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The developed optimization methods for the geometric parameters of the piston rings enables a theoretical determination of a gliding profile for a proper friction behaviour and for a minimum piston ring wear.

The optimization of design parameters of the piston sealing enables either an efficiency increase of 25-30% or a reduction of the number of sealing elements.

The piston ring of a reciprocating compressor which usually consists of two or three piston rings and one oil scraper ring is one of the main limitations for the compressor's durability. For the optimization of the geometric parameters of this piston sealing and its elements a complex theoretical and experimental investigation of the working conditions of a piston ring in a refrigerant compressor was carried out in a wide range of design characteristics and operation modes. The principles of hydrodynamic lubrication theory have been applied for the theoretical analysis of the dependence of the oil film thickness on the compressor's operation mode and on the angle of rotation, respectively.

Hydrodynamic process of lubricating the friction couple piston ring - cylinder being analyzed preliminary has shown that a bilateral skewing of the gliding surface of the piston ring is necessary for the formation of the oil wedge in integration during the move of the piston at one revolution of the compressor's shaft. The carrying capacity of the oil layer in integration is determined by the geometrical parameters of such a bilateral conical skewing together with the gliding characteristics of the working process in the compressor's cylinder: this brings us to the idea of modelling and optimization.

For modelling the operation of a piston ring its gliding surface was described mathematically with a wide variation of its geometric parameters. The parabolic gliding profile was presupposed for a theoretical investigations because it offers the best opportunity for this variation.

The accepted parabolic profile of the piston ring misaligned from the median (Fig.1) offers opportunities for discussing the geometry of the gliding surface on the lubrication conditions in the integration piston ring - cylinder. For such a profile the equation of the oil layer's thickness between the cylinder and the piston ring acquires the form of:

$$H = h + a_0\left(\frac{z}{a}\right)^2 \qquad (1)$$

where H - the oil layer's thickness at the distance of z from the top of the parabolic profile of the piston ring's gliding surface, m; z - floating distance of the gap's cross-section along the height of the ring from the top of the profile, m; a - the width of the lower part of the piston ring's profile, m.

Reynolds' equation for computing hydraulic pressure's force in the oil layer being used we hold the lubricant's viscosity to be constant along the perimeter of the piston ring but the viscosity changes in dependence on temperature along the cylinder's element. We also hold that the system is symmetric in the direction of rotation, i.e. the problem is solved as being one-dimensional.

Now we are solving Reynolds' equation under the assumptions made. Then we find the median of the hydrodynamic pressure P_{rg} for two cases: when the piston moves from the lower dead centre to the upper dead centre

$$P_{rg} = \frac{1}{L}\left(\int_{-b}^{0} p\,dz + P_2 a\right) \qquad (2)$$

and when the piston goes from the upper dead centre towards the lower dead centre:

$$P_{rg} = \frac{1}{L}\left(\int_{0}^{a} p\,dz + P_1 b\right) \qquad (3)$$

where L - the piston ring's height, m; p - the oil pressure in the wedge gap; b - the width of the upper part of the piston ring's profile, m; P_1, P_2 - gas pressure above and under the piston ring, Pa. The minimal thickness between the oil layer between the piston ring and cylinder h of the rotation angle of the compressor's shaft α is being established as a result of mathematical transformations of equations and solution of Reynolds' equation. When the piston moves from the lower dead centre towards the upper dead centre

$$\frac{dh}{d\alpha}=\frac{Ah+Q_{\ell}\,\mathrm{arctg}\left(\frac{\ell}{a}\sqrt{\frac{a_0}{h}}\right)+T_{\ell}+\frac{3\mu v R_{\ell}}{L\gamma\ell}\left[\frac{\ell}{hB_1}+\frac{a}{h\sqrt{ha_0}}\,\mathrm{arctg}\left(\frac{\ell}{a}\sqrt{\frac{a_0}{h}}\right)\right]}{\frac{3\mu a\omega}{La_0}\left[\frac{\ell}{2hB_1}+\frac{a}{2h\sqrt{ha_0}}\,\mathrm{arctg}\left(\frac{\ell}{a}\sqrt{\frac{a_0}{h}}\right)+\frac{R_{\ell}}{\gamma_{\ell}}\left(\frac{1}{B_1^2}-\frac{1}{h^2}\right)+\frac{\ell}{h^2}\right]} \qquad (4)$$

where

$$A=\frac{Et_{\kappa}}{r_{\kappa}^2}$$

$$Q_{\ell}=-\frac{3\mu v a\ell}{Lh\sqrt{ha_0}}$$

$$T_{\ell}=p_2\left(\frac{R_{\ell}}{L\gamma_{\ell}}-1\right)-p_1\frac{R_{\ell}}{L\gamma_{\ell}}+p_r+p_{Tp}+p_{ynp}$$

$$\gamma_{\ell}=\frac{2h\ell}{B_1^2}+\frac{3\ell}{B_1}-\frac{3a}{\sqrt{ha_0}}\,\mathrm{arctg}\left(\frac{\ell}{a}\sqrt{\frac{a_0}{h}}\right)$$

$$R_{\ell}=\frac{a^2}{a_0}\left[\frac{h}{B_1}-\frac{1}{a}+3\ell\sqrt{\frac{a_0}{h}}\,\mathrm{arctg}\left(\frac{\ell}{a}\sqrt{\frac{a_0}{h}}\right)\right]$$

$$B_1=h+\frac{a_0}{a^2}\ell^2$$

E - elasticity module of the piston ring; t_{κ}, r_{κ} - radial thickness and the radius of the piston ring; μ - factor of the dynamic viskosity of oil; v - the velocity of the piston.

The values of the steam pressure above the piston ring p_1 and under it p_2 is used as the boundary conditions for determining the pressure in the lubricant layer.between the piston ring and the cylinder. A special theoretical investigation has been carried out aiming at establishing the mode of pressure distribution in the piston sealing of the refrigerant compressor. The balance of masses, the equation of gas energy, the equation of the ideal gas state, and also the equations for determining the instantaneous expenditure of gas through a slit have been used as initial equations for this investigation.

The piston sealing of a refrigerant compressor consists of one or more sealing elements, and neighbouring piston rings form boundaries for chambers in between them. Modelling the piston sealing operation we considered at first only one sealing element. The working cycle of the compressor was described by a schematic indicator diagram. The state of the gas between neighbouring piston rings and the current gas flow through the piston ring gap was described mathematically by a set of differential equations.

$$
\begin{cases}
\frac{dM_2}{d\alpha} = \frac{dM_1}{d\alpha} - \frac{dM_3}{d\alpha} \\
\frac{dT_2}{d\alpha} = \frac{1}{M_2}\left[T_1\left(k - \frac{T_2}{T_1}\right)\frac{dM_1}{d\alpha} - T_2(k-1)\frac{dM_3}{d\alpha}\right] \\
\frac{dP_2}{d\alpha} = \frac{R}{V_2}\left(T_2\frac{dM_2}{d\alpha} + M_2\frac{dT_2}{d\alpha}\right) \\
\frac{dM_1}{d\alpha} = \begin{cases} A\frac{P_1}{\sqrt{T_1}}\left(\frac{P_2}{P_1}\right)^{1/k}\sqrt{1-\left(\frac{P_2}{P_1}\right)^{(k-1)/k}} & \text{at } \frac{P_2}{P_1} > \beta_{\kappa p} \\ A\frac{P_1}{\sqrt{T_1}}\,\Psi & \text{at } \frac{P_2}{P_1} \le \beta_{\kappa p} \end{cases} \\
\frac{dM_3}{d\alpha} = \begin{cases} A\frac{f_2}{f_1}\frac{P_2}{\sqrt{T_2}}\left(\frac{P_3}{P_2}\right)^{1/k}\sqrt{1-\left(\frac{P_1}{P_2}\right)^{(k-1)/k}} & \text{at } \frac{P_3}{P_2} > \beta_{\kappa p} \\ A\frac{f_2}{f_1}\frac{P_2}{\sqrt{T_2}}\,\Psi & \text{at } \frac{P_3}{P_2} \le \beta_{\kappa p} \end{cases}
\end{cases}
\tag{5}
$$

where $$A = \frac{\varphi f_1}{\omega}\sqrt{\frac{2k}{(k-1)R}}$$

M – instantaneous expenditure of steam through the gap in the lock of the sealing ring; α – rotation angle of the shaft; k – adiabatic exponent; T – steam temperature; ω – angular velocity of the shaft; P – pressure of steam; V_2 – volume of the chamber between the rings; $\beta_{\kappa p}$ – crucial relation between the pressures; f_1 and f_2 – area of the gaps' sections in the locks of the first and the second sealing rings respectively; φ – expenditure factor; R – gas constant; Ψ – the factor determining the expenditure at $\frac{P_i}{P_{i+1}} \le \beta_{\kappa p}$

For each of the possible cases there is presented a system of differential equations describing the state of steam in the chamber between the rings which is analogous to the system of equations To establish the dependence of changes in the thermodynamic parameters of steam in the sealing element between the rings on the rotation angle of the shaft per one revolution it is necessary to solve the system of differential equations one by one which describes each case of the element operation depending on the stage of the working process in the compressor cylinder. If the piston sealing consists of three or more piston rings two or more combined sets of differential equations have to be established for each space between the sealing elements. By the application of highly efficient computer models it is possible to analyze the piston sealing operation in a wide range of working conditions of refrigerant compressors. The mathematical model was used to investigate compressors with cylinder dimensions between 67.5 mm and 115 mm.

Aiming at making the results of theoretical investigations more specific, all the computations according to the suggested formulas and their experimental checking have been done for a compressor without sealing, the cylinder diameter being 67.5 mm while working with the R22 refrigerant. In this compressor the ring sealing is set up of three rings: two compression and one oil scraper. We disregarded the sealing capacity of the oil scraper ring and therefore we considered the pressure above the upper compression ring to be equal to the pressure in the cylinder. We also consider the pressure under the lower compression ring to be equal to the pressure in the crankcase. We also assume that the chambers in the cylinder and those of the crankcase are connected with the space between the rings only by gaps in the locks of the rings. The changes in the thickness of the oil layer for one revolution of the compressor shaft were computed according to specially worked out programs.

Fig.2 shows the computed relations as to the minimal thickness of the oil layer h/Δ from the geometrical parameter of the steepness of the piston ring profile a_o/a^2. At each fixed relation of the geometrical parameters of the ring L/a the value of h/Δ is at the maximum at $a_o/a^2 = (1-3.3)\ 10^{-6}$. Fig.2 shows that the relationship also determines to a large degree the mode of friction in the integration piston ring - cylinder. The computations have shown that the parabolic gliding profile of the piston ring with the axial line skewed as to the median with $L/a = 3-4$ is to be considered optimal. This profile provides for the sufficient thickness of the oil layer in the integration ring - cylinder, the piston ring possessing the oil scraper capacity, that favours the operation of the compressor and the refrigeration machine as a whole.

The modelling of the single sealing element operation made it possible to establish the dependence of the relative pressure in the space between the rings and the relative mass expenditure of refrigerant through the gap in the lock in the lower piston ring upon the rotation angle of the shaft, the frequency of the shaft revolution and the distance between the sealing rings (Fig.3). This investigation showed that it is possible to reduce the number of piston rings from four to three and from three to two, respectively by an optimization of the piston sealing and the distance between the sealing elements, without any decrease in the volumetric and isentropic efficiency of the compressor.

Fig.--5 shows the influence of the distance between the sealing rings on the efficiency of the sealing operation at different frequency of the compressor shaft revolution. Thus, the optimization of the distances between the separate rings is a considerable reserve for raising the efficiency of the piston sealing operation.

The developed optimization methods for the geometric parameters of the piston rings enables a theoretical determination of a gliding profile for a proper friction behaviour and for a minimum piston ring wear. The gliding surface should be designed as a parabola with its axis 20-25% below the middle of the piston height to ensure hydrodynamic lubrication between the ring and the cylinder wall. Experimental investigations at a refrigerant compressor with standard and with new designed piston rings showed significant advantages for the piston rings with the recommended profile. The wear should be reduced by a factor of 2-2.5.

The optimization of design parameters of the piston sealing enables either an efficiency increase of 25-30% or a reduction of the number of sealing elements. Because of a considerable influence of leakage at the piston ring clearance on the efficiency of the piston sealing the piston ring design should be fitted well to the considered application which is especially important for R22 compressors for the low temperature range.

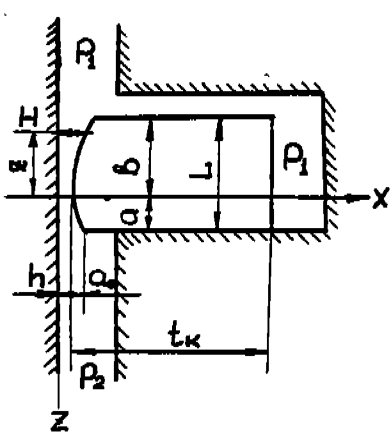


Fig.1 Profile of the piston ring's gliding surface

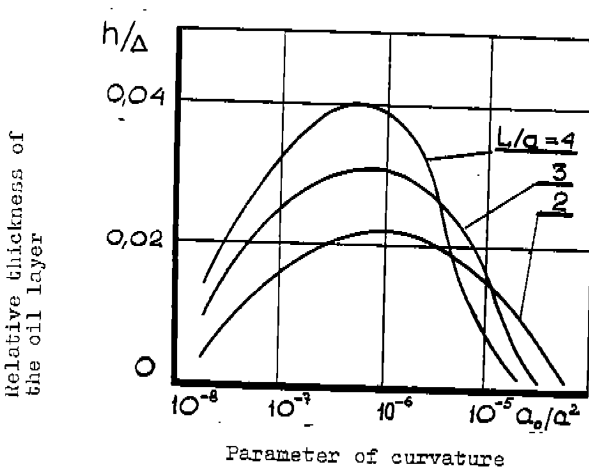


Fig.2 Computed dependence of the relative minimal oil layer h/Δ on the geometric parameter of the working profile's steepness a_0/a^2 of the piston ring at different values of the relation of geometric parameters of the profile L/a.

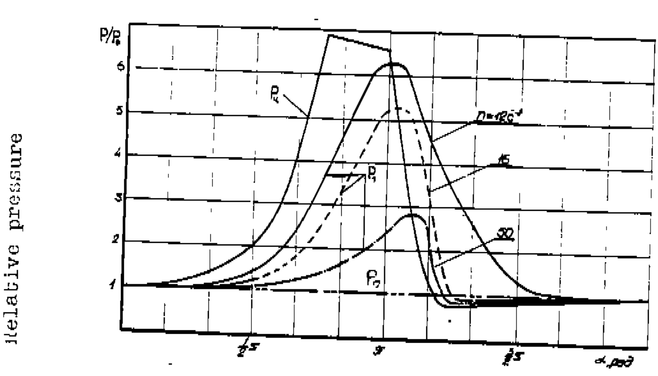
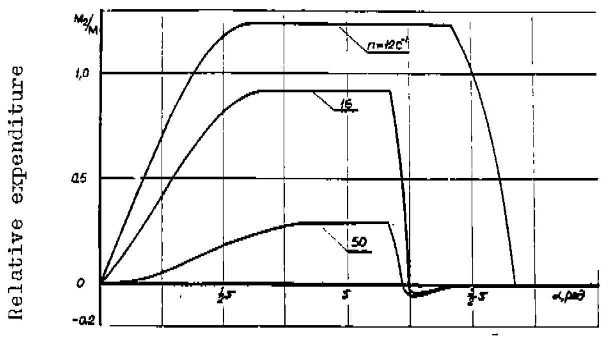
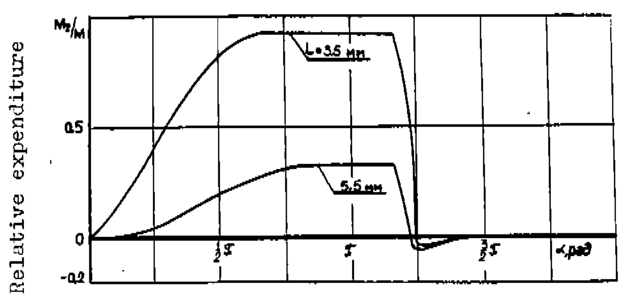


Fig.3 The dependence of the relative pressure in the between-therings cavity P_1/P_0 and in the cylinder of the compressor P_u/P_0 on the rotation angle of the shaft α at different frequence of the rotation of the shaft n :
P_u, P_1, P_0 -the pressure of steam in the cylinder, in the between-the-rings cavity and in the compressor's crankcase.



Angle of the shaft rotation

a



Angle of the shaft rotation

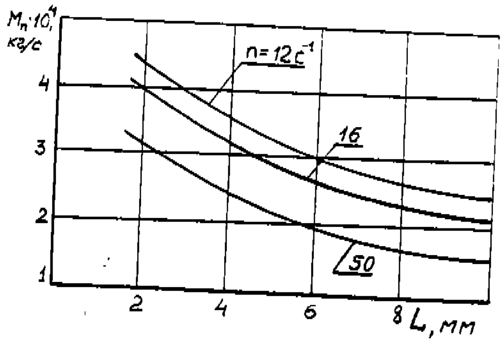
b

Fig.4 The dependence of the relative mass expenditure M_2/M through the gap of the lower sealing ring on the rotation angle of the shaft α.

a -at different values of the frequence of the rotation of the shaft n;

b -at different values of the difference between the sealing rings L; M - the mass of the steam in the between-the-rings cavity under the conditions of suction.

Amount of losses



Distance between the piston rings

Fig.5 The dependence of the value of steam leakages M_n through the piston packing of the compressor without sealings, the packing consisting of two rings, on the distance between the piston rings L at different rotation frequence of the shaft n, compression ratio π = 6.4 and the operation of the compressor using R 22 refrigerant.