The Effect of Hydraulic Accumulator on the Performance of Hydraulic System

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ABSTRACT

The purpose of this paper is to depict the effect of adding a hydraulic accumulator to a hydraulic system. The experimental work includes using measuring devices with interface to measure the pressure and the vibration of the system directly by computer so as to show the effect of accumulator graphically for real conditions, also the effects of hydraulic accumulator for different applications have been tested. A simulation analysis of the hydraulic control system using MATLAB.R2010b to study was made to study the stability of the system depending on the transfer function, to estimate the effect of adding the accumulator on stability of the system. A physical simulation test was made for the hydraulic system using MATLAB to show the effect of the accumulator when it's connected to the system for different parameters and compare it with a PID controller. The hydraulic system has been simulated and tested using Automation Studio (AS) to measure different data such as the linear speed of hydraulic cylinder and the effect of connecting the accumulator to the system. All the results showed that the hydraulic accumulator has a great benefits and a large enhancement to the hydraulic system.

Keywords: hydraulic system, accumulator, directional valve, relief valve, energy storage

تأثير مجمع الضغط الهيدروليكي على أداء منظومه هيدروليكيه

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الخلاصة

في هذا البحث تم دراسة تأثير اضافة مجمع الظغط على مختلف التطبيقات. محاكاة تحليلية لنظام التحكم الهيدروليكي للمنظومة باستخدام برنامج (R2010b .MATLAB) لدراسة استقرارية النظام أثناء العمل اعتمادا على دالة الانتقال وذلك لتقدير تأثير إضافة مجمع الضغط الهيدروليكي على استقرار النظام. وقد تم عمل محاكاة للمكونات الفيزيائية للنظام باستخدام برنامج(MATLAB). لإظهار أثر مجمع الضغط عندما يكون متصلا بها ولعدة قيم مستنتجه ومقارنتها في حالة اضافة وحدة تحكم للمنظومة, تحليل و محاكاة النظام الهيدروليكي باستخدام برنامج المحاكاة Automation Studio ولنفس المنظومة التي تم اختبارها عمليا حيث تم بناءها واختبارها على جهاز الحاسوب لقياس بيانات مختلفة مثل سرعة المكبس الهيدروليكي الخطية ودراسة تأثير ربط مجمع الضغط الهيدروليكي على النظام. وأظهرت النتائج أن مجمع الضغط الهيدروليكي لديه اهميه في تعزيز وتحسين أستقراية النظام الهيدروليكي .

كلمات رئيسيه: منظومه هيدروليكيه, مجمع ضغط, صمام اتجاهي, صمام تنفيس, خزان للطاقه.

1. INTRODUCTION

In the modern world of today, hydraulics plays a very important role in the day-to-day lives of people. Any device operated by a hydraulic fluid may be called a hydraulic device, but a distinction has to be made between the devices which utilize the impact or momentum of a moving fluid and those operated by a thrust on a confined fluid, i.e. by pressure. This leads us to the subsequent categorization of the field of hydraulics into:

- Hydrodynamics
- Hydrostatics, **Ravi 2005.**

The most susceptible components in any hydraulic system are the pump, valves, rotary actuators (motors) or linear actuators (cylinders), reservoirs and connection lines. In addition, some systems that have a **hydraulic accumulator, Arthur 2006.** A hydraulic accumulator is a device which stores pressurized hydraulic fluid. That way, the pump does not have to be powerful enough to cope with a sudden surge in demand. Instead, it can keep steadily pumping hydraulic fluid and rely on the accumulator to provide extra hydraulic fluid when it is needed.

Accumulators can perform several functions for hydraulic systems such as:

- Supply oil for high transient flow demands when pump can't keep up.
- Help to reduce pump ripple and pressure transients.
- Absorb hydraulic shock waves (due to valve closures or actuators hitting stops).
- Used as a primary power source for small (low demand) systems.
- Help system to accommodate thermal
- Compensate for system leakage expansion of the fluid **,Isaiah 2009.**

There are three main types of accumulators as shown in **Fig .1** bladder, diaphragm bladder and piston The choice of accumulator to use in a given application depends on required speed Of accumulator response, weight, reliability and cost.

The bladder accumulator is commonly used in hydraulic systems because of the main advantages of a bladder accumulator such as, fast acting, no hysteresis, not susceptible to contamination and consistent behavior under similar conditions. Hence, bladder accumulators are the best choice for pressure pulsation damping. **web1,2012.** So we choose this type of accumulator to be used in my experimental branch test. **Yudong X., Y. 2009,** presented a dynamic design of electro -hydraulic control valve with accumulator based on a physical simulation model, he found That velocity oscillation of the electro-hydraulic actuator results from the inter coupling effect of the flow pressure pulsation. In order to reduce the velocity overshoot of the hydraulic actuator, an accumulator can be used to absorb the pressure pulsations to weaken the inter coupling effect. **Xiangdong, k., 2010,** presented a simulation and experimental study on the effects of adding accumulator to the fast forging hydraulic control system. By using the mathematical model of fast forging system and do simulation study, **Minav,T.A et.al. 2012,** presented how to use the hydraulic accumulator as a energy storage device that recovered from an electro-hydraulic forklift truck. The braking energy can be stored in the hydraulic accumulator for a long time, and the efficiency of the system increase from 5% to 32%.

2. THEORETICAL MODELING AND SIMULATION

Theoretical model for the main parts of the hydraulic system will be studied, to determine the transfer function for the system.i-Four-way directional valve controlled cylinder modeling:

The directional valves are one types of the spool valve; the general relations and performance for the valve have been derived and studied.

Considering power matching of hydraulic cylinder and directional valve, **Fig. 2** shows a schematic diagram was made using (Auto CAD 2012) of a valve-piston combination. If orifices area of slide valve is matching and symmetrical as shown in **Fig.2,** with zero lap, then the flow pressure equation in the valve is: **Herbert, 1967.**

$$Q_L = C_d A_1 \sqrt{\frac{1}{\rho}(P_S - P_L)} - C_d A_2 \sqrt{\frac{1}{\rho}(P_S + P_L)} \quad (1)$$

$$Q_S = C_d A_1 \sqrt{\frac{1}{\rho}(P_S - P_L)} + C_d A_2 \sqrt{\frac{1}{\rho}(P_S + P_L)} \quad (2)$$

Where, the sum of the line pressures **(P_1)** and **(P_2)** is approximately equal to the supply pressure **(P_s)**. and the same for the supply flow. **Valdmier, 2006.** Applying the continuity equation to each chamber of the cylinder yields,

$$Q_1 - C_{ip}(P_1 - P_2) = \frac{dV_1}{dt} + \frac{V_1}{\beta_e}\frac{dP_1}{dt} \quad (3)$$

$$C_{ip}(P_1 - P_2) - C_{Ep}P_2 - Q_2 = \frac{dV_2}{dt} + \frac{V_2}{\beta_e}\frac{dP_2}{dt} \quad (4)$$

The equations above solved simultaneously gives,

$$Q_L = A_P S X_P + C_t P_L + \frac{V_t}{4B_e} S\, P_L$$

$$= K_q X_V - K_C P_L \quad (5)$$

Now by applying Newton's second law to the forces on the piston, the resulting force equation Laplas transformed, is

$$F_g = A_P P_L = M_t S^2 X_P + B_P S X_P + K X_P + F_L \quad (6)$$

$$P_L = \frac{F_g}{A_P} = \frac{M_t S^2 X_P}{A_P} + \frac{B_P S X_P}{A_P} + \frac{K X_P}{A_P} + \frac{F_L}{A_P} \quad (7)$$

Now substitute Eq. (7) into Eq. (5) gives,

$$A_P S X_P + C_{tp}\left[\frac{M_t S^2 X_P}{A_P} + \frac{\beta_e S X_P}{A_P} + \frac{K X_P}{A_P} + \frac{F_L}{A_P}\right] + \frac{V_t}{4B_e} S\left[\frac{M_t S^2 X_P}{A_P} + \frac{\beta_e S X_P}{A_P} + \frac{K X_P}{A_P} + \frac{F_L}{A_P}\right] \quad (8)$$

$$= K_q X_V - K_{ce}\left[\frac{M_t S^2 X_P}{A_P} + \frac{\beta_e S X_P}{A_P} + \frac{K X_P}{A_P} + \frac{F_L}{A_P}\right]$$

Where, $k_{ce} = k_{\mathrm{c}} + c_{\mathrm{ip}} + c_{ep}/2$

The hydraulic natural frequency

$$\omega_h = \sqrt{\frac{4\beta_e A_P{}^2}{V_t M_t}}(rad/\sec) \tag{9}$$

And the damping ratio, is

$$\delta_h = \frac{K_{ce}}{A_P}\sqrt{\frac{\beta_e M_t}{V_t}} + \frac{B_P}{4A_P}\sqrt{\frac{V_t}{\beta_e M_t}} \tag{10}$$

Now substitute Eqs. (9) and (10) into Eq. (8) and simplify, the transfer function of the valve controlled cylinder is given by :

$$G_{v,c}(s) = \frac{X_P}{X_v} = \frac{\frac{K_q}{A_P}}{S(\frac{S^2}{\omega_h{}^2} + \frac{2\delta_n}{\omega_h}S + 1)} \tag{11}$$

3.THE LONG PIPE LINE MODELING

The precise model of fluid transmission pipeline is a dissipative friction model which is related to the frequency, and it includes a complex Bessel function and a hyperbolic function, as a result, it is very difficult to get accurate analytical solutions. Therefore, in engineering, the influences of pipeline for hydraulic system dynamic dynamic behavior are always neglected, which is unfavorable for system control under the situation of long pipeline. Considering fluid motion feature and physical properties in pipeline such as mass, damping and pressure, **Jiang, 2006**. So that the simple mass-spring-damping dynamic model can be used to simulate the liquid in pipeline. The model is shown in **Fig.3.**(m_o) denotes liquid mass,(B_o) damping coefficient, (K_o)spring rate, ($F_{(t)}$)external force and ($X_{(t)}$) displacement. The transfer function model is derived as follows,

$$G_p(s) = \frac{\frac{1}{K_0}}{\frac{s^2}{\omega_0} + \frac{2\delta_0}{\omega_0}s + 1} \tag{12}$$

Where, **(ω_o**) is natural frequency of long pipeline, $\omega_o = \sqrt{\frac{K_o}{m_o}}$**,** (**$\delta_o$)** damping ratio,

$\delta_o = \sqrt{(B_o{}^2}/4m_oK_o)$.

4. PRESSURE RELIEF VALVE MODELING

Fig. 4 shows a schematic diagram was made using**, Auto CAD, 2012**. of a single-stage pressure control valves (relief valve). The equations describing spool motion**, Herbert, 1967.** is,

$$F_1 - AP_c = M_v s^2 x + K_e x \tag{13}$$

And, the linearized continuity equations at the sensed pressure chamber being controlled are

$$Q_c = K_1(P_s - P_c) = \frac{V_c}{\beta_e}\, s\, P_c - A_s x \tag{14}$$

$$Q_P - Q_L - K_l P_s - K_c P_s - K_1(P_s - P_c) + K_q x = \frac{V_t}{\beta_e} s\, P_s \tag{15}$$

Where,

$K_c = \frac{\partial Q}{\partial P} = \frac{C_d\, w\, x_0 \sqrt{2/\rho}}{\sqrt{P_{s0} - P_{R0}}} = \frac{Q_0}{2(P_{s0} - P_{R0})}$ = Flow-pressure coefficient of main orifice.

$K_q = \frac{\partial Q}{\partial x} = C_d\, w \sqrt{\frac{2}{\rho}(P_{s0} - P_{R0})}$ = flow gain of main orifice.

Now, solving Eq. 14 for **(P_c)**, and substituting into Eq. (15) yields after some manipulation, the transfer function of pressure control valve **(relief valve)** is,

$$G_r(s) = \frac{x}{F_1} = \frac{\frac{1}{K_e}\left(1 + \frac{s}{\omega_1}\right)}{\frac{s^3}{\omega_m^2\, \omega_1} + \frac{s^2}{\omega_m^2} + \left(\frac{1}{\omega_1} + \frac{1}{\omega_2}\right)s + 1} \tag{16}$$

Where,

$\omega_1 = \frac{\beta_e K_1}{V_c}$ Break frequency of sensing chamber.

$\omega_2 = \frac{\beta_e K_{ce}}{V_t}$ Break frequency of main volume.

$K_e = K_c + K_l$ Equivalent flow-pressure

$\omega_m = \sqrt{\frac{K_e}{M_v}}$ Mechanical natural frequency

5. ACCUMULATOR MODELING

According to gas law **web1,2012.**

$$P_{xo} v_{xo}^{\ n} = P_a v_a^{\ n} \tag{17}$$

So that $P_a = P_{xo} v_{xo}^{\ n} v_a^{\ -n}$ (18)

When the state of accumulator changes from condition **o** to **a**, **Fig.5** the pressure increment is:

$$\Delta p = P_a - P_{xo} = P_{xo} v_{xo}^{\ n} v_a^{\ -n} - P_{xo} \tag{19}$$

Now for accumulator modeling one can regard accumulator as a gas spring aerodynamic damping model **{9},** and the state can be considered adiabatically, ***n*=1.4**. So that the equation of accumulator is,

$$(P_s - P_a) = \frac{m_a \ddot{V}_a + B_a \dot{V}_a + C_a \dot{V}_a + K_a V_a}{A^2} \tag{20}$$

And, the resulting equation Laplas transformed, is

$$P_s - P_a = \frac{m_a\, s^2 x + B_a\, s\, x + C_a\, s\, x + k_a\, x}{A^2} \tag{21}$$

$$G_a(s)\, \frac{x}{P_s - P_a} = \frac{A_a^{\,2}}{m_a\, s^2 + B_a\, s\; + C_a\, s + k_a} \tag{22}$$

Now, by combining Eqs. (11), (12), (16) and (22), the Transfer function for the hydraulic system can be derived:

1- Transfer function for the hydraulic system with accumulator for short pipe.

$$G_{(s)} = \frac{x_p}{P_s} = \frac{\frac{K_q}{A_p}\,\frac{1}{K_e}\left(1+\frac{s}{\omega_1}\right) A_a^{\,2}}{s\left(\frac{s^2}{\omega_h^{\,2}}+\frac{2\delta_h}{\omega_h}s+1\right)\left(\frac{s^3}{\omega_m^{\,2}\,\omega_1}+\frac{s^2}{\omega_m^{\,2}}\right)\left(\frac{1}{\omega_1}+\frac{1}{\omega_2}\right)s\left(m_a\, s^2+B_a\, s\; +C_a\, s+k_a\right)} \tag{23}$$

2- Transfer function for the hydraulic system without accumulator for short pipe

$$G_{(s)} = \frac{x_p}{P_s} = \frac{\frac{K_q}{A_p}\,\frac{1}{K_e}\left(1+\frac{s}{\omega_1}\right)}{s\left(\frac{s^2}{\omega_h^{\,2}}+\frac{2\delta_h}{\omega_h}s+1\right)\left(\frac{s^3}{\omega_m^{\,2}\,\omega_1}+\frac{s^2}{\omega_m^{\,2}}\right)\left(\frac{1}{\omega_1}+\frac{1}{\omega_2}\right)s} \tag{24}$$

3- Transfer function for the hydraulic system with accumulator for long pipe.

$$\boldsymbol{G}_{(s)} = \frac{x_p}{P_s} = \frac{\frac{K_q}{A_p}\,\frac{1}{K_0}\,\frac{1}{K_e}\left(1+\frac{s}{\omega_1}\right)}{s\left(\frac{s^2}{\omega_h^{\,2}}+\frac{2\delta_h}{\omega_h}s+1\right)\frac{s^2}{\omega_0}+\frac{2\delta_0}{\omega_0}s+1\left(\frac{s^3}{\omega_m^{\,2}\,\omega_1}+\frac{s^2}{\omega_m^{\,2}}\right)}\,..$$

$$..\frac{A_a^{\,2}}{\left(\frac{1}{\omega_1}+\frac{1}{\omega_2}\right)s\left(m_a\, s^2+B_a\, s\; +C_a\, s+k_a\right)} \tag{25}$$

4- Transfer function for the hydraulic system without accumulator for long pipe. $G_{(s)} = \frac{x_p}{P_s} =$

$$\frac{\frac{K_q}{A_p}\,\frac{1}{K_0}\,\frac{1}{K_e}\left(1+\frac{s}{\omega_1}\right)}{s\left(\frac{s^2}{\omega_h^{\,2}}+\frac{2\delta_h}{\omega_h}s+1\right)\frac{s^2}{\omega_0}+\frac{2\delta_0}{\omega_0}s+1\left(\frac{s^3}{\omega_m^{\,2}\,\omega_1}+\frac{s^2}{\omega_m^{\,2}}\right)\left(\frac{1}{\omega_1}+\frac{1}{\omega_2}\right)s} \tag{26}$$

Fig. 6 shows a simple block diagram of the system.

6. RESULT AND DISCUSSION

The hydraulic accumulator has many benefits to the hydraulic system and some of these benefits which have been tested experimentally are:

1-The use of accumulator as an storage device energy

Fig.7 represents the time required the hydraulic cylinder to extend for both cases with and without accumulator and it's clearly shows that the time is decreased when the accumulator is used. The time required the hydraulic cylinder to extend at system pressure **(40 bar)** is **(2.26 sec),** without using the accumulator and at the same pressure but with connecting the accumulator to the system the time is **(1.72 sec)**. the decreasing in time required is about **(33%)**.the same result shown at system pressure

(**30, 20 and 10** bar). **Fig 8** represents the speed of extending of hydraulic cylinder with pressure for both cases with and without accumulator, and it clearly shows that the speed increased when the accumulator connected to the system. This has been happened because the potential energy stored in accumulator

2- The use of accumulator as a leakage compensator:

The accumulator acts as a compensator, by compensating for losses due to internal or external leakage that might occur during the operation. Also pressure losses happened due to the friction in pipes and connections, also due to the increase in oil temperature which affect on the performance of the hydraulic system. At pump pressures of **40, 30, 20 and10 bars,** the decreasing in pressures drop are about **13.3%, 13 %, 12.7% and 12.5%**respestivly ,as shown in **Fig.9.**

3-The use of accumulator to cushion the vibration of the system:

In this test the effect of adding the accumulator on the vibration of the system have been studied. **Figs. 10** and **11** show the velocity and acceleration of vibration with and without using the accumulator .At system pressure **40 bar**, the velocity and acceleration of vibration is **1.6** $\boldsymbol{mm}$**/sec** and **4.3**$\boldsymbol{m/sec^2}$ respectively, So it's clearly that the accumulator cushions the vibration of the system, also the same results at system pressures **(30, 20 and 10 bar)**. Also a graphical test using vibration meter with interface software **(sw-u801wn)** by **(lutron company)**. was made at point before the relief valve with system pressure **10 bars, Figs. 12** and **13**. From the results above it's clearly that the accumulator reduced the vibration of the system.

4-The use of accumulator as shock absorber:

One of the most important industrial applications of accumulators is in the elimination of high-pressure pulsations or hydraulic shocks. **Quan, 2007.** To test this Phenomena using a graphical chart display using the pressure meter with interface software **(sw-u801 wn)** to view the behavior of pressure at the cylinder when it's suddenly stops at the end of the stroke. The set pressure is **30 bars,** as shown in **Fig.14.** When the cylinder reached the end of the stroke, the stop without using the accumulator is suddenly happened and very fast and causes a hydraulic line shock, but with using the accumulator and from **Fig.15.** it can be seen that the cylinder stops at the end of the stroke fluently. So the benefits of adding accumulators to the system are to damp pressure spikes from pumps

.

7. THE SYSTEM STABILITY TEST USING MATLAB PACKAGE V1.1(R2010B):

Fig .16 represents the Bode diagram for the system without connecting the accumulator for short pipeline, and the result shows that the system is unstable because of the pulsation at the system response and the over shoot is big. **Fig.17** represents the Bode diagram for the system with connecting the accumulator for the short pipeline, and the result shows that the system is stable with phase margin of **87.2 deg** at frequency of **0.06 rad/sec .**Also for a long pipeline **Fig.18** represents the bode diagram for the system without connecting the accumulator and the result shows that the hydraulic system is unstable. **Fig.19** represents the bode diagram for the system with connecting the accumulator, and the result shows that the system is stable with phase margin **of 180 deg.** at frequency of **10.7 rad/sec**.

The physical simulations for the hydraulic system using Matlab V7.11 (R2010b):
A simulation **Fig.20** shows the effect of the accumulator when it's connected to the system for different parameters like cylinder pressure, cyl- inder load, and displacement of cylinder, and the simulated results are shown in **Figs. 21, 22 and 23** respectively. To compare the effect of accumulator and a PID controller to the system the effect of PID controller are shown in **Figs.24, 25 and 25** which makes a self tuner to the system. The self - tuned parameters are found using MATLAB as **P= 0.9, I= 1.2 and D= 0.1.** But when we connected the accumulator the system become more stable and less fluctuation, as shown in **Figs. 7, 28 and 29**. From the figures above it's clearly that the accumulator makes the system steadier than the PID controller; for this case.

The Simulation analysis with automation studio package V5.2
The hydraulic system has been built with Automation Studio Package V5.2**(AS 2008)** to measure different data, such as the linear speed of hydraulic cylinder and to study the effect of connecting the accumulator to the system. **Figs . 30** and **31.** show a compression between the effect of connecting or disconnecting the accumulator on the linear speed at set pressure **of 40 bars**. The results of simulation shows that the liner speed increases by **(31%)** and the response become much faster. Also to estimate the effect of using the accumulator as shock absorber, **Fig. 32** shows the response of the cylinder pressure at the end of stroke. It's clearly that the hydraulic cylinder stops suddenly and so fast which causes a pressure shock at the cylinder, but when the accumulator is connected to the system the rise of the pressure until the cylinder reached to the end of the stroke become more smoothly as shown in **Fig.33** the set pressure is 10 bar.

8. CONCLUSIONS
The present, theoretical simulation analysis and experimental investigation show several conclusions these conclusions can be summarized as below:-
1-The experimental tests showed that the performance of the hydraulic system clearly improvement by connecting the accumulator to the system and the results showed that the accumulator can be used in a wide variety of applications such as:

a- Energy storage
b- Leakage compensation
c- Cushion the vibration
d- Emergency operation
e- Shock absorption
It's found that the effect of accumulator as an energy storage is the most common application than the others.

2- A model equation for the hydraulic system combination has been derived. The theoretical simulation analysis using the bode diagram showed that the system become stable with connecting the accumulator.

3- A physical simulation test using *Matlab V7.11 (R2010b)* was made for the hydraulic system to show the effect of the accumulator when it's connected to the system for the different parameters, the results showed that when the accumulator is connected, the system become more stable and less fluctuation also than the PID Controller, for this case.

4- The practical results and simulation using (AS) program are clearly convergence. This leads us to the possibility of using this program for testing and analysis and design of any hydraulic system.

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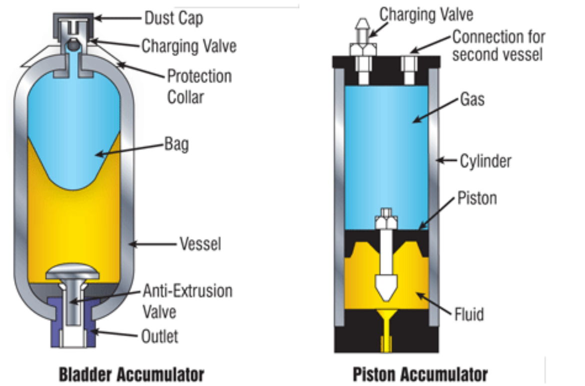
NOMENCLATURE

A_1 ,A_2	orifice area gradient
A_p	piston Surface area
A_a	area of accumulator plan
A_s	Spool end area
B_p	viscous damping coefficient of piston
P	proportional gain
P_L	load
P_c	chamber pressure
P_s,P_a:	pressure at system and condition a
P_1, P_2	pressure of Port A and B
P	pressure difference
Q_1 ,Q_2	the flow through proportional valve
$Q_{p\ ideal}$	ideal flow rate of the hydraulic pump
$Q_{p\ actual}$	actual flow rate of the hydraulic pump
Q_L,Q_s	load and system flow
c_a	gas damping factor
C_d	discharge coefficient of the valve
C_{ip} , C_{ep}	internal and external leakage coefficient
D_p	volume displacement rate
D	derivative gain
F_L	load force on piston
F_g	force generated by piston
I	integral gain
k_q	valve flow gain
k_e	equivalent spring rate
k_a	gas stiffness factor
k_o	spring rate of pipe
K_c	valve pressure gain
K_{ce}	total flow-pressure coefficient
K_1	flow-pressure coefficient of restrictor
K_l	leakage coefficient
m_a	liquid equivalent mass
M_t	total mass of piston
M_v	spool mass
N	rotational speed of pump
t	time
V_t	total hydraulic oil volume in cylinder
V_1 , V_2	forward and return chamber volume
V_c	sensing chamber volume
V_o	gas volume
V_a	volume of accumulator
X_p	piston displacement
X_v	valve displacement
x	spool displacement

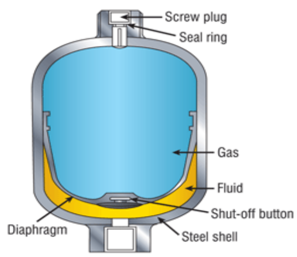
GREEK SYMBOLS

δ Damping coefficient

δ_0 Damping ratio of pipe



Bladder Accumulator **Piston Accumulator**



Diaphragm Accumulator

Figure 1. The main types of accumulator**, Automation Studio, 2008.**

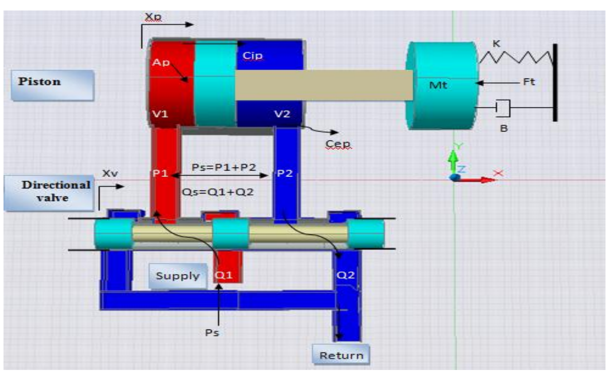


Figure2. Schematic of a valve-piston combination.

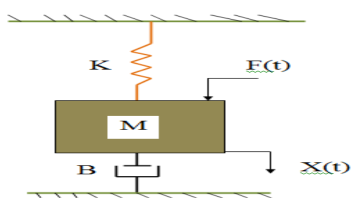


Figure3. Simulation model of liquid in pipeline, **Xiang dong, 2010**.

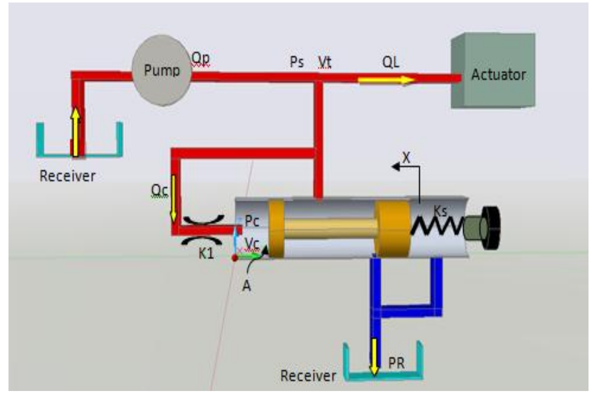


Figure4. Schematic of a single-stage pressure relief valve**.**

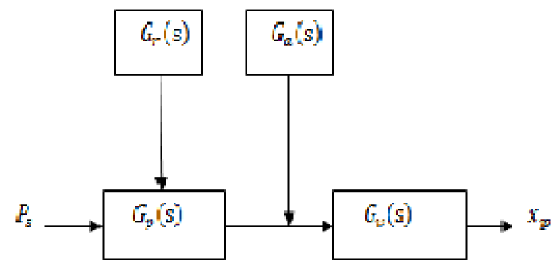


Figure5. Bladder type hydraulic accumulator, **Isaiah David, 2009.**

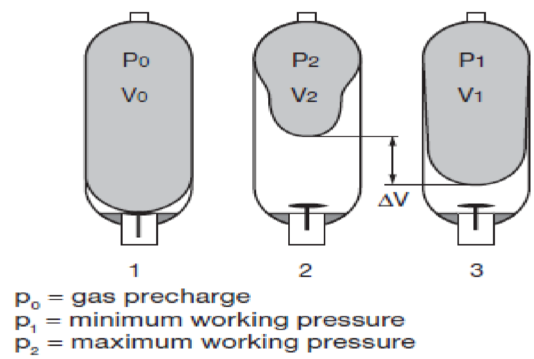


Figure.6 Simple block diagram for the hydraulic system.

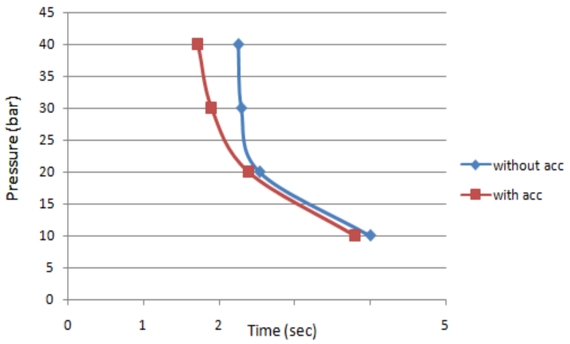


Figure7. The time required the hydraulic cylinder to extend for both cases with and without accumulator.

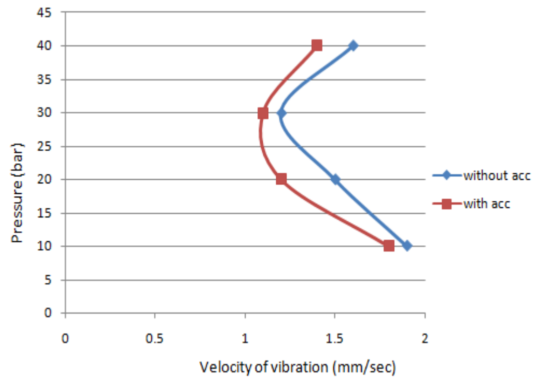


Figure10. The velocity of vibration for both cases, with and without connecting the accumulator.

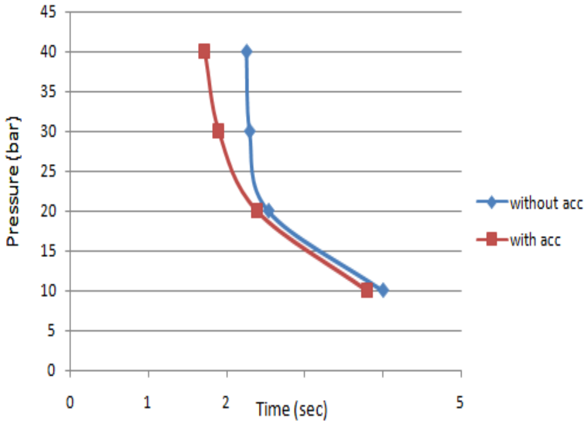


Figure8. The speed of extending of hydraulic cylinder for both cases with and without accumulator.

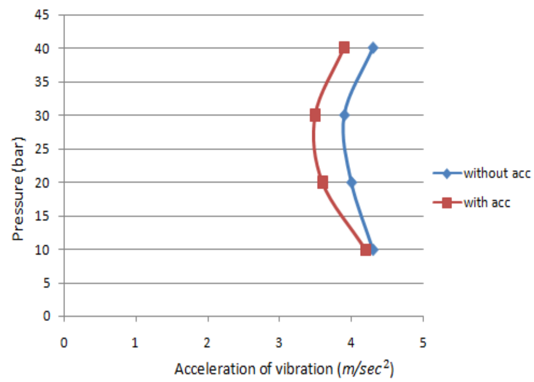


Figure11. The acceleration of vibration for both cases, with and without connecting the accumulator.

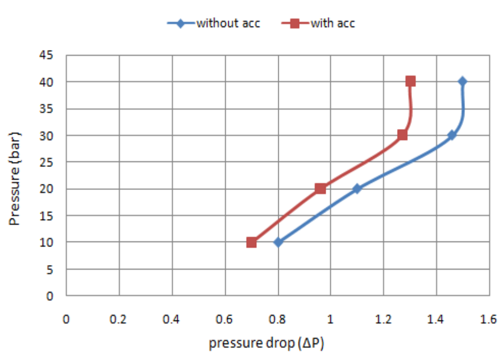


Figure9. The pressure drops across the hydraulic system.

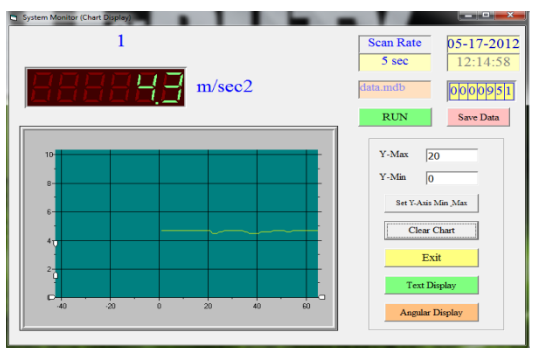


Figure12. Acceleration of vibration at set pressure (10) bar without connecting the accumulator.

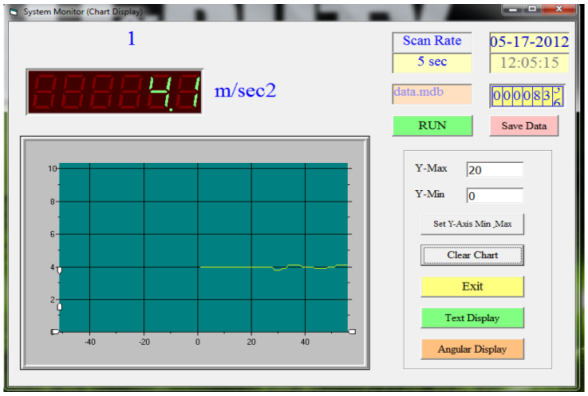


Figure13. Acceleration of vibration at set pressure (10) bar with connecting the accumulator**.**

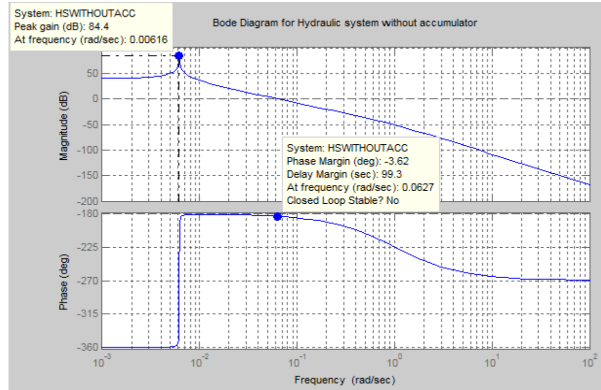


Figure16. The bode diagram for the (tf) of the hydraulic system without connecting the accumulator for short pipeline.

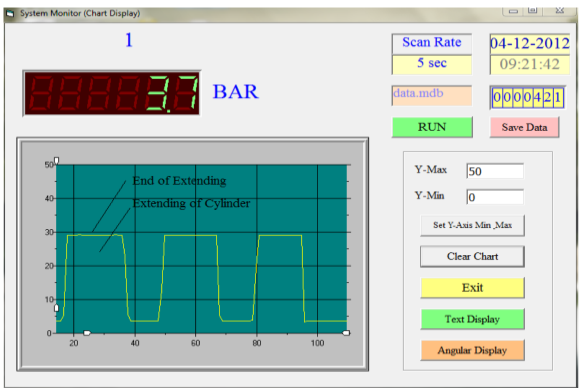


Figure14. Pressure measured at the cylinder without connecting the accumulator, the set pressure is 30bar.

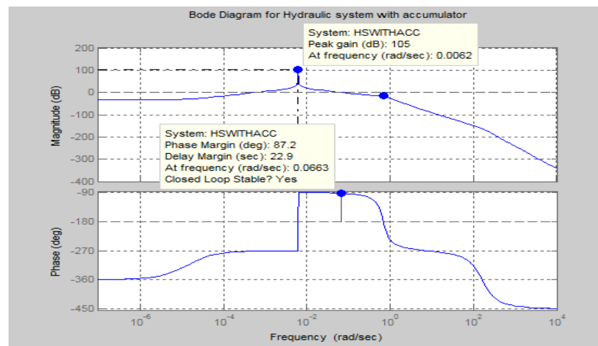


Figure17. The bode diagram for the (tf) of the hydraulic system with connecting the accumulator for short pipeline.

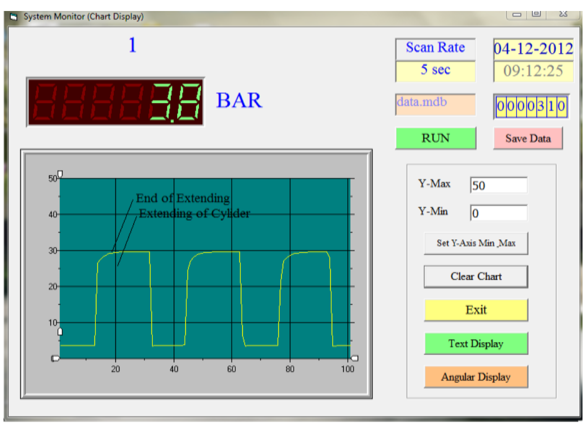


Figure15. Pressure measured at the cylinder with connecting the accumulator.

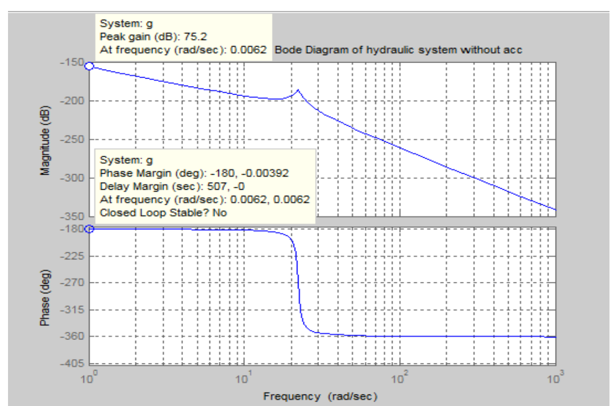


Figure18. The bode diagram for the (tf) of the hydraulic system without connecting the accumulator for long pipeline.

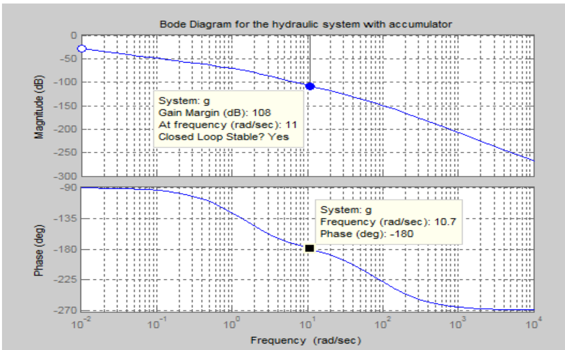


Figure19. The bode diagram for the (tf) of the hydraulic system with connecting the accumulator for long pipeline.

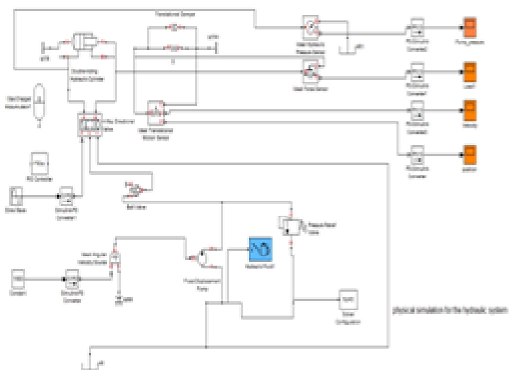


Figure20. Physical simulation for the hydraulic system.

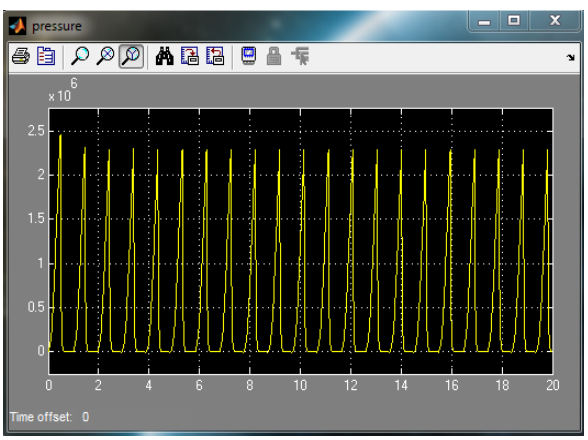


Figure21. The cylinder pressure of hydraulic system without the pid controller or the accumulator.

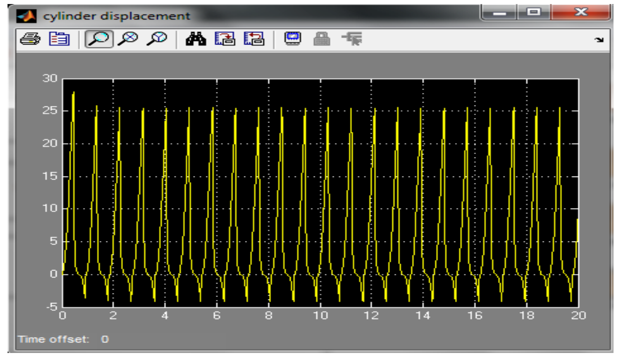


Figure22. The cylinder load of hydraulic system without the pid controller or the accumulator.

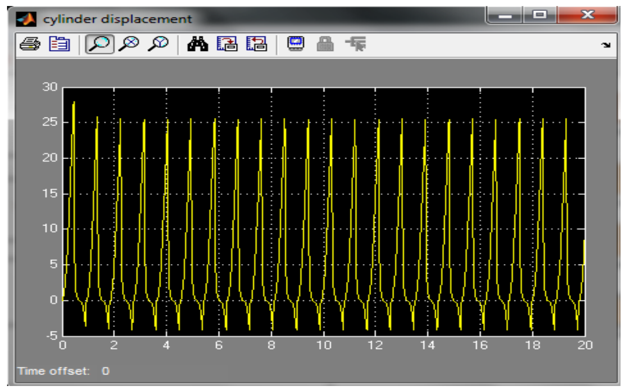


Figure23. The cylinder displacement without connecting the pid controller or the accumulator.

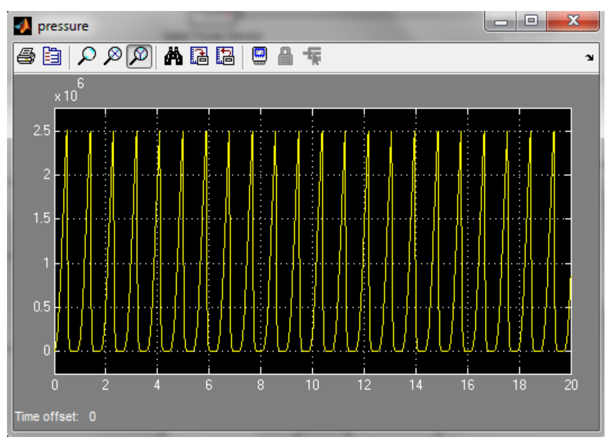


Figure24. The cylinder pressure of hydraulic system with connecting the pid controller.

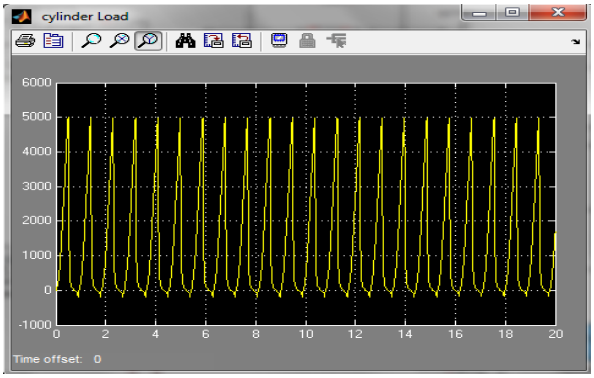


Figure25. The cylinder load of hydraulic system with connecting the pid controller.

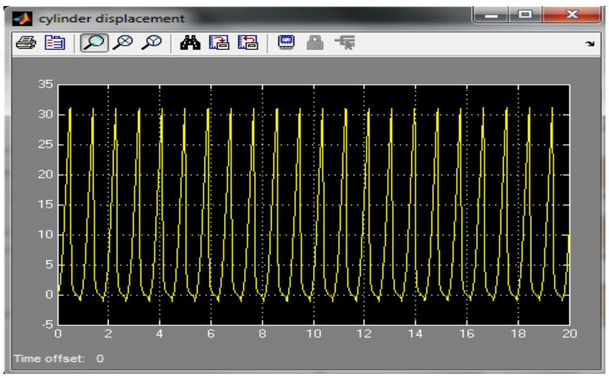


Figure26. The cylinder displacement of hydraulic system with connecting the pid controller.

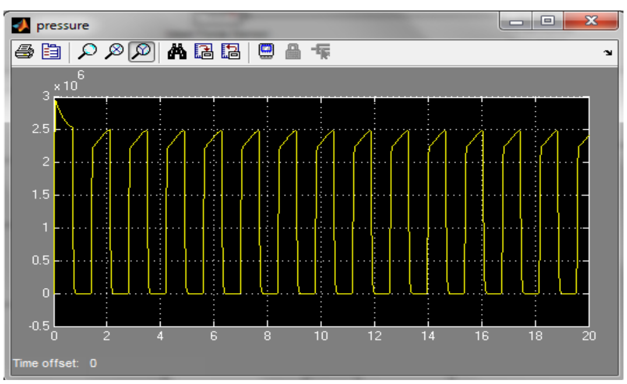


Figure27. The cylinder pressure of hydraulic system with connecting the accumulator.

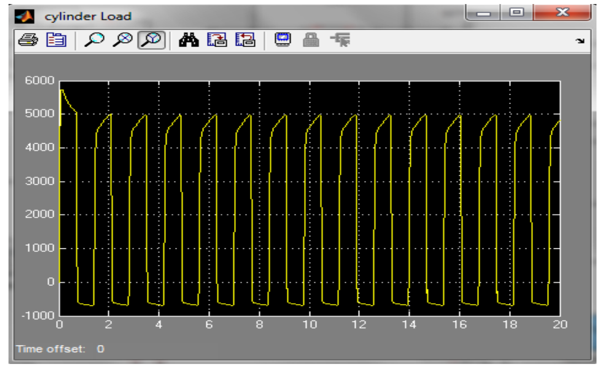


Figure28. The cylinder displacement of hydraulic system with connecting the pid controller.

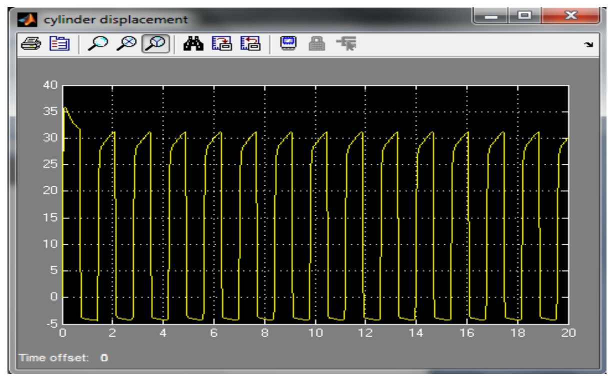


Figure29. The cylinder displacement of hydraulic system with connecting the accumulator.

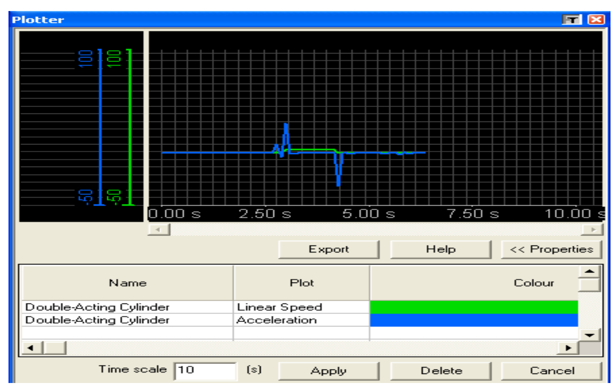


Figure30. The linear speed and acceleration of cylinder without connecting the accumulator using (as).

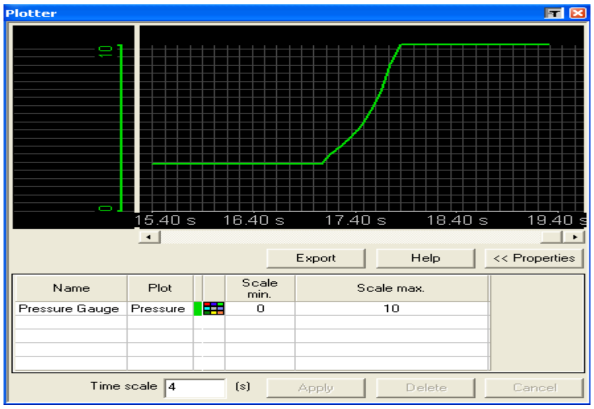


Figure31. The linear speed and acceleration of hydraulic with connecting the accumulator using (as **).**

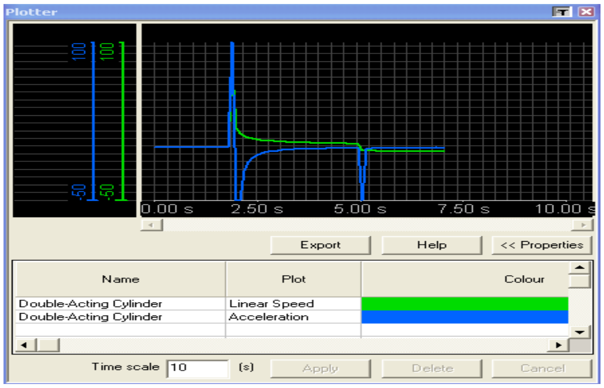


Figure32. The cylinder pressure at the end of stroke without connecting the accumulator using (as).

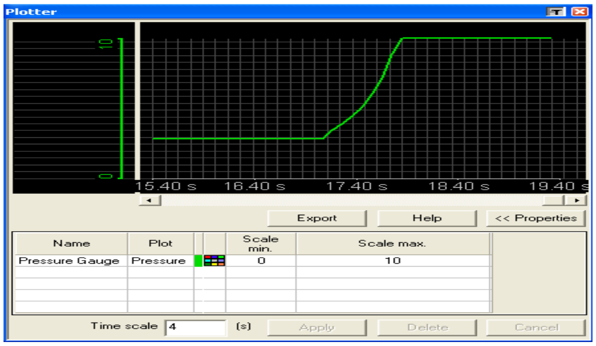


Figure 33. The cylinder pressure at the end of stroke with connecting the accumulator using (as).