PNEUMATIKUS HENGEREK VIZUÁLIS TERMIKUS SZIMULÁCIÓJA LABVIEW HASZNÁLATÁVAL

VISUAL SIMULATION OF THERMODYNAMIC EFFICIENCY OF PNEUMATIC CYLINDER USING LABVIEW

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ABSTRACT
Performance of pneumatically driven vehicle depends on multiple factors, and efficiency of energy to work conversion in the pneumatic cylinder is one of them. This paper describes thermodynamic model of pneumatic cylinder. Equations that define cylinder's geometric properties, gas supply control parameters and energy conversion are presented. *MATLAB* scripts are included. *LabVIEW* user interface is presented, which allows altering of geometric and control parameters of the pneumatic cylinder in real time. The results of the simulation can be presented graphically and numerically.

1. INTRODUCTION

Linear pneumatic actuators are significant part of industrial machinery, that provide fast acting and relatively low cost solutions to automation tasks [1]. Improvement of pneumatic system efficiency is a topical research subject. Selection of the right dimensions of pneumatic cylinders and control valves for the given task are emphasized and discussed in previous work [2–4].

Loses in the control and supply circuit are important, especially in the large industrial installations. Efficiency of the pneumatic control valve and its improvement were presented in the work of Estonian researchers [5].

In a typical application of pneumatic cylinder, at the end of the expansion cycle gas is released to the surroundings. Energy recovery from the released gas is explored by various researchers. Luo *et al*. propose application of rotating compressors to convert gas energy to electrical energy. They experimentally achieved energy efficacy improvement up to 18.1% using scroll-type compressor and DC generator [6].

Letai *et al*. presented design of pneumatic motor that uses four sequentially working paired cylinders, where at the end of expansion cycle under constant gas supply, smaller cylinder transfers gas to the larger cylinder where the gas expands. The energy efficiency improvement was not fully described in the paper [7].

Industrial application of the pneumatic systems has been around for a significant time and some design principles and traditional assumptions has been settled in [1,3]. *Emerson Electric Co* and its division *AVENTICS Hungary Kft* has been organising Pneumatic Vehicle competition in Eger, Hungary since 2008, which promotes out-of-the-box thinking on pneumatic actuator and control system application and design [8]. Unusual area of application demands for lightweight, compact and highly effective solutions. Use of dry, clean nitrogen gas allows control strategies that include high ratio of gas expansion, as the risk of water ice accumulation in the pneumatic system is reduced, comparing to use of compressed air.

Simon presented results of theoretical and experimental work, describing performance and efficiency of pneumatically driven vehicle. The level of detail of presented mathematical model is limited [9,10]. Szakacs developed and presented computer model of entire pneumatic system of pneumatic vehicle. Although the model appears to be highly sophisticated, governing equations, the results and wider conclusions were not included [11].

There is a lack of practical models that explain impact of geometric and control parameters on efficiency of pneumatic cylinder. Mathematical model, computer code and user interface of thermodynamically based model of pneumatic double acting cylinder has been developed. Computer codes for practical implementation using relatively popular commercial software *MATLAB* are presented.

2. MATERIALS AND METHODS

Gas energy conversion to mechanical work in double action pneumatic cylinder is modelled, excluding turbulence effects, friction and heat exchange. Pneumatic diagram of the proposed system is shown in Fig. 1.

Only process in the cylinder is modelled. Assumption is made that process in the cylinder is quasistatic, and calculations are performed by discrete steps. Theoretical pressure – volume diagram of the working cycle of the pneumatic cylinder is presented in Fig. 2. According to the model, gas is supplied at constant pressure up to the specific point in the piston travel, which is marked with index *3*. The gas supply is cut at the specific point to ensure that desired pressure is reached at the end of expansion, at the point marked as index *4*.

where *sgc* – specific gas consumption, g·kJ^{-1}.

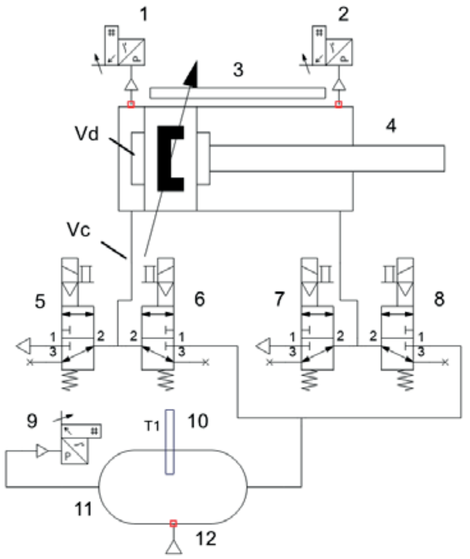

Figure 1. Diagram of the simulated pneumatic circuit

1, 2, 9 – pressure sensors; 3 – piston position sensor; 4 – pneumatic cylinder; 5, 7 – exhaust valves; 6, 8 – inlet valves; 10 – temperature sensor; 11 – gas buffer; 12 – gas inlet, V_d – cylinder displacement volume; V_c – volume of supply pipe, control valve and cylinder cushioning volume

Energy conversion efficiency is the ratio of work done to energy supplied:

$$\mathrm{ece} = \frac{\mathrm{W_w} - \mathrm{W_e}}{\mathrm{E}} = \frac{\mathrm{W_w} - \mathrm{W_e}}{\mathrm{U_{1\_2}} + \mathrm{H_{2\_3}}} \qquad (1)$$

where W_w – gas work on the piston, J; W_e – piston work on the gas at expulsion side, J; E – energy, J; U – internal energy, J; H – enthalpy, J.

Specific gas consumption is the ratio of gas mass supplied in the cycle to work done:

$$sgc = \frac{m_{1\_2} + m_{2\_3}}{W_w - W_e} \qquad (2)$$

Indexes (for instance U_{1_2}) in this and further equations correspond to the indexes presented in Fig. 2.

Supplied energy depends on type of gas, amount of gas and its temperature. Internal energy of the gas stored in the supply pipe and control valve (in the volume V_c) is calculated [12]:

$$U = c_v \cdot m \cdot T \qquad (3)$$

where c_v – specific heat capacity at constant volume, J kg^{-1} K^{-1}; m – mass of gas, kg; T – temperature, K.

When gas is supplied in constant pressure process, additional work on surroundings is performed, which is accounted in the enthalpy. Enthalpy of the gas in the displacement volume of pneumatic cylinder working side [12]:

$$H = c_p \cdot m \cdot T \qquad (4)$$

where c_p – specific heat capacity at constant pressure, J kg^{-1} K^{-1}.

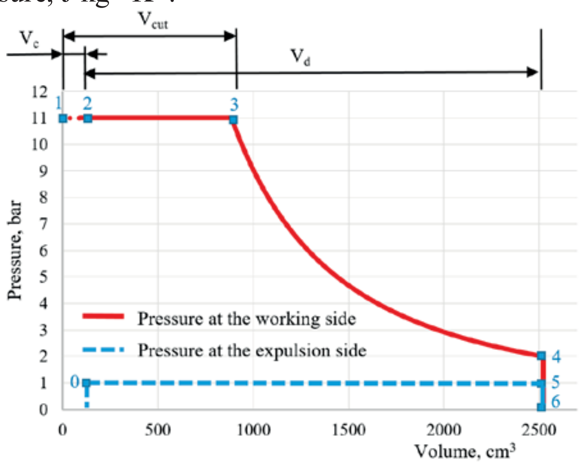

Figure 2. Pressure – volume diagram of the simulated pneumatic circuit. The numbers at the marked points represent index in the equations

V_d – cylinder displacement volume; V_c – volume of supply pipe, control valve and cylinder cushioning volume; V_{cut} – volume at which the gas supply is stopped

Mass of the gas in the supply pipe and control valve, and separately in cylinder up to the point where the gas supply is cut (index *3* in Fig. 2), is calculated according to ideal gas law [12]:

$$m = \frac{P \cdot V}{R \cdot T} \tag{5}$$

where P – pressure, Pa; V – volume, m^3; R – specific gas constant, J kg^{-1} K^{-1}.

Area of the piston at the top side is calculated[1]:

$$A_w = \frac{\pi \cdot B^2}{4} \tag{6}$$

where B – cylinder bore, m.

Area of the piston at the rod side is calculated[1]:

$$A_e = \frac{\pi \cdot (B^2 - b^2)}{4} \tag{7}$$

where b – rod diameter, m.

Cylinder volume at any piston position is calculated [13]:

$$V_i = V_c + A \cdot s_i \tag{8}$$

where i – calculation step; V_c – total volume of supply pipe, control valve and cylinder cushioning volume, m^3; A – area of piston, m^2; s – instantaneous piston position, m.

Volume V_{cut}, at which the gas supply is stopped to ensure desired gas pressure at the end of expansion on the working side, is calculated:

$$V_{cut} = \left(\frac{P_4 (V_c + V_d)^\gamma}{P_1}\right)^{\frac{1}{\gamma}} \tag{9}$$

where γ – ratio of heat capacities, ($\gamma = c_p/c_v$).

Ratio of heat capacities for nitrogen (N_2) at the temperature range, that is typical for pneumatic cylinder, is approximately γ = 1.4. The authors analysed the experimentally obtained data, and found that due to heat exchange between the gas and cylinder walls, apparent value is approximately γ = 1.16.

Changes in the pressure during gas expansion are calculated:

$$dP_{i+1} = \frac{\gamma \cdot P_i \cdot dV_{i+1}}{V_i} \tag{10}$$

where dV – change in cylinder volume, m^3.

Changes in cylinder volume dV are calculated by simple numeric differentiation. Pressure at the working side of the cylinder during expansion is then calculated by numerical integration:

$$P_w = P_3 + \int_{P3}^{P4} dP_i \ . \tag{11}$$

Pressure at the beginning on the expulsion side is close to the final pressure of the working side from the previous cycle, as the working and expulsion sides switch between cycles. Then expulsion side pressure reduces and eventually reaches value that is close to surrounding pressure. The pressure changes can be modelled using relations of pressure and volume in the isentropic process [14]:

$$P_{e\,i+1} = \frac{P_{e\,i} \cdot V_i^k}{V_{i+1}^k} \tag{12}$$

where k – constant; P_e – pressure at the expulsion side, Pa.

To imitate conditions of gas expansion at the expulsion side, value of constant can be set at approximately k=0.2...0.5 Limits to the pressure drop should be included in the model code, to keep expulsion side pressure above ambient pressure.

Work on the both sides of the piston is equal to the area under pressure – volume curve. It is calculated using following equation [13]:

$$W = \int_{V2}^{V4} P \cdot dV \tag{13}$$

The work on both piston sides is calculated using trapezoidal numerical integration and subtracted to find total work.

Instantaneous force on the piston rod is calculated using following equation:

$$F_i = A_w \cdot P_{w\,i} - A_e \cdot P_{e\,i} \tag{14}$$

where P_w – pressure at the working side, Pa.

The equations are implemented in the *MATLAB* code. The code can also be used in the open-source software environment, such as *Octave*. User interface is realised in *LabVIEW*, deploying *MATLAB* code in the *MathScript* module. Two instances of the simulation run continuously. For the separation of the workspace each instance of the main script and its functions are named distinctly. In that way two design and control scenarios can be compared simultaneously.

3. RESULTS

The user interface of computer simulation is shown in Figs 3 and 4. Several parameters are available for the user to control.

Geometrical parameters:

- Piston stroke;
- Cylinder bore;
- Length of supply pipe.

Gas parameters:

- Initial temperature;
- Initial pressure;

- Pressure at the end of expansion.
- Pressure difference at the end of expansion and beginning of expulsion;
- Isentropic constant at the expulsion side.

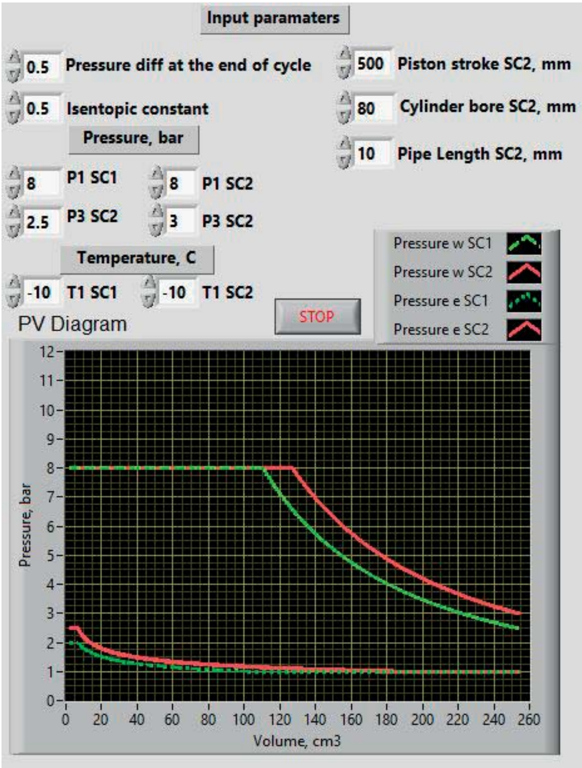

Figure 3. User interface of the model for entering of input parameters and observing pressure – volume diagram

Performance parameters are then calculated for two scenarios and displayed to the user:

- Pressure – volume diagram;
- Force – volume diagram;
- Specific gas consumption (*sgc*);
- Energy conversion efficiency (*ece*);
- Relative difference in *sgc* of two scenarios.

MATLAB code of the model is presented in the appendix of this paper. The analysis of the impact of various geometric and control parameters on the efficiency of pneumatic cylinder will be presented and discussed in the future work of the authors.

4. CONCLUSIONS

Developed code can be used in *MATLAB* or compatible open-source software environment *Octave*. User interface and continuous execution of two instances of the simulation are realised in *LabVIEW*. Effect of pneumatic cylinder bore, piston stroke and gas supply pipe volume on its work, force and energy conversion efficiency can be evaluated. Further, impact of gas temperature and cycle pressure parameters on pneumatic cylinder performance can be investigated. In the future development of this model, heat exchange, turbulence effects and also dynamics of moving parts should be accounted.

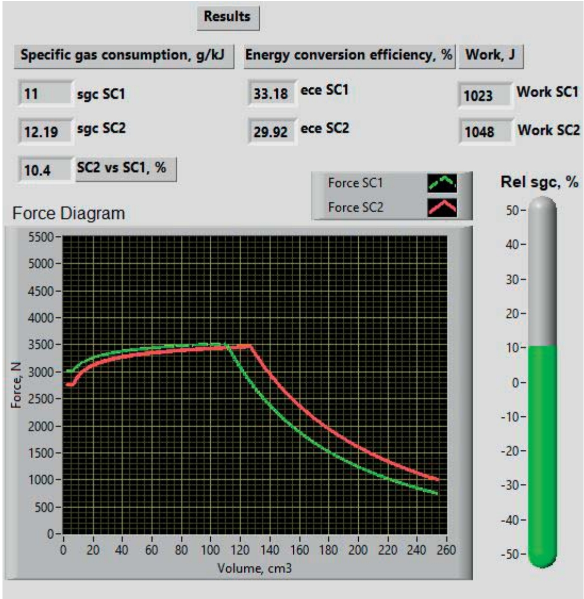

Figure 4. User interface of the model. Force – volume diagram and calculated efficiency parameters are shown

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APPENDIX

MATALB script consists of main part and three user-defined functions.

Main script

```
clear
T1=20+273.15; %Initial temperature, K
Pr=0.5; %Pressure difference at the end and beginning of the cycle
k=0.20; % Isentropic constant
P1=11; %Initial pressure, bar
P3=2; % Final pressure, bar
s=0.5; %Stroke m
B=0.08; %Bore, m
Lp=0.1; %Length of pipe, m
%calling external function to perform the calculation
[ece, Volume, Pressure_w, Work, sgc, Pressure_e, Force]= energy (T1, Pr, P1, P3, B, s, Lp, k);
```

Function ‘energy’

```
function [ece, VFi, Pressure_w, Work, Vd, sgc, Pressure_e, Force]=energy ( T1,Pr, P1, P3, B, s, Lp, k)
P1kPa=P1*100; %kPa
P1Pa=P1kPa*1000; %Pa
P3Pa=P3*10^5; %Pa
MolarMass=28.01348;%g/mol
Ru=8.314; %J/molK
Ri=Ru/(MolarMass/1000); %J/kgK
CalcSteps=1000;
Dp=16/1000; %diameter of supply pipe, m
%calling function to calculate volume
[ VFi,dVFi,Vd, Vc, A] = volume ( B, s, CalcSteps , Lp, Dp);
Smm=s*1000;
Smm_vect=linspace(0,Smm,CalcSteps);
cv=745.6739; %J/kgK
cp=cv+Ri; %J/kgK
Gamma=cp/cv; Vmax=Vd+Vc;
Vcut=((P3Pa*Vmax^Gamma)/P1Pa)^(1/Gamma);
CutOff_mm=round((((Vcut-Vc)*4)/(pi*B^2))*1000);
CutOff_CalcStep=min(find(Smm_vect>
CutOff_mm));%, 'first');
% Reallocation of vector size
P(1:CalcSteps)=P1kPa.*1000;
Pbar(1:CalcSteps)=P1kPa.*10^(-2);
Po1(1:CalcSteps)=Pr;
Po(1:CalcSteps)=Pr;
VFi(1:CalcSteps)=VFi;
dp(1:CalcSteps)= P1kPa*10^3;
Scut=CutOff_CalcStep;
mVc=(P1kPa*1000*Vc)/(Ri*T1); %kg
mVSupply=(P1kPa*1000*(Vcut-Vc))/(Ri*T1); %kg
UVc=mVc*T1*cv; %J
H=mVSupply*T1*cp; %J
USupply=UVc+H; %J
for ii=1:20
   Po1(ii)=P3-Pr;
   Po(ii)=Po1(ii);
end
for ii=20:(CalcSteps)
nn=ii-1;
Po1(ii)=Po(nn)*VFi(nn)^(k)/VFi(ii)^k;
Po(ii)=Po1(nn);
 if Po(ii)<1
    Po(ii)=1;
end
end
for ii=Scut:(CalcSteps-1)
Smm(ii+1)=Smm_vect(ii+1);
dp(ii+1)=((Gamma/VFi(ii))*P(ii)*dVFi(ii+1))*(-1);
P(ii+1)=P(ii)+dp(ii+1);
Pbar(ii+1)=P(ii)*10^(-5);
end
Pressure_e=Pbar;
Pressure_o=Po;
%calling function to calculate work
[Work_w ] = work (Pressure_w, dVFi);
[Work_e ] = work (Pressure_e, dVFi);
ece=(( Work_w - Work_e)/USupply)*100; %%
sgc=((mVSupply+mVc)*10^6)/( Work_w - Work_e);
% g/kJ
Work= Work_w - Work_e; %J
Force=A*Pressure*10^5-A*Pressure_o*10^5; %N
```

Function ‘volume’

```
function [ VFi,dVFi,Vd, Vc, A] = volume ( B, s, s_mm, Lp, Dp )
A=pi*(B/2)^2;
Vpipe=((pi*Dp^2)/4)*Lp; % Volume of supply pipe
Vc=Vpipe;
Vd=((pi*B^2)/4)*s; % Volume of displacement
S_travel=linspace(0.00,s,s_mm);
VFi=Vc+Ak*S_travel;
VFim=VFi([2:(s_mm) 1:1]);
dVFi=VFim - VFi;
```

Function ‘work’

```
function [W] = work ( Pressure, dVFi,)
P_Pa=Pressure*10^5;
PdV_Pa=P_Pa.*dVFi;
W=(trapz(PdV_Pa)); %work, J
```