PNEUMATIKUS HENGER SZABÁLYOZÁSA

CONTROL OF A PNEUMATIC CYLINDER

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ABSTRACT
A modular design MATLAB/Simulink® model has been created in order to model pneumatic cylinder motion, and cylinder control for speed, and position. A simple control model was developed, which can be realized by PLC, or an Arduino like embedded controller. Simulation results are also presented in this paper.

1. INTRODUCTION

A Matlab/Simulink® pneumatic cylinder model has been developed, validated, and presented in the last years of ACIPV conferences.

The in this paper presented pneumatic system description is based on the air mass-flow between the connecting components. The mass flow is driven by pressure difference, and the consumption of one component, is the supply on the other. The model consists of a compressed air supply, pressure reducing valve, T-joints, directional control valves cylinder, and load. The main part of the model is a double acting pneumatic cylinder, which converts pneumatic energy into mechanical one. There are different types of load, and control systems also modelled.

Pneumatic industrial solutions is a fast-developing area, which has many advantages compared to other solutions, like hydraulics, mechanical systems, or electronic drive, and control.

Pneumatic process control, and drive have great history. Besides industrial manufacturing solution, there are other mainly vehicle industrial technical solutions too [1]. For example the first self-propelled airplane, the first self-propelled U-boat had air-motor, but there were compressed-air engine propelled locomotives, trams, cars bicycles, etc. Since 2008 Bosch firm (and successors, Rexroth, Aventics, and Emerson) are organizing the pneumobile competitions, which is a race for pneumatically driven vehicles [2]

Of course, pneumatic solutions have disadvantages too. The main drawbacks are originated from the fact, that the working medium is compressed air.

Compressed air is a mixture of different gases, mainly oxygen, hydrogen, sometimes moisture, and particles as well, and it can be treated as a single component gas. In certain circumferences air can be treated as ideal gas without phase changes, condensation of moisture, etc.

Even with this simplification air dynamic is a lot more difficult to describe, and model, than fluid mechanics. [11] All the thermodynamic state variables of compressed air, like specific, and absolute volume, density, pressure, and temperature are greatly depend on each other. It is hard to find any parameter, which can be base of the model description. Practically the gas constant is the only parameter, which remains constant.

Modelling of a pneumatic system must therefore be based on mass conservation law, and all the thermodynamic properties must be calculated based on the state changes. If at least the change of density would remain close to constant, like in case of fluid mechanics, modelling and control of pneumatic systems would be as easy as in case of hydraulic systems, but because density and thus specific volume greatly depend on pressure and temperature precise position and force control of pneumatic systems are difficult. This complexity can be seen in work of Ferenc Szlivka: Different Mathematical Solutions on Gas Oscillation [8]. Eszter Sárközi publicated a mathematical model of pneumatic piston using using Stribeck Friction [3]. A completely analytical mathematical modelling is described by Vladislav Blagojević, Miodrag Stojiljković [4], and Spartak Poçari, and Andonaq Londo [10]

In general, pneumatic systems are not force and position controlled technical equipment. Usually pneumatic cylinders are making from-end-to end stoke motions without force and position control, except sensing the and-stroke positions.

Modelling a pneumatic cylinder as if it was a hydraulic one draws many mistakes. First of all, the most popular piston force balance equation is not complete, therefore not exact.

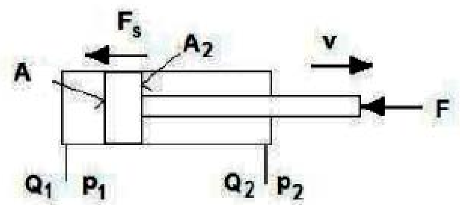

Fig. 1 General representation of a double-acting cylinder [5]

The annotations taken from the original source [5] are the followings: A is the greater, A_2 is the smaller area of the piston. F_s is the friction force, F is the load. p_1, and p_2 are the pressures in the corresponding cylinder chambers, Q_1, and Q_2 are the volume flows. The force balance of the cylinder according to the literature is:

$$F = p_1 \cdot A - p_2 \cdot A_2 - F_s \qquad (1)$$

Using D as the piston, and d for the rod diameter, A, and A_2 surfaces are:

$$A = \frac{D^2\pi}{4} \; ; \; A_2 = \frac{(D-d)^2\pi}{4} \qquad (2)$$

Because of A_2 is $\frac{d^2\pi}{4}$ smaller than A_1. Putting such a cylinder both lines open to the environmental pressure (p_1, and p_2 both are p_0) it is obvious that such a model will produce an $F = p_0 \cdot \frac{d^2\pi}{4}$ piston force, which having F load =0, would – according to the model – move outwards. It is an obvious modelling mistake.

The corrected force balance equation is:

$$F = p_1 \cdot A - p_2 \cdot A_2 - F_s - p_0(A - A_2) \qquad (3)$$

An other often overlooked problem is the pressure in the opposite chamber (p_2), relative to the piston motion direction, especially when a long stroke piston takes a fast forward motion. (Fig 2)

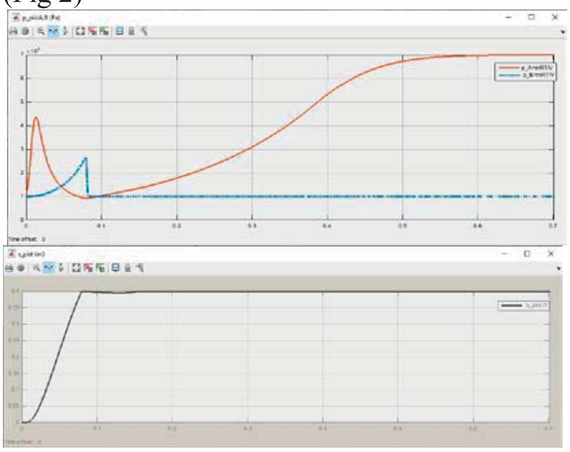

Fig. 2 Pressure, and motion of a fast-moving long-stroke fast moving cylinder

(Description of the simulation to produce the results can be found in the Results chapter.) It can be seen that the dashed line, which represents the p_2 pressure (marked p_B in the picture) reaches about one half of the p pressure (p_A). The longer the way of the exhausted gas from chamber B to the environment, the more the pressure increase is. The pressure increase is also proportional to the piston speed, and the A_1/A_2 ratio.

2. PROBLEM DESCRIPTION

In order to develop a precise control system for pneumatic piston position and force control an adequate model of the pneumatic system has to be derived.

The Matlab/Simulink® model used in this work consist of blocks representing two types of air sources, which are compressed air bottle, or a compressed air pipeline network. In both cases air consumption is calculated, in case of bottle supply the current bottle pressure is calculated as well. Both sources are connected to a pressure reducing valve, which is often called regulator. There are pneumatic pipelines, T-joints, and directional regulating valves also modelled. The main components in the model are the double acting two-chamber linear cylinder, connected to the load-model.

The overall structure of the model shown in Fig. 3.

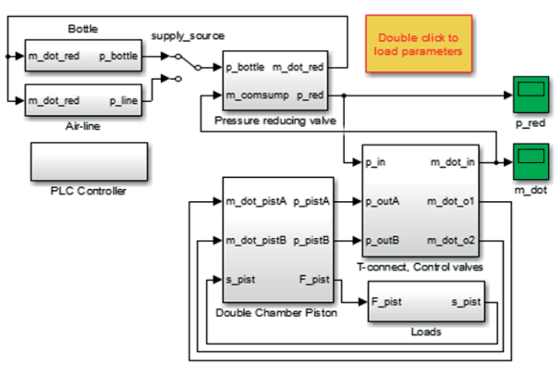

Fig. 3 Overall structure of the model

The compressed air tank is a constant volume, changing mass thermodynamic system. The operation of the model: the filled bottle of V volume contains m kg amount of gas which is characterized by R gas constant. (Fig. 4)

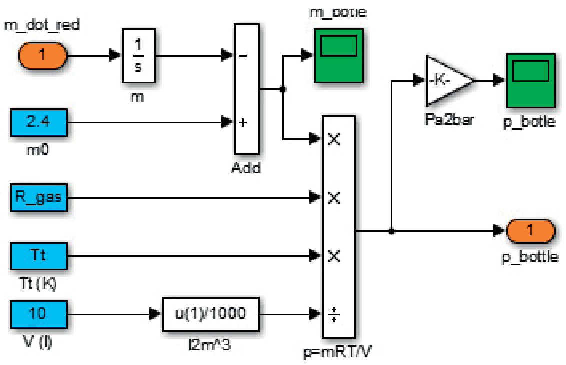

Fig. 4 The bottle model [6]

The pressure in the tank can be calculated as:

$$p = \frac{mRT}{V} \qquad (4)$$

The temperature in the tank is considered constant, and equal to the environmental temperature. The $\dot{m}$ gas consumption is calculated in the pressure reducing valve submodel. (Fig. 6). Having the mass flow of the consumption integrated in time, the mass reduction of the enclosed gas is calculated, from which the pressure of the tank is updated in each simulation step.

In this figure the function annotations, and the color codes will be explained. Orange color is used for input, and output signals in the submodels. Light blue blocks are the constants, greens are scopes. Yellow blocks are Goto, and From pairs, mainly used for control signals (See later in Fig. 10), except the reduced gas temperature, which is used by all the systems in the model (See later in Fig. 8)

Using of colors are significantly increasing reading comfort, and lucidity of the model. Naming the blocks after their function equations, or signal names also help the better understanding the components, and their relations.

It can be seen that the inputs, and constants are aligned to the left, and all the output and scopes are lined up on the right side. All the important signals are scoped for better understanding, and validation. there are 40+ scopes in the model currently

The compressed air line model is very simple (Fig. 5)

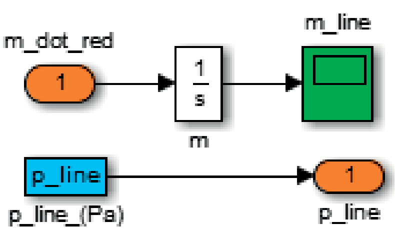

Fig. 5 Compressed air line model

This model calculates comsumpted air from integrating mass flow of consumption and sets line pressure to p_{line}.

The model is fully parametric. There no handtyped constant values, (except 0, 1, and 2) each constant value is handled by global variables from Matlab surface.

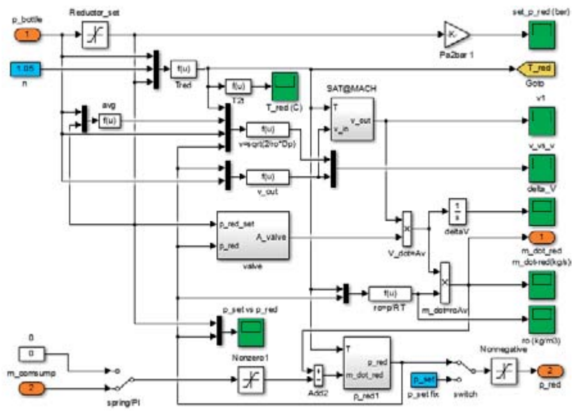

Fig. 6 The pressure reducing valve model [7]

The pressure reducing valve model shown in Fig. 6 has the following tasks to fulfill

- To determine the sound speed corresponding to the gas state
- To determine the reduced pressure gas temperature in the working chamber
- To calculate the pressure in the regulator working chamber
- To model the valve closing element motion in the pressure reducing valve
- To calculate the gas speed, volume-, and mass flow.

The dual chamber piston model consist of two air chambers. Chamber A, and B. Instead of using Fig. 1 it is recommended to use Fig. 7

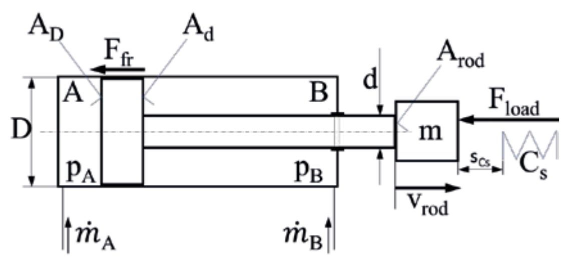

Fig. 7 The dual chamber piston used in the model

The variables used in the modelling are: A is the chamber opposite the piston rod, B is the chamber when the piston rod is A_D is the greater, A_d is the smaller area of the piston. F_{fr} is the friction force, F_{load} is the load. p_A, and p_B are the pressures in the corresponding cylinder chambers, $\dot{m}_A$, and $\dot{m}_B$ are the mass flows, m-is the mass of the load, F_{load} is the external load force, C_s, is the load spring stiffness (if there is any), s_{Cs} is the spring distance.

The force balance equation is described in Eq 3, using the above annotations is:

$$F = p_A \cdot A_D - p_B \cdot A_d - F_{fr} - p_0 \cdot A_{rod} \qquad (5)$$

The complete model can be seen in Fig. 8

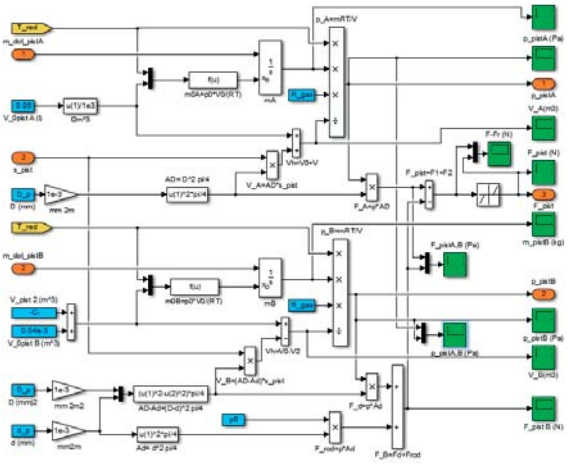

Fig. 8 The dual-chamber piston model

For better understanding model, one half of the piston force model is shown in Fig. 9.

As it can be seen, the initial chamber pressure, which is p_0 is provided by setting the integrator of mass to initial m_0 equivalent mass of air in a V_0 piston chamber volume at p_0 pressure. The corresponding equation is:

$$m_0 = \frac{p_0 V_0}{RT} \qquad (6)$$

$$V_h = V_0 + A_D \cdot s_{pist} \qquad (7)$$

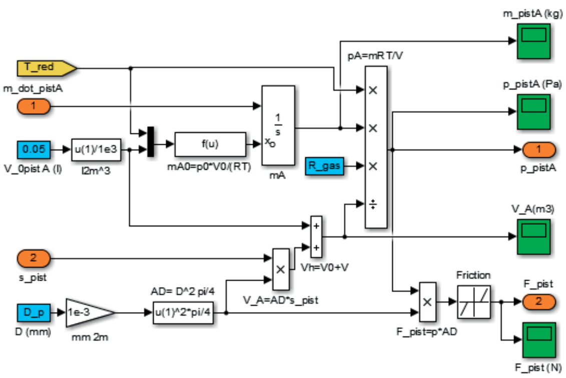

Fig. 9 Chamber A part of the dual-chamber piston model

The piston force is then calculated as pressure times working area:

$$F_{pist} = p \cdot A_D \qquad (8)$$

The chamber B force is calculated in the same way, except the working area is smaller, because of the piston rod.

The two-chamber piston required to develop a T-junction model, because the air had to divide in between the chambers. The chambers are alternatively connected to the supplied air-, or the environment pressure, or in case of X/3 type of directional control valve, the gas amount in the chamber can be enclosed. Usually one chamber is filled, the other is exhausting, like when using 4/2, or 5/2 bi-stable directional control valves. In some cases ,4/3, or 5/3 central locked, valves are used, when in middle position all the channels are closed, or two pieces 5/2 type monostable valves are used, which allows individually chamber A, and B to be charged, locked, or exhausted. This modell uses on/off solenoid valves [10]

To control which one is filled and which is exhausting the pressure there was need to develop directional control valves. The T-junction, and the directional control valves are realized for practical reason in one submodel. The total model is shown in Fig. 10.

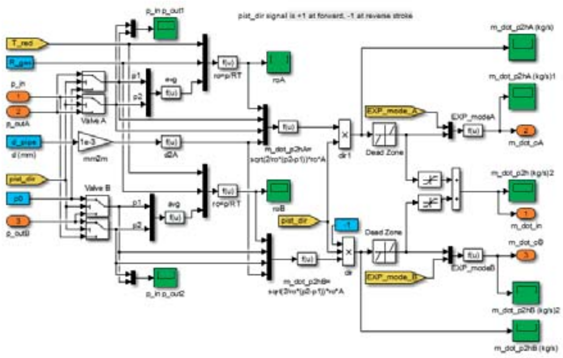

Fig. 10 5/3 Valves, and T-junction model

Fig. 11 shows a 4/2, or 5/2 type of realization of the directional control valve. This figure is a cut from Fig. *10*. This part of the model connecting chamber A with p_{in} (charge), and B with P_0 (exhaust) when piston stroke is forward, and oppositely in reverse stroke.

Fig. 12 shows the solution of T-junction, which connects chamber A, and B consumption $\dot{m}_{oA,B}$ when chambers are filled, and connects consummated mass flow ($\dot{m}_{in}$) to pressure reducing valve block.

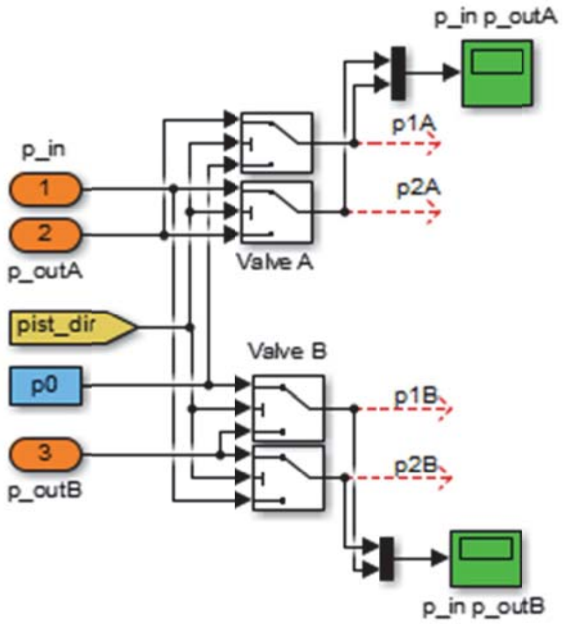

Fig. 11 4/X or 5/X type of directional control valves

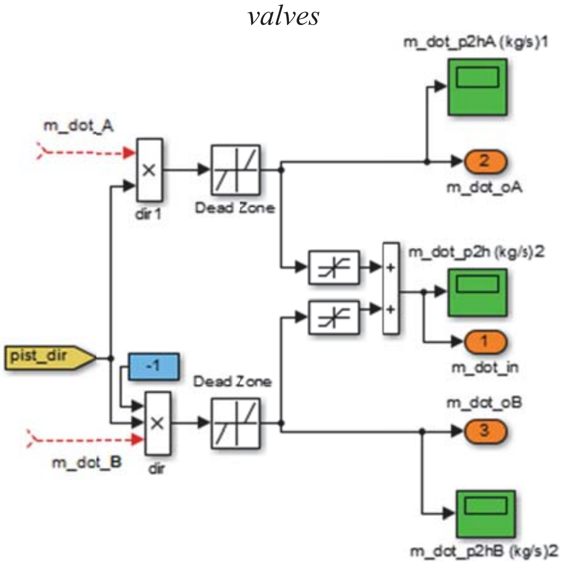

Fig. 12 Check valves, and linearized flow resistance in T-joint, X/2 variant

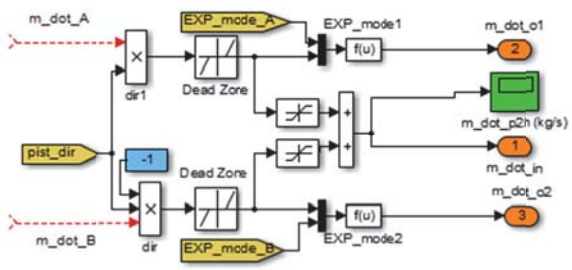

Fig. 13 Check valves, and linearized flow resistance in T-joint, X/3 locked variant

Fig. 13 shows solution, when instead of 4/2 (or 5/2) a 4/3, (or 5/3) valve are used. The X/3 valves have floating, or locked neutral position. The figure shows the locked version, when in middle position gas flows are locked in all channels. In this case EXP mode A, and B values are 0.

The consumption of piston chambers A, and B are calculated inFig. *14*. The input parameter for the calculation is the pressure difference between chamber, and connected pressure ($p_{reductor}$, or p_0). Its relating equations is:

$$\dot{m}_{p\rightarrow hA,B} = \sqrt{\frac{2}{\rho}(p_2 - p_1)} \cdot \rho A \qquad (9)$$

The gas average density required by Eq. 9 is calculated from average pressure as follows:

$$\rho = \frac{\bar{p}}{RT} \qquad (10)$$

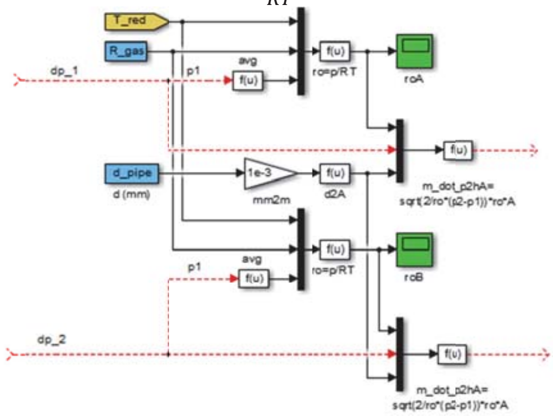

Fig. 14 Calculating mass flows to chamber A, and B from pressure differences

The load model is a simple Newtonian calculation using an M load which is accelerated linearly. Acceleration of the mass is calculated by a=F/m. Acceleration is integrated to have piston speed, and piston travelled distance. The speed integrator is reseted at stroke ends. Stroke end is detected by saturating piston distance at stroke end, and stroke=0. Reaching stroke end can be se to reverse piston motion.

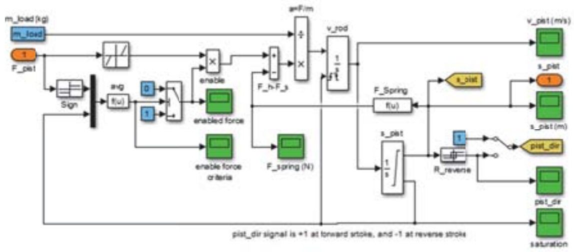

Fig. 15 Load, and piston motion model

Beside inertia load, there is a spring load also developed (see Fig. 7) When piston stroke exceeds s_{Cs}, then spring load force is also calculated.

3. RESULTS

There are 40+ scopes in the model. After simulation run all the important parameters can be investigated and compared. In the results chapter only three scopes will be shown, which are Chamber A, and B pressures, piston motion, and piston force.

The Fig. 2, which is showing the pressure build-up in a fast-moving cylinder is a result of the above described model.

The parameters used to produce the figure are:

% ** General properties **
R_gas=287.05; % Gas constant (J/(kgK))
Tt=273.15+20; % Tank air temperature (K)
p0=1e5; % Environmental temperature p0= 10^5(Pa) = 1 Bar
p_set=7e5; % Reductor set absolute presure (Pa)
p_line=9e5; % Pneumatic line absolute presure (Pa)
d_pipe=10; % Pneumatic pipe diameter (mm)

% ** Piston properties **
h_p=0.4; % Sroke (m)
D_p=80; % Piston diameter (mm)
d_p=36; % Piston rod diameret (mm)
F_pr=40; % Piston friction force (N)

% ** Load properties **
m_load=5; % Mass of load (kg)
mu_mech=0.8 % Mechanism friction coefficient (-)
Cs_load=0; % Load spring coeddicient (N/m)
s_Cs=0; % load spring distance from inposition (m)

During this simulation the reductor pressure is significantly decreasing, caused by the increasing air consumption. Most of the cases when fast piston movements are required, the reductor is the bottleneck of the system. In such cases using of a puffer tank is recommended.

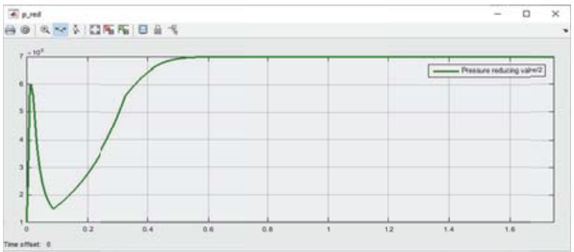

Fig. 16 Reductor pressure during simulation

Changing the supply pressure to constant pressure, which is possible to switch the source switch to $p_{set\ fix}$ in the reductor model (Fig. 6))the same simulation is providing the following results:

The chamber pressures and piston position shown on Fig. 2, is changed to Fig. 17.

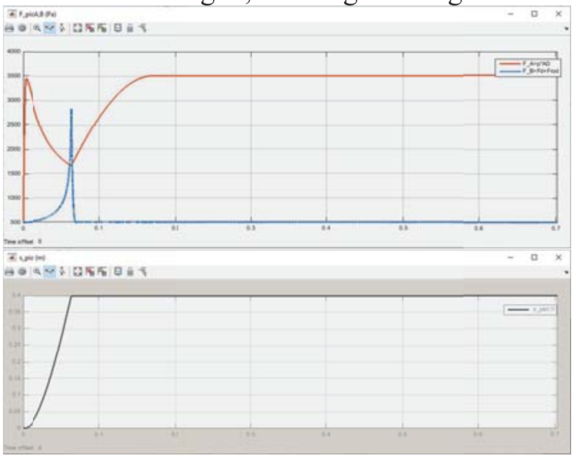

Fig. 17 Pressure, and motion of a fast-moving long-stroke fast moving cylinder

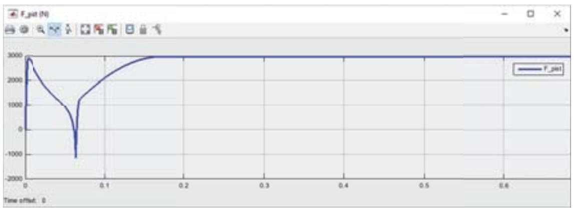

Fig. 18 The piston force during the simulation

4. CONCLUSIONS

Having the model developed, and tested, the net step is to develop a control system, which is able to control the piston force, and position. Instead controlling the pressure in chamber A, to provide force to move the piston to position, it is recommended to control both A, and B chamber pressures, to maintain maximum pressure in chamber B, and a Δp pressure in between the chambers. in this way air pressure, and as a consequence air density is also maximized, reducing the piston position sensitiveness from load change.

The model will be extended by user interface, and animation of piston, and load motion for education purposes.

Cylinder friction will be modified from Coulomb to Coulomb+ Static+ viscous friction or Stribeck friction model [3], [9]

5. SUMMARY

MATLAB/Simulink® model was created in other to simulate mechanic, thermodynamic, and fluid-mechanic behavior of a pneumatic system used in industrial application. Pneumatic system specific modelling aspects has been described, model has been developed, and introduced.

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