Features of the operation of a refrigeration screw compressor for capacity control

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Abstract. To ensure the required temperature of the cooled object by maintaining the heat balance between the heat gain and the refrigerating capacity of the refrigerating machine, a compressor performance control system is used. The article presents the dependencies of the change in efficiency coefficient at full and partial performance with joint regulation of productivity and geometric compression ratio. The laws of the optimal change in the geometric degree of compression with a change in performance are given. The experimental characteristics of a screw compressor running on R22 freon were used.

1. Introduction

Oil-flooded screw compressors (OSC) are widely used in various fields which use artificial cold [1]. In order to maintain the temperature of the cooled object in a predetermined temperature range by ensuring the heat balance between the heat influx and the refrigerating capacity of the refrigerating machine when the temperature of its operation changes, a system for controlling the performance of an oil-flooded screw compressor is used.

There are the following methods for regulating the performance of refrigeration screw compressors: using external devices; changing rotor speed; using built-in devices that change the volume of the cavity of the screws when they are cut off from the suction window.

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Regulation with the use of external devices is carried out by bypassing the compressed refrigerant or by suction throttling.

A method for controlling OSC performance by changing the rotor speed in the range of optimal peripheral speed is quite economical, since the geometric compression ratio does not change. Currently, regulation of rotor speed using frequency converters has become widespread. In [2, 3], performance control was considered by changing the rotational speed of the screws, but with
a decrease in the rotational speed, with a small volumetric capacity of the screw compressor, the operational efficiency is significantly reduced due to large internal refrigerant leaks.

One of the advantages of screw compressors over other types of compressors is the ability to control the cooling capacity of the built-in devices.

The problem of choosing a performance regulator and geometric compression ratio of a screw compressor, which ensures efficient operation of the chiller while reducing productivity and changing the boiling and condensation temperatures, is relevant.

2. Results and discussion

Widespread regulation by one spool leads to significant loss of work due to a decrease in the geometric degree of compression $\varepsilon_G$ with a decrease in productivity [2]. This paper presents the dependences of the compressor efficiency at 100% capacity with a constant geometric compression ratio $\varepsilon_G = 5.0$ and when adjusting the geometric degree by changing the cylindrical part of the discharge window. When the compressor is operating at full capacity with an SRM D standard profile with external screw diameters of $D_1 = D_2 = 204$ mm on $R22$ freon at a boiling point of 10 °C and a condensation temperature of 350 °C, the indicator efficiency of the compressor without adjusting the geometric compression ratio by 4.75% higher than $\varepsilon_G$-controlled compressor.

A decrease in compressor efficiency was due to a decrease in the discharge window area.

The performance control by the slide valve and the thrust bearing has a limited area of variation in the geometric degree of compression, which does not allow to obtain the optimal law of its change with a decrease in productivity [2, 4].

To solve the problem of regulating the capacity and geometrical compression ratio of a screw compressor, which ensures the efficient operation of the refrigerating machine, it is necessary to choose the design of the capacity regulator and geometrical compression ratio to minimize losses from mismatch between the internal pressure of the refrigerant vapor and the condensing pressure. It is also necessary to evaluate the energy efficiency of the proposed regulator when regulating compressor performance.

3. Objects of study and research methods

The authors propose the use of a performance and geometrical compression ratio regulator for a screw compressor, consisting of a slide valve and two rotary valves. Such a regulator allows one not only to regulate the geometric degree of compression, but also to obtain the necessary law of variation of $\varepsilon_G$ with a decrease in productivity. Figure 1 shows the design of such a regulator.

![Figure 1. The design of the regulator of performance and the geometric degree of compression by the valve and rotary dampers: 1 – spool; 2 – rotary shutters; 3 – the bearing of rotary shutters.](image-url)
To evaluate the performance of a refrigerating machine with an oil-flooded screw compressor and a regulator consisting of a slide valve and rotary dampers, it is necessary to analyze the characteristics of a screw compressor with full and partial performance in various modes of operation.

At the department of Low-Temperature Energy of Saint-Petersburg National Research University of Information Technologies, Mechanics and Optics an experimental study of refrigeration OSC on R22 freon was carried out with the possibility of changing the performance and geometric compression ratio and its characteristics were obtained.

Studies of the refrigerating OSC were carried out at the experimental bench. The experimental bench is a cascade-refrigerating machine. The upper branch of the cascade consists of an oil-flooded screw compressor $S$ 3-900 from Kühlautomat (Germany), a condenser, a control valve and a condenser-evaporator. The condenser-evaporator is a condenser for the lower branch of the cascade. An experimental OSC and a water heat exchanger enter the lower branch of the cascade-refrigerating machine.

Tests of the experimental compressor were carried out on a gas ring. After compression in the experimental compressor, the refrigerant vapor is throttled by the valve and cooled in the water heat exchanger and condenser-evaporator, not reaching the saturation state, after which it is sucked into the compressor. The suction temperature is set by cooling the refrigerant vapor with an upper stage-refrigerating machine. The suction and discharge valves set suction and discharge pressures.

An asynchronous electric motor was used as the drive of the experimental OSC. The bench was equipped with the necessary measuring equipment. Standard pressure gauges with a maximum permissible measurement error of 0.4%, and temperature determined the refrigerant vapor pressure at the nodal points by laboratory thermometers with a division value of 0.10 °C.

To determine the mass flow rate of the refrigerant vapor, a narrowing device, a diaphragm, was used. The rotational speed of the electric motor shaft of the experimental OSC was determined by a frequency meter using an optoelectronic pair.

The power consumed by the experimental OSC was determined by the electric power of the electric motor, by means of a measuring complex for measuring current, voltage and power with a maximum permissible measurement error of 0.5%. When determining the power on the compressor shaft, catalog data on the coefficient of efficiency of the electric motor were used. The root mean square relative errors in determining the effective power $Ne$ (power on the compressor shaft) do not exceed 3%. The refrigerant used was R22 freon.

The experimental screw compressor has the following characteristics: the number of teeth of the lead and driven screws 4/6; outer diameters of screws 160 mm; screw length 144 mm; rotational speed of the drive screw is 49.2 s$^{-1}$; theoretical volumetric capacity is 0.0841 m$^3$/s; asymmetrical screw tooth profiles RDC Turbocompressor (Kazan, Russia). A slide valve and interchangeable inserts simulating rotary dampers carried out performance control. The position of the spool and the interchangeable inserts provided a different geometric degree of compression. The compressor efficiency was estimated by the efficiency factor $\eta_e = N_s/N_o$, where $N_s$ is isentropic power.

### 4. Research results and discussion

As a result of an experimental study of a screw oil-flooded compressor, the values of the efficiency factor were obtained from the degree of pressure increase at full capacity and relative productivity equal to 75% and 50% of the maximum.

At the first stage of the experiment, the injection window remained unchanged, with the initial value of the geometric compression ratio $\varepsilon_G = 2.6$. The obtained dependences $\eta_e$, at a boiling point of $t_0 = -20 \degree C$, on the external compression ratio $\pi_n = p_d/p_s$, where $p_d$ and $p_s$ are the discharge and suction pressures, are shown in figure 2. The current value of the geometric degree of compression changes along the line a – b (figure 3).

At full capacity, the maximum $\eta_e$ is achieved at $\pi_n = 3$, which corresponds to the coincidence of the discharge pressure and the internal compression pressure. The maxima $\eta_e$ with a decrease in productivity are shifted towards lower values of $\pi_n$ due to a decrease in $\varepsilon_G$.
Figure 2. The dependence of the efficiency factor \( \eta_e \) of the screw compressor from the external compression ratio \( \pi_n \) when controlled by one valve with a constant geometric compression ratio \( \varepsilon_G \) for \( \pi_n/V_i = 1.0; \quad \pi_n/V_i = 0.75; \quad \pi_n/V_i = 0.5 \).

At the second stage of the experiment, the injection window was changed so that the ratio of the volumes of the paired cavity at the beginning and at the end of the compression process is the current value of the geometric degree of compression \( \varepsilon_{Gi} \), with a relative volumetric productivity of 75% and 50%, it remains equal to 2.6 (line a–c in figure 3).

As a result of the experiment, an insufficient increase in \( \eta_e \) was obtained, which is explained by significant losses during expulsion of steam into the suction chamber during the contraction of the paired cavity until it is cut off from the suction window [5, 6]. The increase in pressure in the cavity at the beginning of the compression process and the decrease in the internal compression ratio is \( \pi_a = p_a/p_n \), where \( p_n \) is the pressure in the cavity at the moment of its connection with the discharge window.

At the third stage of the experiment, in order to determine the optimal law for changing the current value of the geometric compression ratio \( \varepsilon_{Gi} \) from the relative productivity \( V_i/V_t \), a study was carried out with various discharge windows. A graph of \( \varepsilon_{Gi} \) versus \( V_i/V_t \) at which the compressor’s efficiency at a boiling point \( t_0 = -20 \, ^\circ C \) is maximal, is presented in figure 3 (line a–d).

Figure 3. The dependence of the change in the geometric degree of compression \( \varepsilon_G \) on the relative productivity \( V_i/V_t \) for \( a - b - \varepsilon_G = const; \quad a - c - \varepsilon_{Gi} = const; \quad a - d - \varepsilon_{Gi} = var. \)
Figure 4 shows the experimental dependences of the efficiency factor of the compressor $\eta_e$ on the external compression ratio $\pi_H$ at the boiling point of the refrigerant $t_0 = -20 ^\circ C$ when working with a relative capacity of $V_{ti}/V_T = 1.0; 0.75$ and $0.5$, where $V_T$ is the total theoretical volumetric capacity, $V_{ti}$ is the partial theoretical volumetric capacity. The current value of the geometric compression ratio varied along the line $a - d$. This ensured the equality of the external and internal compression ratio.

![Figure 4](image)

**Figure 4.** The dependence of the efficiency factor $\eta_e$ of a screw compressor on the external compression ratio $\pi_e$ when changing the geometric compression ratio $\varepsilon_{Gi}$ along the line $a - d$, the designations are the same as in figure 2.

Figure 5 shows the dependences of $\eta_e$ on the relative productivity $V_{ti}/V_T$ at different boiling points and performance control with one valve. Figure 6 shows the same dependences when controlling the capacity with the slide valve and rotary dampers when $\varepsilon_{Gi}$ changes along the line $a - d$.

![Figure 5](image)

**Figure 5.** The dependence of the efficiency factor $\eta_e$ of a screw compressor on the relative volumetric productivity $V_{ti}/V_T$ when controlling with one spool. $\bullet - t_0 = -10 ^\circ C; \blacksquare - t_0 = -20 ^\circ C$. 

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Figure 6. The dependence of the efficiency factor $\eta_e$ of a screw compressor on the relative theoretical volumetric capacity $V_{Ti}/V_T$ when changing the geometric compression ratio $\varepsilon_{Gi}$ along the line $a – d$, the designations are the same as in figure 5.

From these dependencies it can be seen that the most effective way to regulate the performance with a decrease in the condensation temperature is the slide valve with rotary valves. Moreover, the change in the current geometric degree of compression occurs along the line $a – d$.

Compared to regulating the productivity by one valve with a relative productivity of 75%, the efficiency factor $\eta_e$ increases by 5.4% (from 0.538 to 0.567) at a boiling point of $t_0 = –10^\circ C$ and by 7.2% (from 0.513 to 0.55) at a temperature of $t_0 = –20^\circ C$. At 50% productivity, the efficiency factor $\eta_e$ increases by 14% (from 0.46 to 0.525) at a boiling point $t_0 = –10^\circ C$ and by 19% (from 0.42 to 0.5) at a temperature $t_0 = –20^\circ C$.

The obtained dependencies can be used in solving the problem of evaluating the operation of a refrigerating machine in a spread of external and internal parameters. In this case, a promising method is the method of the theory of entropy potentials [7].

5. Conclusion
When ensuring the efficient operation of the refrigerating machine when regulating performance, an effective regulator is a regulator consisting of a slide valve and rotary dampers. With a decrease in productivity by 75%, the value of efficiency factor increases to 7.2%, and with a decrease in productivity by 50% – up to 19% compared with a regulator consisting of one valve with a constant geometric compression ratio.

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