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WEDGING IN ROTARY VANE COMPRESSOR AS A RESULT OF SELF-OSCILLATION OF THE VANES

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ABSTRACT:

Rotary vane compressor mechanism is regarded as an oscillatory system. Even slight imperfections of technology may cause the increase of noise or wedging of the mechanism. The method of normal oscillation forms is used to simulate self-oscillations of the vanes. The main measures directed to prevent the dangerous phenomenon are recommended.

The increase of noise, decrease of capacity and even wedging are watched in some rotary vane compressors if the speed of rotation of the shaft exceeds 1500 r.p.m. In this connection thorough study of the rotary vane compressor was required, which was realized by means of mathematical modelling the operating processes with computer for the compressor with half-axis of oval 36.5 and 28mm and 5 vanes 4.2 x 44 x 16.5 mm. One revolution of the compressor shaft was divided into small angles and a special program provided the computations of instantaneous values of the forces, acting upon the vanes.

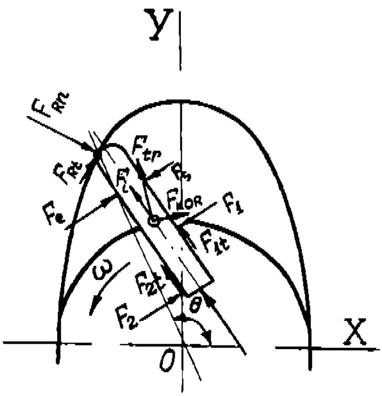

Fig.1 Calculation scheme for the forces exerted upon the vane

The computable scheme to define these forces at any angle of the shaft turn is shown in Fig.1. Solution of the according block of equations made it possible to compute its values as a function of angle θ. Fig.2 presents an usual plot of the forces needed to solve the said problem.

The computations proved that in wide range of the speed of rotation as well as in wide range of evaporating or condensing temperatures there is no danger for the vanes inclined in 8-12 deg. to the radius to be wedged in the slots even if the friction factor is up to 0.25.

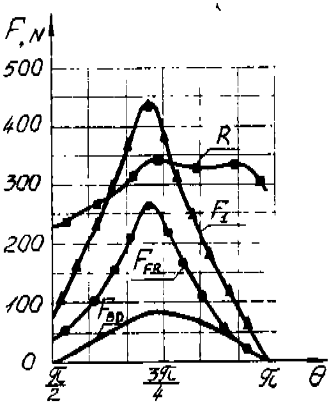

Fig.2. Change of the forces defining deformations of the vane and rotor segment with angle of shaft turn θ

(p_{suc} = 0.204MPa, p_{exh} = 1.51MPa)

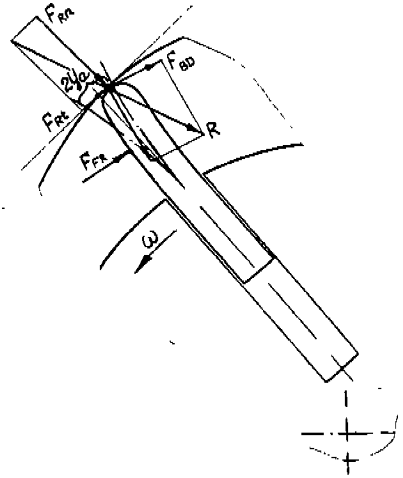

Fig.3. Vane deformation caused by force components F_{BD} and F_{FR}

To explain the excessive noise and the wedging at high speed range the possible vibration of the vanes was learned. At speed range above 1500 r.p.m. the intensity of the load application F_{BD}, F_{FR} (Fig.2) is approaching to a stroke. That stroke provokes the rapid bend of a vane as well as a part of segment of rotor (Fig.3, Fig.4) and certain oscillation of the cantilever with free frequency are unavoidable. These oscillations are damped in short time but its intensification is possible too if there is a variable force changing with the same frequency.The friction of the vane edge on the cylinder wall can become that very recurrant force because the friction factor is dependent upon relative velocity of contacting surfaces (Fig.5).This dangerous phenomenon may take place if the velocity of the edge of a vane in its oscillatory movement is approximating to the velocity of the edge in its rotational movement with the rotor together and there is a moment when the rubbing surfaces are relatively immovable.

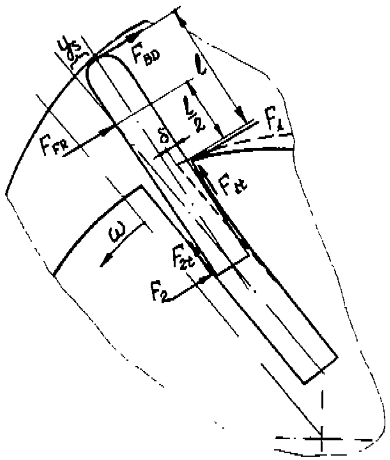

Fig.4. Rotor segment deformation δ and vane shift y_s

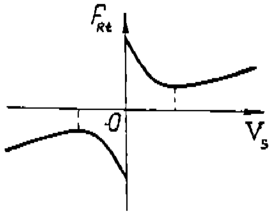

Fig.5. Friction force as a function of velocity

The maximum velocity V_{max} of the edge in its oscillatory movement is determined by the amplitude values of vibration y_{ai} and the natural frequency p_i of the cantilever that the projecting part of a vane is.

To simulate this phenomenon the method of normal oscillation forms was used [2], according to which the dynamic flexure y_a of the unjamed end of the cantilever rod with the length of l and changing load F_V applied to its free end may be defined by equation:

$$y_a = \frac{F_V\, l^3}{EI} \sum_{i=1}^{\infty} \frac{\beta_i \,(X_i)^2_{x=l}}{(k_i\, l)^4} , \qquad (1)$$

where:

X_i - Function defining i-th form of free oscillations of a pivot (main or normal function)

EI - Bending stiffness of a vane

k_i - Designation introducing into the differential equation for the calculation of transverse free oscillations of pivots

$$\frac{d^4 X}{dx^4} - \frac{p^2}{a^2} X = 0, \quad k_i^{\,4} = \frac{p^2}{a^2} ,$$

p_i - circular frequency of i-th oscillation form

$$a = \sqrt{\frac{EI}{\rho S}}$$

ρ - Density of vane material

S - Cross section of vane

β_i - Factor of intensification $\quad \beta_i = \dfrac{1}{1 - \frac{p_i}{\omega_i}}$;

ω_i - Conditional critical circular frequency of the cantilever: at such frequency maximum velocity of vane edge in its vibration movement is equal to its velocity in rotative movement.

To calculate values of normal functions X_i and frequencies p_i (or k_i) the common solution of differential equation for transverse oscillation may be used:

$$X = C_1 \sin kx + C_2 \cos kx + C_3 sh\, kx + C_4 ch\, kx, \qquad (2)$$

where constants C_1, C_2, C_3 and C_4 are arbitrary and ones must be determined at every definite case in accordance with conditions given at the ends of a pivot.

As at the moment of the stroke it is not only the vane that is bended but the part of rotor segment is bended too (Fig.3, Fig.4) the vane is considered to be a beam with elastic jamed one end and free another. Computable scheme (Fig.6) has two springs with stiffness factors r_1 and r_2 working for pressing and torsion conformably. The values of these stiffness factors were found in accordance with the well-known method [1]: $r_1 = 7{,}33\times10^8\ N/m$, $r_2 = 1{,}25\times10^5\ Nm/rad$.

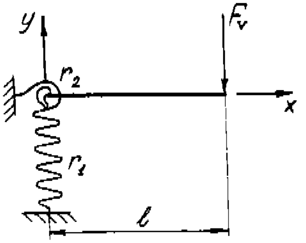

Fig.6.Computable scheme for the vane vibration

The end conditions may be expressed due to values of transverse load V and bending moment M [2]. In our case:

$$\begin{aligned} V_{x=0} &= EI(X''')_{x=0} = -r_1 (X)_{x=0} \\ M_{x=0} &= EI(X'')_{x=0} = r_2 (X')_{x=0} \\ V_{x=\ell} &= EI(X''')_{x=\ell} = 0 \\ M_{x=\ell} &= EI(X'')_{x=\ell} = 0 \end{aligned} \qquad (3)$$

Substitute equation X and its corresponding derivations into these conditions and the following block of 4 homogeneous equations can be defined:

$$\begin{aligned} &k^3 EI C_1 - r_1 C_2 - k^3 EI C_3 - r_1 C_4 = 0 \\ &-r_2 C_1 - k EI C_2 - r_2 C_3 + k EI C_4 = 0 \end{aligned} \qquad (3')$$

$$-\sin kl C_1 - \cos kl C_2 + sh\, kl C_3 + ch\, kl C_4 = 0$$
$$-\cos kl C_1 + \sin kl C_2 + ch\, kl C_3 + sh\, kl C_4 = 0$$

This block should have non-trivial solution only if the determinant of matrice defined with the coefficients attached to C_1, C_2, C_3 and C_4 is equal to nought. Then factorizing this determinant the frequency equation for the case can be defined. The file with the consequent roots of this equation for the first 4 forms of oscillations is represented below:

$k_1 l$	$k_2 l$	$k_3 l$	$k_4 l$
1.8335	4.5862	7.6493	10.641

The corresponding values of coefficients C_{ji} where i is an ordinal number of oscillation form are determined by substitution of the roots into equations (3'):

$$C_{ji} = \begin{bmatrix} -0.801 & -1.266 & -1.526 & -2.177 \\ 1 & 1 & 1 & 1 \\ 0.715 & 1.051 & 1.146 & 1.548 \\ -1.001 & -1.028 & -1.147 & -1.547 \end{bmatrix}$$

Substituting the roots of frequency equation $k_i l$ and coefficients C_{ji} back into equation (2) values of normal function X can be computed. First 3 forms of oscillations computed by this way are shown in Fig.7 and the values of X_i for the vane edge $(x = l)$ are presented below:

$$X_1 = -2.065 \quad X_2 = 2.248 \quad X_3 = -2.341 \quad X_4 = 22.601$$

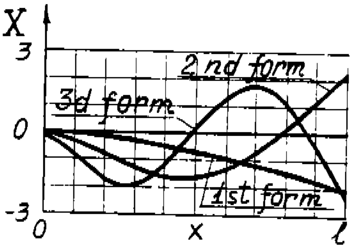

Fig.7. First three forms of vane oscillations

To compute actual deflection of the vane edge as a result of sudden applied load F_v it is necessary to the value y_{ai} defined by each item of equation (1) to add shift of the vane as absolute solid that takes place due to deformation of rotor segment and the turn of the vane to corresponding angle (Fig.4).

As a value of beam deflection caused by suddenly applied load is twice more than static one the summary shift of vane edge is found as:

$$y_j = 2(y_{aj} + y_{sj}) \qquad (4)$$

There is no problems to determine the shift of vane edge y_s in its turning movement as solid. The momental square method [1] can be used to establish deflection value δ of a loaded rotor segment. The maximum value δ caused by the load shown in Fig.2 is $6.02 \times 10^{-7} m$ that is in keeping with $y_s \approx 1 \times 10^{-6} m$. The value of y_{01} is similar. Obviously value y_1 is the initial amplitude that determines maximum value of vane edge velocity in its oscillation movement with circle frequency p_1:

$$V_{max\,1} = y_1 \cdot p_1 \qquad (5)$$

Separate items of expression (1) allow to find higher frequency amplitudes too. Though its values are decreasing quickly but considerably higher frequencies p_i necessitate computations of corresponding tangential velocities.

In Fig.8 the curves for maximum values of vane edge velocities V_{yi} in its vibration movement for first three forms are laid over tangential velocity V_t of the same edge but in its rotative movement with rotor together at n = 1500 r.p.m. The difference of values V_t and V_y is the stock that makes possible normal run of the mechanism without dangerous intensification of vanes vibration.

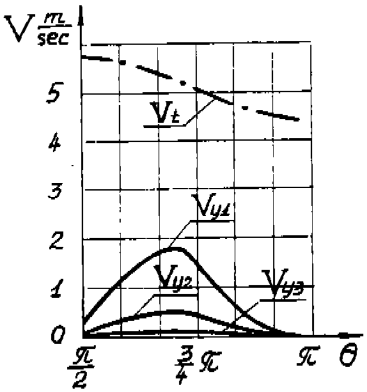

Fig.8. Change of vane edge velocities V_{yi} in its oscillative movement and in its tangential movement V_t with angle of shaft turn θ

Considerable value of such stock in Fig.8 reflects the assumed ideal case when the vane edge exactly follows cylinder wall while the friction factor in all conjugations is 0.1. In fact internal dynamical overloads are unavoidable because of strokes as a result of possible heavy run of vanes in slots. In this case the amplitudes y_i and corresponding velocities V_{yi} are increased. As internal dynamical overloads are displayed roughly at rising rotational speed, the critical value of the speed can be watched when noise is incresing essentially and even wedging of vanes can be the result. It takes place due to increasing of free oscillation amplitudes by synchronized changes of the friction force between vanes and cylinder wall that is selfoscillations of the vanes.

The following main measures directed to prevent that dangerous phenomenon arise from the performed analysis:

- Easy run of vanes in the slots of the rotor is to be guaranteed by careful execution of all necessary measures such as extreme precise manufacturing, plane smooth surfaces and optimal clearance in the conjugation.
- The materials of the vanes and the rotor are not to be inclined to wedging, the clearance in the conjugation must not be changed in wide range of temperatures.
- The surface of the cylinder wall is to be smooth enough to reduce the said fluctuation of the force exerted upon the edge of the vane.
- The optimization of vane edge profile performed in [3] to reduce friction loss also may appear highly effective to stave off vibration as the value of changing force maintaining oscillations would be diminished.

The recommended measures have been learned by the manufacturer of rotary-vane compressors and as a result of it some improving technology arrangements are inculcated. Following the improvements it became possible to enlarge speed range of the compressors.

For sure simulation of the oscillating processes in the investigated mechanism it is necessary to take into account the transverse shift of vane in the slot that needs to regard its non-linear oscillations. Continuation of the research work is to determine the dynamics of amplitude increase in the investigated oscillation processes and to predict possible failure conditioned by the wedging of the vanes in the slots.

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