Power interactions of scroll compressor elements

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Abstract. The subject of our research is a scroll compressor. With its large number of advantages over reciprocating compressors in the form of overall, mass and production characteristics, they have their own difficulties associated with the theory of stability of a movable scroll during the operation process. As you know, a complex system of forces acts on the spiral, is very variable in magnitude and direction, for the period of one orbital revolution of the spiral. It should be noted that in modern designs of scroll compressors in the axial direction, a movable scroll has most often a one-sided rigid support - a thrust bearing. Moving the movable spiral in the opposite direction is limited only by the force of the gas axial pressure, which does not guarantee it saving its original position, at least within the gaps, touching the movable and stationary spirals, which is unacceptable. Next, we consider the basic calculation equations associated with the stability of a moving spiral.

1. Introduction

The emergence in 1985 of Scroll Compressors (SCC) in the Japanese market aroused great interest in commercial and industrial circles [1]. The Japanese company Hitachi managed to realize the idea of spiral machines, patented by the Frenchman Leon Kreuks back in 1905, although before that it had taken more than ten years of searching for structural forms and testing the technology for making the machine [2]. During this time, serial production of three standard sizes of SEC for domestic and industrial air conditioners with a cooling capacity of up to 12 kW, operating on R22 refrigerant was established [3-4].

Currently, a number of companies in the USA and Western Europe conduct intensive research and development. The cooling capacity of the SEC was significantly increased to 35-50 kW and higher [5-6].

The main advantage of SECs is their high energy, production and operational efficiency. Compared with low-performance piston refrigeration compressors, SECs have higher (by 10-15%) indicator efficiency and feed rate (20-30%); smaller dimensions (30-49%) and compressor mass (10-15%); 5 dB lower sound pressure level (A); fewer details and a number of other advantages [7-8].

One of the important questions of the theory of SEC is the stability of a mobile spiral (PSP) during compressor operation, since it is affected by a complex system of forces, variable in magnitude and direction, for a period of one orbital revolution of the spiral, namely [9-10]:

- gas tangential forces directed perpendicular to the axis of rotation of the PSP (and the axis of the eccentric shaft) and do not intersect the latter. They determine the effect of torque on the compressor shaft. The vector of these forces, variable in magnitude and direction;
- centrifugal forces $\sum R_C$ directed parallel to the eccentricity line and also perpendicular to the axis of the shaft. They are constant (at $\omega = \text{const}$) in magnitude and also variable in direction;
- gas axial forces $\sum R_A$ directed parallel to the shaft axis, variable in magnitude.

2. Materials and methods
Gas tangential and centrifugal forces create a tipping moment $M_{\text{OPR}}$ acting on the PSP. At the same time, the gas axial forces create a moment opposite to the overturning moment of the sign. We call it the moment of stability $M_y$. Obviously, it is necessary (Figure 1):

$$K_y \cdot M_{\text{OPR}} \leq M_y,$$

where the stability coefficient $K_U = 1.1 \div 1.2$.

![Figure 1](image.png)

**Figure 1.** The total forces and moments acting on the moving spiral at different D/h ratios with the same volume of W_B cells.

In Figure 1(a-b), two possible variants of movable spirals are presented, which provide the same given SPK productivity.

Let us compare these options at the same gas pressures, performance, and frequency of orbital motion of the PSP. It can be argued that (with the same scale of drawings):

$$M_{\text{OPR}}^a > M_{\text{OPR}}^b,$$  \hspace{1cm} (2)

$$M_y^a < M_y^b.$$  \hspace{1cm} (3)

Deflection of the tops of the feathers of spirals:

$$\Delta_{\text{PSP}}^a > \Delta_{\text{PSP}}^b.$$  \hspace{1cm} (4)

On the other hand:

$$\sum R_A^a > \sum R_A^b.$$  \hspace{1cm} (5)
3. The discussion of the results

Assuming the same loss as a first approximation, we assume that the theoretical volumetric performance of both options is the same:

$$V_T = W_B n_C,$$  \hspace{1cm} (7)

where, $W_B$ – the volume of two suction cells SPK; $n_C$ rotation frequency $W_B = f(D, h, t, \delta).$

In its turn:

$$D = f(\theta_\circ, t, \delta),$$ \hspace{1cm} (8)

where $t$ - spiral pitch, $h$ and $\delta$ the height and thickness of the spiral, respectively.

We can accept $\delta = \text{const}; \theta_\circ$ - twist angle of spirals.

We accept $\theta_\circ = 4\pi$ for both options.

Thus, the volume of the suction cells, which determines the theoretical volumetric productivity under the indicated conditions.

To ensure the remaining parameters of the SPK, to determine the forces and moments acting on the PSP and other elements of the compressor, it is necessary to reveal this dependence. According to the dependences obtained in the theory of SPK:

$$W_B f = 2h \left[ (f_{\varphi_2} - f_{\varphi_1}) - f_{CN} \right],$$  \hspace{1cm} (9)

where $(f_{\varphi_2} - f_{\varphi_1})$ - helix area between twist angles $\varphi_2$ and $\varphi_1$; $f_{CN}$ – the end area of the feather of the spirals within the specified angles.

Spiral area:

$$f_\varphi = 0.5 r_\circ^2 \int_{\varphi_1}^{\varphi_2} \varphi^2 dy = \frac{(\rho_2^3 - \rho_1^3)}{6 r_\circ}.$$ \hspace{1cm} (10)

where $r_\circ$ - radius of the main circle of the spiral; $\rho_1$ and $\rho_2$ the radius vectors of the spiral, relevant angles $\varphi_2$ and $\varphi_1$.

At $\varphi_2 = 4\pi$ area bounded by external spiral arcs $\varphi_2 = 4\pi$ and $\varphi_1 = 3\pi$, according to the formula (10), will be:

$$f_{4\pi} - f_{3\pi} = 6.17 \pi^3 r_\circ^2 = 4.84 r^2 = C_d t^2,$$ \hspace{1cm} (11)
as \( r_p = \frac{t}{2\pi} \).

Area:

\[
\int_{CN} = \delta S_{\phi i}^2 f(\varphi)
\]

where \( S_{\phi i}^2 f(\varphi) \) – spiral length between corners \( \varphi_2 = 4\pi \) and \( \varphi_1 = 3\pi \).

\[
S_{3\pi}^4 f(\varphi) = r_p \int_{3\pi}^{4\pi} \sqrt{1 + \varphi^2} \, d\varphi = 0.5r_p \left[ \varphi\sqrt{1 + \varphi^2} + \ln(\varphi + \sqrt{1 + \varphi^2}) \right]_{3\pi}^{4\pi} = c_1 r_p
\]

where \( c_1 = \text{const} \),

\[
\int_{CN} = \delta c_1 r_p = \frac{\delta c_1 t}{2\pi} = c_2 t
\]

where \( c_2 = \text{const} \), \( (c_2\) is a named number).

So:

\[
W_B = 2h(c_1 t^2 - c_2 t^2)
\]

According to equation (2), the volumetric theoretical productivity of the SPK:

\[
V_t = 2n_c h(c_o t^2 - c_2 t^2)
\]

This dependency allows you to build graphs \( t=f(h) \) and \( \varepsilon_o = f(h) \) at \( V_t = \text{const} \) and \( n_c = \text{const} \) (Figure 2, \( \varepsilon_o \) - radius of the orbital circle).

By choosing \( t \) and \( h \) you can calculate the dimensions of the spirals, find the required cell areas and their volumes, then the location of the cells relative to the tipping axes, determine the forces acting on the spiral, and the \( \sum R_c \) forces are reduced to the plane of the force \( \sum R_f \).

![Graph](image)

**Figure 2.** The relationship between the main parameters of the spiral

After that, we should calculate \( M_{\text{OPR}} \) and \( M_y \), build graphs in the function \( h \), find \( K_y \) (Figure 3-4).
When calculating $M_{OPR}$ you must first determine the resultant force vector $\sum R_y$ and $\sum R_c$, i.e. tipping force:

$$
\sum R = \sqrt{(\sum R_y)^2 + (\sum R_c)^2 + 2\sum R_y \sum R_c \cos(\sum R_y; \sum R_c)}
$$

(17)

and its direction.

**Figure 3.** The nature of the change in the forces acting on the moving spiral when changing $h$

**Figure 4.** Graph for determining the optimal value of the height of the ribs of the spirals

Next, several checks of the SRP for stability should be performed. To do this, without changing the thermal regime ($t_0$ and $t_K$), it is necessary to check the stability of the SRP at other angles of rotation of
the SRP, i.e. at other positions of the SRP in the limit of one orbital revolution. So, if the previous calculation was made, for example \( \varphi = 90^\circ \), it is necessary to do the same calculations with other values \( \varphi \), for example, through 45° or 90° from 0° to 2\( \pi \). We find \((M_{\text{OPR}})_{\text{max}} \) and \((M_{\text{OPR}})_{\text{min}} \), corresponding values \( M'_{y}, M'_{y} \) and \( K'_{y} \).

4. Conclusion
The determination of the optimal value of \( h \) does not end there, since under other thermal conditions the ratio between \( M_{y} \) and \( M_{\text{OPR}} \) (i.e., the \( K_{y} \) value) will be different. Since with a decrease in the pressure difference \( \Delta p = p_k - p_o \), the stability coefficient \( K_{y} \) will decrease.

In fact, with a decrease in \( \Delta p \), \( M_{y} \) also decreases in proportion to the change in \( \Delta p \) (for each position of the SRP on the orbital path).

\( M_{\text{OPR}} \) will also decrease, but in a smaller proportion, because of the two components of the overturning force, the tangential force \( \sum R_{r} \) decreases in proportion to change \( \Delta p \), and the total centrifugal force \( \sum R_{c} \) does not change (at \( \omega = \text{const} \)). Therefore, when decreasing \( \Delta p \) will decrease and \( K_{y} \).

From the foregoing, it follows that after a preliminary determination of the optimal value \( h \) it is necessary to check this parameter in other thermal conditions with lower values \( \Delta p \) and upon receipt \( K_{y} \geq K_{y} \) finally take the value \( h \).

5. Conflict of interest
The authors declare that they have no conflict of interest on the content of this paper.

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