Impact of Capturing Used Air on the Dynamics of Actuator Drive

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Abstract: This paper discusses the process of capturing the used air from the pneumatic actuator in a separate reservoir for its later use in order to get a higher energy efficiency of the pneumatic system.
A mathematical model is developed and MATLAB simulations is performed. Certain conditions for collecting of air pressure under which the capturing can be used, without the loss of actuators dynamic characteristics, were determined by modeling and experimentally. A good correlation of results of modeling and the results obtained experimentally are shown.

Keywords: Capturing used air, Energy efficiency, Pneumatic system, Limit pressure ratio, Time constant

1. INTRODUCTION

Energy efficiency has long been neglected in the field of application of pneumatic systems. In a usual working cycle of the pneumatic components compressed air is released into the atmosphere after the use. The release of exhausted, but still compressed, air in the atmosphere is a waste of energy. Energy efficiency of pneumatic systems is around 20%, which is low compared with about 80% energy efficiency of electrical systems and 40% energy efficiency of hydraulic systems (Belforte, 2000; Luo et al., 2011).

The main reason for low energy efficiency of pneumatic systems is the open-loop circuit of the pressurized air circulation in the pneumatic systems. Because the air is released in the atmosphere instead of circulation in the pneumatic system it waste energy and pollute the atmosphere with an exhausted airborne contaminants.

More recently, great interest arouses for this issue, because of worldwide trend of energy saving. Saving energy and improving the energy efficiency of pneumatic systems can be achieved in a number of ways (Seslija et al., 2001) and one of them is reuse of exhausted air. A large number of papers have been published on the subject of re-use of compressed air (Blagojevic et al., 2011; Al-Dakkan et al., 2003; Wang et al., 1999; Blagojevic et al., 2013).

Seslija D. et al. (2001) identifies the possibilities for increasing energy efficiency in compressed air system and suggests the measures that can improve the energy efficiency. (Blagojevic et al., 2011, 2013) presented way for reducing air consumption in execution part of pneumatic system by restoring energy based on applying by-pass valve between a cylinder chambers.

One way to save energy in these systems is the use of two levels of pressure. It is possible in a cases where the actuator performs useful work in only one direction and a return stroke is at no-load, so a lower pressure can be used. Another possibility for re-using of the air released from the actuator is by collecting the exhausted air in the reservoir and subsequently reusing it in a pneumatic circuits.

In recent years, several papers appeared which consider the capturing of discharged air, after finishing their work in the pneumatic cylinder, and storing the compressed air in an additional tank for later use (Shi et al., 2005; Li et al., 2005; Siminiati, 2010).

It is possible to use a pressure booster valve, to amplify the pressure sufficiently for the needs of other pneumatic devices which need higher pressure. Boosting is achieved at the expense of reducing the flow so that it takes into account the balance between consumption and the amount of air collected in the tank.

(Shi et al., 2005) presents one exhausted-air reclaiming system for pneumatic cylinders. An air tank is introduced to receive the air from discharging chamber. In order to reclaim more energy and make less influence on the cylinder velocity it is necessary to control precisely the connecting of discharging chamber with receiver to avoid stick-slip motion. The time moment when the connecting stops is called switch point. It is showed that the switch point has to be properly controlled to avoid bad influence on the velocity stability only, but the piston motion time is increased and piston velocity is reduced.

(Luo et al., 2011) proposed pneumatic structure with intermediate air tank for buffering the down-stream negative effects to the up-stream. It means to avoid negative effects on dynamics of the cylinder. The impact of the intermediate air tank on the dynamics was not studied in this article.

(Li et al., 2005) proposed three ways for recover and subsequent use of the captured air. First one is to sand the recovered air back to air source main reservoir, second is to send the recovered air back to recover accumulator and third way is directly recover the air from the exhausted hole without a control valve and than use boost valve to send the air back to recover accumulator. All three ways include air charge accumulator to collect exhausted air before boosting pressure for future use.

(Siminiati, 2010) suggests that the compressed air, collected in additional tank, can be used to form a closed pneumatic circuit. The compressor sucks air from the tank instead of the atmosphere. The experimental results are not presented. In all these above mentioned cases it is very important to avoid to reduce the dynamic characteristics of cylinder movement and to save the work ability (or the exergy) of captured compressed air at the points of used pneumatic actuators (Eret et al., 2010).

This paper treats the problem of collecting used air in separate reservoir but in this research the problem of dynamic behavior of the actuator, during the capturing process, is investigated too. This article presents the results of the research of the influence of capturing used air on the dynamics of the actuators which released the output air in additional reservoir instead in the atmosphere.

The new approach of control strategy of capturing air, which improve energy efficiency and save system dynamics during capturing air, is proposed. The approach is different from the previously described methods and it includes an estimation of limit pressure of capturing air for appropriate control procedure for the pneumatic system.

This article considres only collecting discharged compressed air to be reused in different possible solutions (Li et al., 2005), but without a loss of dynamic characteristics of pneumatic actuator from which the air is collected.

2. MATHEMATICAL MODEL

Analysis of the influence of the capturing used air on the dynamics of the actuator requires a mathematical description of pneumatic system used to collect pressurized air. It is typical pneumatic system with actuator driving an inertial load as it is shown in Fig 7. The system consists of pneumatic valve, working cylinder, inertial load and additional reservoir. The mathematical model consists of two parts: the pressurized air expiration model and mechanical model piston's movement with a load. In the first part is included additional reservoir according to pneumatic scheme in Fig.8. The system described in (Cajetinac, 2012) is supplemented here with mathematical description of the pressure change in additional reservoir.

Air flow through the valve opening A_v from the area of higher pressure to a lower pressure area can be done with sonic or subsonic velocity depending on the ratio of these two pressures.

If the ratio is less than or equal to the critical pressure ratio value C_r, expiration air velocity is equal to the sonic velocity. Mass flow rate in this case depends on the upstream pressure p_g, discharge valve coefficient C_f, area of valve opening A_v and air temperature at the upstream pressure. If the ratio is greater than the critical limit ratio, the air speed is subsonic velocity and mass flow rate also depends on the downstream pressure p_d.

Equations that express it are as follows (Richer and Hurmuzlu, 2000; Shen et al., 2006):

$$\dot{m}(p_u, p_d) = \sqrt{\frac{k}{RT}\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}} C_f p_u A_v, \quad \frac{p_d}{p_u} \le C_r$$

and

$$\dot{m}(p_u, p_d) = \sqrt{\frac{2k}{RT(k-1)}} \sqrt{1-\left(\frac{p_d}{p_u}\right)^{\frac{k-1}{k}}} \cdot \left(\frac{p_d}{p_u}\right)^{\frac{1}{k}} C_f p_u A_v, \quad \frac{p_d}{p_u} > C_r \tag{1}$$

The meaning of the variables in equation (1) are:

C_f – discharge valve coefficient,

p_u , p_d – upstream and downstream pressure,

A_v – air discharge effective area of the valve opening,

C_r – critical pressure ratio witch divide the flow regimes,

k – ratio of specific heats at constant pressure and constant volume respectively.

Critical pressure ratio for air is

$$C_r = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \approx 0{,}528 \tag{2}$$

Calculation of the pressure in the cylinder chamber starts from the an ideal gas state equation

$$pV = mRT \tag{3}$$

where R is the universal gas constant, T is the absolute temperature, m is the mass of gas and V is the volume of the chamber in which the gas is located. Solving for the pressure gives:

$$p = \frac{mRT}{V} \tag{4}$$

The time derivative of the left and right sides of the equation (4), if the process is isothermal, gives:

$$\dot{p} = RT\,\frac{d}{dt}\left(\frac{m}{V}\right) = RT\,\frac{\dot{m}}{V} - p\,\frac{\dot{V}}{V} \tag{5}$$

The equation (5) shows that the change in pressure in the cylinder chamber depends on the mass flow rate, the volume of chamber, rate of change of the volume and the piston displacement x, consequently. By integrating equation (5) we get the pressure value p_a and p_b in cylinder chambers a and b respectively, which is used to calculate the piston movement.

Dynamic behavior is defined by the equation balance of forces on the piston:

$$M\ddot{x} + B\dot{x} = p_a A_a - p_b A_b \tag{6}$$

where are:

M – mass of moving parts,

B – coefficient of viscous friction,

p_a –pressure of air in the chamber a ,

A_a – surface of the piston in the chamber a,

p_b –pressure of air in chamber b,

A_b – surface of the piston in the chamber b.

By calculating the time derivative of left and the right side of equation (6) is obtained:

$$M\dddot{x} + B\ddot{x} = \dot{p}_a A_a - \dot{p}_b A_b \tag{7}$$

Solving for $\dddot{x}$ gives:

$$\dddot{x} = \frac{A_a}{M}\dot{p}_a - \frac{A_b}{M}\dot{p}_b - \frac{B}{M}\ddot{x} \tag{8}$$

where the pressure time derivative $\dot{p}_a$ and $\dot{p}_b$ are calculated using (5). The experiment was performed with the pass-through piston rod, which means that the surfaces of the piston on both sides are equal ($A_a = A_b$).

3. SIMULATION

Computer simulation is used to assess the characteristics of the system and assess the impact of changing the parameters on the behavior and properties of the real system. Here was used partly modified simulation model described in (Cajetinac, 2012). The modification consists of adding another block that calculates the pressure change in a separate reservoir which is connected to discharge cylinder chamber and collects the used air.

The pressure in the additional reservoir is taken to be the downstream pressure p_d in equation (1). For the calculation of the pressure in additional reservoir equation (5) was used. Because the volume is constant, the equation takes the form:

$$\dot{p}_r = RT\frac{\dot{m}}{V_r} \tag{9}$$

where V_r is volume of additional reservoir.

Based on equations (1) to (9), simulations of the pneumatic system in SIMULINK are performed.

Simplified block diagram of the simulation, which was performed, is given in Fig.1.

For each cylinder's chamber mass flow described by (1), (2) is simulated and pressure described by (3) to (5). Mechanical piston system is simulated by (8) and the pressure change in the additional tank by (9). Series of rectangular pulses simulate command from the PLC and the operation of pneumatic valves. The pulses are used to control the switch blocks that connect different values of the upper and lower pressure to the blocks for mass flow. Delay in operation of pneumatic valves was ignored in the simulation, because high speed valves with 2ms response time is used. As the simulation outputs are observed displacement and piston speed, pressure changes in both chambers, the pressure in the additional reservoir and the differential pressure.

Table 1 contains the values of the physical quantities that are used in the simulation.

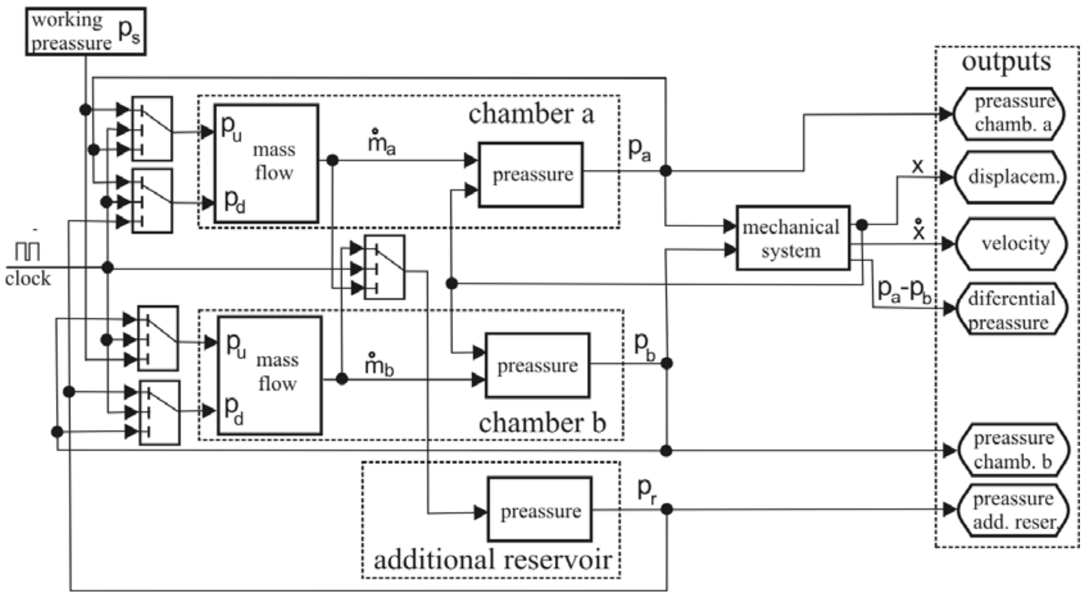

Fig. 1. Block diagram of the simulation.

Table 1. The values used for simulation

Sym.	Variable	Value	Unit
R	Universal gas constant	287	J/ kgK
T	Temperature	293	K
C_r	Critical pressure ratio	0.528	dimensionless
A_v	Surface expiration	$2.5 \cdot 10^{-6}$	m^2
A	Piston Area	$8 \cdot 10^{-4}$	m^2
M	Mass of moving parts	0.790	kg
P_s	Working pressure	$5 \cdot 10^5$	Pa
V_r	Additional reservoir volume	$2 \cdot 10^{-3}$	m^3
B	Coefficient of viscous friction	20	kg s^{-1}
C_f	Coefficient expiration	0.65	dimensionless
k	Ratio of spec. heat air at const. pressure and volume	1.4	dimensionless

Simulation of the piston movement when the air is released to the atmosphere is shown in Fig. 2. The piston continuously makes the cycles.

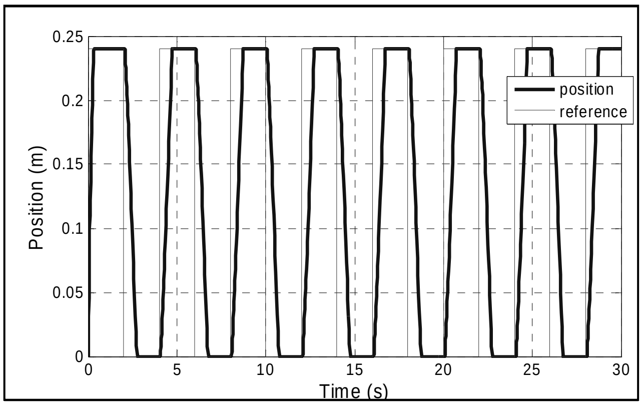

Fig. 2. Reference signal and position of piston without capturing.

Simulation of the piston movement, when the air is released from the cylinder into an additional reservoir, gives the results shown in Fig. 3. It can be seen that the piston makes four cycles (eight shifts) and it stops.

Simulation of the pressures in the chambers of the cylinder is shown in Fig.4. Each chamber is fed alternately to the working pressure and the reached pressure in the additional reservoir.

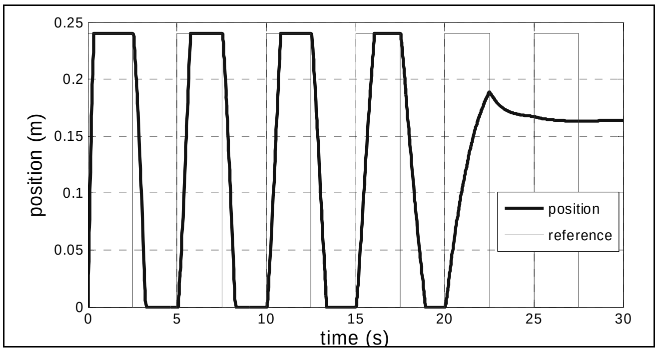

Fig. 3. Reference signal and position of piston (simulation).

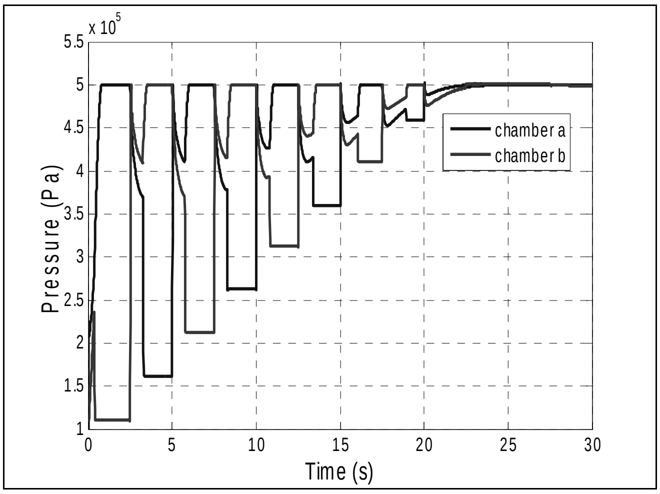

Fig.4. Simulation of pressure in the chambers of the cylinder.

The piston stops due to equalization of the reservoir pressure and supply pressure in the system, so that the differential pressure becomes zero (Fig. 5).

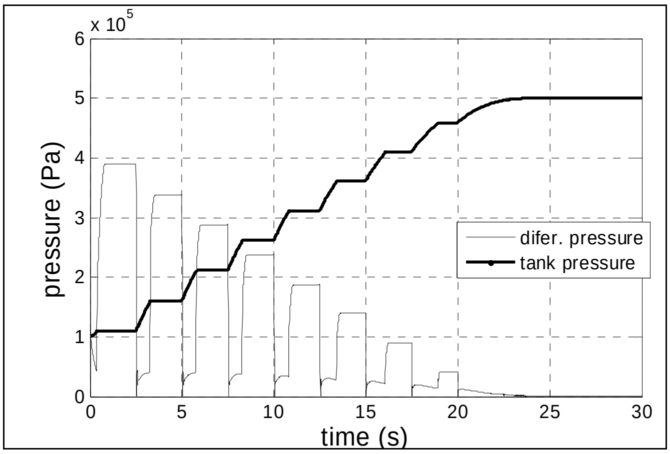

Fig. 5. Additional tank pressure and differential pressure.

The simulation speed of the piston is shown in Fig. 6. It can be seen that the maximum speed is reduced in each successive cycle so that the piston stops after 24 s.

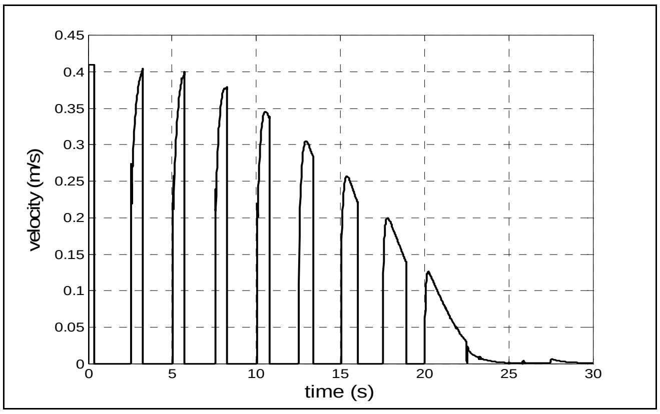

Fig. 6. Velocity of piston.

It is noticeable that the maximum speed in the first eight seconds remained approximately the same (value about 0.4 m/s). This corresponds to the value of the pressure in the tank of 2.6 bar, as can be seen in Fig. 5.

4. EXPERIMENTAL SYSTEM

In order to investigate the process of capturing exhausted air from the pneumatic actuator, that previously was simulated, the experimental pneumatic installation with additional tank is formed. The photo of experimental settings is shown in Fig. 7.

Pneumatic installation shown in Fig. 7 consists of :

- cylinder FESTO DNN 32-350PPV-A (1),
- potentiometer for measuring the piston position (2),
- two valves FESTO MHE2 (3),
- two non return valves (4),
- additional tank (5),
- pressure transducer (6),
- reservoir’s pressure regulating valve (7),
- compressor unit (8),
- programable logic controler OMRON PLC CP1 (9),
- data acquisition module CP1W-MAD11 (10).

The installation is supplied by compressor power unit (8). Data acquisition and control is performed using a programmable controller OMRON PLC CP1 (9) and module for data acquisition CP1W-MAD11 (10) (Cajetinac et al, 2012). The output air from the cylinder is collected in the tank (5) instead of releasing it to the atmosphere. During each piston’s movement the air from one of the chambers is completely discharged into additional tank (5) through non return valves (4). The pressure in the tank increases with each movement. The pneumatic scheme of the installation is shown in Fig. 8.

The process of filling the additional tank with the exhausted air from the cylinder and impact on dynamic properties of the same cylinder is investigated.

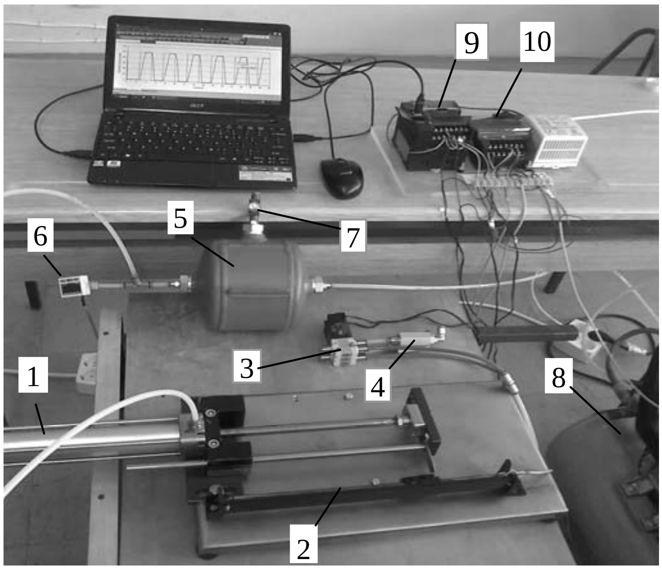

Fig. 7. Photo of experimental installation.

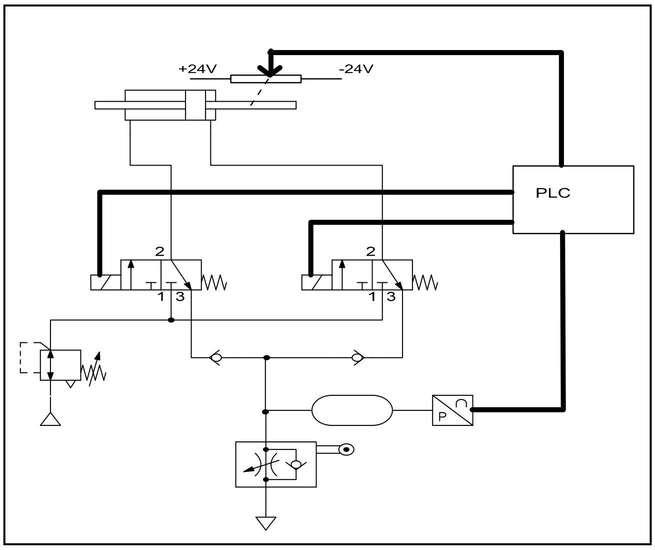

Fig. **8.** Pneumatic, measuremnent and control scheme of experimental installation.

Firstly, in several experiments the exhausted output air is released to atmosphere or to the auxiliary reservoir in order to compare dynamic properties of the cylinder in these two conditions.

Two types of experiments with additional reservoir were performed:

a) The air is firstly discharged from the actuator to the additional tank, alternatively from both chambers and the pressure in the tank increases gradually until the actuator stops;

b) In the next group of experiments the pressure in the additional tank is previously fixed to constant initial value and than the actuator is activated by step input signal. This procedure is repeated for several different values of initial reservoir pressure to investigate the possible change of dynamic characteristics of the actuator during the process of filling the reservoir and the consequential increase tank’s pressure. The initial value of the pressure in the additional tank is set by filling from external pressure supply trough the check-choke valve (Fig 8) before each experiment and the tank is vented after the experiment by the valve.

All the process is govern with PC and programmable logic controller.

5. EXPERIMENTAL RESULTS

Several experiments were performed in order to verify the simulation results. Firstly, movement of the piston was measured when the air is released in the atmosphere. The piston continuously cyclical moves full stroke (Fig. 9).

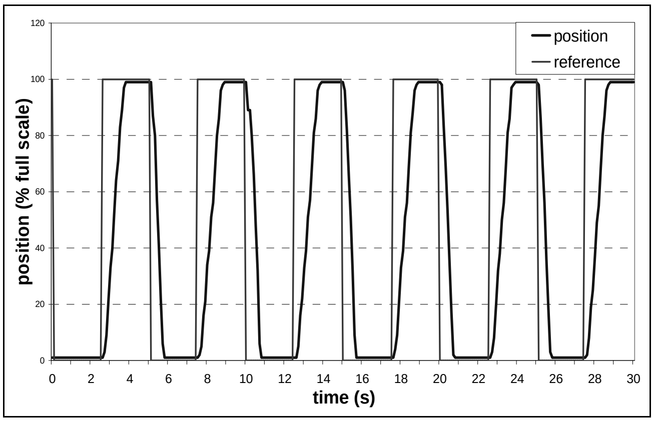

Fig. 9. Reference and piston position without capturing air.

Piston speed during the cyclic motion is shown in Figure 10. The speed is not directly measured by separate speed transducer, then the existing signal from the position sensor is used to calculate the current value of piston speed.

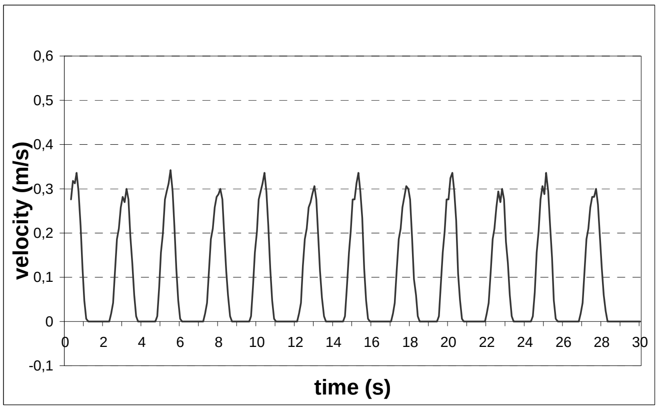

Fig. 10. Velocity of piston without capturing air.

It is evident that the maximum speed is different when the piston moves in one direction (about 0.34m / s) and the other (about 0.3 m / s) due to the asymmetry of friction.

With increasing number of cycles the piston speed remains unchanged as is shown on Fig. 10.

In the next experiment, the air is discharged into the tank trough the check valve (4) Fig.7. Due to the filling of the reservoir pressure in it increases.

The movement of the piston when the pressure in separate reservoir opposes the discharge air from the cylinder was checked with measurement piston position and speed. In this experiment, a separate reservoir was with the volume of 2 liters. The pressure is constantly increasing for approximately 0.5 bar during the movement of the piston in one direction.

Fig. 11 shows the results of measuring the piston displacement. After 7 to 8 displacements (or 3.5 to 4 cycles), the piston is stopped due to increased pressure in the auxiliary reservoir. This is in good agreement with the simulation results in section III, except that, in the simulation, the piston stops completely but in the real experiment piston continues with small oscillations during two next cycles.

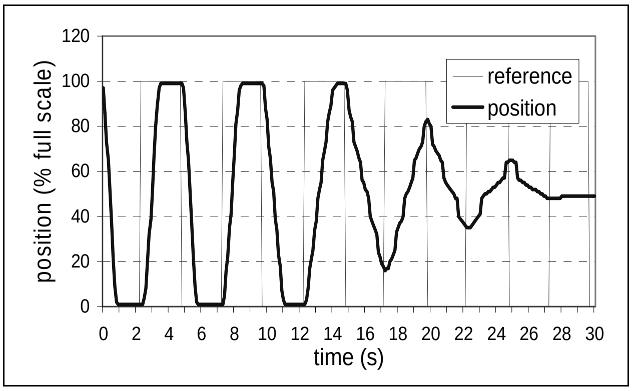

Fig. 11. Reference signal and position of piston durig capturing (experiment).

After every piston movement the pressure of the captured air in the reservoir would be increased and piston velocity would be decreased gradually. The variations of piston speed and pressure in the additional reservoir are shown in Fig. 12. The stick-slip motion, when the piston velocity is less then 0.07m/s, is observed.

Maximum piston speed obtained by measuring ranges from 0.3 m/s to 0.34 m/s in the same way as during the air discharge into the atmosphere. But detailed analysis of the experimental results shows that the maximum speed of the piston does not depend on the pressure in the auxiliary reservoir until the pressure does not exceed the value of approximately 0.50 of the working pressure value.

Further more detailed analyzing of capturing process and pressure level in additional reservoir influence on piston velocity is investigated, in the new experiments performed with testing piston movement on different pressure level in the reservoir connected to the discharging chamber.

In the second group of experiments, rather than cyclical, moving the piston is done only in one direction and with the prior selected preset pressure values in the additional reservoir. This value is preset by using control pressure valve (7) in Fig. 7, to simulate each one particular pressure in the reservoir as initial condition for capturing process start.

The procedure enables to estimate the critical value tank pressure when have to stop air capturing because the process capturing starts to decrease piston velocity.

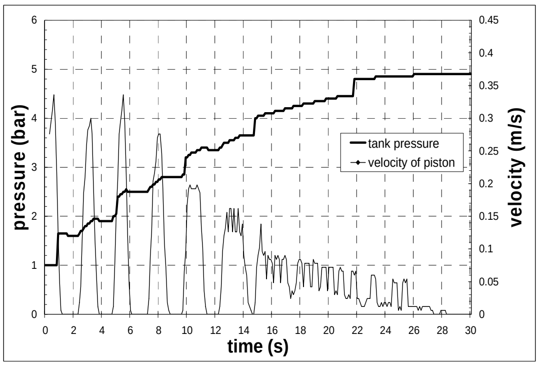

Fig. 12. Velocity of piston and tank pressure.

The cylinder is then launched with a step signal excitation, 0 to 100% of full displacement of the piston. The piston displacement, as the resulting response on this step input, is measured for different values of the initial pressure in additional reservoir. The resulting variations of the step responses are shown in Fig. 13.

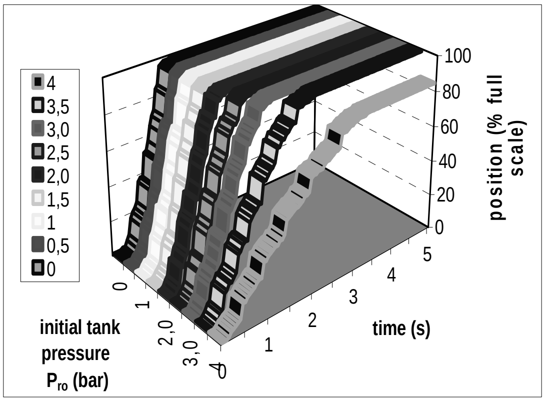

Fig. 13. Position of piston versus initial reservoir pressure.

The diagrams show the displacement of the piston for the different size of the initial pressure P_{r0} in a additional reservoir. The initial pressure P_{r0} changed from 0 to 4 bar with step increment 0.5 bar.

These experiments show that considerable changes in the dynamic characteristics of the cylinder appears only if the initial value of the pressure in the tank has a value of over half value of working pressure approximately. This is in good agreement with results obtained in first experiments with cyclic piston motion.

These experiments were conducted in the installation on Fig. 7 with different supply pressures (4, 5, 6 bar) to determine the threshold value of the limit pressure in the tank to which it is possible to collect the air without lowering dynamic characteristics of the cylinder.

The parameter known as the *time constant* - τ_c is used for the estimation the influence of the pressure in the additional reservoir on the performance of the actuator witch releases the air from the discharge chamber to the reservoir instead in the atmosphere.

The pressure in the added reservoir, during piston movement, continuously increases relative to the preset initial value. Consequently, operating conditions for the capturing are varied during the piston movement. But a time constant - τ_c can be used to estimate the effect of the specific preset initial pressure on the speed of the cylinder. Time constant is convenient for the estimation because τ_c is the measure of the step response's line slope at t = 0, when step input signal starts and chosen initial pressure is applied (Beater, 2007). This parameter is equal to the time that a system of first order (Bolton, 1996) takes to obtain the level of $1 - e^{-1} \approx 0.632$ of its asymptotic value (Bolton, 1996; Vyhlidal et al., 2001) caused by the step input. Time constant τ_c of the tested cylinder is measured and calculated for different values of initial pressure fixed in the additional reservoir and different values of supply pressure in the system.

Diagrams of the time constant τ_c for different values of the initial pressure and different values supply working pressure (4, 5, 6 bar) are given in Fig. 14.

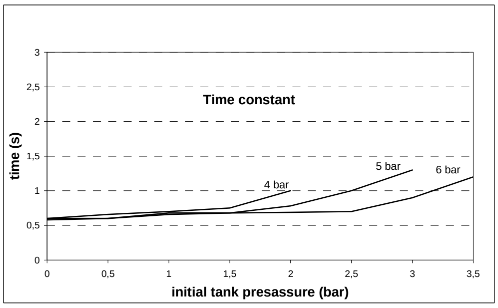

Fig. 14. Time constant value versus initial reservoir pressure.

It can be seen that τ_c is approximately constant when the initial value of the pressure in the tank of the used air is in the range from 0 to the half value of working pressure where it appears the knee on the time constant line. This means that this range of pressure change does not affect the speed of piston. Substantial increase in the value of τ_c and decrease of piston speed is observed with initial pressure greater than this limit pressure.

These results of the research suggests one new practical method to determine the limit pressure in additional reservoir of used air when have to stop capturing air from the cylinder. It is the pressure value p_{r0} when the system time constant rapidly increases. The value of the time constant is monotonically increasing function of the pressure in additional reservoir (Fig. 14). The curve of the time constant has one obvious "knee" when p_{r0} reach the limit acceptable value. The knee determines limit pressure value and enables a calculation limit pressure ratio. The limit pressure in additional reservoir is measured with pressure transducer 6 (Fig. 7). In above procedure, it is calculated the value of the estimated limit pressure ratio as

$$C_{re} = \frac{p_{r0}}{p_s} \tag{10}$$

Obviously, when the supply pressure changes, the higher supply pressure would enable a higher limit pressure in tank which receives exhausted air.

This is the ratio of the limit pressure of collected used air in the additional reservoir and supply pressure in the system.

Above this value the process of capturing deteriorates the dynamic properties of the cylinder and is not recommended to continue to capture the air from the cylinder. This value represents a practical guide for control of collecting exhausted air and its return to a pneumatic system. For the tested experimental installation is estimated $C_{re} \approx 0.50$.

The limit pressure p_{r0}, for one pneumatic drive application, should be determined with the proposed method, for each one actuator drive, considering a real pneumatic system including capturing installation and additional tank used for collecting compressed air.

6. CONCLUSION

Based on the modeling of pneumatic system for capturing of exhausted air and analyzing results of the experiments is concluded that the dynamic properties of the tested cylinder was not exacerbated by the installation of additional tank until ratio air pressure in the tank and working pressure is less than estimated limit pressure ratio C_{re}. In an actual application, limit pressure ratio (10) have to be determined before capturing discharged air from an actuator.

Compressed air that was collected in the additional reservoir can be used to drive another cylinder, in case that the lower value of the pressure from the additional reservoir is sufficient. In subsequent experiments, applying pressure boosters to amplify the pressure in additional reservoir on the sufficient level, have to be investigated.

These results can be used in a control strategy of filling and emptying the additional reservoir while achieving savings in the consumption of compressed air and improvement of energy efficiency of pneumatic systems but without loss of the dynamic characteristics of pneumatic actuators.

The method presented in this article could be adopted for a particular pneumatic drive in specific operation conditions and apply to define one limit pressure in conducting of a capturing process for many actuators at the same time.

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