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The Thermodynamic Parameters Analysis of Helical Compressors

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Abstract

As compared to the classical air compressors the helical compressors have gained more ground lately in the entire world owing to their advantages. In Romania this type of compressors is used in various economic fields Before starting the process of designing and developing the helical compressor, there are other more important steps to be followed, namely: to design and produce rotor gear profile, mathematical and numeric simulation of the real process of compression resulting into establishing the geometry with of the suction and cranking chambers and of the operational constructive and technical parameters.

Key words: *helical compressors, numeric simulation, technical parameters*

Introduction

The physical conversion of gas during the compression is done with the sensitive variation of the mass, due to the discharge phenomena which characterize the flow of the fluid through the functional slots between the rotors, and the compressors CAS.

If the effect of the mass waste were neglected, the conversion could be considered polytrophic and the only difficulty consists in establishing the polytrophic exponent owing to the mass variation of the gas.

The method suggested by this paper consists in establishing by numerical simulation of the real phenomenon, of the intermediate parameters of the gas as if the polytrophic exponent varied continuously during the gas conversion between over adiabatic or/and determining the functional parameters of helical compressors.

Before starting the process of designing and developing the helical compressor, there are other more important steps to be followed namely: to design and produce rotor gear profile, mathematical and numerical simulation of the real process of compression resulting into establishing the geometry at of the suction and cranking chambers and of the operational constructive and helical parameters. Among the companies that develop revolving helical compressor we must mention: Svenska Rotor Maschiner from Sweden, GHH from Germany, Aerzener from Austria.

The Analysis of Gas Compression Processes

The thermodynamic change from which the method begins is the adiabatic change. The gas is considered ideal, inheres the rotor speed being veer high it is considered that there is no heat exchange between the walls of the compression chambers and the gas.

The compression chambers is divided into precincts and the following equation can be written for each precinct.

$$d(p \cdot V) = D(G \cdot \mathrm{R} \cdot T) \tag{1}$$

$$p \cdot \mathrm{d}(V) + V \cdot \mathrm{d}(p) = \mathrm{d}(G) \cdot \mathrm{R} \cdot T + G \cdot \mathrm{R} \cdot \mathrm{d}(T) \tag{2}$$

$$\mathrm{d}(p) = \frac{G \cdot \mathrm{R} \cdot \mathrm{d}(T) + \mathrm{R} \cdot T \cdot \mathrm{d}(G) + p \cdot \mathrm{d}(V)}{V} \tag{3}$$

$$G = \mu \cdot F \cdot W \cdot \gamma \cdot t \tag{4}$$

p is the pressure, V − volume, G − gas mass, μ − consumption coefficient of the slot, considered invariable in a precinct, F − slot sectional area, W − flow speed through the slot, R − gas rate (R= 8310 J/Kmol K), t − time.

The relation (3) written for usual precinct i obtained by the division of the compression chambers in a finite number of m precincts.

$$\mathrm{d}(p_i) = \frac{(G_i + \mathrm{R} \cdot \mathrm{d}(T_i) + \mathrm{R} \cdot T_i \cdot \mathrm{d}(G_i) - p_i \cdot \mathrm{d}(V_i))}{V_i} \tag{5}$$

The Gas Mass During the Process

The discreet variation of the gas mass corresponding to a usual precinct i is:

$$\mathrm{d}(G_i) = \mu \cdot F_i \cdot W_i \cdot \gamma_i \cdot \mathrm{d}t \tag{6}$$

μ is invariable in precinct i, m − numbers of rotors, d(G_i) − gas mass in current precinct i, F_i− sectional area of the equivalent slot of communication between the current precint i and the two adjacencies i-1 and i+1 as well as the suction chamber.

This area is determinate by surmising all areas of the communication dots defined above as follows:

$$F_i = 4\left[F - \frac{(F \cdot i)}{m}\right] + F_{asp} + F_{tr} \tag{7}$$

$$F = \delta \cdot L_C = \delta \sqrt{\left(L^2 + \left(L + \frac{D_e}{1_p}\right)^2\right)} \tag{8}$$

δ is operational clearance between rotors, L − rotors length, 1_p − axial pitch of the leading rotor, F_{asp} − slot on the line of contact between the compression chambers and the suction cavity

$F_{asp} = \delta \cdot L_e$, F_{tr} − triangular slot $F_{tr} = dat$, ε − contact clearance between rotors, L_c − contact line length between rotors,

W_i − adiabatic flow speed of the gas through the slots in the case of below – critical ratio of pressures if :

$$p_i / p_{i+1} \geq \left[\frac{2}{K+1}\right]^{\frac{K}{K+1}} \tag{9}$$

$$W_i = \sqrt{2g \cdot R \cdot T_{i+1}\left(\frac{K}{K-1}\right) \cdot \left[1 - \left(\frac{p_i}{p_{i+1}}\right)^{\frac{K}{K+1}}\right]} \tag{10}$$

another:

$$W_i = \sqrt{2g \cdot R \cdot T_{i+1}\left(\frac{K}{K+1}\right)} \tag{11}$$

K is adiabatic exponent, g − gravitational acceleration, (p,T) − pressure and temperature followed by the current index of cavity, γ_i − gas density under pressure and temperature conditions

$$\gamma_i = p_{i+1} / (R \cdot T_{i+1}) \tag{12}$$

For the consumption coefficient of the slot there is the hypothesis that it is invariable whim the precinct *i* and it has an average value.

Setting the Mathematical Pattern

The mathematical pattern used to described the phenomena of gas compression and cranking comprises the gas next equation of the heat balance in the current cavity *i*, [1] :

$$dp_i = \left[\frac{(G_i R dT_i + RT_i dG_i - p_i dV_i)}{V_i}\right] \tag{13}$$

$$dT_i = dT_{ad} + dT_{intr} + dT_c + dT_t \tag{14}$$

dT_i is temperature function differential; dT_{ad} − temperature variation after the adiabatic compression:

$$dT_{ad} = \frac{K-1}{K} \cdot \frac{T_i}{p_i} dp_i \tag{15}$$

dT_{intr} − temperature variation in the cavity after filling it with gas of a higher temperature:

$$dT_{intr} = \frac{(GT)_i + d(GT)_i}{G_i + dG_{intr}} - T_i \tag{16}$$

dT_c − temperature variation by gas wrote out of cavity [1]:

$$dT_c = (K-1) \cdot \frac{T}{G} \cdot \Delta G \tag{17}$$

dT_t – temperature variation by gas wrote out of cavity, [6]:

$$dT_t = \alpha \frac{(T - T_{per}) \cdot F_i}{C_V} \tag{18}$$

Numerical Analysis of the Mathematical Pattern

The system of differential equations (13),(14) was integrated through Runge - Kutta method of fourth order witch Math software and the algorithm is applied several times in a cycle of iterations in order to calculate the gas parameters at each interval. The diagram emphasise the pressure and temperature waste during gas evolution, the increase in temperature during the process.

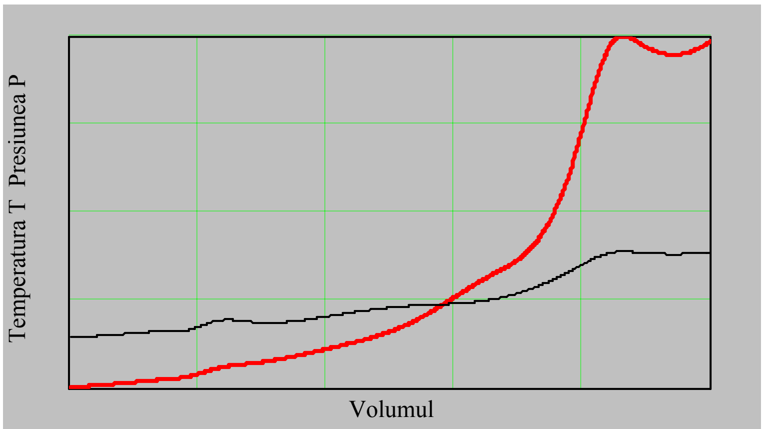

Fig. 1 Diagramele p(V) , T(V)

Conclusions

As for as the presented method is concerned, it lies the basis in the helical compressors research and design the technique used in solving the raised issues, aligns to the computer-assisted analysis and design of the complex systems. Its advantage consists in testing a great number of version. This analysis is also the starting point in structural analysis due to the fact that it establishes the exploitation conditions pressure and temperature, leading to an accurate proportioning.

References

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Analiza parametrilor funcţionali ai compresoarelor elicoidale

Contribuţiile aduse în acest articol referitor la studiul funcţionaliăţii compresoarelor elicoidale reprezintă o continuare a unei cercetari anterioare reşind de data aceasta punerea în evidenţă din punct de vedere fenomenologic a evoluţiilor presiunii şi temperaturii la refularea compresorului elicoidal ţinand cont de pierderile de masă de gaz in timpul comprimarii si de influenţa construcţiei compresorului asupra ***fenomenului de recomprimare*** *a gazului evacuat.*