Improving the efficiency of a vapor refrigerating machine with a screw compressor while adjusting the performance

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Abstract. When using an air condenser in a refrigeration machine, the condensation temperature significantly depends on the parameters of the outside air. To ensure stable operation of the throttling device, the condensation temperature must be maintained and remain no lower than a certain value. This leads to unnecessary energy consumption for steam compression in the compressor. The article presents a scheme of a refrigerating machine with a screw compressor which allows to reduce energy consumption for cold production by reducing the condensation pressure when the ambient temperature drops and installing a pump in front of a throttle device. Operation of the refrigerating machine with a screw oil-filled compressor with full capacity on different modes has peculiarities. The article describes the regulator of the screw compressor, which consists of a spool and two butterfly valves. This design allows for adjustment of the geometric degree of compression with full performance and allows to get arbitrary laws of change with decreasing performance. The article presents the dependences of cooling capacity, effective power and coefficient of performance at full and 50% performance of a screw compressor with a decrease in condensation temperature. Experimental and calculated characteristics of a refrigeration oil-filled screw compressor operating on Freon R₂₂ were used.

Using a scheme of a vapor refrigerating machine with a screw compressor pump in front of the throttle valve can significantly increase the energy efficiency of its work.

1. Introduction

Nowadays, oil-filled screw compressors (OSC) are widely used in vapor refrigerating machines. When the refrigerating machine with an air condenser is operating, the condensation temperature and pressure is determined by the outside air temperature and, as it is lowered, the machine's operating efficiency rises. Under reduced condensation pressure, unstable operation of thermostatic valves (TV) occurs, which leads to an uneven supply of liquid refrigerant to the evaporator. In order to ensure the stable operation of thermostatic valves, the temperature and pressure of condensation is maintained not lower than a certain value. Danfoss recommends maintaining the condensation temperature in a medium-temperature refrigerating machine at a temperature not lower than 30°C, and in a low-temperature refrigerating machine not lower than 20°C. Thus, when compressing steam in the compressor, extra work is expended.

In work [1] operation of the refrigerating machine is shown, the scheme of which is shown in figure 1. A pump is installed between the condenser and the control valve, it increases the pressure of the liquid
refrigerant to a value corresponding to the saturation pressure of the refrigerant at a temperature of 30°C, which ensures stable operation of the control valve. When the condensing temperature decreases, the pressure in the condenser decreases, which leads to an increase in the efficiency of the refrigerating machine. The work expended by the pump on increasing the pressure of the liquid refrigerant is much less than the additional work expended on compressing the steam in the compressor. Work under this scheme allows for maintaining the pressure necessary for normal operation of the thermostatic valves when the outside air temperature decreases. The article provides an assessment of the effectiveness of the operation of the refrigeration machine according to this scheme with a scroll and a piston compressor.

2. Materials and methods
Operation of the refrigerating machine with a screw oil-filled compressor with full capacity on different modes has peculiarities. A significant advantage of the OSC is the ability to control the performance using internal devices. When adjusting the performance of one spool together with a decrease in performance, the geometrical degree of compression $\varepsilon_G$ decreases, which leads to an increase in work losses associated with the lack of refrigerant vapor [2].

![Diagram](image)

**Figure 1.** Scheme (a) and the operation cycle (b) of a vapor refrigerating machine $CD$ – condenser; $HE$ – heat exchanger; $P$ – pump; $CV$ – control valve; $E$ – evaporator; $CM$ – compressor; the operation cycle of the refrigerating machine without a pump $1’-2’-3’-4’-5’$; cycle of operation of the refrigerating machine with a pump $1-2-3-4-5-6$.

In another work [3], a regulator consisting of a spool and two butterfly valves was suggested. This design of the regulator allows for changing the geometric degree of compression $\varepsilon_G$ with full performance and get arbitrary laws of change of $\varepsilon_G$ with decreasing performance.

To assess the performance of a vapor refrigerating machine with a pump in front of a control valve, it is necessary to calculate the characteristics of a OSC with a full and 50% capacity.

For this, experimental and calculated dependencies of cooling capacity, effective power and coefficient of performance of compressors operating on Freon $R22$ were determined. Experimental characteristics were obtained at the Department of Low-Temperature Engineering and Renewable Energy of the ITMO University [3].

The experimental compressor has the following geometrical characteristics: the number of settings of the leading screw (LS) and the driven screw (DS) is 4/6; outer diameters of the LS and DS screws is 160 mm; screw length is 144 mm; stroke of the screw is 144 mm; screw rotation speed is 2940 r/m;
theoretical volume capacity is 5.1 m³/min; profiles of the teeth of the screws are of the standard range. Performance regulation was carried out by a spool and butterfly valves [3].

Calculations were made for a screw compressor having the following geometrical characteristics: the number of settings of LS and DS is 5/6; outer diameter of the LS is 137.5 mm; outer diameter of the DS is 107.6 mm; stroke of the LS is 215 mm; screw length is 200 mm; screw rotation speed is 2940 r/min; theoretical volumetric capacity is 4.8 m³/min; the profiles of the teeth of the screws are made according to [4]. In calculating the effective efficiency, the dependencies given in [2, 3, 4, 5, 6] were used.

In Figure 2 the dependences of the effective efficiency $\eta_e$ of compressors on the external degree of pressure increase $\pi_H$ when operating at full capacity and relative performance $V_{Ti}/V_T = 0.5$ at the boiling temperature of the refrigerant $t_0 = -20°C$ are shown.

![Figure 2](image)

**Figure 2.** The dependence of the effective efficiency $\eta_e$ of the screw compressor on the external compression ratio $\pi_H$.

$- V_{Ti}/V_T = 1.0; - V_{Ti}/V_T = 0.5;$

--- OSC by size range (experiment)

--- OSC with the ratio of the number of teeth 5/6 R 22 (calculation)

To determine the law of change in the current value of the geometric compression ratio $\varepsilon_{Gi}$ from the relative volumetric capacity of the compressor $V_{Ti}/V_T$, at which the values of the effective efficiency $\eta_e$ have the greatest values, an experimental study of the OSC with different end discharge windows that were changed using interchangeable inserts was conducted.

![Figure 3](image)

**Figure 3.** Dependence of the effective efficiency $\eta_e$ of an experimental screw compressor on the geometric compression ratio $\varepsilon_{Gi}$.

$\pi_N = 2.5; - \pi_N = 3.0; - \pi_N = 4.0.$

Figure 3 shows the dependences of $\eta_e$ on $\varepsilon_{Gi}$ with relative performance $V_{Ti}/V_T = 0.5$ for various external degrees of compression and boiling point $t_0 = -20°C$. From the examination of the figures it
can be seen that the curves shown have maxima, the position and absolute value of which depend on the external compression ratio $\pi_H$. When $\pi_H = 2,5 ... 4$, the largest values of $\eta_{\text{Soc}}$ occur at $\varepsilon_{\text{Gi}} = 2,0 ... 2,2$. The dependence $\eta_e = f(\varepsilon_{\text{Gi}})$ is shown in Figure 1 in [10] and is determined by the line $f - q$.

In figure 4, 5, 6 shows the dependences of the cooling capacity $Q_0$, the effective power $N_e$ and the cooling coefficient of the cooling machine with OSC on the condensation temperature $t_b$ with full and partial performance. Boiling point $t_0 = -20^\circ\text{C}$.

**Figure 4.** Dependency of refrigerating capacity of compressor $Q_0$ on the boiling temperature $t_b$. Designations are the same as in figure 2.

**Figure 5.** Dependency of effective power of compressor $N_e$ on the boiling temperature $t_b$. Designations are the same as in figure 2.
3. Discussion of the research results

From the presented dependences it is clear that the refrigeration coefficient of a vapor refrigerating machine with a screw compressor in full and partial capacity increases while maintaining the condensation temperature in accordance with the outdoor air temperature.

When the condensation temperature decreases from $t_c = 31\, ^\circ\text{C}$ to $t_c = 12.6\, ^\circ\text{C}$, the refrigeration coefficient increases in full capacity from $\varepsilon_e = 1.48$ to $\varepsilon_e = 3.9$ for the experimental compressor and from $\varepsilon_e = 2.02$ to $\varepsilon_e = 4.5$ according to the calculation results, and at 50% capacity from $\varepsilon_e = 0.93$ to $\varepsilon_e = 3.34$ for the experimental compressor and from $\varepsilon_e = 1.32$ to $\varepsilon_e = 3.62$ as calculated.

The increase in the refrigeration coefficient of a vapor refrigerating machine occurs due to a decrease in the operation of the compressor, since the outlay of work for increasing the fluid pressure by the pump is significantly less than the outlay of compressing steam. The magnitude of the pump does not exceed 2.5% of the magnitude of the compressor.

Thus, the use of a circuit with an oil-filled screw compressor and pump leads to an increase in the operating efficiency of the vapor refrigerating machine with full and partial performance while reducing the outside air temperature.

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