Pneumatic Piston Control Modelling and Optimization

Tamás Szakács

Óbuda, University, Bánki Donát Faculty of Mechanical and Safety Engineering - BGK, MEI, 1034 Budapest, Bécsi út 96/b, szakacs.tamas@bgk.uni-obuda.hu

Abstract: Piston control can be used for position, speed, acceleration, and force control. The main difference between pneumatic, and hydraulic cylinders is that the hydraulic control uses fluid instead of gas, and fluids are practically incompressible. Therefore, hydraulic cylinders can be more accurately controlled for position and speed. Pneumatic cylinders are showing greater dependence on external loads. Pneumatic systems in the contrary are simpler as construction, cheaper than hydraulic ones, and more wide-spread in laboratories and shops, but usually they are used for end-to end motions without exact position control. This paper introduces a pneumatic piston control that decreases the drawbacks of the position control of pneumatic cylinders, making it allow for simple application.

Keywords: pneumatic piston; position; control; vehicle height control

1 Introduction

Modelling of a hydraulic cylinder is not an easy task it involves thermodynamic, fluidmechanic, and mechanical theories. Modelling of a pneumatic cylinder is even more difficult, because working medium in pneumatics is compressible, and because of that the piston position cannot be exactly controlled, piston position and behaviour is greatly depending on the changes on external loads. Based on this, pneumatic cylinders are not usually used for position control.

Recently a special interest can be seen on pneumatics due to development of many affected areas, like robotics, and pneumobiles [1], and conventional vehicles. Robotics requires accurate control, and sensorics, same as pneumobiles, where cylinder force, and motion are controlled. Haiato, et al. [2] is describing a discrete-valued model-predictive control (DVMPC) algorithm with promising results. Power & Motion Tech [3] describes an ElectroPneumatic Positioning System (EPPS) using Rexroth (Aventics) components. The article summarises advantages of EPPS compared to ElectroMechanical positioning (EMPS). It draws a conclusion of a frequently overlooked phenomenon: “With directional flow control, minor leaks cause the pneumatic cylinder piston to move slightly.

The positioning system's controller receives motion feedback from the sensor, and the controller responds by slightly adjusting flow to compensate for piston movement. This slight, but constant back-and-forth motion, known as dithering, negatively affects system accuracy and functionality." Their control schematics is described in Figure 1.

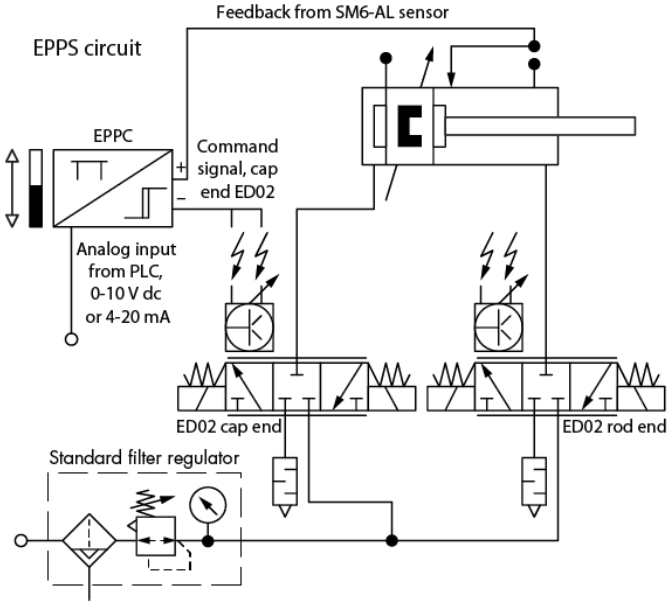


Figure 1
Typical EPPS system using a rod-type pneumatic cylinder [3]

Controlling pneumatic cylinders has drawn interest of many other researchers and skilled amateurs as well.

There are studies of pneumatic system control modelling using Graph model-based analysis of technical systems [4], and [5].

The density of the working medium in hydraulics is 750-800 kg/m^3, while in pneumatics is 1.2-1.3 kg/m^3 depending on the temperature, and pressure. The difference is in the range about three great-orders. In most of the cases low pressures are used, if the cylinder is controller for position, or speed. Gas density if a function of pressure. As the pressure increases, so does the density. Because the load dependency is related to the gas density, if the operating pressure could be increased, so would the load dependency be decreased.

If moving a piston would be realized not by filling one cylinder chamber, and releasing the opposite, but filling both sides with controlled pressure, the piston would move the same way with having significantly greater pressures in both chambers. In this way position dependency of external load would be significantly decreased as well.

A possible application of this control system can be sky-hook like purely pneumatic vehicle body control, or platform height control in industry, or home applications, or in robotics.

In this paper a pneumatic cylinder modelling and control for industrial applications will be introduced.

2 Descriptions

In the ACIPV conference series different pneumatic modelling theories have been introduced by a variety of authors, mostly for pneumobile pneumatic system modelling, and control. This model of an industrial control model is based on an earlier published model, which has been updated, and modified for the actual task.

Properties of the 2023 compared to earlier published models [6] are:

- The cylinder is a dual chamber type, to be able to modell, and simulate exhaust pressure drop, and pre-charging in both chamber sides.
- A controller block is added to simulate PLC, or any kind of microcontroller. In some case pats of program codes can be copy-pasted from PLC, or controller development environments.
- Directional valves are added, and updated to control gas flow, and direction.
- Mechanical load block has been re-worked to fulfill the special industrial task.

During the modell creation it was always kept in focus that the model should remain as simple as possible, and easy to understand for students, and colleagues.

Figure 2 shows the pneumatic schematics of the modelled system. Its detailed description can be found in earlier ACPIV publications [e.g.: 1].

Figures 3 and 4 show the difference between the top layers of the original and the advanced model. The most significant changes on the top layer are the controller block, and the air-line is added to the air-source beside the bottle.

The bottle, and reductor submodels have not changed because these parts are identical in both systems.

The bottle, body and reductor models are described in the ACIPV 2018 Conference paper. (Szakács, 2018 [6])

The 2019, two-chamber cylinder model is shown on Figure 5.

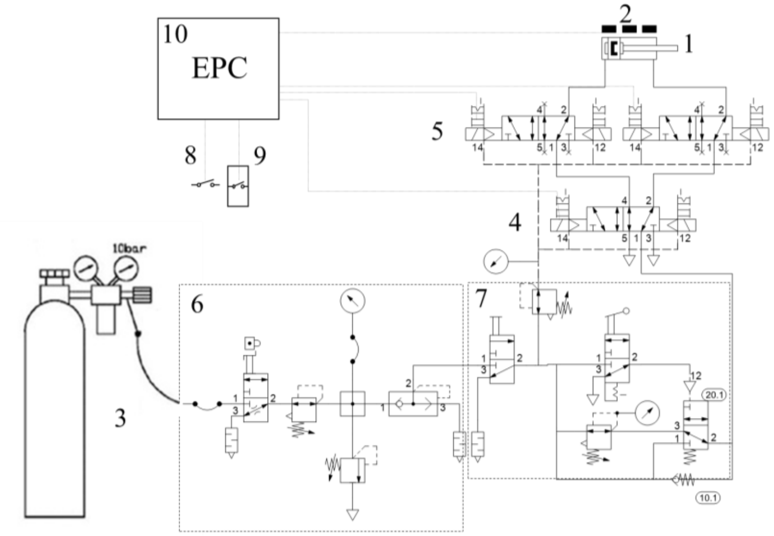


Figure 2
Schematics of the pneumatics

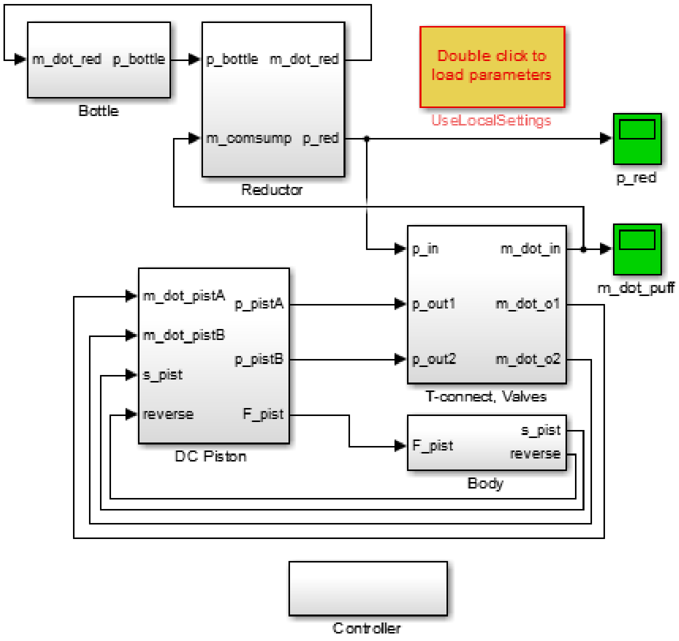


Figure 3
The 2019 Pneumobile pneumatic system modell

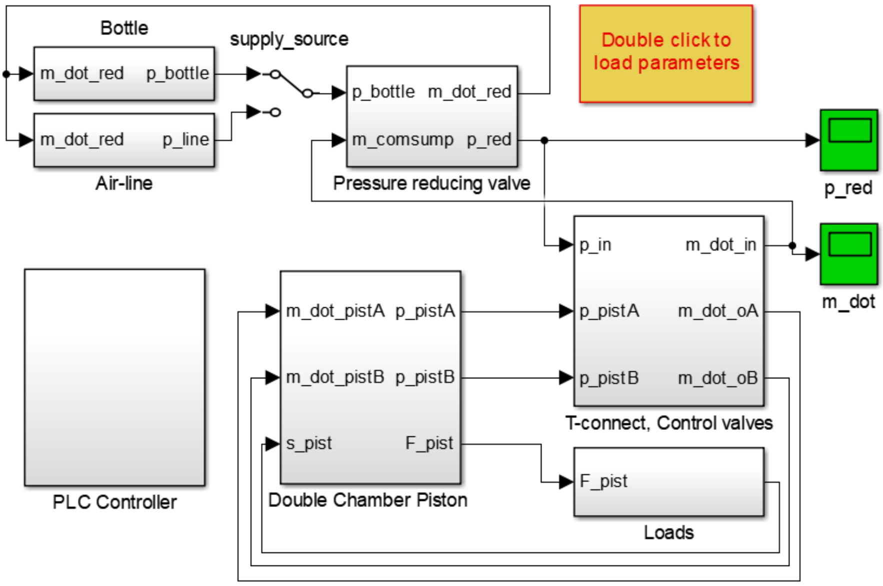


Figure 4
The current industrial piston control modell

The chambers usually are alternatively connected to the supply air, or the environment. One chamber is filled, the other is exhausting. In the new model the cylinder chambers can be independently connected to supply air, or the exhaust, or the inner air can be locked in. (See the schematics on Figure 1) To control which chamber is filled, locked or, which is exhausting the pressure the directional control valves model has been developed.

The 2019 controller model was representing a 5/3 type of directional valve to alternatively charge, and release the two chambers, and locking the air flow. This latest has been originally developed for ECO expansion control during long distance run in the pneumobile competition [1]. The piston direction control signal is calculated in the body submodel. At that state there was no pre-chare control on the engine. See on Figure 6.

The operation of the 5/3 directional valve is explained in Figures 6-8. Both channels fill, or release on the same time possible only in that modell, 5/2, 5/3 valves are not able to operate in that way.

Figure 9 shows a case when the chambers are alternately filled/released, as real valves do.

In the 2023 model the 5/3 directional valve has been replaced by a dual 3/3 one. By this mean both chambers of the cylinder can be independently filled, locked, or released not only alternately, but completely independently, also parallel if so needed.

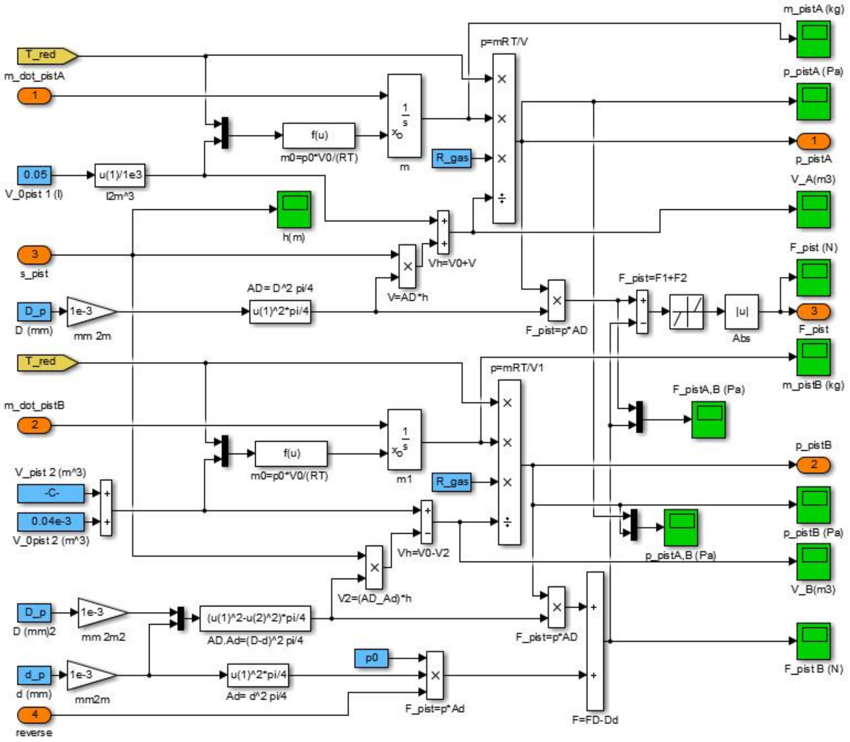


Figure 5

The 2019, dual chamber piston modell

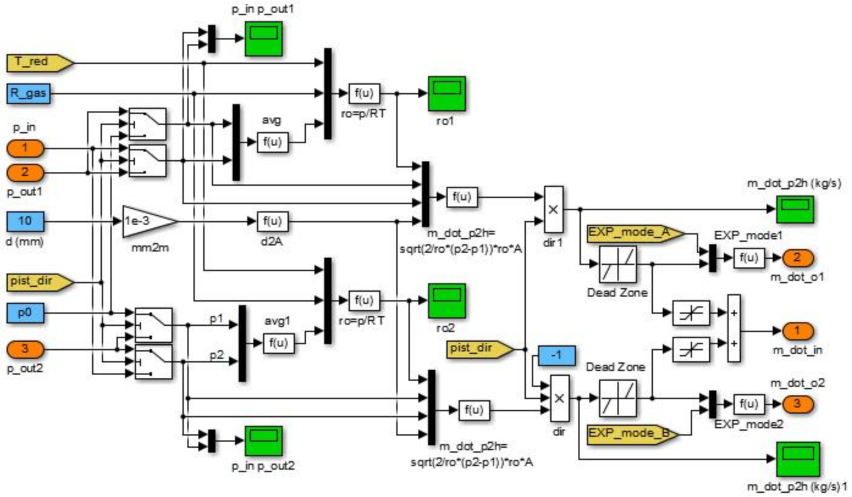


Figure 6

5/3 valves, and T-junction modell

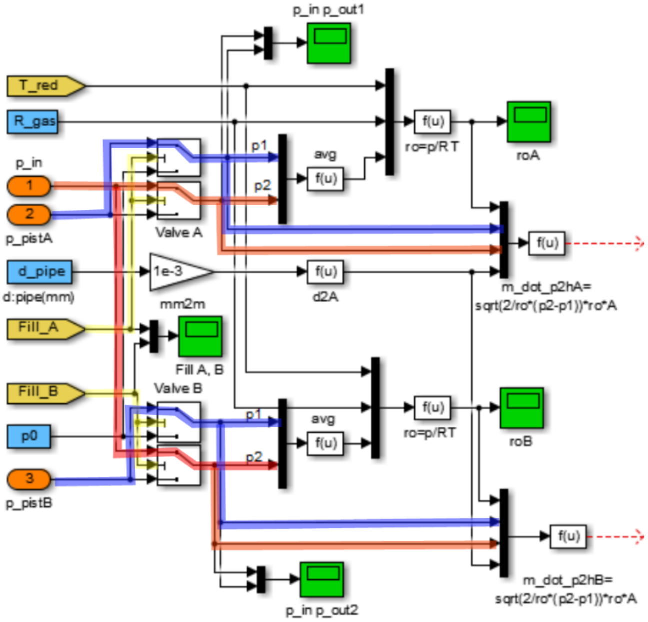


Figure 7
Operation of 5/3 valves: both chamber fill

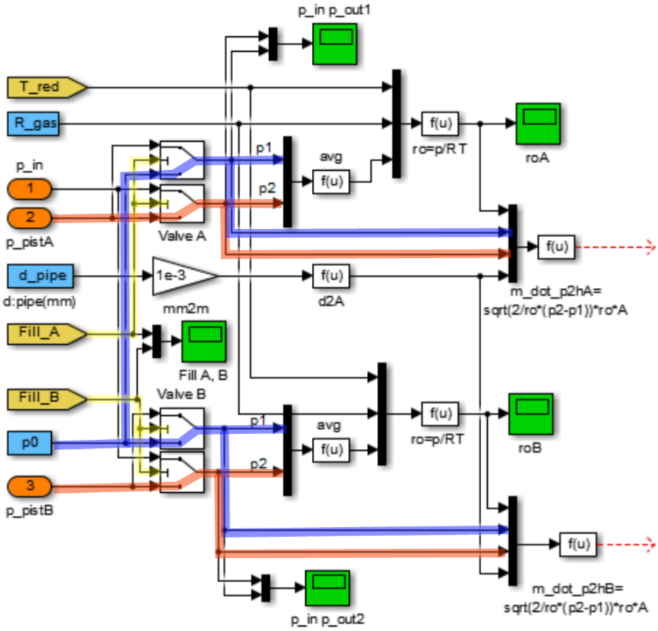


Figure 8
Operation of 5/3 valves: both chamber release

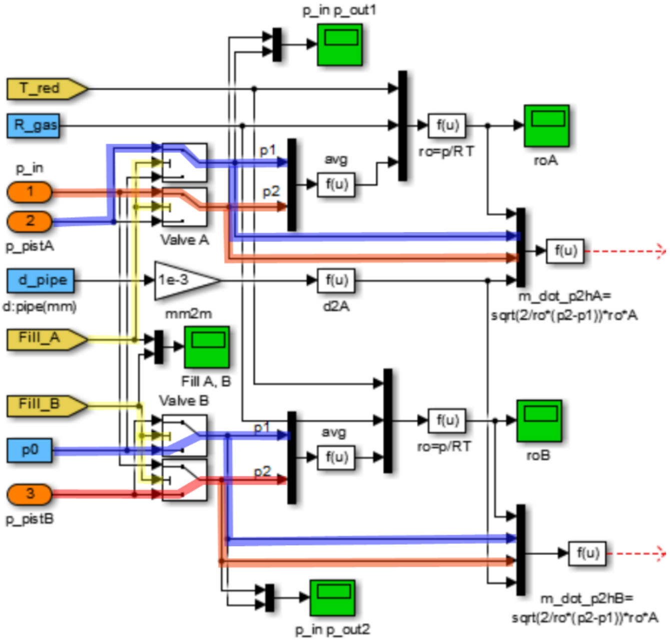


Figure 9
Operation of 5/3 valves: Chamber A fill, B release

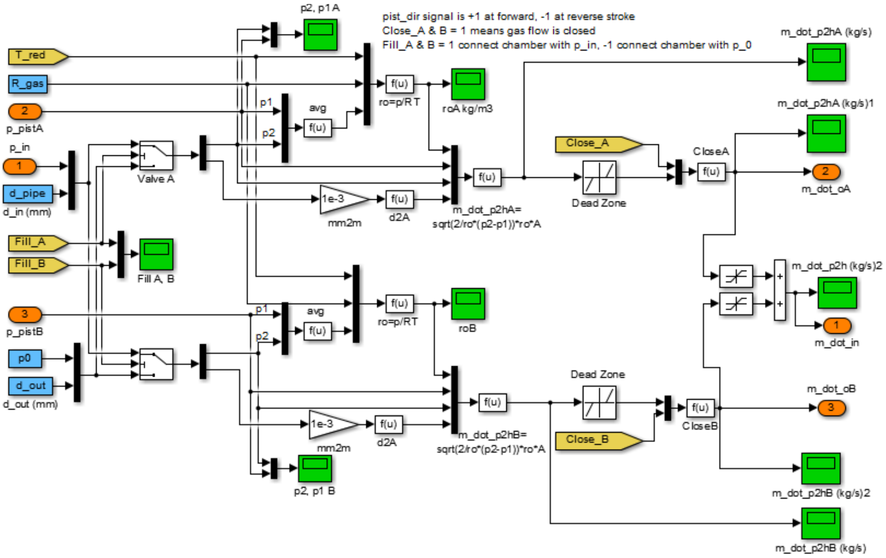

pist_dir signal is +1 at forward, -1 at reverse stroke
Close_A & B = 1 means gas flow is closed
Fill_A & B = 1 connect chamber with p_in, -1 connect chamber with p_0

Figure 10
Dual 3/3 valves with differential intake and exhaust

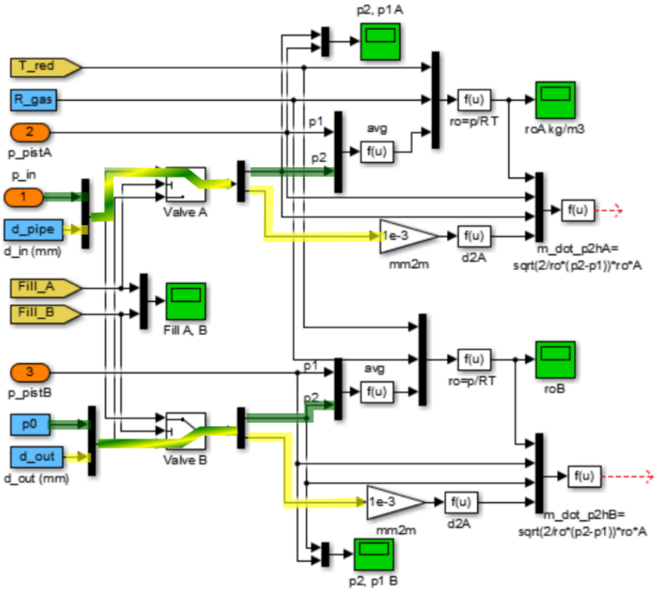


Figure 11

Dual 3/3 valves with differential intake and exhaust

Figure 11 shows the mechanical, and load model of the cylinder. In this modell block the mechanical motion of the piston, and the connected load is modelled. Acceleration, speed, and piston-rod position are calculated, as well as friction force, and damping. Piston end positions are detected, which limits piston motion, and further accelerating forces.

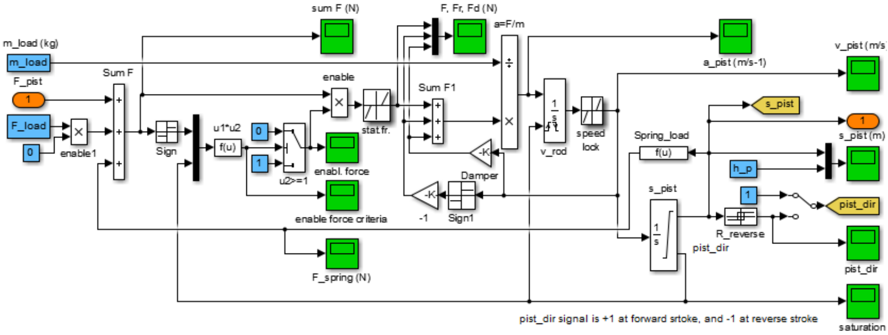


Figure 12

The mechanical motion, and load modell

Figure 13 shows the external loads, spring force (can be disabled in the modell) friction force, and variables used in the modelling.

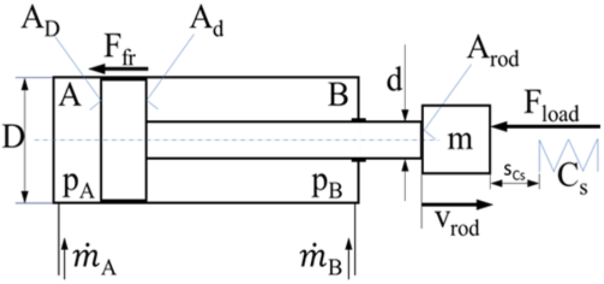


Figure 13
Annotation of the piston used in the modell

$$F = p_A \cdot A_D - p_B \cdot A_d - F_{fr} - p_0 \cdot A_{rod} \tag{1}$$

Equation (1) explains the force balance (including spring load, but excluding damping force).

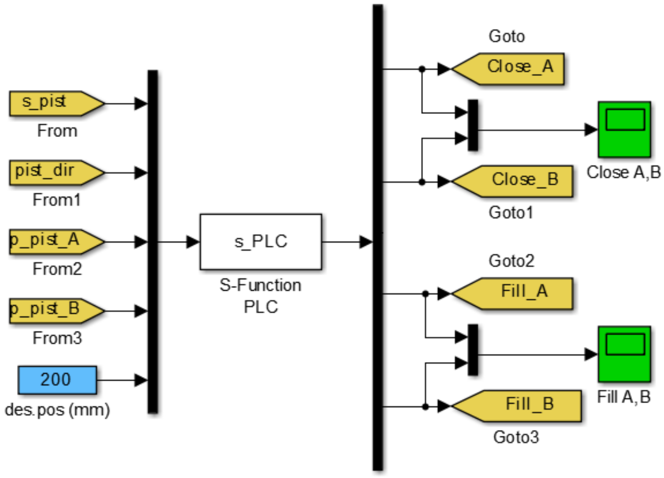
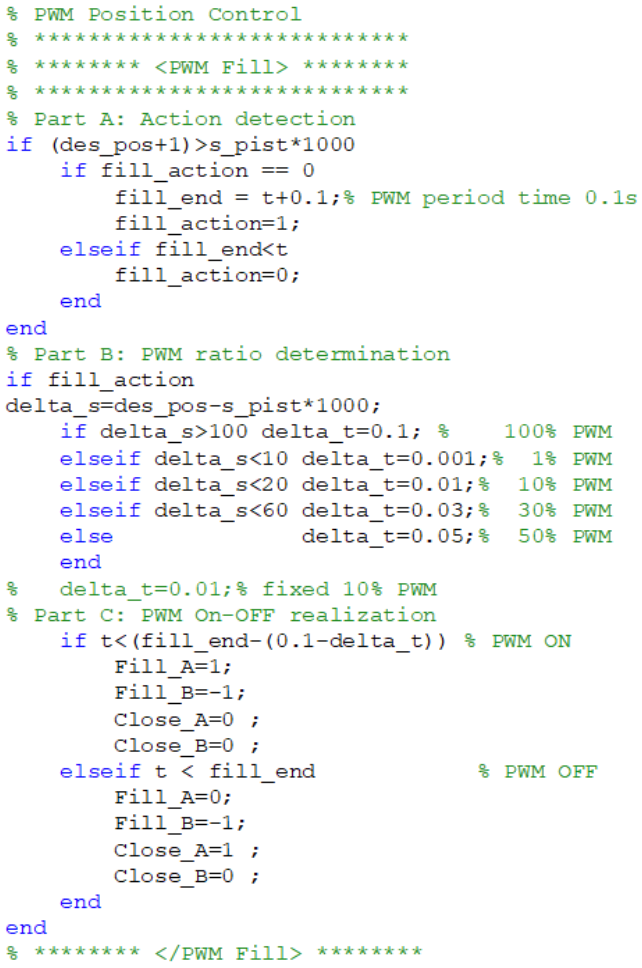


Figure 14
The controller modell

The controller is not connected to the other blocks of the model by data line, which represent that the electric control does not mix pneumatic, or mechanic lines in the modell. Figure 14 shows an example of the PWM position control code.

In part “A: Action detection” the fill action is detected, and the action trigger time is recorded based on trigger occurrence timepoint and PWM period length Fill action is pulled high, so in this period no more action detection is done till the action end.



```
% PWM Position Control
% ****************************
% ******** <PWM Fill> ********
% ****************************
% Part A: Action detection
if (des_pos+1)>s_pist*1000
    if fill_action == 0
        fill_end = t+0.1;% PWM period time 0.1s
        fill_action=1;
    elseif fill_end<t
        fill_action=0;
    end
end
% Part B: PWM ratio determination
if fill_action
delta_s=des_pos-s_pist*1000;
    if delta_s>100 delta_t=0.1; %    100% PWM
    elseif delta_s<10 delta_t=0.001;%  1% PWM
    elseif delta_s<20 delta_t=0.01;%  10% PWM
    elseif delta_s<60 delta_t=0.03;%  30% PWM
    else              delta_t=0.05;%  50% PWM
    end
%   delta_t=0.01;% fixed 10% PWM
% Part C: PWM On-OFF realization
    if t<(fill_end-(0.1-delta_t)) % PWM ON
        Fill_A=1;
        Fill_B=-1;
        Close_A=0 ;
        Close_B=0 ;
    elseif t < fill_end            % PWM OFF
        Fill_A=0;
        Fill_B=-1;
        Close_A=1 ;
        Close_B=0 ;
    end
end
% ******** </PWM Fill> ********
```

Figure 15

Example code for PWM cylinder chamber fill

In part “B: PWM ratio determination” the PWM ON period is calculated based on the difference between desired, and actual piston position. PWM ON period is greater when difference is higher, and getting less when desired position is being achieved.

In part “C: PWM ON-OFF realization” as the name indicates the fill, and the hold states are switched. For the PWM ON period fill is ON, and hold (close) is OFF. For the PWM OFF period fill is ON, and hold (close) is also ON, so there is no actual fill for this period.

This modell code does not include directional control valve switch-on (tF), and switch-off (tE) times. In case of the Aventics 3/2-directional valve, Series CD12 (Part No.: 5724500920) tF=34, tE=90 ms [7].

The control code can control directional valve states either by time, or by event.

Event control can be based on piston position, chamber pressure, piston force, etc. These variables can be passed through the PLC code as global variables, or through the input of the s-function. In Figure 14 variables passed through the input are piston position, piston motion direction, chamber pressure A, and B, and the desired position.

Examples for time control can be seen in the results chapter Simulation 1 part.

3 Results

The first simulations have been run to present the differences between released, and filled counter-chamber (chamber B)

Simulation 1:

In this pair of simulation simple piston motion will be presented once regular fill, and then fill against pressurized opposite chamber.

p_{set}=4 bar, d_{in}=3 mm, d_{out}=10 m, m_{load}=500 kg, h_p=400 mm, D_p=80 mm, d_p=36 mm

Set 1.1:

```
if and (t>=0.5,t<1)
    Fill_A=1;
    Fill_B=-1;
    Close_A=0;
    Close_B=0;
end
```

Figure 16

Piston rod position (Scope s_pist (m) in Loads modell) set1.1

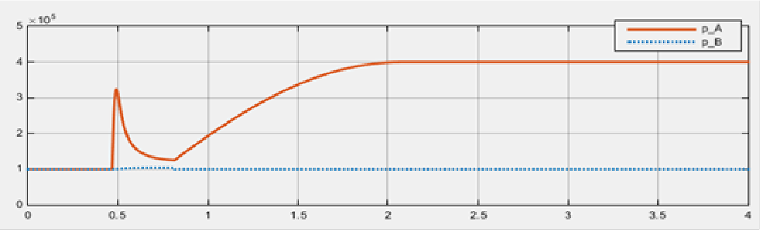


Figure 17
Chamber A, and B pressures (p_pistA,B (Pa) scope in Double Chamber Piston modell), set1.1

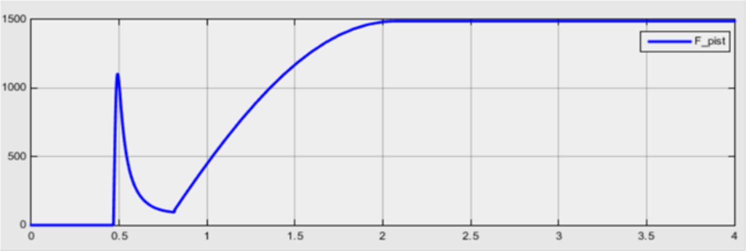


Figure 18
Piston force (F_pist (N) scope in Double Chamber Piston modell), set1.1

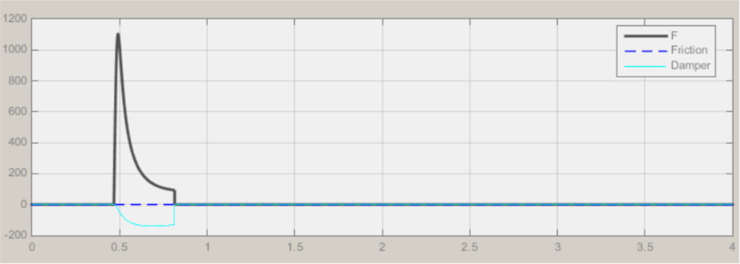


Figure 19
Acting force, and resistances (Scope F, Fr, Fd (N) in Loads modell) set1.1

Set 1.2:

```
if and (t>=0.5,t<1)
    Fill_A=1;
    Fill_B=1;
    Close_A=0;
    Close_B=0;
end
```

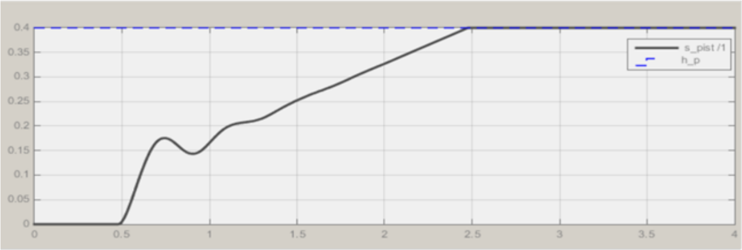


Figure 20
Piston rod position (Scope s_pist (m) in Loads modell) set1.2

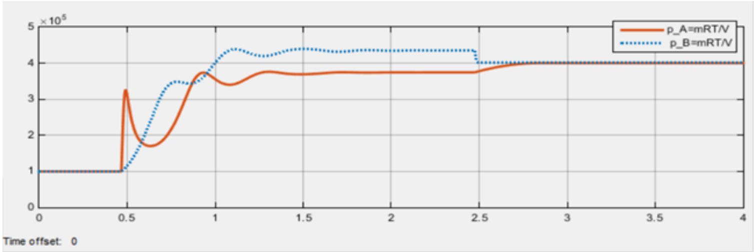


Figure 21
Chamber A, and B pressures (p_pistA,B (Pa) scope in Double Chamber Piston modell), set1.2

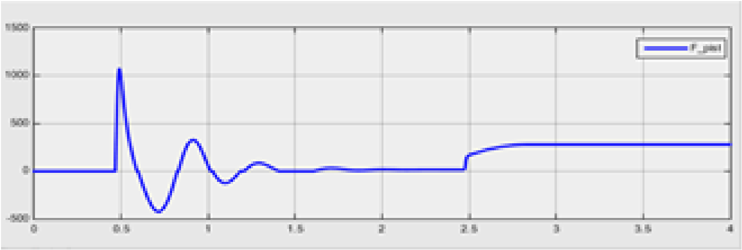


Figure 22
Piston force (F_pist (N) scope in Double Chamber Piston modell), set1.2

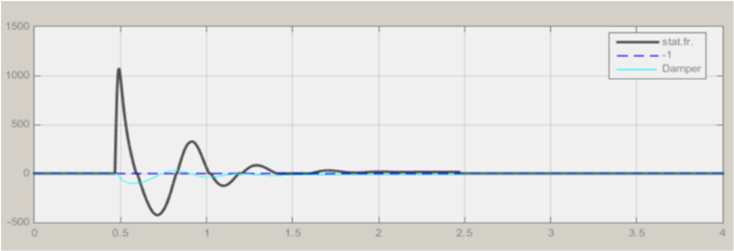


Figure 23
Acting force, and resistances (Scope F, Fr, Fd (N) in Loads modell) set1.2

Evaluation of simulation 1, sets 1.1, and 1.2:

The main difference between sets 1.1, and 1.2 can be seen is figure 17 and 21 regarding the chamber pressures, and figure 16 and 20 regarding piston position. The piston travelled at lower speed of course, but the chamber pressures are significantly greater on both sides. This results higher gas density, thus less dependency on external loads.

Simulation 2:

In this simulation the PWM piston position control will be demonstrated without opposite chamber pressuration.

p_{set}=6 bar, d_{in}=2 mm, d_{out}=10 mm, m_{load}=20 kg, h_p=400 mm, D_p=80 mm, d_p=36 mm

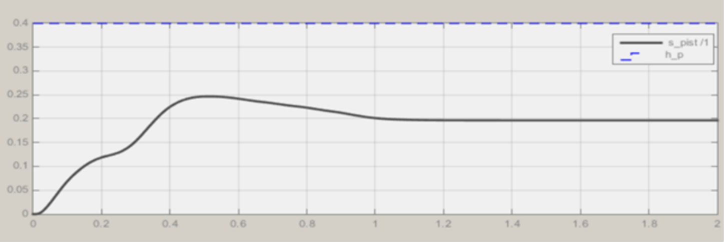


Figure 24

Piston rod position (Scope s_pist (m) in Loads modell) set2

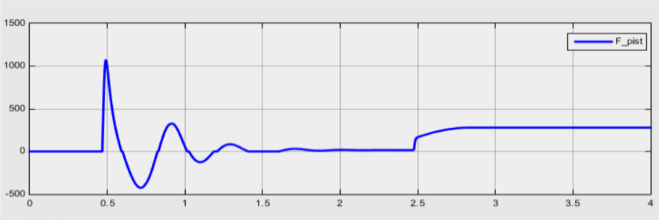


Figure 25

Chamber A, and B pressures (p_pistA,B (Pa) scope in Double Chamber Piston modell), set2

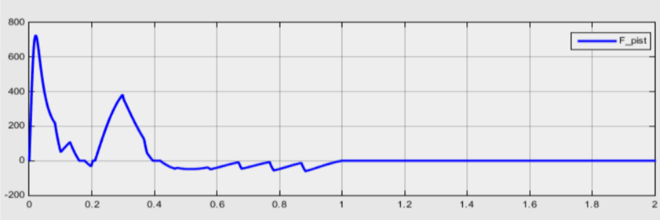


Figure 26

Piston force (F_pist (N) scope in Double Chamber Piston modell), set2

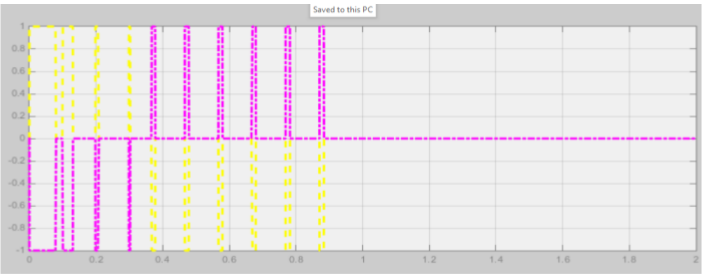


Figure 27
A and B chamber Fill (Scope Fill A,B in PLC Control modell) set2

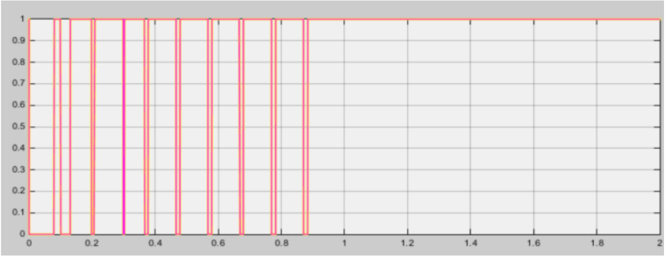


Figure 28
A and B chamber Close (Scope Close A,B in PLC Control modell) set2

4 Outlook

The further development of this modell will include a Graphical User Interface (GUI), and a graphical representation of the cylinder including piston motion, and chamber pressure indication by colors to help better usage even for those who are not expert in coding in m-code, and programming in Simulink.

Fine-tuning of the controller logic is needed, and more simulation must be performed for PWM-control with pressurized opposite chambers, to modell load changing reactions, and comparison of position control with, and without pressurized opposite chamber in varying external load conditions.

Switch-on and switch-off delays of pneumatic directional valves has to be included in the modell, and control code must be corrected accordingly.

Air leakage have to be included in the modell to simulate dithering.

Conclusions

This research has proved that position controlling of pneumatic cylinders using PWM position control is possible. Also, that positioning the piston with a controller that pressurizes both cylinder chambers decrease external load dependency with a factor of 5-10.

Expectations during modell, and controller development indicated that vibrations may have an issue for control. Usage of a vibration damper in the controller system may be necessary.

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