КОНТИНУАЛНО УПРАВЉАЊЕ ЕЛЕКТРОПНЕУМАТСКИМ СЕРВО УРЕЂАЈИМА

CONTINUOUS CONTROLLING OF ELECTRO-PNEUMATIC SERVO ACTUATORS

Petar MIŠLJEN*, **Miloš PAVIĆ, Nikola RADOSAVLJEVIĆ, Katarina NESTOROVIĆ**
Military Technical Institute, Belgrade, Serbia

https://doi.org

У овом раду је разматрана могућност унапређења управљања радом пнеуматског покретача. Због стишљивости ваздуха пнеуматски покретачи се најчешће користе у дискретном режиму рада, односно радни клип покретача заузима један од крајњих положаја. Кретање клипа покретача настаје као последица разлике притисака у коморама клипа покретача. Проблем кретања клипа покретача у складу са жељеним законом управљања је решен контролом полажаја млазнице ваздуха помоћу електромагнетног покретача. На основу тренутног положаја клипа пнеуматског покретача и његовог жељеног положаја, микрононтролер генерише PWM (pulse width modulation) сигнал и исти прослеђује енергетском претварачу који напаја електромагнетни покретач млазнице. Енергетски претварач је реализован помоћу прекидачких биполарних транзистора са изолованим гејтом (IGBT). Тестирање рада електропнеуматског покретача је извршено у лабораторијским условима и постигнути су задовољавајући резултати у жељеној области примене. Даљи рад треба усмерити на развој управљачких структура у циљу оптимизације рада електропнеуматског покретача.

Ključne reči: *Пнеуматски покретач, микроконтролер; PWM; електромагнетни покретач*

This paper considers the possibility of improving the control of the operation of a pneumatic actuator. Due to the compressibility of air, pneumatic actuators are most often used in discrete operating mode, that is, the working piston of the actuator occupies one of the final positions. Movement of the actuator piston occurs as a result of the difference in pressure in the piston chambers. The problem of moving the piston stroke in accordance with the desired law of control is solved by controlling the position of the air nozzle using an electromagnetic actuator. Based on the current position of the pneumatic actuator piston and its desired position, the microcontroller generates a PWM (pulse width modulation) signal and passes it to the energy converter that supplies the electromagnetic actuator of the nozzle. The energy converter is realized by switching bipolar transistors with isolated gates (IGBT). Testing the operation of the electro-pneumatic actuator was carried out in laboratory conditions and satisfactory results were achieved in the desired field of application. Further work should be focused on the development of control structures in order to optimize the operation of the electro-pneumatic actuator.

Key words: *Pneumatic actuator; microcontroller; PWM; electromagnetic actuator*

1. Introduction

Processes controlled by pressure often meet in nature. Both in plant and animal life. Many actuators are developed, which in their foundation are subject to change in pressure. One of the recent concepts of cross-border pressure application is the tensor actuator described in [1]. The average efficiency of this facility, in its operating range, is about 30%.

* Çorresponding author, e-mail: petarmisljen@gmail.com

Pneumatic actuators have found application in soft robotic systems. For this purpose, a special type of soft actuators have been developed, which should simulate the muscular structure of living organisms. This is the biggest challenge in the development of these actuators. The expected use of these actuators is in medicine, especially for the manufacture of dentures and rehabilitation devices. The efficiency of these actuators depends on their shape. In literature [2], the optimization of the shape of such an actuator was made. The research published in [2] refers to a glove that should be used for rehabilitation of the arm.

The development of robotic surgery led to the development of soft pneumatic actuators. 4 DOF pneumatic actuator with 4 chambers is presented in literature [4]. The literature [9] suggested the application of a rotary pneumatic actuator in order to improve the ergonomics of the pneumatically driven exoskeleton.

In the literature [5], two types of pneumatic actuators, rotary and linear, were analyzed, with the dynamics of their dynamic characteristics. The modeling of pressure change in chambers was carried out, deriving from the equations of energy maintenance and the equation of motion of the piston of the actuator.

The pronounced non-linearity of the linear pneumatic actuator is emphasized in [6]. The author proposed the linear pneumatic actuator model and presented the simulation results. The simulation results of a robust PID controller are shown in [8].

The subject of this paper is an electro-pneumatic actuator (EPA) consisting of piston, cylinder and electroventil. These actuators have a quick response, but due to air compressibility they are used to move the piston from one end to the other, that is, they are difficult to control [3, 7]. The aim of this paper is to reduce the impact of air compression and to ensure continuous control of the position of the actuator piston by using a controlled valve.

2. Model of an electro-pneumatic actuator

The functional diagram of the electropneumatic servo distribution is shown in Figure 1.

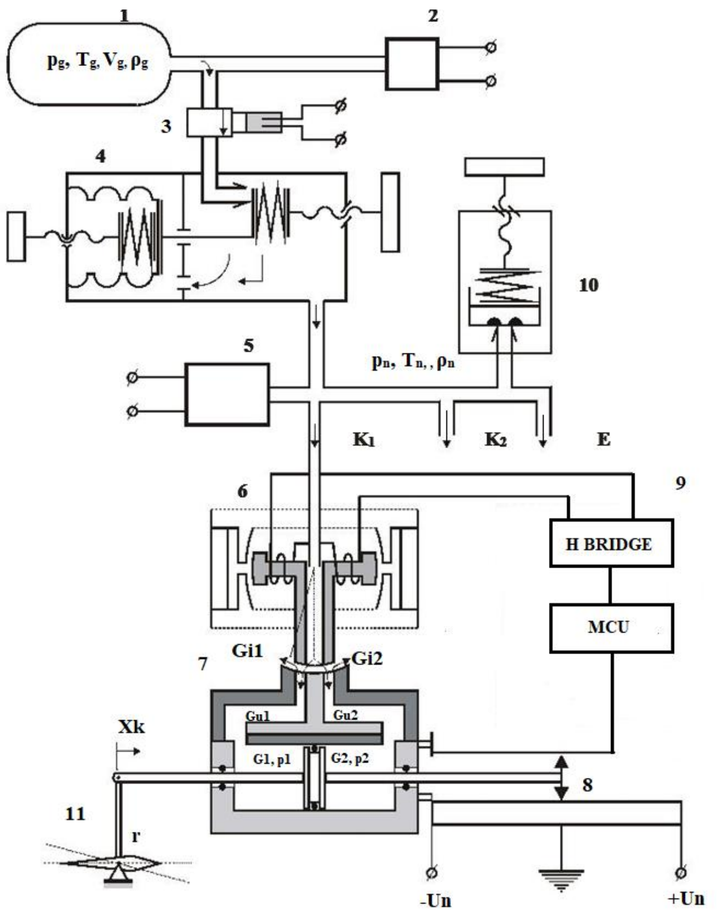

Figure 1. Functional diagram of the electropneumatic servo distributor

The basic elements of the system shown in Figure 1 are: reservoir (1); pressure transducer (2); start valve (3); pressure regulator (4); regulator pressure regulator (5); electropneumatic feeder (6); pneumatic piston engine (7); piston position sensor (8); control electronics (9); pressure relief valve (10); management object (11).

Elements (1), (3), (4), and (10) constitute a block of air devices and they provide pressure for air supply by EPA. The high pressure gas tank 1 is the primary source of energy. The start valve (3) includes the system operation process. It is activated by bringing the voltage to the contacts.

The compressed air enters the distributor (6) and expands through the nozzle into the piston engine (7). Electricity through the electric windings of the distributor (6) turns the nozzle. The piston engine is rotated around one point, and at the other end the executive element is attached - the handlebar. A position sensor (8) is directly mounted on the piston rod of the servomotor. The electronics block (9) (H bridge and MCU) performs the error signal processing and the formation of a control signal. The stabilized DC source is powered by electronic components.

The mass flow at the output of the pressure regulator is divided into three actuators, i.e. on three channels of management:

$$G_n = G_{n1} + G_{n2} + G_{n3} \tag{1}$$

If the losses in the supply pipe are neglected, then each control channel is valid:

$$G_n = G_{u1} + G_{u2} \tag{2}$$

where Gu1 and Gu2 are mass flows at the inlet openings of the cylinder with a piston. The index is omitted here and the flow through the branch to one control object is observed. This flow is divided into the part entering and expiring from the cylinder:

$$\begin{aligned} G_{u1} &= G_1 + G_{i1} \\ G_{u2} &= G_2 + G_{i2} \end{aligned} \tag{3}$$

where G1 and G2 are mass flows in the cylinder itself, and Gi1 and Gi2 are mass flows at the outlet openings of the cylinder with a piston.

Nonlinear equations of flow are:

$$\begin{aligned} G_{u1} &= C_d S_{u1} \frac{p_n}{\sqrt{T_n}} \sqrt{\frac{2\kappa}{R(\kappa-1)}} \sqrt{\left(\frac{p_1}{p_n}\right)^{\frac{2}{\kappa}} - \left(\frac{p_1}{p_n}\right)^{\frac{\kappa+1}{\kappa}}} \\ G_{i1} &= C_d S_{i1} \frac{p_1}{\sqrt{T_1}} \sqrt{\frac{2\kappa}{R(\kappa-1)}} \sqrt{\left(\frac{p_a}{p_1}\right)^{\frac{2}{\kappa}} - \left(\frac{p_a}{p_1}\right)^{\frac{\kappa+1}{\kappa}}} \\ G_{u2} &= C_d S_{u2} \frac{p_n}{\sqrt{T_n}} \sqrt{\frac{2\kappa}{R(\kappa-1)}} \sqrt{\left(\frac{p_2}{p_n}\right)^{\frac{2}{\kappa}} - \left(\frac{p_2}{p_n}\right)^{\frac{\kappa+1}{\kappa}}} \\ G_{i2} &= C_d S_{i2} \frac{p_1}{\sqrt{T_1}} \sqrt{\frac{2\kappa}{R(\kappa-1)}} \sqrt{\left(\frac{p_a}{p_2}\right)^{\frac{2}{\kappa}} - \left(\frac{p_a}{p_2}\right)^{\frac{\kappa+1}{\kappa}}} \end{aligned} \tag{4}$$

where: Sd is the flow coefficient; S = S (α) - the relevant surface for calculating flow G (normal to flow direction); p - input and output pressures; Su1 - the input surface of the first flow hole; Si1 -

outlet surface of the first flow opening; Su2 - the input surface of the second flow hole; Si2 - the output surface of the second flow opening; Gu1 - Inlet mass flow on the first opening of the cylinder with a piston; Gu2 - Inlet mass flow at the second opening of the cylinder with a piston; Gi1 - Output mass flow from the first opening of the cylinder with the piston and Gi2 - Output mass flow from the second piston opening with the piston.

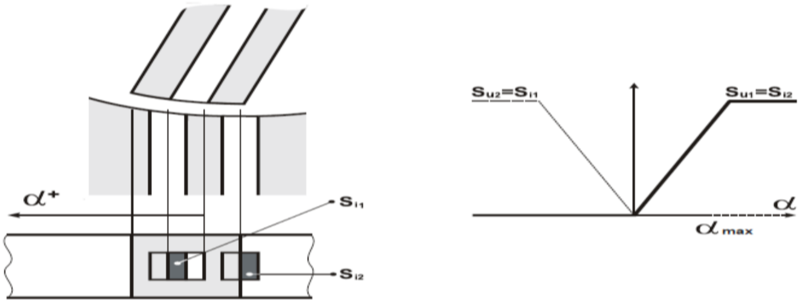

Figure 2. Change of flow surfaces

Based on the equations of state, continuity and energy maintenance in the pneumatic cylinder chambers, it is concluded that the change in the mass flow depends on the change in pressure in the chambers and on the movement of the piston.

Control volumes are:

$$V_1 = V_0 + S_k \cdot x_k$$
$$V_2 = V_0 + S_k \cdot x_k \tag{5}$$

where: V_1 and V_2 are the volume of the chamber relative to the position of the piston; V0 - volume of the cylinder chamber in the zero position; Sk - the surface of the piston; xk - piston position relative to zero position.

The continuity equation for a cylinder is:

$$\frac{d}{dt}\left(\rho_1 V_1\right) - \frac{d}{dt}\left(\rho_2 V_2\right) = \Delta \dot{G} \tag{6}$$

where: ρ_1 and ρ_2 - air density in chambers 1 and 2, ΔG - change of mass flow.

Piston movement in the pneumatic cylinder depends on the external load. The moment of the external load consists of the moment of inertia, the moment of friction and the hinged torque.

Since the cylinder process is isothermal, this is true:

$$\frac{p}{\rho} = const \Rightarrow \frac{dp}{\rho} - \frac{pdp}{\rho^2} = 0 \Rightarrow d\rho = \frac{\rho}{p} dp \tag{7}$$

It can be considered to be

$$\rho_1 = \rho_2 = \rho_0 , \tag{8}$$

so it is:

$$\Delta G = \frac{d}{dt}\left(\rho_1 V_1\right) - \frac{d}{dt}\left(\rho_2 V_2\right) = V_1 \frac{d}{dt}\rho_1 + \rho_1 \frac{d}{dt}\left(V_0 + S_k \cdot x_k\right) - V_2 \frac{d}{dt}\rho_2 - \rho_2 \frac{d}{dt}\left(V_0 - S_k \cdot x_k\right)$$
$$\Delta G = \rho_0 \frac{V_0}{p_o} \frac{d}{dt} \Delta p + 2\rho_0 S_k \frac{d}{dt} x_k \tag{9}$$

The schematic of the control unit and the H bridge is shown in Figure 3.

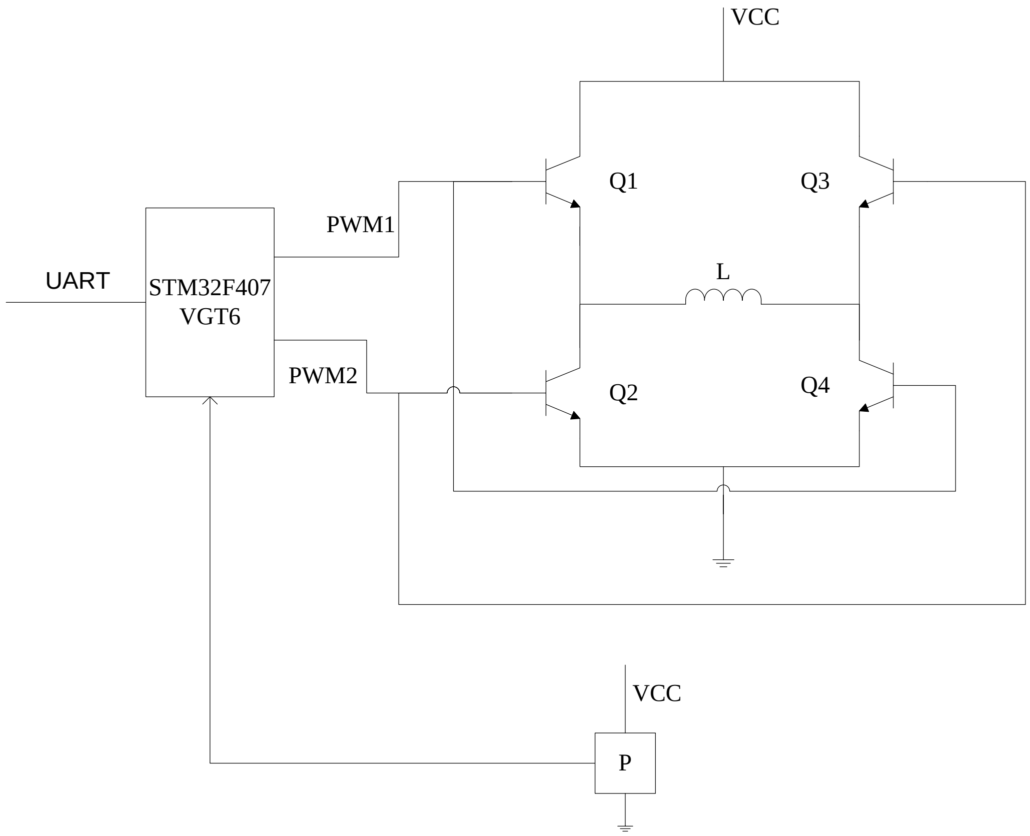

Figure 3. Control unit and H bridge

Through the serial data exchange protocol (UART), the microcontroller obtains information about the desired position of the piston driver. Based on the desired position of the actuator piston and the data obtained from the piston position sensor, the microcontroller calculates the required value of the control signal. The program code of the microcontroller implemented a proportional integral (PI) management law. The control signal is the degree of filling PWM signal that polarizes switching transistors in the H bridge. The PWM1 and PWM2 signals can not be active at the same time. If it is necessary to rotate the actuator nozzle in the positive direction, the PWM1 signal is active, and the PWM2 signal is activated to rotate the nozzle in the negative direction. Since the PWM signal strength depends on the mean value of the turning coil current, the change in the PWM signal fill rate changes the angle of rotation of the nozzle and the angle of rotation of the nozzle during one cycle of the microcontroller.

3. Experimental results

During the experiment, the position of the piston stroke was induced in the form of superposition of the sinus function and some constant value of 15 mm. The sine function amplitude is 10, and the frequency is changed in order to determine the bandwidth of the actuator. The obtained experimental results are shown in Figures 4-7. Signal *poz_1* is a signal from the piston position sensor, the signal *signal_P_1* represents the setpoint of the piston position.

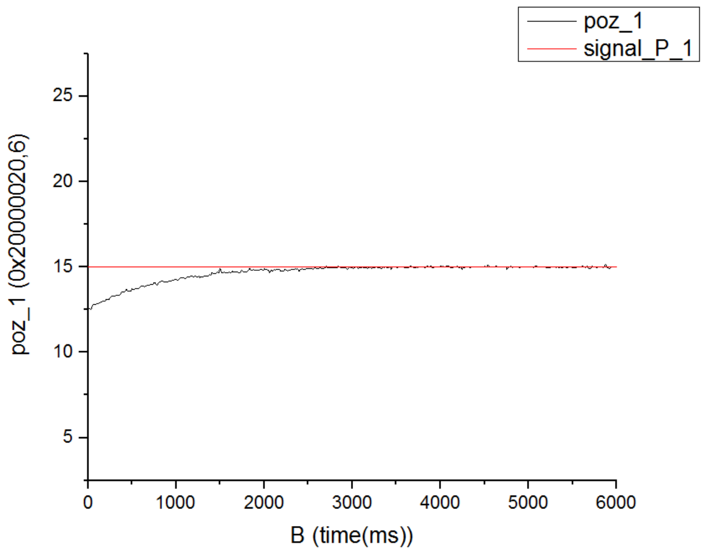

Figure 4. Pneumatic actuator response for the specified frequency 0 Hz.

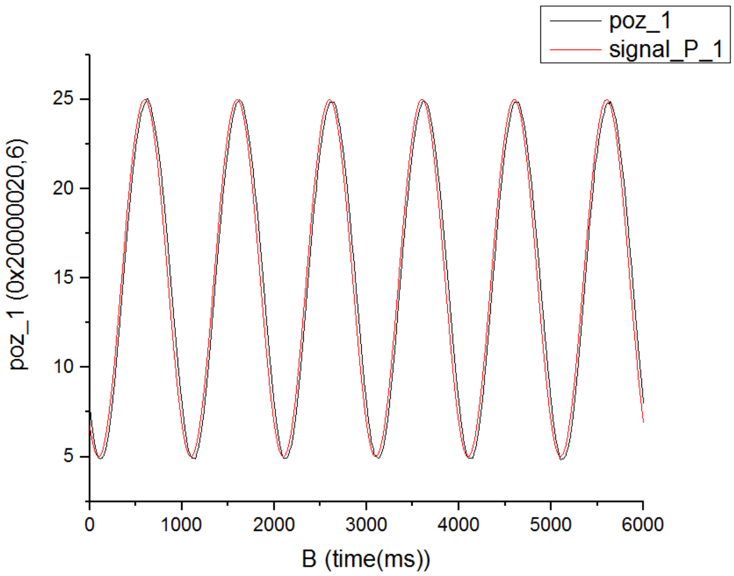

Figure 5. Pneumatic actuator response for the specified frequency 1 Hz.

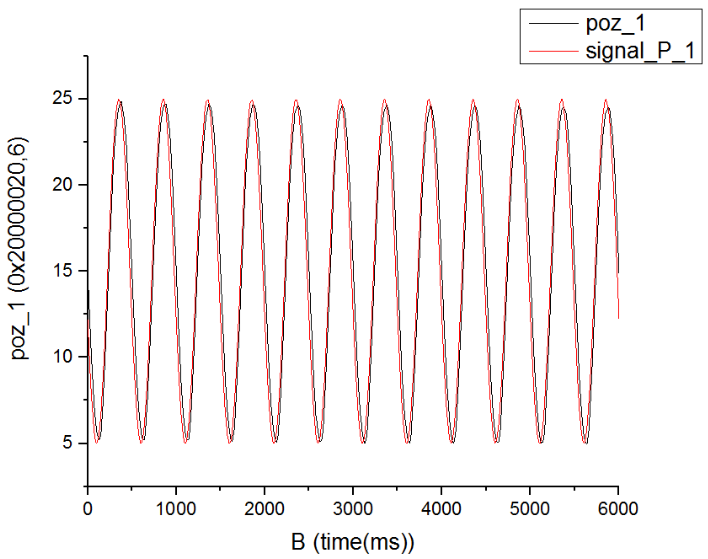

Figure 6. Pneumatic actuator response for the specified frequency 2 Hz.

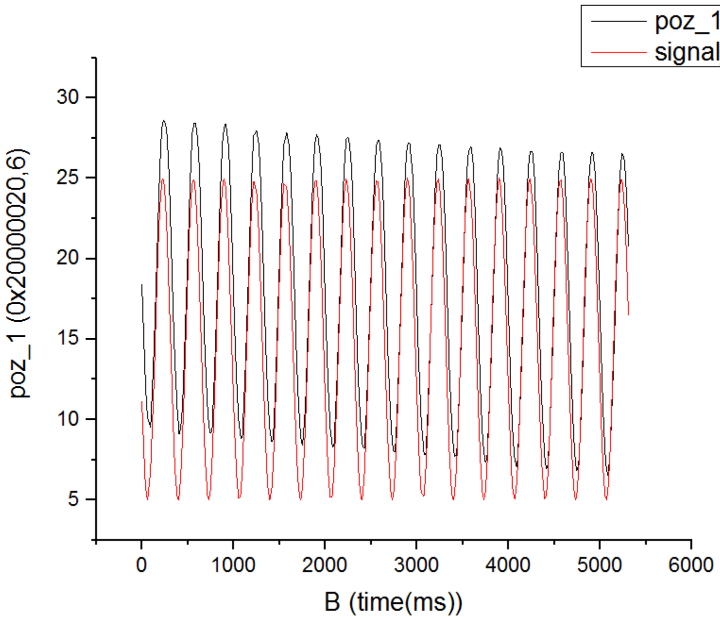

Figure 7. Pneumatic actuator response for the specified frequency 3 Hz.

4. Conclusion

In this paper, the set goal has been achieved. Continuous monitoring of the position of the pneumatic actuator piston was realized. From Figure 4-7 it is clear that the tested electropneumatic actuator closely follows the set position values up to 3 Hz. The goal is achieved using the microcontroller and the H bridge. In the further work, it is necessary to improve the management structure in order to expand the bandwidth.

5. References

[1] **Luchsinger R. H., Bräker M.**, A novel pneumatic actuator with Tensairity, *WIT Transactions on Ecology and the Environment*, *Vol 138*, 2010, pp. 365-374

[2] **Weiping Hu, Rahim Mutlu, Weihua Li, Gursel Alici**, A Structural Optimisation Method for a Soft Pneumatic Actuator, *MDPI Robotics 2018*, 7, 24

[3] **Ionel Cristian VLADU, Daniela ROȘCA, Nicu BÎZDOACĂ, Viorel STOIAN**, DYNAMIC CONTROL OF PNEUMATIC ACTUATOR SYSTEM, *Scientific Research and Education in the Air Force – AFASES, 2016*, pp. 399-404.

[4] **Luigi Manfredi1, Fabrizio Putzu, Saygun Guler, Yu Huan, Alfred Cuschieri**, 4 DOFs Hollow Soft Pneumatic Actuator – HOSE, *Materials Research Express*, 2018.

[5] **M. Sorli, L. Gastaldi, E. Codina, S. de las Heras**, Dynamic analysis of pneumatic actuators, *Simulation Practice and Theory 7* (1999) 589-602.

[6] **Mihai Avram, Constantin Nițu, Constantin Bucşan, Bogdan Grămescu**, Linear pneumatic actuator, *MATEC Web of Conferences 121*, 08001 (2017).

[7] **Steve Davis**, Pneumatic Actuators, *MDPI Actuators 2018*, *7*, 62.

[8] **Michael G. Skarpetis, Fotis N. Koumboulis, George Panagiotakis, Nikolaos D. Kouvakas**, Robust PID Controller for a Pneumatic Actuator, *MATEC Web of Conferences 41, 02001, 2016.*

[9] **Jobin Varghese, Akhil V.M., Rajendrakumar P.K., Sivanandan K.S.**, A rotary pneumatic actuator for the actuation of the exoskeleton knee joint, *Theoretical & Applied Mechanics Letters 7* (2017) 222–230.