Simplified Engineering Calculation Methods of the Temperature Distribution Along the Screw Compressor Rotors

Timur Mustafin and Ruslan Yakupov

Abstract A purpose of a screw compressor experimental prototype development, rotor profiles design of optimization has many engineering problems. One of the main of them is trying to predict on this stage the compressor construction parameters, which are as close to the optimal as possible. Using of the compressor mathematical model for this aim sometimes is impossible, because of hardness of new mathematical model design and necessary of their accuracy checking. Therefore there is a necessary to have not only high-accuracy calculation tools, such as mathematical model, but also have some simplified engineering methods for the fast calculation of compressor parts' conditions, which have enough accuracy for the first stage of the development at same time. These conditions also include the temperature field of the compressor rotors. The presented paper dedicated to the analyses of possibility to obtain approximation equations, which can be used for the calculation of the preliminary value of the screw compressor rotors' temperature fields depending on the operation mode and construction parameters of the screw compressors.

Keywords Screw compressor · Rotors' temperature fields · Simplified engineering calculation methods

1 Introduction

Setting safe gaps in displacement compressors is an important issue since this determines not only the reliability of the unit but also its thermodynamic characteristics. When designing compressors, one has to compromise between reducing the probability of jamming, which requires increased gaps between the operating components,

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and reduced leakages, which is achieved by reducing the gaps. Setting safe gaps is impossible without calculating thermal deformations and, consequently, the temperature distribution among the compressor components. Even for compressors with simple-shape operating components, e.g. piston compressors, this is not a trivial task since it is required to correctly define the conditions in which heat exchange between the operating element and the walls of the working chamber occurs. In case of scroll or screw compressors, this task is further complicated by the complex shape of their operating elements. Thermal expansion of one scroll is compensated by thermal expansion of the second one in scroll compressors, screw compressors see the effects of thermal deformations of their rotors on the profile gaps added up.

Two methods for calculating thermal deformations of rotors can be distinguished nowadays: the method of calculation based on empirical dependencies [1] and the one based on building a mathematical model of the thermal state of rotors [2–6]. The purpose hereof is to approximate the data suitable for a preliminary rapid assessment of the thermal state of rotors.

2 Calculation of Temperature Fields of Rotors and Their Thermal Deformations

A whole range of works discusses the calculation of temperature fields of screw compressor rotors, with some of them revealing the impact of various factors on the temperature field of compressor rotors to some extent. The most comprehensive approach to calculating the temperature fields is presented in [2–8].

Thermal state of rotors is determined by the heat conduction equation:

$$\frac{\partial T}{\partial \tau} = \alpha \cdot \nabla^2 T, \tag{1}$$

where α is the thermal diffusivity coefficient of the rotor material, T is temperature, and ∇ is nabla. Thermal load on compressor rotors is cyclical. However, the period of a single cycle is rather short; as a result, the temperature field of the rotor does not virtually change with time. The latter is confirmed by the studies in [2, 3, 7]. This makes it possible to assume that thermal processes in a rotor are pseudostationary. Therefore, Eq. (1) can be simplified as follows:

$$\nabla^2 T = 0. \tag{2}$$

The boundary conditions for Eq. (2) are determined based on equation $\lambda_M \frac{\partial T}{\partial n} = q$, λ_M is the heat conductivity of the rotor material, where $\frac{\partial T}{\partial n}$ is the normal derivative with respect to the rotor surface which is affected by the heat flow q. However, one should distinguish between the heat gains from the heat exchange of the gas-oil medium with the rotor surface in the working cavity, the heat exchange of the

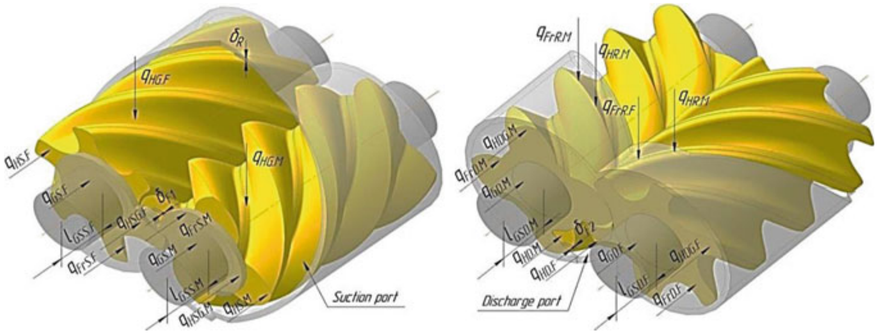


Fig. 1 Calculation scheme for determining the boundary conditions

end rotor surfaces with the gas-oil medium in the suction cavities through the end surfaces of the suction and discharge ports, the heat gain from the friction of the rotors against the gas-oil medium in the end surfaces and radial gaps, as well as the heat gain along the shaft sections from the bearings. All of these heat gains are reflected in the calculation scheme in Fig. 1 and are described in much detail in [7–12].

To set the boundary conditions when calculating the temperature field of the rotors, it is necessary to calculate the average gas temperature in the end section along the rotor length.

The rotor is divided into n sections for this purpose. The distance between the sections is equal to

$$\Delta z = \Delta\phi \cdot H, \tag{3}$$

where $\Delta\phi$ is the angular pitch of the mathematical model, and H is the screw pitch of the corresponding rotor (4).

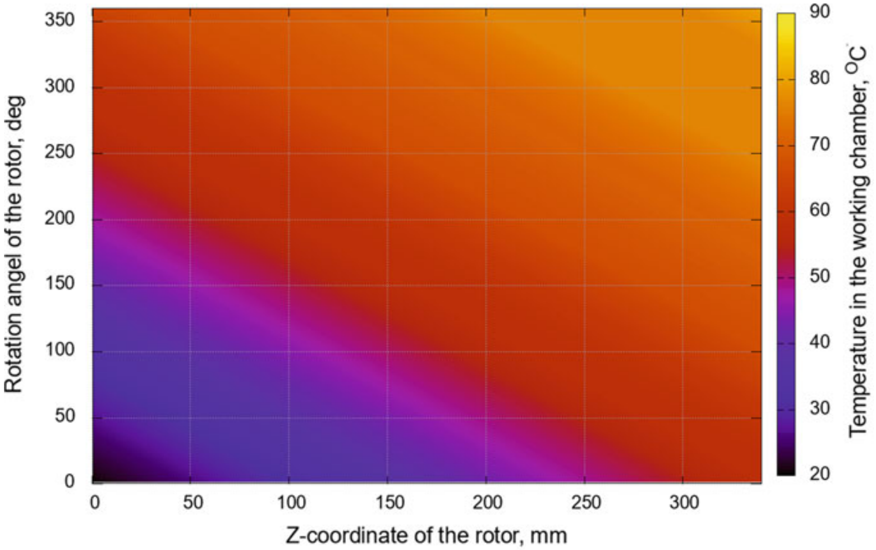
First, the temperature of the cavities at each angle of rotation of the rotor is determined for each rotor section. Since the same processes occur in the cavities shifted by a 90-degree angle, the temperature of one working cavity can be considered to determine the average gas temperature in each section.

The average air temperature in each section is calculated as

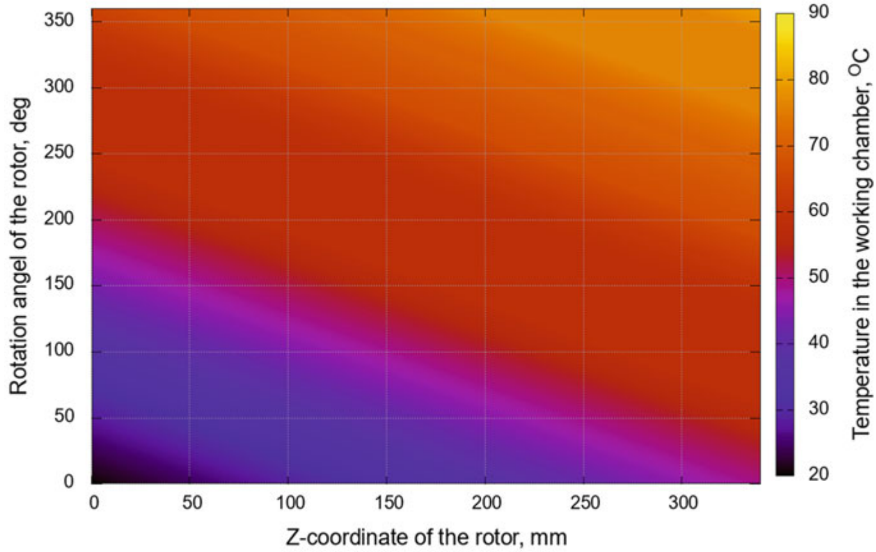
$$T_{CP} = \frac{\sum_{i=1}^{360/\Delta\phi} T_i}{360/\Delta\phi}, \tag{4}$$

where T_i is the air temperature in the working cavity of the section at the i angle of rotation of the rotor.

Figure 2 exemplifies the medium temperature distribution along the rotor length at each angle of rotation. Figure 3 exemplifies the distribution of the average medium temperature in the working cavity along the rotor length.



(a)



(b)

Fig. 2 Medium temperature distribution in the working cavity along the length of the male (**a**) and female (**b**) rotors and the angles of rotation of the rotors

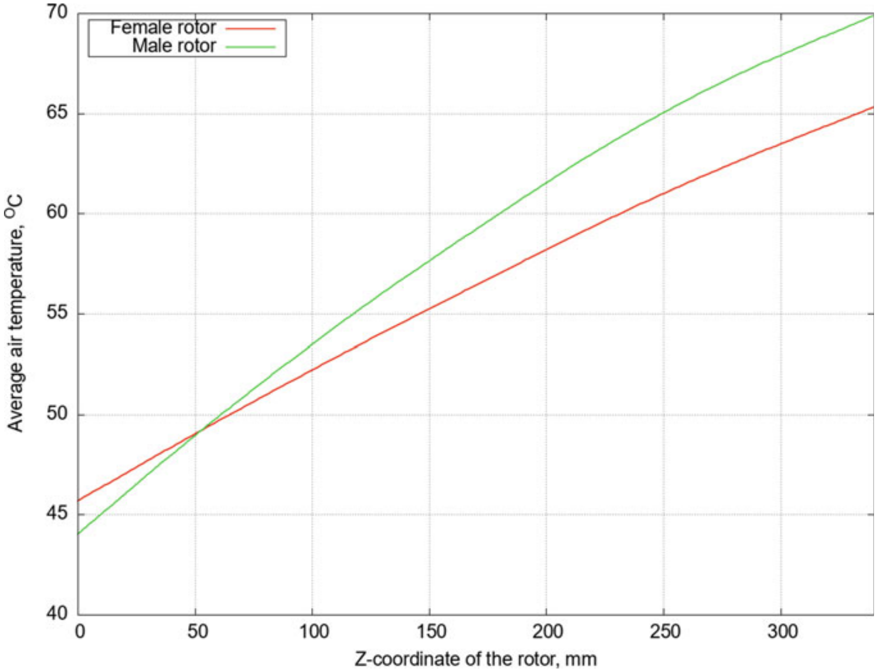


Fig. 3 Average medium temperature in the working cavity along the rotor length

2.1 Impact of the Compressor Design Parameters, As Well As the Operating Mode on the Thermal State of the Rotors

Figure 4 shows the impact of the design parameters as exemplified by the presence of the suction port on radial housing boring. The results of the temperature field calculations show that the radial tops of the rotors in the radial gap are subject to the most intense heating. It should be noted that the temperature of the gas-oil medium at the suction is lower; consequently, its viscosity is much higher, and therefore, the friction heat in the gap is also higher. This explains the higher temperature of the rotors in the housing without any suction port on the radial housing boring.

Similar conclusions can be made by analyzing the effect of the geometric compression ratio of the compressor on the temperature fields of the rotors. Its effect reflects not only through a change in the temperature of the compressed medium but also determines the discharge port dimensions. Significant geometric compression ratio is achieved by the smaller dimensions of the discharge port, which consequently leads to greater friction of the end surfaces of the rotor against the gas-oil mixture on the discharge side. The latter leads to additional heating of the rotor ends on the discharge side.

The impact of the values of the end and radial gaps on the temperature fields of the rotors is determined by the friction against the gas-oil mixture in them, which leads

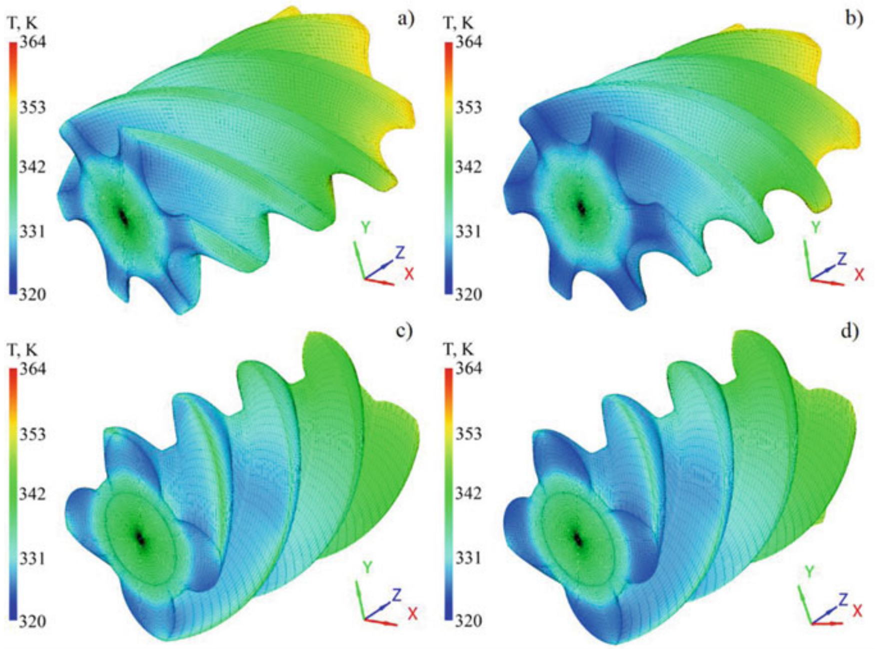


Fig. 4 Impact of the compressor design parameters on the temperature fields of the rotors (degree of compression P = 9, driving rotor speed of rotation n = 3000 rpm): **a** female rotor in a housing without any radial housing boring for the suction port; **b** female rotor in a housing with radial housing boring for the suction port; **c** male rotor in a housing without any radial housing boring for the suction port; **d** male rotor in a housing with radial housing boring for the suction port

to the conclusion that they are inversely proportional to the values of said gaps, all other things being equal. In fact, a smaller gap leads to an increased gas-oil mixture temperature in the gap and, therefore, to its viscosity drop. Thus, the smaller the gap the higher the friction heat, but its growth is non-linear.

This article uses the profiles of the female rotor without any seal ledges as an illustration for the calculations. The presence of the latter reduces the friction area on the radial surface against the gas-oil mixture, while the main heating along the radial surface from the friction heat can be seen on said ledges, which reduces the unevenness of the temperature field of the rotor in the end section as a whole.

The impact of the operating parameters on the temperature fields of the rotors is shown in Fig. 5 as exemplified by a driven rotor without any seal band in the housings and without radial boring for the suction port. It should be noted that their main effect shows through the temperature of the gas in the working cavities. This explains the higher temperature gradient between the suction and discharge ends of the rotors at high-pressure ratios. Reduced speed of rotation of the rotors leads to the equalization of the temperature of the rotors in different end sections, but the total rotor temperature increases at the same time. This can be explained by the fact that

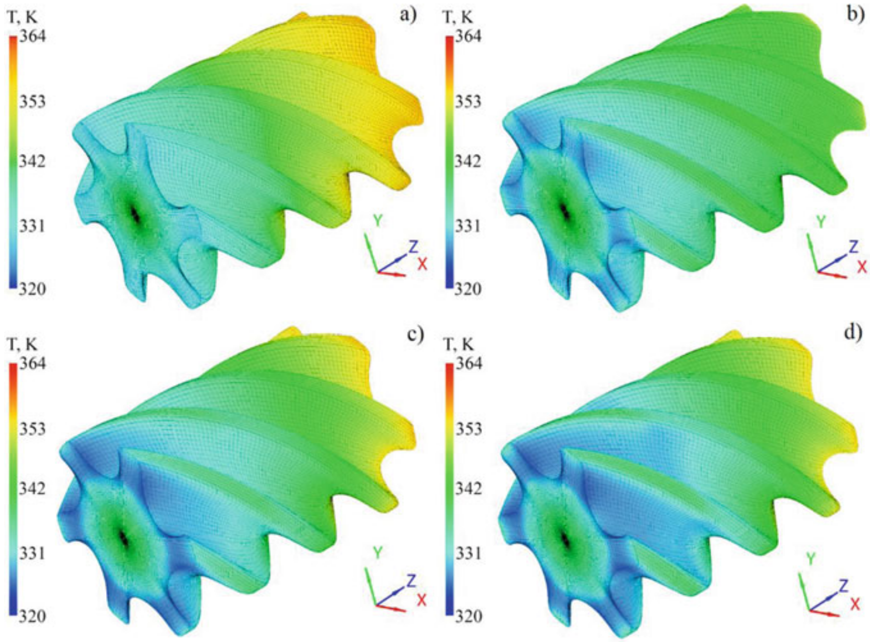


Fig. 5 Impact of the compressor operation mode on the temperature fields of the female rotor. **a** degree of compression P = 9, driving rotor speed of rotation n = 1500 rpm. **b** Degree of compression P = 4, driving rotor speed of rotation n = 3000 rpm. **c** Degree of compression P = 9, driving rotor speed of rotation n = 3000 rpm. **d** Degree of compression P = 9, driving rotor speed of rotation n = 3600 rpm

the smaller the rotor speed the higher the temperature of the compressed gas in the working cavities and the smaller the friction of the rotors against the gas-oil mixture in the gaps.

Figure 6 shows the effect of the amount of injected oil on the temperature field of the female rotor. The screw compressor without radial boring of the suction port in the cylinder block as described above was selected for the analysis. The suction and oil parameters also correspond to those described, and the assumed compression ratio is equal to P = 9, and while the speed of rotation is n = 1500 rpm. A change in the amount and temperature of the injected oil affects not only the temperature of the compressed medium in the working cavity but also its viscosity. The latter, as noted above, has a strong effect on the friction heat in the gaps and consequently affects the unevenness of the temperature fields of the rotors in their end sections. Less oil injected leads to decreased gas temperature on the suction side and to increased discharge-side temperature, which results in an increased temperature gradient in the rotor between the suction and discharge ends. More oil injected leads to decreased discharge-side temperature and increased gas temperature on the suction side. Further increase in the amount of gas injected does not lead to any significant temperature decrease on the discharge side, however, slightly increases the gas temperature on

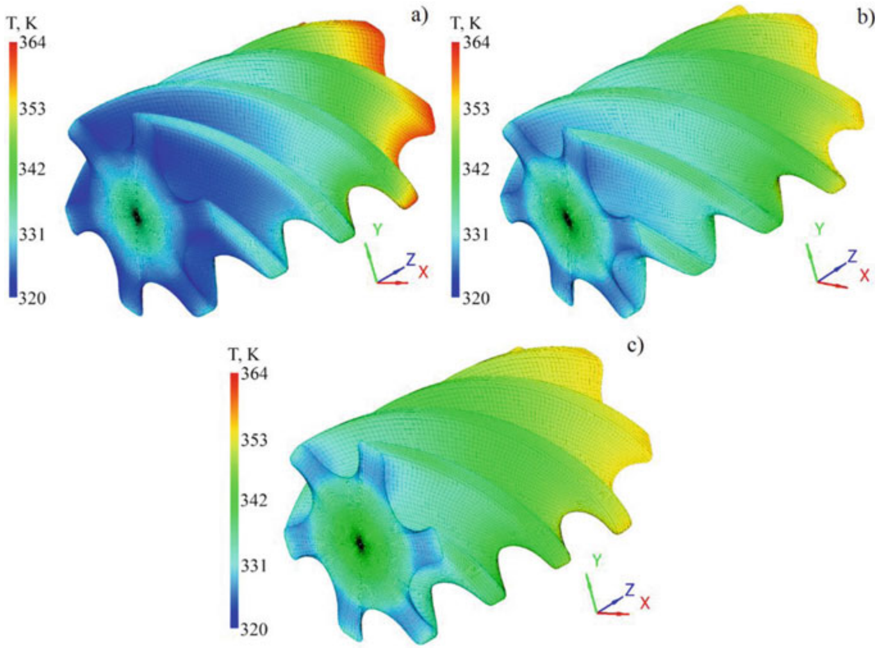


Fig. 6 Effect of the amount of injected oil on the temperature field of the female rotor (degree of compression P = 9, driving rotor speed of rotation n = 3000 rpm): **a** gas-oil ratio: 1.95. **b** Gas-oil ratio: 3.9. **c** Gas-oil ratio: 7.8

the suction side due to the initial temperature of the injected oil. This results in a more even temperature field throughout the rotor.

2.2 Calculation Results

As can be seen from the results, the thermal state of the rotors is determined by many factors, with their majority unable of being taken into account in the final empirical dependence. The central part of the rotor is affected by the heat gain from the bearings, beyond which the unevenness of the temperature field increases approaching the tops of the rotor lobes. This unevenness also decreases the higher the distances from the rotor ends. This is due to the effect of friction against the gas-oil mixture in the gaps. This makes it possible to determine—with some confidence—the following dependence for determining the average temperature of the rotors in the sections along the length of the rotor for any i section according to the following method:

$$T_{Mi} = (T_S - 1.35) + 0.45(T_D - T_S) + 20.76\left(\frac{l_i}{l_R}\right)^2 - 10.15\left(\frac{l_i}{l_R}\right) \quad (5)$$

$$T_{Fi} = (T_S + 3.39) + 0.40(T_D - T_S) + 32.49\left(\frac{l_i}{l_R}\right)^2 - 26.76\left(\frac{l_i}{l_R}\right) \quad (6)$$

where T_S, T_D, are the suction and discharge temperatures, respectively; l_i is the distance from the suction end to i section of the rotor, and l_R is the length of the profile part of the rotor. The Eq. (5) describes the male rotor average temperature in section, the Eq. (6) should be used for the female rotor average temperature in section calculation. All temperatures have °C dimension.

3 Conclusions

The computational model discussed enables the calculation of the temperature fields of the rotors: it makes it possible to assess the impact of both individual factors and the combination thereof, including not only the operating mode but also the compressor design features. The dependences for the approximate calculation of the average temperature in the rotor section have been obtained based on the approximation of the calculation data.

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