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# Reuse Remaining Air Pressure in Classical Pneumatic System

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ABSTRACT: In practical industrial applications of pneumatic systems, the issues of energy saving are receiving attention due to the considerable amount of compressed air wasted in operation. This research investigates a configuration of pneumatic systems while a special emphasis is put on the saving of energy in the execution. A method of recovering energy is also presented. In order to improve the performance and energy efficiency of the pneumatic drive, the by-pass valve control scheme is designed and studied. For pneumatic positioning application, in the constant speed phase, the inlet and outlet chambers will be connected via by-pass valve to reduce the overshoot and allow exhaust compressed air to be reused. Three supply pressures used5, 7 and 9 bars, compared with the traditional control of the motion of the asymmetric cylinder in which maximum energy saving obtained at 5, 7 and 9barare 16.7% 23.44% and 23.5% respectively.

KEYWORDS: Reuse Residual, Air Pressure and Conventional Pneumatic System

## INTRODUCTION

With the increasing consciousness of the want for environmental defense, energy-saving and low-carbon technologies have been the present point of progress. Pneumatic systems, that account for the industrial energy use around 10 - 20 %, are unavoidably addressed as a power-saving point. Low power efficiency is the key factor limiting more improvement of pneumatic systems. Research reveals that power consumption can be decreased by around 8% for every 0.10 MPa decrease in the compressor supply pressure [1]. Pneumatic actuators can provide enhanced options to hydraulic or electrical actuators for enormous uses. Pneumatic actuator has the benefits of high power to weight ratio, maintenance ease, low cost, cleanliness, easily available; operate at great speed and an affordable energy source. Owing to many benefits, pneumatic actuators are generally employed in industrial automation for sequential control.Pneumatic system power saving can be seen effective in reducing air consumption via reducing losses in components, or by recovering the energy via reusing the exhaust air. A group researchers proposed a novel booster regulator along with energy recapture, and investigated its energy efficiency via introducing the air power concept [2].

A increased the efficiency of the pneumatic system by choosing the best components, and then using a suitable controller which have high reliability and response in order to reduce power consumption by the compressor [3]. On the other hand a researchers investigated a novel method for energy saving in a point-to-point actuation of a pneumatic system [4]. The method predicted the system’s actuation using the Gas Low and the actuator model, and commits air supply cut-off at the time when the energy in the actuator was sufficient to complete the actuation task. However cutoff at end-stroke and cut-off using model prediction can reduce the amount of air potential energy wasted in conventional actuation by up to 43.5% and 80.2% respectively at expense of the actuation time of predicted cut-off in which increases by up to 25%.

Some researchers proposed that the exhaust pressure of the cylinder hold middle level (0.2~0.5) MPa, and so the downstream flow available for running air blow guns or any application, in which assuming that exhaust flow was used effectively [5]. It was shown that the method improved the immediacy of response (caused by higher initial cylinder pressure on the driving side) and reduces the amount of air consumption as long as the condition of choked flow and lower maximum driving force are noted. The performance and energy efficiency of the pneumatic drive by using by-pass valve control scheme for pneumatic positioning application, in the constant speed phase [6]. The energy efficiency can be improved compared with the traditional control system. The experimental results showed that 12-28% energy saving can be achieved by using the by- pass valve control method under the same working conditions as the traditional control system. Some researhcers presented a new hybrid pneumatic-electrical system aiming at energy efficiency improvement by recovering exhaust air energy from pneumatic actuator outlets to generate electricity [7]. A closed-loop control strategy is proved to be essential to ensure the exhaust energy recovery work properly and to maintain existed actuator operations simultaneously.

A portable energy-saving type air supply system. A variable volume tank is developed in order to drive a pneumatic actuator with a low discharge pressure in a tank.it was observed that the variable volume tank can drive the actuator at a lower supply pressure than with the constant volume tank [8]. Due to nonlinear model of the pneumatic cylinder, 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 [9,10]. A group researchers proposed a new experimental model to examine the energy efficiency of the traditional pneumatic systems in addition to the servo systems, regardless of actuator type [11]. It consisted of two parts, the electric and pneumatic parts. The carried experiments have confirmed that this novel experimental model provided adequate results of actuator placement and also the power efficiency. Some researchers described and compared two different methods to estimate compressed air leakage by infrared and ultrasound thermography [12]. The abilities and constraints of these technologies were investigated, as well as the dependability and results accuracy thus obtained. In this study, an air booster is utilized prior to the inlet main valve and reuses the booster exhaust air through the actuator circuit entrance. This method leads to reduce consumption of air flow rate of the system hence increases energy efficiency.

## EXPERIMENTAL SYSTEM SET-UP

Figure.1 shows the experimental Rig for energy efficiency examination in which consists of two main important parts electric and pneumatic part. Electric part consists of PC, data acquisition and control module, amplifier and safety module, proximity switch; flow pressure sensors while pneumatic part consists of the necessary components for conventional pneumatic system.

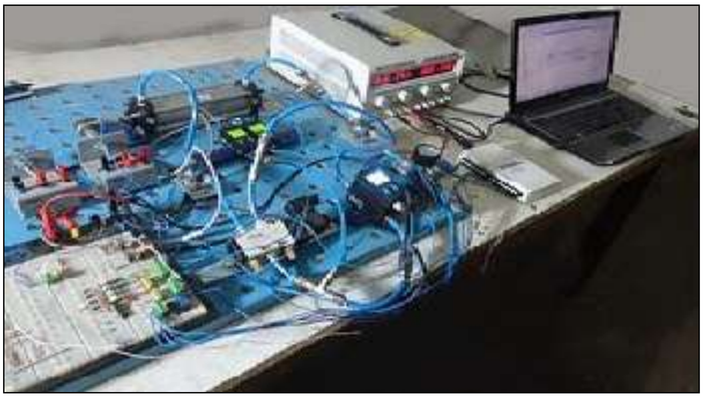


**Figure 1.** Experimental Rig

Pneumatic part consists of main components for conventional pneumatic system. Double acting cylinder with single rod type of CDA1-L50-200 with and without load 50 N was used as actuator. To control the air flow in extending and retracting strokes of actuator a 5/3 way directional control valve normally closed with solenoid operation and spring return was used while two one way directional control valve normally closed with solenoid operation and spring return was used to achieve abridging link between extend and retract process. To protect a system from excessive pressure a Feston on return valve type KAM-08 was provided on the other hand to adjustable air pressure supply at a limit quantity a pressure regulator type Exp flex AR-200 was utilize, A Festo flow sensor type SFAB-200U-WQ8-25A-M12 and two Festo pressure sensor type SDE1-D10-G2-HQ4-C-P2-M8 was used to transfer analog signal to data acquisition type NI USB-6212 hence to P.C in which record the variations of flow rate and air pressure with time continuously. A Festo type variable non-return throttle valve manual operated was used to study conventional pneumatic system, in order to control the speed of stroke in case of extended and retraction which has rang pressure 0.1-10 bar also A Festo position sensor type MLO-POT-225-

LWG was used to control the accurate position of pneumatic cylinder used in study of conventional pneumatic system, the schematic diagram of the circuits in which be used shown in figure 2 and3.

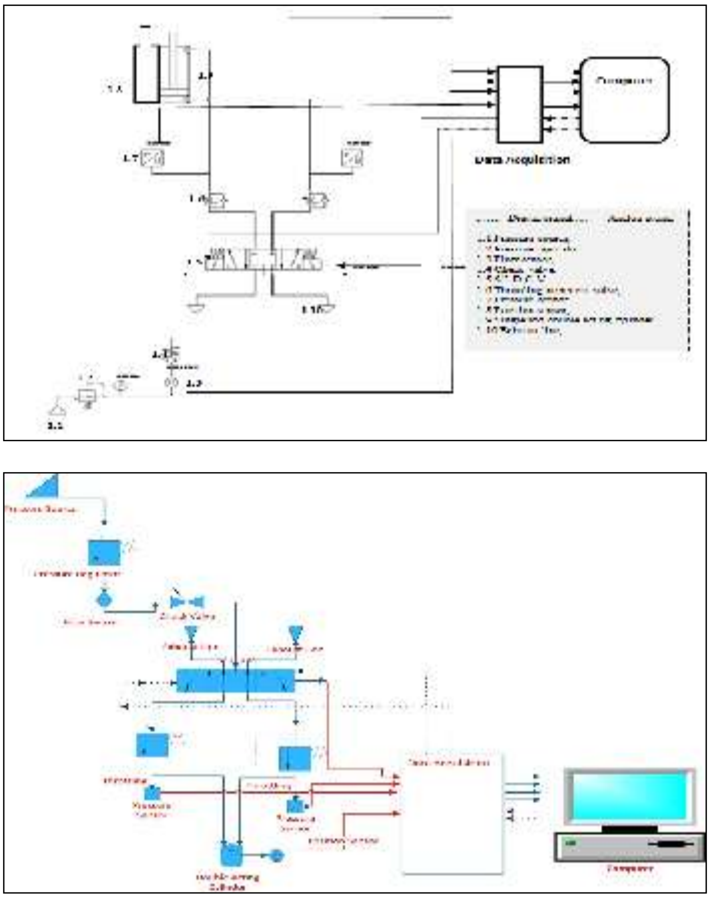
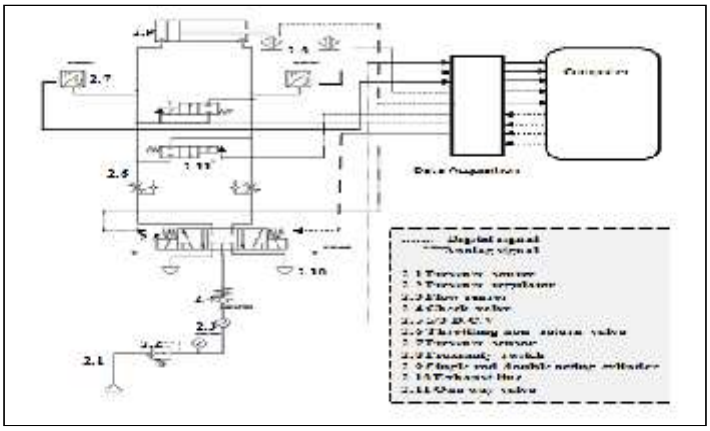


**Figure 2.** Conventional Pneumatic System circuit without bridging valve (circuits No1)



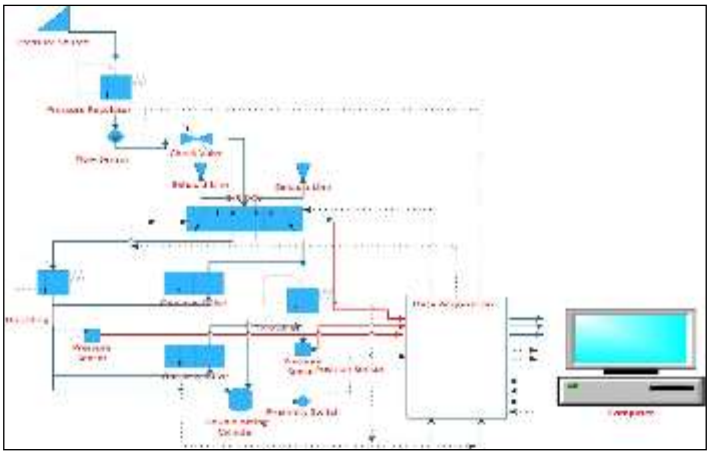


**Figure 3.** Conventional Pneumatic System circuit with bridging valve (circuits No2)

Control circuit

MATLAB SIMULIK tool used to control circuits in Fig.2 and 3 with four digital outputs and five analog inputs, one solenoid valve digital signal for extend and other for retract while one signal for both one way directional control valves. Two analog signal for pressure sensors and one for flow meter and two of proximity switches. Control circuits also included Number of stroke, the working time for the inlet and outlet digital orders and inlet analog.

Theoretical analysis

Mass Flow Rate Model

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

$$C1=\sqrt{\frac{K}{R}(\frac{2}{K+1})^{\frac{K+1}{K-1}}} \quad ; \quad C2=\sqrt{\frac{2K}{R(K-1)}} \quad ; \quad Pcr=(\frac{2}{K+1})^{\frac{K}{K-1}}$$

For air (k = 1.4) we have C1 = 0.040418, C2 = 0.156174, and Pcr= 0.528, T=300 k,Cf = 0.25[13]

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

$$\dot{m}_v=\begin{cases} 34.43 * 10^{-9} * P_u & \text{if} \quad \frac{P_d}{P_u} \le 0.528 \\ 133 * 10^{-9} * P_u * (\frac{P_d}{P_u})^{0.714} \sqrt{1-(\frac{P_d}{P_u})^{0.285}} & \text{if} \quad \frac{P_d}{P_u} > 0.528 \end{cases} \quad (1)$$

For extend stroke to calculate $\dot{m}_{vin}, P_u = P_s$&Pd=P1 and$\dot{m}_{vout}, P_u = P_2$&Pd=Pa while in retract for $\dot{m}_{vin}, P_u = P_s$& Pd=P2 for$\dot{m}_{vout}, P_u = P_1$& Pd=Pa

Cylinder Chambers Model:

For cylinder chamber 1 [14]

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

Active length for side1 L =0.2m, active area for side1 A1 =1.96*10-3 m3, Inactive volume for side1 Vo1 = 1.6*10-5m3, ambient temperature T=300 k gas constant R=287 J/kg.k,$\alpha_{in}$=1.4,$\alpha_{out} = 2, \alpha = 1.2$,the model is simplified to follow form:

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

In which for extend

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

$$-1.2\frac{1.96*10^{-3}*P_1}{1.6*10^{-5}+1.96*10^{-3}*x}\dot{x} \qquad (2)$$

And for retract

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

$$-1.2\frac{1.96*10^{-3}*P_1}{1.6*10^{-5}+1.96*10^{-3}*x}\dot{x} \qquad (3)$$

And for cylinder chamber 2

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

Substitute active length for side 2, L =0.2m, active area for side 1 A2 =1.646*10-3 m2, inactive volume for side1 Vo2 = 1.03*10-5m3, ambient temperature T=300 k, Gas constant R=287 J/kg.k ,$\alpha_{in}$=1.4,$\alpha_{out} = 2, \alpha = 1.2$ we get

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

For extend

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

$$-1.2\frac{1.646*10^{-3}*P_2}{1.03*10^{-5}+1.646*10^{-3}(0.2-x)}\dot{x} \qquad (4)$$

For retract

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

$$-1.2\frac{1.646*10^{-3}*P_2}{1.03*10^{-5}+1.646*10^{-3}(0.2-x)}\dot{x} \qquad (5)$$

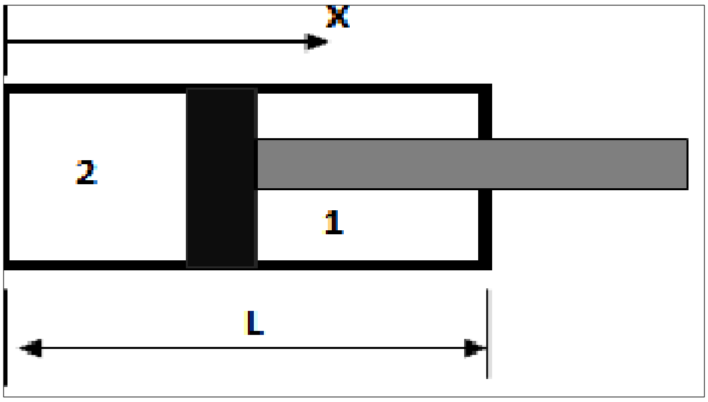



**Figure 4.** Air cylinder coordinate

Piston-Load Dynamics Model

$$M\ddot{x} + F_f = P_1.A_1 - P_2.A_2 - P_a.A_r$$

$$\ddot{x} = \frac{1}{M}(P_1.A_1 - P_2.A_2 - P_a.A_r - F_f) \quad (6)$$

$$Fstatic=0.67\frac{N}{mm}.dbore \quad , \quad Fdynamic=0.4\frac{N}{mm}.dbore[15\ ]$$

ford bore =0.05m , F static and F dynamic are (33.5 and 20)N respectively, Actuator cross-sectional area side 1 A1= 1.96*10-3 m2,Actuator cross-sectional area for side 1 A2= A1-Ar=1.96*10-3-0.314*10-3=1.646*10-3 m2, mass piston & rod & load assembly = 6.165 kg.

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

$$\ddot{x} = \frac{1}{1.165}(1.96 * 10^{-3} * P_1 - 1.646 * 10^{-3} * P_2 - 100.5) \quad (7)$$

B.C: extend (1) $L \geq x \geq 0$

retract (2) $L \geq 0.2\text{-}x \geq 0$

Simulation results

Due to complexity of the system mathematical models were developed based on some design assumptions that, in most cases, simplifies the conditions and functionality. Figure 5 & Figure 6 show subsystem models of nonlinear mathematical model of the conventional pneumatic actuator using MATLAB/SIMULINK with and without valve bridging while fig7, 8, and 9 shows the difference in air consumption for simulation between circuits No1 (no saving) and No2 (bridging method) for variable pressure supply (5, 7, and 7) bar at constant velocity. In extending stroke the flow rate drops from ((2.53) to (1.85), (3.54) to (2.59) and (4.5) to (3.34)) l/min respectively while air consumption dropped along retracting stroke from ((1.26) to (0.92), (1.7) to (1.3) and (2.27) to (1.6)) l/min respectively. The consumption rate was reduced due to the difference in effective area. This led to mean value of air consumption drop from ((1.55) to (1.137), (1.99) to (1.46), and (2.57) to (1.88)) l/min. Therefore air saving in ten stroke for (5, 7, and 9) bar were (26.5%), (26%), and (27%), respectively.

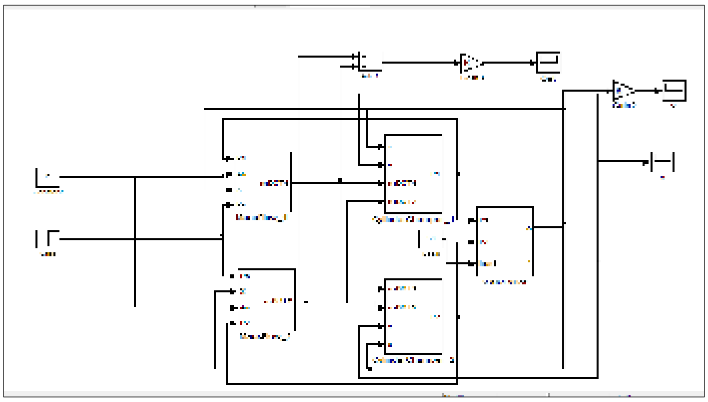


**Figure 5.** Simulation model of nonlinear mathematical model of the conventional Pneumatic actuator without bridging valve

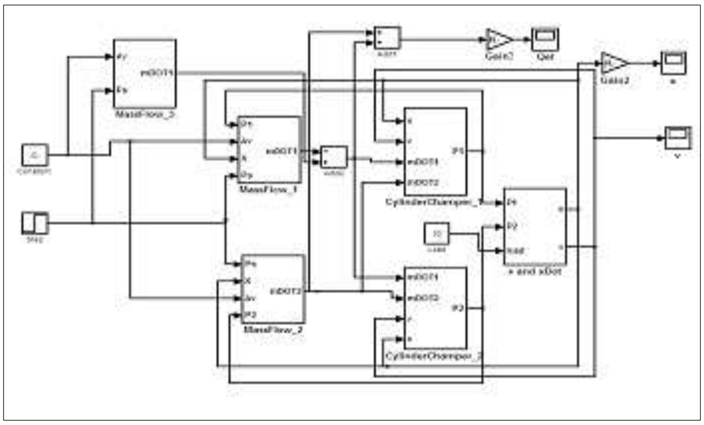



**Figure 6.** Simulation model of nonlinear mathematical model of the conventional Pneumatic actuator bridging valve

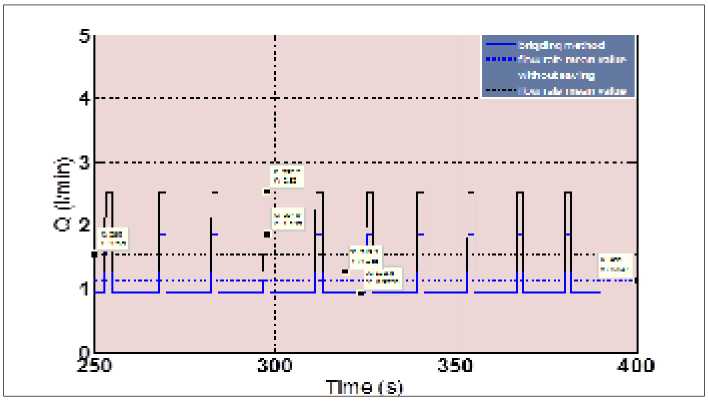



**Figure 7.** Air consumption with and without bridging valve (5bar)

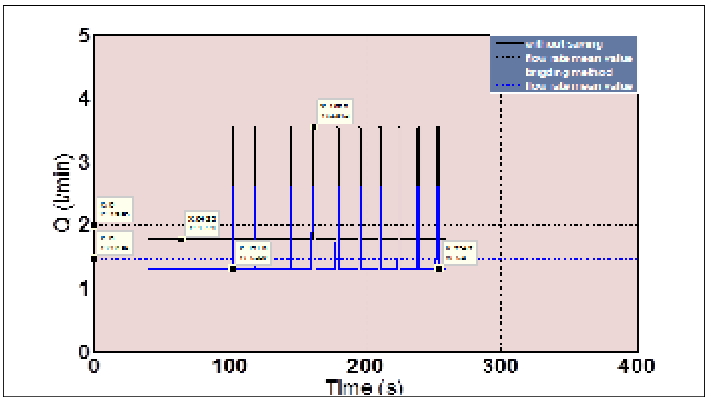



**Figure 8.** Air consumption with and without bridging valve (7bar)

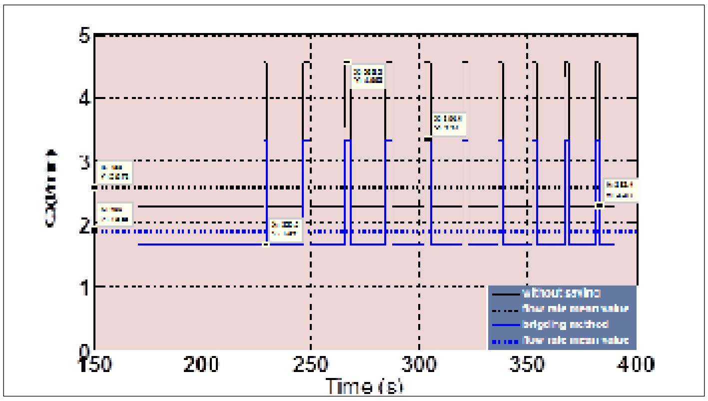
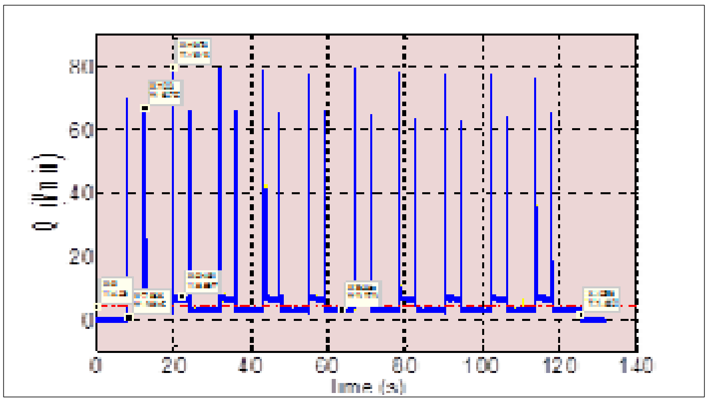



**Figure 9.** Air consumption with and without bridging valve (9bar)

Experimental results

For a ten stroke with stroke length 200 mm and pressure supply5, 7 , and 9 bar at constant average velocity and ambient temperature with vertical load 50N.To reduce flow rate consumption in both extends and retracts overshoot, the pressure difference between pressure line and pressure supply could be reduced. One option is to add one way directional control valve between extending pressure line and retracting pressure line and adding another in the opposite direction circuit No2.In extending stroke when the piston arrives at execution position the flow cutoff then one way valve opens to pass flow from extending line to retracting line at a specific time period. This time period depends on the amount of pressure difference between the two sides when the pressures between the two lines are equal the valve closes and retracting pressure supply line opens and repeat the same procedure at retracting execution position. Figure 10 illustrates the difference in the flow rate consumption by using circuit No (1 and 2) in which savings concentrated only on reducing altitude of extending and retracting overshoot for example the 5 bar pressure supply, constant velocity, ten strokes, without load, the maximum height of extending overshoot for ten strokes in two cases are (79.74, 59.2) l/min respectively. While in retracting overshoot 66.75 (l/min) and 44.27 (l/min), flow rate drop from (6.897 to 6.045) l/min for extending stroke and from (3.273 to 2.53) l/min which lead to reduce mean flow rate from (4.29 to 3.263) l/min. This difference saved air consumption in both overshoot types 28.3% and 16% related to total air consumption but at the expense of increasing operation time period by amount equals to 17.5 second, which represents the time required for equivalent pressure between the extending and retracting line at the end of stroke which depends on the supply pressure.

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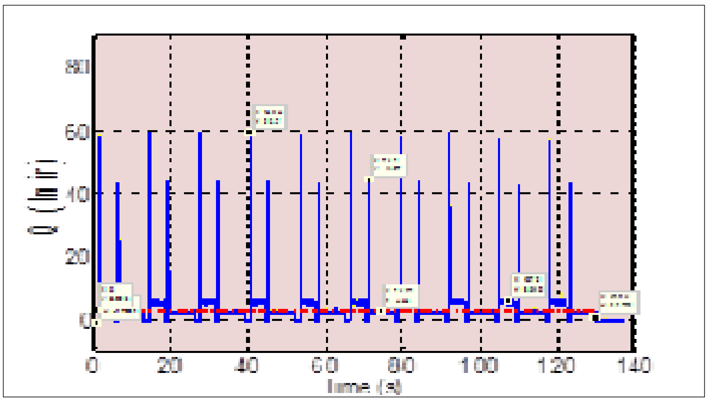
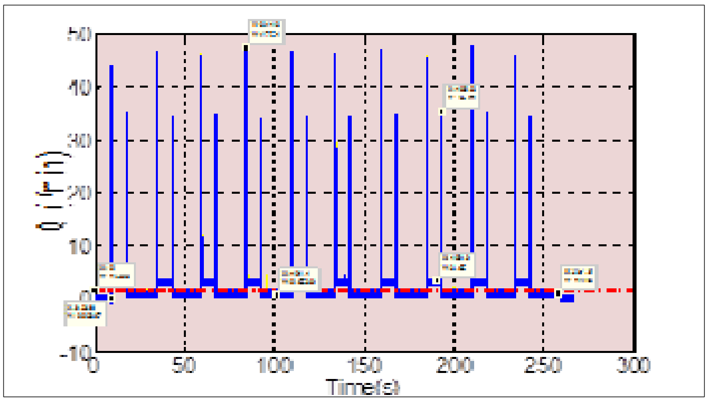



Without saving

**Figure 10.** Flow rate change with time (5 bar)

Figure 11shows the difference in saving of flow rate consumption at variable supply pressure. Where, a flow rate saving at (7 and 9) bar in both extending and retracting overshoot are equal to 21.3% and 27% respectively. The mean air flow rate decreased 0.426 l/min to save 23.44% in 7 bar while 0.967 l/min to save 23.5% in 9 bar, This led to the conclusion that the increase in pressure supply increases efficiency saving. This is because the increased pressure supply leads to increased pressure accumulated at the end of the stroke, and therefore needs high flow rate to overcome this pressure from the other side.

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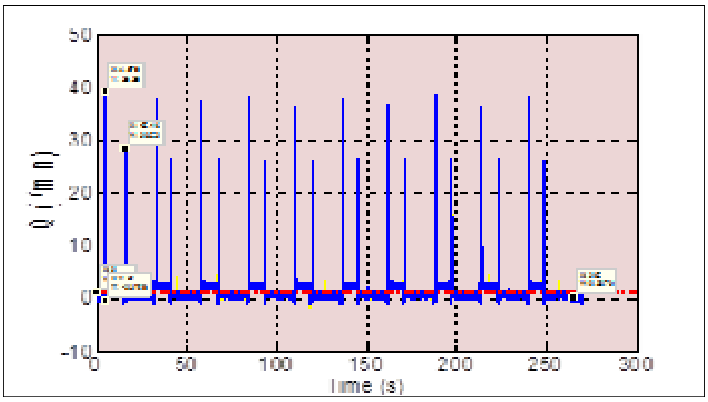


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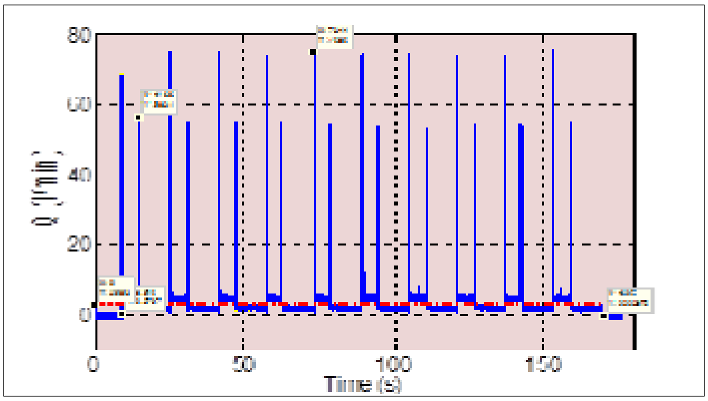


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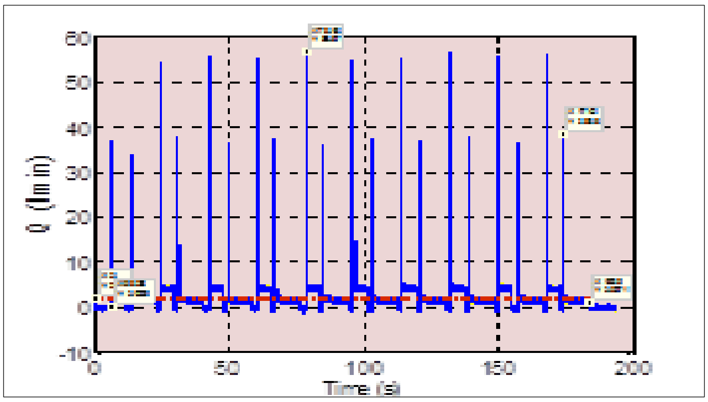


With saving

**Figure 11.** Air consumption with and without bridging valve (7&9) bar

CONCLUSION

The preceding methods of providing pneumatic cylinders are such that the use of compressed air from the cylinder chambers is always released into the atmosphere. Energy losses that have emerged during releasing compressed air into the atmosphere at the end of the process, and change of direction of movement cylinder rod, were important. The presented way of restoring energy of compressed air through the bridge double acting cylinder chamber shows good characteristics. the forgoing discussion in this paper shows that the system performance and energy efficiency can be significantly improved. Evidence of these comes from both simulation and experimental studies. Theoretical study for a conventional pneumatic system shows that maximum air saving for bridging method at 9 bar 27%. Experimentally it's concluded that most of air saving methods occurs at expense of time increment. One of the important observations derived from the study is the importance of response time at the end of the stroke.

Delay signal time lead to losses in air energy at both extending and retracting side because in drive at extending side unuseful pressure accumulate in this delay time which lead to increase air consumption at extending stroke as well as increasing required pressure to overcome it at retracting stroke and hence increase flow rate overshoot. Experimentally maximum air saving for bridging method at 9 bar is 23.5%.

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