Analysis of Operation of Muldsliding-Vane Vacuum Pumps

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ANALYSIS OF OPERATION OF MULTISLIDING-VANE VACUUM PUMPS

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Abstract

The paper presents thermodynamic analysis of processes that occur in multisliding-vane vacuum pumps, operating in the tank-emptying layout. For a pump with a clearance space, limit value of vacuum obtainable has been determined, as well as tank emptying rate and power required to drive the pump. A formula for egzergetic efficiency of pumping process has been derived. An experimental method of indicator diagram determination has been discussed for a multisliding-vane vacuum pump without a clearance space.

Nomenclature:
b – vane thickness
B – egzergy
e – eccentricity
L – length of working chamber
m – mass
n_{ob} – rotational speed
N – power
p – pressure
r – rotor radius
R – cylinder radius
R_i – individual gas constant
T – temperature
V – volume
y – radial clearance
$Z_k(\varphi)$ – relative cross-sectional area of vane machine
α_1 – α_4 – angles describing positions of control edges
λ – angle between successive vanes
σ – compression ratio
τ – time
φ – angular coordinate

1. Introduction

Multisliding-vane rotational vacuum pumps are usually regarded [1,2,6] as a special case of multisliding-vane rotational compressors. This is justified by the fact, that processes within a vacuum pump are similar, or sometimes identical, to those within compressors. However, when taking the function of a pump within a

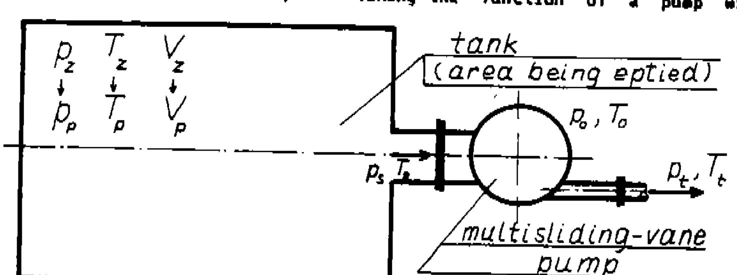

Fig. 1. Outline of a system, from which gas is being removed

technical system into consideration, one can point out a number of the latter's functional goals, which justify the need for separate treatment as far as thermodynamic analysis of multisliding-vane rotational vacuum pumps is concerned. One of them is the process of tank emptying. It is present in many branches of technology and in laboratories. Fig. 1 outlines a system, from which gas is being removed. Its initial thermodynamic parameters are p_Z, T_Z and V_Z. Parameters of an area, into which gas is being transferred, are p_0, T_0 and $V = \infty$. The design idea of the pump is presented in Fig. 2. Not specified in the picture is the design

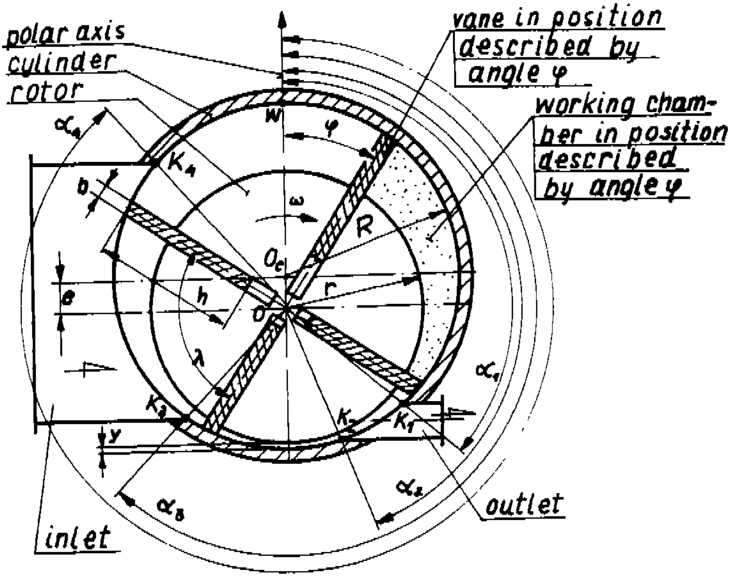

Fig. 2. Design idea of multisliding-vane vacuum pump.

parameter L, i.e. length of the working chamber. Pump timing control (Fig. 3) is achieved by means of proper localization of the control edges (K_1 to K_4). Their positions are characterized by angles $\alpha_1 .. \alpha_4$ and are constant for a given pump type. Therefore, for multisliding-vane vacuum pumps one can reasonably speak of a constant value characteristic of them, which is internal compression ratio [6]. It is determined by formulae:

- for adiabatic compression pumps:

$$\sigma_{wA} = \frac{p_k}{p_w} = \left(\frac{V_w}{V_k} \right)^k = \left[\frac{Z_k(\alpha_4)}{Z_k(\alpha_1 - \lambda)} \right]^k \qquad (1)$$

- for isothermal compression pumps:

$$\sigma_{wT} = \frac{p_k}{p_w} = \frac{V_w}{V_k} = \frac{Z_k(\alpha_4)}{Z_k(\alpha_1 - \lambda)} \qquad (2)$$

where: p_w, V_w – gas pressure and working chamber volume at the beginning of compression,

p_k, V_k – gas pressure and working chamber volume at the end of compression,

$Z_k(\ )$ – relative cross-sectional area of the working chamber as a function of angular position specified in the parentheses [4].

The relative cross-sectional area $Z_k()$ depends on design parameters; consequently σ_w is dependent on the parameters and the type of compression process.

In a tank/vacuum pump/ambient system, introduction of the external compression ratio [6] is also justifiable according to formula:

$$\sigma_z = \frac{p_t}{p_s} \tag{3}$$

where: p_t - gas pressure in the discharge pipe,

p_s - gas pressure in the suction pipe.

Multisliding-vane vacuum pumps, which remove gas from a closed tank, operate at variable external compression ratio ($p_s = p_p \neq$ idem). This results in practical inequality of external and internal compression ratios, otherwise being the most favourable case.

2. Multisliding-vane Vacuum Pump with a Clearance Space

Depending on radial clearance magnitude y, various physical models of multisliding-vane vacuum pump operation are relevant. If $y \approx 0$, no medium flux flows back from the delivery area to the suction area. If $y > 0$ and $\alpha_3 - \alpha_2 > \lambda$ (see Fig. 2), there exists a clearance space within the multisliding-vane pump, due to which some part of the medium exhibiting parameters like in the delivery area (or similar) returns to the suction area. Presence of the return gas flux implies, that pressure in the system (tank) lowers until the mass m_{ss} of the working medium within the working chamber being just closed during filling period equals the reverse flux mass m_r. In the absence of other types of return flux, the m_r mass amounts to:

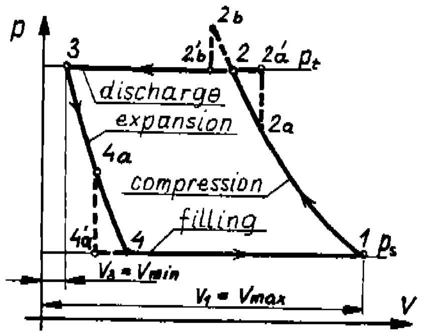

Fig. 3. p - V diagram of a multisliding-vane vacuum pump with clearance space

$$m_{rt} = R^2 \cdot L \cdot Z_k(\alpha_2) \cdot \frac{p_o}{R_i \cdot T_o} \tag{4}$$

Moreover,

$$m_{ss} = R^2 \cdot L \cdot Z_k(\alpha_4) \cdot \frac{p_p}{R_i \cdot T_p} \tag{5}$$

Allowing for (4) and (5) and equation $m_{ss} = m_r$, the following is obtained after transformations:

$$p_p = p_o \cdot \frac{T_p}{T_o} \cdot \frac{Z_k(\alpha_2)}{Z_k(\alpha_4)} \tag{6}$$

In the above formulae, p_p and T_p are gas pressure and temperature in the tank at any moment, relatively. Relationship between the limit pressure obtainable p_p and number of vanes, ensuing from equation (6), is shown in Fig. 4. The formula can also be used to assess the effect other design parameters have on p_p.

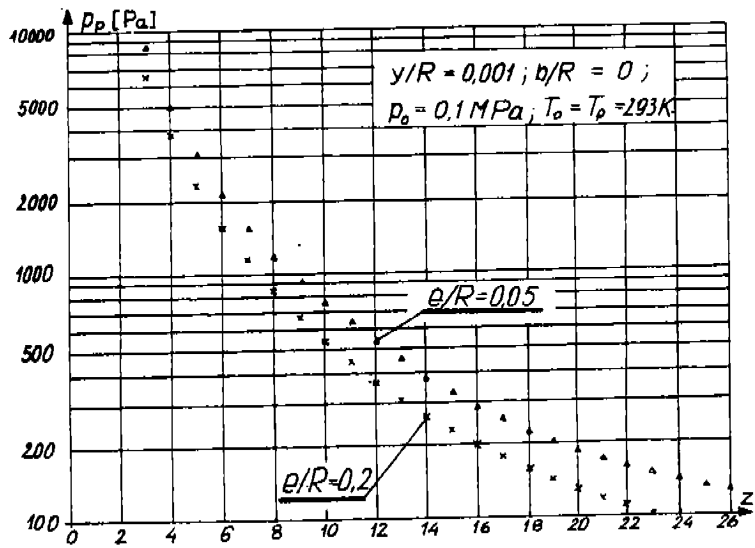

Fig. 4. Relationship between $p_{p\ min}$ and number of vanes.

Similar diagram can be made, when all return fluxes m_{ri} are known. If

$$m_{\Sigma r} = \sum_i \int_{\tau_o}^{\tau} \dot{m}_{ri} \cdot d\tau \tag{7}$$

then the equation $m_{ss} = m_{\Sigma r}$ should be used.

Dynamics of pumping process depends both on properties of a multisliding-vane vacuum pump, and volume of an area being evacuated. Assuming $T_o = T_Z$ and isothermal compression/expansion, gas pressure within the tank at any moment τ can be determined from the formula [4]:

$$p_p(\tau) = p_o \cdot \left\{ \frac{Z_k(\alpha_4) - Z_k(\alpha_2)}{Z_k(\alpha_4)} \left[1 - \frac{Z_k(\alpha_4)}{\bar{V}} \right]^{z \cdot n_{ob} \cdot \tau} + \frac{Z_k(\alpha_2)}{Z_k(\alpha_4)} \right\} \tag{8}$$

where: $\bar{V} = \frac{V}{R^2 L}$ - relative volume of the tank being evacuated.

The time elapsed until pressure value p is obtained amounts to:

$$\tau = \frac{\ln \left[\frac{\frac{p_o}{p} \cdot Z_k(\alpha_4) - Z_k(\alpha_2)}{Z_k(\alpha_4) - Z_k(\alpha_2)} \right]}{z \cdot n_{ob} \cdot \ln \left[1 - \frac{Z_k(\alpha_4)}{\bar{V}} \right]} \tag{9}$$

Momentary delivery of a multisliding-vane vacuum pump is evaluated from the equation:

$$\dot{m}_p(\tau) = R^2 \cdot L \cdot \frac{p_p(\tau)}{R_i \cdot T_o} \cdot n_{ob} \cdot \bar{m}_p(\tau) \tag{10}$$

where:

$$\bar{m}_p(\tau) = z \cdot \left[Z_k(\alpha_4) - \frac{p_o}{p_p(\tau)} \cdot Z_k(\alpha_2) \right] \tag{11}$$

Momentary driving power of the pump is as follows:

$$N_p(\tau) = R^2 \cdot L \cdot n_{ob} \cdot p_p(\tau) \cdot |\bar{N}_p(\tau)| + N_f \quad (12)$$

where the relative driving power $|\bar{N}_p(\tau)|$ can be evaluated from the relationship:

$$\bar{N}_p(\tau) = z \cdot Z_k(\alpha_4) \cdot \ln \frac{Z_k(\alpha_1-\lambda)}{Z_k(\alpha_4)} + z \cdot \sigma_z(\tau) \cdot Z_k(\alpha_2) \cdot \ln \frac{Z_k(\alpha_3-\lambda)}{Z_k(\alpha_2)} +$$

$$+ z \cdot Z_k(\alpha_3-\lambda) \cdot \left\{ \sigma_z(\tau) \cdot \left[\frac{Z_k(\alpha_2)}{Z_k(\alpha_3-\lambda)} \right] - 1 \right\} +$$

$$- z \cdot Z_k(\alpha_1-\lambda) \cdot \left\{ \sigma_z(\tau) - \left[\frac{Z_k(\alpha_4)}{Z_k(\alpha_1-\lambda)} \right] \right\} \quad (13)$$

where N_f – work dissipated due to friction of vanes [2, 4].

Momentary, time-varying external compression ratio is equal to:

$$\sigma_z(\tau) = \frac{p_o}{p_p(\tau)} \quad (14)$$

Fig. 5 shows relationships existing between $p_p(\tau)$, $\bar{m}_p(\tau)$ and $|\bar{N}_p(\tau)|$ for a selected set of design variables.

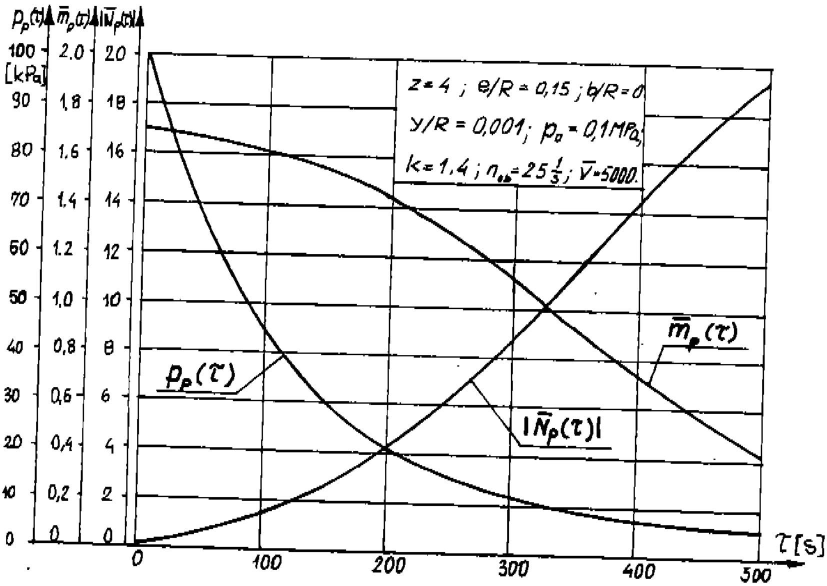

Fig. 5. Relation between $p_p(\tau)$, $\bar{m}_p(\tau)$ and $|\bar{N}_p(\tau)|$ for a multisliding-vane vacuum pump with clearance space

In order to asses thermodynamic effectiveness of the pump, the same values can be used as those for compressors [1, 2, 6]. The effectiveness of gas removal from the vacuum area can be estimated, on the other hand, with the use of egzergetic efficiency [5]:

$$\eta_{bp} = \frac{\Delta B_{zb}}{L_{nap}} \tag{15}$$

where: ΔB_{zb} - egzergy increase within a closed system limited by tank walls,
L_{nap} - work of the pump-drive engine.

Egzergy increase within a closed system limited by tank walls amounts to [5]:

$$\Delta B_{zb} = B - V\,(p_p - p_o) = R^2 \cdot L \cdot \bar{V} \cdot \left\{ \frac{p_p}{R_i \cdot T_p} \cdot \left[c_p \cdot \left(T_p - T_o - T_o \cdot \ln \frac{T_p}{T_o}\right) + T_o \cdot R_i \cdot \ln \frac{p_p}{p_o} \right] - (p_p - p_o) \right\} \tag{16}$$

where: B - the sum of potential, kinetic and thermic egzergy for gas contained in the tank.

3. Multisliding-vane Vacuum Pump without Clearance Space

As already mentioned, a vacuum pump with $y \approx 0$ is conspicuous by the fact, that (almost) no medium flux flows back through the radial clearance. This makes necessary to introduce a correction into the $p = p(V)$ relation representation for any working chamber. In a vacuum pump with non-zero clearance space, volume of the working chamber never reaches null. In a vacuum pump with no clearance space, volume of the working chamber changes from $V = 0$ to $V = V_{max}$ and then again to $V = 0$ (see Fig. 6).

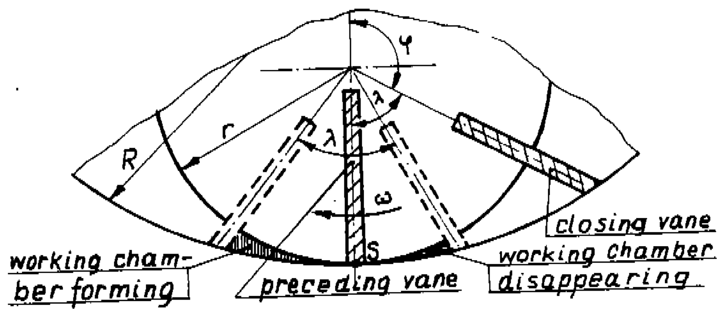

Fig. 6. Formation and disappearance of the working chamber for a pump without clearance space

A new working chamber starts forming, when the preceding vane of this working chamber, which position corresponds to the angle of $\varphi = \pi - \lambda$, passes cylinder/rotor contact point. It is fully formed, when the closing vane comes to $\varphi = \pi$ position. The chamber starts to vanish, when the closing vane has turned by the angle of $\varphi = 3\pi - \lambda$, and ends when $\varphi = 3\pi$. The rotor turns by an angle of ε during the period between formation beginning and complete disappearance of the chamber, ε being equal to:

$$\varepsilon = 3 \cdot \pi - (\pi - \lambda) = 2 \cdot \pi + \lambda = 2 \cdot \pi \cdot \left(\frac{z+1}{z} \right) \tag{17}$$

This means, that the full working cycle duration of a multisliding-vane vacuum pump with $y \approx 0$ depends on number of vanes and corresponds to the rotor rotation by an angle of ε. This relationship holds also for one- and two-vane pumps.

The necessity above mentioned of introducing a correction into the $p = p(V)$ relation representation for the $y \approx 0$ case arose from interpreting experimental data obtained from multisliding-vane vacuum pump tests [3]. Fig. 7 presents a plot of experimentally derived relationship of pressure versus chamber position for a selected type of a foursliding-vane vacuum pump without clearance space. The plot

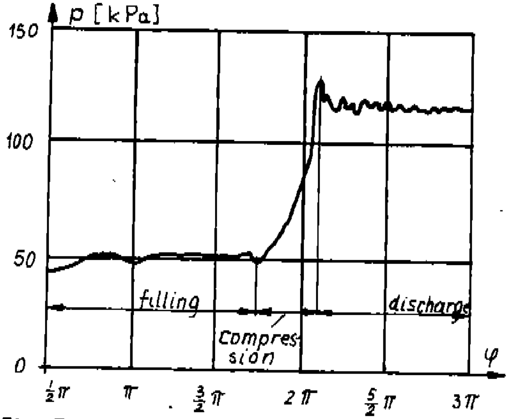

Fig. 7. p = f(φ) relation for a four-vane pump

has been obtained by pasting together a number of pressure vs. time curves taken from an oscilloscope screen and produced by measuring points arranged radially in the pump body.

The relation thus derived has been next transformed to the p = f(φ) relation. Number and arrangement of the measuring points have been so selected, as to obtain the pressure sensing of working chamber pressure during the whole working cycle. Fig. 8 shows p - V diagram derived by analyzing the plot of Fig. 7.

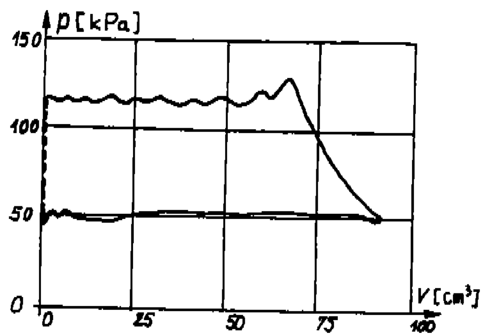

Fig. 8. p - V diagram for a four-vane pump with $y \approx 0$

The diagrams indicate that the process running in the working chamber can be divided into three stages, that of filling, compressing and discharging. Absence of expansion stage provides justification for regarding a vacuum pump with $y \approx 0$ as an ideal compressor.

4. Final Remarks

Multisliding-vane vacuum pumps, apart from drawbacks (friction), display a number of advantages, owing to which they are readily used as initial or low vacuum pumps. Their thermodynamic description however is not completely consistent with that of multisliding-vane compressors, especially if they are used in the tank-emptying layout. Multisliding-vane pumps with ($y \neq 0$) and without ($y \approx 0$) clearance space should be distinguished between and differently described. The authors propose also the employment of egzergetic efficiency to evaluate tank-emptying effectiveness.

5. References

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