Study on the influence of supplying compressed air channels and evicting channels on pneumatical oscillation systems for vibromooshing

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Abstract. The paper presents a pneumatic system with two oscillating masses. The system is composed of a cylinder (framework) with mass m₁, which has a piston with mass m₂ inside. The cylinder (framework system) has one supplying channel for compressed air and one evicting channel for each work chamber (left and right of the piston). Functionality of the piston position comparatively with the cylinder (framework) is possible through the supplying or evicting of compressed air. The variable force that keeps the movement depends on variation of the pressure that is changing depending on the piston position according to the cylinder (framework) and to the section form that is supplying and evicting channels with compressed air.

The paper presents the physical model/pattern, the mathematical model/pattern (differential equations) and numerical solution of the differential equations in hypothesis with the section form of supplying and evicting channels with compressed air is rectangular (variation linear) or circular (variation nonlinear).

1. Introduction

The mathematical modeling of the work of some mechanic systems, frequently used in the industry, such as the vibrosmoothing device with pneumatic operation, is very important, being the first condition in achievement of the mathematical model that will be used in computer simulation. The conditions for running, execution and functioning of this device allow the formulation of some simplifying hypothesis satisfying the design. These hypotheses are defined by the vibrosmoothing device and by the elastic elements.

The vibrosmoothing device represents a perfectly centered system, meaning that the result of perturbation (elastic and dissipative) forces is oriented through the mass center of system. Therefore, all the work organs make only translational movements;

The elastic elements are considered perfectly linear and the differences between their elastic characteristics are neglected. The whole system is moving around the stable equilibrium position.

2. The mathematical calculation models

The vibrosmoothing device is represented schematically in Figure 1.
The differential equations of the system movement for the pneumatic oscillation system are (Figure 2, 3 and 4):

\[ \ddot{x} = \frac{K_e}{m_1} \cdot x - \frac{A}{m_1} (P_1 - P_4) - \frac{F_a}{m_1} \cdot \text{sign}(x) \]  

(1)

\[ u = \frac{K_e}{m_1} \cdot x + \frac{F_a}{m_1} \cdot \text{sign}(x) + A (P_1 - P_4) \left( \frac{1}{m_1} + \frac{1}{m_2} \right) \]  

(2)

where:

- \( m_1 \) is the framework mass;
- \( m_2 \) is the piston mass;
- \( K_e \) is the elastic constant of the springs mounted in parallel;
- \( F_a \) is the constant horizontal force;
- \( P_1 \) is the pressure on the left frontal surface;
- \( P_4 \) is the pressure on the right frontal surface;
- \( P_r \) is the pressure for compressed air supplying;
- \( A \) is the supplying channel for compressed air;

It is imperative to take into consideration each type of production program (range), to restrain as much as possible the area that we are using and to minimize the time of execution, all of these in order to satisfy the client’s needs, to try to classify them in order to be able to define a global software (with general rules) that is expect to fulfill each client’s needs (Figure 5) [1-6].
Figure 2. The vibrosmoothing device

Figure 3. The upper view of vibrosmoothing device

Figure 4. The front view of vibrosmoothing device
Figure 5. Schematically principle of pneumatically oscillation system

where:
1. Electromagnet
2. Cover frame of pneumatical system
3. Framework mass \( m_1 \)
4. Piston mass \( m_2 \), with diameter \( D \) and length \( l \)
5. The supplying channel for compressed air to the right chamber of the piston
6. The principal supplying channel for compressed air
7. Pressure for compressed air supplying mass \( P_r \)
8. The supplying channel for compressed air to the left chamber of the piston
9. Electromagnet
10. Fixed mass
11. Spring, \( u \) compressed distance
12. Evicting channel for compressed air from the left chamber of the piston
13. Evicting channel for compressed air from the right chamber of the piston
14. Pressure compartment \( (P_1, P_2) \)

3. The mathematical calculation model
The framework of mass \( m_1 \) is executing an absolute movement in comparison with the fixed system \( Oxy \). The relative movement of the piston of mass, \( m_2 \) is described in comparison with the system \( O_1uv \).

The differential equations of the system movement are [1-3]:

\[
\ddot{u} = \frac{K_c}{m_1} \cdot x + \frac{F_a}{m_1} \cdot \sin g(\dot{x}) + A(\dot{P}_1 - \dot{P}_2) \cdot \left( \frac{1}{m_1} + \frac{1}{m_2} \right) \tag{3}
\]

We can see that \( m_1, m_2, A, K_c, F_a \) remain constant during the movement, but the pressures \( P_1 \) and \( P_2 \) are changing in function of the piston position in comparison with the framework.
3.1. The variation of the pressures $P_1$ and $P_2$

The variable pressures $P_1$ and $P_2$ are determined by studying the system as a pneumatic cylinder with double effect. Considering the transformation process as an adiabatic process the pressures $P_1$ and $P_2$ alternating linearly in hypothesis with the section form of supplying and evicting channels with compressed air is rectangular and the pressures $P_1$ and $P_2$ alternating nonlinearly in hypothesis with the section form of supplying and evicting channels with compressed air is circular, Figures 6 and 7 [2-5].

- if: $u_{\text{min}} \leq u \leq u_1 \Rightarrow P_1 = P_0 \left( \frac{u_{2} - u_{1}}{u_{2} - u_1} \right)^x \quad P_2 = P_r \left( \frac{1 - l_1 - u_{7} + u_1}{l - l_1 - u_7 + u} \right)^x$ (4)

- if: $u_1 \leq u \leq u_2 \Rightarrow P_1 = P_0; \quad P_4 = P_r$ (5)

- if: $u_2 \leq u \leq u_3 \Rightarrow P_1 = P_0 + \frac{3}{4} P_r \left( \frac{u_{7} - u_{3}}{u_{7} - u_3} \right)^x - P_0 \frac{u - u_2}{u_3 - u_2}; \quad P_4 = P_r - \frac{1}{4} P_r \left( \frac{u - u_2}{u_3 - u_2} \right)^x$ (6)

- if: $u_3 \leq u \leq u_4 \Rightarrow P_1 = \frac{3}{4} P_r \left( \frac{u_{7} - u_{4}}{u_{7} - u_4} \right)^x \quad P_4 = \frac{3}{4} P_r \left( \frac{1 - l_1 - u_{7} + u_3}{l - l_1 - u_7 + u} \right)^x$ (7)

- if: $u_4 \leq u \leq u_5 \Rightarrow P_1 = \frac{3}{4} P_r \left( \frac{1 - u_{4}}{u_{5} - u_4} \right)^x \quad P_4 = \frac{3}{4} P_r \left( \frac{1 - l_1 + u_3 - u_7}{l - l_1 + u_4 + u_7} \right)^x + P_0 \frac{u - u_4}{u_5 - u_4}$ (8)

- if: $u_5 \leq u \leq u_6 \Rightarrow P_1 = P_r; \quad P_4 = P_0$ (9)

- if: $u_4 \leq u \leq u_4 \Rightarrow P_1 = \frac{3}{4} P_r \left( \frac{u - u_4}{u_{5} - u_4} \right)^x \quad P_4 = \frac{3}{4} P_r \left( \frac{1 - l_1 + u_3 - u_7}{l - l_1 + u_4 + u_7} \right)^x + P_0 \frac{u - u_4}{u_5 - u_4}$ (10)

![Figure 6](image)

**Figure 6.** The resultant in a transversal cross section
4. Numerical solution of the differential equations

The differential equations system was solved numerically using the Runge-Kutta method with the use of utilitarian MathCAD. Introducing the following initial values [7-9]:

- \( u_1 = 0.014 \) m;
- \( u_2 = 0.018 \) m;
- \( u_3 = 0.022 \) m;
- \( u_4 = 0.032 \) m;
- \( u_5 = 0.036 \) m;
- \( u_6 = 0.040 \) m;
- \( U_{\text{min}} = 0 \) m;
- \( U_{\text{max}} = 0.054 \) m;
- \( D = 0.050 \) m;
- \( l = 0.102 \) m;
- \( X = 1.4 \);
- \( P_r = 4 \times 10^5 \) N/m\(^2\);
- \( P_0 = 1 \times 10^5 \) N/m\(^2\);
- \( m_1 = 4.5 \) kg;
- \( m_2 = 2.5 \) kg;
- \( K_e = 80-104 \) N/m;
- \( F_a = 25 \) N

We obtained the graphics of the framework displacement (Figure 8) in hypothesis with the section form of supplying and evicting channels with compressed air is rectangular and the graphics of the framework displacement (Figure 9) in hypothesis with the section form of supplying and evicting channels with compressed air is circular [10-12].
Figure 8. The graphics of the framework displacement in rectangular section

Figure 9. The graphics of the framework displacement in circular section

5. Conclusion
The physical model of the oscillation system pneumatical is used for constructing the vibrosmoothing device with pneumatical actioning of the movement. The mathematical modelling allows us to choose the constructing and running optimal parameters in order to obtain a movement with desired amplitude and frequency for the framework, satisfying the technological process demands.

References
[1] Radu I 1997 Vibrosmoothing device with pneumatical actioning, Romanian Academy, The annual symposium of the institute of solid mechanics "SISOM 95", Bucharest, Romania, pp 391-394
[2] Radu I 2001 Physical and mathematical pattern of a pneumatical oscillation system with two masses unlinear-D.N.4, Magyar Tudomanyos Akademia, Kutatasi es fejlesztesi tanacskozas, 25(3), Kotet, Godolo, Hungary, pp 305-309
[3] Radu I 2001 *Physical and mathematical pattern of a vibrosmoothing device with two masses, unlinear,with pneumatic operation D.N. 2*, 6th International Scientific Symposium, Quality and Reliability of Machines, Nitra, Slovakia, pp 179-182.

[4] Radu I and Ursu-Fischer N 1997 *Unele aspecte privind modelarea matematidi a unui dispozitiv de vibronetezire la variatia liniar-neliniara a presiunii de alimentare (partel+2)*. Conferinta știintifica „Tehnologii și produse noi în constructia de maini TEHNOMUS IX”, Suceava, Romania, pp 110-122.

[5] Radu I and Glavan D 2002 *Study Of Line System For Vibromoothing Device D.N.6-1*, 7th International Scientific Symposium, Quality and Reliability of Machines, Nitra, Slovakia, pp 77-80.

[6] Hordieievo D and Karmalita A 2014 *Study of use application vibration equipment for oil-retaining holes at the neck of the trees*, *Problems of Applied Sciences* 2 95-100.

[7] Spadlo S, Młynarczyk P and Bańkowski D 2014 *Analysis of the Effect of Processing Vibro-abrasive Finishing on the Geometric Structure Surface Scales Ammunition and Sharp Edges*, *Journal of Achievements in Materials and Manufacturing Engineering* 66 (1) 39-44.

[8] Spadlo S and Bańkowski D 2015 *Influence Of The Smoothing Conditions In Vibro-Abrasive Finishing And Deburring Process for Geometric Structure of the Surface*, Proceedings of 24th International Conference on Metallurgy and Materials, METAL, Machine Parts Made Of Aluminum Alloys, pp 1062-1068.

[9] Glavan D O and Radu I 2001 *Vibrații mecanice în tehnică*, Edita Universitatii „Aurel Vlaicu”Arad, pp 75-92.

[10] Glavan D O and Radu I 2001 *Elemente de vibrații mecanice*, Edita Universitatii „Aurel Vlaicu”Arad, pp 29-34.

[11] Kovalev V D, Vasilchenko Y V and Dașiü P 2014 *Adaptive optimal control of a heavy lathe operation*, *Journal of Mechanics Engineering and Automation* 4(4) 269-275.

[12] Hakansson L 1999 *Adaptive Active Control of Machine-Tool Vibration in a Lathe*, Analysis and experiments, Lund University, Sweden.