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ABSTRACT

In the paper the criteria which should be satisfied by a reciprocating compressor inlet piping system have been pressented. The inlet pipeline, proposed by the author, with a proper pressure pulsation and noise damper satisfying these criteria has been discussed. Dependences making it possible to calculate the relative filling factor which is a quantitative measure of the compressor dynamic supercharging have been derived. A method of calculating gas mass natural vibration frequency in the inlet piping system has been discussed, which makes it possible to determine proper geometric dimensions of the piping system. The results of experimental measurements of the effect of the proposed piping system on the compressor work and the level of noise emitted by its inlet to the environment have been presented.

NOMENCLATURE

- a - speed of sound
- A_r - pipeline cross-section field
- A_t - piston field
- C - piston volumetric speed
- D - pipeline inside diameter
- i - number of cylinders
- l - lenght
- m - ordinal number of harmonic component
- p - variable pressure of gas
- ρ - gas variable density
- p_o, ρ_o - mean gas pressure and density, respectively
- r - crankthrow
- V - volume
- w - gas variable velocity
- x - displacement coordinate
- Z - mechanical impedance
- $\varkappa$ - adiabate exponent
- τ - time

INTRODUCTION

The delivery of a reciprocating compressor can be increased by dynamic supercharging, i.e. by using wave processes in its inlet piping system. These phenomena affect the compressor delivery most during the natural vibration frequency of gas mass in the inlet piping system or its part directly adjacent to the machine with the forcing frequency or its higher harmonic components. Such a case is called resonance supercharging. Resonance phenomena in inlet piping systems are accompanied by an increase of the total level of the noise emitted by the compressor to the environment, which is due to the increase of the aerodynamic noise level. This requires pressure pulsation and noise dampers of higher damping efficiency to be installed in reciprocating compressor inlet piping systems. Such dampers sometimes cause considerable resistance of flow, which leads to a decrease of compressor delivery and increased power demand [1], [2], [3], [4], [6], [7].

RELATIVE FILLING FACTOR

The changes of the compressor delivery due to dynamic supercharging are referred to the delivery of a compressor working without inlet piping system, i.e. one sucking in the gas directly from a tank of an infinitely large volume, in which the pressure is constant. That is why the concept of a relative filling factor λ_w is introduced, defined as a ratio between the gas mass sucked into a cylinder with the consideration of pressure pulsation in the inlet piping system m'_g and the gas mass sucked in the conditions of constant pressure m_g.

$$\lambda_w = \frac{m'_g}{m_g} \tag{1}$$

In order to derive the dependence making it possible to define the value of the relative filling factor it is necessary to solve differential equations describing the gas pulsatory flow in the inlet piping system with corresponding boundary conditions. The differential equations describing the gas pulsatory flow in a straight section of the pipeline of constant cross-section are derived with numerous simplifying assumptions on the basis of equation of motion, equation of continuity and equation of state [4]. The equation of state covers the gas specific volume changes under the influence of pressure waves. We assume the adiabatic nature of these changes. For subsonic flow $w \ll a$, neglecting friction and assuming minor density changes due to pressure pulsation, after adequate transformations we obtain the following equations:

$$\left.\begin{aligned} -\frac{\partial p}{\partial x} &= \varrho_o \frac{\partial w}{\partial \tau} \\ -\frac{\partial w}{\partial x} &= \frac{1}{\varrho_o a^2} \frac{\partial p}{\partial \tau} \end{aligned}\right\} \tag{2}$$

where

$$a = \sqrt{\varkappa R T} \tag{3}$$

The boundary conditions for an inlet piping system are determined by calculating its displacement coordinate from its open end. For this end the variable pressure is assumed to be zero.

$$x = 0, \quad p = 0 \tag{4}$$

The boundary condition from the side of the compressor is given in the form of a dependence of piston volumetric speed $C = dV/d\tau$ on the crank angle or time.

$$\left.\begin{aligned} & x = 1 \\ & \frac{dC}{d\tau} = -\frac{V_o}{a^2 \rho_o}\left(\frac{d^2 p}{d\tau^2} + \frac{\varkappa C_o}{V_o}\frac{dp}{d\tau} + \frac{a^2 A_r}{V_o}\left.\frac{\partial p}{\partial x}\right|_{x=1}\right)\end{aligned}\right\} \tag{5}$$

Assuming farther that: the suction valve operates perfectly, the gas is sucked during crankshaft half-turn, the cylinder dimensions are small in relation to pressure pulsation wave length and the connecting rod is infinitely long, after numerous transformations we receive the dependences defining the relative filling factor.

1. For a single-acting single-cylinder compressor:

$$\lambda_w = 1 - \frac{1}{\varkappa}\sum_{m=2}^{\infty}\frac{\varkappa - m^2}{(m^2-1)^2}\cos^4\frac{m\pi}{2}\cos^2\varphi_m + $$

$$- \frac{\pi^2}{16}\left[(\varkappa - 1)\sin\varphi_1\cos\varphi_1 + \frac{8}{\pi}\cos^2\varphi_1\right] \tag{6}$$

2. For a compressor with two single-acting cylinders with antiphase of 180° or a single double-acting cylinder:

$$\lambda_w = 1 - 2\sum_{m=2}^{\infty}\left(1 - \frac{m^2}{\varkappa}\right)\frac{\cos^2\varphi_m}{(m^2-1)^2}\cos^4\frac{m\pi}{2} \tag{7}$$

The value φ_m occurring in dependences (6) and (7) is calculated (for $m = 1, 2, 3, \ldots$) from the dependence:

$$\mathrm{tg}\,\varphi_m = \frac{m^2\omega_k^2 - \omega_o^2}{2\delta m \omega_k} \tag{8}$$

in which

$$2\delta = \varkappa\frac{C_o}{V_o} \tag{9}$$

and

$$V_o = \frac{A_t r}{2} i \tag{10}$$

$$C_o = \frac{\omega_k A_t r}{\pi} i \tag{11}$$

During resonance the natural vibration frequency of gas mass in the inlet piping system ω_o is equal to the forcing frequency or its higher harmonic component $\omega_o = m\omega_k$. The forcing frequency is determined by the compressor crankshaft rotational speed. In such case dependences (6) and (7) have the following forms.

- For a single cylinder:

$$\left.\begin{aligned}\lambda_w &= 1 - \frac{\pi}{2\varkappa} && (\text{for } m = 1)\\ \lambda_w &= 1 - \frac{1}{\varkappa}\ \frac{\varkappa - m^2}{(m^2-1)^2}\ \cos^4\frac{m\pi}{2} && (\text{for } m = 2,3,\dots)\end{aligned}\right\} \quad (12)$$

- For two cylinders:

$$\lambda_w = 1 - \frac{2}{(m^2-1)^2}\left(1 - \frac{m^2}{\varkappa}\right)\cos^4\frac{m\pi}{2} \qquad (13)$$

$$(\text{for } m = 2,3,\dots)$$

Dependences (12) and (13) can be successfully used for determining theoretical possibilities of compressor dynamic supercharging. They make it possible to calculate theoretically largest increase or decrease of compressor delivery during the resonance of natural vibration frequency of gas mass in the inlet pipeline with the forcing frequency or its higher harmonic components. The effect of dynamic supercharging depends on the type of gas under compression, since the value of the relative filling factor λ_w depends on the exponent of the adiabate $\varkappa$ whose value may change over a broad range. For a popular refrigerating medium Freon 22 (R22) $\varkappa = 1,14$, for air $\varkappa = 1,40$ and single monoatomic gases $\varkappa = 1,67$. By means of dependences (12) and (13) the values of the relative filling factor have been calculated for two values of adiabate exponent as follows: 1,14 and 1,40. The results of these calculations have been shown in Fig. 1 in the form of dependence $\lambda_w = f(m)$.

CALCULATION OF NATURAL VIBRATION FREQUENCY OF GAS MASS IN A PIPING SYSTEM

In order to calculate natural vibration frequency of gas mass in a piping system it is necessary to solve partial differentialy equations (2) with boundary conditions adequate to the piping system in question. Eliminating the velocity from the first equation, and the pressure from the second one, after a few simple transformations equations (2) will take the form of wave equations:

$$\left.\begin{aligned}\frac{\partial^2 p}{\partial \tau} &= a^2\,\frac{\partial^2 p}{\partial x^2}\\ \frac{\partial^2 w}{\partial \tau^2} &= a^2\,\frac{\partial^2 w}{\partial x^2}\end{aligned}\right\} \qquad (14)$$

The boundary conditions at the beginning and end of an actual piping system are given in the form of mechanical impedance. For a completely acoustically closed end impedance $Z = \infty$. The condition of such end is satisfied by for instance connecting the pipeline to the compressor if we disregard the cylinder and valve chamber volume or a strongly throttled valve within the pipeline. For a completely acoustically open end impedance $Z = 0$. Such an end is found in case of, for instance, a pipeline end through which the gas is sucked in or forced out to the atmosphere or a tank of large volume, the so-called acoustically infinite volume. The mechanical impedance of an infinitely long pipeline of constant cross-section is determined by dependence $Z = A_r \varrho_0 a$ and is called pipeline wave impedance.

The mechanical impedance of a closed tank of volume V, placed in the pipeline, is calculated disregarding the active component from dependence:

$$Z = \frac{\varrho_0 a^2 A_r^2}{j \omega V} \tag{15}$$

The effect of certain fittings elements (valves, dampers, etc.) on pulsatory flow can be modelled by a flat throttling plate placed across the pipeline of an equivalent flow area. The mechanical impedance of such a plate is calculated disregarding the active component from the dependence:

$$Z = j \omega \frac{\varrho_0 A_r^2}{d F\left(\frac{d}{D}\right)} \tag{16}$$

The value of Fock function $F\left(\frac{d}{D}\right)$ can be calculated with satisfactory approximation from the following empirical dependence:

$$F\left(\frac{d}{D}\right) \approx \left(1 - 1{,}47 \frac{d}{D} + 0{,}47 \frac{d^3}{D^3}\right)^{-1} \tag{17}$$

A tank of a suitable volume mounted into a piping system divides it into two separate vibrating systems. In such case calculations should be done for each part separately. The condition of pipeline division into two parts is fulfilled when:

$$\Delta\omega = \left|\frac{\omega_{\infty} - \omega_v}{\omega_{\infty}}\right| \leqslant 0{,}04 \tag{18}$$

where:

ω_{∞} - natural vibration frequency of gas mass in the piping system between compressor and dividing volume, calculated without this volume, i.e. with the assumption that its volume $V = \infty$ (completely open end),

ω_v - natural vibration frequency of gas mass in the piping system between compressor and dividing volume, calculated with this volume. The piping system behind the tank of the dividing volume in both cases in disregarded [4].

INLET PIPING SYSTEM

A reciprocating compressor inlet piping system is usually composed of several sections of the pipeline of a cross-section of properly selected field, one or more pressure pulsation and noise dampers and a filter purifying the sucked gas. Such a system should ensure the compressor dynamic supercharging and prevent exceeding the level of noise emitted to the environment beyond the allowable one. This leads to the fact that the resonance should be induced in the section of the inlet piping between the compressor and the pressure pulsation and noise damper. The damper, on the other hand, should ensure efficient damping of aerodynamic and mechanical noises emitted by the inlet piping system to the environment. Inducing the resonance phenomena in the section between the compressor and damper should at least eliminate the delivery loss caused by the resistance of flow through the filter and the pressure pulsation and noise damper.

On the basis of the analysis of compressor dynamic supercharging process done do far we can formulate the requirements that should be met by the inlet piping system of a reciprocating compressor.

1. The geometric dimensions of the inlet piping system and place of mounting the damper should be chosen in such a way that during the normal compressor work the natural vibration frequency of the gas mass in it was near the resonance with the second harmonic component of the forcing frequency.

2. The inlet piping system should have low hydraulic resistance so that it reduced the effect of supercharging to the smallest possible degree. This consequently means that the pressure pulsation and noise dampers installed in the inlet piping system should have low hydraulic resistance.

3. The inlet piping system must ensure efficient damping of aerodynamic and mechanical noises emitted to the environment. This requirement means that dampers installed in inlet piping must be highly efficient in damping over a wide range of frequencies.

As can be seen the requirements for reciprocating compressor inlet piping system are often contradictory so an optimal system must be a compromising solution.

In Fig. 2 a diagram of the proposed inlet piping system with pressure pulsation and noise damper, well meeting the above requirements, has been shown. The damper used in it is a solution patented by the author [5]. The inlet piping system shown in Fig. 2 is composed of a two-chamber damper 1 and inlet pipeline 2 with a resonance cavity 10 connected with the pipeline with a ring gap or ports. In the damper there are two chambers: pre-damping chamber 3 and chamber 2 with a tubular damping element 5 located off-centre, the chambers being separated by a plate 4 located crosswise. The entry of damping element 6 is in the shape of a nozzle widening towards pre-damping chamber 3, the wall of the damping element is perforated and the end closed with a scattering cone 7. The inlet connector 8 is driven through the bottom to the pre-damping chamber 3, off-centre and opposite to nozzle 6.

EXPERIMENTAL MEASUREMENTS OF THE PROPOSED INLET PIPING SYSTEM

The aim of the measurements was to determine the effect of the proposed inlet piping system on the compressor work and the level of noise emitted by its inlet to the environment. The measurements were carried out on a small double acting reciprocating compressor of air. A compressor of special design with slide timing gear was used in order to eliminate supercharging loss caused by the effect of pressure pulsation resonance on the automatic inlet valves operating. So the effect of supercharging was reduced only by the resistance of flow through the inlet piping system.

The measurements consisted in comparing the results obtained during compressor work without inlet piping system and one with the system, at different rotational speeds and different outlet pressure. The compressor delivery values $\dot{V}_d$ on the basis of which relative filling factors λ_w were calculated, power demand on the compressor crankshaft N_e and unitary energy demand for compression q_e were compared. The unitary energy demand was defined as:

$$q_e = \frac{N_e}{\dot{V}_d} \tag{19}$$

The tested inlet piping was designed and dimensioned in such a way that the natural vibrations frequency resonance with the second harmonic component of the forcing frequency occurred at the compressor shaft rotational speed of ca. 3250 [rpm]. The tests were done at three different compressor rotational speeds: 3245, 3015 and 2520 [rpm] and four different outlet pressures of absolute values: 0,70 , 0,80 , 0,85 and 0,90 [MPa].

The results of the measurements have been shown in Fig. 3 as a diagram of dependence $\lambda_w = f(p_w)$ for particular speeds. Fig. 4 shows diagram of dependences $N_e = f(p_w)$, $\dot{V}_d = f(p_w)$ and $q_e = f(p_w)$ for one speed $\dot{n} = 3015$ [rpm].

As can be seen from the diagrams of dependence $\lambda_w = f(p_w)$ presented in Fig. 3, the applications of the proposed inlet piping system increases the compressor delivery by about 12 % in case of the resonance of natural vibration frequency with the second harmonic component of the forcing frequency. The farther the resonance, the smaller the effect of supercharging. In all the tested cases the application of the inlet piping system increased the compressor delivery and compressing power demand, which was illustrated by diagrams in Fig. 4. The changes of compressor delivery and power demand due to the application of the inlet piping system always resulted in a minor decrease of unitary energy demand q_e.

The use of the inlet piping system reduced the level of noise emitted to the environment by 24 ÷ 26 [dB]. The intensity of noise emitted by the compressor inlet was always lower than 81 [dB A] when the inlet piping was used.

CONCLUSIONS

The presented and tested inlet piping system with pressure pulsation and noise damper satisfactorily meets the requirements for inlet piping system. It is an example of practical realization of inlet piping system by means of which the effect of dynamic supercharging eliminates loss due to an increase of inlet flow resistance. The changes of compressor delivery and power demand caused by the application of the inlet piping system result in a minor decrease of unitary compressing energy demand. The damper, which is an element of the inlet piping system, ensures sufficient damping of noise emitted by this system to the environment.

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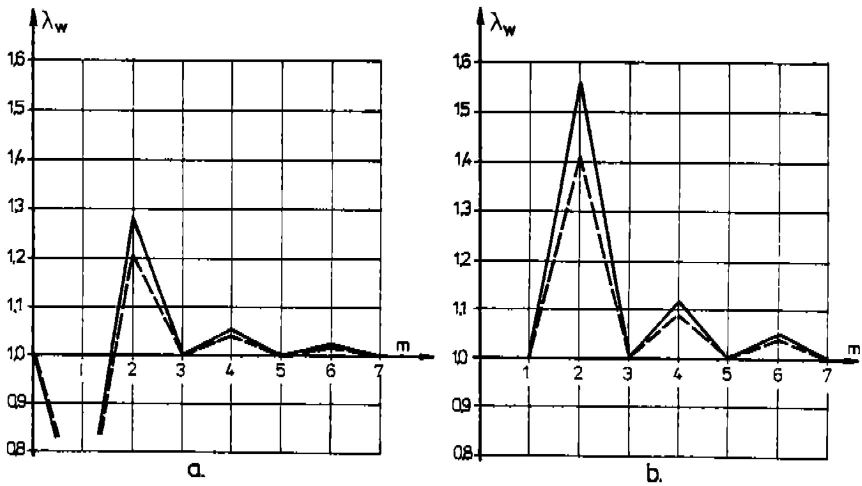

Fig. 1. Theoretical values of relative filling factor calculated disregarding the non-resonating harmonic components:
a - for single acting cylinder, b - for double acting cylinder; ——— - for $\varkappa = 1,14$
— — — - for $\varkappa = 1,40$

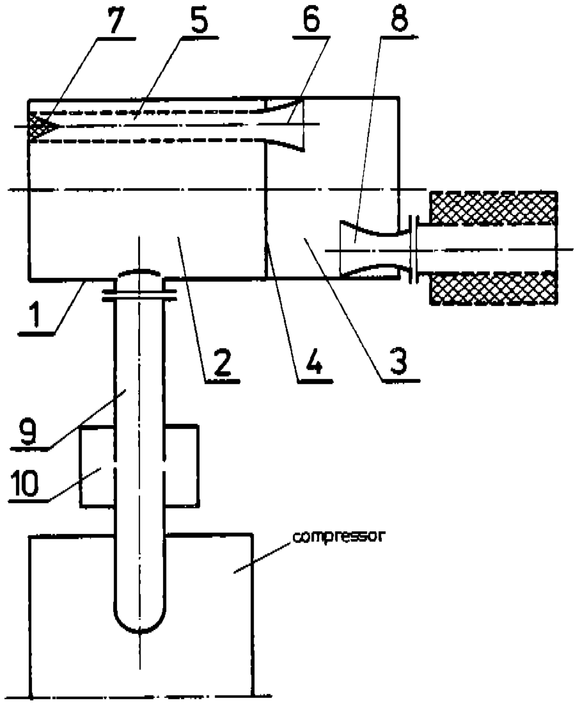

Fig. 2. Diagram of the proposed inlet piping system.

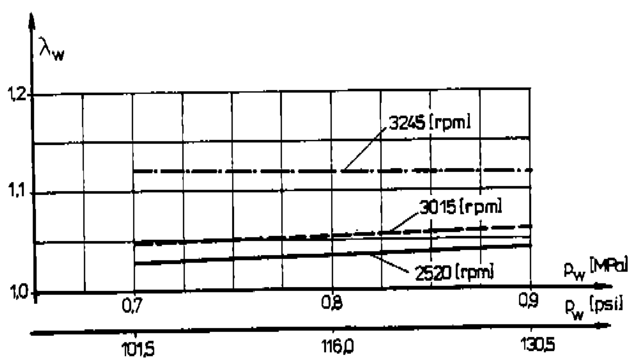

Fig. 3. Diagrams of dependence $\lambda_w = f(p_w)$ for various rotational speeds of compressor.

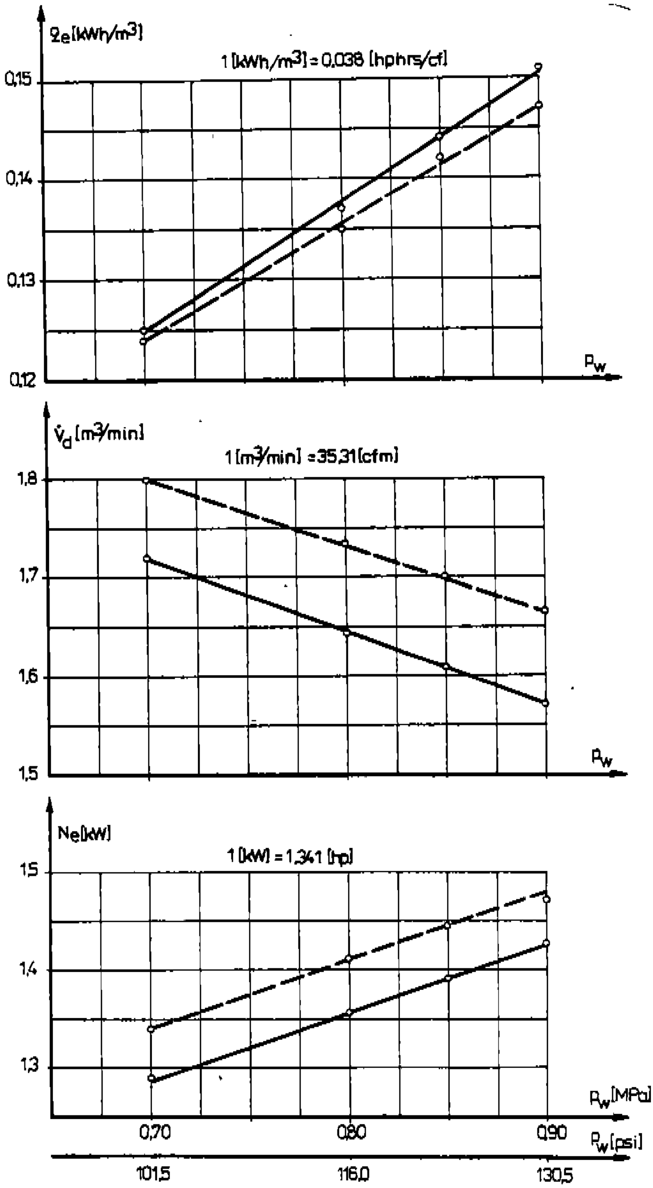

Fig. 4. Diagrams of dependences $N_e = f(p_w)$, $\dot{V}_d = f(p_w)$ and $q_e = f(p_w)$ for rotational speed $\dot{n} = 3015$ (rpm);
——— - without inlet piping.
– – – - with inlet piping.