PNEUMOPULSE DRIVE VIBRATING BUNKER CONTROL OVER THE MOVEMENT. BASIS DESIGN CALCULATIONS

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Abstract: *The article presents the method of calculating the pneumopulse drive of the original design vibrating hopper.*

Keywords: vibrating hopper, pneumopulse drive, the method of calculation

1. Introduction

The level of modern production and therefore competitiveness depends on the degree of an automation of production processes. One of the additional means of automation manufacture is bootable and orientation devices, among them, the most perspective are vibrating hoppers [1]. The widespread use of vibrating equipment for blanks and details' supply during manufacturing operations is due to various kinds of high efficiency of vibration mode for their artificial seizure, orientation and transportation. The vibration hoppers are built on the basis of the different drives' types – mechanical, electromagnetic, pneumatic, hydraulic and others. Among the all types of drives for bunkers with large carrying capacity should be used the vibrating hoppers with pneumatic or hydraulic drives, including their varieties – pneumopulse and hydropulse, the advantages of which are proved in comparison with the other types of actuators driven manufacturable machine vibration [2, 3, 4].

Therefore developing of new designs of vibrating hoppers on basis of the pneumopulse drive and development the appropriate method of calculation are relevant engineering and scientific tasks.

2. Basic part

Estimated diagram pneumopulse drive vibrating bunker control over the movement of shown in Figure 1.

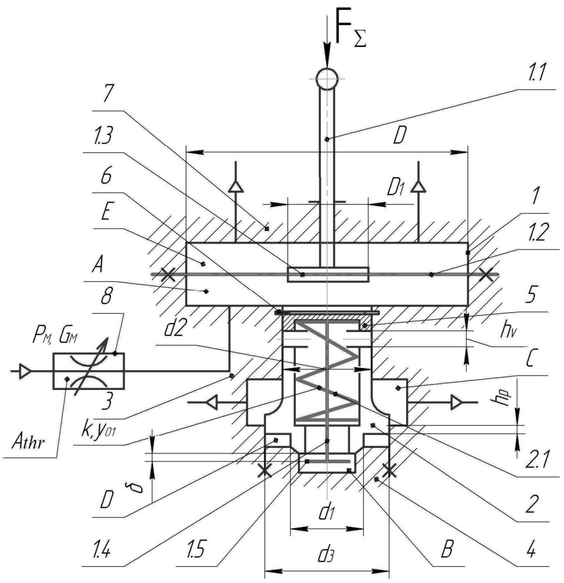

Figure 1: Estimated diagram pneumopulse drive vibrating hopper control over the movement

The main drive links are membrane pneumoengines 1, rod 1.1 which passes vibration load on the tank (conditionally not shown) winning effort F_{Σ}, which in the first approximation and neglecting friction forces in the guide rod and internal friction in the membrane pneumoengines can be represented as a constant total power

$$F_{\Sigma} = G_h + F_{sp_b} + F_{м} , \quad (1)$$

where G_h – the total weight of hopper with a rated load parts subject to vibration transport; F_{sp_b} – total force of the springs (flat), connecting with the base vibrating hopper; $F_{м}$ – strength elastic of the membrane 1.2 at the end of the stroke direct 1.1.

Manage vibrating movement pneumoengines pneumopulse drive performed three stepped valve 2 with chamfer-spool sealing elements.

Valve 2 is located in the central axial groove case 3 drive so that it connects to the shaft bore casing 3 in diameter d_2, a spool of the valve body 3 in contact with the bore diameter by d_3. Chamfer valve 2 part in contact with the saddle 4 rigidly fixed in the housing 3.

Spring 2.1 loaded valve 2, one end face of which (at the lower figure 1) rests on the bottom of the bore of the valve 2, and the second (top figure by 1) in the bottom of the bushing 5, enshrined in the housing 3 spring ring 6.

The membrane pneumoengines 1 contains a membrane 1.2, tightly and rigidly inserted between the housing 3 and lid 7.

With tough membrane 1.2 center 1.3 rigidly connected to the rod 1.1 and control rod 1.4 centered 1.5, located in the initial position at a distance (clearance) from the end facet valve 2.

Compressed air from the air-through adjustable throttle 8 is fed into the cavity *A* under the membrane 1.2 pneumoengines 1 and through the bore in the housing 3 and a central bore in the valve 2 and enters the cavity *B* clearance.

From exhaust cavity *C* freely connected to the atmosphere, the cavity *B* in the initial state of the valve 2 spool is separated, namely its positive overlap h_{∂} in diameter d_3. In the initial position of the valve 2 between the spool and valve face parts formed intermediate cavity *D*. Cavity *E* loosely connected with the atmosphere.

The valve 2 in the initial position is fixed to the saddle 4 spring 2.1 and pressure compressed air. This fixation can be described by the following inequality

$$p \cdot A_2 + k \cdot y_{01} > p \cdot A_1 , \quad (2)$$

or

$$p(A_2 - A_1) + k \cdot y_{01} > 0 , \quad (3)$$

where p – pressure air in the network; $A_1 = \pi d_1^2 / 4$ (here, in order to maximize assumed that the valve 2 ~is sealing chamfer carried by the largest diameter d_1 chamfer) – a sectional area of the valve 2 in the diameter d_1; $A_2 = \pi d_2^2 / 4$ – cross-sectional area of the valve 2 shank; k, y_{01} – respectively, stiffness and deformity previous spring 2.1.

In order to a movement of the rod 1.1 pneumoengines 1 pressure p compressed air in the pressure cavity *A* must satisfy the inequality

$$p > F_{\Sigma} / A_{ef} , \quad (4)$$

Where $A_{ef} = \left[\pi(\mathrm{D}^2 + D \cdot D_1 + D_1^2)\right] / 72 = 0,262(\mathrm{D}^2 + D \cdot D_1 + D_1^2)$ – effective membrane 1.2 area [5, 6]; D – diameter membrane 1.2 pinching the edge between the body 3 and cap 7; D_1 – diameter of the hard center 1.3. during the movement of the membrane 1.2 upwards (on figure 1), together with a rod 1.1 and a rod control 1.4 stop 1.5 the gap passes and opens the valve seat 2 of 4, as a result the cavity *B* and *D* are connected to the compressed air and begins to operate in an area A_2 of the valve spool 2 through it by the force

$$F_{pn} = p(A_3 - A_2) - 0,5k \cdot (y_{01} + h_p) - F_f \quad (5)$$

begins to move on the path positive overlap h_∂ (where $A_3 = \pi d_3^2 / 4$ – cross-sectional area of the valve spool 2; F_f – friction between the surfaces of the valve 2 and bore its location in the housing 3. During the design calculations friction force F_f can be ignored because it is small compared with drivers of air pressure and positional power spring 2.1.

Full power $F_{P\Sigma}$, acting on the valve 2 during its opening (forward motion) can be estimated dependence (for neglecting friction forces)

$$F_{P\Sigma} = F_{p\Pi} + p \cdot A_{ef}\,, \qquad (6)$$

where $$p_{1_{\min}} = 1,2 \cdot F_\Sigma / A_{ef} \qquad (7)$$

– minimum operating pressure of compressed air in the cavity *A*, the assumption that it is 20% more stationary pressure $p_0 = F_\Sigma / \mathrm{A}_{ef}$, when all moving drive links are in a state of static equilibrium.

After passing through the valve 2 overlap positive h_p the cavity *A* through cavities *B* and *D* is connected to the exhaust cavity *C*. For full movement $h_v = h_p + h_n$ the valve 2, where h_n – negative valve 2 overlap, air pressure in the cavity reduced pressure p_2 "closure" of the valve 2, which can be estimated approximately by the assumption that rod 1.1, membrane 1.2 and control core 1.4 with stop 1.5 began reverse motion under the force F_Σ. Under this assumption and disregard friction forces:

$$\begin{cases} p_2 \le F_\Sigma / \mathrm{A}_{ef} = p_0; & (8) \\ p_2 \le k \cdot (y_{01} + h_v) / (A_3 - A_2). & (9) \end{cases}$$

Comparing (8) and (9), can get the formula for calculating the spring stiffness 2.1:

$$k = \frac{F_\Sigma \cdot (A_3 - A_2)}{(\mathrm{y}_{01} + h_v) \cdot A_{ef}} \qquad (10)$$

Comparing, for example, (7) and (8), can find that the minimum pressure "closure" satisfies the inequality

$$p_{2_{\min}} \le 0,83 \cdot p_{1_{\min}}\,. \qquad (11)$$

Since the proposed air pulse drive is controlled by the movement, then from the scheme of figure 1 it is clear that the maximum possible amplitude of the rod 1.1 $H_{\max} = h_v$.

The requested effective area of the membrane A_{ef} can be calculated the assumption that the average work force pressure of compressed air for double move in $2h_k$

$$A_p = \overline{p} \cdot A_{ef} \cdot h_v + \overline{p}(A_3 - A_2) \cdot h_v \qquad (12)$$

It is full vibrational energy pneumoengines 1 and valve 2 provided that these oscillations harmonic, ie

$$E_k = m_\Sigma \omega^2 \cdot h_v^2 = 4\pi^2 m_\Sigma \nu^2 \cdot h_v^2\,, \qquad (13)$$

where $\overline{p} = (p_{1\min} + p_{2\min}) = 0,92\, p_{1\min}$ – average pressure for the cycle; $\omega = 2\pi\nu$ – angular frequency harmonic oscillations; ν – linear frequency harmonic oscillations; $m_\Sigma = m_{eng} + m_h + m_v$ – oscillating mass (here: m_{eng} – total mass pneumoengines 1, including masses membrane 1.2, rod 1.1, hard center 1.3 and control rod 1.4 with stop 1.5; m_h – the total mass of the vibrating hopper with details with account the masses attached moving parts; m_v – erected mass the valve 2 with spring 2.1, hub 5, etc.).

Equating (12) and (13), can find the formula for calculating the required minimum effective membrane 1.2 area:

$$A_{ef_{\min}} = \frac{4\pi^2 m_\Sigma \nu^2 h_v}{\overline{p}} - (A_3 - A_2)\,, \qquad (14)$$

and with account (7), will have

$$A_{ef_{\min}} \frac{1,1 \cdot F_\Sigma (A_3 - A_2)}{4\pi^2 m_\Sigma \nu^2 \cdot h_v - 1,1 \cdot F_\Sigma}\,, \qquad (15)$$

or

$$A_{ef_{\min}} = \frac{A_3 - A_2}{\frac{4\pi^2 m_\Sigma \nu^2 \cdot h_\nu}{1{,}1 \cdot F_\Sigma} - 1} \qquad (16)$$

From (16) it is clear that there $A_{ef_{\min}}$ will be a rational if

$$\alpha = \frac{4\pi^2 m_\Sigma \nu^2 \cdot h_\nu}{1{,}1 \cdot F_\Sigma} = \frac{35{,}85 m_\Sigma \nu^2 \cdot h_\nu}{F_\Sigma} > 1, \qquad (17)$$

but not significantly, and thus to make a difference $(\alpha - 1)$ was less than one, but more than zero. This value α can be achieved by selecting certain m_Σ, ν, h_ν i F_Σ. Areas A_3 and A_2 can be selected and design considerations and the ability to quickly skip valve 2 great air flow.

Operating experience of pneumatic actuators [6, 7] shows that the accepted costs of energy (compressed air) are provided for $h_p^{opt} = 2...2{,}5мм$ and landing d_3 6…7 of accuracy standart tolerance grades. Obviously to be effective emissions assign $h_n^{opt} = h_p^{opt}$, then $h_\nu^{opt} = 4mm$. With this value h_ν

$$\alpha = 0{,}1434 \cdot m_\Sigma \nu^2 / F_\Sigma > 1, \qquad (18)$$

or

$$m_\Sigma / F_\Sigma > 6{,}97 \cdot \nu^{-2}, \qquad (19)$$

where the coefficient 6.97 has dimension in SI [m].

Inequality (19) to evaluate the range of the rational ratio m_Σ / F_Σ, depending on the frequency of vibration (figure 2). The graph $m_\Sigma / F_\Sigma = f(\nu)$ shows that $0 > (\alpha - 1) < 1$ will be at relatively high frequency vibration, which are expedient selected in the zone 40 ... 100 Hz (shaded area). The ratio m_Σ / F_Σ for the physical meaning magnitude similar to the reciprocal of the of vibration acceleration $\ddot{y}_{пм}^{-1}$ working link vibrating hopper (hopper).

Assessment for calculating the dependence (19) can be written in this form

$$m_\Sigma / F_\Sigma = \ddot{y}_{пм}^{-1} > 6{,}97 \cdot \nu^{-2}, \qquad (20)$$

or

$$m_\Sigma / F_\Sigma = \ddot{y}_{пм} > 0{,}143 \cdot \nu^2, \qquad (21)$$

where the numerical coefficient has the dimension of 0,143 in SI [m].

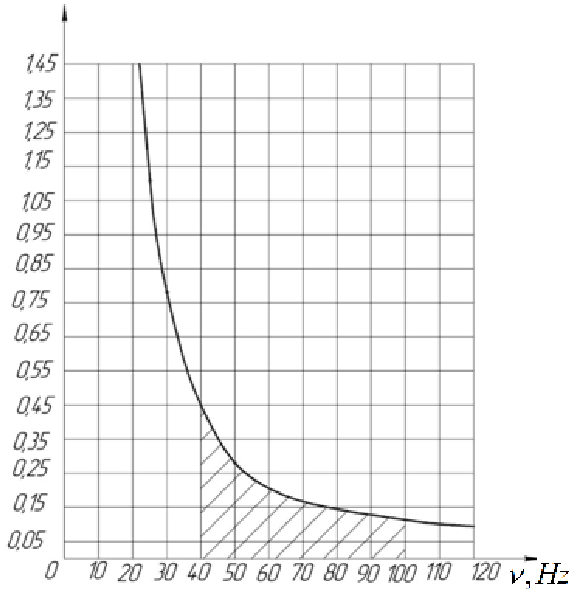

Figure 2 - Theoretical graph for selecting m_Σ / F_Σ the frequency ν

With the help graphics, showed in figure 2, can set the rational range of variation $\ddot{y}_{пм}$ in the frequency range (40 ... 100) Hz. For example ν =40Hz limit acceleration $\ddot{y}_{пм}$ <228,5 m/s^2, and for ν = 100 Hz – $\ddot{y}_{пм}$ <1430 m/s^2.

According to (21) the total effort

$$F_\Sigma < 0{,}143 \cdot m_\Sigma \cdot \nu^2. \qquad (22)$$

For $\nu = 40$Hz and $m_\Sigma = 10$kg total effort and can not exceed the limit $F_{\Sigma\max} < 2285Н$, and for $\nu = 100$Hz i $m_\Sigma = 100$kg – $F_{\Sigma\max} < 143000Н$. Obviously, for vibration frequencies close to 100Hz and large values of masses m_Σ must be very powerful air pulse drive of this type, for which the nominal values of air pressure p = (0.4 ... 0.6) MPa will have large dimensions or those parameters that are impossible to realize. In this case it is better to use hydropulse drive. Described drive should be used for vibration bunkers $m_{\Sigma} = (10...20)$ kg of $\nu = 40...60$ Hz frequency range, possible values for which efforts F_{Σ} will be in the range $m_{\Sigma} = 10$kg and and $\nu = (40...60)$Hz – $F_{\Sigma\max} = (2285...3575)$ H; $m_{\Sigma} = 20$ kg and $\nu = (40...60)$Hz – $F_{\Sigma\max} = (4570...7150)$ H. These boundary values can be used to calculate the formula (16) effective membrane 1.2 area of 1.2 (figure 1).

If the relationship (7) to accept that $p_{1\min}$ lies within the standard of the pressure in the air network p = (0.4 ... 0.6) MPa, then the values found $F_{\Sigma\max}$ for m=(10...20) kg and $\nu = 40...60$Hz and can estimate the required range of effective membrane area 1.2:

$$A_{ef} = 1,2F_{\Sigma\max} / P_{1\min}. \qquad (23)$$

For : $p_{1\min} = 0,4$ MPa i $F_{\Sigma\max} = (2285...3575)$ H –

$$A_{ef} = \frac{1,2(2285...2575)}{4\cdot 10^5} = (6,86\cdot 10^{-3}...1,07\cdot 10^{-2})\text{m}^2;$$

$p_{1\min} = 0,6$ i $F_{\Sigma\max} = (4570...7150)$ H –

$$A_{ef} = \frac{1,2(4570...7150)}{6\cdot 10^5} = (9,14\cdot 10^{-3}...1,43\cdot 10^{-2})\text{m}^2$$

If structural considerations appoint diameter D_1 hard membrane center, with well-known formula for A_{ef} (see deciphering the formula (4)) can be determined diameter D clamping membrane 1.2 for addiction

$$D = -0,5D_1 + \sqrt{(0,5D_1)^2 - (D_1^2 - 3,82\cdot A_{ef})}. \qquad (24)$$

For the average load vibration bunkers in large automated production $m_{\Sigma} = 5...6$kg (small size parts) and adjust the frequency range of vibration (40...60)Hz and pressure $p_{1\min} = (0,4...0,6)$MPa boundary value $F_{\Sigma} < (1144...2574)$ H. Choosing the intermediate value $F_{\Sigma} = 1300$ H and the diameter of rigid center $D_1 = 42$mm calculated by the (24) diameter D:

$$D = -0,5\cdot 42 + \sqrt{(0,5\cdot 42)^2 - (42^2 - 3,82\cdot 2,6\cdot 10^3)} = 71,8mm,$$

де $A_{ef} = \frac{1,2\cdot F_{\Sigma}}{p_{1\min}} = \frac{1,2\cdot 1300}{6\cdot 10^5} = 2,6\cdot 10^{-3}\text{m}^2.$

Diameter D rounded to integer $D = 74$ mm, which includes the size chamfers in the housing 3 and lid 7.

Cycle of the workflow vibrating hopper in the established mode can be described by a simplified sequence diagram, showed in figure 3, which shows two pulses of pressure 1 changes in the pressure cavity A (see figure 1) and moving 2 the valve 2 with rod 1.1.

Assuming that the movement *h* of the rod 1.1 without phase shifts monitors the law of change pressure energy (compressed air) in the cavity *A*, period *T* the pressure change *p* and movement *h* can determine according to diagram, a simple dependence:

$$T = \nu^{-1} = t_{ep} + t_{gr} + t_{pr} + t_{rd} = t_p + t_r + t_{er}, \quad (25)$$

where t_{gr} – time the growth pressure energy (second pulse) in the cavity *A* from $p_{2\min}$ to $p_{1\min}$; t_{pr} – time preservation energy level $p_{1\min}$ for which the valve 2 shaft with rod 1.1 are approximate distance $(\text{h}_p - \delta) + 0,5h_n$; t_{ep} – time of endurance pressure level $p_{2\min}$, at which the rod 1.1 and valve 2 returns to its original position, that is $t_{ep} = t_r$ (here t_r – time a flyback valve 2 and rod 1.1); t_{rd} – time reducing the pressure energy in the cavity *A* of the level $p_{1\min}$ to $p_{2\min}$; $t_{er} = t'_{p_0}$ – time of endurance rod 1.1 and valve 2 in the lower initial position until the pressure energy in the cavity *A* does not grow from level $p_{2\min}$ to level p_0 (time growth t'_{p_0}).

Theoretically, after moving rod 1.1 with stop 1.5 on distance gap δ and undermining of the valve 2 seat, the latter can move at a faster rate than the rod 1.1, since the pressure energy begins to act on a large area A_2 (see dependence 5). On the sequence diagram (curve 2) at this point showed fracture dependence *h(t)*.

To estimate the components *T* can assumed that the movement of the valve 2 and rod 1.1 at sites of direct and reverse of moves are of uniformly accelerated, then, for neglecting friction force, the known formulas of physics of uniformly accelerated motion, can find:

$$t_{p_1} = t_h = \sqrt{\frac{2\delta \cdot m_{\Sigma_1}}{\mathrm{F}_h}} = \sqrt{\frac{2\delta(m_{eng} + m_h)}{(p_{1_{\min}} \cdot A_{ef})}}\ ; \quad (26)$$

$$t'_{в} = t_{pr} = \sqrt{\frac{2(\mathrm{h}_h - \delta + 0,5h_n) \cdot m_\Sigma}{(F'_{pn} + p_{1\min} \cdot A_{ef})}}\ ; \quad (27)$$

$$t''_{в} = t_{rd} = \sqrt{2 \cdot 0,5h_n \cdot m_\Sigma / F'_{рн}} =$$

$$= \sqrt{\frac{h_n \cdot m_\Sigma}{\overline{p}(A_3 - A_2 + A_{ef} - 0,5k(y_{01} + h_v)}}\ , \quad (28)$$

$$t_r = t_{ep} = \sqrt{\frac{2 \cdot h_v \cdot m_\Sigma}{\left[-p_{2\min}(A_3 - A_2) + 0,5k(y_{01} + h_v)\right]}}\ , \quad (29)$$

where $F_h = p_{1\min} \cdot A_{ef}$ the force that moves the rod 1.1 towards δ ;
$F'_{pn} = p_{1\min}(A_3 - A_2) - 0,5k(y_{01} + h_v - 0,5h_n)$
Component t_{pr} of the cycle of vibration drive can be measured by a simple ratio, found from (25), if given the frequency of vibration ν and the calculated values t_p and t_r the formulas (26)…(29).

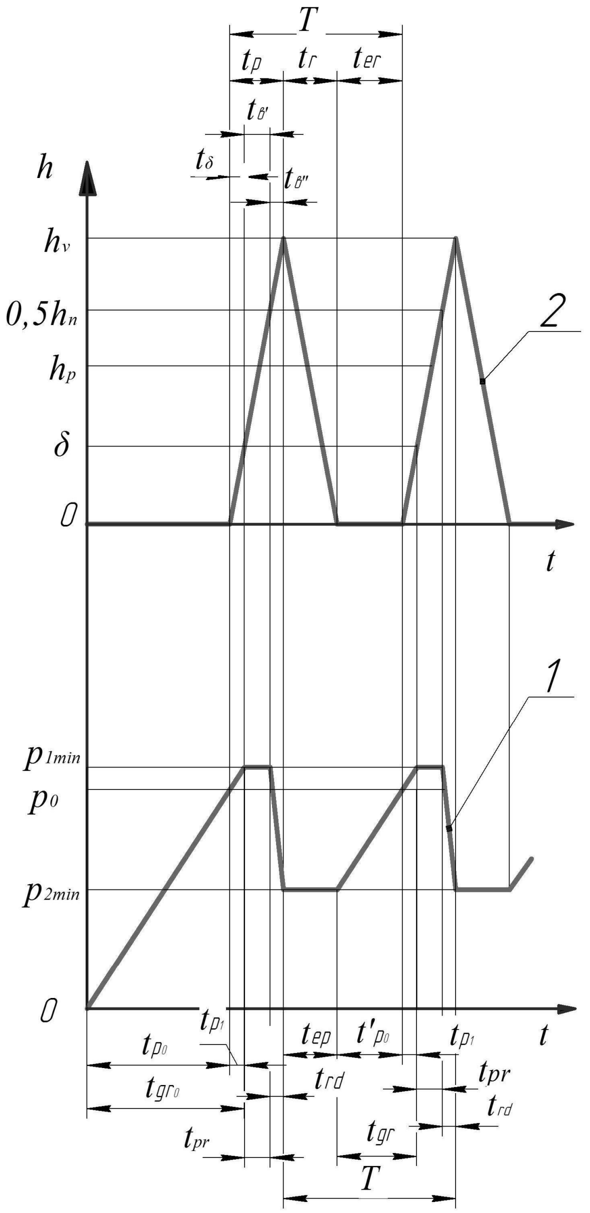

Figure 3 – The simplified sequence diagram of a vibrating hopper drive control of the movement

According to the sequence diagram (see figure 3)

$$t_p = t_\delta + t'_{в} + t''_{в}, \quad (30)$$

then

$$t_{er} = T - (\mathrm{t}_p + t_r) = \nu^{-1} - (\mathrm{t}_p + t_r)\ . \quad (31)$$

It is known [5] that if the pneumatic systems pressure drop of energy is small and few number (criterion) of Mach, $M = V / V_s$ (here V – flow speed, $V_s = \sqrt{\kappa / \rho}$ – the speed of sound in the same spot flow), then the compressibility of air can not take into account all the calculations hold for incompressibility liquid. Usually, the air supply networks in manufacturing is carried out on large receivers, then, assuming that thermodynamic process in vibration actuators adiabatic, flow speed V_A, supplied to the cavity A, can be estimated from the dependence [5]:

$$V_A = \mu \sqrt{2 \cdot RT \cdot \frac{\kappa_a}{\kappa_a - 1} [1 - \left(\frac{P_{A_2}}{P_{A_1}} \right)^{\frac{k_a - 1}{k_a}}} , \quad (32)$$

where $k_a = 1,4$ [1] – adiabatic index; R=287,14 J/(kg·K) – gas constant for air [5]; $T = T_1 = T_2$ – absolute temperature in the air network T_1 and T_2 the cavity A, K. Can make the calculations for project $T_1 = 293K$; p_{A_1}, p_{A_2} – air pressure energy in the network and in the cavity A and at the time of energy feed; $\mu = 1 / \sqrt{1 + \xi}$ – coefficient expenses (here ξ – coefficient of resistance throttle input 8 (see figure 1, can be taken $\xi = 0,4...0,8$ [5, 7]).

Assuming that at the time of filing of energy $p_{A_2} = p_a$ (here $p_a = 0,1$МПа – atmospheric pressure), then for $p_{A_1} = 0,6$ MPa can find

$$V_A = 0,79 \sqrt{2 \cdot 287,14 \cdot 293 \frac{1,4}{1,4 - 1} \left[1 - \left(\frac{0,1}{0,6} \right) \frac{1,4 - 1}{1,4} \right]} = 383,4 m/s,$$

where $\overline{\xi} = (0,4 + 0,8) / 2 = 0,6$ – average coefficient of resistance, in which $\overline{\mu} = 1\sqrt{1 + 0,6} = 0,79$.

With this speed Mach's number $M = 383,4 / 340 = 1,13$, that more than 0.25 and compressibility of energy should be considered [5].

Since the when valve opening 2, exhaust carried out in the atmosphere, then $p_{1\min} = 0,4...0,6$ MPa and $p_a = 0,1$ MPa, $\beta_1 = p_a / p_{1\min} = 0,1 / (0,40...0,60) = 0,25...0,17$ i $\beta_2 = p_a / p_{2\min} = p_a / (0,83 p_{1\min}) =$ $0,1 / (0,33...0,50) = 0,30...0,2$, that according to the works [5 –7] when the calculated values of less than 0.5 means that the mode of energy leakage through the valve 2 is supercritical and energy expense through the valve 2 depends on the pressure before the valve 2. For such values of mass flow G_{ex} through the open valve 2 can be calculated by the formula [5]

$$G_{ex} = \overline{\mu}_v \cdot A_v \cdot p_{2\min} \sqrt{\frac{1}{2R \cdot T}} , \qquad (33)$$

where $A_v \approx \pi d_1 \cdot h_v \cdot \sin(\alpha_v / 2)$ – area slit a throttle open valve 2, $\overline{\mu}_v = 0,6$ – average rate of expenses through conical valve; $\alpha_v = 60^\circ ...90^\circ$ – angle conical chamfer valve 2; $h_v = 4 \cdot 10^{-3} m$ – movement of valve 2 (opening).

If the thermodynamic process during the adiabatic (as we assumed), then mass ΔM energy entering the cavity vibration actuator (A, B, and bore of the valve 2 and housing 3 (see figure 1) and are thrown during the exhaust can be estimated from the known dependence [5]

$$\Delta M = W_A (p_{1\min} - p_{2\min}) / (\mathrm{RT}) =$$

$$= 0,17 \cdot \mathrm{W}_A \cdot p_{1\min} / (RT), \qquad (34)$$

where W_A – the total amount of the initial pressure valve 2 cavities.

According to the accepted duty cycle sequence diagram vibration actuator by us and dependence (33) components t_{gr} and t_{rd} cycle period T can also be estimated by the formula [5]:

$$t_{gr} = \frac{\Delta M}{G_{rec}} = \frac{0,17 \cdot W_A p_{1\min}}{\bar{\mu} A_{thr} p_{1\min} \sqrt{2RT}} = \frac{0,17 W_A}{\bar{\mu} \cdot A_{thr} \sqrt{2RT}}, \quad (35)$$

$$t_{н} = \frac{\Delta M}{G_{ex}} = \frac{0,17 \cdot W_A p_{1\min}}{\bar{\mu} A_v p_{2\min} \sqrt{2RT}} = \frac{0,17 W_A}{\bar{\mu}_к \cdot A_v \sqrt{2RT}}, \quad (36)$$

where A_{thr} – an adjustable cross sectional area of the throttle 8 (see. figure 1).

Shown in sequence diagram (see figure 3) relatively slow set pressure and quick exhaust energy carrier and creates the most favorable mode of operation vibration hopper, that promotes fast moving of details on the guide rails hopper. During the project can calculate approximately assume that $t_{gr} \approx 5 \cdot t_{rd}$ та $t_{op} = t_{ep} \approx 3t_{rd}$, then

$$T = 3t_{rd} + St_{rd} + 3t_{rd} + t_{rd} = 12 \cdot t_{rd}. \quad (37)$$

where

$$t_{rd} = T/12 = 0,083T = 0,083/\nu. \quad (38)$$

If we take the value of the volume W_A (it is clear that for high frequency vibration, this volume should be as small as possible), is based on the dependence (36) and (38) get the formula to calculate the necessary area A_v:

$$A_v = \frac{0,17 W_A}{\bar{\mu}_v \cdot t_{rd} \sqrt{2RT}} = \frac{2,05 \cdot W_A \cdot \nu}{\bar{\mu}_v \sqrt{2RT}} \approx \frac{1,45 \cdot W_A \cdot \nu}{\bar{\mu}_v \sqrt{RT}}. \quad (39)$$

For the resulting area A_v, determine the diameter d_1 of the valve 2 of known dependence:

$$d_v = \frac{A_v}{\pi h_v \sin(\alpha_v / 2)}. \quad (40)$$

In the proposed design of vibration hopper and constructively accepted control rod 1.4 sizes and stops 1.5, sometimes appropriate for a given diameter d_1, area of slit A_v, angle α and move h_v the valve 2, calculate the limits of changes W_A for the frequency range of vibration $[\nu_{\min}, \nu_{\max}]$ by dependence

$$W_A = \frac{A_v \bar{\mu}_v \sqrt{RT}}{1,45 \cdot \nu} \approx 0,69 \cdot A_v \bar{\mu}_v \sqrt{RT} \cdot \nu^{-1}, \quad (41)$$

for example, $\nu = 40...60\text{Hz}$,

$$A_v = \pi d_1 h_v \cdot \sin(\alpha_v / 2) =$$
$$= 3,14 \cdot 22 \cdot 4 \cdot \sin(60^\circ / 2) = 138 \cdot 10^{-4} m^2,$$

$d_1 = 22mm,\ h_v = 4mm$ i $\alpha_m = 60^\circ$;

$$W_A = \frac{1,38 \cdot 10^{-4} \cdot 0,6 \sqrt{287,14 \cdot 293}}{1,45(40...60)} = (2,8...4,1) \cdot 10^{-4} m^3.$$

During the project area calculation Vibration actuator difference ($A_3 - A_2$) can be determined from (28) as $t_{rd} = t''_в$:

$$A_3 - A_2 = \frac{1}{\bar{p}} \left[\frac{m_\Sigma \cdot h_n}{t^2_{rd}} + 0,5(y_{01} + h_v) \right] =$$
$$= \frac{(m_\Sigma h_n - \bar{p} A_{ef} t^2_{rd}) \cdot A_{ef}}{t^2_{rd} (\bar{p} A_{ef} - 0,5 F_\Sigma)} =$$
$$= \frac{(m_\Sigma \cdot h_n \cdot \nu^2 - 6,34 \cdot 10^{-3} p_{1\min} A_{ef}) A_{ef}}{6,34 \cdot 10^{-3} (p_{1\min} A_{ef} - 0,5 F_\Sigma)}. \quad (42)$$

Dimensions of the quantities in the formula (41) in the international system of units.

For the resulting formula (41) values ($A_3 - A_2$) and dependence (10) calculate spring stiffness 2.1 (figure 1), and from inequality (2), turning it into equality introduction of safety factor for $K_{sf} = 1,1...1,2$ and equating $p = p_{1\min}$, determine the cross-sectional area A_2 of the valve 2:

$$p_{1\min} \cdot A_2 + k \cdot y_{01} = K_{sf}\, p_{1\min} A_1 ,$$

where

$$A_2 = \frac{K_{sf}\, p_{1\min} \cdot A_1 - k \cdot y_{01}}{p_{1\min}} = \frac{K_{sf} \cdot A_1 - k \cdot y_{01}}{p_{1\min}}. \quad (43)$$

Similar methods were approbated during the creation of vibrating machines scientific school "Theory and Calculation of Vibration and development processes and equipment" Vinnytsia National Technical University. The inaccuracy presented in the article engineering methods of calculation vibrating hopper about 5%.

Conclusion

The technique of calculating air pulse drive vibration hopper allows to define all the basic geometric parameters of the drive using simple ratios. Examined can refine depending on the results of theoretical and experimental studies of the dynamics of this.

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