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The Effect of Variations in the Amount of Working Medium Contained Within the Working Chamber on the Operation of a Rotary-Vane Vacuum Pump

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THE EFFECT OF VARIATIONS IN THE AMOUNT OF WORKING MEDIUM CONTAINED WITHIN THE WORKING CHAMBER ON THE OPERATION OF A ROTARY-VANE VACUUM PUMP

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ABSTRACT

In the paper, some problems related to the operation of rotary-vane vacuum pumps are analyzed. Geometrical relationships are discussed which are required to describe the operation of such pumps and thermodynamic processes that take place in their working chambers. Special attention is paid to the specificity of two-vane gas-ballast pumps. These pumps are used to compress gas mixtures which contain vapors, e.g., water vapor. The compression process of an increasing amount of gas contained within the working chamber is also discussed. A formula for the temperature and pressure of the gas is derived. On the basis of the formula, other parameters that characterize the operation of a vacuum pump are determined.

1. INTRODUCTION

Fig. 1 shows the cross section of a two-vane vacuum pump. It is made up of a cylinder and a rotor with radii of R and r, respectively. The axis of the rotor is shifted in relation to the cylinder's axis by eccentricity e. In radius grooves of the rotor two vanes operate (each of height h and thickness b) pressed to the surface of the cylinder by a spring and centrifugal force. The integral parts of the machine are two side covers which make its boundaries in the front side perpendicularly to the cylinder's axis. The vanes and the working part of the rotor are of length L. There are two ports on the surface of the cylinder: the inlet and outlet ports. The

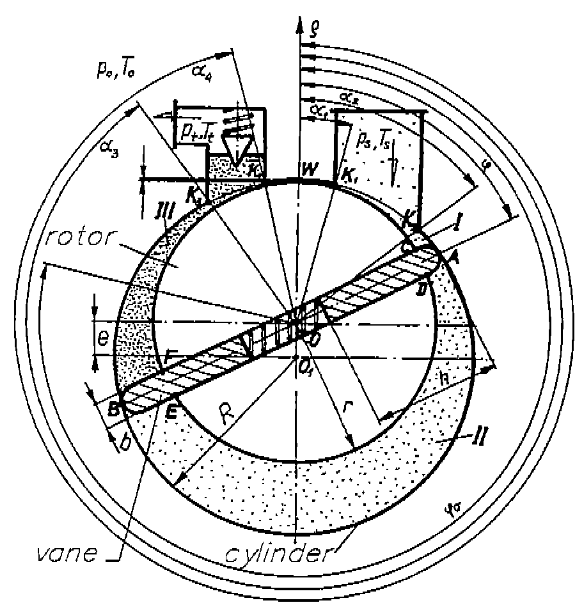

Fig. 1 Cross section of a two-vane vacuum pump.

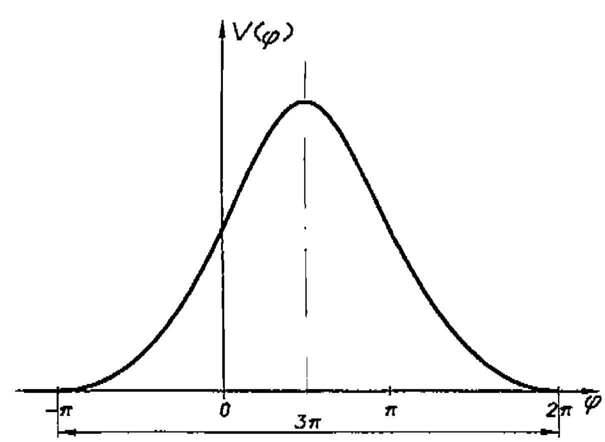

Fig. 2 Example of a dependence of a two-vanes vacuum pump's working chamber's volume on the position of the chamber

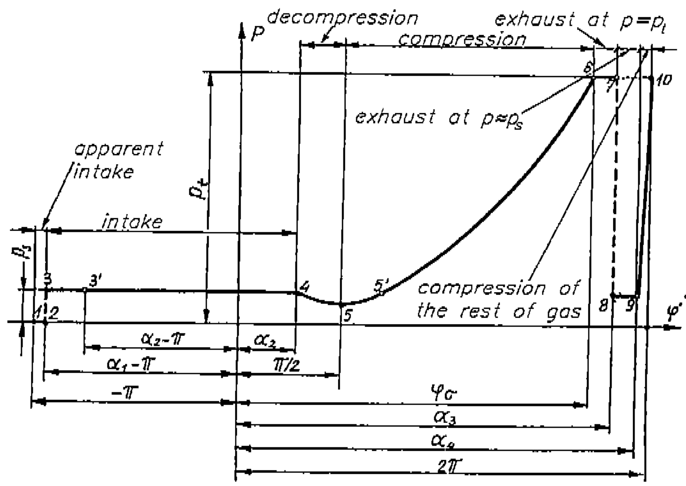

Fig. 3 Change of gas pressure in the working chamber of an ideal two-vane vacuum pump

latter has a valve whose head is pressed to the seat by a spring. The extreme position of the aforementioned ports is described by angles $\alpha_1 \div \alpha_4$. In order to describe geometrically the vane machines a polar coordinates system of ρ, φ related to the rotor was applied. The pole is the centre of the rotor and the polar axis crosses this point on the surface of the cylinder which is the closest to the pole.

The radius groove y is very small and it can be assumed that $y \approx 0$. This fact determines the existence of three spaces called working chambers in the rotary-vane pump. In the situation shown in Fig. 1, chamber I is the space confined by the surface of the cylinder (curve WA), surface of the rotor (curve WC), surface of the vane (line CA), and side covers. Chamber II is made up of the space contained between the cylinder and the rotor and the one between the vanes and the side covers (in the cross section figure ABEDA). Chamber III is confined by the surface of the cylinder (curve FW), vane (line BF), and the surface of the side covers. In the chambers, whose volume changes from $V = 0$ to $V_{\max}$ and again to $V = 0$, cyclic processes take place in intake, compression and exhaust of the working medium. To analyse them it is necessary to specify the way of describing the position of a chosen chamber within a full working cycle. In vane machines the position of the chamber can be described by giving the position of the vanes which confine it, especially the position of one of them. The most convenient way is to choose the vane which closes the analysed space according to the motion of the rotor [1].

The volume of the working chamber in any of its positions can be calculated using the factor specified for multi-vane machines, called a relative chamber's cross section area $Z(\varphi_u)$:

$$V(\varphi) = R^2 L Z(\varphi_u) \tag{1}$$

Fig. 2 shows the relation between the volume of the working chamber and its position.

The description of the processes taking place within the operation of a two-vanes vacuum pump and its energy effects is possible through the analysis of processes taking place in one working chamber within the full cycle of change of its volume. It can be assumed that the chambers are hermetic, and work put in overcoming the friction is insignificant.

Fig. 3 shows an example of changes of pressure in working chamber of a rotary-vane vacuum pump.

2. NECESSITY OF CHANGING THE AMOUNT OF WORKING MEDIUM IN WORKING CHAMBER IN SOME VACUUM PUMPS

The two-vane vacuum pumps enable one to achieve vacuum of absolute pressure p_s of 1 to 10 *Pa*. This means that the achieved external compression is 10^4 to 10^5.

Each volumetric compressing machine is also characterised by the internal degree of compression specified by formula [2, 3]

$$\sigma_w = \frac{p_{ks}}{p_{ps}}, \tag{2}$$

where:

p_{ks} — pressure in working chamber at the end of the compression process just before the valve or outlet port is opened,

p_{ps} — pressure in working chamber at the beginning of the compression process.

External compression is a function of the machine's geometry, formulated by the degree of compression ε and the type of process. The degree of compression is defined as follows:

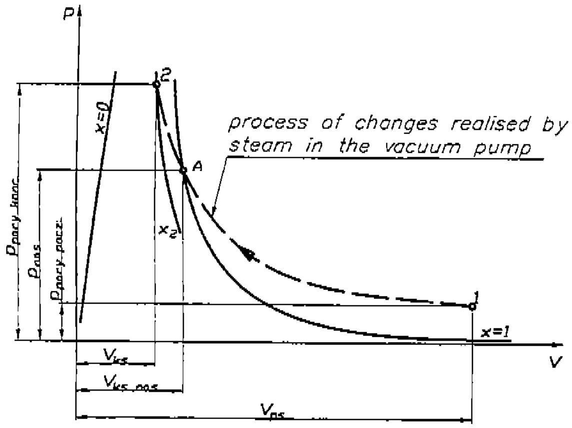

Fig. 4 Process of compression changes realised b steam in the mixture.

$$\varepsilon = \frac{V_{ps}}{V_{ks}}, \tag{3}$$

where V_{ps} and V_{ks} are the volumes of the working chamber in the aforementioned conditions.

The high value of compression at the process of sucking out different gas mixtures containing vapour disturb the operation of the pump, which consists in outdropping a volatile phase in the working chamber (Fig. 4). When the compression degree in the pump is higher than ε_n — the saturation compression degree ($\varepsilon_n = V_{ps}/V_{ksnas}$), then the final compression condition is in the area of humid vapour ($x_2 < 1$). On the other hand, without any intervention in the processes of compression of the whole gas mixture, it is impossible to achieve gas pressure which would allow for opening the outlet port before reaching the degree of saturation compression. The simplest form of such an intervention is the application of the so-called "blowdown". It consists in taking in ambient air into the chamber in the phase of compression [4, 5]. Fig. 5 shows the structural diagram of a two-vane blowdown vacuum pump. In its working area more and more quantity of gas is compressed. It makes it possible to achieve pressure p_t at the lower values of the compression degree. The classical description of the change processes taking place in the working chamber of a vacuum pump does not take the changes of the mass of compressed gas into account. An attempt of such a description is made in point 3.

3. PROCESSES OF CHANGES OF GAS IN THE WORKING CHAMBER OF A BLOWDOWN VACUUM PUMP

Analysing the thermodynamic system set by the balance shield (Fig. 5), which is the gas included in the working chamber, the following can be specified:

- mass of the gas m_p closed in it at the moment of the position of the chamber described by angle $\varphi = \alpha_2$; its parameters are: pressure p_s and temperature $T_0 = T_s$,
- m_a — the flux of the gas (air) mass flowing in from the environment; its parameters are p_0, T_0,
- $L_{z\pi}$ — work which has to be put into the system in order to change the volume of the working chamber,
- $Q_{z\pi}$ — heat exchanged with the environment,
- $E_{a\pi}$ — energy put into the system with blowdown gas,
- $E_{e\pi}$ — energy put out of the system including the flowing-out gas.

The thermodynamic parameters of gas at any of the positions of the working chamber are $m(\varphi)$, $T(\varphi)$, $p(\varphi)$, and its volume is $V(\varphi)$.

The basic factor, that brings about the change of gas parameters during compression, is the law specifying the change of the gas mass in the chamber during this process. The ratio of pressures $p(\varphi)/p_0$ is in a longer part of the compression time smaller than the critical pressure ratio. Therefore the mass flow rate $\dot{m}_a$ which is used for the blowdown of the pump and which is flowing through the non-return valve built into the wall of the working chamber can be calculated from the following equation [3]:

$$\dot{m}_a = \mu A_{otw} \psi_{s\max} \frac{p_0}{\sqrt{R_i T_0}}, \tag{4}$$

where:

μ — loss factor of the mass flow characteristic of the valve,

A_{otw} — area of a minimum cross section of the seat of a non-return valve,

$\psi_{s\max}$ — isentropic flow number.

Since these values do not depend on the position of the chamber, the mass of the gas present in it can be calculated from the following expressions:

$$m(\varphi) = m_p + \int_{\tau_p}^{\tau} \dot{m}_a d\tau, \tag{5}$$

or after taking Eqn. (4) and the relationship into account

$$d\tau = \frac{1}{2\pi n_{ob}} d\varphi, \tag{6}$$

from:

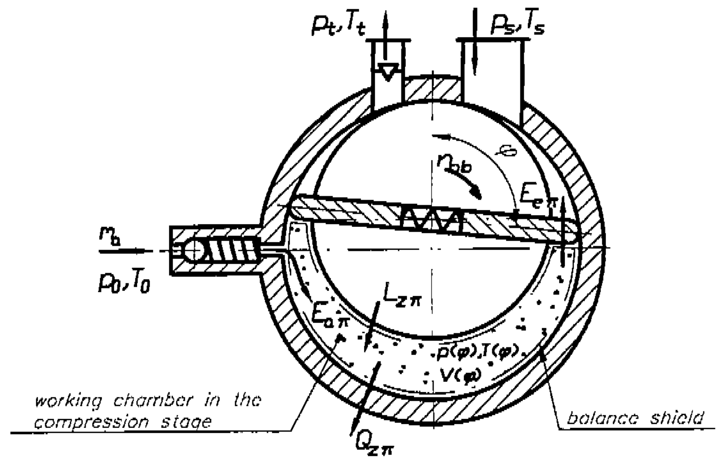

Fig. 5 Two-vanes blowdown vacuum pump.

$$m(\varphi) = m_p + \frac{\mu A_{otw} \psi_{s\max} (\varphi - \varphi_p)}{2\pi n_{ob}} \cdot \frac{p_0}{\sqrt{R_i T_0}}, \tag{7}$$

where:

φ_p — angle specifying the position of the chamber at which the blowdown starts,

n_{ob} — rotational speed of the rotor.

The thermodynamic gas parameters, i.e. pressure $p(\varphi)$ and temperature $T(\varphi)$ at any of the positions of working chamber can be determined from formulas [3]:

$$\frac{dp}{d\varphi} = \frac{k-1}{V(\varphi)} \left(\frac{dQ_{z\pi}}{d\varphi} + \frac{dE_{a\pi}}{d\varphi} - \frac{dE_{e\pi}}{d\varphi} - \frac{k}{k-1} \cdot \frac{dL_{z\pi}}{d\varphi} \right), \tag{8}$$

$$\frac{dT}{d\varphi} = (k-1) \left(R_i T \frac{dm}{d\varphi} - \frac{dL_{z\pi}}{d\varphi} \right) \frac{1-\varphi_\pi}{R_i m}, \tag{9}$$

where:

k — adiabatic exponent,

φ_π — ratio defined as

$$\varphi_\pi \frac{\frac{dQ_{z\pi}}{d\varphi} \omega - (T_0 - T) c_p \dot{m}_a}{\frac{dL_{z\pi}}{dgj} \omega - (\dot{m}_a - \dot{m}_e) p v}. \tag{10}$$

where:

ω — means the angular velocity of the rotor

$\dot{m}_e$ — mass flow rate of gas flowing out of the chamber,

c_p — specific heat,

v — specific volume.

For the pump described here the following were assumed: $dQ_{z\pi} \approx 0$ and $dE_{e\pi} \approx 0$.

The remaining derivatives in Eqns. (8) and (9) have the form of:

$$\frac{dE_{a\pi}}{d\varphi} = c_p T_0 \dot{m}_a \frac{1}{2\pi n_{ob}}, \tag{11}$$

$$\frac{dL_{z\pi}}{d\varphi} = pR^2 LZ'(\varphi), \tag{12}$$

where $Z'(\varphi)$ is a derivative of $Z(\varphi)$.

After introduction of Eqns. (10) to (12) into (8) and proper transformation one obtains:

$$\frac{dp}{d\varphi} + k\frac{Z'(\varphi)}{Z(\varphi)}p = \frac{(k-1)c_p T_0 \dot{m}_a}{2\pi R^2 L n_{ob}} \cdot \frac{1}{Z(\varphi)}. \tag{13}$$

The solution of this differential equation is of the form:

$$p(\varphi) = \left[\frac{1}{Z(\varphi)}\right]^k \left\{ p_s [Z(\alpha_2)]^k + \frac{(k-1)c_p T_0 \dot{m}_a}{2\pi R^2 L n_{ob}} \int_{\varphi_p}^{\varphi} [Z(\varphi)]^{k-1} d\varphi \right\}. \tag{14}$$

Whereas the following differential equation can be derived after transformation of Eqn. (9):

$$[T - a(\varphi)]\frac{dT}{d\varphi} - k(\varphi)T^2 + m(\varphi, p)T + n(\varphi, p) = 0, \tag{15}$$

where $a(\varphi)$, $k(\varphi)$, $m(\varphi, p)$, $n(\varphi, p)$ are functions of the chamber's position and gas pressure which include the properties of the medium, the valve characteristics, environment parameters, and others.

The convenient way to solve Eqn. (15) is a numeric one.

Using Eqn. (14) the analysis of work of an exhaust vacuum pump with blowdown can be made. Fig. 6 shows the processes of changes of gas compression in the pump, at different fluxes of the blown-in air.

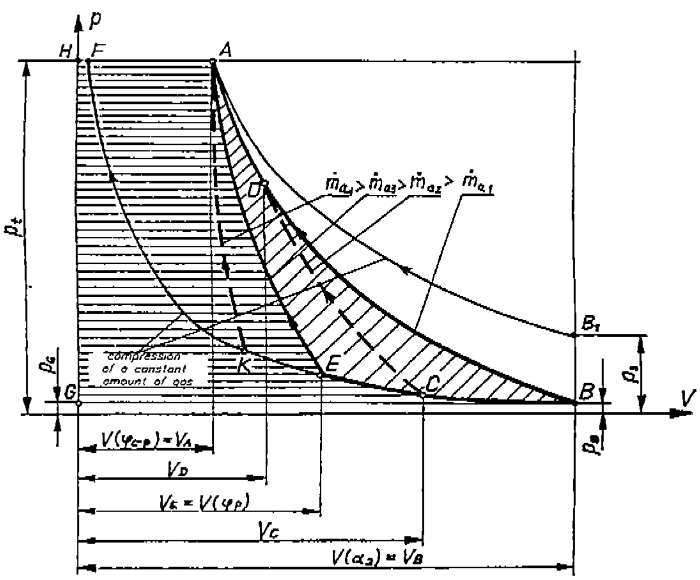

Fig. 6 Compression in a two-vanes blowdown vacuum pump.

At compression of a constant amount of medium, the gas has to be compressed to volume V_A to achieve the compression rate ε_n. This can be reached for the assumed compression processes (process B_1A), when the medium sucked in has the pressure $p_s = p_{B1}$. It is much higher than the pressure that one can obtain using the two-vanes pump, but it is the lowest at such compression of constant amount of medium that allows the condition A to be obtained. Compression of a constant amount of gas from pressure p_B will make it possible to obtain pressure p_t at volume V_F, which fails to fulfil the requirement of vapour non-outdropping. Supplying a constant flux of gas mass m_{a1} into the working chamber during the whole period of compression will permit the assumed volume V_A at pressure p_t. Technical work of compression of this changing amount of gas (area $BAHGB$) is bigger than technical work of the change process BF (area $BFHGB$). It is possible to lower the amount of this work and at the same time to obtain the volume V_A by changing the value of the stream of the mass of the flowing-in gas. In the beginning the working medium i compressed after BE which is part of the process BF. For such a position of the chamber, that the volume is $V_E = V(\varphi_p)$, a constant flux of gas mass $m_{a_3} > m_{a_1}$ is supplied into the system. Thus, the final state will be reached, and the technical work of the process BEA will be smaller than that of the process BA. Accepting the flux of gas mass $m_{a4} > m_{a3}$ makes it possible to open the non-return valve later, and for m_{a2} fulfilling the requirement $m_{a1} < m_{a2} < m_{a3}$ the non-return valve should be opened when the volume of the chamber changes from V_C to V_D.

The technical work of the compression process taking place in a two-vane vacuum blowdown pump (BEA) can be calculated from the equation

$$L_{tBEA} = L_{t\pi BE} + \sum_{i=1}^{r} \frac{1}{2} \{p(\varphi_E + i\Delta\varphi) - p[\varphi_E + (i-1)\Delta\varphi]\}\{V(\varphi_E + i\Delta\varphi) + V[\varphi_E + (i-1)\Delta\varphi]\} \quad (16)$$

where:

$L_{t\pi BE}$ —technical work of the compression process of a constant amount of the medium sucked from the vacuum area,

$\Delta\varphi$ — calculation step of a variable that determines the position of the working chamber which is as follows:

$$\Delta\varphi = \frac{\varphi_A - \varphi_E}{r} \quad (17)$$

where r — the number of calculation steps.

CONCLUSIONS

The operation of a rotary-vane vacuum pump designed for compression of mixtures containing vapour (e.g. water vapour) was analysed. The analysis shows that lowering the degree of compression (which is necessary to prevent vapour outdropping) can be made most conveniently by supplying the ambient air into the working chamber. The formulas were derived for such a case that describes the change of the thermodynamic parameters of gas ($m(\varphi), p(\varphi), T(\varphi)$). The $p-V$ diagram presents the processes carried out in the working chamber at different values of the flux of the flowing-in mass of air. It can be deduced from this diagram that the most convenient way is to supply the medium in a short time and with a large flux of mass. The formula for the technical work of the compression process was derived.

REFERENCES

[1] Gnutek Z., Kalinowski E., Some aspects of describing processes in sliding-vane rotary machines. Proceedings 1992 International Compressor Engineering Conference at Purdue, p. 1199-1205. Hamilton Purdue University, West Lafayette, Indiana 1992.

[2] Golovincow A.G.: Rotacjonnyje kompressory, Maszinostrojenije, Moskva, 1963

[3] Gnutek Z.: Rotary-vane machines—development of selected elements of uni-dimensional theory. OWPWr, Wrocław 1997

[4] Groszkowski J.: Technology of high vacuum, WNT, Warsaw 1978

[5] Hałas A.: Technology of high vacuum, PWN, Warsaw 1980