conclusions

The sound speed in the discharge flow of an oil-injected screw compressor was analytically evaluated based on the spectrum of the pressure pulsations measured in three equally spaced points. At the same time the solution delivered the pressure field of the waves travelling in both directions between the compressor and oil separator. It was confirmed that due to the wave reflections the amplitude of the resultant wave is position dependent.

It was found that the influence of the oil on the sound speed in the screw compressor discharge flow is rather weak. The sound speed obtained from the measurements deviated from that in the dry gas less than on 10%. It could be stated that the sound speed value for the dry gas can be used in computations of the natural frequencies for the discharge piping of oil-injected screw compressors. The results were used to solve the resonance problem related to a real screw compressor package.

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