Influence of suction port parameters on integral characteristics of screw-type compressor

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Abstract. At engineering screw-type machines different tasks on search for more effective and reliable design solutions arise connected with: organization of work process; selection of design shapes of the flow channel; strength calculation of machine components and selection of material. Considered in the work is enhancement of suction process efficiency by the use of dynamic head of the suction gas flow. Influence of rotor length to diameter ratio and suction port opening angle on integral characteristics of the compressor is considered. Calculation of work process in the compressor is performed by means of mathematical simulation. According to the work results it was determined that change in performance coefficient may reach 5%. At increase in suction port opening angle the leakage from compression cavities to a suction chamber increases and may reach 20%.

1. Object

Screw-type compressors (SC) are widely used in different spheres of industry. This is conditioned by high technical and economical parameters of SC. Screw-type compressors are constantly improved, their volumetric and power characteristics, reliability and service life, automation degree are increased, and overall dimensions and specific amount of metal, noise and vibration level are decreased. All this leads to a high demand for SC as compared to other displacement compressors.

At engineering screw-type machines different tasks on search for more effective and reliable design solutions arise connected with: organization of work process [1, 2, 3]; selection of design shapes of the flow channel [4, 5]; strength calculation of machine components and selection of material [6]. The SC application range is expanded by capacity and end pressure, new screw shapes are created, design and manufacturing technology are improved.

The purpose of the present work is:
- increase in SC working process efficiency by means of suction port opening angle optimization;
- derivation of dependence of SC performance coefficient change on suction port opening angle;
- elaboration of recommendations on suction port designing.

Organization of the effective suction process in screw-type compressors is a challenging task [3, 7]. Gas flow from the compressor connecting branch flange to the suction port makes a turn, gas acquires a significant speed in the suction port, at that, gas speed becomes irregular due to discrete supply. Poor stream line of screw teeth at the cavity inlet and absence of the necessary directivity of gas flow leads to shocks and vortex. Gas motion through the screw channels leads to its throw back to the circumference by centrifugal forces. Simultaneously, which is quite significant, substantial volume of heated gas
flowing through slots comes to the suction cavity. All this adversely affects the compressor characteristics and its power (economical) parameters.

2. Calculation method

The actual mass capacity of the compressor is less than the theoretical one, i.e. \( m_a = \lambda m_t \), where \( \lambda < 1 \) — performance coefficient [8, 9].

The performance coefficient for SC can be represented as (1) [8]

\[
\lambda = \lambda_p \cdot \lambda_t \cdot (1 - \nu_u) - \nu_l - \nu_{el},
\]

where \( \lambda_p \) is pressure factor, \( \lambda_t \) is temperature factor, \( \nu_u \) is utilization factor, \( \nu_l \) is leakage factor, \( \nu_{el} \) is external leakage factor.

Pressure factor is a factor taking into account pressure losses at gas flow through the suction port, which is determined by equation (2)

\[
\lambda_p = 1 - \frac{\Delta p}{p_s},
\]

where \( \Delta p \) is pressure losses at flow through the suction port.

Temperature factor \( \lambda_t \) considers the increase of suction gas temperature. A rise of temperature leads to a decrease of the intake gas density and, as a consequence, a decrease of capacity.

Utilization factor \( \nu_u \) shows how much of the theoretically possible volume is used in the suction process.

Leakage factor \( \nu_l \) considers gas leakage from the high-pressure cavities in the low-pressure cavity. Leaks reduce the volume of the suction gas, as they themselves occupy part of the suction cavity volume and heat the intake gas.

External leakage factor \( \nu_{el} \) considers the volume of gas that flows from the compressor's working cavity to the environment.

Pressure losses can be determined by equation (3)

\[
\Delta p = \frac{\xi_s \cdot \rho_s \cdot c_s^2}{2},
\]

where \( \xi_s = \xi_s(Re) \) is coefficient of hydrodynamic resistance at suction, determined by approximation of graphs [10].

Average speed of gas directed motion in the cavity can be found from continuity equation (4) [7].

\[
c_s = \frac{2 \pi \cdot l \cdot n}{\alpha_{1s}}
\]

where \( \alpha_{1s} \) is central angle of suction port (see figure 1).

These formula indicate that in order to increase the pressure factor it is necessary to increase the suction port angle or decrease the rotor length and number of turns. At that, it is necessary to take into account that \( l \) and \( n \) values cannot be used freely at any time. It should be noted that the suction port increase leads to decrease in gas dynamic losses and, consequently, to increase in pressure factor. On the other hand, this leads to suction process delay and increase in mass exchange in the work cavities, which shall lead to gas heating and decrease in suction gas fresh portion. On the one hand, decrease in suction port leads to reduced leakages, and on the other hand to increase in volume underutilization coefficient. This paper is devoted to the optimization of the central angle of suction port \( \alpha_{1s} \).

Besides, it is advisable to design the suction port so that the time within which the suction port connects the suction chamber with the double cavity after its complete disengagement from the double screw tooth is equal to the time for shock wave passing from the discharge end to the suction end. The dynamic head of suction gas flow thus created shall lead to improvement of cavity filling with fresh gas.
Shock wave creation can be represented as follows. Gas moving in the suction cavity meets a fixed compressor housing - end on the discharge side. Flow disturbance thus created propagates in the double cavity of screws towards suction side. As is known, a speed of small disturbances propagation equals to local sound velocity. Consequently, when gas flow runs in the housing end wall the combination of sound waves constantly following each other creates a compression wave which is called the shock wave.

Thus, it is necessary to solve the optimization task which makes it possible to find the most rational parameters of the suction port. To this end, a mathematical model of work process and the software program were developed.

In order to simplify calculations the following assumptions have been made:
1. gas is considered as ideal one;
2. process is quasi-static, i.e. the parameters are the same at any time and in each point of the double cavity;
3. pressure and temperature in the suction and discharge branches remain constant;
4. dry gas compression process without internal cooling is considered. The amount of heat removed through the compressor housing in the process of compression shall be ignored.

At the ends of the calculation area the following boundary conditions are set:
1. at inlet – gas pressure in front of the suction port is equal to gas pressure in the suction branch;
2. at outlet – gas discharge process takes place under internal compression pressure equal to the pressure in the discharge branch (nominal mode).

In order to describe the work process in the screw double cavity the following equations shall be used: state, first law of variable-mass body thermodynamics, adiabatic egress.

The finite equation of parameter change in the double cavity looks as follows

\[
\begin{align*}
\frac{dT}{d\tau} &= \frac{T(k-1)}{p \cdot V} \left[ \frac{dQ_0}{d\tau} + i_{in}(\dot{m}_{in} - \dot{m}_{out}) - \frac{dL_0}{d\tau} - i_{out} \frac{k-1}{k}(\dot{m}_{in} - \dot{m}_{out}) \right], \\
\frac{dp}{d\tau} &= \frac{k-1}{V} \left( \frac{dQ_0}{d\tau} - \frac{k-1}{k} \frac{dL_0}{d\tau} + i_{in} \cdot \dot{m}_{in} - i_{out} \cdot \dot{m}_{out} \right).
\end{align*}
\]

where \(dQ_0\) is heat supplied from surfaces limiting the working cavity, \(dL_0\) is work performed by heat.

The current volume value \(V\) is a function of drive rotor turning angle and is determined by analytical, graphical or numerical method [7, 11]. The analytical method stated in [8] is used in the work.

Gas leakage through gaps which make significant part of the useful capacity takes place in the screw-type compressors [12, 13, 14]. Increase in the number of rotor turns leads to reduction of the relative leakage value, however it leads to increase in losses at suction. Therefore, decrease in the absolute value
of gas losses through slots is one of the main factors of SC efficiency improvement. The following issues are of great significance in SC theory:

1. study of qualitative influence of leakages on the machine parameters and their classification;
2. determination of gas amount flowing through slots of different shape [15];
3. determination of quantitative relations between the total leakage value and the most important machine parameters, as well as analysis of the obtained dependences.

Mass gas flow rate through slots shall be determined by the Saint-Venant - Wantzel equation:

- for pre-critical egress by equation (6)
\[
\dot{m} = \mu \cdot f \cdot \frac{p_2}{T_2} \sqrt{\frac{2k \cdot T_1}{k + 1 \cdot R}} \left( 1 - \left( \frac{p_2}{p_1} \right)^{\frac{k-1}{k}} \right),
\]

- for critical egress by equation (7)
\[
\dot{m} = \mu \cdot f \cdot \frac{k}{R \cdot T_2} \left( \frac{2}{k + 1} \right)^{\frac{k+1}{k-1}}.
\]

In order to determine leakages it is necessary to know the change in sealing lip length along the contact lines and screw edges for one double cavity and the gap between screws. Figure 2 contains projections of the engagement lines and screw contact for two-sided asymmetric profile of teeth [1, 5].

The compressor screws actually do not contact each other, there is some gap between them all the time [16] which is necessary to ensure compressor safe operation. Meant by the contact lines are the imagined lines used to set minimum gaps between the operating screws. It is not simple to determine them by calculation. In the practice of calculations and investigations the contact lines are the tangent lines of screws made in strict accordance with theoretical sizes.

The contact line length is an important parameter of screws engagement since this makes it possible to calculate slot cross-section by screws.

The contact lines must be continuous to ensure safe separation of areas with the increased gas pressure and the suction area. The task of contact lines determination is a spatial one. The suction port must provide maximum filling of cavities with fresh gas. The suction port dimension is characterized by central angles \(\alpha_{1s}\) and \(\alpha_{2s}\) (see figure 1).

Usually, the double cavities of the driving and driven screws are disconnected from the suction chamber simultaneously. For screws with asymmetric shape of teeth [7] by equation (8)
\[
\alpha_{2s} = \frac{\alpha_{1s} + \frac{2\pi}{m_1}}{i_{12}} - \left( \theta_{III} + \theta_{IV} \right),
\]

where \(i_{12}\) is gear ratio, \(\theta_{III}\) and \(\theta_{IV}\) is profile angles.

Connection of suction angles of driving and driven screws makes it possible to bring the task of rational port sizes determination to search optimal sizes of suction angles for one of screws. This is convenient to be done for the driving screw (angle \(\alpha_{1s}\)).

With account of the overlap angle the driving rotor suction angle will be equal to
\[
\alpha_{1s} = \tau_{1st} + \beta_{01} + \Delta\alpha_{1s},
\]

where \(\tau_{1st}\) is swirl angle, \(\beta_{01}\) is angle between the center line and the beam drawn through the rotation center of the driving screw and the tooth top in position of the gas compression start in the double cavity (compression start angle).
Figure 2. Engagement and contact lines of screws with two-sided asymmetrical shape of teeth.

The time within which the double cavity remains connected with the suction branch after complete disengagement of the cavity from the tooth is equal to [7]

\[ t = \frac{\Delta \alpha_{ls}}{2\pi n_1}, \]  

where \( \Delta \alpha_{ls} \) is additional value of suction angle (overlap angle) (see figure 1), \( n_1 \) is number of driving screw turns.

The overlap angle is equal to the rotation angle to which the screw will turn after complete disengagement of the cavity under consideration till its disconnection from the suction chamber. The time of shock wave passing from the discharge end to the section end

\[ t = \frac{l}{c^*}, \]  

where \( l \) is screw length, \( c^* \) is shock wave propagation speed.

By equating the time (10) and (11) we obtain

\[ c^* = \frac{2\pi \cdot l \cdot n_1}{\Delta \alpha_{ls}}. \]
Then the additional value of the suction angle is determined by equation (13)

\[ \Delta \alpha_{s} \approx \frac{2\pi \cdot I \cdot \eta}{\sqrt{kRT}}. \]  

(13)

Solution of the differential equations represented above is not possible in general terms since the equation system is not immediately differentiable. Thus, the special software program was developed for solution of these equations. The differential equations shall be solved by numerical method of step-by-step approximations.

The initial parameters in each chamber must be set in the first approximation for numerical calculation. They can be selected randomly, but this will increase a number of iterations necessary to obtain the required accuracy. That is why the initial parameters in each chamber shall be determined by the adiabatic equation. With a knowledge of initial parameters and having selected the count length the main differential equations can be solved and the final parameters in each chamber and all intermediate values of rotation angle can be found, i.e. the indicator diagram of the first approximation can be obtained. This diagram is not actual yet.

Upon its obtaining the calculation of the second approximation begins; at that, the initial parameters are values obtained at calculation of the first approximation, except for the first compression chamber. The initial parameters in this chamber for the second and subsequent approximations shall be taken as follows: the compression start pressure in each approximation on the basis of the accepted assumptions shall be taken equal to the pressure in the suction branch; the compression start temperature shall be determined according to the calculation results of the previous approximation and shall take into account the gas heating due to leakage.

The program makes it possible to obtain the following dependences: end area of rotor cavities (depending on the driving rotor turning angle), volume of double cavity (depending on the driving rotor turning angle), change in pressure and temperature due to the double cavity volume, change in contact lines (depending on the driving rotor turning angle), change in mass leakage through the contact lines (depending on the driving rotor turning angle).

3. Calculated data representation

The basic variant in calculations is a dry SC with asymmetric shape (see figure 2) of teeth according to 4+6 scheme (four teeth in the driving rotor and six teeth in the driven rotor) [7]. The suction gas - air, suction pressure 0.1 MPa, discharge pressure 0.3 MPa, compressor volumetric capacity 20 (m³/min), rotor length to diameter ratio \( l/d = 1 \), rotor turning angle 300 deg. The geometric compression ratio for all variants is constant. Calculations to reveal dependencies of the performance coefficient and its component parts at change in geometrical and mode parameters has been carried out.

The following parameters have been selected as variable ones: ratio \( l/d \) (rotor length to diameter); discharge pressure within 0.25 to 0.35 MPa; suction temperature within 293 to 313 K. Change in test pressure, suction temperature has taken place at different values of \( l/d \). Given below are graphs of change in performance coefficient and its component parts at change in ratio \( l/d \).
Figure 3. Change in pressure coefficient depending on suction port opening angle.

Figure 4. Change in leakage coefficient depending on suction port opening angle.
Figure 5. Change in utilization coefficient depending on suction port opening angle.

Figure 6. Change in performance coefficient depending on suction port opening angle.

In the process of gas suction to the compressor machine the gas pressure decrease takes place and, as a consequence, slight decrease in its temperature. But simultaneously heat exchange between gas and compressor parts, which are more heated, as a rule, takes place. Therefore, gas temperature is slightly increased. For different values of the central angle of suction port, the components influence of the performance coefficient is different.

The results of the calculations show that changing of the central angle of suction port divides the graph of the performance coefficient function into three zones (figure 6). On the left - the effect of gas-dynamic resistances (figure 3) and underutilization of the volume (figure 5), on the right - the effect of mass leakage (figure 4), the middle part depends on both the first and the second factor.
4. Conclusion
According to work results it was established that the performance coefficient (see figure 6) has an expressed maximum depending on the suction port parameters. Depending on ratio \( l/d \) the maximum value shall be shifted to the right (\( l/d=1.35 \)) or to the left (\( l/d=1.5 \)). Change in the performance coefficient at different modes in the considered intervals reaches 3-5%.

Influence on the performance coefficient constituents:
1. influence on pressure factor (see figure 3): change in pressure factor reaches 1÷3% on the average, the less values correspond to \( l/d=1 \), the more – to \( l/d=1.5 \);
2. influence on utilization factor (see figure 4): At increase in the suction port opening angle the utilization factor is decreased, tending to zero;
3. influence on leakage factor (see figure 5): at increase in the suction port opening angle the leakage factor is increased due to higher leakage from the compression cavities to the suction chamber. Change in the considered ranges reaches 20%.

It should be noted that calculations have been performed at constant volumetric capacity. At that, the compressor geometrical parameters with different \( l/d \) did not coincide. Difference between the pressure factors at \( l/d=1 \) and other values (see figure 3) is connected exactly with this fact.

The recommended values of \( \alpha_1s \): for ratio \( l/d=1 \) the value \( \alpha_1s \) can be equal to 295 deg; \( l/d=1.35 - \alpha_1s=305 \); \( l/d=1.5 - \alpha_1s=315 \). The given values are true for dry compression compressors with a capacity of up to 100 m³/min and ratio \( \pi \) up to 4.

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List of symbols and abbreviations

| Symbols | Subscripts |
|---------|------------|
| c       | 1          | gas parameters upstream the port |
| d       | 2          | gas parameters downstream the port |
| i       | in         | incoming gas |
| f       | out        | outgoing gas |
| k       | s          | suction |
| l       |            | |
| m       |            | |
| m       |            | |
| n       |            | |
| p       |            | |
| R       |            | |
| T       |            | |
| V       |            | |
| \( \rho \) |            | |
| \( \tau \) |            | |
| \( \mu \) |            | |

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