Improving Energy Saving in Conventional Pneumatic Systems by Using Air Booster Experimentally

Dr Jafar Mehdi Hassan
Mechanical Engineering Department,University of Technology /Baghdad.
Dr. Majid Ahmed Oleiwi
Control and System Engineering Department ,University of Technology/Baghdad.
Amer Abed Mansour
Control and System Engineering Department ,University of Technology/Baghdad.
Walaa Mousa Hashim[*]
Electromechanical Engineering Department,University of Technology/ Baghdad.
Emai;:mousawalaa@yahoo.com

Received on:28/10/2014 & Accepted on:9/7/2015

Abstract

Pneumatic actuators are widely used in industry and many other applications, whereas low energy efficiency has been recognized as a critical drawback compared with corresponding hydraulic and electrical actuators. The paper presents a study to improve the performance and energy efficiency of the traditional pneumatic drive with vertical load 50N, by added air booster type IPR (Input Pressure Reduce) to the. Four supply pressures used 2, 4, 6 and 8 bar, compared with the traditional control of the motion of the asymmetric cylinder in which maximum energy saving obtained at 6 bar 16.7%.
Keywards: Energy saving, Conventional Pneumatic Systems and Air Booster

تحسين كفاءة المنظومات الهوائية ذات الدوائر المفتوحة باستخدام مكبر الضغط عمليا

الخلاصة

للمشغلات الهوائية تطبيقات واسعة في المجالات الصناعية والمجالات الاخرى وذلك لما يميزها عن المنظومات الاخرى حيث تتميز بانها خفيفة الوزن،بيئة نظيفة،رخيصة الثمن مقارنة مع المنظومات الهيدروليكية.في هذا البحث تم دراسة طريقة جديدة في تحسين كفاءة المنظومة الهوائية ولما لها من مردود في تقليل معدل الصرف للطاقة من خلال ربط مكبرضغط الهواء عند مدخل الصمام لدائرة هوائية بحمل عمودي 50N واعادة تدوير هواء العادم للمكبر عند مدخل المشغل ولضغوط تجهيز مختلفة 2،4،6،8 بار حيث تم الحصول على اعلى توفير بالطاقة عند 6 بار وبمقدار 16.7%.

https://doi.org/10.30684/etj.2015.108845
2412-0758/University of Technology-Iraq, Baghdad, Iraq

INTRODUCTION

Pneumatic actuators can offer better alternatives to electrical or hydraulic actuators for great many applications. Pneumatic actuators have the advantages of low cost, high power-to-weight ratio, ease of maintenance, Cleanliness, having a readily available; operate at high speed and cheap power source. Owing to numerous advantages, pneumatic actuators and systems are widely applied in industrial automation for the so-called sequential control. Energy saving in pneumatic system is divided in three parts first reduce air consumption to minimum value as possible by reduced the losses of components or use some device or method, second reused exhaust air by recovery the energy with some device or method, third in servo pneumatic system applied advanced control system to obtain accurate position in less time as possible and hence reduce in useful air consumption of delay time.

The model of the pneumatic cylinder is nonlinear, for this, Jia. K. et [1].al and Jihong. W. et.al [2] use a developing and energy-efficient control strategy to avoid the problem of solving the complicated nonlinear differential equations which transfer to linear system description which lead to poor air saving.

Khalid.A. et.al [3], Ming.H .T et.al [4], J. Gyeviki et.al [5], Vladislav et.al [6] using sliding mode control and hybrid fuzzy sliding mode control with and without rod double acting cylinder They proved that pneumatic servo systems can be used for the accurate robust position control, not only for the movement between two hard stops. The experimental results showed that proposed sliding-mode controller gives fast response and good transient performance. Furthermore, the controlled system was robust to the variations of the system parameters and external disturbances and they do not require accurate modeling..

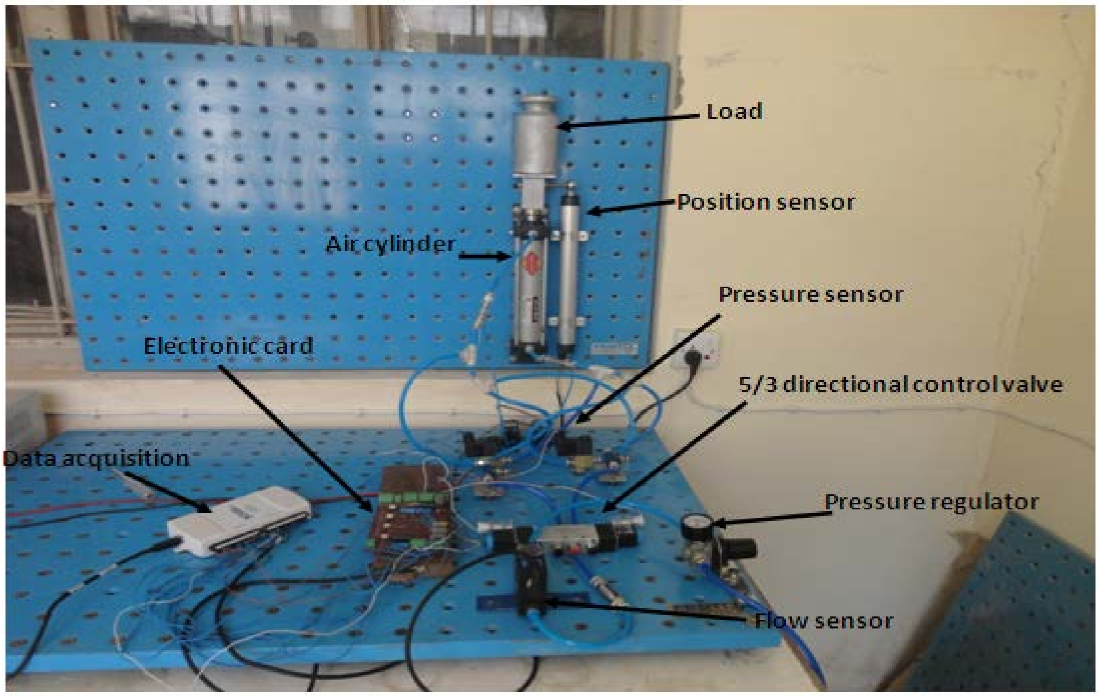
Vladislav .B. et.al [7] and Khalid .D et.al [8] reduce pressure difference between pressure supply and active pressure line at the stroke start motion by adding bridging two ways valve with rodless cylinder and horizontal load.

In present study air booster is used before inlet main valve and recycling exhaust air booster at the entrance to the actuator circuit. This method leads to increase in supply air pressure ratio 1:2 and benefit from exhaust booster to reduce air flow rate consumption.

Experimental system set-up

Figure.1 shows a pneumatic system rig in which designed and constructed at the laboratory of Post Graduate Studies of the Department of mechanical Engineering at the University of Technology to present the experimental work of saving energy in pneumatic system. It consists of two important parts, such as electric and pneumatic part. Electric part consists of PC, data acquisition and control module, amplifier and safety module, position sensor; flow sensor and pneumatic part consists of the necessary components for conventional pneumatic system, such as various actuators, valves, tubes and other accessories

.



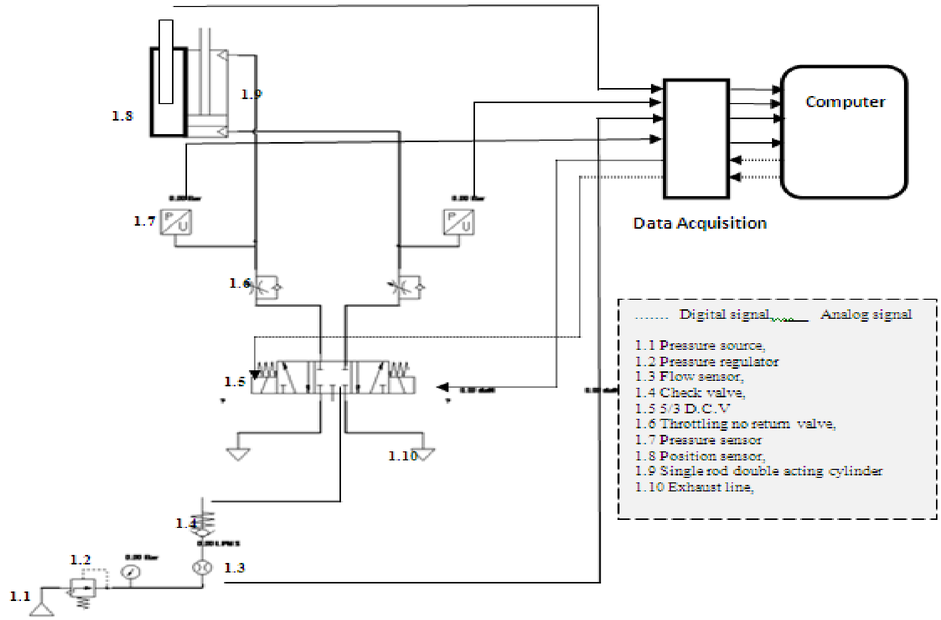
Figure(1). Pneumatic experimental Rig

Pneumatic part consists of the necessary components for conventional pneumatic system, such as double acting cylinder type of CDA1-L50-200, 5/3 way directional control valve type, one way directional control valve type , load 50N, A Festo variable non return throttle valve type, Non return valve type KAM-08, Pressure regulator type Expflex AR-200, A Festo flow sensor type SFAB-200U-WQ8-25A-M12, A Festo pressure sensor type SDE1-D10-G2-HQ4-C-P2-M8, a Festo Linear Variable Differential Transformer (Position Sensor) type MLO-POT-225-LWG, data acquisition type NI USB-6212, Power supply type MCH-505D, and tubes, the schematic diagram of the circuits in which be used shown in figure 2 and 3.

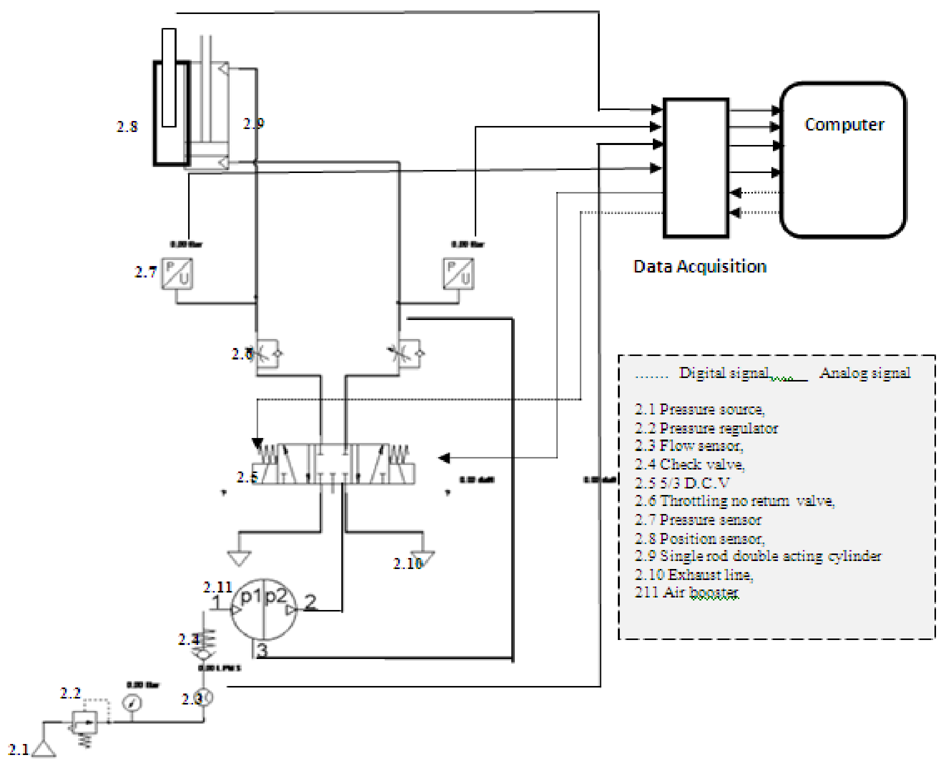
Control circuit

MATLAB SIMULIK program used to control circuits in Fig.1 and 2 with 2 digital outputs and 4 analog inputs, in which included,

1- Sequence operation of digital and analog signals.
1- On-off switch.
2- Number of strokes.
3- Strokes reset.
3- Length of stroke.



Figure(2). CPS circuit without air booster(circuit No1)

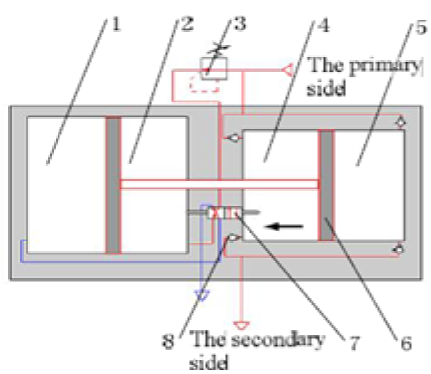


Figure(3). CPS circuit with air booster(circuit No2)

IPR air booster

Because of its compact structure, small size, no external power supply, etc, pneumatic booster is widely used in locally pressure boosting applications where there is a need for a small amount of higher pressure air [9][10] . The most common booster is called input pressure reduced (IPR) booster. The output pressure is set by adjusting the input pressure of the driving chambers by a regulator, with the development of energy saving technologies of the pneumatic system, the importance of booster has become more and more obvious [11][12]. However this type of booster has its own shortages, such as its small output flow, when the boosting ratio is higher, the shortage becomes more distinct Furthermore.

A typical IPR booster, as shown in Fig. 4, is composed of a regulator, a piston, driving chambers, boosting chambers, a reversing valve, and four check valves When the driving chamber A is connected to the atmosphere through the reversing valve, part of the compressed air charged from the primary side (the suction side, as shown in Fig. 4) flows directly into the boosting chamber B while the remaining air flows into the driving chamber B through the regulator and the reversing valve successively. As a result of the regulator, the air pressure in driving chamber B will be reduced. The compressed air in boosting chamber B and driving chamber B drives the piston to move toward the left. The air pressure in boosting chamber A increases until the pressure is higher than that of the secondary side Thereafter, the higher pressure compressed air is discharged from the chamber A to the secondary side. When the piston reaches its travel destination and impacts the rod of the reversing valve, the reversing valve changes its state, and causes the air in the driving chamber B to flow to the atmosphere. The air charged from the primary side flows directly into boosting chamber A and flows into the driving chamber A through the regulator and the reversing valve. The air in the boosting chamber A and the driving chamber A then drives the piston to move toward the right while the air pressure in the boosting chamber B increases. Finally, the higher pressure compressed air in the boosting chamber B is discharged to the secondary side. The booster continues to deliver higher pressure compressed air by repeating the process discussed above [13].



1. Driving chamber A , 2. Driving chamber B 3. Regulator,
4.Boosting chamber A , 5. Boosting chamber B 6. Piston
7. Reversing valve , 8. Check valve

Figure(4). schematic diagram of air booster

Theoretical analysis

Air booster model

Assumption:

1- The working fluid (air) of the system follows all ideal gas lows.

2- There is no leakage between the chambers, the area of the piston rod end is too small to be considered, and the effective areas of all intake and exhaust ports are the same.

3- Supply temperature is equal to atmosphere temperature (i.e. T=constant).

4- The flow of air moving into and out of the chambers is a stable one dimensional flow that is equivalent to the flow of air through the nozzle contraction.

Each chamber is considered as one control unit and the whole booster is considered as the coordinate system, all of the chambers do not exhaust and charge air simultaneously. According to the ratio (P_{b1}/P_{b2}) the flow equation for the flow through a restriction can be written as follows [13]

$$\dot{m} = \begin{cases} CA\sqrt{2\rho_1 P_{b1}\left(\frac{k}{k-1}\right)(P_{b1}-P_{b2})\left[\left(\frac{P_{b2}}{P_{b1}}\right)^{\frac{2}{k}}-\left(\frac{P_{b2}}{P_{b1}}\right)^{\frac{k+1}{k}}\right]}, & \frac{P_{b1}}{P_{b2}}>0.5283 \\ CA\sqrt{k\rho_o P_o\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}, & \frac{P_{b1}}{P_{b2}}\leq 0.5283 \end{cases} \quad \ldots(1)$$

Where: $\dot{m}$ mass flow rate (at any cross-section)in kg/s,

- C orifice flow coefficient, dimensionless,
- A cross-sectional area of the air booster orifice in m²,
- ρ_1 real air density under upstream conditions, kg/m³,
- P_{b1} air upstream pressure in Pa,
- P_{b2} air downstream pressure in Pa,
- ρ_o real air (total) density at total pressure P_o and total temperature T_o, in kg/m³,
- P_o absolute upstream total pressure of the air, in Pa,
- T_o absolute upstream total temperature of the air in K.

Assume air density is constant=1.22kg/m^3, adiabatic process k=1.4, C=0.4, A=5*10^{-5}m^3, for high pressure side $\mathrm{P_{b1}} =$ pressure supply 2, 4, 6, 8 bar and for low pressure side $\mathrm{P_{b1}} =$ atmospheric pressure (1bar), the above equation is simplified to below form

$$\dot{m} = 5.8 * 10^{-5}\sqrt{P_{b1}(P_{b1}-P_{b2})\left[\left(\frac{P_{b2}}{P_{b1}}\right)^{1.4}-\left(\frac{P_{b2}}{P_{b1}}\right)^{1.7}\right]} \quad \ldots(2)$$

Mass Flow Rate Model [13]

$$\dot{m}_{v}=\begin{cases} C_f * A_v * C_1 * \frac{P_u}{\sqrt{T}} & if \ \frac{Pd}{Pu} \leq P_{cr} \\ C_f * A_v * C_2 * \frac{P_u}{\sqrt{T}} (\frac{P_d}{P_u})^{\frac{1}{k}} \sqrt{1-(\frac{P_d}{P_u})^{(k-1)/k}} & if \ \frac{Pd}{Pu} > P_{cr} \end{cases} \quad \ldots (3)$$

$$C_1=\sqrt{\frac{K}{R}(\frac{2}{K+1})^{\frac{K+1}{K-1}}} \quad ; \quad C_2=\sqrt{\frac{2K}{R(K-1)}} \quad ; \quad P_{cr}=(\frac{2}{K+1})^{\frac{K}{K-1}}$$

For air (k = 1.4) we have C_1 = 0.040418, C_2 = 0.156174, and P_{cr} = 0.528, T=300 k, $C_f = 0.25$ [14]

$$A_v = d^2 * \frac{\pi}{4} = (8*10^{-3})^2 * \frac{\pi}{4} = 5.02*10^{-5} m^2$$

$$\dot{m}_{v}=\begin{cases} 34.43*10^{-9} * P_u & if \ \frac{Pd}{Pu} \leq 0.528 \\ 133*10^{-9} * P_u * (\frac{P_d}{P_u})^{0.714} \sqrt{1-(\frac{P_d}{P_u})^{0.285}} & if \ \frac{Pd}{Pu} > 0.528 \end{cases} \quad \ldots (4)$$

For extend stroke to calculate $\dot{m}_{vin}$, $P_u = P_s$& $P_d=P_1$ and $\dot{m}_{vout}$, $P_u = P_2$ & $P_d=P_a$ while in retract for $\dot{m}_{vin}$, $P_u = P_s$& $P_d=P_2$ for $\dot{m}_{vout}$, $P_u = P_1$& $P_d=P_a$

Cylinder Chambers Model:

For cylinder chamber1 [14]

$$\dot{P}_I = \frac{RT}{V_{o1}+A_1 x}(\dot{m}_{in}\alpha_{in} - \dot{m}_{out}\alpha_{out}) - \alpha \frac{\pm A_1 P_1}{V_{o1}+A_1 x}\dot{x} \quad \ldots (5)$$

Active length for side1 L =0.2 m , active area for side1 A_1 =1.96*10^{-3} m^2, Inactive volume for side1 V_{o1} = 1.6*10^{-5}m^3, ambient temperature T=300 k gas constant R=287 J/kg.k, α_{in}=1.4, $\alpha_{out} = 2$, $\alpha = 1.2$, the model is simplified to follow form:

$$\dot{P}_1 = \frac{86100}{1.6*1^{-5}+1.96*10^{-3}*x}(1.4*\dot{m}_{in} - 2*\dot{m}_{out}) - 1.2\frac{\pm 1.96*10^{-3}*P_1}{1.6*1^{-5}+1.96*10^{-3}*x}\dot{x} --(3.52)$$

In which for extend

$$\dot{P}_1 = \frac{86100}{1.6*10^{-5}+1.96*10^{-3}*x*}(\dot{m}_{in}*1.4 - \dot{m}_{out}*2) - 1.2\frac{1.96*10^{-3}*P_1}{1.6*10^{-5}+1.96*10^{-3}*x}\dot{x} \quad \ldots (6)$$

And for retract

$$\dot{P}_1 = \frac{86100}{1.6*10^{-5}+1.96*10^{-3}*x*}(\dot{m}_{in}*1.4 - \dot{m}_{out}*2)1.2\frac{1.96*10^{-3}*P_1}{1.6*10^{-5}+1.96*10^{-3}*x}\dot{x} \quad \ldots (7)$$

And for cylinder chamber 2

$$\dot{P}_2 = \frac{RT}{V_{o2}+A_2(L-x)}(\dot{m}_{in}\alpha_{in} - \dot{m}_{out}\alpha_{out}) - \alpha \frac{\pm A_2 P_2}{V_{o2}+A_2(L-x)}\dot{x} \quad \ldots (8)$$

Substitute active length for side 2, L =0.2 m , active area for side 1 A_2 =1.646*10^{-3} m^{2,} inactive volume for side1 V_{o2} = 1.03*10^{-5}m^3, ambient temperature T=300 k, Gas constant R=287 J/kg.k , α_{in}=1.4, $\alpha_{out} = 2$, $\alpha = 1.2$ we get

$$\dot{P}_2 = \frac{86100}{1.03*1^{-5}+1.646*10^{-3}(0.2\pm x)}(1.4* \dot{m}_{in} - 2* \dot{m}_{out}) - 1.2\frac{\pm 1.646*10^{-3}*P_2}{1.03*1^{-5}+1.646*10^{-3}(0.2\pm x)}\dot{x} \quad \text{-- (9)}$$

For extend

$$\dot{P}_2 = \frac{86100}{1.03*10^{-5}+1.646*10^{-3}(0.2-x)}(1.4*\dot{m}_{in} - 2*\dot{m}_{out}) - 1.2\frac{1.646*10^{-3}*P_2}{1.03*10^{-5}+1.646*10^{-3}(0.2-x)}\dot{x} \quad \text{--- (10)}$$

For retract

$$\dot{P}_2 = \frac{86100}{1.03*10^{-5}+1.646*10^{-3}(0.2-x)}(1.4*\dot{m}_{in} - 2*\dot{m}_{out}) + 1.2\frac{1.646*10^{-3}*P_2}{1.03*10^{-5}+1.646*10^{-3}(0.2-x)}\dot{x} \quad \text{---- (11)}$$

Piston-Load Dynamics Model [15]

$$M\ddot{x} + F_f = P_1.A_1 - P_2.A_2 - P_a.A_r \quad \dots (12)$$

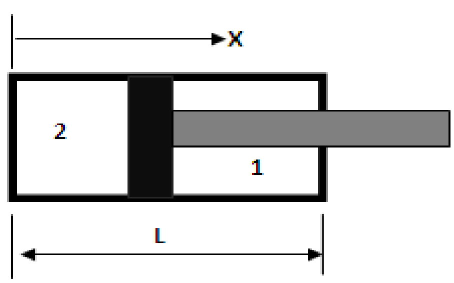
$$\ddot{x} = \frac{1}{M}\left(P_1.A_1 - P_2.A_2 - P_a.A_r - F_f\right) \quad \dots (13)$$

$$F_{static}=0.67\frac{N}{mm}.d_{bore} \quad , \quad F_{dynamic}=0.4\frac{N}{mm}.d_{bore} \quad \dots (14)$$

for d_{bore} =0.05m , F_{static} and $F_{dynamic}$ are (33.5 and 20)N respectively, Actuator cross-sectional area side 1 A_1= 1.96*10^{-3} m^2,Actuator cross-sectional area for side 1 A_2= A_1-A_r=1.96*10^{-3}-0.314*10^{-3}= 1.646*10^{-3} m^2 , mass piston & rod & load assembly = 6.165 kg.

By neglecting atmospheric force acting on the piston rod (P_aA_r) the model is simplified to follow form in start motion:

$$\ddot{x} = \frac{1}{1.165}(1.96*10^{-3}*P_1 - 1.646*10^{-3}*P_2 - 100.5) \quad \dots (15)$$

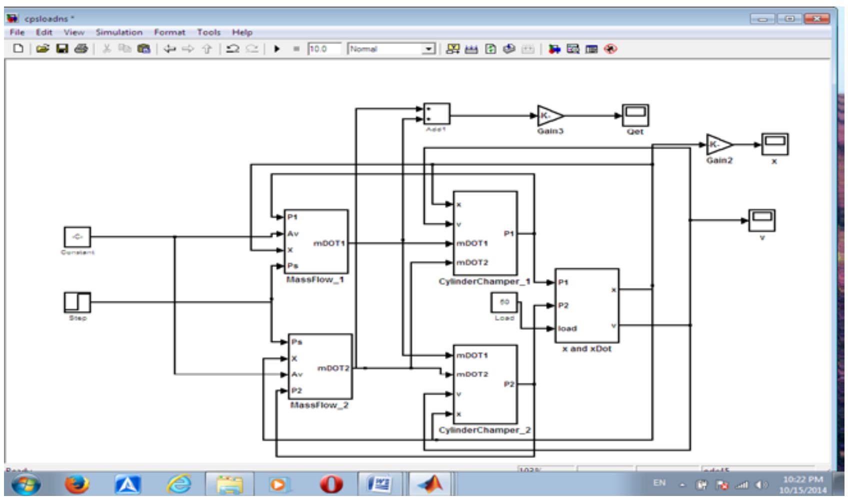


Figure(5). air cylinder coordinate

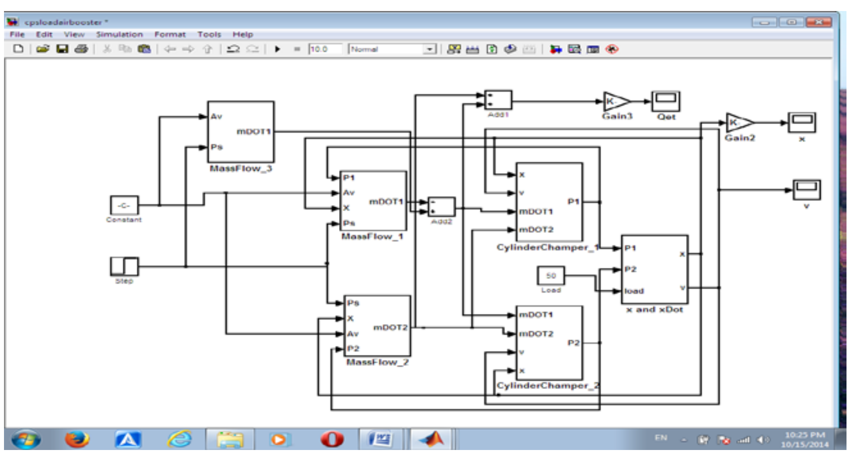
From Fig. 5 B.C: extend (1) $L \geq x \geq 0$
retract (2) $L \geq 0.2\text{-}x \geq 0$

Simulation results

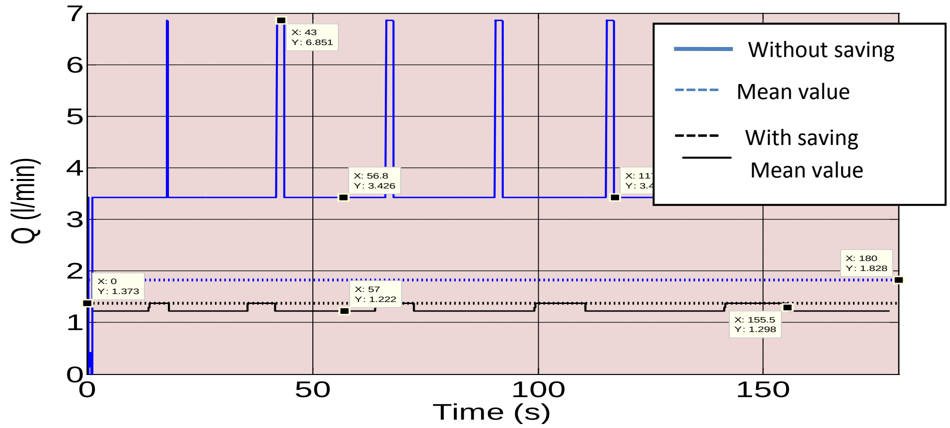
Fig. 6 & Fig.7 show subsystem models of nonlinear mathematical model of the conventional pneumatic actuator using MATLAB/SIMULINK with and without air booster while Fig 8, 9, 10 show simulation results obtained at pressure supply 4, 6, 8 bar respectively, due to the extend and retract air consumption drop a mean value reduces from (1.82 to 1.3732, 3.65 to 1.794, and 4.2 to 2.313) l/min in which lead to air saving 6.75%, 19%, and 7% for pressure supply (4, 6, and 8) bar respectively..



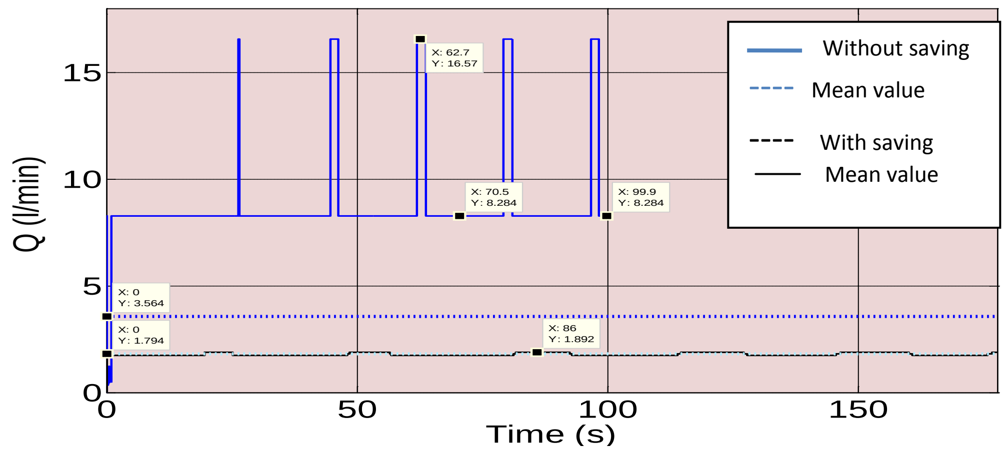
Figure(6). Simulation model of nonlinear mathematical model of the conventional Pneumatic actuator without air booster



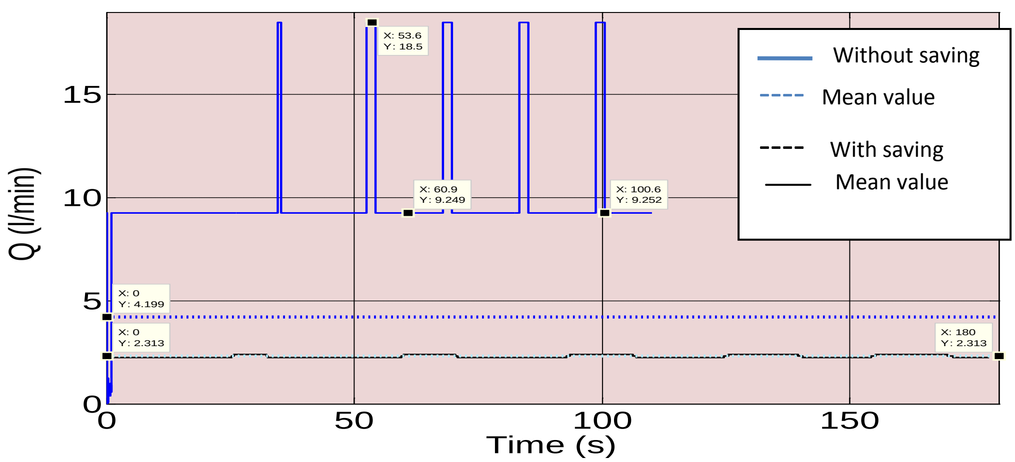
Figure(7). Simulation model of nonlinear mathematical model of the conventional Pneumatic actuator with air booster



Figure(8). air consumption with and without air booster (4bar)



Figure(9). air consumption with and without air booster (6bar)



Figure(10). air consumption with and without air booster (8bar)

Experimental results

For a ten stroke with 200 mm stroke length and pressure supply 4 bar at constant average velocity and ambient temperature with vertical load 50N, Fig.11 show comparison in results obtained with and without air booster.

1- It is clear a nonlinearly of air consumption with time in comparison with the Fig.8 due to high air compressibility inside the booster.

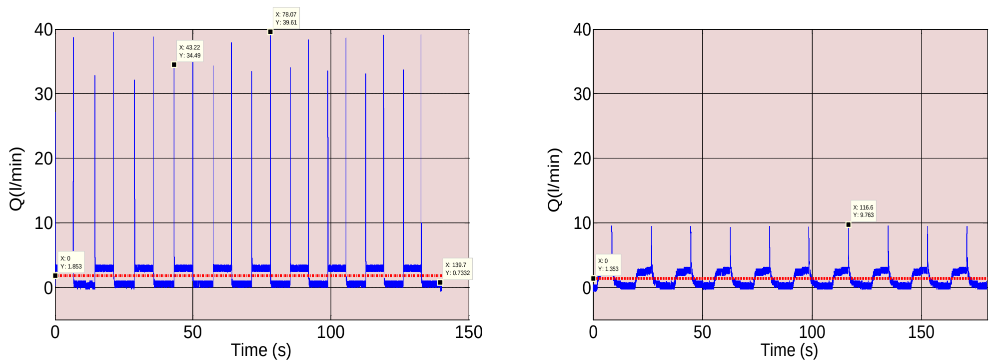
2-Extend and retract overshoot drop from (39.61 and 34.49) l/min to constant value in both type is 9.763l/min while mean flow rate drop from (1.853 to1.353)l/min, this lead to save in air consumption 6% with time increment30 second.

Air consumption is calculated from the Figure.11 a & b by multiplying mean flow rate value with period time for one stroke then for ten strokes as follow:

Air consumption=mean value*time*10

Air consumption is determined from divided the difference in air consumption to air consumption without saving as follow:

Air saving = difference in air consumption/air consumption without saving



Without saving **With saving**

Figurte(11). flow rate change with time

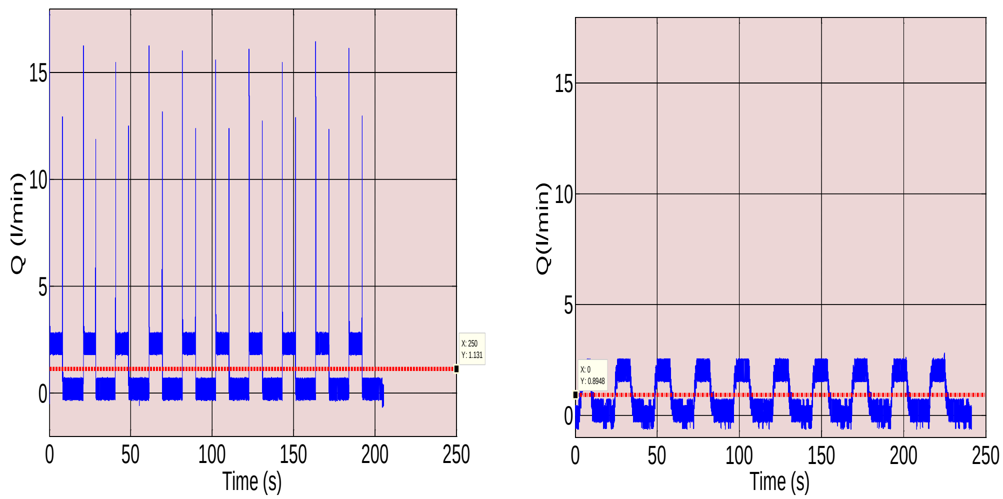
Fig. 12, 13, 14, 15 show comparison in results obtained by using variable pressure supply (2, 6, 4 and 8) bar, its note that at low pressure value there is no extend and retract overshoot and increases with pressure, saving in air consumption for (2, 4, 6, and 8) bar are 7.6%, 6%, 16.7%, and 2% respectively at expense time (36, 35, 25 and 7) second. Its noted that more efficient occur at moderate pressure (6 bar) because of when the air supply pressure are set at 6 bar the output flow of the air booster ascends stably with increase of the terminal pressure of the air in the driving chamber of the booster.

Comparison Results

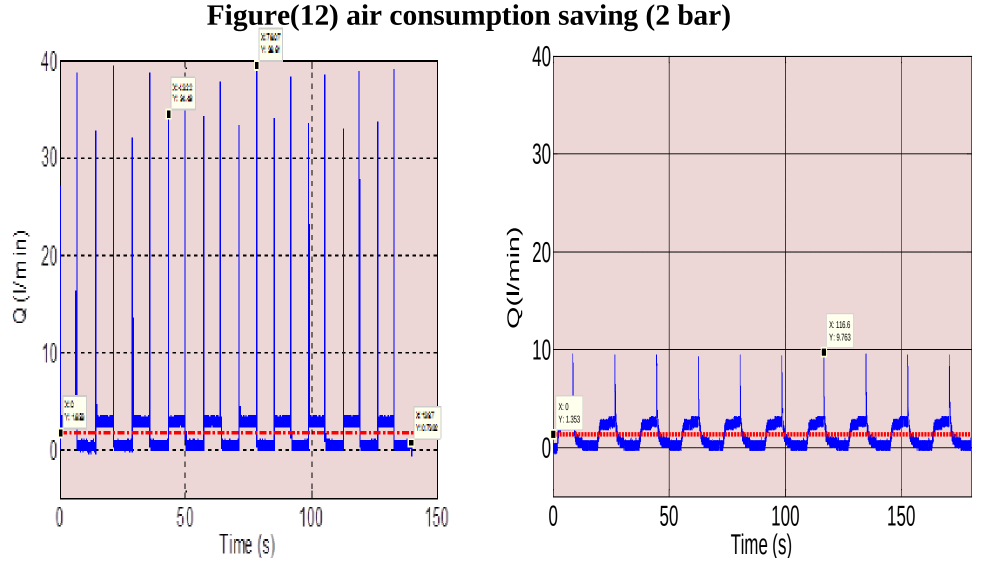
Table.1 shows a difference between the theoretical and experimental results due to high air compressibility, neglecting friction effects and some assumptions and approximation used to simplify the solution.

Table .(1) comparison results

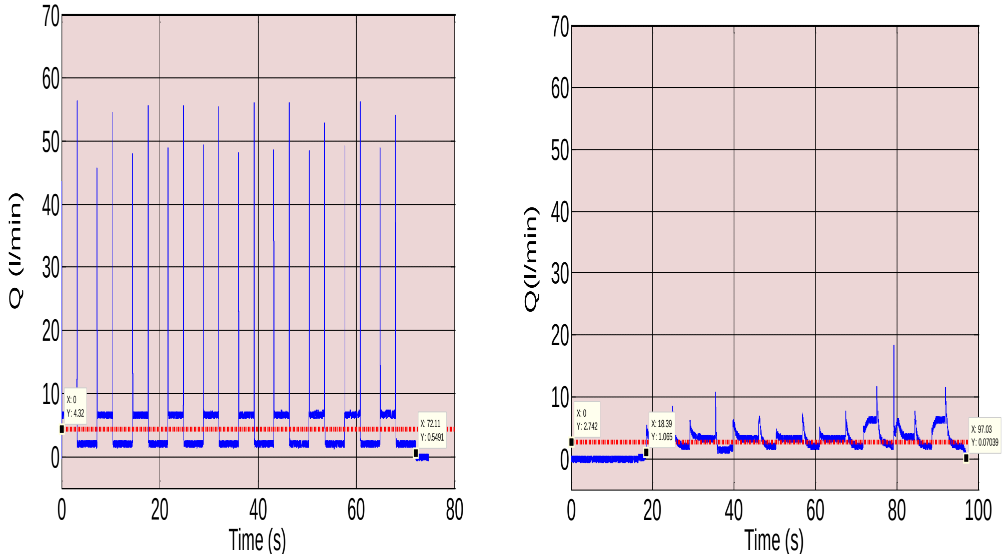
CPS with booster &vertical load	**Pressure supply(bar)**		**4**	**6**	**8**
	Air saving %	**Theoretical**	**6.75**	**19**	**7**
		Experimental	**6**	**16.7**	**2**
		Difference	**0.75**	**2.3**	**5**



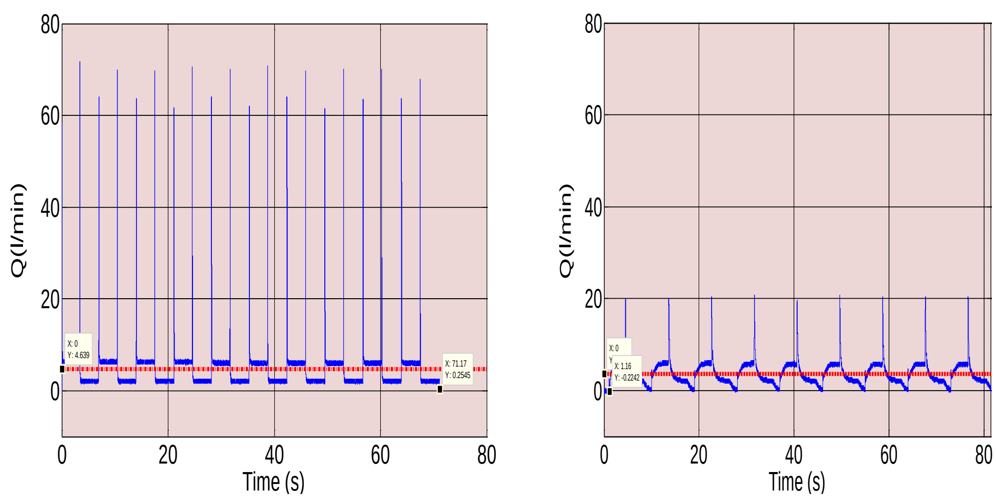
Figure(12) air consumption saving (2 bar)



Figure(13) air consumption saving (4 bar)



Figure(14) air consumption saving (6 bar)



Figure(15) air consumption saving (8bar)

Electrical cost saving

Table.2 illustrates saving in electric cost in one year by using air booster for 16 hour operation in one day, one kilowatt to produce air in 6 bar [16].

Table (2) electrical cost saving

Energy needed to produce compressed air in 1 day at 6 bar	1kW*16 h = 16kWh
Cost of electricity	£0.045/kWh
Hours of operation in 1 year	Two shifts, 5 days per week, 4160 h per year
Energy cost of 1 year	£0.045*4160 = £187.2
One year cost saving	£187.2*16.7 = £3126

Conclusion

The paper presents an experimental model for energy efficient examination of conventional pneumatic system. Architecture of experimental model for energy efficiency examination is a modular with the possibility of flexible adjustment for measurement of compressed air consumption (flow) at various linear and rotary or swivel pneumatic actuators. One of the main purposes of experimental model is to provide energy efficiency using air booster. This is the result of increase pressure in booster chamber in ratio 1:2 and recycling exhaust air chamber in to cylinder.

Refrences

[1] Ke.J, K.Thanapalan, J.Wang, and H.Wu "Development of Energy Efficient Optimal Control of Servo Pneumatic Cylinders" Control 2004, University of Bath, UK, September 2004.

[2] Wang.J, S.Member, IEEE, and T.Gordon "Energy-Efficient Tracking Control of Pneumatic Cylinders" 50th IEEE Conference on Decision and Control and European Control Conference (CDC-ECC) Orlando, FL, USA, pp 12-15, December 2011.

[3] Khalid.A. Al-Dakkan ,E. J. Barth. and M.Goldfarb "A Multi-Objective Sliding Mode Approach for the Energy Saving Control of Pneumatic Servo Systems" ASME International Mechanical Engineering Congress & Exposition Washington, D.C ,pp15-21., November 2003.

[4] Tsai M.H, C.N.heng, and M. C.Shih "Design and Control for the Pneumatic Cylinder Precision Positioning Under Vertical Loading" National Science Council of Taiwan ,NSC 94-2212-E- pp 006-033 2006.

[5] Gyeviki.J, J. sarosi, T. Endrody , E. Forgacs, and P. Toman " LabView Based Position Control for A Pneumatic Cylinder" Annals of Faculty Engineering Hunedoara – International Journal of Engineering, pp. 58 –65, 2110.

[6] Blagojevic.V, M.Stojiljkovic1, B.Veselic2 “Advantages of Digital Sliding Mode Position Control of the Rodless Pneumatic Cylinder” Automatic Control and Robotics Vol. 9, No 1, pp. 95 - 102, 2010.

[7] Blagojevic.V, and M.Stojiljkovic “Increasing Eenergy Efficiency of the Execution Part of Pneumatic System by Restoring Energy” Mechanical Engineering Vol. 6, No 1, pp. 37 – 44, 2008.

[8] Khalid.A. AI-Dakkan, M.Goldfarb, and E.J. Barth “Energy Saving Control for Pneumatic Servo Systems” 2W3 IEE/ASME International Conference on Advanced Intelligent Mechatronics2003, pp 284-289, July 21, 2009.

[9] SMC (China) Co., Ltd. Modern practical pneumatic technology(3)[M]. Beijing: China Machine Press, 2008. (in Chinese).

[10] CHEN Weiguo, DING Xusheng, GUO Xiangfeng. Study on principle and method of fluid pressurization technology[J]. Machine Tool & Hydraulics, 2001(2): 54–55. (in Chinese).

[11] ONEYAMA N. Pneumatic manufacturer’s attempts on energy saving[J]. Hydraulics and Pneumatics, 1996, 27(3): 372–377.

[12] SHI Yan, CAI Maolin, WANG Gaoping. Study on air supplied with different pressure and locally pressure boosting technology of pneumatic system[J]. Machine Toll and Hydraulics, 2010, 28(9) 57–59. (in Chinese).

[13] SHI. Y , C.Maolin ,X.Weiqing, and L.Jie “Expansion Energy Used Pneumatic Booster” Chines Journal of Mechanical Engineering, May 16, 2012.

[14] Edmond . R and Yildirim . H “A High Performance Pneumatic Force Actuator System Part 1 - Nonlinear Mathematical Model” ASME Journal of Dynamic Systems Measurement and Control, Vol. 122, No.3, pp. 416-425, February 12, 2001.

[15] Fleischer H “Manual of pneumatic system operation” McGraw-Hill. New York San Francisco Washington Auckland Bogotá Caracas Lisbon London 1995.

[16] Yang.A ,J.Pu,C.B.Wong,P.Moore “By-pass Valve Control to Improve Energy Efficiency of Pneumatic Drive System” Journal of Elsevier, Control Engineering Practice 17, pp 623–628,2009.