Energy efficiency of pneumatic cylinder for pneumatic vehicle motor

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Abstract

Compressed gas is a relatively expensive source of energy. When compressed gas is used for propelling of pneumatically driven vehicle, efficient gas utilization is favoured.

The design and control strategy of pneumatic cylinder, with the emphasis on **effective energy conversion** is being discussed.

Mathematical model, results of computer simulation and experimental work are provided and discussed.

Experimental research is performed on the competition vehicle, equipped with pneumatically driven piston motor and instrumented.

Relative significance of various geometric and control parameters of pneumatic cylinder on efficiency of energy conversion is presented.

Materials and Methods

Theory and modelling

To develop relations between control parameters and efficiency of energy conversion in the pneumatic cylinder, a simplified theoretical model is developed. Graphical representation of the modelled pneumatic cylinder is shown in the Fig. 2. The numbers in the diagram correspond to index used for parameters in the equations.

Energy conversion efficacy

Energy conversion efficacy is the ratio of work done and energy supplied:

$$\eta_{th} = \frac{W}{E} = \frac{U_{1\_2} + H_{2\_3} - U_{4\_6}}{U_{1\_2} + H_{2\_3}} = 1 - \frac{U_{4\_6}}{U_{1\_2} + H_{2\_3}}$$

where W – work, J; E – energy, J, U – internal energy, J, H – enthalpy, J.

Applying well known formulas for internal energy, enthalpy, the expression can be rewritten:

$$\eta_{th} = 1 - \frac{c_v \cdot T_4 \cdot (n_{1\_2} + n_{2\_3})}{c_v \cdot T_1 \cdot n_{1\_2} + c_p \cdot T_1 \cdot n_{2\_3}}$$

where c_v – specific molar heat capacity at constant volume, J mol^{-1} K^{-1},

c_p – specific molar heat capacity at constant pressure, J mol^{-1} K^{-1} n – amount of gas, mol, T – temperature, K.

The ratio of expansion r_e is defined: $r_e = \frac{V_c + V_d}{V_{cut}}$

where V_c – total volume of supply pipe and control valve, m^3; V_d – displacement volume, m^3, V_{cut} – volume at which the gas supply is stopped, m^3.

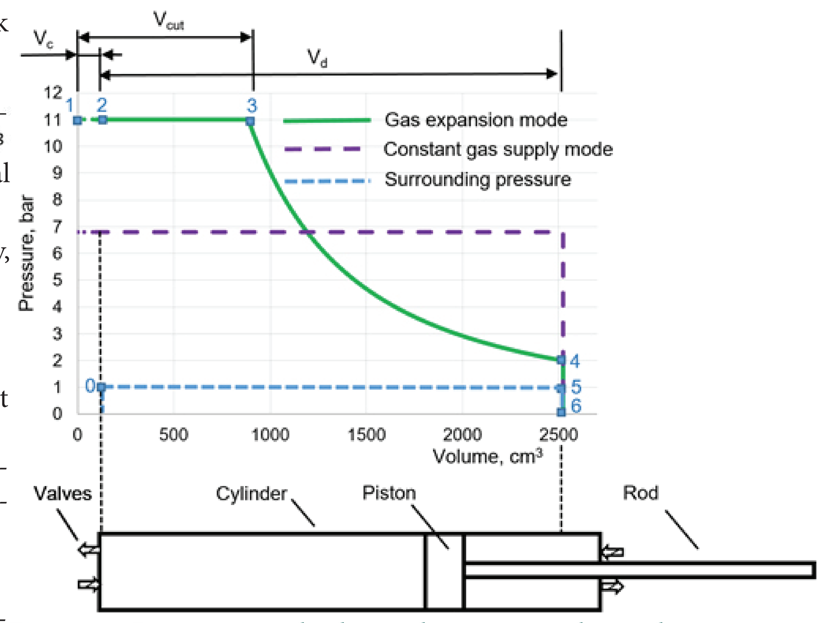

Figure 2. Pneumatic cylinder and pressure-volume diagram. V_c – volume of supply pipe and control valve; V_d – displacement volume, V_{cut} – volume at which gas supply is stopped.

Using the ratio of heat capacities $\gamma = c_p/c_v$, if initial pressure P_1 and desirable final pressure P_4 are known, V_{cut} can be found, basing on pressure and volume relations during isoenthropic and adiabatic process, where from point 3 to 4 in the Fig. 2, PVγ=const:

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

As the gas is supplied at constant temperature T_1 and constant pressure P_1 and R is a constant, we can re-write energy conversion efficacy:

$$\eta_{th} = 1 - \frac{{r_e}^{1-\gamma} \cdot (V_c + V_{cut} - V_c)}{V_c + \gamma \cdot (V_{cut} - V_c)} = 1 - \frac{{r_e}^{1-\gamma} \cdot V_{cut}}{V_c + \gamma \cdot (V_{cut} - V_c)}$$

In the case of practical application of pneumatic cylinder as energy conversion device for pneumatic vehicle, consumption (which corresponds to work done by certain amount of gas) and efficiency using specific type of gas might be subjects of interest. Also additional work to expel gas from the cylinder or perform work on surrounding should be accounted for. Specific work W_m can be used for that purpose and it is expressed as ratio of effective work done to molar quantity of gas:

$$W_m = \frac{T_1 \cdot R \cdot \left((V_c + \gamma \cdot (V_{cut} - V_c)) - {r_e}^{1-\gamma} \cdot V_{cut}\right)}{V_{cut} \cdot (\gamma - 1)} - \frac{p_0 \cdot V_d \cdot R \cdot T_1}{P_1 \cdot V_{cut}}$$

where p_0 – surrounding pressure, bar.

Computer modelling of energy efficacy and specific work

A computer model was developed in the software environment MATLAB. It is based on the equations that are presented in previous sub-chapter. The purpose of this computer simulation is to simplify and automatize input parameter sweep, speed-up calculation and present numerical and graphical results. The main script sets initial parameters:

- Gas type;
- Initial temperature;
- Initial pressure;
- Final pressure;
- Geometric parameters of the pneumatic cylinder and supply pipe;
- Number of simulations and variable parameter.

Any initial parameter can be set as variable, and its value then is gradually changed during the simulations.

Separate scripts contains code to calculate changes in cylinder volume, gas pressure, temperature, heat capacity and internal energy during one the course of piston movement from minimal to maximal cylinder volume. Calculation is performed at the resolution of 1000 points per cycle. As the heat capacity of the gas changes with the temperature, an external database is called at each calculation step to get thermodynamic parameters of the gas. Changes in gas pressure and temperature inside the cylinder during piston movement are calculated using following equations:

$$dP_{i+1} = \frac{\gamma_{i+1} \cdot P_i \cdot dV_{i+1}}{V_i} \qquad dT_{i+1} = \frac{T \cdot dV_{i+1}}{V_i} + \frac{T \cdot dP_{i+1}}{P_i}$$

Changes in pressure and temperature are then numerically integrated.

Geometric parameters of the simulated pneumatic cylinder are following: bore 80 mm; stroke 500 mm; volume of supply pipe V_c = 20 cm^3. Initial temperature of the gas was 20°C, initial pressure 11 bar and minimal pressure after expansion 2 bar. Three sweeps were simulated: 1. Gradual change of final pressure from 2 bar to 11 bar; 2. Gradual change of initial temperature from 20°C to -30°C; 3. Five-fold increase of supply valve and pipe volume V_c from 20 cm^3 to 100 cm^3 and gradual change of final pressure from 2 bar to 11 bar. Each sweep was simulated using Nitrogen and then repeated using Argon.

Experimental setup

The experimental work was performed on the instrumented pneumatic vehicle, designed according to the rules of International AVENTICS Pneumobile Competition. In this research the vehicle served as a realistic load for the pneumatic cylinder and the parameters of vehicle movement like acceleration and velocity were not analysed. Pressure in both chambers of the pneumatic cylinder and also temperature and pressure in the buffer tank were measured synchronously with piston position. The aim of experimental research was to validate the findings of mathematical modelling, particularly the effect of gas expansion on efficiency in real world conditions. Other reason for the experiments was to identify other significant factors, which affect efficiency of pneumatic cylinder and can be addressed in the future work. For this purpose mathematical modelling was performed assuming ideal gas behaviour, without heat exchange, turbulence and friction, and results compared with experimental ones. Analysis of the differences between modelling and experiment results were used to identify next research area.

Diagram of the experimental pneumatic circuit is shown in Fig. 3. Relative pressure sensors Aventics PE5 were installed in the buffer and connected to the cylinder heads by specially made holes. Absolute pressure sensor AC Delco 213-3205 was used to measure ambient air pressure, which was added to relative pressure. Aventics SM6-AL piston position sensor was used. Aventics PRA double acting cylinder with bore 80 mm, stroke 500 mm and diameter of rod 25 mm was used. Separate valves of type Aventics CD12 3/2 were used for inlet and exhaust gas supply control. Only Nitrogen was used as the working gas for experimental research. Use of Argon was dismissed, as the results of computer simulation showed, that work done by specific amount of gas will be smaller, comparing Argon to Nitrogen. The control and data acquisition system is shown on Fig. 4. It is based on National Instruments modular hardware, and programmed in LabVIEW. The experiments were performed on the test track, at ambient temperature around 22°C. Gas temperature in the buffer was approximately 18 – 21°C.

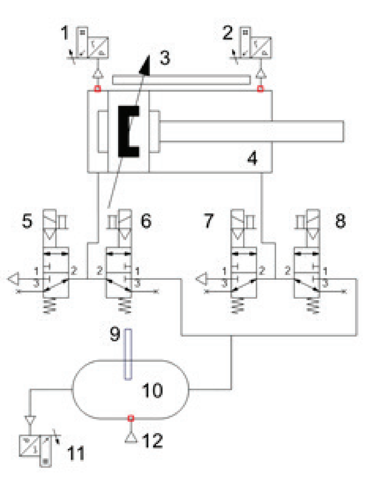

Figure 3. Pneumatic circuit diagram. 1, 2, 11 – pressure sensors; 3 – piston position sensor; 4 – pneumatic cylinder; 5, 7 – exhaust valves; 6, 8 – inlet valves; 9 – temperature sensor; 10 – gas buffer tank; 12 – gas inlet

Experimental data were acquired in piston movement domain, with a resolution approximately 1 mm.

An analogue piston position sensor was used, it led to some uncertainty.

Cylinder volume and its differential at each data point were calculated.

Work in the instroke and outstroke sides of the cylinder was calculated.

Force was calculated by taking into account difference in piston areas on both sides.

Instroke side work was subtracted from the outstroke side work to find effective work.

The same principle was used for calculating effective force.

Molar quantity of the supplied gas was calculated at each data point and then numerically integrated.

Mean piston speed was calculated using time data, which were recorded for each data point with resolution 1 µs.

Specific work was calculated.

All the calculations were performed in MATLAB.

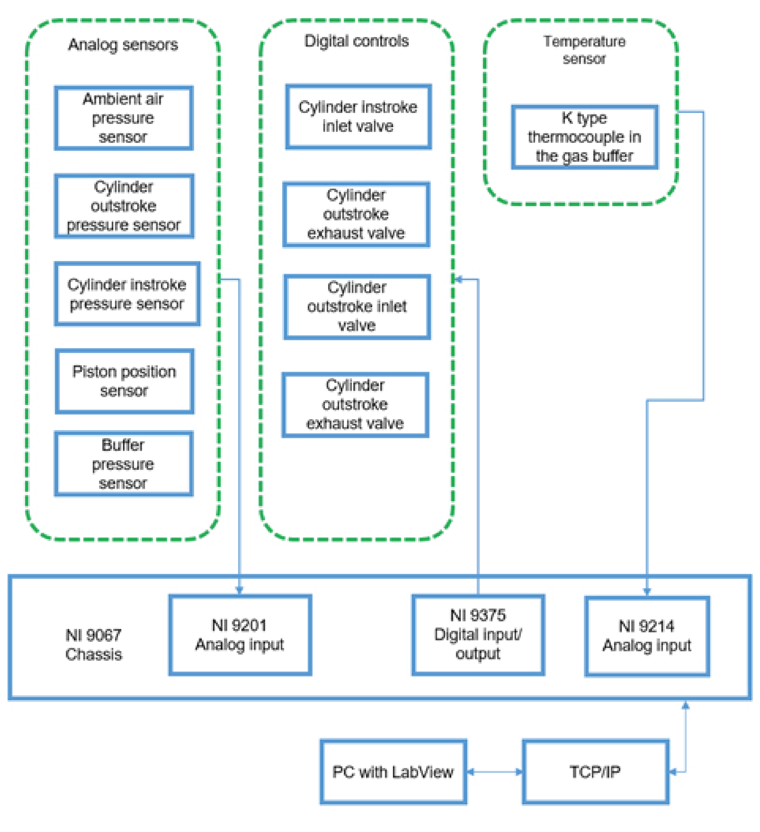

Figure 4. Layout of control and data acquisition system

Introduction

This research is focused on the improvement of the efficiency of pneumatic motor for pneumatic vehicle, built for participation in International AVENTICS Pneumobile Competition. The competition is organised by the company AVENTICS Hungary Ltd since 2008. The pneumatic vehicle used in this research is shown on Fig. 1. Recent results of this competition shows large variation of the results of different teams. Energy source and design rules are the same to all teams. Efficiency of mechanical and pneumatic systems, correct operation of control system along with level of team training and some luck makes the difference.

The subject of this research is the efficiency of double acting pneumatic cylinder, used in the gas expansion mode, in the conditions that are specific to a pneumatic vehicle. The novelty of the research consists of finding relative significance of gas type and temperature, geometric and control parameters on the efficiency of energy to work conversion. Another novelty of this paper is the use of normalised parameters, which are not dependent on cylinder size, absolute amount of work done or quantity of gas used. Such parameters are well known in other fields of research, but rarely used in relation with pneumatic actuators. Results of mathematical modelling, computer simulation and experimental research are presented and discussed.

Figure 1. Pneumatic vehicle during the competition, Eger, Hungary

Results and Discussion

Mathematical and computer modelling

Energy conversion efficacy η_{th}, increases when ratio of expansion r_e and ratio of heat capacities γ increase. Increase in additional volume V_c which includes supply pipe volume and control valve volume, decreases η_{th}. Those findings might be useful for efficacy considerations for general use of pneumatic cylinders. One may assume that use of gas which has higher γ, for instance, Argon instead of Nitrogen, can give an advantage and lead to higher work output per gas amount used in the cycle. This might not be true, as shown in Fig. 7.

In specific case of competition pneumatic vehicle the gas (Nitrogen N_2) is supplied in metal bottles which have an internal volume of 10 l and pressure of 200 bar at a temperature of 20°C. According to ideal gas law, independently to the gas type, molar quantity in equal thermodynamic conditions will be the same.

Specific work W_m follows the same tendency which applies to energy conversion efficiency η_{th}, except the ratio of heat capacities, γ. Ratio of heat capacities increases with decrease of heat capacity of the gas, as shown in equation 20. Specific work will increase with increase of initial gas temperature and heat capacity of the gas. Use of gas with diatomic molecules instead of monatomic gas, for instance, Nitrogen instead of Argon, will increase specific work and therefore the driving range of the pneumatic vehicle, assuming that gas bottle of similar volume and pressure is used. Use of heat exchanger and increase of inlet gas temperature will also increase specific work. Relative significance of the work on surrounding reduces by increase of initial gas pressure.

The pneumatic cylinder can be operated by continuously supplying it with gas at constant pressure, or by cutting the gas supply at some point of piston travel and letting the gas to expand. In practical application of pneumatic cylinder as part of pneumatic vehicle motor, maximal pressure at the beginning of the cycle is limited for safety reasons. The minimal pressure at the end of the cycle should be:

- Above atmospheric pressure of the surroundings;
- Sufficient to create force to overcome mechanical resistance.

Those two conditions ensure that the piston of the pneumatic cylinder finishes its course.

Effects of gas type and operational mode on gas amount per work done are shown in Fig. 5.

In the operating mode with gas expansion, certain pressure at the end of expansion is reached by cutting gas supply upon reaching certain cylinder volume. Mean effective pressure (MEP) and gas amount per cycle depends from gas type. Using gas with higher ratio of heat capacities (γ), Argon instead of Nitrogen, MEP is larger and amount of gas per cycle greater.

In the operating mode with constant gas supply in the cylinder, shown for comparison at the same MEP as for Nitrogen in expansion mode, gas amount per cycle is almost two times larger, comparing to Nitrogen in expansion mode, and does not depend on gas type.

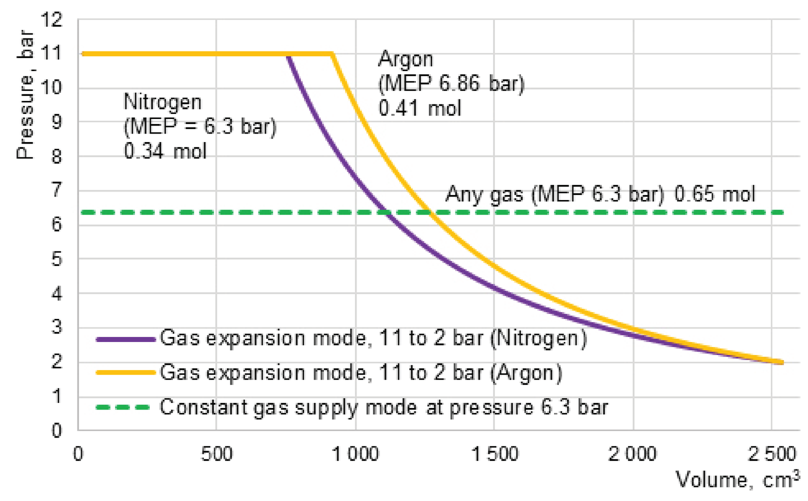

Figure 5. Effect of gas type and operational mode on gas amount per cycle

Drop of gas temperature during expansion is shown in Fig. 6, which explains higher energy conversion efficiency using expansion process instead of constant pressure process. Gas with lower heat capacity, Argon exhibits larger temperature reduction during expansion.

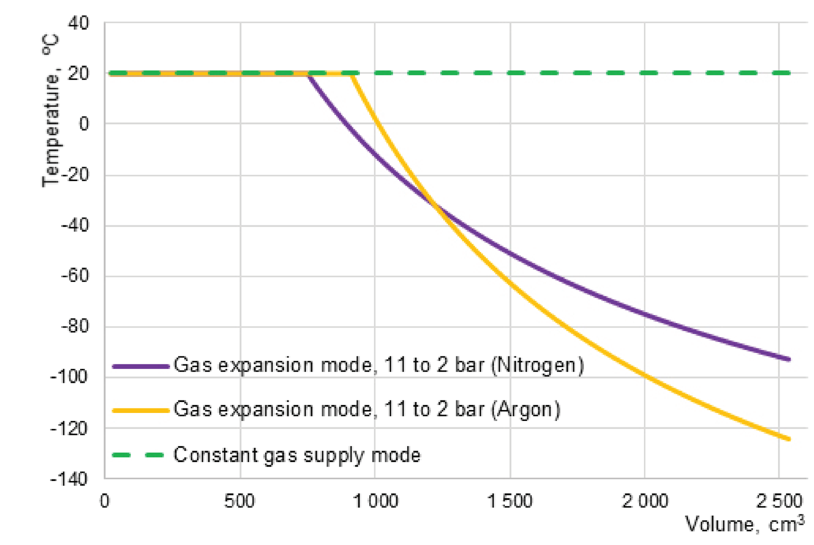

Figure 6. Effect of gas type and operational mode on gas temperature

Effects of gas type and expansion ratio on energy conversion efficiency and specific work are shown in Fig. 7. A practical parameter – difference of gas pressure in the beginning and end of the piston travel is used in the diagram instead of more theoretical parameter ratio of expansion (r_e). Simulation was performed at constant initial pressure P_1=1 bar, in 90 steps with gradual increase of cycle final pressure P_4 from 2 bar to 11 bar.

Further in the article mathematical and computer modelling section the effect of gas temperature increase on specific work and the effect of volume V_c increase on specific work are evaluated.

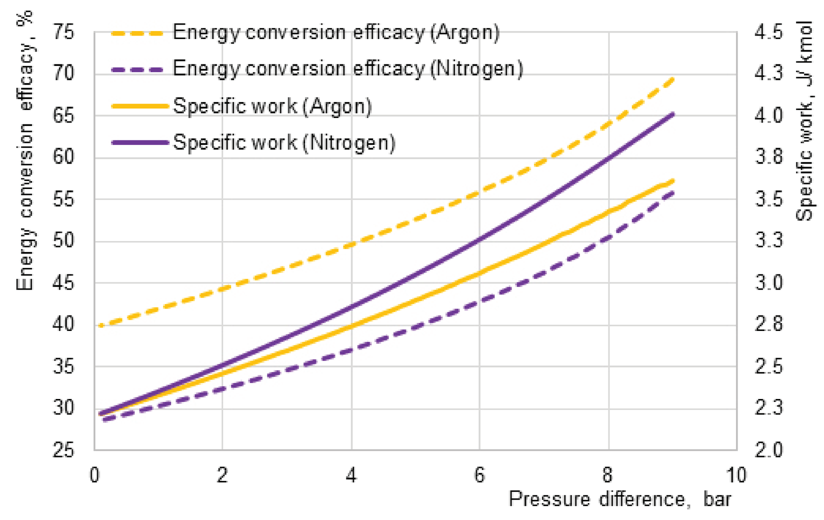

Figure 7. Effect of gas temperature increase by one degree on specific work depending on initial gas temperature

Experimental results

Experimentally obtained results show effects of different gas supply strategies on the efficiency of energy conversion into work. Only one extension cycle for each case was selected, as the data were acquired during the test drive, and each repeated cycle had slightly different piston speed and gas temperature. Pressure volume diagrams of the extending stroke (outstroke) are shown in Figs. 10. – 12.

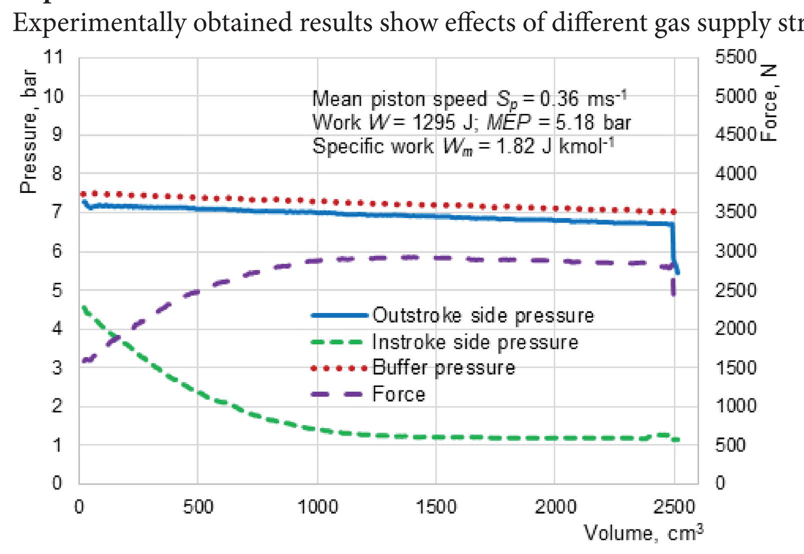

Figure 10. Pressure–volume diagram at uninterrupted gas supply mode, piston speed 0.36 ms^{-1}

By supplying the gas throughout the full stroke of the piston movement, in a traditional control mode, cylinder operation resembles the one modelled in the scenario of constant pressure, shown in Fig. 2. Pressure changes in both sides of the piston and the buffer during relatively slow piston speed, S_p = 0.36 m s^{-1}, are shown in Fig. 10. Restriction to the gas flow by control valves and pipes creates pressure difference of approximately 0.2 bar between buffer and pneumatic cylinder, which is relatively stable within the course of piston travel. In the chamber on the other side of the piston, gas flow restriction by exhaust valve creates gradual pressure drop until the pressure drops close to ambient pressure. Specific work during constant pressure operation, predicted by mathematical model according to equation 22 and the computer model, shown in Fig. 7, should be 2.21 J kmol^{-1}. Experimentally obtained result for specific work is 1.82 J kmol^{-1}, which is smaller than predicted. That can be explained by the increased pressure on the other side of the piston during initial part of the piston travel. Force on the rod increases gradually, then stabilises for the rest of the piston travel.

Pressure changes during similar operation mode, with uninterrupted gas supply, are shown in Fig. 11. Mean piston speed was almost two time larger, S_p = 0.79 m s^{-1}, comparing to case with the slower piston speed. The difference in cylinder outstroke side pressure and buffer pressure is approximately 1.2 bar, which is significant increase comparing to case of the lower piston speed. Restriction, created by exhaust valve or pipe on the instroke side, leads to slow, almost linear drop of the gas pressure on instroke side during whole cycle. Instroke side pressure reaches ambient pressure only at the very end of cycle, when piston has stopped. Outstroke side pressure diagram is similar to the one in case of slower piston speed, showed in Fig. 10. Considering this, gas amount per cycle in both cases is similar too. Force diagram is in significantly different shape.

Indicated (calculated from pressure and volume changes) and specific work are reduced approximately by 20%, comparing to slower piston speed case. Sudden increase of instroke side pressure at the end of the cycle probably is caused by pneumatic piston cushioning.

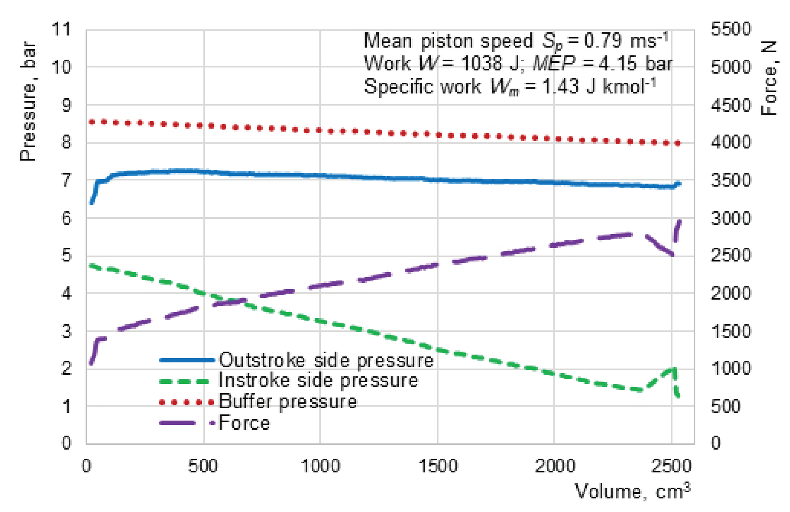

Figure 11. Pressure–volume diagram at uninterrupted gas supply mode, piston speed 0.79 ms^{-1}

Optimisation of pneumatic cylinder operation for industrial purposes significantly differs from objectives set for pneumatic cylinder as the part of pneumatic competition vehicle motor. Linearity of piston movement and longitude stiffness of pneumatic cylinder plays more significant role in industrial applications comparing to competition vehicle.

Pressure in the buffer, cylinder and force on the piston during gas expansion mode are presented in Fig. 12. In this operating mode gas supply is cut at certain point of piston travel, and pressure in the buffer is returning to the initial value, as gas is resupplied to the buffer from the pressure bottle. Pressure on the instroke side is 2 bar at the beginning of the cycle and quickly stabilises at the value which close to ambient pressure. Force diagram thus mostly follows the value of outstroke pressure. Specific work is 3.58 J kmol^{-1}, which shows a more efficient operation when, compared to the previous cases of interrupted gas supply without expansion.

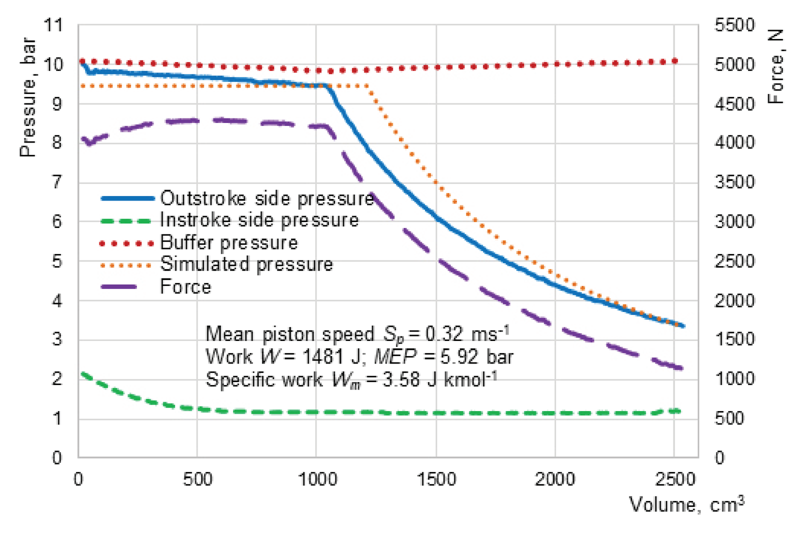

Figure 12. Pressure–volume diagram at gas expansion mode

The value of specific work, predicted by theoretical model for such geometry, initial pressure, pressure difference and gas type is 3.38 J kmol^{-1}. It is unusual to have theoretical efficiency smaller then experimentally obtained, but this effect will be explained. Computer simulation of gas pressure with similar pressure at the beginning and end of the gas expansion process shows different V_{cut} value and path of pressure changes. Pressure – volume diagram of gas expansion part in logarithmic scale (logP logV) is shown in Fig. 13. The term of the slope in the equation of linear regression corresponds to apparent ratio of heat capacities. In case of simulated data ratio of heat capacities is as expected for the gas type (Nitrogen) in the given temperature range, γ = 1.403. For experimental data its value is γ = 1.158. As the gas type in both cases is the same, difference in the pressure changes can be explained by effect of heat transfer, which is not included in computer simulation. As the gas temperature decreases during the expansion, difference in temperature and heat transfer intensity between cylinder wall and gas increases. Relative significance of heat transfer increases by increase of ratio of expansion.

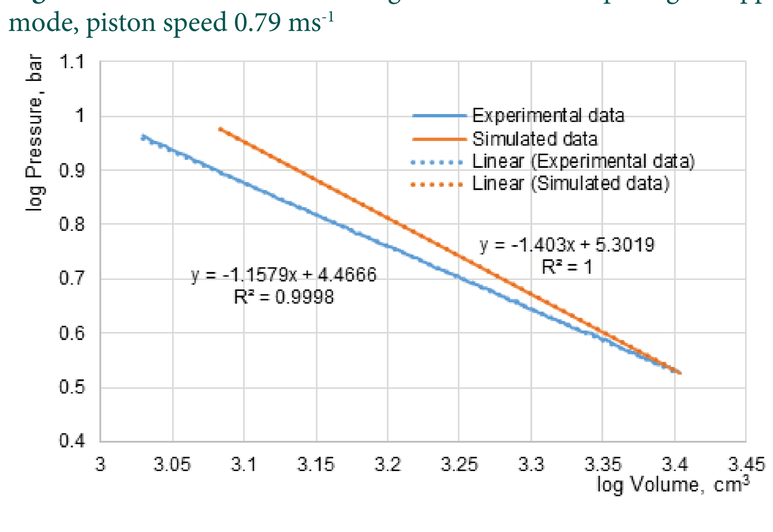

Figure 13. Pressure–volume diagram in logarithmic scale

Conclusions

1. A mathematical model was developed for assessment of effect of gas type, geometric and control parameters on pneumatic motor energy conversion efficacy. Efficacy increases when:
   - Ratio of expansion r_e increases;
   - Ratio of gas heat capacities γ increases.
2. Another mathematical model was developed to evaluate effect of pneumatic motor control parameters on specific work, which is the work that can be done by certain amount of gas.
   - Specific work can be increased by using a gas with larger heat capacity. In practical terms it means that Nitrogen is a suitable choice.
   - Increase of gas temperature by 10°C at the same supply pressure will increase specific work by approximately 3.8%
   - Geometric parameters of the pneumatic cylinder, such as bore and stroke do not have direct effect on specific work of pneumatic cylinder.
   - Relative size of additional volume Vc, connected to pneumatic cylinder inlet, such as volume of gas supply pipe and inlet control valve have mild effect on specific work. For practical pneumatic motor fivefold increase of additional volume can decrease specific work by approximately 2.9%.
   - Specific work will increase by approximately 80% by increasing the rate of expansion r_e from 1 to 3.38 at initial pressure P_1 = 11 bar, which corresponds to range of pressure at the end of the cycle P_4 from 11 to 2 bar.
3. In continuous gas supply mode experimentally obtained specific work was lower than predicted by computer simulation using ideal gas model. Specific work is reduced by additional work to expel gas from cylinder on the other side of the piston. This additional work depends on final gas pressure of the previous cycle, piston speed and gas flow restriction by output hole, pipe and valve.
   - In the case of relatively slow piston speed, S_p = 0.36 m s^{-1}, experimentally obtained specific work was by 17.6% lower than predicted.
   - In the case piston speed S_p = 0.79 m s-1, relative difference between measured and predicted specific work was 35.3%.
   - The differences in modelled and experimental results demonstrate strong effect of restrictions to the gas flow on the efficiency of a pneumatic motor.
4. In gas expanding mode experimentally obtained specific work was by 5.9% higher than predicted by computer simulation of ideal gas model. It can be explained by cooling of the gas during expansion, which leads to increased intensity and significance of heat transfer between cylinder walls and gas.
5. Low gas pressure at the end of cycle as the result of gas expansion leads to reduced work to expel gas from the cylinder on the other side of the piston.

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