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CALCULATION OF OPERATING SPEED FOR TWO-STAGE LIQUID SLIDING-VANE VACUUM PUMP

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Key words and phrases: gas leakage through the hinge clearances; labyrinth seal; liquid sliding-vane vacuum pump; operating speed; real gas state.

Abstract: The paper deals with the calculation of operating speed for liquid sliding-vane vacuum pump and stating the loss factor. The mathematical model of real gas flow through the hinge seal is given. The paper presents the laboratory-scale set-up for calculating experimentally the volume of gas leakage through the hinge seal.

Liquid sliding-vane vacuum pump (**LSVVP**) designed by the "Theory of Machines and Mechanisms, Machines Components" department offers several advantages: high vacuum and smaller dimensions [1, 2]. There is the presence of hinges in the stage rotor. The clearances of the hinge are the principal source of gas leakage from the second stage of the pump (within the stage rotor) into the first one. The design differs in that blade butt-ends of one of them are fitted with the flexible coating separating gas phase and liquid ring thus keeping the water steam from entering the pump feed area of the pump which significantly enhance higher vacuum.

The major performance qualities of the vacuum pump are high operating speed and maximal high vacuum obtained in the process. Operating speed is the gas space pumped out. It is calculated as the product of maximum volume of the first stage working cell by the rotational speed of the stage rotor

$$S = S_1 = V_{\max}\, n\, \lambda_{c1}\, \lambda_{c2}\,, \tag{1}$$

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where λ_{c1} – is the coefficient of the first stage capacity of the pump; λ_{c2} – is the coefficient of the second stage capacity; n – is the rotational speed of the stage rotor, s^{-1};

$$\lambda_{c2} = 1 - \lambda_1 - \lambda_2, \qquad (2)$$

where λ_1 – is the loss coefficient due to the gas leakage through the hinge clearances λ_2 – is the loss coefficient due to liquid evaporation in the pump. For the pump with the flexible coating the equation is $\lambda_2 = 0$. Hence, the gas leakage through the hinge clearances is the principal cause of the reduction in operating speed.

The most gas leakage through the hinge clearances occurs when the hinge is positioned in the suction area of the first stage; the pump feed area of the second stage being located with the same turning angle of the stage rotor (Fig. 1). In this case gas pressure on the outside of the hinge is the lowest in the pump (suction area of the stage) while the gas pressure on the inside of the hinge is the highest (pump feed area of the second stage). Consequently gas leakage from the pump feed area of the second stage into the suction area of the first stage occurs through the hinge resulting in the fall in the compression ratio of the second stage and the reduction of operating speed as well as the ultimate obtainable vacuum (Fig. 2).

To calculate the gas parameters in the hinge pocket with regard to the rotating angle of the shaft the gas flow via the seal is considered as the labyrinth seal flow [4].

It is assumed that the flow sections areas of the seal in the hinge and their geometrical dimensions are constant; the process of the gas leakage through the seals is quasi-stationary.

The development of the theoretical functions characterizing the accepted model of gas flow through the hinge is based on the Beattie-Bridgeman equation of the real gas condition in the form of Bogolyubov–Mayer, mass and energy balance for the gas volume within the hinge pocket: heat exchange of gas with the hinge components [3, 5].

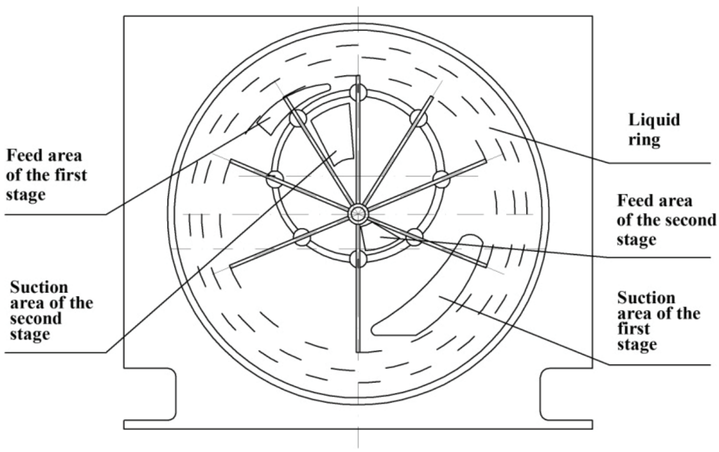

Fig. 1. Fundamental diagram of liquid sliding-vane pump

The equation of the real gas condition for the inside of the hinge volume is as follows

$$pV = mRzT\,, \tag{3}$$

where p – is the pressure in the hinge, Pa; V – is the pocket volume in the hinge, m^3; m – is gas mass, being in the hinge pocket, kg; R – is gas constant for air, $R = 287{,}2$ J/(kg·C°); z, T – is the compressibility factor and gas temperature, K.

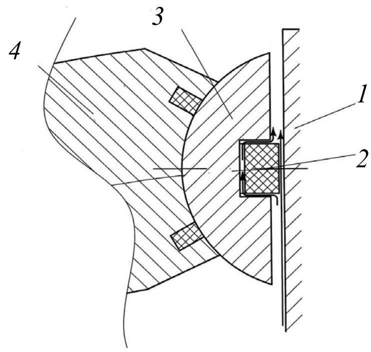

Fig. 2. Gas leakage through the hinge clearances:
1 – blade; *2* – seal; *3* – hinge; *4* – stage rotor

Having applied the approximation of real gas as an ideal in the area of the condition diagram where the process under study is taking place, we will consider the product zT or the conventional temperature as the single complex and differentiate both parts of equality (1) with respect to time

$$\frac{Vdp}{d\tau} = RzT\frac{dm}{d\tau} + Rm\frac{d(zT)}{d\tau}\,; \tag{4}$$

and derive only the derivative into the left part of equality

$$\frac{dp}{d\tau} = \frac{RzT}{V}\frac{dm}{d\tau} + \frac{Rm}{V}\frac{d(zT)}{d\tau}\,. \tag{5}$$

Change of the conventional temperature is found in the following way: write down the first thermodynamics principle for gas in the hinge pocket

$$dQ + (i_1G_1 - i_2G_2)d\tau = dU\,, \tag{6}$$

where dQ – is the heat quantity supplied to the gas in the hinge pocket in $d\tau$ time; dU – is change in the internal gas energy in the hinge pocket in $d\tau$ time; i_1, i_2 – are gas enthalpies streaming inward and backward respectively. From the equation (5)

$$\frac{dQ}{d\tau} + i_1G_1 - i_2G_2 = \frac{dU}{d\tau}\,. \tag{7}$$

The first term of the equation can be presented as

$$\frac{dQ}{d\tau} = \alpha_y\left[F_\text{b}(T_\text{b} - T) + F_\text{h}(T_\text{h} - T)\right], \tag{8}$$

where α_y – is the conventional average coefficient of convective heat from gas to blade and hinge, W/(m^2·K); F_b, F_h – are respectively the surface areas of the blade and hinge being in contact with gas, m^2; T_b – is the average temperature of the blade surface, K; T_h – is the average temperature of the inside face of the hinge, K; T – is the thermodynamic temperature of gas, K.

Change in the internal gas energy is found differentiating the relation of internal mass energy m with respect to time $U = mu$

$$\frac{dU}{d\tau} = m\frac{du}{d\tau} + u\frac{dm}{d\tau}, \tag{9}$$

where u – is the specific internal gas energy.

Substitute the enthalpy and specific internal gas energy in the equations (7) and (9) for the following relation

$$i = \frac{k}{k-1}RzT; \quad u = \frac{1}{k-1}RzT; \quad du = \frac{1}{k-1}Rd(zT). \tag{10}$$

Substituting successively the relation (10) in the equations (7) and (9) into (10) and the value $\frac{dU}{d\tau}$ from (9) in the equation (7) and replacing

$$d\tau = \frac{d\varphi}{\omega},$$

having conducted transformations we will derive the following differential equation

$$\frac{d(zT)}{d\varphi} = \frac{1}{m\omega}\left[\frac{k-1}{R}\frac{dQ}{d\tau} + (kz_1T_1 - zT)G_1 + (1-k)G_2zT\right]. \tag{11}$$

Solving equation (11) integrally with equation (10) and substituting

$$m = \frac{pV}{RzT}$$

we will arrive at the differential equation of temperature and gas pressure change in the hinge in relation to the turning angle of the pump rotor:

$$\frac{dp}{d\varphi} = \frac{kR}{V\omega}\left(z_1T_1G_1 - zTG_2 + \frac{k-1}{kR}\frac{dQ}{d\tau}\right); \tag{12}$$

$$\frac{d(zT)}{d\varphi} = \frac{RzT}{pV\omega}\left[(kz_1T_1 - zT)G_1 + (1-k)zTG_2 + \frac{k-1}{R}\frac{dQ}{d\tau}\right]. \tag{13}$$

In order to obtain experimental data on the leakage in the hinge, the set-up was designed and developed (Fig. 3). It is intended for experimental evaluation of gas leakage amount through the hinge from the pump feed area of the second stage to the suction area of the first stage.

The set-up comprises the case *1*, housing tubes *2* fitted with the spring-loaded plates *3* imitating hinge seals; vacuum pump *4*, compressor *5* globe valves *6* and *7*, vacuum gauge *8* and manometer *9*. It runs in such a way: the valve *7* being shut off, the vacuum pump *4* pumps down the air from the set, then valve *6* is shut off, following the valve *7* is opened and the compressor *5* pumps up the pressure of 0,4 MPa into the area ahead of the spring-loaded plate. Thereafter, the valves being shut off, the vacuum drop is observed in the back of the spring-loaded plate area.

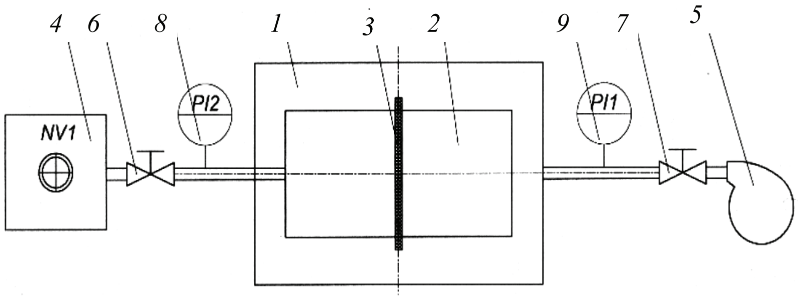

Fig. 3. Set-up for experimental determination of gas leakage amount through the hinge seals

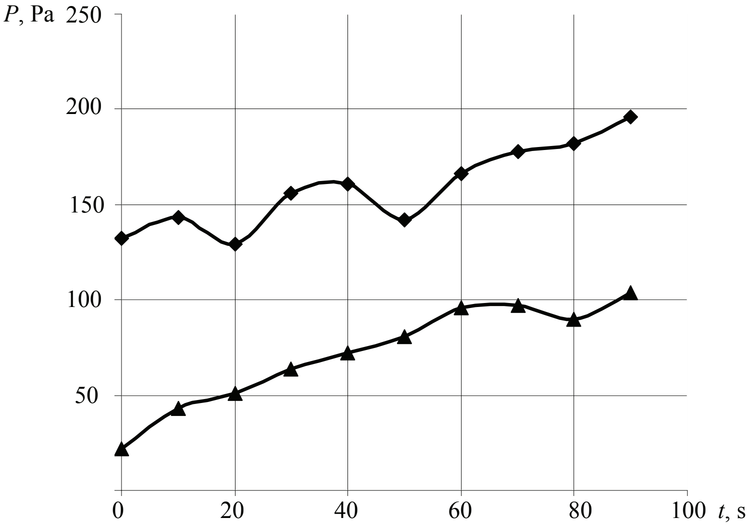

Fig. 4. The relation of vacuum drop with time in the first stage:
– Experimental; – Theoretical

In the context of research project the procedure of experimental determination of gas leakage amount out off the pump feed area of the second stage into the suction area of the first one was produced.

In the ahead of the spring-loaded plate area the pressure up to 150 kPa was produced employing the compressor. In the area behind the spring-loaded plate the high vacuum up to 2 kPa was given rise with the help of the sliding-vane rotary vacuum pump. Afterwards, the 10 minute drop in the high vacuum was noted with the vacuum gauge.

Fig. 4 shows the relationship of drop in high vacuum, Pa, to time. It is theoretically calculated and is experimentally obtained by means of the set-up.

The theoretical and experimental investigations carried out by the research team give grounds to the following conclusions: the experimentally obtained and theoretically calculated data are within the allowable error. Thus, the mathematical model of gas leakage through the hinge clearances which was put forward can be used to calculate the pump capacity factor.

As a result, the amount and cost of experimental work can be cut while the calculation accuracy of LSVP at the design development stage can be improved.

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К расчету быстроты действия двухступенчатого жидкостно-пластинчатого вакуумного насоса

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Ключевые слова и фразы: быстрота действия; жидкостно-пластинчатый вакуумный насос; истечение газа через зазоры в шарнире; лабиринтное уплотнение; уравнение состояния реального газа.

Аннотация: Рассмотрен расчет быстроты действия жидкостно-пластинчатого вакуумного насоса, определение коэффициента потерь. Приведена математическая модель течения реального газа через уплотнение в шарнире. Показана лабораторная установка для экспериментального определения величины переточек газа через уплотнение в шарнире.
