IOP Conf. Series: Materials Science and Engineering **779** (2020) 012025 doi:10.1088/1757-899X/779/1/012025

Avanepneumatic motor modification to reduce torque ripples

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Annotation. The problem of determining and possible ways to reduce the amplitude of torque ripples in a vane pneumatic motor without a significant reduction of its energy efficiency is considered. This problem is important for improving the ergonomic performance of manual mechanized pneumatic tools and improving the accuracy of pneumatic servosystems. The results are presented for the traditional layout scheme of a vane pneumatic motor, as well as for the proposed modified design. The decision was made using a detailed mathematical model of a pneumatic motor. The prospects of the modified pneumatic motor to reduce the torque ripples at higher energy efficiency indicators are shown, compared with the original model with an increased number of vanes.

Introduction

Vane pneumatic motors (PM) received the most widespread application among volumetric pneumatic motors: they occupy more than 75% of the PM market in power range from 0.1 to 18 kW. This is due to the simplicity and reliability of their design, relatively low cost and high developed capacity per unit of weight and volume. The most widespread segment of vane PMs is manual tools (screws, grinders, drills, mixers, etc.). Vane PMs are used in explosive industries, in the pharmaceutical and food industries, because this type of PM can be subjected to washing and sterilization. PMs in a special non-magnetic version are used as a part of servos for operation in strong magnetic fields [1].

One of the problems associated with the use of vane PM for some applications is the presence of torque ripples within one revolution of the shaft. The occurrence of these ripples is associated with a finite number of PM working chambers as well as with a sharp pressure change in chambers when they are connected to supply or exhaust lines. Torque ripples can cause vibrations that adversely affect both the resource of a PM itself and the device where it is installed. For manual tools, it is possible to reduce the quality of processed products as well as a harmful effect on the working personnel [2].

Note that similar problems with torque ripples are typical for different types of electric motors [3]. Here, to reduce the amplitude of these ripples, both constructive solutions are used, for example, changing the shape and number of poles, and more common solutions based on the formation of a certain law of current change in the windings of an electric motor with the help of a feeding device.

The catalogs of the main manufacturers of PMs (Atlas Copco, Ingersoll Rand, Deprag, etc.) usually contain their mechanical and flow characteristics, i.e. the dependence of the developed torque, power and compressed airflow on the steady angular velocity at a given supply pressure. The values of the torque ripples amplitude for each particular model are not given; that is this indicator is not regulated. The customer can choose only a model with a certain number of vanes. It is not possible to estimate possible torque ripples from catalogue data.

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Problem statement

For vane PMs (by analogy with electric motors), an approach associated only with design solutions is possible, since the speed of existing devices for controlling the pressure and flow of compressed pneumatic is not enough.

The simplest and most obvious way is to increase a number of vanes, i.e. increase a number of working chambers. But increasing a number of vanes, together with a decreasing torque rip pleamplitude, leads to increasing friction losses and decreasing energy efficiency. The original approach is presented in [4], where the stator has a non-circular profile, the geometry of which is chosen taking in account minimum torque ripple amplitude. But this requires a more complex and expensive stator manufacturing technology, and can also reduce the life of vanes due to the variable vane — stator contact area.

The purpose of this paper is to develop a method for solving the problem of estimating the level of vane PM torque ripples and the possibility of reducing them while maintaining sufficient energy efficiency. The energy efficiency index is extremely important here, because the cost of pneumatic energy is 5–6 times higher than electric energy. Here, as indicators of torque ripple, we will use its amplitude, and then-the specific flow of compressed air (flow per unit of developed power).

To solve this problem, it is necessary to produce several prototypes of PM with a different number of vanes and perform a sufficiently large amount of expensive experimental research. To reduce the costs and volume of these works, mathematical modeling methods have been developed that allow predicting the characteristics of vane PM depending on its geometric parameters and supply pressure [5]–[9]. At the same time, the models in these sources differ in the degree of detail of both structural details and in the description of thermodynamic processes in working chambers. On the other hand, mathematical models of a vane PM contain a number of parameters that are quite difficult to estimate experimentally or to obtain analytical expressions for their calculation. In our case, these are the parameters characterizing the friction in the various nodes of PMs, and the cross-sections leaks and overflows in the gaps of PMs. To determine them, the method of vector identification [10] was used which allows restoring the values of the required parameters on the basis of a limited number of experimental data and a mathematical model. Mathematical modeling is widely used for the analysis and synthesis of various machines based on volumetric and dynamic principle of action [11],[12].

Therefore, the solution of the problem was carried out in the following sequence:

–determination of mechanical and consumable characteristics of a specific PM model on a typical experimental stand;

– obtaining an adequate mathematical model of vane PM;

– carrying out the procedure of vector identification of effective cross-sections of channels of overflows and leaks values, and also coefficients of friction of vanes about a stator and rotors lotes;

–determination of torque ripples and energy efficiency of the original PM model with different number of vanes and parameters determined during the identification procedure, as well as its modified design.

It is reasonable to assume that the identified parameters do not change for PMs with different number of vanes.

The vane PM basic design and its modification

The РП42-55 PM model was used as an object of study, the structural diagram of which is shown in Figure1.

In the modification design the working volume of the PM is divided into two equal parts, and the rotor consists of two halves with slotes for vanes turned at an angle π/z, where z is the number of vanes. So, for $z = 4$ (see Figure 2) this angle is 45 °. The supply of compressed air to the chambers and exhaust are carried out through the channels in the stator as in the original design. The analysis of this scheme showed that all parts of the modified PM can be manufactured on the same equipment as for the original model, with the required quality of the working surfaces, followed by assembly into the finished product. The disadvantages of this scheme are the greater number of parts (correspondingly slightly higher cost), as well as the appearance of additional channels for the flow of compressed air, which will negatively affect energy efficiency. The main geometric parameters of the initial PM model are given in [8].

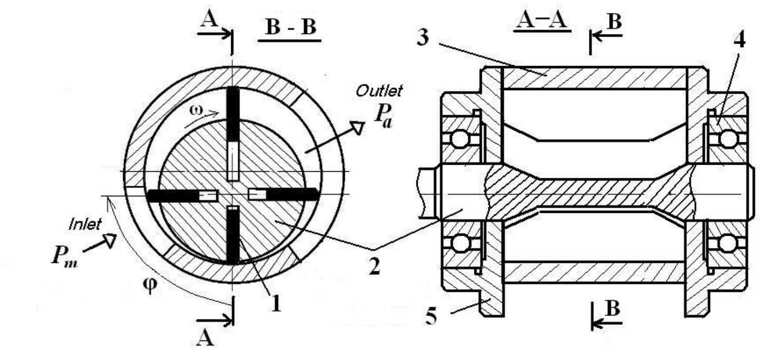

Figure1. Design scheme of PM model RP42-55.1 — plate, 2 — rotor, 3 — stator, 4 — bearings, 5 — covers

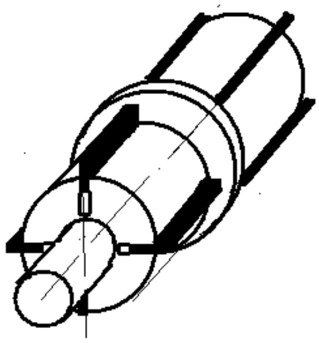

Figure2. The modified PMrotor

Test System and Experimental Results

The test rig constructive scheme of the mechanical and pneumatic part is shown in Figure3.The test rig consists of a rigid metal frame, on which the PM (model RP42-55), the torque sensor and the angular velocity TM-307, as well as the loading device, an electromagnetic brake ANV-12, are coaxially fixed. ShaftsofPM, sensor and brake are connected using compensating couplings MIC-5-2470.Sensor, brake and couplings are products of “Magtrol” (Germany).

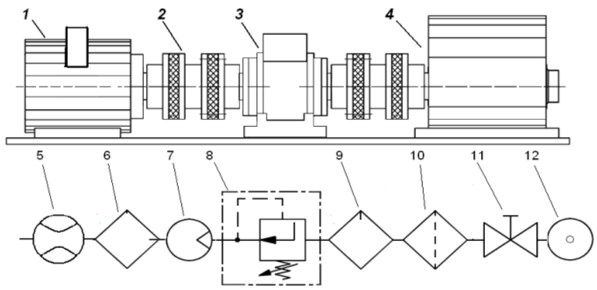

Figure 3. Scheme of test rig. 1 — air motor, 2— coupling, 3— torque sensor, 4— electromagnetic brake, 5— flow meter, oil separator, 7— air motor, 8— pressure regulator, 9— lubricator, 10— filter, 11— valve, 12— supply.

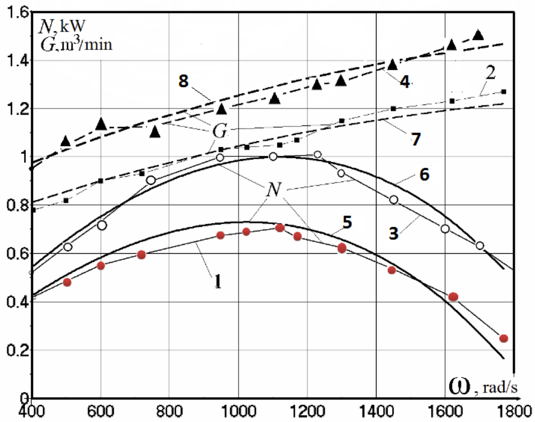

Figure 4. Experimental and calculated mechanical characteristics and air consumption performance.

Compressed air consumption was measured with a ‘testo-6460” flow meter (Germany). The readings were taken in the angular velocity range from 400 to 1800 $s^{-1}$for supply pressures of 0.5 and 0.6 MPa. The experiments were carried out when adjusting the lubricator to supply 35–40 mm^3 of “BEZAN-pneumo” oil per 1 m^3 of compressed air (innormal conditions).Figure 4 presents experimental (marked with markers and connected by lines) mechanical and air consumption characteristics - dependences of developed power N and consumed compressed air consumption G on steady angular velocity.In Figure 4 curves 1 and 2 - respectively power, flow rate at a supply pressure of 0.5 MPa. Curves 3 and 4 are similar dependences at a supply pressure of 0.6 MPa.

Mathematical Modeling

Model consists of the following system of equations:

– two differential thermodynamic equations that describe the change in pressure and temperature of air in the working chamber are obtained based on the energy conservation law and equation of state for the compressed air in the chamber in differential form. Each working chamber of PM is considered as a cavity of variable volume with variable input and output flow zones;

– geometric ratios describing the law of change in the chambers volume and the position of the vanes as functions of the shaft angle;

– Saint-Venant ratios for compressed air consumption at inlet and outlet, as well as for the flow of compressed air through the gaps between the chambers and leaks into the atmosphere;

– logical relations describing the condition of connecting each working chamber to the pressure line and exhaust, as a function of the rotation angle and the angles of air distribution;

– equation of motion of the PM shaft;

– relations describing friction losses in bearings, as well as the forces acting on the vane, when it moves in the rotor slote.

Below we dwell in more detail on the equation of motion , because the proposed approach makes it possible to more accurately calculate friction losses in a vane PM and to describe torque ripples than in the models considered in [5]–[9].

The equation of motion of the PM rotor is compiled on the basis of the Lagrange equation:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\varphi}}\right)-\frac{\partial T}{\partial \varphi}=M \quad (1),$$

where φ is the angle of rotor rotation , measured from the vertical axis (see Figure 1), which also uniquely determines the position of the vanes.

The kinetic energy T is defined as the sum of the kinetic energies of rotor rotation and vanes in relative and figurative movements:

$$T=\frac{1}{2}\left[J_0\omega^2+\sum_{i=1}^{z} m\left(e\omega sin\varphi_i\right)^2+\sum_{i=1}^{z} m\left(R_1-ecos\varphi_i\right)^2\omega^2\right],$$

where J_o— the moment of inertia of the rotor, m— the mass of the vane, e— the eccentricity, z— the number of vanes, φ_i— the angle determining the position of the i-th vane: $\varphi_i=\varphi_1+2\pi\left(i-1\right)/z.$ The radius vector from the center of the rotor to the vane — stator contact pointis $R_{2i} = R_c– ecos\varphi_i$[9], where R_c— the stator radius. The quantity R_{1i} is the radius vector of the gravity center ofthe vane. Fora rectangular vane: $R_{1i} = R_{2i}$ - $b/2$, b— the height of the vane. The equation $e\omega sin\varphi_i$— the speed of the i-th vane relative to the rotor slote.

The generalized torque M in formula (1) is calculated as $M=M_d$- M_m - $M_н$, where M_d is the driving torque, M_m— the friction torque, which consists of the friction torque values in the bearings M_r and the friction torque of the vanes on the stator and the rotor slots, $M_н$ is payload torque.

The driving torque created by the i-th vane from the pressure of compressed air in two adjacent chambers p_i and p_{i+1} (see Figure5) is $M_{di} = (p_i — p_{i+1})[(R_{2i})^2 — r^2]l/2$, where $r$and l are the rotor radius and its length.

The friction torque in bearings is usually determined by empirical relationships given in the materials of manufacturers. Here we used the dependencies given in [13]:

$$M_p=f_1F_p\,{}^{d}\!/_{2}+0{,}97910^{-10}\,f_c\left(vn\right)^{2/3}D^3,$$

where $Fp\ (p_m, \omega)$— the radial load on the bearing as a function of supply pressure and angular velocity. The radial load on the rotor and, accordingly, on the bearings arises due to the fact that the pressure in the working chambers of the PM in the inlet and expansion zone (see Figure1) is higher than in the chambers communicating with the atmosphere. The resultant from the air pressure forces acting on the rotor and directed to its center along the symmetry axis of the i-thchamber will be equal to $F_i = 2p_i rl sin(\pi/z)$.

$$F_1 = k_1 N_1 \cos\beta + k_{11}\left(R_c - e\cos\varphi_i\right)\omega,$$

$$\beta = \arcsin\left(R_c^{-1} e \sin\varphi\right),$$

$F_2=k_2N_2$; $F_3=k_2N_3$—correspondingfriction forces, k_1 and k_2— the dry friction coefficients, k_{11}— the viscous friction coefficient;
$F_4=m\omega^2(R_1-e\cos\varphi_i)$ — centrifugal force;
$F_5=me[(d\omega/dt)sin\varphi_i + \omega^2 cos\varphi_i]$ —inertial force from the radial movement of the vane;
$F_6=2me\omega^2\sin\varphi_i$— Coriolis force.

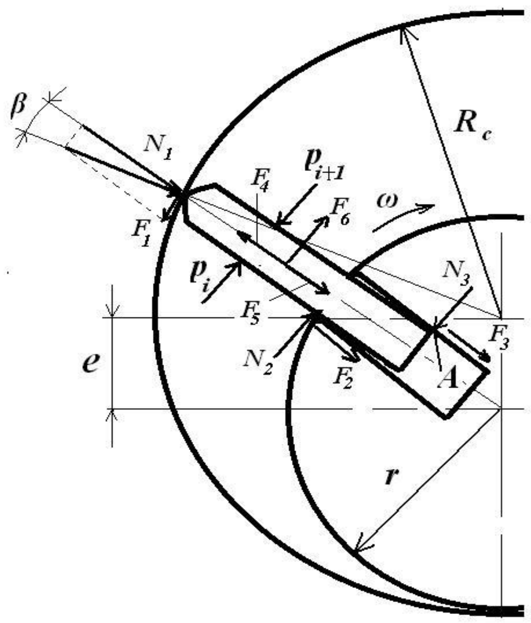

Figure5.A vane loading scheme

The radial load in the support of the rotor will be equal to half vector sum of forces F_i from all chambers, and its average value per revolution is(for the considered PM model, see [8] in more detail):

$$F_p = \frac{1}{2\pi}\int_0^{2\pi}\sum_{i=1}^{z} 2p_i(\varphi) rl sin\left(\pi/z\right) d\varphi$$

Where d = 12mm— the inner diameter of the bearing, f_t = 0,002 (a coefficient of friction for a ball radial bearing), f_c— a coefficient depending on the method of bearing lubrication, v = 35 mm^2/s— the kinematic viscosity of the oil, n— the number of revolutions per minute, D = 22mm— the diameter passing through the centers of the rolling elements.

The given torque from the friction forces of the vane on the stator and the rotor slots is determined as follows. Figure 5 shows a diagram of forces acting on the vane. In determining the friction forces, we were guided by the following provisions. The value of the friction force in the general case is determined by the Stribek model. For vane PMs, more than 90% of friction losses are attributable to the vane — stator friction pair. Therefore, in the vane — rotor friction pair, the viscous component can be ignored, which should not lead to significant errors. In addition, in our case, the relative velocity in the vane — stator pair is significantly higher than those speeds where the Stribek effect is manifested.

Composing three equations of vane equilibrium, namely, the sum of the projections of the forces on the axis of the vane, on the orthogonal direction and the sum of the moments relative to point A, we obtain a system of equations for determining N_1, N_2 and N_3. The given moment from the friction forces acting on the vane:

$$M_t = F_1\left(R_c - e\cos\varphi_i\right) + \left(F_2 + F_3\right) e \sin\varphi_i .$$

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Procedure of vector identification of PM parameters

The identification procedure is implemented in the software complex MOVI (Multicriteria Optimization and Vector Identification), which is based on the Sobol method of studying the parameter space. This method is widely used to solve problems of optimization of parameters of hydraulic and pneumatic machines [14]–[17].

As data for comparison, the calculated values of the power on the shaft $N_{1j}(\omega)$ and the consumption flow rate $G_{1j}(\omega)$ and the corresponding experimental values of these indicators (with index 2), at a supply pressure p_m= 0.5 MPa (j=1) and p_m = 0.6 MPa (j=2) are taken. The values of their maximum divergence are accepted as criteria of conformity:

$$\Delta_1 \leq max \left| N_{1j}(\omega i) — N_{2j}(\omega_i) \right| ,$$

$$\Delta_2 \leq max \left| G_{1j}(\omega i) — G_{2j}(\omega_i) \right| ,$$

where the index i means the value of the value at the point with the number, when sampling the studied interval of change in the angular velocity of the rotor (i=1-12). Friction coefficients k_1, k_2, k_{11} as well as effective cross — sectional areas of leakage channels from the chamber — f_u and flows between the chambers-$f_p$were taken as the required parameters of the mathematical model. At the first stage, when using the MOVI software package, the values Δ_1 and Δ_2 are set quite arbitrarily. If as a result of calculations in the test table there is no Pareto–optimal solution vector (k_1, k_2, k_{11}, f_u, f_p), then Δ_1 and $\Delta_2$should be increased. If there are several Pareto–optimal vectors, then it is necessary to reduce Δ_1 and $\Delta_2$so that there is one Pareto–optimal vector, which will give the desired values of the parameters. Several series of calculations were carried out with the number of tests from 256 to 2048 (for greater reliability) and the range of parameters k_1= 0.1-0.2; k_2= 0.1 — 0.2; k_{11}=0-0.12 Ns/m, f_u=0-37^{10-6} m^2, f_p =0-37^{10-6} m^2. As a result, the following values were obtained: k_1 =0.145, k_2 =0.132; k_{11}=0.026 Ns/m, f_u=$4.^{1210-6}m^2$; f_p =$8.2^{10-6}m^2$. In Figure4 the graphs of the developed power and consumed flow rate of 0.5 MPa (curves 5, 7) and 0.6 MPa (curves 6, 8) calculated according to the identified mathematical model are presented. The coincidence of the calculated and experimental curves can be considered quite satisfactory.

The main results and conclusions

After determining the required parameters for the PM with z = 4, using the mathematical model, we calculated the characteristics of the PM with a different number of vanes (from 4 to 9), as well as a modified PM, under the assumption that the identified parameters remain almost unchanged. The calculation results are presented in table 1 and in Figure6. The analysis shows that increasing the number of vanes leads to certain decreasing in the amplitude of torque ripples within one revolution of the shaft.

With the variation of the number of vanes, the optimal values of the air distribution angles can take slightly different values than for a motor with z = 4. Using the MOVI program, optimization calculations of the inlet end angle and the start exhaust angle for PM with vanes z = 4–9 were carried out. As quality criteria were selected: developed capacity $N \geq N_1$; specific consumption $q \leq q_1$; and amplitude of torque ripples $\delta \leq \delta_1$(criteria with index 1 relate to PM with air distribution angles). Figure 6 shows graphs illustrating the torque ripples in the nominal mode for the RP42-55 model with z = 4 (curve 1) and the modified PM, also with z = 4 (curve 2), as well as the PM with z = 5, respectively curves 3 and 4. We can state a significant decrease in the amplitude of torque ripples and a doubling of the frequency for the modified PM, with a slight increase in the specific consumption of compressed air.

The choiceof the number of vanes or use a modified design of a vane PM depends on the task, i.e. requirements for PM on the amplitude of the torque ripples and the specific flow rate of compressed air and for each specific case is decided individually. In the future, it is planned to carry out similar work with vane PM models having different ratios of geometric parameters and, accordingly, other nominal revolutions and powers, as well as with a special antifriction coating on the stator working surface and composite plate material providing greater bending strength with a smaller thickness and, accordingly, mass.

Table 1. Indicators in the nominal operation mode of the PM with optimized values of the air distribution angles

Number of vanes z	Corner speed ω, c^{-1}	Power N_{max},кWt	Specific consumption q,m^3/(min кWt)	M_{max}-M_{min},Nm	Intake end angle.grad.	Exhaust start angle,grad..
4	1000	0,72	1,42	0,128	70	221
5	950	0,7	1,51	0,08	90	214
7	850	0,614	1,77	0,023	97	206
8	800	0,54	1,96	0,016	89	223
9	750	0,435	2,27	0,011	78	243
			Modified pneumatic motor			
Z=4	1000	0,717	1,56	0,05	70	221
Z=5	950	0,694	1,63	0,028	90	214

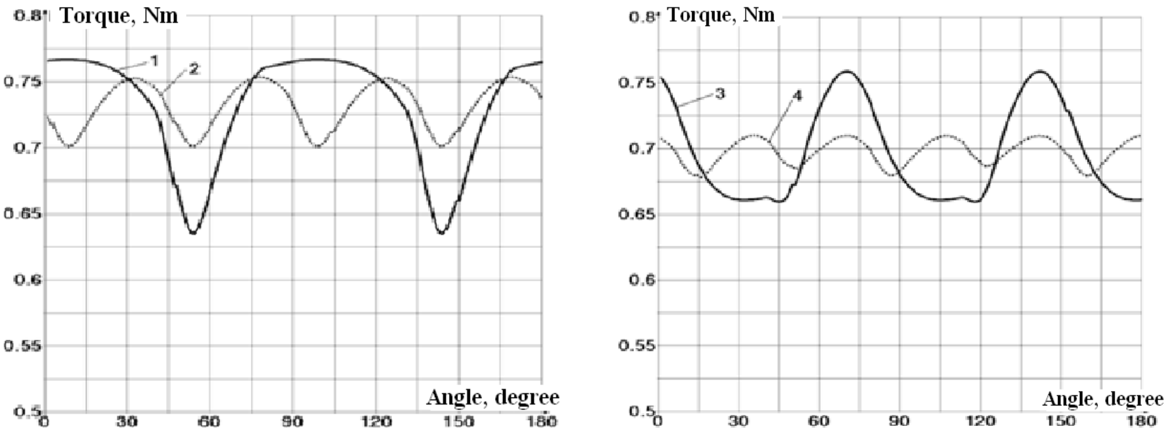

Figure6. The amplitude of fluctuations in torque for the original and modified PM

Select the number of plates or the use of a modified design of the plate depends on PM tasks, i.e. requirements for PM amplitude of fluctuations in torque and specific consumption of compressed air and each case is decided individually. In the future, it is planned to carry out similar work with models of plate PM, having other ratios of geometric parameters and, accordingly, other nominal revolutions and power, as well as with a special antifriction coating of the working surface of the stator and the composite material of the plates, providing greater bending strength at a lower thickness and, accordingly, weight.

The research was supported by Russian Foundation for Basic Research, project No. 18-29-10072 mk.

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