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DYNAMIC ANALYSIS AND GEOMETRICAL OPTIMIZATION OF THE DETAILS  
OF ROTARY COMPRESSOR WITH ROLLING PISTON

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ABSTRACT:

The dynamic analysis of rotary compressor mechanism was realized by means of mathematical modelling of its operating processes with computer. The modelling procedure combined with variations of the geometrical parameters of the details proved the possible improvement of compressor efficiency and reliability. The results of modelling were used in the line of new refrigerating compressors with rolling piston manufacturing in Riga (Latvia).

NOMENCLATURE:

- $d_c$  - Diameter of internal surface of cylinder
- $d_e$  - Diameter of eccentric
- $d_R$  - Diameter of external surface of rotor
- $e$  - Eccentricity
- $F_{n1}, F_{t1}$  - Normal and tangential force components acting upon the rotor from the side of the eccentric, respectively
- $F_{n2}, F_{t2}$  - Normal and tangential force components acting upon the rotor from the side of cylinder wall
- $F_{n3}, F_{t3}$  - Normal and tangential components of the pressure force which the vane exerts on the rotor
- $F_{FR}$  - Resultant force of the refrigerant pressure exerted upon the vane
- $F_P$  - Force of the refrigerant pressure on the back edge of the vane
- $F_{SR}$  - Force of the compressed spring exerted upon the back edge of the vane
- $F_V$  - Force acting upon the vane from the side of the rotor
- $F_1, F_2$  - Normal force components acting upon the vane from the side of cylinder
- $P$  - Resultant force of the refrigerant pressure exerted on the rotor
- $P_M$  - Mechanical power which compressor consumes
- $P_m$  - Conditional unit pressure in the conjugation passage which rotor surface runs
- $V_R$  - Velocity of external rotor surface
- $\epsilon_i$  - Instantaneous angular acceleration of the rotor
- $\Delta_1$  - Instantaneous value of the clearance between the rotor and eccentric
- $\Delta_{1min}$  - Minimum operation clearance in the sliding bearing: rotor - eccentric
- $\Delta_2$  - Instantaneous value of the clearance between the rotor and the cylinder
- $\nu$  - Dynamic viscosity of the oil lubricating
- $(\alpha - \varphi)$  - The angle between the vector of the forces  $F_{n1}$  and  $F_{n2}$
- $\psi$  - The angle of rotation of the cylinder radius connecting its centre with the point of contact of the rotor and cylinder, relative to X-axis

$\omega$  - Rotational speed of the shaft

$\omega_R$  - Rotational speed of the rotor

$\omega_{RE}$  - Rotational speed of rotor relative to eccentric:  $\omega_{RE_i} = \omega - \omega_{R_i}$

## INTRODUCTION

The trend of using new refrigerants (R134a, R22 instead of R12) in the domestic, commercial and transport refrigerating units caused the thorough study of the rotary compressor with a rolling piston as the corresponding increase in the load exerted upon the rotor may result in excessive wear of the contacting surfaces. The same are the inner surface of a rotor and the edge of a vane.

To solve the problem of analyzing the forces acting in the mechanism was possible, and corresponding velocities of the contacting surfaces were defined. In spite of all this compressor efficiency is considered to be one of the most important subjects. This paper describes how mathematical modelling of dynamic processes in rotary compressors was realized with computer to optimize dimensions of the main details of the mechanism. As a result, a highly efficient, reliable and compact refrigerating rotary compressor was developed.

## CRITERIA OF OPTIMIZATION

There is a number of demands that define the relative standard of a compressor. The main of them are the following:

1. The compressor efficiency is to be as high as possible in a wide range of temperature conditions of refrigerating unit.
2. The reliability and the safety as well as the longevity of a compressor are to be high and long as possible.
3. The weight and dimensions indexes are to be minimal.
4. The standard of noise and vibration is to be as low as possible.

Performance of the mentioned demands is not to be accompanied by sharp rise of compressor cost. The terms "high or long or low as possible" mean objective limitations superimposed by the concrete conditions: the possibility of using inexpensive materials and accessible technology.

It is known that the mentioned demands are contradictory but computation of a simple optimal ratio for these demands is hardly possible because of its different nature. Count of each demand in economical calculations is possible and, in our opinion, it is expedient to optimize the mechanism conformably to each demand independently of the others.

### Main criterion of optimization

Proceeding from such an environment that energy efficiency im-

provement is strongly desired the compressor efficiency is considered as the main criterion of optimization.

Side by side with the coefficient of fullness defining the power effectiveness of a compressor, in practice it is customary to evaluate the efficiency of the whole refrigeration units as energy efficiency ratio, i.e., refrigerating capacity  $C$  to input power:

$$E.E.R. = \frac{C}{P_c + P_v} \quad (1)$$

where  $P_c, P_v$  is the input power of a compressor and that of a ventilator, respectively,  $W$ .

Even if  $SI$  is used the shortcoming of  $EER$  is evident as its value depends on evaporating and condensing temperatures hampering to compare tests results which are run in different temperature ranges. Therefore, it is more preferable to use the exergy efficiency ratio  $\eta_{EX}$  that is exergy of cold  $E_{CL}$  to input power [4]:

$$\eta_{EX} = \frac{E_{CL}}{P_c + P_v} \quad (2)$$

where

$$E_{CL} = \left( \frac{T_o}{T_{EV}} - 1 \right) \times C$$

$T_o$  - Ambient temperature, °K  
 $T_{EV}$  - Evaporating temperature, °K

(3)

It follows from equation 1, 2 and 3 that energy and exergy efficiency ratios are linked to each other with the temperature coefficient ( if  $SI$  is used):

$$\eta_{EX} = E.E.R. \times \left( \frac{T_o}{T_{EV}} - 1 \right) \quad (4)$$

#### The second criterion of optimization

High efficiency of a compressor is to be in concord with its reliability and longevity. It is known that by excessive wear the most vulnerable contacting surfaces of a mechanism are the inner surface of a rotor (rolling piston) and the edge of a vane. That is linked with the main peculiarity of a rolling piston compressor mechanism in which the sliding velocity of a rotor and an eccentric is not a guaranteed value but it is determined by friction forces exerted upon the rotor.

As the conjugation rotor - eccentric may be considered a sleeve bearing; therefore, all theoretical working out and experimental data for sleeve bearings may be applied to investigation of this conjugation.

The well-known condition to secure the liquid flow friction in a journal bearing may be defined by the inequality

$$\Delta_{1min} \geq 2(R_{z1} + R_{z2}) \quad (5)$$

where

$\Delta_{1min}$  - Minimal fluid-dynamic clearance in the conjugation  
 $R_{z1}, R_{z2}$  - Medium heights of micro-roughness of a journal and an insert, respectively.

The greater value  $\Delta_{1min}$  defines the reliable run not only in the rotor-eccentric conjugation but also, to a certain extent, it determines the wear intensity of a vane edge. The thing is that  $\Delta_{1min}$  depends not only on the load  $F_{nl}$  but on rotational speed of a rotor to an eccentric  $\omega_{RE}$  which determines friction loss in the conjugation vane - rotor.

Therefore, the value  $\Delta_{1min}$  is accepted as the second criterion of optimization.

#### Other criteria of optimization

As the sliding velocity of the rotor to the vane is comparatively not essential and rigidity of the details is sufficient the vibration of a vane and a rotor or high frequency noise are hardly possible. Therefore, minimal compressor dimensions and mass are linked chiefly with values of the main and second criteria.

The special criterion of optimization is to be accepted to form the vane edge profile and to find its width. It is possible to consider that wear intensity of a vane edge is directly proportional to multiplication of contact tension and sliding velocity in the conjugation rotor - vane:  $G_H \times V_R$ .

### GEOMETRICAL OPTIMIZATION OF THE MAIN DETAILS OF A COMPRESSOR MECHANISM

#### Simulation of dynamic processes in the compressor mechanism

Main details dimension ratios in a rotary compressor mechanism not only determine values of forces and relative velocities of contacting surfaces defining friction losses but also are connected with steadiness of mechanism work, its reliability and longevity. Decreasing the rotor sliding velocity relative to an eccentric causes decreasing the fluid-dynamic clearance in this conjugation and at the moment of maximal load a contact and even adherence of the surfaces become possible that may cause the damage of a compressor.

To optimize dimensions of the main details of the mechanism in accordance with criterions mentioned above the model of dynamic

processes in a compressor was worked out. One revolution of the compressor shaft was divided into small angles and a special programme provided the computations of instantaneous values of the forces acting upon the rotor and the vane. The computable schemes to define these forces at any angle of the shaft turn  $\Psi$  are shown in Fig.1, 2 and 3.

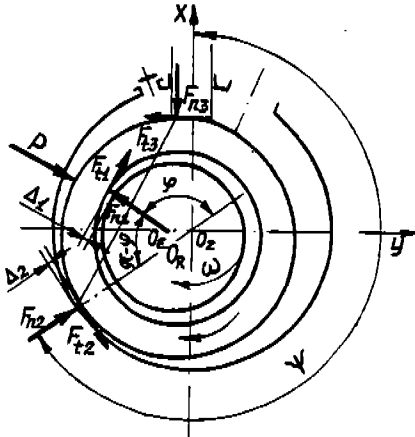


Fig.1. Calculation scheme for the forces exerted upon the rotor

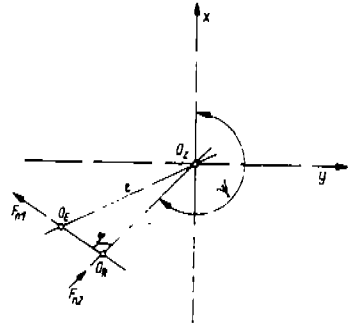


Fig.2. Layout of the rotor, eccentric and cylinder geometrical centres

In accordance with the D'Alembertian principle two equations were written down by projecting the forces, acting upon the rotor and its inertial force - on X and Y axes and one equation of the torques (Fig.1).

$$\begin{aligned} \sum F_{iX} &= 0 \\ \sum F_{iY} &= 0 \\ \sum M_{iO} &= 0 \end{aligned} \quad (6)$$

On deriving the system of equations it was assumed that the mechanical trajectory of a rotor is represented by the circumference with the centre of a cylinder and the instantaneous radius value  $(R_C - R_R - \Delta_2)$  where:

$\Delta_2$  - Instantaneous value of a clearance between the rotor and the cylinder

$R_C, R_R$  - Radii of the cylinder and of the external surface of the rotor, respectively.

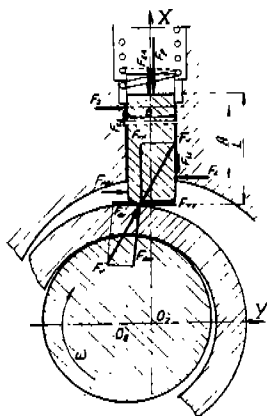


Fig. 3. Calculation scheme for the forces exerted upon the vane

On the rotation of the shaft the angle of an eccentric turn advances the angle of a rotor shifting  $\psi$  (Fig. 1) by a certain value.  $(\pi - \varphi)$ . The angle  $-\varphi$  can be expressed by applying the trigonometrical function, because the sides of triangle having the tops which serve as centres of the rotor  $O_R$ , of the eccentric  $O_E$  and that of the cylinder  $O_C$  (Fig. 2) are expressed by the geometrical sizes of the rotor, cylinder and the functional clearances. On adding the term  $\Delta_1$  we obtain:

$$\Delta_2 = R_c - R_r - \frac{e \sin \left\{ \pi - \varphi - \arcsin \left[ (R_1 - R_2 - \Delta_1) \frac{\sin \varphi}{e} \right] \right\}}{\sin \varphi} \quad (7)$$

where

$R_1, R_2$  - Radius of the internal rotor surface and that of the eccentric surface, respectively.

The value of  $\Delta_1$  can be determined by applying the existing methods of computation for the sleeve bearings [2]:

$$\Delta_{1 \min} = \xi (R_1 - R_2) \quad (8)$$

where

$\xi$  - Relative eccentricity of a sliding bearing which is defined in dependence of the bearing stress,  $C_F$ , and of the relative length of the journal:  $L / (2R_2)$ .

Instantaneous rotational speed of a rotor is a result of action of instantaneous angle acceleration  $\varepsilon_i$  and the previous value of this speed  $\omega_{R(i-1)}$ :

$$\omega_{Ri} = \omega_{R(i-1)} + \varepsilon_i \times \Delta t \quad (9)$$

where

$\Delta t$  - The time of  $\varepsilon_i$  action

$J_R$  - Rotor moment of inertia

$$\epsilon_c = \frac{F_{t1} R_L - F_{t2} R_R - F_{t3} R_R \operatorname{sign}[\omega(R_C - R_R - \Delta_2) \cos \psi + \omega_{i-1} R_R]}{J_R} R_R$$

Fig.4 presents an example of the usual plot of modelling parameters. The comparatively slight rotor velocity relative to the vane edge  $V_j$  and the corresponding length  $S_R$  that a rotor passes has been confirmed experimentally by means of accelerated filming.

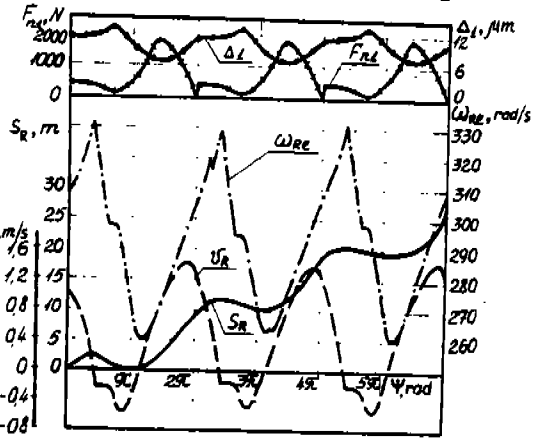


Fig.4. Change of the modelling parameters with the angle of shaft turn  $\psi$

$$(R22, V_0 = 13 \text{ cm}^3 (0.793 \text{ in}^3) \tau_R, \text{ m/s}$$

$$t_0 = -10^\circ \text{C} (14^\circ \text{F}),$$

$$t_c = 55^\circ \text{C} (131^\circ \text{F})$$

#### Modelling of vane dynamic

A block of equations of forces and torques similar to the equations (5) was derived in accordance with the computable scheme shown in Fig.3. Unified with the algorithm described above the common block of these equations made it possible to learn in details the reciprocity of a rotor and a vane. As a result, we saw a good accordance with the modelling and experimental data. Fig.5 shows a usual plot of unit friction power  $\sigma_H \times V_j$  in the contact of a vane edge and a rotor. The integrating value of this function during one revolution of the shaft was taken into account by optimizing the vane edge form and its thickness.

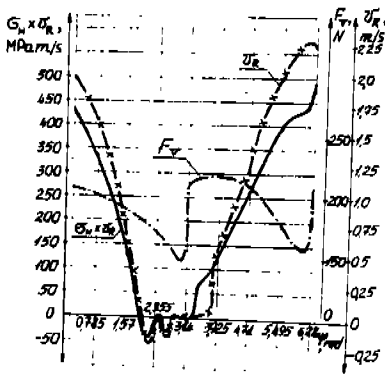


Fig.5. Change of the force of interaction between the vane and the rotor  $F_V$ , velocity of external rotor surface  $V_R$  and unit friction power  $\sigma_H \times V_R$  with angle of shaft turn  $\psi$  (R22,  $V_0 = 13 \text{ cm}^3$ ,  $t_0 = -15^\circ \text{C} (5^\circ \text{F})$ ,  $t_c = 30^\circ \text{C} (86^\circ \text{F})$ ,  $d_c = 48 \text{ mm} (1.89 \text{ in})$ ,  $e = 4 \text{ mm} (0.16 \text{ in})$ )



## Main results of geometrical optimization of the details

The described above dynamic analysis and modelling the processes in a rotary vane compressor with a computer is especially efficient due to the good possibility to optimize the mechanism varying values of geometrical parameters.

The following main problems are picked out in the number of solving ones:

- Ascertainment of the optimal value of the eccentricity
- Ascertainment of the optimal value of an eccentric diameter
- Ascertainment of the optimal value of a cylinder opening diameter
- Ascertainment of the optimal values of the vane thickness and its edge radius.

Such values have been ascertained conformably to a number of new rotary compressors with the following cylinder volumes: 7(0.427), 8.5(0.519), 11(0.671), 13(0.793)  $cm^3$  (inch<sup>3</sup>). There are two modifications of a compressor intended for R134a and R22. R22. The motor speed is 3000 r.p.m. in all units.

As a result of modelling the detailed information about interdependence of the parts dimensions and reliability of the most vulnerable conjugation rotor - eccentric is found. Separate curves are shown in Fig.6, 7 and 8.

The plot in Fig.8 proves that values  $\Delta_1$  and  $P_M$  are practically not changed with the value of an external rotor surface parameter  $d_R$  (cylinder volume  $V_0$  is fixed). That may be resumed with an important practical conclusion: the external diameter of a rotary compressor is determined chiefly by motor size.

The optimization of the vane thickness  $B$  and radius of its edge  $r$  has shown that there is an optimal ratio of these values:  $\frac{r}{B} \approx 2$ . The thickness  $B=6mm$  (0.236) is taken for all the compressors.

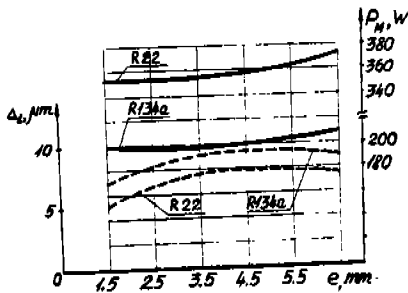


Fig.6. Change of minimal operating clearance between rotor and eccentric  $\Delta_{1min}$  and input of mechanical power  $P_M$  with the value of eccentricity  $e$  ( $V=13 cm^3$  (0.671  $in^3$ ),  $t = -15^\circ C$  (5F),  $t_c = 30^\circ C$  (86°F))

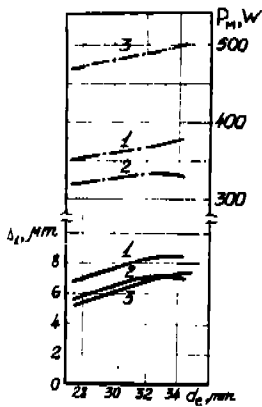


Fig. 7. Change of minimal operating clearance between rotor and eccentric  $\Delta_{1min}$  and input mechanical power  $P_M$  with the value of eccentric diameter  $d_e$  ( $V_0=13\text{cm}^3$ , 1-R12,  $t_0=+5^\circ\text{C}$  ( $41^\circ\text{F}$ ),  $t_c=55^\circ\text{C}$  ( $131^\circ\text{F}$ ); 2-R502,  $t_0=-35^\circ\text{C}$  ( $-31^\circ\text{F}$ ),  $t_c=55^\circ\text{C}$  ( $131^\circ\text{F}$ ); 3-R22,  $t_0=-10^\circ\text{C}$ ,  $t_c=55^\circ\text{C}$  ( $131^\circ\text{F}$ ))

New Riga refrigerating compressors

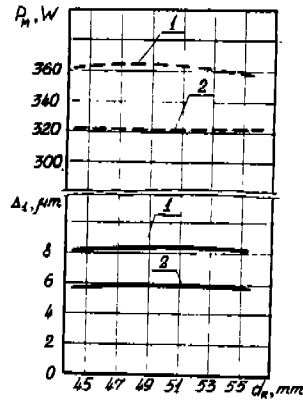


Fig. 8. Change of minimal operating clearance between rotor and eccentric  $\Delta_{1min}$  and input mechanical power  $P_M$  with the external rotor diameter  $d_R$  ( $V=13\text{cm}^3$ , 1-R12,  $t_0=+5^\circ\text{C}$  ( $41^\circ\text{F}$ ),  $t_c=55^\circ\text{C}$  ( $131^\circ\text{F}$ ); 2-R502,  $t_0=-35^\circ\text{C}$  ( $-31^\circ\text{F}$ ),  $t_c=55^\circ\text{C}$  ( $131^\circ\text{F}$ ))

Optimal dimensions of the main details of the mechanism found descriptively were realized in the number of new compressors developed at Riga compressor factory. These units are characterized by the improved efficiency, small-size and lightweight formation (20% reduction over the conventional), higher reliability even if ecology safe R22 is used as well as by low noise and vibration level.

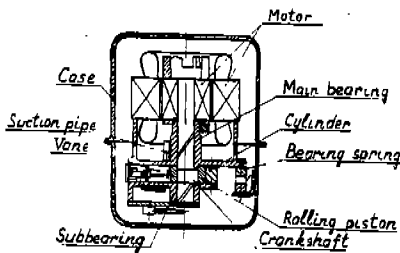


Fig. 9. Sectional view of Riga compressor

The basic structure of the compressor is shown in Fig. 9. Some of its specification and performance factors are listed in Table 1 indicating that in all the compressors the high efficiency

has been achieved at standard temperatures: evaporating  $t_0 = -15^{\circ}\text{C}$  ( $5^{\circ}\text{F}$ ),  
 condensing  $t_c = 30^{\circ}\text{C}$  ( $86^{\circ}\text{F}$ ), suction  $t_s = 20^{\circ}\text{C}$  ( $68^{\circ}\text{F}$ ).

Table 1

Capacity R134a W (BTU/h)	Power Supply (ph-Hz-V)	E.E.R. W/W (BTU/W)	Energy effici- ency ratio $\eta_{EX}$
315 (1,075)	1 - 50 - 220	2.3 (7.85)	0.312
400 (1,365)	1 - 50 - 220	2.3 (7.85)	0.312
500 (1,706)	1 - 50 - 220	2.3 (7.85)	0.312
630 (2,150)	1 - 50 - 220	2.3 (7.85)	0.312

There is a certain drop in efficiency because of the accepted mo-  
 motor cooling scheme with a refrigerant at suction pressure and low  
 temperature, respectively. It is known that the discharge pressure  
 in this case means decrease of heat exchange irreversibility loss  
 caused by less temperatures difference of the refrigerant and the  
 motor coil windings. But in the reviewed compressors suction pressu-  
 re in the case secures noise reduction in a simple and reliable de-  
 sign. The sizes and assembly dimensions of the new compressors are  
 shown in Fig.10.

Low noise and vibration level

Inside the sealed case the compressor is supported by 3 bearing  
 springs that reduces noise and particularly vibration so peculiar to  
 rotary compressors due to its light weight. A jet type muffler in the  
 suction line with the said bearing springs together make it possible  
 to obtain the noise level not more than 45dB at 1 m from the shell  
 surface and vibration index 100dB.

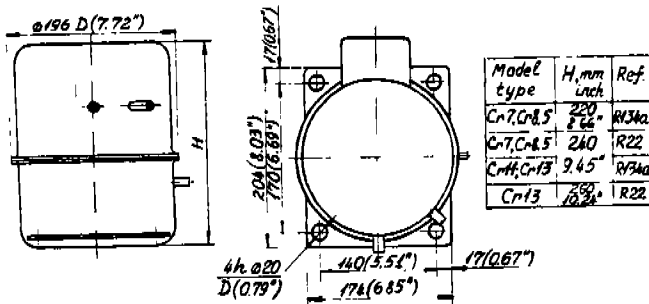


Fig.10. Dimensional Sketch

## CONCLUSIONS

New highly efficient and reliable rotary compressors were developed through investigations of relationship between dimensions of main parts and values of criteria of optimization. The noise and vibration level were considerably reduced due to bearing springs supporting the mechanism inside the case.

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