ESTIMATION OF PISTON DYNAMIC CHARACTERISTICS FOR REVERSIBLE AXIAL - PISTON AIR MOTOR

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Abstract‒ We discuss the stability problem of a piston of the reversible axial-piston pneumatic motor called DAR-14M. We performed the full analysis of stress and deformation characteristics of the piston near a resonant frequency with the help of FEM ANSYS. We estimated the own frequency of the piston analytically for the given motor, this estimate was confirmed by numerical methods. It was also shown in harmonic analysis that loaded end face of the piston`s behaves very different in comparison to the unloaded end face. As expected, maximum deformation was found in the area of the holes for the bearing shaft, indicating an optimization necessary in that region of the motor.

Keywords - Piston pneumatic motor, stress and deformation characteristics of the piston, resonance frequency

I. INTRODUCTION

We consider reversible axial-piston air motor of DAR series, firstly engineered by Kaufman M.S. [1] Such motors are widely used for power drive of underground mining machines: drilling rigs and carriages [2], charging units in mining, loading and loading-transport machines, winches [3]. The motors of this kind are 1. 5-2 times smaller in weight and size than the radial-piston pneumatic ones. The bilateral action pistons create a torque of the rotor [4]. The rotor is a cam disk with two-way sinusoidal surface for converting reciprocal movement of the piston into rotation of the rotor. The piston makes three double movements along the rotor axis at the time of one full rotation of the rotor. Each piston has two bearings. The outer rings of the bearings are moving on sinusoidal profile surfaces of the rotor (that shown in Fig. 1). This motion causes the rotor to move. The main dynamic processes in rotor and piston group were investigated in paper [4]. The pressure in working chamber, the frequency of a rotor rotation, influence of the gap between the piston and a rotor on dynamics of engine, an acceleration of pistons were investigated. As a result of this analysis ways of engines improvement were outlined: use of more effective air filters and installation on the engine of bearings with bigger fatigue durability.

The bearings and the piston make a reciprocating movement jointly. Main drawbacks of these motors are a noise [5] and a high rate of the piston breakages in an area of the holes for a bearing shaft (80% from all malfunctions) [6]. It is thereforeuseful to analyse the system behaviour under the influence of the periodic driving force. The study of resonant phenomena in the piston gives us the information about stability of our system. In our work we investigate the deformation problem, the stresses and the frequency of the piston own oscillation. We find, for example, an amplitude increase; sharp changes of the deformation intensity. A system can be destroyed when a frequency of the periodic driving force approaches to the system own frequency.

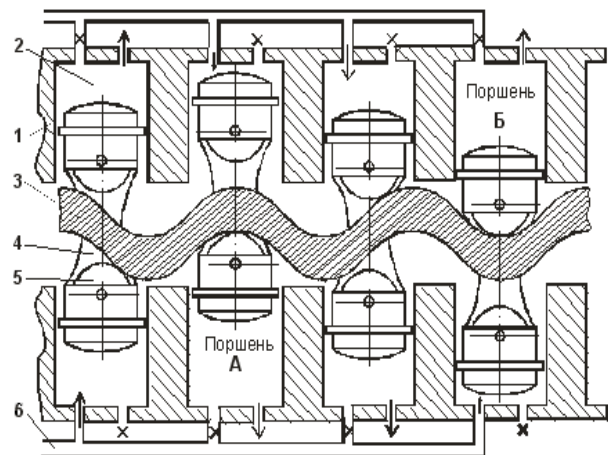We provide an estimate of such frequency depending on the system parameters.

Fig. 1 Engine axially reverse (unfolding)
1 - body; 2 - Cylinder; 3 - cam rotor; 4 - piston;
5 - roller (bearing); 6- gas pipeline.

II. ENGINEERING DESIGN

In reality deformed mechanical systems have an infinite number of degrees of freedom described by generalised coordinates. For analytical calculations it is necessary to limit this number by taking some justified approximations such as considering most rigid parts as absolutely undeformed. In our work we consider the piston's body in assembly as absolutely rigid in contrast to the bearing shaft. According to our estimates the bearing shaft stiffness coefficient is by two orders of magnitude less than the piston's stiffness coefficient. We take a rigidity of the shaft as the rigidity of an entire model system. The bearing shaft is subjected to bending deformation. Periodically changing pressure of compressed air pushes ends of the piston alternately. The piston makes oscillations. Thus external actions are the

pressures in the working chamber $P_1(t)$ and in the exhaust chamber $P_2(t)$. Neglecting a friction of the sliding piston on cylindrical surface we got the following differential equation [7]

$$mx'' + b(x' - x_0') + c(x - x_0) = \pi d^2 [P_1(t) - P_2(t)]/4, \quad (1)$$

where m is the piston mass*; x, x_0* are the coordinates of the piston's butt-end central point before and after deformation respectively*; x' - x'_0* is a relative movement velocity of this point; d is a piston diameter; c is a stiffness coefficient; b is a damping coefficient. We obtain then a time dependence of the coordinate x_0

$$x_0(t) = l_I + 0.5H(1 - \cos \varphi(t))$$
$$+ (R_0 - R_D)(D_W - A(t))/A(t) \quad (2)$$

where l_I is an initial gap between end surface of the cylinder and the piston end face, H is the piston stroke; $\varphi(t)$ is a varying angle of the sine curve piston's profile, Ro is a radius of the grinding wheel, R_D is the piston radius, D_W - average diameter of the rotor wave. Parameter $A(t)$ depends on the number n of the sinusoid's waves:

$$A(t) = (n^2 H^2 \sin^2 \varphi(t) + D_W^2)^{1/2}. \quad (3)$$

Taking into account dynamical loading in our calculations we obtain an estimate for the first resonance frequency approximately at 600 Hz. For getting more accurate result we carry out numerical calculations using the finite element method. This method allows increasing manifold mechanical system's degrees of freedom and provides best dynamical analysis.

We've used the complete method of FEM ANSYS [8]. The equations were solved by means of the sparse matrix method. We consider the homogenous piston of the pneumatic motor DAR-14M. Technical data of the pneumatic motor DAR-14M power - 10 kW; rotational frequency - 400 r/min; air pressure - 0.5 MN/m2; piston diameter – 0.082 m; piston length - 0.160 m; the hole diameter - 0.017 m. The piston is molten (its alloy consists of Aluminum, Silicon - 7% and Copper - 2%). The alloy properties are: density = 2,800 kg/m 3 [9]; elasticity modulus =71,000 MN/m2; Poisson coefficient =0.31. The own frequencies were found using the block Lanczos algorithm, which is much more efficient for large symmetric problems than the iteration method. The driving force amplitude was determined from the conditions of loading and was chosen as P=2,640 N, yet the piston loaded by its own weight. All characteristics for the piston stress-strain state were calculated in every node of the finite elements grid (SOLID 92).

III. RESULTS

We used the Static Analysis to get a stress field, strain field and displacement field in the piston. We model the piston with its right loaded end face. We determine a normal stress, a shear stress, and a principal stress. We analyze how these stresses vary with position throughout a piston. For example the X-axial stress field is shown in Fig. 2 (a). The principal stress field is shown in Fig. 2 (b).
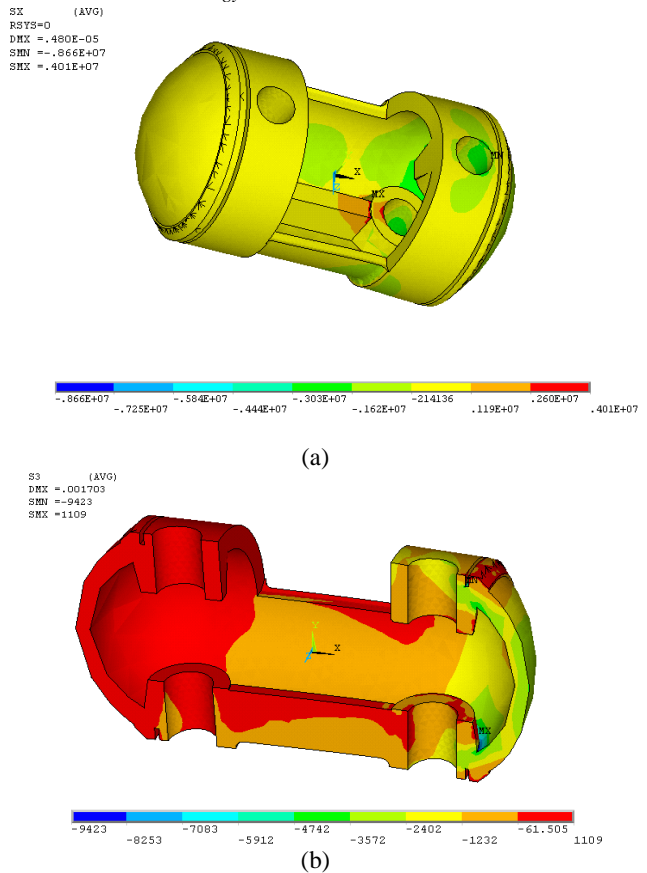

(a)

(b)

Fig. 2 The normal stress field along X axis in a piston body (a), the principal stress field (b)

One can see that a maximum stress appears near the holes for the bearing shaft as expected. This part of the piston is subjected to a deformation bending and not stretching and compression deformations only. In order to decrease this deformation it is necessary to optimize the form and parameters of the stiffening rib.

The strain field is shown in Fig. 3,4. Maximum deformation is in a stiffening rib and near the hole for a bearing shaft. See Fig. 3, Fig. 4.
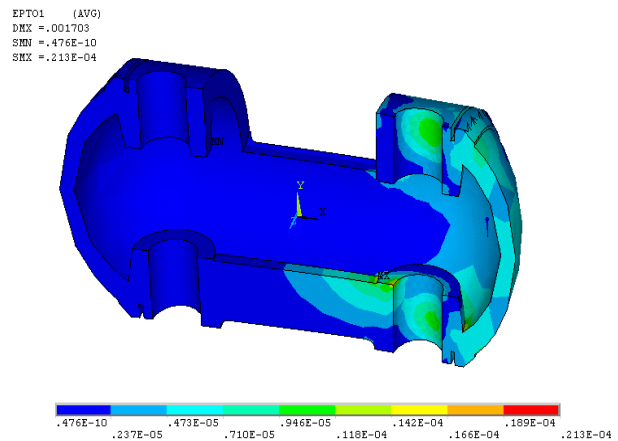

Fig. 3 Total strain field, main total deformations ε_1.
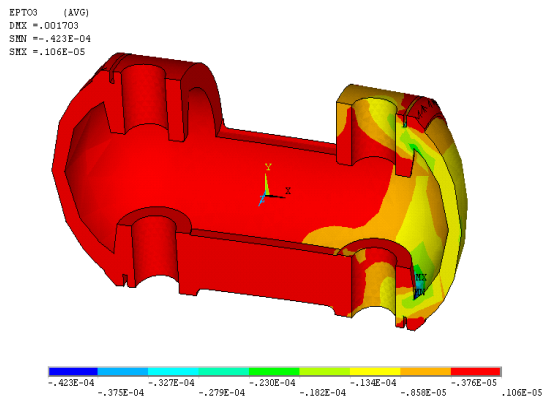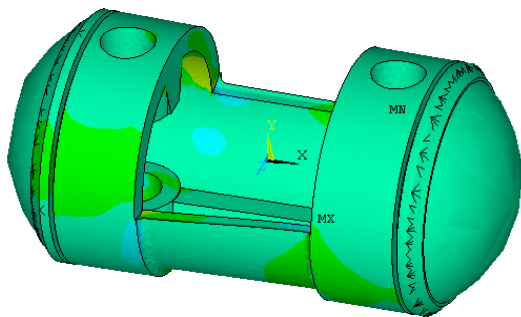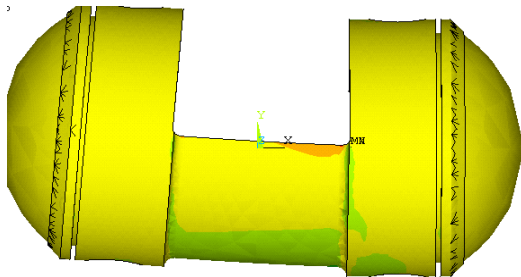
EPTO3   (AVG)
DMX =.001703
SMN =-.423E-04
SMX =.106E-05

-.423E-04   -.327E-04   -.230E-04   -.134E-04   -.376E-05
    -.375E-04   -.279E-04   -.182E-04   -.858E-05   .106E-05

Fig. 4   Total strain field, main total deformations ε_3 .
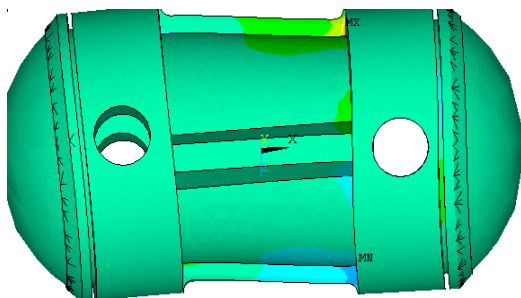
We used the Modal Analysis to get the own frequency of system and the vibrations modes.

We obtained three first own frequencies and the corresponding oscillations. The results for the own frequencies are: 685.54 Hz; 1053.8 Hz; 2317.1 Hz. Three first oscillation modes are shown in Fig. 5. The elastic deformations for the first mode of oscillation about X-axis is shown in Fig. 5 (a), the second mode of oscillation about Z-axis is shown in Fig. 5 (b),   the third mode of oscillation about Y-axis is in Fig. 5 (c). The axis Z is a perpendicular to a surface of the figure.

The first resonance frequency turned out to be equal to 685 Hz in accordance with our analytical estimate. We used   the Harmonic Analysis to determine the piston behavior near the resonance. The load (pressure) was varied   according   to harmonic   law and the   system's response was considered at the frequencies interval 670-700 Hz with a step 6 Hz. Some curves for the piston stress-strain state are shown in Fig. 6.
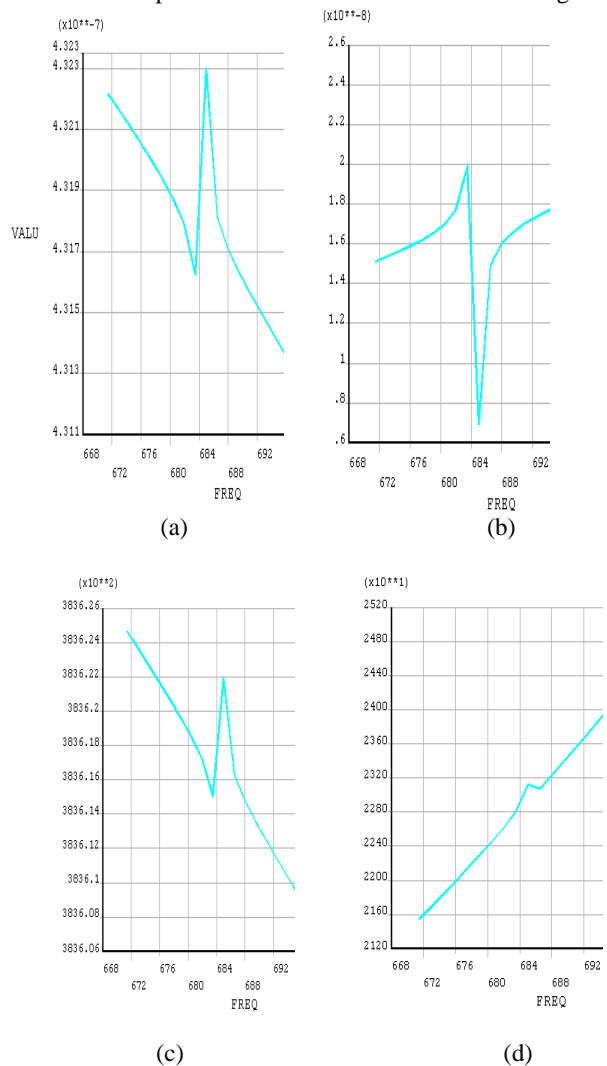

(a) - frequency 685.54 Hz,

(b) - frequency 1053.8 Hz,

(c) - frequency 2317.1 Hz

Fig. 5   First three own frequencies and oscillation modes

(a)

(b)

(c)

(d)

Fig. 6   XY- elastic deformation on the loaded end face (a), and on unloaded end face of the piston (b), the principal stress $\sigma_1 < 0$ (N/m^2) on the loaded end face of the piston (c), and the principal stress $\sigma_1 > 0$ (N/m^2) on the unloaded end face of the piston (d)

IV. DISCUSSION

As a result we obtained all characteristics for the piston stress-strain state. The concentration of stresses is in the stiffening rib and near the hole for the bearing shaft. Peak dependencies were found for the stresses relative to coordinate axes, planes, and also principal stresses near the resonant frequency. Dependence of the normal stresses on the driving force frequency was different for central point of the loaded and unloaded piston's sides and at the same time in both cases we observed the stress-jumps. The stress of a central part of unloaded piston was slightly changed near the resonant frequency. Elastic deformations were also calculated to all directions and planes. We find the different behaviour for the loaded end and unloaded end of the piston near the resonant frequency.

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