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AN ORIGINAL SOLUTION FOR A PNEUMATIC TOOL USED IN FINISHING AND REPAIRING ON CONSTRUCTION ACTIVITY

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ABSTRACT

In this paper an original solution for a pneumatic tool used in finishing and repairing on construction activity is presented. After assembly equipment description, some theoretical aspects for establish the main functional parameters are presented.

1.Introduction

A problem that frequently occurs in a building site is the demolishing, repairing and finishing operations.

For executing the up mentioned activities regarding the specific conditions of each site, different kinds of technological equipments and action procedures are used. The tool that makes the object of this paper is pneumatically driven and the active part of it uses the repeatable mechanical shock.

2.General presentation

The percussion technological equipment used in repairing and finishing operations are pneumatic, hydraulic, electrical or diesel engine percussion hammer.

There are two types of those equipments:

-portable direct used by human operator

-attached on the working equipment of some construction machinery

The pneumatic tool as can be seen in pic.1, is a modular type and can be used both in percussion or rotation-percussion mode.

In standard option the tool is made by two main modules: the percussion module and the rotation module.

For the drive of the percussion module a double action directional valve in the active and also in dead stroke is used for increased strike energy.

The percussion module is composed by: the plunger 1, springs 2, piston 3 and main shaft 4.

The compressed air gets through orifice "a", through channel "b" or channel "c" moving alternative the plunger that strikes the piston 3.

For rotation the Ljungstrom principle was used by means of the rotor has a number of constructive mobile throttles in which the air detents.

The rotation movement of the main shaft is obtained as a result of air flowing through angled throttle "A" of the rotor "R". The flowing is caused by the difference of pressure before and after the throttle.

When the air is getting through the throttle it detents and rotates the rotor "R". The loses of air are diminished by using special seals. The radial forces that appear as a result of motion resistance and the radial forces are taken by bearings. The motor has an increased adiabatic efficiency.

3.Calculation elements

The drive calculation for linear or rotaries motors is difficult because all the parameters that characterize the functioning are variable and the air flow regime between the inlet and outlet is not permanent. The initial phase of the percussion begins when the piston starts to move. In this moment the force generated by pressure is balanced by the force that opposes the movement of the plunger. The final phase is when the plunger is at the end of the stroke. For calculation we consider that the plunger goes to the left side.The force balance is between the elastic force generated by the spring and the force generated by the resistance at movement (friction).

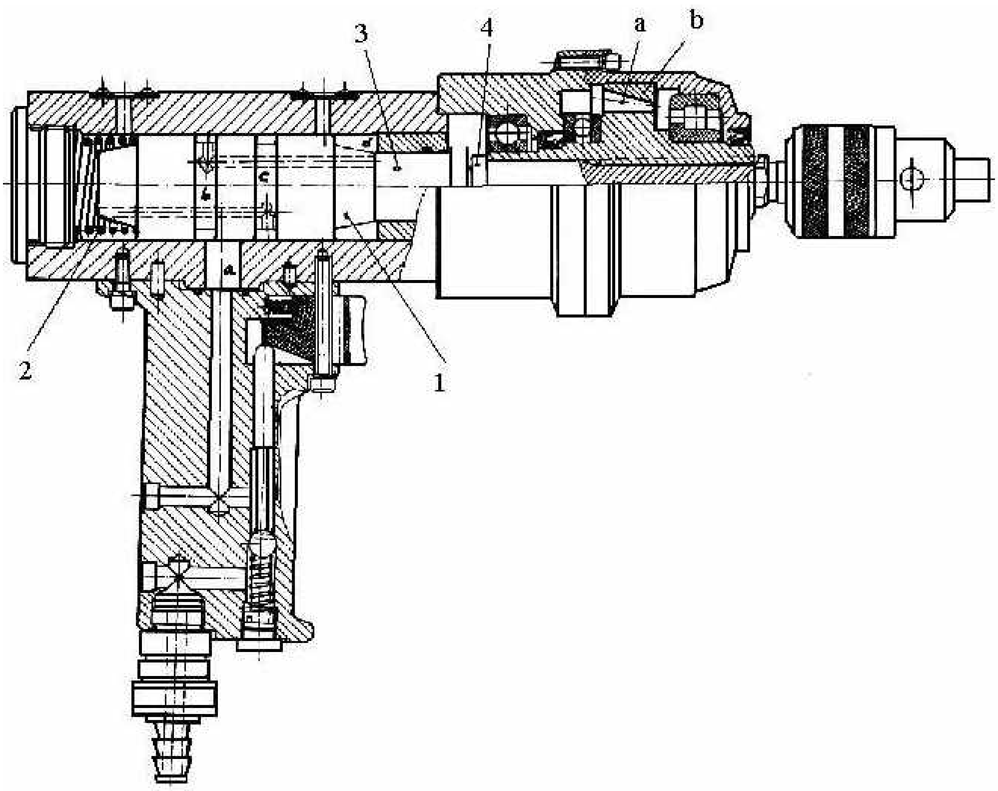

Figure. 1

The movement equation that can be written as:

$$m\ddot{x} = S(p - p_o) - k(x - x_o) - F \, , \qquad (1)$$

where m - is movement system mass, s – is active surface, p – is the pressure, p_0 – is the counter pressure, k – is elastic constant of spring, x – is the stroke of the plunger, x_0 – is pre compression of the spring; the force balance is: $F = F_0 + F_f + F_u$ (F_0 – pre compression spring force, F_f – resistance at movement, F_u – active force)

The variation of air pressure equation can be written as:

$$Vdp + \chi pdV = \chi T_1 R \, dG \, , \qquad (2)$$

where V – is enclosure volume, dp – is pressure variation, γ – adiabatic constant, dV – is volume variation, T – is air temperature, R – is air constant, dG – is air quantity variation, p – is enclosure pressure.

From relation (2) by making the changes we can obtain:

$$dp = \frac{\chi}{x}\left(\frac{RT_1}{S}\dot{G} - p\dot{x}\right)dt \, . \qquad (3)$$

The equations (1) and (3) can be integrated with finite interval method, and for the time intervals Δt_i we can obtain the following expressions:

$$\Delta p_i = \frac{\chi}{x_{i-1}}\left(\frac{RT_1}{S}\dot{G}_{i-1} - p_{i-1}x_{i-1}\right)\Delta t_i \, , \qquad (4)$$

$$p_i = p_{i-1} + \Delta p_i \, , \qquad (5)$$

$$\ddot{x}_i = \frac{1}{m}\left[(p_i - p_o)S - k(x_{i-1} - x_o) - F\right], \qquad (6)$$

$$\ddot{x}_{ic} = 0{,}5(\ddot{x}_i + \ddot{x}_{i-1}), \qquad (7)$$

$$\Delta\dot{x}_i = 0{,}5(\ddot{x}_i + \ddot{x}_{i-1})\Delta t_i \, , \qquad (8)$$

$$\dot{x}_i = \dot{x}_{i-1} + \Delta\dot{x}_i \, , \qquad (9)$$

$$\dot{x}_{ic} = 0{,}5(\dot{x}_i + \dot{x}_{i-1}), \qquad (10)$$

$$\Delta x_i = \dot{x}_{ic}\Delta t_i + 0{,}5\ddot{x}_{ic}\Delta t_i^2 \, , \qquad (11)$$

$$x_i = x_{i-1} + \Delta x_i \, , \qquad (12)$$

where Δp_i – pressure variation for finite time intervals, x_i – system acceleration in i_i interval, x_{ic} – constant system acceleration in Δt_i

interval, Δx_i - increase of plunger velocity in Δt_i interval, x_i – final velocity, x_{ic} – constant system velocity in Δt_i interval, Δx_i – system movement in Δt_i interval, x_i – system stroke.

At the end of i interval the value of air flow G_i that pass through the percussion mechanism can be obtained with the following relations:

-for the sub critic flow regime

$$\dot{G} = 8{,}28\mu f \sqrt{p_1\rho_1\left[\left(\frac{p}{p_1}\right)^{\frac{2}{\chi}} - \left(\frac{p}{p_1}\right)^{\frac{\chi+1}{\chi}}\right]}, \quad (13)$$

-for critic flow regime

$$\dot{G} = 2{,}145\mu f\sqrt{p_1\rho_1}, \quad (14)$$

where μ – is flow coefficient, f – is inlet flow area, p_1 – inlet air pressure, ρ – air density.

The total time of drive phase can be obtained with:

$$t_a = t_1 + t_2 + t_3, \quad (15)$$

where t_1 – is starting time, t_2 – is drive time, t_3 – is post filling time

The total time of functioning cycle is:

$$t = t_a + t_p, \quad (16)$$

where t_p – is passive time

Only a part of the plunger striking energy is transmitted to the active part due the percussion process because of internal losses:

$$E = \Delta E + E_1 + E_2, \quad (17)$$

where ΔE – is energy losses in striking moment, E_1 – plunger energy after striking, E_2 – active part energy after striking.

It is well known that:

$$\begin{gathered} E = 0{,}5 m_s v_s^2; \\ \Delta E = \frac{m'}{m_s + m'} E(1-c_r)^2; \\ E_1 = m_s v_s'^2; \\ E_2 = 0{,}5 m' v'^2, \end{gathered} \quad (18)$$

where m_s - is plunger mass, m' – is active part mass, c_r – is restitution coefficient, v_s – is plunger velocity in the moment of strike, v_s' – is plunger velocity after strike.

The power of the strike can be written as:

$$P = \frac{E \cdot n}{6 \cdot 10^3}, \quad (19)$$

where energy E is in daNm, and n is the number of strikes per minute

$$n = \frac{60}{t}. \quad (20)$$

The efficiency of the tool can be calculated with:

$$\eta = \frac{E_2}{E}. \quad (21)$$

The rotational – pneumatic - static motor is a volumetrically motor that transforms the pneumatic energy – potential energy – in mechanical energy that is transmitted to the ending shaft.

The theoretical power of the motor can be written as:

$$P = \dot{G} R T \eta \frac{n}{n-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right], \quad (22)$$

where n – is the exponent, p_1 – is the air pressure on inlet, p_2 – is the air pressure at outlet.

Because the throttle has a small length, considering the air velocity through it high, a heat exchange is not possible, so an adiabatic detent take place.

For pressure p1 and p2 calculation the following considerations were taken into account:

-at Mach 1, the aerodynamic process is very complex, this complexity is resulting from the instability of the process. In this kind of process undesirable things like shock waves and vibrations may occur, so the values round of 0.8 –1.0 must be avoided.

-it was accepted that the capacity of air working machinery decreases with the square of pressure and the pneumatic tools are designed to work at optimum to a 7 bar pressure.

After all we' ve mentioned above we can clearly say that for obtaining a higher power at the final shaft the pressure difference must be raised.

Also to avoid vibration, the critical velocity must be a sub sonic one.

The air velocity at the outlet of the throttle is:

$$w_2 = \sqrt{2g\frac{\chi}{\chi-1}\cdot\frac{p_1}{\gamma_1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{\chi-1}{\chi}}\right]}, \quad (23)$$

where g – is gravity acceleration, γ_1 – specific mass at pressure p_1

The calculation of critical pressure can be done with:

$$p_{cr} = p_1 \left(\frac{2}{\chi + 1} \right)^{\frac{\chi}{\chi - 1}} . \qquad (24)$$

If $p_a > p_{cr}$ and $w_2 < w_{cr}$, the throttle must be a convergent one.

The air flow can be written as:

$$\dot{G} = \mu f \sqrt{\frac{2\rho_1 \chi}{\chi - 1} p_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{2}{\chi}} - \left(\frac{p_2}{p_1} \right)^{\frac{\chi + 1}{\chi}} \right]} , \qquad (25)$$

where f – is the throttle aria, μ – is air flow coefficient depending on the angle of convergence of the throttle

The air flow consumption is:

$$Q = 15{,}83 V_m n_m \frac{p_1}{T_1} , \qquad (26)$$

where V_m – is motor capacity, n_m – is the shaft rotational speed.

The capacity of the motor is:

$$V_m = z \frac{\dot{G}}{\rho_n} , \qquad (27)$$

where z – is the number of throttles on the rotor, ρ_n – is air density in normal conditions of pressure and temperature.

The power consumption at the shaft is:

$$P_r = k_1 P z \eta_m , \qquad (28)$$

where k_1 – is the transforming coefficient from pneumatic detent to rotation, η_m – is the mechanical efficiency.

4. Conclusions

Experiments shown that the modular tool functioned at calculated parameters. Depending on the characteristics of the operations different kind of tools can be used.

The construction that is proposed has a high efficiency, and from the technical and economical point of view has sufficient working and constructional arguments.

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