Implementing the principles of operating processes schematization and of performance loss distribution when designing long-stroke reciprocating compressor stages

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Abstract. To compress gases up to 10.0 MPa and higher, diaphragm and multistage reciprocating compressor units are used. They are heavy, massive and complicated. The results of analyzing laboratory examples of slow-speed long-stroke stages of reciprocating compressors while gas pressurizing up to such pressure have shown the possibility of solving the problems mentioned. Such stages differ from others, that is why we cannot use common design methods in this case. The performance design method for slow-speed long-stroke compressor stages with a discharge pressure of up to 10.0 MPa is based on the well-known principles of schematization of operating processes and distribution of performance losses. With regard to the experimental studies results (indicator and temperature diagrams, integral characteristics) we have found: the volumetric efficiency, which characterizes the ratio between real and theoretical compressor performance, its main components: the volume coefficient, the pressure coefficient, the temperature coefficient, and the leakage coefficient. Dependences are obtained for calculating these coefficients, which differ significantly from the known dependences for calculating high-speed compressor stages. The analysis of the influence of various factors on the performance losses of low-capacity long-stroke compressor stages of medium and high pressure revealed that the gaps in the working chamber have the maximum impact on the performance losses, the leakage coefficient significantly depending on the discharge pressure change. The volume coefficient has much less impact due to the increased stroke-to-diameter ratio of the piston, which provides a small amount of relative dead volume. The developed method can be used for preliminary calculation of compressor stages of the considered type.

1. Introduction

The study results of laboratory samples of reciprocating compressors slow-speed long-stroke stages, performed at the Department of Refrigeration and Compressor Equipment and Technology of the Omsk State Technical University, proved the possibility of compressing air in one stage from atmospheric pressure to pressures of at least 10.0 MPa at a temperature of the injected gas that does not exceed the regulatory requirements [1-3]. In accordance with their specific design and operational differences, such stages can be called long-stroke (since parameter Sp/Dc > 10) low-speed (since the working cycle time τ>1s) [2, 4]. These parameters play a key role in the organization of the working process, providing a mode of operation close to isothermal. It should be noted that in the high-pressure stages of multi-stage compressors, the parameter Sp/Dc can also be close to 10. However, in such stages, the ratio of the discharge pressure to the suction pressure usually does not exceed 3...5, while the working cycle time is 0.2...0.02 s [5, 6].
The expected advantages of single-stage compressors of this type over multi-stage piston and diaphragm compressors of a similar purpose are: reducing the number of compression stages and interstage components (heat exchangers, pipelines, oil and condensate separators, filters, safety valves, etc.), increasing the degree of unification of the unit, as well as reducing the weight and overall dimensions [5]. In addition, such units do not have normal forces in the cylinder-piston group, which positively affects their performance characteristics [5].

When designing stages of reciprocating compressors, their main dimensions and parameters are often calculated based on the method of determining performance losses, including the calculation of the volumetric efficiency (Frenkel M. I., Plastinin P. I., Minta M., Barclay M., Davies R., John F. McLaren, Corberan, J. M., Hafner, J., Gaspersic, B., Luszczycyki, M., Prakash, R., and others) [6-9]. One of the options for implementing this method is to determine the volumetric efficiency according to the following dependence [8]:

\[ \lambda = \lambda_0 \cdot \lambda_p \cdot \lambda_T \cdot \lambda_h \cdot \lambda_L, \]

where \( \lambda_0 \) is the volume coefficient, \( \lambda_p \) is the pressure coefficient, \( \lambda_T \) is the temperature coefficient, \( \lambda_h \) is the humidity coefficient and \( \lambda_L \) is the leakage coefficient.

At the same time, each of the coefficients included in equation (1) is determined according to the known ratios or recommendations that are valid for existing high-speed compressor stages [8]. It can be assumed with sufficient probability that the significant differences between the long-stroke slow-speed compressor stages considered in this work and the high-speed analogues do not allow us to correctly calculate the volumetric efficiency of such a stage according to the known method. Therefore, the task of this study is to obtain dependencies and recommendations for determining each of the above-mentioned components in formula (1).

2. Object of the study

The object of the study is the stage of a low-capacity compressor (the described volume is about 2.0...3.0 m³/h) long-stroke slow-speed piston compressor without feeding lubricant to the flow part. The design of the stage is described in detail in [10, 11]; the lip piston seals are made of self-lubricating material; the automatic suction and discharge valves are of the poppet type. The experimental stage has the following main dimensions and parameters: cylinder diameter is 0.05 m, piston stroke is 0.5 m; cycle time is 2...3 s; relative dead volume is 0.026 %. External cylinder cooling is water; water temperature is 290 K; compressed gas is air; intake gas temperature is 290 K; suction pressure is 0.1 MPa; discharge pressure is 0.5…12.0 MPa.

3. Experimental research methodology

The scheme of the experimental stage is shown in (Fig. 1). Piston 1 is driven through rod 2 from the rod of the hydraulic cylinder, which in turn is driven from the hydraulic accumulator station. The suction and discharge valves are located in the valve plate and are not shown conditionally. An experimental unit with a linear (hydraulic) drive has been developed for experimental studies [12]. Experimental studies of the working processes of the stage include measurement of the instantaneous parameters of the state of the working gas in the cylinder and its integral characteristics. The pressure sensor 9 and the temperature sensor 8 allow us to measure the corresponding instantaneous parameters in the working compression chamber. The flow sensor 12 allows us to determine the actual compressor performance of the experimental stage. To measure the rapidly changing gas pressure in the working chamber of the stage, silicon pressure sensors 9 (Fig. 1) of the D16 type were used [12-14]. The temperature measurement was performed by thermistors 8, 11 (Fig.1) with a negative temperature coefficient of resistance [15-17].

The actual gas flow rate at the stage discharge was determined by the flow sensor 12 (Fig.1) of the AWM720P1 type. Data from the temperature, the pressure, and the flow sensors is transmitted to the digital oscilloscope 14 via amplifier 13 and personal computer 15.
Let us estimate the error of measuring temperature, pressure, and flow. The error of the pressure sensor is determined by the formula \[ \delta_p = \sqrt{\delta_p^2 + \delta_{PG}^2 + \delta_o^2} \] (2)

where \( \delta_p \) is the relative error of the pressure sensor, \( \% \); \( \delta_{PG} \) is the relative error of the sample pressure gauge, \( \% \); \( \delta_o \) is the relative error of the oscilloscope, \( \% \).

The relative error of the pressure sensor according to the passport is \( \delta_p = 1.4 \% \). The relative error of the sample pressure gauge is \( \delta_{PG} = 1.5 \% \). The relative error for the oscilloscope is determined according to the passport as \( \delta_o = 3 \% \). Then the total error of the pressure sensor is equal to:

\[ \delta_p = \sqrt{1.4^2 + 1.5^2 + 3.0^2} = 3.63 \% \]

The air flow rate is measured by the AWM720P1 thermal flow sensor with an intrinsic accuracy \( \delta_{VTD} = 0.3 \% \). Then the total error of the flow measurement is determined by the formula:

\[ \delta_v = \sqrt{\delta_v^2 + \delta_{VTD}^2} = \sqrt{3.0^2 + 0.3^2} = 3.02 \% \]

To measure the instantaneous air temperature in the working chamber of the stage, sensors based on a bead thermistor of st1-18A type were used [16, 17, 25]. The data from the temperature sensor is transmitted to the digital oscilloscope via an amplifier. We will find the total accuracy of the temperature sensor based on a bead thermistor [19-22]:

\[ \delta_T = \sqrt{\delta_M^2 + \delta_l^2 + \delta_v^2 + \delta_{F^2}} \] (3)

where \( \delta_M \) is the relative accuracy of the oscilloscope, 0.05 \%; \( \delta_l \) is the thermometer accuracy, determined by the accuracy of the device, 0.1 \%; \( \delta_v \) is the voltmeter accuracy, 0.3 \%; \( \delta_F \) is the accuracy taking into account non-linear correlation between voltage and temperature, 1.5\%.

Thus, the accuracy in measuring the instantaneous air temperature in the working chamber of the experimental stage will be:

\[ \delta_T = \sqrt{0.05^2 + 0.1^2 + 0.3^2 + 1.5^2} = 1.53 \% \]

The bead design of the temperature sensors ensures a minimum dead volume in the working chamber of the stage, which is crucial for high values of the ratio of the discharge pressure to the suction pressure. Note that this measurement method does not allow temperature measurements in high-speed stages due to the significant inertia of the bead sensing elements; however, in low-speed stages of the type under consideration, the working cycle time of which is one or two orders of magnitude longer, the use of bead thermistors is well-justified.
4. Method for estimation the components of the volumetric efficiency

It is known that the volumetric efficiency can be represented as [24]:
\[
\lambda = \lambda_{ind} \cdot \lambda_{hu}
\]  (4)

where \( \lambda_{ind} = \lambda_0 \cdot \lambda_p \) is the coefficient that takes into account the indicator compressor performance loss; \( \lambda_{hu} = \lambda_T \cdot \lambda_L \cdot \lambda_h \) is the coefficient that takes into account the so-called “hidden” compressor performance loss.

We will use the results of our experimental studies to determine the components of the volumetric efficiency listed above

5. Estimation of indicator compressor performance loss

To calculate the volume coefficient, we apply the existing calculation method [23]:
\[
\lambda_0 = 1 - a_m \left( \frac{P_{dis}}{P_{suc}} \right)^{m/m} - 1,
\]  (5)

where \( P_{suc} \) is the suction pressure, Pa; \( P_{dis} \) is the discharge pressure, Pa; \( m \) is the “polytropy index of the boundary parameters”; \( a_m \) is the value of the relative dead volume.

In formula (5), all values except \( m \) are set according to the design and operating mode of the compressor stage. Let us find the polytropy index of the final parameters \( m \) using the obtained experimental indicator diagrams. For this purpose, let us apply the well-known principle of schematization of the process of reverse expansion by replacing the indicator diagram of a valid reverse process of expansion occurring with varying polytropic exponent, on a curve with a relatively constant polytropic inverse exponent \( m \), with both curves start and end points of the expansion process must be the same [23]. In this case, the schematized process allows us to quantify, with an acceptable margin of error, the actual performance losses caused by the expansion of the gas from the dead volume (Fig. 2).

![Figure 2. An example of schematization of the final parameters by polytropic at P_{suc} = 0.1 MPa; P_{dis} = 10.0 MPa; Dc = 0.05m; Sp = 0.5m; \tau = 3 s](image)

According to the well-known recommendations, the formula (6) can be used to calculate the “polytropy index of the final parameters” [22, 23]:
\[
m = 1 + A(k-1),
\]  (6)

where \( A \) is the empirical coefficient.

The analysis of the obtained experimental indicator diagrams (Fig. 3...6, Table 1) allows for the considered range of operating modes of the air slow-speed long-stroke stage to preliminarily accept the following recommendations: for \( P_{dis} \leq 5.0 \) MPa – \( A = 0.125 \) (when \( m \approx 1.05 \)); for \( 5.0 \) MPa < \( P_{dis} \leq 12.0 \) MPa – \( A = 0.25 \) (when \( m \approx 1.1 \)). The volume coefficient in this case varies from 0.98 at a pressure of 5.0 MPa to 0.95 at a pressure of 12.0 MPa.
Note that for high-speed air stages, the value of $m$ at $P_{\text{dis}} < 3.0$ MPa lies in the range of 1.2...1.4, at $P_{\text{dis}} > 3.0$ MPa is assumed to be 1.4 [26].

Figure 3. Experimental indicator diagram at $P_{\text{dis}}=5.0$ MPa, $\tau=3$ s

Figure 4. Experimental indicator diagram at $P_{\text{dis}}=10.0$ MPa, $\tau=3$ s

Figure 5. Experimental indicator diagram at $P_{\text{dis}}=12.0$ MPa, $\tau=3$ s

Figure 6. Experimental indicator diagram at $P_{\text{dis}}=12.0$ MPa, $\tau=2$ s
Table 1. Indicator volumetric efficiency and its components

| Experiment No. | Pdis, MPa | Cycle time, s | λ_{ind} | λ₀ | λᵣ |
|---------------|-----------|--------------|---------|----|----|
| 1             | 5         | 3            | 0.97    | 0.98 | 0.99 |
| 2             | 10        | 3            | 0.96    | 0.97 | 0.985 |
| 3             | 12        | 3            | 0.93    | 0.95 | 0.984 |
| 4             | 12        | 2            | 0.93    | 0.95 | 0.981 |

Throttle compressor performance losses are determined on the indicator diagram (Fig. 2) by the segment 3 – 3’ [23], because during this period of the working process, the volume of the working chamber decreases when the suction valve is open. The values of the pressure coefficient, with an error are in the range of 0.98...0.99.

6. Estimation of hidden compressor performance loss

The temperature coefficient characterizes the reduced mass of the new portion of the intake air that fills the working chamber of the piston during the process of absorption, due to its heated surfaces forming the flow path of a stage. It is defined as the ratio of the air temperature in the standard point of suction to the air temperature in the stage working chamber at the end of the suction process [23]:

$$\lambda_T = \frac{T_{suc2}}{T_{suc3}},$$

(7)

where $T_{suc}$ is the air temperature at the standard suction point, $T_{suc3}$ is the air temperature in the working chamber at the end of the suction process.

This component of the volumetric efficiency should be estimated based on the results of the experimental studies, which resulted in temperature diagrams that allow us to estimate the change in air temperature in the working chamber of the stage during the working cycle of the compressor stage (Fig. 7 – 9). Table 2 shows the results of the obtained experimental data processing.

The experimental data obtained made it possible to determine the gas temperature at the end of the suction process and calculate the heating coefficient for the considered range of the operating parameters using formula (7).

Figure 7. Experimental temperature diagram at a discharge pressure of 3.0 MPa, cycle time is 3 s

Figure 8. Experimental temperature diagram at a discharge pressure of 12.0 MPa, cycle time is 3 s
Figure 9. Experimental temperature diagram at a discharge pressure of 10.0 MPa, cycle time of 2 s

Table 2. Experimental data for calculating of the temperature coefficient

| Experiment No. | $P_{dis}$ MPa | Cycle time, s | $T_{Inc}$ | $T_{Inc,T}$ | $\lambda_T$ |
|----------------|---------------|---------------|-----------|-------------|------------|
| 1              | 0.5           | 3             | 293       | 294.5       | 0.996      |
| 2              | 3             | 3             | 293       | 300         | 0.975      |
| 3              | 6             | 3             | 293       | 307.5       | 0.953      |
| 4              | 10            | 3             | 293       | 321         | 0.91       |
| 5              | 12            | 3             | 293       | 329         | 0.89       |
| 6              | 0.5           | 2             | 293       | 296         | 0.99       |
| 7              | 3             | 2             | 293       | 305         | 0.96       |
| 8              | 6             | 2             | 293       | 310         | 0.945      |
| 9              | 10            | 2             | 293       | 326         | 0.9        |
| 10             | 12            | 2             | 293       | 333         | 0.88       |

In the engineering method of calculating high-speed reciprocating compressor stage, the formula for calculating the heating coefficient is as follows [23]:

$$\lambda_T = 1 - B (\varepsilon - 1)$$  \hspace{1cm} (8)

where $\varepsilon$ is the ratio of the pressure value at the end of the compression process to the pressure value at the beginning of the compression process; $B$ is the empirical coefficient, the value of which is approximately 0.01 when calculating high-speed stages [23]. According to the results of processing the experimental data presented in Table 2, in relation to slow-speed long-stroke stages, it is preliminarily recommended to assign $B=0.001$.

Unlike the volume coefficient, the pressure coefficient and the temperature coefficient, the leakage coefficient cannot be estimated from experimental indicator and temperature diagrams. One of the ways to solve this problem is to determine the leakage coefficient from the known values of the volumetric efficiency [23, 24] and its four components:

$$\lambda_L = \frac{\lambda}{\lambda_0 \cdot \lambda_p \cdot \lambda_T \cdot \lambda_h}$$  \hspace{1cm} (9)

In this case, the total volumetric efficiency (which is in the numerator) can be defined as the ratio of the measured actual productivity to the value of the theoretical productivity calculated from the known basic dimensions and parameters of the stage [23]. In the denominator, the three coefficients are defined above, and the humidity coefficient does not depend on the design features of the compressor stage and is determined in all cases by the known thermodynamic dependences [22-24].

Some results of processing the experimental data obtained for the considered range of the design and operating parameters of the air slow-speed long-stroke compressor stage are presented in Tables 3 and 4.
Table 3. The base for calculation the leakage coefficient at cycle time $\tau=2$

| $\varepsilon$ | $\lambda_0$ | $\lambda_p$ | $\lambda_T$ | $\lambda_h$ | $\lambda_L$ |
|---------------|-------------|-------------|-------------|-------------|-------------|
| 5             | 0.996       | 0.99        | 0.96        | 0.94        | 0.92        |
| 40            | 0.96        | 0.98        | 0.97        | 0.92        | 0.91        |
| 60            | 0.99        | 0.99        | 0.99        | 0.99        | 0.99        |
| 100           | 0.99        | 0.99        | 0.99        | 0.99        | 0.99        |

$\lambda_L = \frac{\lambda_0 \lambda_p \lambda_T \lambda_h}{\lambda}$

Table 4. The base for calculation the leakage coefficient at cycle time $\tau=3$

| $\varepsilon$ | $\lambda_0$ | $\lambda_p$ | $\lambda_T$ | $\lambda_h$ | $\lambda_L$ |
|---------------|-------------|-------------|-------------|-------------|-------------|
| 5             | 0.98        | 0.99        | 0.92        | 0.99        | 0.95        |
| 40            | 0.96        | 0.97        | 0.98        | 0.99        | 0.75        |
| 60            | 0.94        | 0.96        | 0.98        | 0.99        | 0.52        |
| 100           | 0.92        | 0.96        | 0.99        | 0.99        | 0.15        |

$\lambda_L = \frac{\lambda_0 \lambda_p \lambda_T \lambda_h}{\lambda}$

The analysis of these results showed that it is convenient to represent the leakage coefficient in the form of the following expression to estimate the productivity losses, taking into account the effects of air flows through the gaps in the working chamber:

$$\lambda_L = 1 - x \cdot \varepsilon,$$

(10)

where $x$ is the empirical coefficient (for the considered range of design and operating parameters $x=0.008$); $\varepsilon$ is the ratio of the discharge pressure to the suction pressure.

The method of calculating the actual performance slow-speed long-stroke compressor stage of, based on the definition of the flow coefficient as a set of factors reflecting the influence of various factors on the loss of productivity allows us to estimate the influence of these factors on the performance loss of the object. Figure 10 shows data on the ratio of individual components of performance losses depending on the degree of pressure increase for the considered slow-speed long-stroke stage (Fig. 10a) and for the known high-speed stages (Fig. 10b); in relation to them, the presented comparative results show essential difference in the influence of individual factors on the compressor performance losses.

It is known, in high-speed reciprocating compressor stages, having satisfactory technical condition, the determining factor is the value of the relative dead volume, the effect of which on the compressor performance losses is reflected (presented) by the volume coefficient [6, 8, 23, 24], which is confirmed by the calculation results presented in Figure 10b. In the considered slow-speed stages, this factor is minimized by increasing the ratio of the stroke of the piston to its diameter (in this case, the relative dead volume becomes less than 0.1%); but the long working cycle time and, accordingly, long time of the processes of air flow through the gaps, leads to the fact that the leakage coefficient becomes a key one (Fig. 10a).

7. Conclusion

The analysis and processing of the results of the experimental studies made it possible to develop a method for calculating the actual volumetric efficiency of slow-speed long-stroke air compressor stages, based on the principles of schematization of working processes and separation of losses. It is shown that for the object under consideration, in contrast to the high-speed reciprocating compressors the most important factor in determining the loss of productivity is the leakage coefficient, reflecting the growing influence of mass transfer through the gaps in the working chamber with increasing time cycle stage.

The results obtained determine the current directions for improving the design of slow-speed reciprocating compressor stages, influencing the development of promising sealed self-acting valves and cylinder-piston seals.
Figure 10. Dependence of the components of the flow coefficient on the values of the discharge pressure to the suction pressure a) slow-speed compressor stage (DC=0.05 m; Sp=0.5 m; τ=3 c; \(a_m=0.00026\)); b) high-speed compressor stage (DC=0.05 m; Sp=0.05 m; τ=0.02 c; \(a_m=0.05\))

Obviously, the described results is only the first step in improving the existing methodology for calculating piston compressor stages in relation to slow-speed long-stroke piston compressor stages. Further experimental and theoretical studies will be aimed at clarifying the total volumetric efficiency and its components, including when compressing gases with different properties.

List of symbols

| Symbol | Description                  |
|--------|------------------------------|
| λ₀     | volume coefficient           |
| λₚ     | pressure coefficient         |
| λₜ     | temperature coefficient      |
| λ₉     | humidity coefficient         |
| λₙ     | leakage coefficient          |
| λₙ₀    | indicator volumetric capacity|
| λₙ₁    | "hidden" volumetric capacity |
| Pᵣₙ   | suction pressure             |
| Pₚₖ   | discharge pressure           |
| k      | ratio of specific heats      |
| aₘᵣ   | relative dead volume         |
| $Sp$   | stroke of the piston         |
| Dc     | diameter of cylinder         |
| δₚ    | relative error of the pressure sensor |
| δ     | accuracy                     |
| m      | polytropy index of the ultimate, boundary parameters |
| τ      | working cycle time           |
| $T_{sw}$ | air temperature at the standard suction point |
| $T_{sw,l}$ | air temperature in the working chamber at the end of the suction process |
| $\varepsilon_c$ | ratio of the pressure value at the end of the compression process to the pressure value at the beginning of the compression process |
| $\varepsilon$ | ratio of the discharge pressure to the suction pressure |
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