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Research on energy efficiency of pneumatic cylinder for pneumatic vehicle motor

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Abstract. Compressed gas is relatively expensive source of energy. When compressed gas is used for propelling of pneumatically driven vehicle, efficient gas utilization is favoured. Design and control strategy of pneumatic cylinder, with the emphasis on effective energy conversion is being discussed in this paper. 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.

Key words: gas consumption, compressed air vehicle, pneumobile, gas expansion, efficacy.

INTRODUCTION

Pneumatic actuators are important part of industrial machinery. Pneumatic cylinders are typically used to move or hold tools or products. Pneumatic cylinders include wide variety of linear actuators, such as spring load single acting, double acting and rodless cylinders (Cambell, 2018).

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 company AVENTICS Hungary Ltd since 2008 (Aventics Hungary Kft, 2019). 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 (Emerson Electric Co, 2019). Energy source and design rules are similar 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.

For typical industrial application one chamber of double acting pneumatic cylinder is continuously supplied with air at certain pressure until the piston reaches the end position. At the end of piston's course air in the chamber is at high pressure and it is then released to the surroundings. To control piston and rod speed, air flow from the chamber on the other side of the piston can be throttled to maintain certain pressure. Such control strategy provide relatively even motion profile (Barber, 1997).

In the industrial applications actuators usually are the last elements in the large and complex system that also contains air filters, compressors, dehumidifiers, distribution pipes and control valves. Air compressor efficiency and losses in the pipelines are discussed in technical information sources (Barber, 1997; SMC Pneumatics, 1997). The efficiency of pneumatic actuators is usually left out of the technical documentation. Energy efficiency of pneumatic actuators is approximately 20–30%, thus affecting overall energy efficiency of compressed air systems(Cai et al., 2001; Wang & Gordon, 2011; Luo et al., 2013; Du et al., 2018). Energy efficiency of the pneumatic cylinders and its improvement is analysed in the previous work by research community.

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

Fleisher developed concept of dual pressurisation that is based on supplying dual action cylinder working side chambers with different pressure during expansion and retraction. In case, when equal force during expansion and retraction is needed, pressure is adjusted to compensate difference in piston surface area. In case when opposite piston movement is needed for positioning only, little force is required and pressure can be significantly reduced. Dual pressurisation can considerably reduce air consumption (Fleischer, 1995; Harris et al., 2012).

Reutilisation of air which otherwise would be released to surroundings at the end of working cycle is proposed by several researchers. Luo developed energy recovery system that utilises scroll expander that converts mechanical energy to electrical, increasing efficiency by 18.1% (Luo et al., 2013). Li et al. proposed use of recovery tank to collect air from cylinder exhaust. The boost valve then is used to increase air pressure and resupply high pressure supply circuit. They reported recovery of up to 40% of air amount (Li et al., 2006). Shen and Goldfarb presented energy saving approach which is based on additional control valve to enable air flow between cylinder chambers or so called crossflow. They reached energy savings up to 25–52% (Shen & Goldfarb, 2007). Létai et al. (2017) presented a design of pneumatic vehicle with four cylinder pneumatic motor. The engine uses sequentially working paired cylinders, where smaller cylinder works in constant gas supply mode at 10 bar pressure, and the gas then is transferred to a larger cylinder where it expands. Proposal of Letai et al. combines gas reutilization and expansion. The authors claim that in this way the energy is reused. Theoretical analysis is not fully developed and presented (Létai et al., 2017).

Doll et al. proposed and discussed pneumatic cylinder control system, in which air supply is cut at certain point of piston travel and expanded until the end of the stroke. They achieved up to 85% savings in air consumption, comparing to regular pneumatic cylinder control system (Doll et al., 2011). Du et al. proposed and experimentally validated bridge-type control system of pneumatic cylinder, in which gas expansion is used and piston motion is controlled. Improvement in energy efficiency by 50–70% where achieved (Du et al., 2018). Harris et al. explored energy efficiency of pneumatic cylinder for vertical load lifting application, using computer model and experimental setup. They compared traditional, dual pressurising and gas expansion control circuits. With dual pressurising they achieved 27% and with gas expansion 29% in savings of compressed air consumption (Harris et al., 2014).

Analysis of previous work has shown, that gas expansion can provide greater efficiency increase, comparing to other known control strategies.

Use of pneumatic cylinder as part of pneumatic vehicle motor has some differences comparing to industrial application. For instance, other gas type can be used, instead of air, and at higher pressure. Furthermore, gas temperature can be increased, using heat exchanger. Equalized motion profile and piston speed limits are non–essential.

The subject of this research is an efficiency of double acting pneumatic cylinder, used in the gas expansion mode, in the conditions that are specific to 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 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 another fields of research, but rarely used in relation with pneumatic actuators. Results of mathematical modelling, computer simulation and experimental research are presented and discussed.

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.

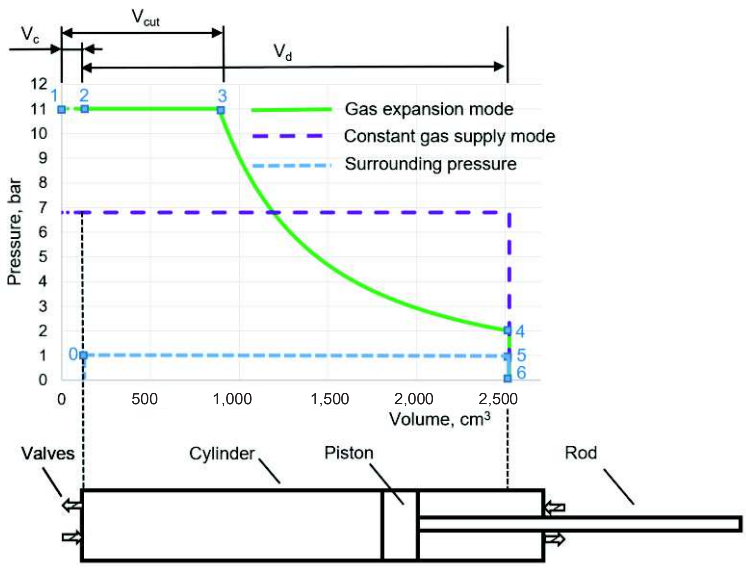

Figure 2. Pneumatic cylinder and pressure-volume diagram. V_c – volume of supply pipe and control valve; V_d – displacement volume; V_{cut} – volume at which gas supply is stopped.

Heat exchange, turbulence effects and friction are not included. Effect of temperature on heat capacity is not accounted.

Energy conversion efficacy

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

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

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

Internal energy can be expressed (Serway & Kirkpatrick, 2014):

$$U = c_v \cdot n \cdot T \tag{2}$$

where c_v – specific molar heat capacity at constant volume, J mol^{-1} K^{-1}; n – amount of gas, mol; T – temperature, K.

Enthalpy can be expressed (Serway & Kirkpatrick, 2014):

$$H = c_p \cdot n \cdot T \tag{3}$$

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

Expression 1 can be rewritten:

$$\eta_{th} = 1 - \frac{c_v \cdot T_4 \cdot (n_{1\_2} + n_{2\_3})}{c_v \cdot T_1 \cdot n_{1\_2} + c_p \cdot T_1 \cdot n_{2\_3}} \tag{4}$$

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

Ratio of heat capacities (Caton, 2016):

$$\gamma = \frac{c_p}{c_v} \tag{5}$$

Ratio of expansion is defined:

$$r_e = \frac{V_c + V_d}{V_{cut}} \tag{6}$$

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.

If initial pressure P_1 and desirable final pressure P_4 is 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, $PV^{\gamma} = const$:

$$V_{cut} = \left(\frac{P_4(V_c + V_d)^{\gamma}}{P_1}\right)^{\frac{1}{\gamma}} \tag{7}$$

where P – pressure, Pa.

Using isentropic relations, stating that from point 3 to 4 in the Fig. 2, $TV^{\gamma-1} = const$ temperature at the end of expansion can calculated (Heywood, 2018):

$$T_4 = T_1 \cdot r_e^{\,1-\gamma} \tag{8}$$

Expression (4) now can be rewritten with temperature T_1 cancelled out:

$$\eta_{th} = 1 - \frac{c_v \cdot r_e^{1-\gamma} \cdot (n_{1\_2} + n_{2\_3})}{c_v \cdot n_{1\_2} + c_p \cdot n_{2\_3}} \quad (9)$$

Heat capacities can be replaced by ratio of heat capacities according to the Eq. 5:

$$\eta_{th} = 1 - \frac{r_e^{1-\gamma} \cdot (n_{1\_2} + n_{2\_3})}{n_{1\_2} + \gamma \cdot n_{2\_3}} \quad (10)$$

According to ideal gas law, molar quantity of gas can be expressed:

$$n = \frac{P \cdot V}{R \cdot T} \quad (11)$$

where R – universal gas constant, 8.314 J mol^{-1} K^{-1}.

As the gas is supplied at constant temperature T_1 and constant pressure P_1 and R is a constant, they cancel out when Eq. 11 is used to replace molar quantity n in the Eq 10:

$$\eta_{th} = 1 - \frac{r_e^{1-\gamma} \cdot (V_c + V_{cut} - V_c)}{V_c + \gamma \cdot (V_{cut} - V_c)} = 1 - \frac{r_e^{1-\gamma} \cdot V_{cut}}{V_c + \gamma \cdot (V_{cut} - V_c)} \quad (12)$$

In 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 can be used for that purpose and it is expressed as ratio of effective work done to molar quantity of gas:

$$W_m = \frac{W_g - W_s}{n} = \frac{U_{1\_2} + H_{2\_3} - U_{4\_6} - W_s}{n_{1\_2} + n_{2\_3}} \quad (13)$$

where W_m – specific work, J mol^{-1}; W_g – work done by gas, J; W_s – work done on surroundings, J.

Combining Eq. 13 with Eqs 2 and 3:

$$W_m = \frac{n_{1\_2} \cdot T_1 \cdot c_v + n_{2\_3} \cdot T_1 \cdot c_p - c_v \cdot T_4 \cdot (n_{1\_2} + n_{2\_3}) - W_s}{n_{1\_2} + n_{2\_3}} \quad (14)$$

Stating T_4 according to the Eq. 8:

$$W_m = \frac{n_{1\_2} \cdot T_1 \cdot c_v + n_{2\_3} \cdot T_1 \cdot c_p - c_v \cdot T_1 \cdot r_e^{1-\gamma} \cdot (n_{1\_2} + n_{2\_3}) - W_s}{n_{1\_2} + n_{2\_3}} \quad (15)$$

Expressing n according to the Eq. 11 and separating work on surroundings:

$$W_m = \frac{\frac{P_1 \cdot V_c}{R \cdot T_1} \cdot T_1 \cdot c_v + \frac{P_1 \cdot (V_{cut} - V_c)}{R \cdot T_1} \cdot T_1 \cdot c_p - c_v \cdot T_1 \cdot r_e^{1-\gamma} \cdot \frac{P_1 (V_c + V_{cut} - V_c)}{R \cdot T_1}}{\frac{P_1 (V_c + V_{cut} - V_c)}{R \cdot T_1}} - \frac{W_s}{\frac{P_1 (V_c + V_{cut} - V_c)}{R \cdot T_1}} \quad (16)$$

Cancelling out P, T and R and rearranging work on surroundings:

$$W_m = \frac{V_c \cdot c_v + (V_{cut} - V_c) \cdot c_p - c_v \cdot r_e^{1-\gamma} \cdot V_{cut}}{\frac{V_{cut}}{T_1}} - \frac{W_s \cdot R \cdot T_1}{P_1 \cdot V_{cut}} \quad (17)$$

Multiplying and dividing the numerator of the first member of equation by c_v:

$$W_m = \frac{\frac{\frac{c_v \cdot \left(V_c \cdot c_v + (V_{cut} - V_c) \cdot c_p - c_v \cdot {r_e}^{1-\gamma} \cdot V_{cut}\right)}{c_v}}{\frac{V_{cut}}{T_1}}}{} - \frac{W_s \cdot R \cdot T_1}{P_1 \cdot V_{cut}} \tag{18}$$

Applying Eq. 5 and rearranging:

$$W_m = \frac{T_1 \cdot c_v \cdot \left((V_c + \gamma \cdot (V_{cut} - V_c)) - {r_e}^{1-\gamma} \cdot V_{cut}\right)}{V_{cut}} - \frac{W_s \cdot R \cdot T_1}{P_1 \cdot V_{cut}} \tag{19}$$

Specific molar heat capacity at constant pressure can be expressed (Caton, 2016):

$$c_p = R + c_v \tag{20}$$

Inserting Eq. 20 in the Eq. 5 and simplifying:

$$c_v = \frac{R}{\gamma - 1} \tag{21}$$

Replacing c_v in Eq. 19 by Eq. 21 and expressing work on surroundings according to the Eq. 25:

$$W_m = \frac{T_1 \cdot R \cdot \left((V_c + \gamma \cdot (V_{cut} - V_c)) - {r_e}^{1-\gamma} \cdot V_{cut}\right)}{V_{cut} \cdot (\gamma - 1)} - \frac{p_0 \cdot V_d \cdot R \cdot T_1}{P_1 \cdot V_{cut}} \tag{22}$$

where p_0 – surrounding pressure, bar.

Computer modelling of energy efficacy and specific work

A computer model was developed in 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 selected number of 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 1,000 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 (Lemmon et al., 2018). Changes in gas pressure inside the cylinder during expansion are calculated using following equation:

$$dP_{i+1} = \frac{\gamma_{i+1} \cdot P_i \cdot dV_{i+1}}{V_i} \tag{23}$$

where i – calculation step.

Temperature change:

$$dT_{i+1} = \frac{T \cdot dV_{i+1}}{V_i} + \frac{T \cdot dP_{i+1}}{P_i} \tag{24}$$

Changes in pressure and temperature are then numerically integrated. Volume, at which the gas supply is cut, to satisfy the requirement of initial and final pressure, is calculated, using Eq. 7. Energy conversion efficiency and specific work are calculated using correspondingly Eqs 1 and 13. For the control of the results, work is also calculated using Eq. 25 (Caton, 2016).

$$W = \int_{Vmin}^{Vmax} P \cdot dV \tag{25}$$

Mean effective pressure is used to measure work output of pneumatic cylinder, regardless of its size (Stone, 1999):

$$MEP = \frac{W}{V_d} \tag{26}$$

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:

- Gradual change of final pressure from 2 bar to 11 bar;
- Gradual change of initial temperature from 20 °C to -30 °C;
- 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 (Aventics Hungary Kft, 2019). In this research vehicle served as realistic load for pneumatic cylinder and the parameters of vehicle movement like acceleration and velocity were not analysed. Pressure in the both chambers of the pneumatic cylinder and also temperature and pressure in the buffer tank was measured synchronously with piston position. The aim of experimental research was to validate findings of mathematical modelling, particularly 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.

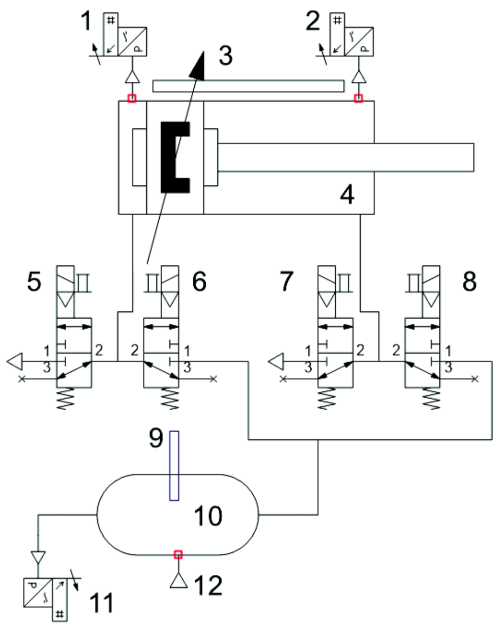

Figure 3. Pneumatic circuit diagram. 1, 2, 11 – pressure sensors; 3 – piston position sensor; 4 – pneumatic cylinder; 5, 7 – exhaust valves; 6, 8 – inlet valves; 9 – temperature sensor; 10 – gas buffer tank; 12 – gas inlet.

Experimental data were acquired in piston movement domain, with resolution approximately 1 mm. As 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, using Eq. 25. Force was calculated by taking into account difference in piston areas on both sides. Instroke side work was subtracted from outstroke side work to find effective work. The same principle was used for calculating effective force. Molar quantity of the supplied gas was calculated using Eq. 11 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 by applying Eq. 13. All the calculations were performed in *MATLAB.* Force on the rod was calculated according to following equation:

$$F_k = A_o \cdot P_{o\,k} - A_n \cdot P_{n\,k} \tag{27}$$

where A_o – piston area at outstroke side, m^2; P_o – pressure at outstroke side, Pa; A_n – piston area at instroke side, m^2; P_n – pressure at instroke side, Pa; k – data point.

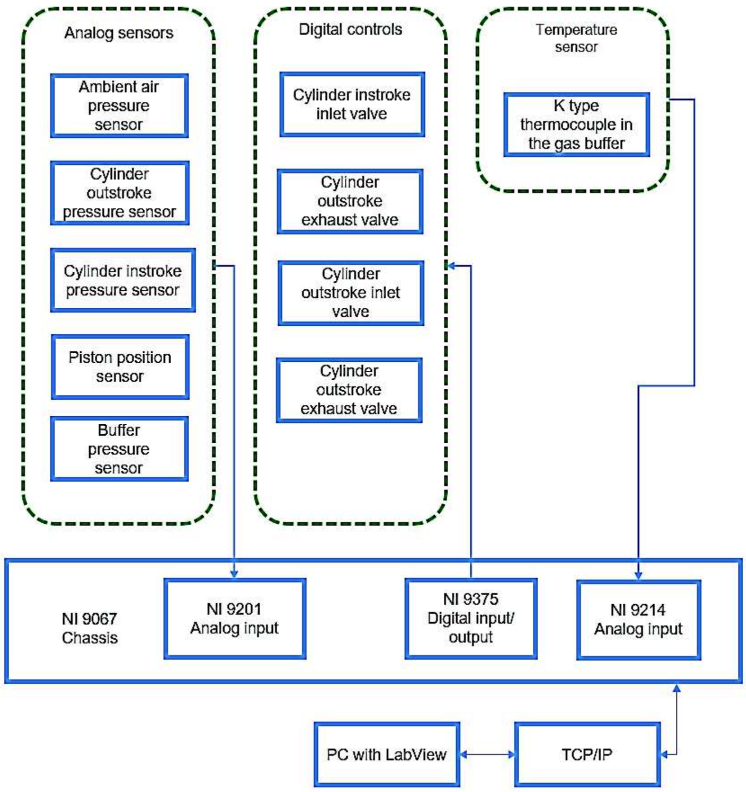

Figure 4. Layout of control and data acquisition system.

RESULTS AND DISCUSSION

Mathematical and computer modelling

Energy conversion efficacy η_{th}, calculated according to Eq. 12, 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 a internal volume of 10 l and pressure of 200 bar at a temperature of 20 °C. According to ideal gas law, show in Eq. 11, independently to the gas type, molar quantity in equal thermodynamic conditions will be the same.

Specific work W_m, calculated according to Eq. 22, 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 Eq. 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, calculated by Eq. 7. 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.

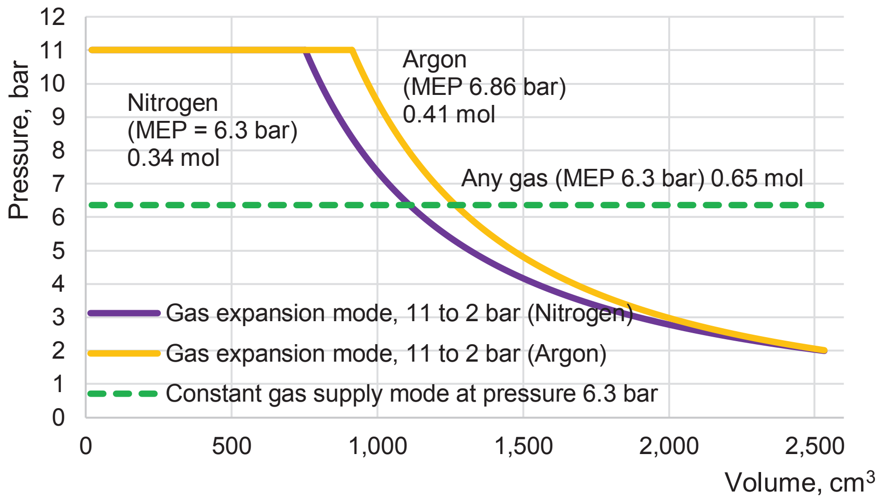

Figure 5. Effect of gas type and operational mode on gas amount per cycle.

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.

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.

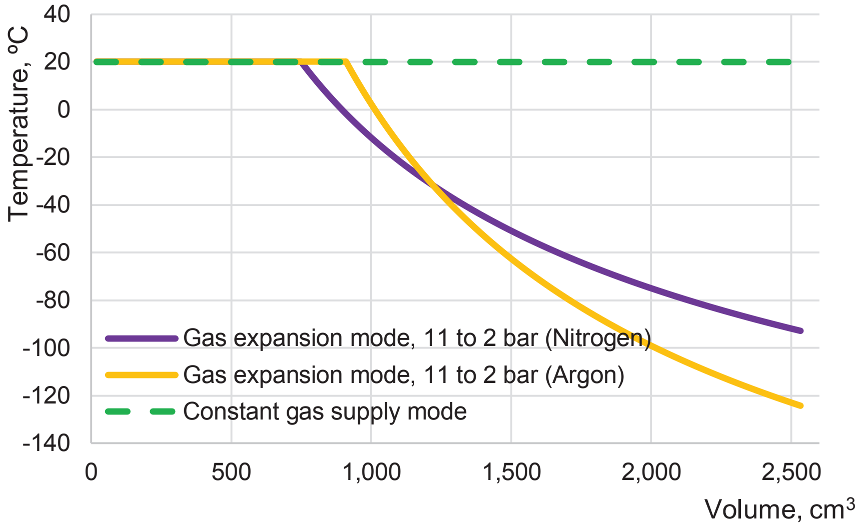

Figure 6. Effect of gas type and operational mode on gas temperature.

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 = 11 bar, in 90 steps with gradual increase of cycle final pressure P_4 from 2 bar to 11 bar.

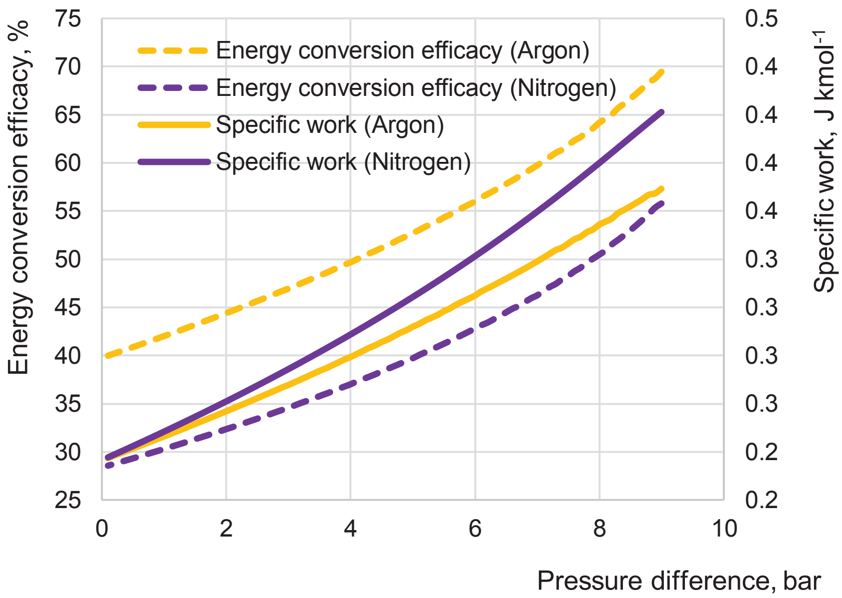

Figure 7. Effect of gas type and expansion on energy conversion efficacy and specific work, initial pressure 11 bar.

As the ratio of expansion or pressure difference increases, energy conversion efficacy and specific work increases. Differences in specific work between gases of different heat capacity increases with increase of ratio of expansion and therefore result in pressure difference. Technical information on pneumatic systems suggest reduction of air pressure to reduce consumption, which is opposite to findings of this research (Li et al., 2006; SMC Pneumatics, 2019). Reduction of pressure for pneumatics might reduce energy consumption of industrial plant, where pneumatic cylinders are operated in constant air supply mode. When gas is stored in the bottle at relatively high pressure (for instance, 200 bar), reduced to safe level for pneumatic components (6 – 11 bar) and cylinder operated in expansion mode, maximising pressure difference between beginning and end expansion will increase efficacy which will also lead to reduce of gas consumption.

Gas temperature is discussed in technical literature concerning water condensation and permissible conditions for polymeric materials (Barber, 1997). Saidur et al. shows that air temperature in the compressor inlet should be reduced to increase its volumetric efficiency (Saidur et al., 2010). To the author's knowledge, there is a little or no discussion in research community on the subject of gas temperature impact on efficiency of pneumatic cylinder. Gas temperature at the high-pressure bottle reducer's output usually is lower than ambient temperature. Heat exchangers can be used to warm up the gas. As the maximal pressure in the pneumatic motor feed circuit is limited at certain value, increase of temperature will lead to reduction of gas density. Effect of temperature increase by one degree on relative changes in specific work is shown in Fig. 8. The effect does not depend on type of gas, ratio of expansion or initial and final pressure difference. The effect depends on initial gas temperature, which can be explained by reduction of gas density.

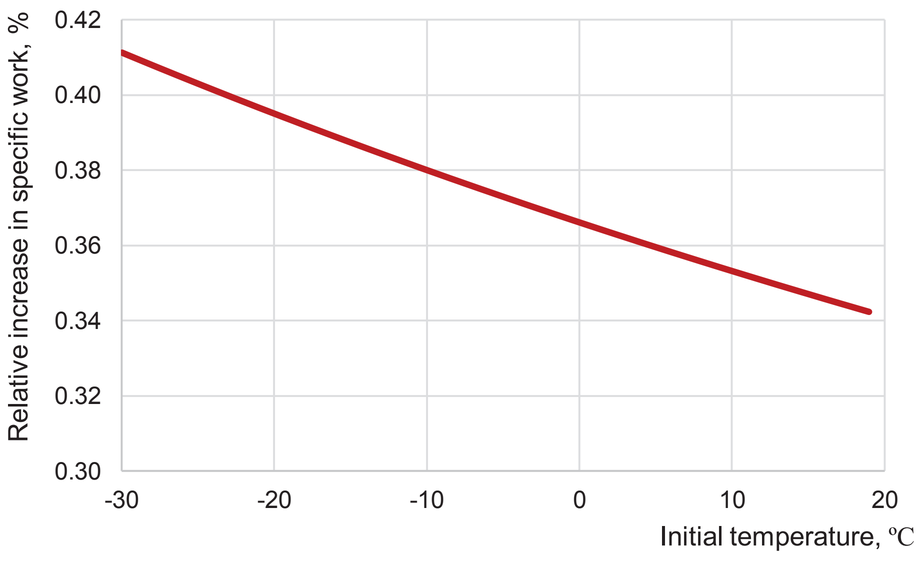

Figure 8. Effect of gas temperature increase by one degree on specific work depending on initial gas temperature.

Increase of V_c reduces specific work and this effect depends mainly on ratio of V_c to V_d. The reduction in specific work is more pronounced when ratio of expansion or initial and final pressure difference increases, as shown in Fig. 9.

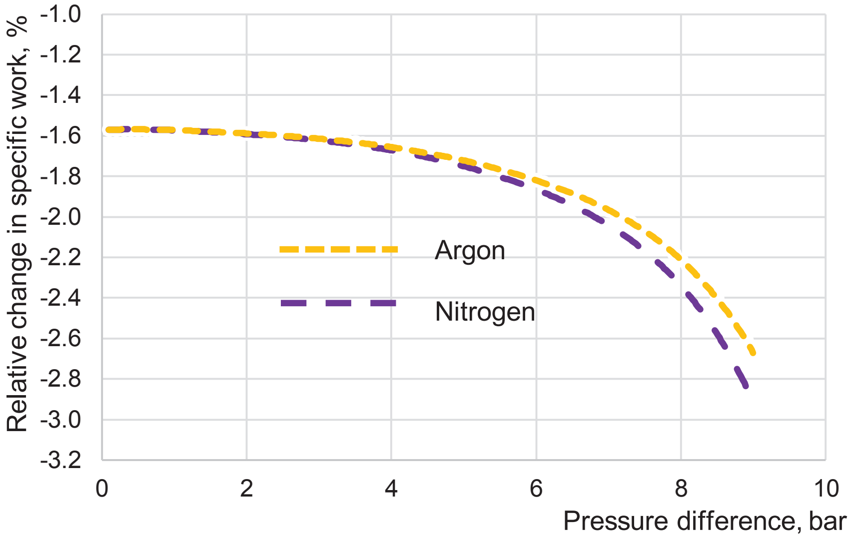

Figure 9. Effect of fivefold increase of volume V_c on specific work.

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.

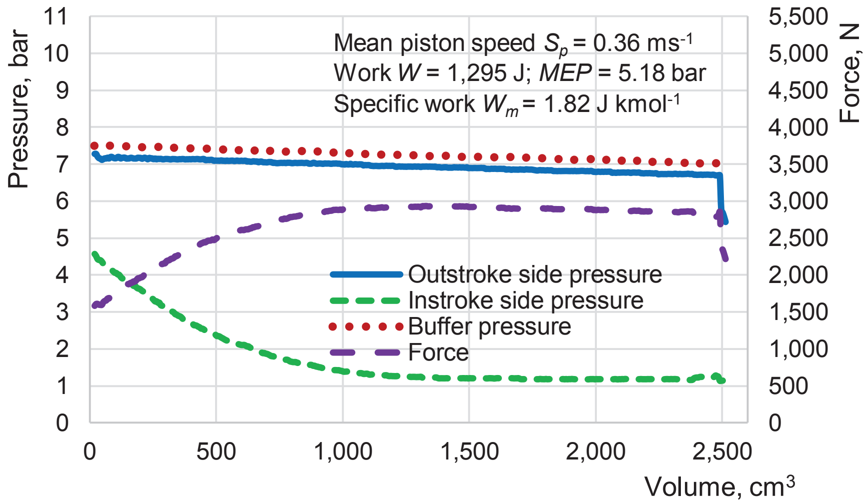

Figure 10. Pressure–volume diagram at uninterrupted gas supply mode, piston speed 0.36 m s^{-1}.

By supplying the gas during all the course of 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 Eq. 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. Experimental results published by Harris et al. using traditional control mode (continuous air supply) for extend stroke show similar outstroke and instroke side pressure curves, although mean piston speed in their case was approximately two times higher. (Harris et al., 2014). From the pressure – time graphs it appears, that they continued to supply the cylinder with air even after the piston has finished its course, increasing pressure in the working chamber and in that way increasing air consumption.

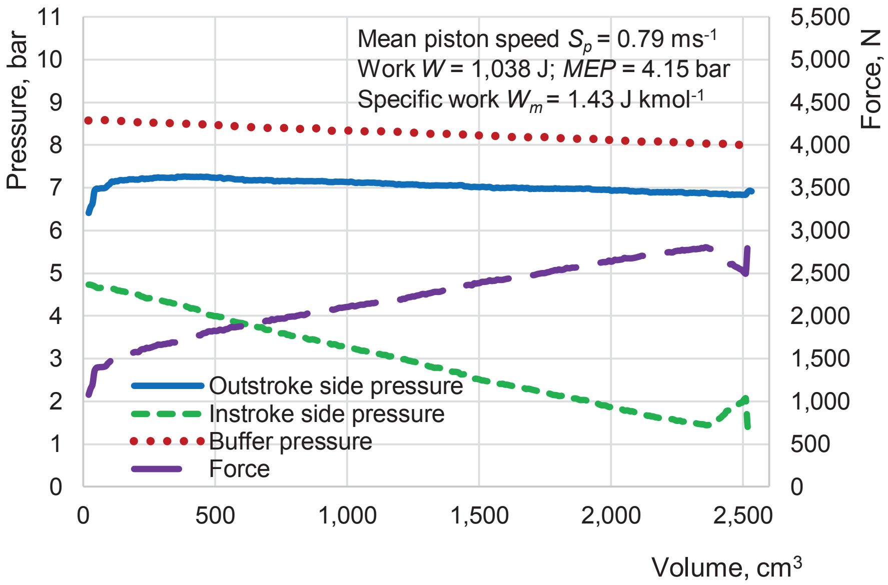

Figure 11. Pressure–volume diagram at uninterrupted gas supply mode, piston speed 0.79 m s^{-1}.

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. Comparing the pressure diagrams in Figs 9 – 11 to the results obtained by other researchers, significant difference in pressure curve in the chamber on the instroke side is noticeable. In the work of Doll et al., Du et al. and Harris et al. initial pressure in the non-working side is close to surrounding pressure (1 absolute or 0 bar relative). It probably means that they present and analyse data of first cycle or there is sufficiently long pause between the cycles for pressure drop to ambient level. Pressure at the end of the piston travel in the non-working side is increased for speed control purposes (Doll et al., 2011; Du et al., 2018; Harris et al., 2014). Optimisation of pneumatic cylinder operation for industrial purposes significantly differs from objectives set for pneumatic cylinder as the part of pneumatic competition vehicle motor. It appears that in trade-off between linearity of motion and pneumatic cylinder efficiency. Linearity of piston movement and longitude stiffness of pneumatic cylinder plays more significant role in industrial applications comparing to competition vehicle.

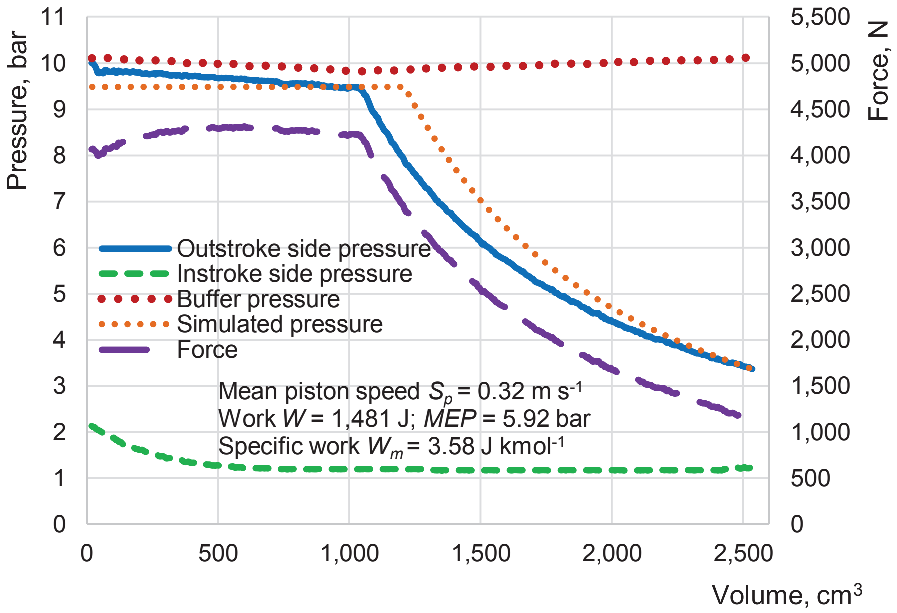

Figure 12. Pressure–volume diagram at gas expansion mode.

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. This finding is in agreement with other researchers, who proposed operational mode with gas expansion (Doll et al., 2011; Harris et al., 2014; Du et al., 2018). Direct comparison of the results is complicated, as most other authors do not use normalised efficiency parameters.

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, $\gamma = 1.403$. For experimental data its value is $\gamma = 1.158$. 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.

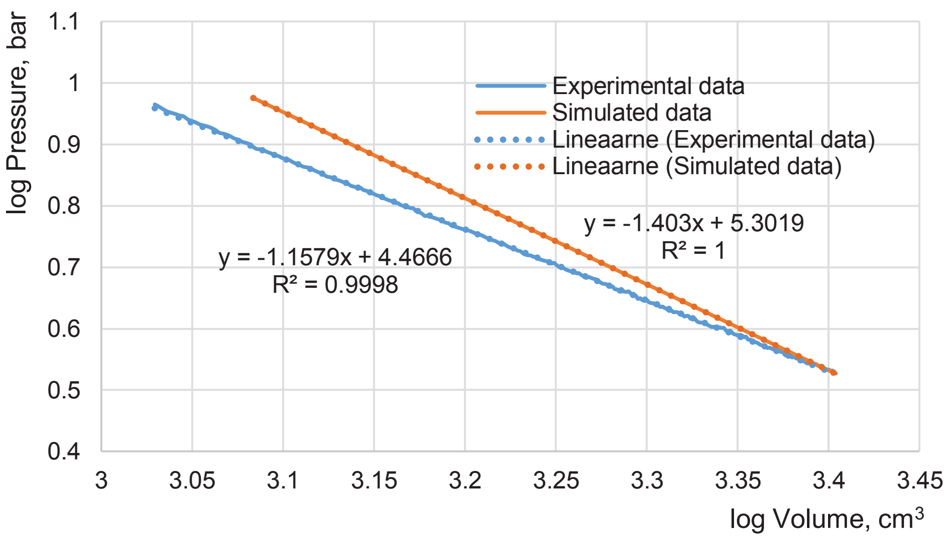

Figure 13. Pressure–volume diagram in logarithmic scale.

For practical calculations of pneumatic cylinder control parameters, such as V_{cut} value, apparent ratio of heat capacities can be calculated from pressure and volume relations on the previous cycle.

CONCLUSIONS

1. Mathematical model was developed for assessment of effect of gas type, geometric and control parameters on energy conversion efficacy of pneumatic motor. Energy conversion 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 gas of 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 V_c, 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 is 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 previous cycle, piston speed and gas flow restriction by output hole, pipe and valve.

- In case of relatively slow piston speed, S_p = 0.36 m s^{-1}, experimentally obtained specific work is by 17.6% lower than predicted.
- In case piston speed S_p = 0.79 m s^{-1}, relative difference between measured and predicted specific work is 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 is 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.

Future work

Prediction and evaluation of heat transfer between components of the pneumatic system, gas and surroundings can lead to deeper understanding and improvement of efficiency of pneumatic motor.

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