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Mathematical modeling of working processes of variable frequency screw compressor with differentiated oil supply into the working chamber

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Abstract. Currently, a method of screw compressors regulation by smooth change of rotors spinning frequency, or so-called frequency regulation, is widely used. The operation process of oil-injected screw compressors at frequency regulation modes were examined in this paper in terms of the relationship of a compressor operation modes and its oil system. The calculations based on the operation process mathematical model of oil-injected screw compressor were aimed at defining its integral characteristics when changing the rotor spinning frequency, the quantity of the oil supplied and the discharge pressure.

1. Introduction

It is known that for improving the efficiency of screw compressors working process, the liquid is supplied into their working chamber. This reduces the intensity of gas leakage through the clearances, which is one of the drawbacks of the screw compressor design. In addition, the injected liquid provides intensive cooling of the compressed gas in the working chamber itself and, accordingly, the permissible value of the discharge gas temperature even if the ratio of the discharge pressure to the suction pressure is in the range of 5...15. Usually, oil or water is used as working liquid. Water provides the most efficient gas cooling, but due to a number of indicators (e.g. poor sealing of clearances due to low viscosity; corrosion activity, etc.) is has limited application [1, 2, etc.]. Screw oil-injected air, gas and refrigerating compressors have been widely used; moreover, in the process of operation there is often a change of both specific mode parameters (performance, suction and discharge pressure, boiling and condensation temperatures, capacity) and their various combinations [3 – 5]. The method of regulation of the screw compressor, applied in one or another case, determines not only the efficiency of its operation, but also the reliability of the design, its maintainability, weight characteristics, the cost of the compressor and the compressor unit as a whole. Nowadays, a method of screw compressors regulation by smooth change of rotors spinning frequency, or so-called frequency regulation, is widely used.

In this connection, it is necessary to note, that presence of oil in a working chamber has not only a positive, but also a negative effect, as it leads to increase in indicator losses of power in the discharge process and losses on hydro-mechanical friction in a compressor flowing part, as well as to additional "pumping" power expenditure. Both the positive and the negative effect of oil injection into the working chamber depends on the amount of oil and its viscosity (i.e. the oil temperature). Since the frequency control of the performance is accompanied in most cases by forced injection of oil under pressure, the amount of oil supplied to the compressor is determined by the fixed geometric dimensions of the elements of the oil system flow part and the difference between the values of suction and discharge pressures, which do not depend on the rotation frequency of the compressor rotors. It can be assumed that the change in the specific amount of oil (relative to the number of working gas) will affect the efficiency of the oil-injected compressor. In order to determine the possibility and feasibility of reducing energy consumption in compressor plants, this article investigates the working process of screw oil-injected compressor on frequency control modes concerning the relationship of the compressor operating modes and its oil system.

2. Method of calculation

Computational and theoretical studies were carried out using a mathematical model of oil-injected screw compressor working process and were aimed to define the capacity, efficiency coefficient, performance, flow coefficient and other characteristics at various modes (when changing the rotor speed, the quantity and the temperature of the supplied oil). The analysis of the current methods of calculating the working processes of screw oil-injected compressors [6-14] allowed to develop a methodology that takes into account the interconnection of the compressor stage and its oil system.

Methodology for calculation of the screw compressor under consideration was chronologically developed in two stages. At the first stage, the working process was analyzed with the widest possible list of physical phenomena accompanying it. Based on the results of its implementation, not very important physical phenomena were identified, which allowed, at the second stage, to simplify the analytical methodology for more effective application.

According to the results of the first stage of developing a mathematical model [11, 12] a physical model of the working process of a screw compressor with a heterogeneous gas-liquid working medium can be presented as follows. During the rotor (rotors) spinning, the working chamber volume changes. Gas (the first stage – air) rotates together with the rotor about its axis. During the compression, dropping liquid (at the first stage, water or oil) is supplied into the working chamber. The gas flow acts upon the dropping liquid that causes the change in its velocity, deformation and fragmentation. When interacting with the surfaces of the body, the droplets precipitate on them and spread out in the form of a film. When interacting with the surfaces of the rotating rotor the liquid may either break up into smaller particles, or precipitate on them, depending on the impact velocity. The working process is accompanied by external mass transfer through the clearances, partially sealed by liquid, and through the gas distribution elements; by the external heat gas exchange with the surfaces of the rotor and the body, and by heat and mass transfer between gas and liquid.

A generalized working process scheme in the considered control volume is shown in figure 1. In the control volume mass and heat flows are conventionally presented by generalized values, which reflect the direction and physical nature of these flows. To simplify the mathematical description of the considered working process and their implementation let us make the following assumptions: gas medium is continuous; pressure of the working liquid in the suction and discharge chambers is constant; dropping liquid is a system of spherical shape droplets of the same diameter suspended in the flow; working gas flow through a gas-distribution elements and constructive clearances is taken as adiabatic and quasi-static (where real gas properties and its compressibility are taken into account by the empirical coefficient); the processes of inter-phase mass transfer in the working chambers are quasi-stationary; the heating of the liquid film takes place with a constant temperature on the surface and in the liquid layer; the absolute motion of the gas phase in the working chamber is defined by translational, caused by the rotor spinning and by relative movement caused by a change of the working length of the screw chamber during the rotors interaction; the motion of droplets is not affected by the hydrodynamic interaction between them; impact of the Magnus force, hydro-mechanical, diffusive and reactive forces on a droplet is negligible; during the interaction with the body surfaces the liquid is precipitated in the form of a film, uniformly distributing over the surface; there are no failure of the film from the surface of the working chamber and the coagulation of the droplets in the gas stream; during the crushing of liquid particles by the rotating surfaces of the rotors the size of the secondary particles is completely determined by the velocity of collision of the liquid and the rotor.

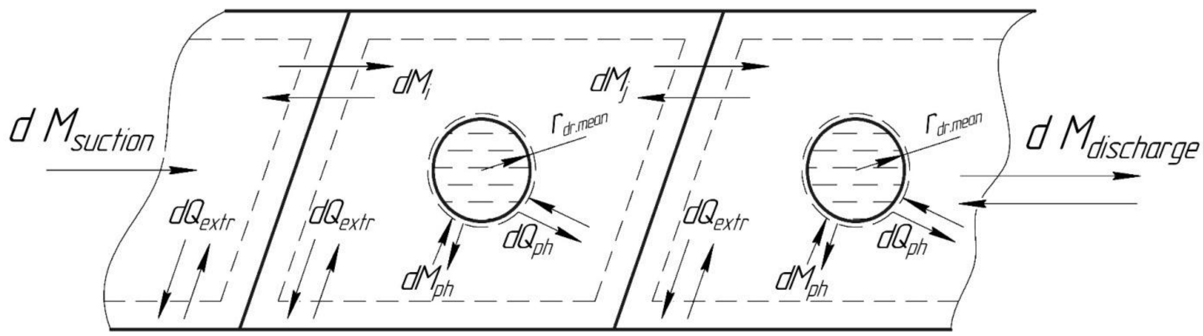

Figure 1. A calculation scheme of a screw compressor with a heterogeneous gas-liquid working medium

Single-valuedness conditions include:

- geometrical conditions (listed below);
- physical conditions (physical characteristics of components of the working medium at the first stage the working medium was a heterogeneous mixture of air and liquid (water or oil), at the second stage - air or R134 and oil;
- initial conditions characterizing the parameters of the working medium in initial time of the working process; for gas these are the settings in the beginning of the process of absorption, which are assumed to be parameters of the gas in the intake chamber; for the liquid–droplet parameters at the moment of entering the coupled chamber from the injection device;
- boundary conditions, characterizing the interaction of working medium with the medium; these are parameters of the gas in the suction and the discharge chambers, surface temperature of compressor parts [11, 13], interacting with the working liquid and liquid parameters before the injection device (injector).

The basic calculation equations used in the mathematical model are the first law of thermodynamics for a body of variable mass, including external and inter-phase mass transfer, law of conservation of mass for gas and liquid component of the working medium; the Newton-Richmann law (external (gas – compressor parts) and inter-phase (gas – liquid) heat exchange); the dependencies for determination of thermo-physical properties of gas and liquids; and the real gas equation [15]. The heat transfer coefficient for a droplet moving relative to the gas phase, in accordance with the assumptions with respect to the averaged parameters of liquid droplets and gas flow, is determined as follows [16]:

$$\alpha = \frac{Nu\lambda}{2r}\ , \tag{1}$$

where the criterion Nusselt number according to [17], is defined by the following expression: $Nu = 2\left(1 + \beta \operatorname{Re}^{1/2} \Pr^{1/3}\right)$ (β is the constant coefficient [17]).

The calculation of transient heating of the dropping liquid is carried out by the heat balance method [16]. The change in mass of the gas phase due to the inter-phase mass transfer, taking into account quasi-stationary processes of evaporation and condensation for the liquid droplet can be measured using the formula of Fressling [18]. The flow of the gas-liquid mixture through the working clearances and the gas distribution valve is described by the equation of Saint-Venant and Wantsel with the introduction of the correction factor that takes into account the compressibility as the compressibility of gas and the presence of liquids [11, 14]. The motion of evaporating droplets in the working chamber is described by the differential equation of a variable mass point motion (the equation of Meshchersky). While droplets breaking by the gas flow the size of the secondary droplets is determined by the recommendations given in [19]; while its breaking by the surfaces of the spinning rotor – by ones supposed in [20].

In addition to the current settings during the calculation of the working process integral characteristics, efficiency factor of the supply, power input to the compressor shaft, taking into account the expenditures of liquid injection are determined [11, 21, 22]:

$$W = W_{\text{ind}} + W_T + W_{\text{add}} \tag{2}$$

where, in relation to the object of study, W_{ind} is the indicated power of the compressor stage; W_{sdd} is the power of the pump that feeds the liquid into the working chamber; $W_{\text{T}} = W_{\text{mech}} + W_{\text{mix}}$ is the friction power of rotors on the gas-liquid mixture (W_{mix}), as well as the friction power in the bearings (W_{mech}).

The explanation is required for the calculation of power losses of the rotors friction on the gas-liquid mixture, which is approximately determined on the basis of the available experimental and theoretical data on the friction of simple configuration bodies of rotation.

Friction power of the disc surfaces of the rotor ends is described by the following expression [22]:

$$W_{\text{mix1}} = 0.5 C_{\text{m}} \rho_{mix} \omega^3 r_{\text{r}}^5 \tag{3}$$

where ρ_{mix} is the density of the gas-liquid mixture, ω is the angular velocity of the rotor; r_{r} is the radius of the rotor; $C_{\text{m}} = f(\text{Re}, r_{\text{r}}, \delta)$ is the coefficient of resistance (δ is the rotor-body clearance).

The power expenditures for the friction of the rotor outer surface on the near-wall liquid layer can be determined as follows [23]:

$$W_{\text{mix2}} = \mu_{\text{mix}} \omega^2 r_{\text{r}}^2 F_{cyl} / \delta \tag{4}$$

where μ_{mix} is the viscosity of gas-liquid mixture in the side-wall layer; F_{cyl} is the area of the outer cylindrical surface of the rotors.

The power required for moving the gas-liquid mixture by the side surfaces of the rotor teeth is determined by analogy with the power expenditure to the sparging of the gas-liquid mixture by toothed cylindrical wheels [24]:

$$W_{\text{mix3}} = 0.015 \omega^3 r_{\text{r}}^5 \left(1 + 2.5 L_{\text{r}} / r_{\text{r}}\right) \cdot 10^{-3} \tag{5}$$

where L_{r} is the rotor length.

For solving the systems of differential equations, described above, the iterated Euler numerical method is used [25].

The convergence of the initial and final values of the calculating cycle is provided by the following conditions [21, 26]:

$$P_{n+1(\varphi_1=0)} - P_{n(\varphi_1=\pi/2)} \le \xi_1 \tag{6}$$

$$T_{n+1(\varphi_1=0)} - T_{n(\varphi_1=\pi/2)} \le \xi_2 \tag{7}$$

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where ξ_1 and ξ_2 is the pre-set accuracy of calculation; P_n, T_n are the parameters of the working medium at the current phase or at the time layer of the working process; P_{n+1}, T_{n+1} are the parameters of the working medium at the following phase or the time layer of the working process.

If the condition of convergence is not met, the adjustment of the initial conditions for each area is made, for example, $P_{n+1(\varphi 1=0)} = P_{n(\varphi 1=\pi/2)}$. At the end of every calculation cycle the value of the gas temperature in the discharge chamber is specified as the bulk temperature of the discharge gas.

The results of the verification of the mathematical model developed at the first stage, by comparing the calculated and experimental indicator diagrams of the working process of air screw compressor with oil or water injection, as well as its integral characteristics, are presented in [11, 12]. They show that the tested methods correctly describe the quality of physical processes, accompanying the investigated screw compressor operation; however, they accurately ensure the calculation of their quantitative characteristics.

At the first phase of the working process modeling analysis it was proved that during oil injection the influence of evaporation and condensation processes is negligible; it was also shown that convective heat transfer between gas and liquid film, between gas and compressor parts surfaces is also negligible in comparison with convective heat exchange between gas and liquid droplets. When analyzing the integral characteristics it was revealed that the compressor efficiency ratio (i.e. input-output ratio) depends on the frequency of the rotors spinning and the relative amount of liquid supplied into the working chamber. The optimal amount of liquid varies with the ratio of discharge and suction pressure and the frequency of the rotors.

Therefore, at the second stage of the modeling the main aim was to analyze the influence of the oil supply system into the screw compressor on its efficiency at frequency regulation modes. Based on the results of the calculation and parametric analysis at the first stage, the mathematical model of the working process of air and refrigeration screw compressor with oil injection into the working chamber was simplified. This provided the efficiency of its practical application. The simplified system of the main design equations on the second stage of mathematical modeling has obtained the following form.

The system of equations for calculating the gas suction into the working chamber considers mainly oil-free gas and has the following form:

$$\begin{cases} \dfrac{dT}{d\varphi} = (k-1)\cdot\dfrac{T}{m}\cdot\dfrac{\dot{m}}{\omega} + \dfrac{\sum\limits_{i\neq 1}(k\cdot T_i - T)\cdot\dot{m}_i}{m\cdot\omega} - (k-1)\dfrac{T}{V}\cdot\dfrac{dV}{d\varphi} \\ \dfrac{dp}{d\varphi} = z\cdot k\cdot\dfrac{p}{m}\cdot\dfrac{\dot{m}}{\omega} + z\cdot\dfrac{p}{m}\cdot\dfrac{k\cdot\sum\limits_{i\neq 1}T_i\cdot\dot{m}_i}{T\cdot\omega} - k\cdot\dfrac{p}{V}\cdot\dfrac{dV}{d\varphi} \\ \dfrac{dV}{d\varphi} = f(\varphi) \\ \dfrac{dm}{d\varphi} = \dfrac{\sum\dot{m}}{\omega} \\ \dfrac{dL}{d\varphi} = p\cdot\dfrac{dV}{d\varphi} \end{cases} \tag{8}$$

The system of equations for calculating the processes of compression and discharge is written as follows:

$$
\begin{cases}
\dfrac{dT}{d\varphi} = \dfrac{3\cdot\alpha\cdot d}{\mathrm{c_V}\cdot \mathrm{r}\cdot\rho_{\mathrm{oil}}}\cdot\dfrac{(T_{oil}-T)}{\omega} + (k-1)\cdot\dfrac{T}{m}\cdot\dfrac{\dot{m}}{\omega} + \dfrac{\sum_{i\neq 1}(k\cdot T_i - T)\cdot\dot{m}_i}{m\cdot\omega} - (k-1)\dfrac{T}{V}\cdot\dfrac{dV}{d\varphi} \\
\dfrac{dp}{d\varphi} = z\cdot(k-1)\cdot\dfrac{3\cdot\alpha\cdot d}{\mathrm{r}\cdot\rho_{\mathrm{oil}}}\cdot\dfrac{m}{V}\cdot\dfrac{(T_{oil}-T)}{\omega} + z\cdot k\cdot\dfrac{p}{m}\cdot\dfrac{\dot{m}}{\omega} + \dfrac{p}{m}\cdot\dfrac{k\cdot\sum_{i\neq 1}T_i\cdot\dot{m}_i}{T\cdot\omega} - k\cdot\dfrac{p}{V}\cdot\dfrac{dV}{d\varphi} \\
\dfrac{dT_{oil}}{d\varphi} = \dfrac{3\cdot\alpha\cdot d}{c_{oil}\cdot\mathrm{r}\cdot\rho_{\mathrm{oil}}}\cdot\dfrac{(T-T_{oil})}{\omega} \\
\dfrac{dV}{d\varphi} = f(\varphi) \\
\dfrac{dm}{d\varphi} = \dfrac{\dot{m}}{\omega} \\
\dfrac{dL}{d\varphi} = p\cdot f(\varphi)
\end{cases}
\tag{9}
$$

In this system, T is the absolute temperature of the gas in the working chamber, K; P is the gas pressure in the working chamber, Pa; V is the volume of the working chamber, m^3; L is the expansion work, J; ω is the angular speed of the driving rotor rotation, deg/s; T_i is the temperature in the i-th chamber of the compressor; m is the mass of gas in the working chamber, kg/s; k is the coefficient of adiabatic compressed gas; $\dot{m}_i$ is the mass flow of gas leakage from the i-chamber to the working one, kg/s; z is the compressibility coefficient (which takes into account gas reality [15]); c_v is the specific mass isochoric heat capacity, J/(kg · k); L is the expansion work, J; α is the coefficient of heat transfer between a droplet of liquid and gas; T_{oil} is the liquid temperature; P_{oil} is the liquid density; m_l is the mass of liquid in the working chamber; r is the droplet radius; c_{oil} is the specific mass of liquid heat capacity, J/(kg · K).

Like the first stage, the structure of the calculated costs of power included the indicator power, determined by the numerical integration and taking into account the indicator losses on pushing the gas-oil mixture through the discharge port; pumping power costs for oil supply to the working chamber; hydro-mechanical losses on the movement of the gas-oil mixture in the working chamber. Such integral characteristics as the productivity, the average temperature of the discharge gas, and the indicator power were determined by numerical integration according to the known scheme, applied at calculation of volumetric action compressors [27]. According to the results of these calculations, the efficiency coefficient of the screw compressor is determined.

3. Results and discussion

The calculation of the working processes of the screw oil-injected compressor and its integral characteristics was carried out for the following initial data: the diameter of the driving rotor is 142 mm; the diameter of the secondary rotor is 110 mm; the length of the rotors is 236 mm; the main (male) rotor has five teeth, the gate (female) one has six spans (the spans profile was designed by "NIIturbokompressor" company, Kazan, Russia), the average height of the profile clearances is 80 µm, of the end clearances – 50 µm, on the tops of teeth – 100 µm; working gas is air or freon R134a.

The main calculation results are presented in figures 2 – 10. In the form of three-dimensional surfaces, the data on calculations of the efficiency coefficient dependence on the injected oil temperature T_{oil} and the mass share of oil *d* which was supplied into the working chamber of the compressor at the beginning of the compression process by an independent pump (or that the same, by displacement) at different speeds of the driving rotor.

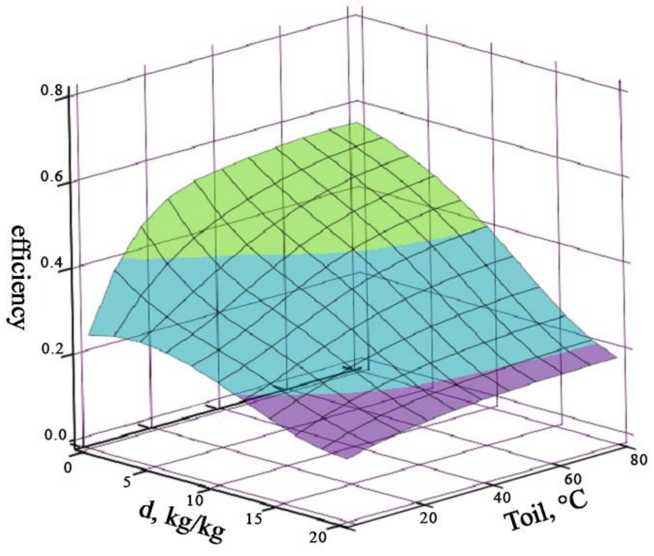

Figure 2. Dependence of the air screw compressor efficiency coefficient on the specific amount of oil in the working chamber d and the temperature of the injected oil Toil at the speed of the screw compressor driving rotor 3000 rpm; the suction pressure 0.1 MPa, and the discharge pressure 0.5 MPa.

As can be seen from the results presented in figure 1, it is not always necessary to strive for efficient cooling of oil in oil coolers. At low temperature of the injected oil its viscosity increases and therefore the pumping costs of power, hydro-mechanical losses and indicator losses in the injection process increase sharply. When there is a small amount of oil (at *d* from 1 to 5 kg/kg), the efficiency coefficient of the working process is drastically reduced by cooling the oil below 40^0 C, at *d* from 5 to 10 kg/kg this effect is also considerable. The large enough clearances in the working chamber, accepted for the calculation, allow to show the negative influence of excessive quantity of oil on the efficiency coefficient. As it is clearly seen in figure 1, at high rotor speeds, the increase in *d* results in a reduction of the compressor performance due to the rise of pumping power expenditure, hydro-mechanical losses in the working chamber and indicator losses during the discharge process. Only in a small range of low oil temperatures a slight increase in *d* provides a slight increase in efficiency by improving the sealing of the clearances. In the remaining range of *d* and T_{oil} at high rotor speeds, the positive effect of the oil on the clearances sealing is worse than the influence of the above losses.

Figures 2 and 3 present the results of the dependence of the screw oil-injected compressor efficiency coefficient on the temperature of the injected oil T_{oil} and the mass fraction of the oil *d* at the rotational speed of the driving rotor 2000 rpm and 1000 rpm. As it can be observed, at decrease in frequency of rotors speed at the expense of frequency regulation the character of oil quantity and temperature influence on the working processes efficiency changes noticeably. It is well seen that the reduction of the rotor speed results in the displacement of the optimal amount of oil towards the increase of *d*. It appears that the increase in *d* and the decrease in the speed make a non-linear reverse ratio: and non-linearity can be provided by automatic regulation of the amount of the oil supplied, if necessary. Note that during performance spool and throttle regulation at constant speed of rotors and constant oil consumption, the value *d* increases, which leads to deviation of the operation mode from a nominal one. The effect of T_{oil} when reducing the speed of rotation is the same: the recommended temperature of the injected oil is in the range of 40 – 80^0 C.

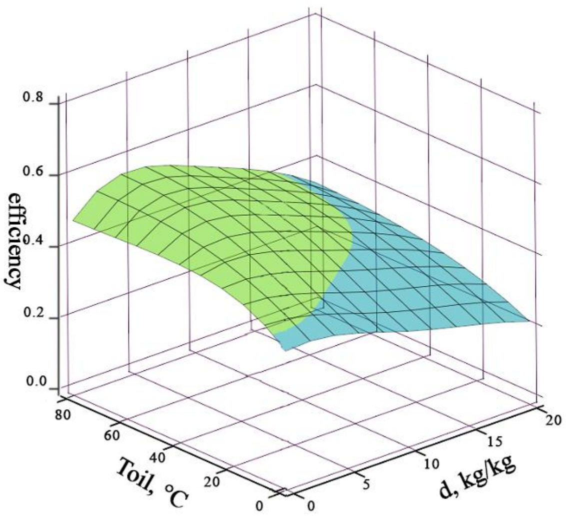

Figure 3. Dependence of the air screw compressor efficiency coefficient on the specific amount of oil in the working chamber d and the temperature of the injected oil Toil at the speed of the screw compressor driving rotor 2000 rpm; at the suction pressure 0.1 MPa, and the discharge pressure 0.5 MPa.

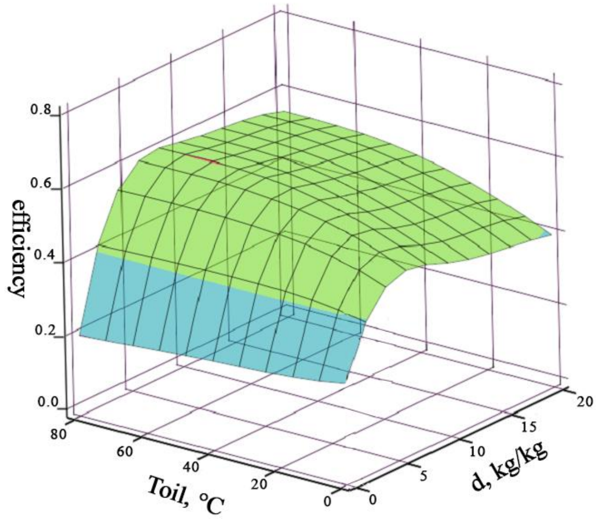

Figure 4. Dependence of the air screw compressor efficiency coefficient on the specific amount of oil in the working chamber d and the temperature of the injected oil Toil at the speed of the screw compressor driving rotor 1000 rpm; the suction pressure 0.1 MPa, and the discharge pressure 0.5 MPa.

Let us assess the influence of the specific amount of oil in the working chamber *d* and the temperature of the injected oil T_{oil} on the efficiency coefficient of the screw oil-injected compressor when compressing various working gases: air and freon R134a. Figures 4 and 5 present the results of calculations at the same suction pressure 0.196 MPa (which for freon R134a corresponds to the boiling point-10^0 C; at the same time, the gas temperature at the inlet of the compressor was taken as overheated by 5^0 C in relation to the boiling temperature); and at the discharge pressure of 1.0 MPa.

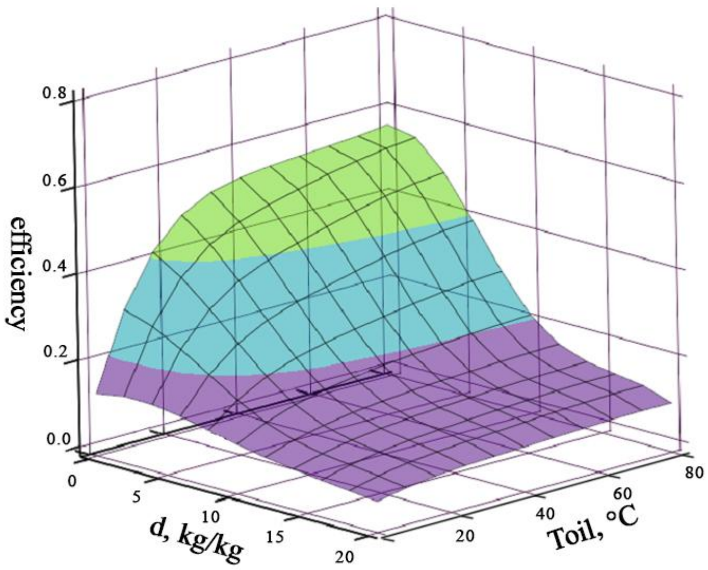

Figure 5. Dependence of the refrigeration screw compressor efficiency coefficient on the specific amount of oil in the working chamber d and the temperature of the injected oil Toil at the speed of the screw compressor driving rotor 3000 rpm; the suction pressure 0.196 MPa, and the discharge pressure 1.0 MPa.

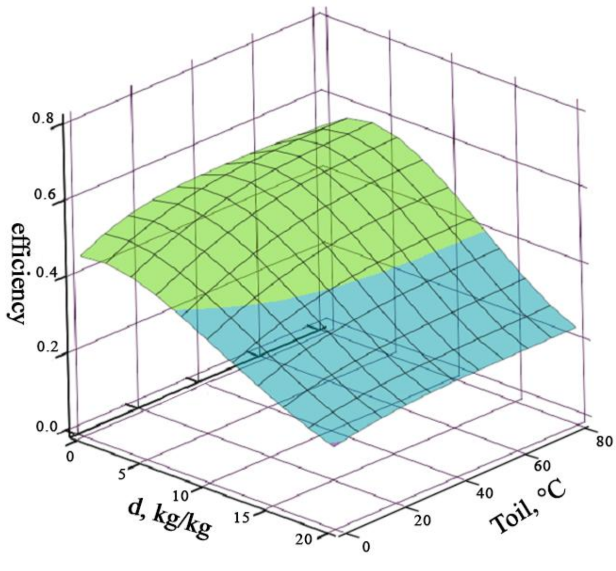

Figure 6. Dependence of the air screw compressor efficiency coefficient on the specific amount of oil in the working chamber d and the temperature of the injected oil Toil at the speed of the screw compressor driving rotor 3000 rpm; the suction pressure 0.196 MPa, and the discharge pressure 1.0 MPa.

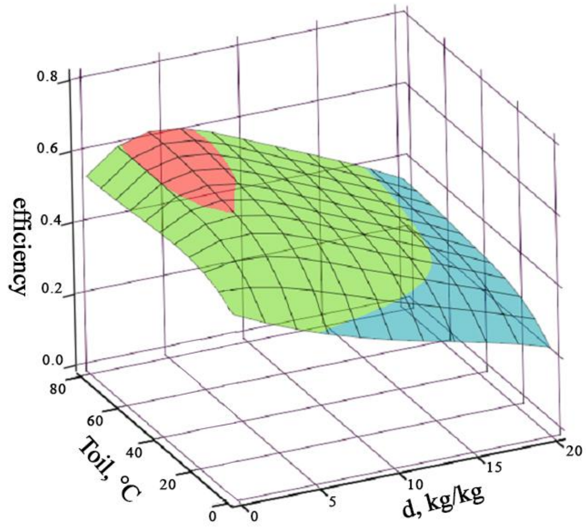

Figure 7. Dependence of the refrigeration screw compressor efficiency coefficient on the specific amount of oil in the working chamber d and the temperature of the injected oil Toil at the speed of the screw compressor driving rotor 1000 rpm; the suction pressure 0.196 MPa, and the discharge pressure 1.0 MPa.

As can be seen from the presented calculation results, in the refrigeration compressor, apparently, because of the working gas greater density, at smaller *d*, higher work efficiency is reached, comparing with air. Still, the qualitative influence of *d* and T_{oil} change on the efficiency coefficient of a screw oil-injected compressor stays the same for both working gases both at a nominal speed, and at its reduction. Note that the determination of the optimal *d* and T_{oil} for the nominal rotation speed is a priority, as the deviation of *d* and T_{oil} from the optimum can be even more critical when reducing the speed of rotation [5].

The influence of *d* and T_{oil} on the temperature of discharge gas is not less important. Let us consider the results presented in figures 8 and 9.

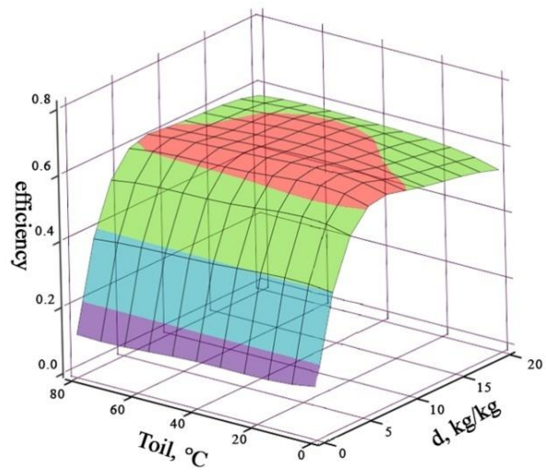

Figure 8. Dependence of the air screw compressor efficiency coefficient on the specific amount of oil in the working chamber d and the temperature of the injected oil Toil at the speed of the screw compressor driving rotor 1000 rpm; the suction pressure 0.196 MPa, and the discharge pressure 1.0 MPa.

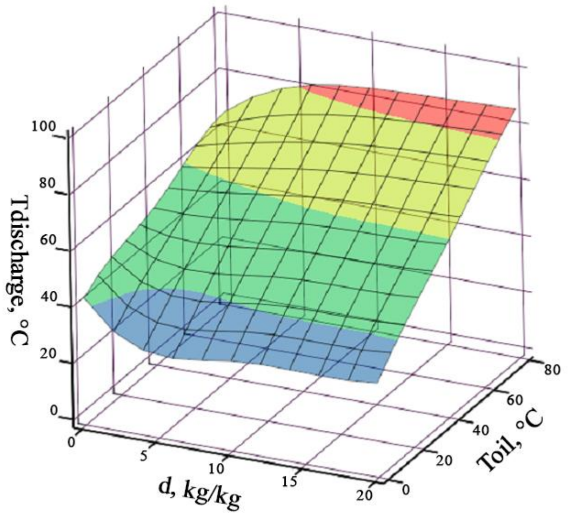

Figure 9. Dependence of the average discharge temperature of the refrigeration screw compressor on the specific amount of oil in the working chamber d and the temperature of the injected oil Toil at the speed of the screw compressor driving rotor 3000 rpm; the suction pressure 0.196 MPa, and the discharge pressure 1.0 MPa.

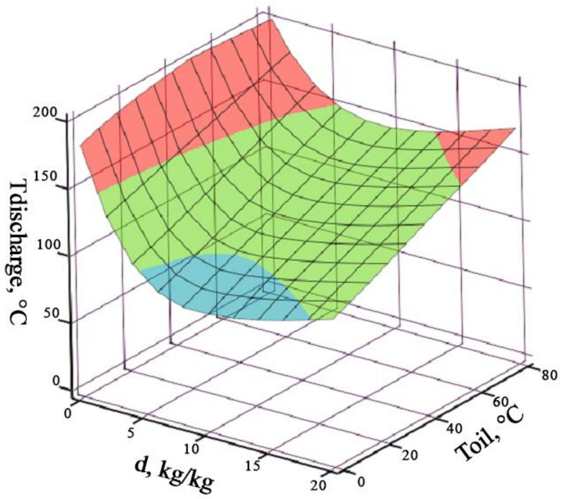

Figure 10. Dependence of the average discharge temperature of the air screw compressor on the specific amount of oil in the working chamber d and the temperature of the injected oil Toil at the speed of the screw compressor driving rotor 3000 rpm; the suction pressure 0.196 MPa, and the discharge pressure 1.0 MPa.

Analyzing the temperature mode, it can be noted that the temperature of the oil, of course, affects the temperature of the discharge gas. Therefore, the choice of the optimal value T_{oil} may in each case be determined depending on the prevailing criterion, in this case, either on the efficiency of work or on the temperature mode. The effect of the oil amount on the discharge temperature for the air compressor is more obvious. At low d the insufficient quantity of heat from compressed gas is transferred; at high d the pressure of the gas in the working chamber increases sharply, which leads to the growth of its temperature.

4. Conclusion

Thus, even taking into account the known errors of theoretical calculations, connected with accepted assumptions and uncertainty on certain conditions of ambiguity, qualitative analysis of a screw oil-injected compressor efficiency at frequency control modes has shown that with independent oil feed to the flow part (supply system or system with independent oil pump) has an advantage in comparison with spool regulation or regulation by throttling gas at suction. It means that with proper selection of the oil supply system flow characteristics, the rotors speed change does not result in any significant change in the screw compressor efficiency coefficient, because the value d is automatically changed and is slightly different from the optimal values over the entire range of speed variation. However, it sets strict requirements for the determination of the optimal value d at nominal rotors speed. The failure to comply with these requirements may lead to reduced efficiency by tens of percent both at nominal rotors speed and its decreasing in the process of compressor frequency control. Essential is the cooling system of oil, which temperature affects both the efficiency of the screw compressor and the temperature of the discharge gas; besides, the effects of injected oil temperature change for these characteristics may not be the same.

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