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PNEUMATIC ACTUATOR WITH CONSTANT VELOCITY MODE IN RECIPROCATING MOTION

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Abstract

Motion systems with speed control mode are widely used in industry. This paper reports on an open loop pneumatic actuator that provides reciprocating motion with constant velocity mode. Computer simulations of the dynamic behavior of these actuators show their acceptable performance and high robustness. In addition, the estimation method, which allows performing the calculation of the actuator key parameters, is described. Design of the pneumatic actuator for a printing machine has been considered as the practical example.

Keywords: pneumatic actuator, open loop, reciprocating motion, constant velocity, printing machine

1 Introduction

When the design engineer begins to develop any form of automatic equipment he is confronted with two important problems: the first one is related to the mechanical and control design of the functional device, the second problem is a commercial one and pertains to designing with reference to the cost of manufacture.

In order to solve the first problem, especially when automatic control of complex motion is required, a wide knowledge of the principles underlying those mechanical movements, which have proved to be successful, is very helpful, even to the design engineer who has had extensive experience.

The second problem mentioned, that of cost, is directly related to the design itself, which should be reduced to the simplest form consistent with successful operation. Simplified designs are usually not only less costly, but more durable. Almost any action or result can be obtained mechanically if there are no restrictions on the number of parts used and the manufacturing cost, but it is evident that a design should pass the commercial as well as the purely mechanical test. For this reason, it is advisable for the design engineer to carefully study dynamic mechanical systems which have previously been applied to commercial machines.

Motion systems with speed control mode are widely used in arc welding machines, painting and printing equipment, in the inspection devices of scanning motion systems, cutting machines for plastic, wood and fabric materials, gluing and others. These systems can generally be divided into two groups, namely open loop and closed loop velocity control. In some applications the position signal is used as an additional signal for the technological application or for the control algorithm in the cascading loops configuration.

In industrial applications electro mechanical motion systems are widely used for speed control, however thanks to advances in pneumatic control theory, specifically the combination of fast-acting valves, advanced electronics, and software; servo pneumatics offer a practical alternative in intelligent-motion applications at a sensible price. In manufacturing and industrial process equipment, open loop pneumatic actuators with velocity control are rarely used. This is due to the limitations of their performance. For instance, the output of such systems is sensitive to plant parameter variations, and in a few cases, to the change of the force disturbance (both external and internal). Therefore, these systems are usually used where the requirement of the output stability is not critical. Such systems may be utilized, for example, in painting equipment, rough inspection devices, wood and fabric material industry, gluing and others.

This paper reports on the open loop pneumatic actuating system that provides the reciprocating motion with constant velocity mode.

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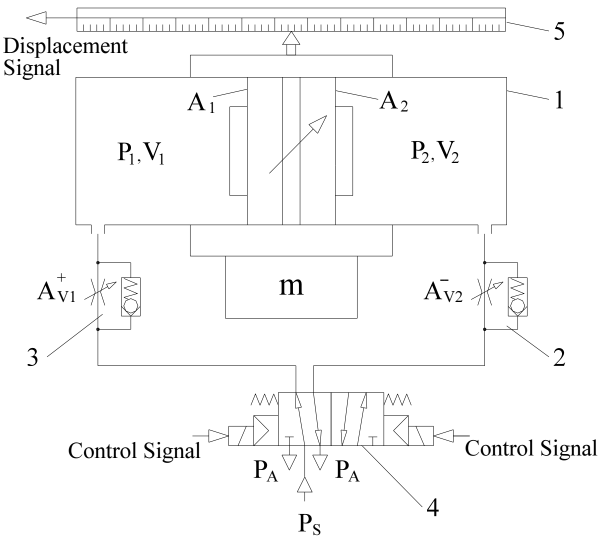
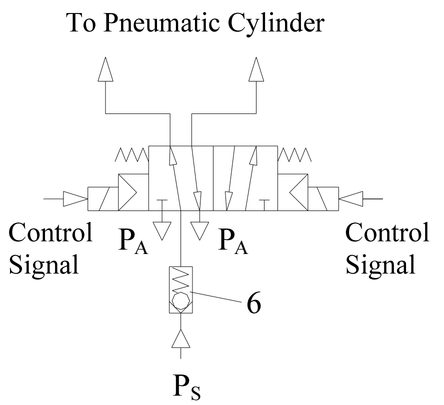
2 Structure of the Pneumatic Actuator with Constant Velocity Motion

In some practical applications such as printing, painting or spraying systems, meeting the stability requirement of the constant velocity motion is not hard. Typically, in these cases the maximum value of the ripple is in the range of 7 to 12 % of the desired velocity magnitude. For these applications, the open loop linear pneumatic actuator may be utilized, and its schematic diagram is shown in Fig. 1.



Actuator with 5/2-way solenoid valve



5/2-way solenoid valve with non return valve

Fig. 1: *Block diagram of the pneumatic actuator with reciprocating motion*

Its construction consists of the pneumatic cylinder (1), two one-way flow control valves (2 and 3), the 5/2-way solenoid control valve (4) and a position transducer (5) which measures the load displacement. The value of the actuator velocity is set by an adjustable throttle in the one-way flow control valve. The solenoid control valve (4) connects the pneumatic cylinder to the supply pressure and the exhaust port according to the control algorithm. The position transducer (5) measures the load displacement, and the control system determines the technological command and forms the control signal of the solenoid valve (4) for the implementation of the reverse motion according to the control algorithm. It is important to note that in this case the position signal is not used for the adjustment of the constant velocity.

Generally in such actuation systems rodless pneumatic cylinders are used as in this design, making the adjustment of the speed in both directions of motion identical and simple (it is very important for these actuators). Using a non-return valve (6) in the supply port (see Fig. 1,b) allows the achievement of energy recovery in the reverse motion process; in this case the kinetic energy of the moving mass is used for its acceleration in the opposite direction.

3 Parameters Estimation of the Actuator with Constant Velocity Motion

It is obvious in the open loop actuator with constant velocity motion that the graph of velocity versus time has the "trapezoidal" form. The dynamic behavior of this actuator is described in Eq. 1 (for actuator with schematic diagram shown in Fig. 1 (Krivits and Krejnin, 2006).

Where m is the mass of the load (moving mass), x is the position of the cylinder piston, b_v is the viscous friction coefficient, F_F is the friction force, F_L is the external force load, P_1 and P_2 are the absolute pressures in the actuator working chambers, P_A is the absolute atmospheric pressure, P_S is the absolute supply pressure, A_1 and A_2 are the effective areas of the actuator piston, V_{01} and V_{02} are the inactive volumes at the end of stroke and admission ports, L_S is the stroke of the pneumatic cylinder, A_V^+ is the effective area of the charging line, A_{V1}^- and A_{V2}^- are the effective areas of the discharge line of the first and second working chambers accordingly, $\varphi(*)$ is the flow function, β is the control coefficient ($\beta = 1$ if the actuator moves in the direction from left to right and $\beta = 0$ if the actuator has the opposite movement), and K_* is the constant coefficient, see Eq. 2.

$$\begin{cases} m \cdot \ddot{x} + \mathrm{b_V} \cdot \dot{x} + F_F + F_L = P_1 \cdot A_1 - P_2 \cdot A_2 \\ \dot{P}_1 = \dfrac{1}{V_{01} + A_1 \cdot x} \cdot [\beta \cdot A_V^+ \cdot P_S \cdot K_* \cdot \varphi(\dfrac{P_1}{P_S}) - (1-\beta) \cdot A_{V1}^- \cdot P_1 \cdot K_* \cdot \varphi(\dfrac{P_A}{P_1}) - P_1 \cdot A_1 \cdot \dot{x}] \\ \dot{P}_2 = \dfrac{1}{V_{02} + A_2 \cdot (L_S - x)} \cdot [(1-\beta) \cdot A_V^+ \cdot P_S \cdot K_* \cdot \varphi(\dfrac{P_2}{P_S}) - \beta \cdot A_{V2}^- \cdot P_2 \cdot K_* \cdot \varphi(\dfrac{P_A}{P_2}) + P_2 \cdot A_2 \cdot \dot{x}] \end{cases} \tag{1}$$

$$K_* = \sqrt{\frac{2 \cdot k \cdot R \cdot T_S}{k-1}} \approx 760 \frac{m}{s} \tag{2}$$

The mathematical model has been obtained by assuming that (Dupont and Dunlap, 1995):

- The hypothesis of an isothermal process is reasonable
- The gas is perfect
- Pressure and temperature within the actuator chambers are homogeneous
- The origin of piston displacement is at the left point of the stroke

In the steady state condition, when the actuator moves with constant velocity, the acceleration and change of pressure in the actuator working chambers are zero. In this case one of the working chambers is connected to the supply line and the other one is connected to the exhaust port. Taking into account these conditions Eq. 1 may be rewritten as (for rodless cylinder):

$$\begin{cases} (P_{10} - P_{20}) \cdot A_P = F_F + F_L + b_V \cdot \dot{x}_C \\ A_V^+ \cdot P_S \cdot K_* \cdot \varphi(\dfrac{P_{10}}{P_S}) = P_{10} \cdot A_P \cdot \dot{x}_C \\ A_V^- \cdot K_* \cdot \varphi(\dfrac{P_A}{P_{20}}) = A_P \cdot \dot{x}_C \end{cases} \tag{3}$$

where P_{10} and P_{20} are the absolute pressures in the actuator working chambers in steady state condition, $\dot{x}_C$ is the actuator constant velocity, and A_P is the effective area of the actuator piston. Since the value of $\dot{x}_C$ in forward and reverse motion is identical, the effective area of the discharge line is:

$$A_{V1}^- = A_{V2}^- = A_V^- \tag{4}$$

In general, in such a pneumatic system there may be some combinations of the upstream and downstream conditions, however when the upstream flow moves at the subsonic condition and the downstream flow moves at the sonic condition, the optimal system characteristics are reached (Ivlev et al., 1985). In this case Eq. 3 has the following form:

$$\begin{cases} (P_{10} - P_{20}) \cdot A_P = F_F + F_L + b_V \cdot \dot{x}_C \\ 2 \cdot A_V^+ \cdot P_S \cdot K_* \cdot \varphi_* \cdot \sqrt{\dfrac{P_{10}}{P_S} \cdot (1 - \dfrac{P_{10}}{P_S})} = P_{10} \cdot A_P \cdot \dot{x}_C \\ A_V^- \cdot K_* \cdot \varphi_* = A_P \cdot \dot{x}_C \end{cases} \tag{5}$$

where $\varphi_* \approx 0.259$.

The value of the constant velocity $\dot{x}_C$ may be obtained by using the third line of Eq. 5:

$$\dot{x}_C = \frac{A_V^- \cdot K_* \cdot \varphi_*}{A_P}. \tag{6}$$

It is important to note that for this combination of upstream and downstream conditions, the value of the constant velocity $\dot{x}_C$ depends only on the value of the effective areas of the discharge line A_V^- and the piston A_P. In this case, variation of the internal and external load force, moving mass and supply pressure have no affect on the actuator velocity in the steady state condition. The actuator has a high robustness and in addition, it is easier to adjust its steady state velocity. This invariance is the significant advantage in the application of the open loop pneumatic actuator for reciprocating motion.

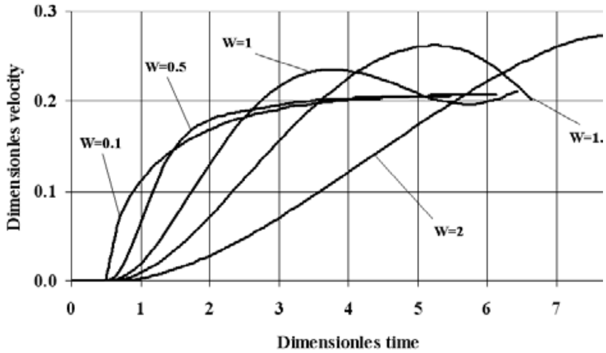


Fig. 2: *Influence of the dimensionless inertial load on the transient response of the actuator with constant velocity motion*

In the process of the actuator parameter estimation, the nature of the transient response and its duration are very significant. Usually actuators operating in the field meet the conditions:

$$\begin{aligned} &0 < \Omega = \frac{A_V^-}{A_V^+} < 1 \\ &P_S \geq 0.4\, MPa \\ &0 < \chi = \frac{F_F + F_L}{P_S \cdot A_P} \leq 0.3 \end{aligned} \tag{7}$$

In these devices, the dimensionless inertial load:

$$W = \frac{A_V^+ \cdot K_*}{A_P} \cdot \sqrt{\frac{m}{P_S \cdot A_P \cdot L_S}} \tag{8}$$

has a strong effect on the nature of the transient response. Figure 2 illustrates the influence of the parameter W on the nature of the transient response where the dimensionless time is:

$$\tau = \frac{A_V^+ \cdot K_*}{A_P \cdot L_S} \cdot t \tag{9}$$

and the dimensionless velocity (Krivits and Krejnin, 2006) is:

$$\dot{\xi} = \frac{A_P}{A_V^+ \cdot K_*} \cdot \dot{x} \tag{10}$$

From this point of view, the range from 0.1 to 0.5 for W is recommended. As seen in this figure for $W \geq 1$, the transient response has a long duration and oscillations, which are unacceptable for proper operation.

The dimensionless valve effective area ratio is:

$$\Omega = \frac{A_V^-}{A_V^+} \tag{11}$$

This parameter has a weak influence on the nature of the transient response, however it has a strong effect on the response time and the value of the steady state velocity (see Fig. 3). For practical applications it is recommended that the valve effective area ratio lie between 0.6 and 1.

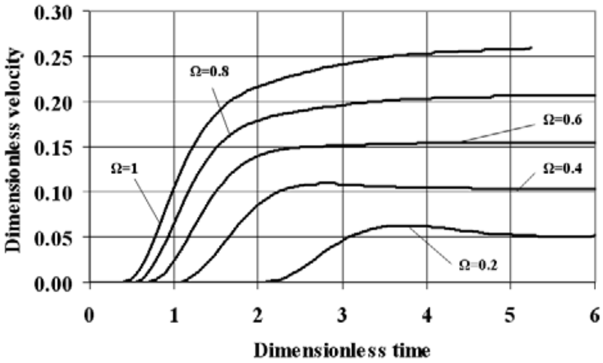


Fig. 3: *Influence of the dimensionless valve effective area ratio on the transient response of the actuator with constant velocity motion*

The magnitude of the dimensionless load χ does not influence the nature of the transient response (see Fig. 4). Increasing this parameter increases only the transient response time, as shown below. According to Fig. 4 the recommended values of χ that are referred to above ($0 \le \chi \le 0.3$) are acceptable.

Another important matter in the design of the open loop actuator with constant velocity is the estimation of its stroke for the acceleration and deceleration parts of motion. In practice, the most essential is the stroke where the actuator moves with constant velocity; however, the total actuator stroke should include the acceleration and deceleration displacements. For the acceleration part, when $0.6 < \Omega < 1$ and $0.1 \le W \le 0.5$ it may be assumed that the equivalent uniform acceleration is:

$$\ddot{x}_S = \frac{\mathrm{B}_{AC} \cdot P_S \cdot A_P}{m} \tag{12}$$

where the coefficient B_{AC} may be defined as $\mathrm{B}_{AC} \approx 0.015$ (this estimation was obtained by analyzing experimental and computer simulation data). Then the displacement on the acceleration part is

$$L_A = \frac{m \cdot \dot{x}_C^2}{2 \cdot \mathrm{B}_{AC} \cdot P_S \cdot A_P} \tag{13}$$

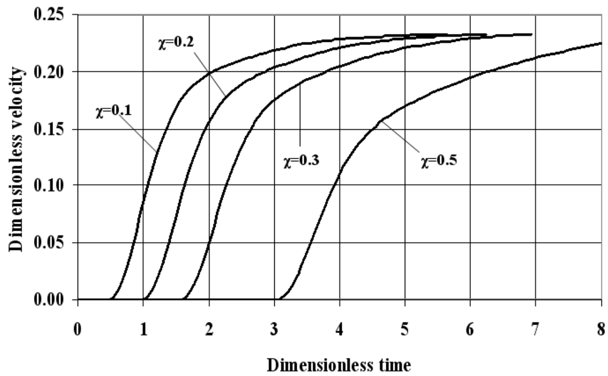


Fig. 4: *Influence of the dimensionless load on the transient response of the actuator with constant velocity motion*

The value of the deceleration ($\ddot{x}_D$) is usually given, and the displacement of this part (L_D) may be estimated by the following equation:

$$L_D = \frac{\dot{x}_S^2}{2 \cdot \ddot{x}_D} \tag{14}$$

Then the total stroke of the actuator is

$$L_S = L_A + L_C + L_D \tag{15}$$

where L_C is the actuator displacement when it moves with constant velocity (usually, it is given).

The key parameters of the actuator, which are the piston effective area ($A_!$) and the effective areas of the control valve (A_V^+ and A_V^-), should be estimated by the following sequence:

Using the given value of the actuator constant velocity and the third line of Eq. 5 the ratio B_A^- is defined:

$$B_A^- = \frac{A_V^-}{A_P} = \frac{\dot{x}_C}{\varphi_* \cdot K_*} \tag{16}$$

For the required parameters Ω ($0.6 < \Omega < 1$) and W ($0.1 \le W \le 0.5$), the ratio B_A^+:

$$B_A^+ = \frac{A_V^+}{A_P} = \frac{B_A^+}{\Omega} \tag{17}$$

and the effective area of the piston:

$$A_P = \frac{(B_A^+)^2 \cdot K_*^2 \cdot m}{2 \cdot W^2 \cdot P_S \cdot L_S} \tag{18}$$

are defined, where the total actuator stroke (L_S) should be taken into consideration in the first estimation as:

$$L_{SE} = L_C + L_D; \left(L_D \approx \frac{\dot{x}_S^2}{2 \cdot \ddot{x}_D} \right)) \tag{19}$$

Using Eq. 13 and 19 the displacement of the acceleration part of the actuator motion (L_A) and the actuator total stroke (L_S) are defined.

Using Eq. 18 the adjusted value of the piston effective area (A_P) is determined and the effective areas of the control valve are defined:

$$A_V^- = B_A^- \cdot A_P \text{ and } A_V^+ = \frac{A_V^-}{\Omega} \tag{20}$$

4 Printing Machine with Open Loop Pneumatic Actuator

The operation of some types of printing machines is based on the reciprocating motion of the print head module, which realizes the printing process during the period when the print head moves with constant velocity. Usually all functions of printing and actuator reversing processes are controlled by a PLC according to the signal of the displacement sensor.

For such a printing machine, the open loop pneumatic actuator allows reaching an inexpensive and robust design. For instance, consider the parameter estimation and dynamic analysis of the pneumatic open loop actuator for printing equipment with the following characteristics:

Table 1: Pneumatic open loop actuator parameter values

Parameter	Symbol	Value	Units
moving mass	m	30	kg
supply pressure	P_S	0.6	MPa
external force	F_L	0	N
constant velocity	$\dot{x}_C$	0.3	m/s
constant velocity displacement	L_C	0.6	m
max deceleration	$\ddot{x}_D$	10	m/s^2

For a first approximation, the following assumptions may be taken into account.

Table 2: Pneumatic open loop actuator assumed parameter values

Parameter	Symbol	Value	Units
actuator friction	F_F	50	N
viscous friction coefficient (Krivits and Krejnin, 2006)	b_V	50	Ns/m
dimensionless inertial load	W	0.25	
dimensionless valve effective area ratio	Ω	1	
area ratio (Eq. 16)	B_A^-	≈ 1.5e^{-3}	
area ratio (Eq. 17)	B_A^+	1.5e^{-3}	
estimated total actuator stroke (Eq. 19)	L_{SE}	≈ 0.61	m
Estimated effective piston area (Eq. 18)	A_{PE}	≈ 0.8e^{-3}	m^2
acceleration displacement (Eq. 13)	L_A	≈ 0.3	m
effective control valve areas	$A_V^+ = A_V^- = B_A^- \cdot A_P$	≈ 1.2e^{-6}	m^2

Therefore, referring to Eq. 15 and 19, the final total actuator stroke should be approximately 1 m.

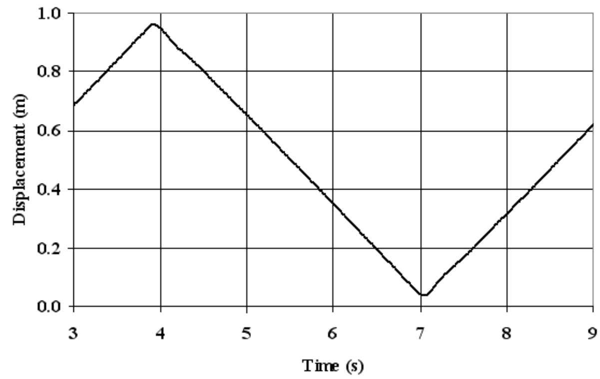
According to this estimation of the actuator parameters shown in Table 2, a rodless pneumatic cylinder with piston diameter of 40 mm (effective area $A_P = 1.256 \cdot 10^{-3}$ m^2) and with stroke of $L_S = 1$ m may be used. Such an actuator is a double acting rodless pneumatic cylinder with adjustable end position cushioning at both ends, with a mechanically coupled connection between the piston and the slide. In this case, the pneumatic cylinder type DGP ("Festo") or ORC ("Koganei") may be used as an actuator.

One 5/2-way double solenoid valve type JMYH-5/2-M5-L-LED ("Festo") may be used as a control element. Its effective areas are $A_V^+ = A_V^- = 3.2 \cdot 10^{-6}\,\text{m}^2$ (standard nominal flow rate is 190 l/min) and the response time is about 0.01 s.

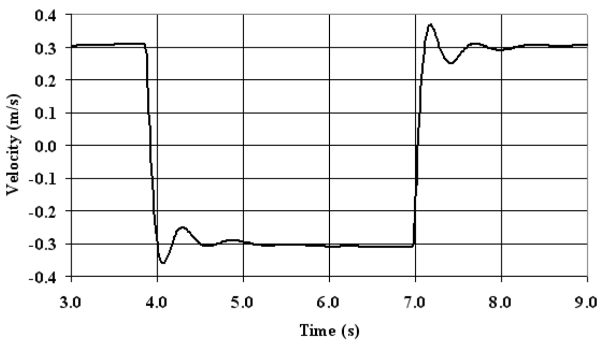
Two swivel flow control valves of type GRLA-M5-QS-4 ("Festo") are used to regulate the exhaust air flow in a double acting cylinder that produces a change in the piston speed. In the free flow direction the effective area is about $A_V^+ = 2.5 \cdot 10^{-6}\,\text{m}^2$ (standard nominal flow rate is 150 l/min) and in the throttle direction the effective area (A_V^-) changes in the range from 0 to $2.4 \cdot 10^{-6}$ m^2 (standard nominal flow rate is from 0 to 140 l/min).

A position transducer with resolution of 0.02 mm may be used as a displacement sensor (for instance, linear encoder type RGH by "Renishaw").

The results of computer simulation of the actuator dynamics demonstrate the effectiveness of the proposed solution. In this case, the fourth rank Runge-Kutte stability criterion with relative error of integration of 0.1 % is used.



a. Position response.



b. Velocity response.

Fig. 5: *Transient response of the open loop actuator with reciprocation motion*

Figure 5 shows the actuator response to the constant velocity motion command, where the steady state velocity value is 0.3 m/s. It can be seen that the position response (see Fig. 5a) has the "triangular" monotonic form (without overshoot) and the velocity (see Fig. 5b) is changed by the "trapezoidal" law with overshoot. However, on the cylinder stroke segment between 0.2 m and 0.8 m (see Fig. 6) the changing of the velocity is in the range between 0.29 m/s and 0.31 m/s reaching about ± 3.3 % of the desired steady state value, which is very acceptable. It is important to note that the motion in this condition has the monotonic form without fluctuations.

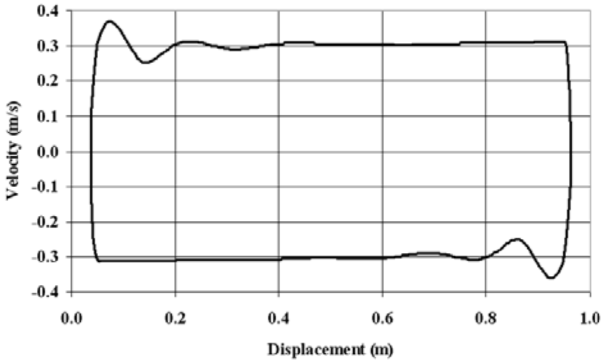


Fig. 6: *Phase-plane trajectory of the open loop actuator with reciprocation motion*

In Fig. 7 the transient responses of the reversing process from 0.3 m/s to -0.3 m/s are shown. These curves are carried out for different values of the friction force. It can be seen that the magnitude of the steady state velocity is invariant in this condition; variation of the friction force affects only on the form the transient response and its time.

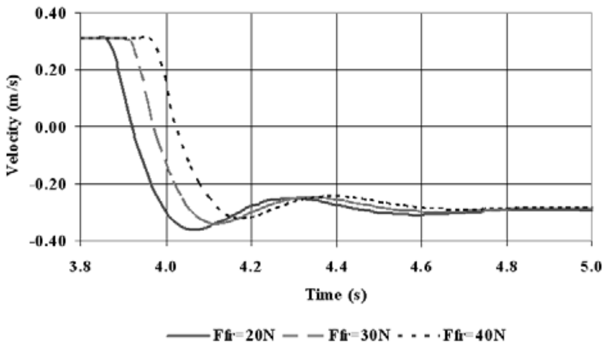


Fig. 7: *Velocity transient responses for different friction forces*

Conclusion

Computer simulation of the dynamic behavior of the pneumatic actuator which has reciprocating motion with constant velocity mode shows the high effectiveness of the proposed solution in equipment, where the deviation of the constant velocity motion should be up to 4 to 5 % of the steady state desired value. The proposed actuator has an open loop control system and uses the standard pneumatic cylinder. This design is not only less costly, but more durable in comparison with closed loop design.

Nomenclature

A	effective area of the actuator piston	[m^2]
A_V	effective area of the control valve	[m^2]
b_V	viscous friction coefficient for linear motion	[Ns/m]
F_F	friction force	[N]
F_L	external force load	[N]
K_*	constant coefficient	[m/s]
k	adiabatic exponent	
L_A	displacement of the actuator acceleration	[m]
L_C	actuator displacement at constant velocity	[m]
L_D	displacement of the actuator deceleration	[m]
L_S	stroke of the pneumatic cylinder	[m]
m	mass of the load (moving mass)	[kg]
P	absolute pressure in the actuator working chamber	[Pa]
P_S	absolute supply pressure	[Pa]
R	gas constant	[J/kgK]
t	time	[s]
T_S	temperature of the air	[K]
W	dimensionless inertial load	
x	position of the cylinder piston	[m]
$\dot{x}_C$	constant velocity of the steady state motion	[m/s]
β	opening coefficient of the control valve	
ξ	actuator dimensionless displacement	
τ	dimensionless time	
$\varphi(*)$	flow function	
φ_*	flow function saturation	
χ	dimensionless force load	
Ω	effective area ratio of the control valve	

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