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Investigation Vibration Damping in the Hydraulic Systems by Using an Accumulator

Abstract- *It is generally accepted that the vibration of fluid power systems considered to be one of the major problems that is normally occurred in the hydraulic system, which causes a noise and short life of its components. Accordingly, it should be reduced the efficiency as well as an increase leakage system. Hence, present study pays more attention to investigate a bladder an accumulator which was successfully added to the hydraulic system in order to reduce the vibrations that might be generated by the system and decelerate the actuator at the end stroke. Which variables are measured before relief and directional valve and at the linear cylinder body. It was found that the maximum percentage damping in vibration velocity at a position before relief and directional control valve and at cylinder body was 20%, 20.8% and 55% at 20 and 15 bar pressure supply respectively. Whereas, it was observed the acceleration was 11.3%, 12.5% and 50% at 40 bar pressure supply. Also, it was found that the piston begins decelerate gradually from distance 25cm in which equal to 1/6 of total stroke length with a period of time 5 seconds.*

Keywords: *Vibration, Damping, Hydraulic Systems, Accumulator.*

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1. Introduction

It is well known that the pumps are normally utilized as a fluid power source in the hydraulic system. However, one of the obstacles is the fluid flow pulsation, which is attributed to the pressure oscillation in the pumps. In fact, the periodical pulsation of fluid damage the components inside the hydraulic system as well as it could be reduced the surface life which is resulted to the main sound sources [1]. The degree of pressure oscillation of the hydraulic circuit can be reduced by utilizing an accumulator of appropriate capacity to properly position and install it in the system.

Control pressure surges in the hydraulic system and energy saving from the surges by using an appropriate accumulator of the hydraulic system was seccesssfully simulated by A.Kumar et.al [1]. Simulation results informed that the ability to absorb the surge and stabilize the system was relatively high in the smaller capacity of the accumulator. While, the energy delivery time was long with larger accumulator size.

T.Hung Ho and K. Kwan [2], used a novel way to recover the kinetic energy without reversion of fluid flow in the accumulator. Both displacement hydraulic pump and motors variables were effectively examined when the system operated in the flow coupling configuration.

It was found that the low efficiency of traditional HSTs under partial load conditions could be improved by utilizing the pressure coupling configuration.

While, M. Barnwal et.al [3], examined the operating parameters of a hydraulic transmission system with and without hydraulic accumulator. The hydraulic system was designed in the open circuit mode. It was concluded that the accumulator in the circuit absorbs pressure surge within the circuit with the certain limitations. Accordingly, it is preferable to use a pre-charger pressure in order to achieve the stability of the system as well as long life.

D. Otis [4], determined a thermal time-constant correlation based on the experimental data gas that charged by hydraulic accumulators. This correlation permits accurate prediction of accumulator thermodynamic losses. The gas pressure and temperature during compression or expansion, pressure was varied between 1.0 and 19.5 MPa. The experimental results revealed that the thermal time constant model was critically predicted the gas temperature (or pressure). It should be noted that the use of a variable time constant could be effectively increased the

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accuracy without adding a great deal of complexity to the analysis.

Mathematical modeling of a hydraulic accumulator for passenger vehicles was effectively simulated. The simulation of the system input functions was conducted by the use of real gas models.

H. Chrostowskiand and K. Kedzia **[5]**, modeled a hydraulic accumulator which was normally working as a secondary source of energy in a multisource driving system. The simulation of the system for characteristic input functions was conducted by the use of real gas models. Simulation results revealed the dependence of energy efficiency of the accumulator which was analyzed in term of its time-constant, time of storage, number of cycles of load and. The results were effectively compared with ideal gas thermodynamic processes. A mathematical modeling of a hydraulic accumulator for passenger vehicles was successfully simulated by A. Pfeffer et.al [6]. It was examined a carbon fiber reinforced plastic body, and aluminum piston. The impact of the radial discretization was mainly recognizable at high piston strokes, the longitudinal discretization was generally necessary for high model accuracy.

F.Teixeira [7], investigated theoretically the thermal boosted concept to improve energy storage capacity of a hydro-pneumatic accumulator. Simulation results indicated that more 60% improvement can be achieved demands. The energy storage capacity and efficiency were effectively decoupled from the power demands of the system

A.Stroganov and L.Sheshin [8], showed an improvement of heat-regenerative accumulators with piston and elastic separators aimed at the expansion of the operational conditions range as well as long service life and durability. Experimental results demonstrated a significant increase in the energy recuperation efficiency in the whole tested regimes when using the improved accumulators.

On the other hand, V.rizar et.al [9], examined hydraulic accumulator during operation and its effects on the energy efficiency of the component. An initial evaluation of the popular thermal time constant model was made. The model provided an interesting close up view to the gas movement and temperature distributions during operation. In addition, the model was successfully validated with experimental data, and provides a repeatable and accurate prediction of the gas states, regardless of the operating conditions, with maximum prediction errors of 10%.

S.Mameic & M.Bogdevieius [10], focused on the pressure pulsations in the hydraulic systems, the means reducing them. Also, an attempt was made to examine the structure of the hydraulic accumulators including their features and differences by using Fortran software. It was found that the addition of one, two or three hydraulic accumulators into the system lead to a significant decrease in the pressure pulsation, 1.78 MPa, 0.48 MPa and 0.43 MPa, respectively. Also, there was an increase in the pressure pulse amplitude from 2 MPa to 5 MPa caused an increase in the pressure pulsation from 0.15 MPa to 0.43 MPa. A variation in the hydraulic accumulator with size from 0.02 m^3 to 0.09 m^3 caused a rise in the pressure pulsation from 0.37 MPa to 0.38 MPa. It was also found that a distance between the accumulator in a mutual connection caused an increase in the pressure pulsation from 0.15 MPa to 3.08 MPa.

Present study pays more attention to examine the effect of adding an accumulator to the hydraulic system in order to reduce the vibrations that might be generated by the system as well as the actuator at the end stroke. Three positions were measured they were before relief, directional valve and at the linear cylinder body.

Nomenclature

A	Cross section area	m^2
c	Sound velocity	m/s
d_1	Inner diameter	m
$F_{1,2}$	Action and reaction force	N
l	Pipe length	m
u	Initial flow velocity	m/s
R	Resultant force	N
p	Pressure	N/m^2
t	Time	Second
p_o	Initial pressure	N/m^2
Δp	Pressure pulsation	N/m^2
ΔR	Pulsation reaction	N
β	Angle of elbow	Degree
ω_n	natural frequency	1/second

2. Experimental Set-Up

Figures (1 , 2) illustrate a schematic diagram and a photo of the hydraulic system rig located at the laboratory of under Graduate Studies of the Department of mechanical Engineering at the University of Technology to present the experimental work in hydraulic system tests.

It consists of two important parts, as electric and hydraulic part. Electric part consists of a PC, Arduino device, amplifier and safety module, position sensor; pressure sensor and hydraulic part consist of the necessary components of the conventional hydraulic system, such as various actuators, valves, tubes and other accessories.

Present study was efficiently used AC motor with a speed of 1500 rpm and input power 3 kW in order to rotate the shaft of hydraulic pump internal gear type G2 (Rexroth Company Production). The flow rate was 14 l/min, pressure 120 bar, displacement 11 cm3/rev and kinematic viscosity limit between (10→300) cSt.

To control the direction of the oil flow , it was used a directional control valve (Rexroth Company Production, 4WE10E1X/LG24NZ4). This means 4/3 D.C.V, normally closed, solenoid operation 24v AC current and with spring return.

A relief valve was critically utilized which was available in the size 6, type DBD (Rexroth Company Production, DBDS6G10/100). In fact, this relieves valve was used to protect a rig from excessive pressure that resulted from the displacement pump.

Further, a flow control valve was also used in the current system (DV(10–i) /OP350) which was a direct mounting on the pressure line, valve's size 10 and its weight was 0.4 kg. This valve works on the oil that has a viscosity limit between (2.8→380) cSt , oil temperatures ranging between (-20→100)°C and the maximum pressure that can be supported by this valve was 350 bar.

Present work also used a check valve type (S) with a return spring, and size of 10, in order to prevent the oil from returning back in the opposite direction . This valve was placed after the pump. It can be mounted directly on the pipelines, and its symbol was (S10A1).

Further, a double acting hydraulic cylinder with single rod was effectively used to convert the hydraulic energy into linear mechanical energy. In fact, this energy was used to push a spring (variable load) placed in the front side of the piston. The stroke length of this cylinder was 250 mm. The piston and rod diameter were designed as 50 mm and 25 mm, respectively.

A bladder type accumulator was successfully used in the test rig (Bosch). The capacity of the accumulator was assumed as 2.3 litters and the maximum pressure was 100 bar. The range of the working oil pressure is (10 – 40) bar, and the pre-charge pressure of the Nitrogen was 8 bar.

Atmosphere reservoir type ELF3 was used with capacity 80 liters of hydraulic oil type HL32. To measure the hydraulic pressure on a digital screen or by computer using a pressure meter with interface cable with software (sw-u801wn) was used with range is (0-100) bar.

To measure the pressure of the system a Borden gage by (new –Tec) was used, the pressure range (0- 100) bar. In order to measure the displacement when advancing and returning the cylinder and transferring the signal to PC over time a computer was added to record the readings and draw relationships by Lab-view program.

A vibration meter Type (VB-8200) (Figure 3) was used with a maximum vibration velocity 200 (mm/sec) and vibration acceleration 200 (m/sec^2).

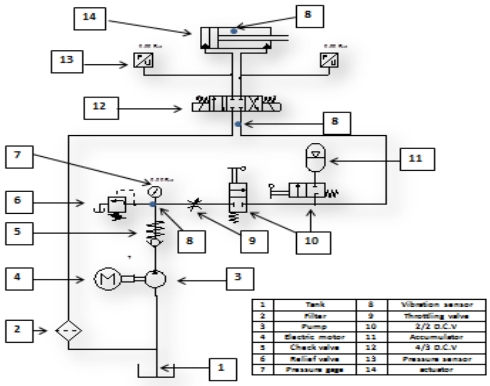


Figure 1: Apparatus schematic diagram

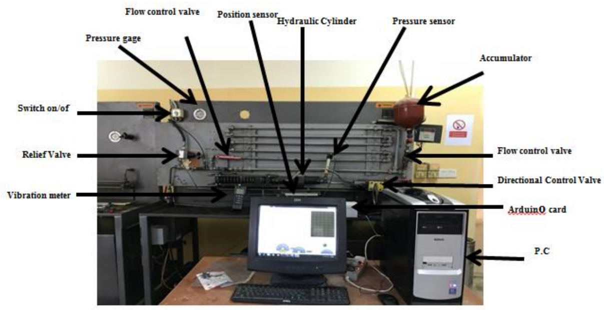


Figure 2: Experimental rig

Figure 3: Vibration Meter

3. Arduino program

The micro-controller on the board is programmed by Arduino Programming language and by Arduino's integrated development environment (Arduino IDE). Arduino can be integrated, Arduino is connected to its sensors and electronic parts only or Arduino is connected to communicate with programs on the computer, such as Processing and Max MSP and lab view Specifications of Arduino Figure (4).

•Microcontroller: AT mega 328
•Working voltage: 5 volts
•Input voltage limits: 6-20 volts and preferably 7-12 volts.
•I / O outlet: 40 mA
•Current voltage in pin 3.3V: 50 mA
•Memory Size: 32 KB
• Speed: 16 MHzP
• Six analog and digital signal

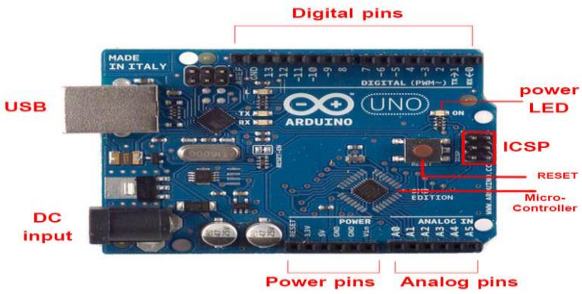


Figure 4: Arduino interface

4. Experimental Process

Present work was effectively used the accumulator in order to store the oil, which was delivered by the pump during the work cycle. The accumulator, then releases the stored oil on demand to complete the cycle, thereby serving as a secondary power source to assist the pump. In this experimental work, expert this application with a linear actuator (hydraulic cylinder) as follows

1- Actuate power unit of rig test.
2- Remain directional control valve in normal closed position.
3- Set pressure system at 10 bar by a relief valve.
4- Open manual directional valve to charge the accumulator to the maximum setting pressure.
5- Wait 5 minutes until the system is steady state.
6- Support vibration sensor on the pipeline at a point before the relief valve.
7- Actuate directional control valve to extend piston cylinder.
8- Record the data obtained from the digital vibration meter with interface directly to the computer.
9- Repeating the previous work with different pressures (20, 25, 30, 35 and 40 bar).
10- Repeat above procedure with fixed vibration sensor before directional control valve and on the cylinder body.

5. Theoretical analysis

The expression of pressure pulsation at a distance (x) is: [11[

$$p_\Delta(x,t) = \left[\frac{-2p}{n\pi}(cosn\pi l - 1)cos\omega_n t + \frac{-2u}{n^2\pi^2 cl}(cosn\pi l - 1)sin\omega_n t\right] sin\frac{n\pi}{l}x \qquad (1)$$

The resultant force of an elbow due to pressure pulsation is:

$$R = 2F_1 sin\left(\frac{\beta}{2}\right) = 2F_2 sin\left(\frac{\beta}{2}\right) = 2\left[\frac{\pi {d_1}^2 p}{4}\right] sin\left[\frac{\beta}{2}\right] \qquad (2)$$

Pressure pulsation $p=p_o+\Delta p$, elbow resultant force is

$$R = 2\left(\frac{\pi d_1^2}{4}\right)(p_o + \Delta p)sin\left(\frac{\beta}{2}\right) = 2\left(\frac{\pi d_1^2}{4}\right)p_o sin\left(\frac{\beta}{2}\right) + 2\left(\frac{\pi d_1^2}{4}\right)\Delta p\, sin\left(\frac{\beta}{2}\right) \qquad (3)$$

Where:-

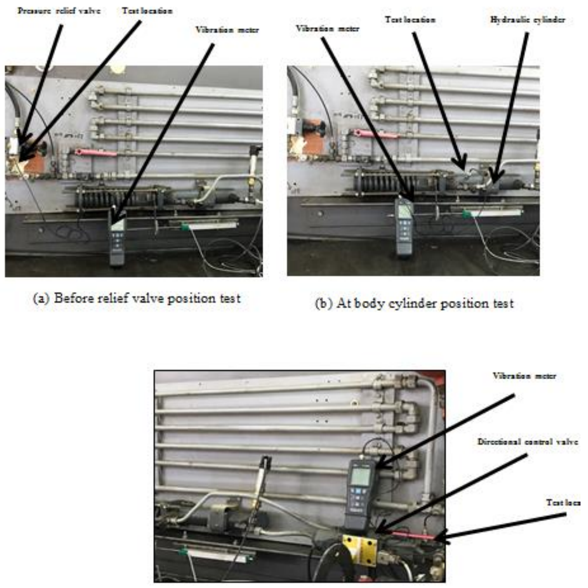
$2\left(\frac{\pi d_1^2}{4}\right)p_o sin\left(\frac{\beta}{2}\right)$ The force produced by static pressure.

$2\left(\frac{\pi d_1^2}{4}\right)\Delta p sin\left(\frac{\beta}{2}\right)$ The alternating force produced by pressure pulsation.

$$\Delta R = 2\left(\frac{\pi d_1^2}{4}\right)\Delta p sin\left(\frac{\beta}{2}\right) = 2A\Delta p\, sin\left(\frac{\beta}{2}\right) \qquad (4)$$

6. Results and Discussion

Present study inspected the effect of adding an accumulator to the hydraulic system, in order to damp the vibration that might be generated in the hydraulic system, due to the presence of pressure oscillation in the pump pulsation, three positions were measured. Fig.5 displays the vibration concentration regions (restriction flow regions) before relief and directional control valves as well as on the cylinder body. Seven values of the pressure supply were critically used (10, 15, 20, 25, 30, 35 and 40) bar.



(a) Before relief valve position test (b) At body cylinder position test

(c) Before directional control valve position test

Figure 5: Test location in the rig a, b, c

It is obvious to see (from equation (1) and Figs. (6, 7)), the vibration extent was decreased significantly as the distance (x) increased. The reason for that was attributed to the pressure oscillations that absorbed with oil flow and the system. Consequently, the amounts of vibration (velocity and acceleration) in body cylinder less than before direction control valve and relief valve.

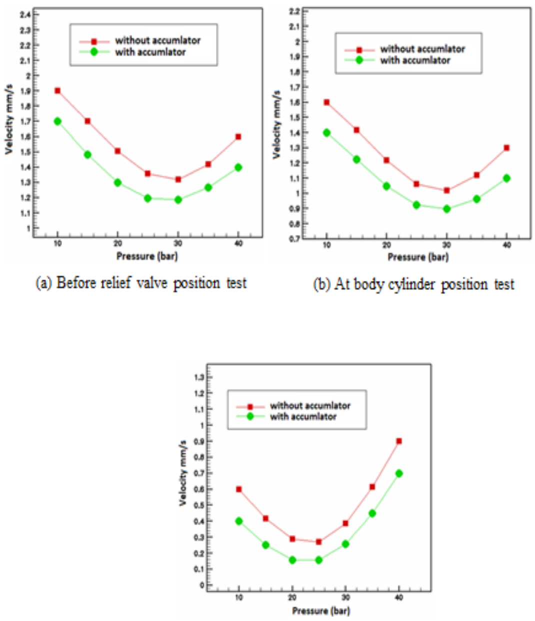
Additionally, it is obviously seen from Figs. (6, 7), that the amount of vibration (velocity and acceleration) increased at high and low pressure in which the maximum values at 10 and 40 bar in all positions . The reason for that was attributed to the rise of the natural frequency (equation.1).

Correspondingly, Fig.6 exhibits the relationship between pressure supply and vibration velocity in the sites selected for test. Before relief valve location the highest value of vibration velocity with and without 1.9 mm/second and 1.7 respectively under pressure 10 bar.

These values begin to decrease gradually with increasing pressure to reach the lowest value in which equal to1.2 and 1.1 mm/second at 30 bar pressure supply. Then the vibration speed begins to increase again with pressure supply increasing to reach its highest value 1.6 and 1.4 mm/second at 40 bar.

The above behavior was repeated with variation of vibrating accelerating with pressure as shown in Fig.7. It is clear to realize that the maximum vibration accelerated at 10 bar which was equal to 4.2mm/s^2 and reduced to 4.1mm/s^2 by using an accumulator. However, the values started to decrease with increasing pressure in both cases (with and without an accumulator) to reach the lowest value 3.9 and 3.5 at 25bar pressure supply respectively, hence the acceleration increases again with pressure to maximum when equal to 4.4 and 3.9mm/s^2 with and without accumulator at a pressure of 40 bar. Figs.(6-7) clarify likewise change of velocity and acceleration of vibration with pressure supply before direction control valve and at cylinder body. It's clear similar trend to the previous site test except lower value was occurring at different pressure value.

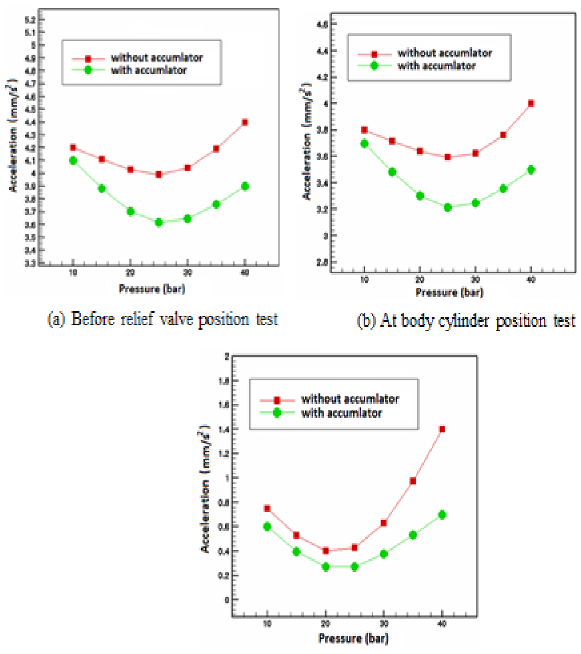
Before inlet directional control valve maximum velocity and acceleration at 10 bar, in which equal 1.4 and 1.6 mm/second, 3.7 and 3.8 mm/s^2 with and without an accumulator, respectively. While, the low velocity at 30 bar 0.85 and 0.9 mm/second and low acceleration observed at 25 bar 3.1 and 3.5 mm/s^2 with and without an accumulator, respectively. At cylinder body the maximum velocity takes place at 25bar in which equivalent to 0.1 and 0.21 while the acceleration observed at 20 bar in which equal 0.15 and 0.25 mm/s^2 with and without an accumulator. It's not from above results minimum values of velocity and acceleration with distance (x) take place with less pressure due to change natural frequency of the oil.



(a) Before relief valve position test (b) At body cylinder position test

(c) Before directional control valve position test

Figure 6: Variation of the Vibration Velocity with Pressure a, b, c



(a) Before relief valve position test

(b) At body cylinder position test

(c) Before directional control valve position test

Figure 7 : Variation of Vibration Acceleration with Pressure a, b, c

Figs (8-9) show damping percentage of velocity and acceleration by using an accumulator. Maximum percentage in velocity reduction of vibration at the position before relief and direction control and at cylinder body was 20%, 20.8% and 55% at 20 and 15 bar respectively. While in acceleration was 11.3%, 12.5% and 50% at 40 bar. This means that vibration damping increase with distance (x) from the pressure source due to more activity of an accumulator in this position.

Owing to the pressure difference between the pressure stock of the accumulator and pressure system, which is excessive pressure of the system suck through an accumulator.

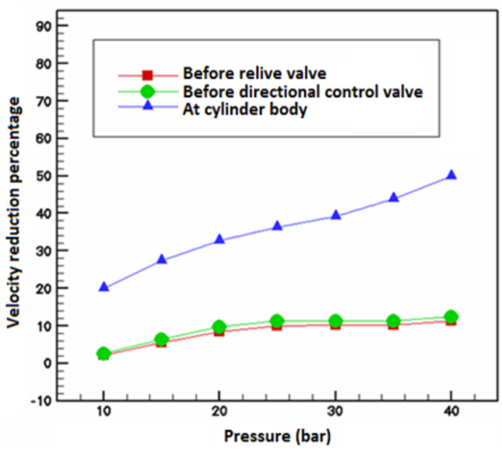


Figure 8 : Reduction Percentage of velocity with pressure

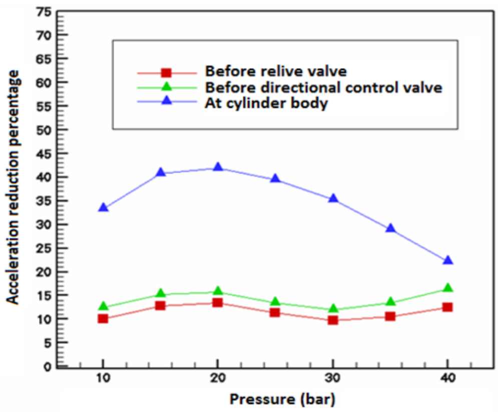


Figure 9 : Reduction percentage of acceleration with pressure

Figs (10, 11) illustrate the variation of actuator displacement with time in extend and retract stroke with and without an accumulator, it's clear difference in speed of end extend stroke after adding an accumulator. While retract stroke was not affected due to load (spring force) is dominant.

Figure.10 shows the change of stroke position with time without an accumulator. It is clearly seen that when the piston reached to the end of the stroke, the stop was occurred suddenly and without any decelerating this lead to a hydraulic line shock. While by using an accumulator figure.11 the piston begins decelerate gradually from a distance 25cm in which equal to 1/6 of total stroke length with a period of time 5 seconds.

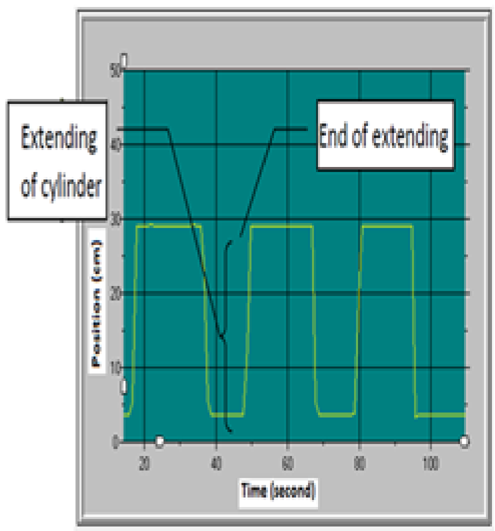


Figure 10: Extending stroke (Without accumulator)

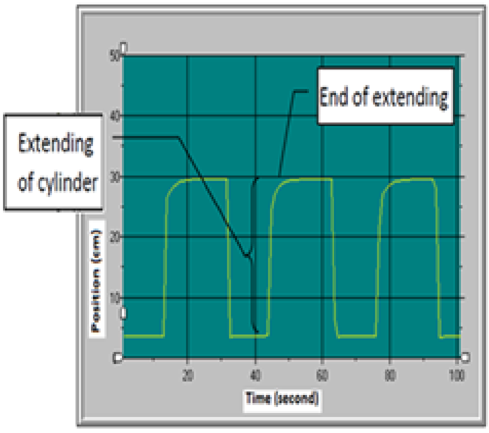


Figure 11: Extending stroke (With accumulator)

7. Conclusions

To reduce the vibrations that might be generated in the hydraulic system and decelerate the actuator piston at the end stroke, an accumulator was critically connected to the hydraulic system. Three positions were measured before relief and directional valve and at the linear cylinder body.

Experimental results revealed that maximum percentage damping in vibration velocity at the position before relief and direction control and at cylinder body was 20, 20.8 and 55% at 20 and 15 bar pressure supply respectively. While, in acceleration was 11.3%, 12.5% and 50% at 40 bar pressure supply. Additionally, it was obviously seen that the piston begins decelerate gradually from distance 25cm in which equal to 1/6 of total stroke length with a period of time 5 seconds.

References

[1] A.Kumar1, J.Das1, K.Dasgupta1and M.Barnwal "Effect of Hydraulic Accumulator on Pressure Surge of a Hydrostatic Transmission System" The Institution of Engineers (India) , 2017.

[2] T.Hung HO and K. Kwan "Modeling and simulation of a hydrostatic transmission system with energy regeneration using hydraulic accumulator" Journal of Mechanical Science and Technology Vol.24, Issue 5, pp (1163~1175), 2010.

[3] M. Barnwal, N. Kumar, A.Kumar and J. Das "Effect of Hydraulic Accumulator on the System Parameters of an Open Loop Transmission System" 5th International & 26th All India Manufacturing Technology, Design and Research Conference (AIMTDR 2014), , IIT Guwahati, Assam, India, December (12–14)Th. 2014.

[4] D. Otis "An Experimental Thermal Time Constant Correlation for Hydraulic Accumulators" Transactions of the ASME 116/Vol. 112, March, 1990.

[5] H. Chrostowski and K.Kedzia "The Analysis of Pneumatic Hydraulic Accumulator Efficiency Applied as Element of Hybrid Driving System" Scientific Papers of the University of Pardubice Series B - The Jan Perner Transport Faculty, 2004.

[6] A.Pfeffer, T.Glueck., W.Kemmetmueller, and A.Kugi "Mathematical Modeling of a Hydraulic Accumulator for Hydraulic Hybrid Drives" Mathematical and Computer Modeling of Dynamical Systems journal Vol 10, 2016.

[7] F.Teixeira Tavares "Thermally Boosted Concept for Improved Energy Storage Capacity of a Hydro-Pneumatic Accumulator" Ph.d, Mechanical Engineering, University of Michigan, 2011.

[8] A.Stroganov and L.Sheshin "Improvement of Heat-Regenerative Hydraulic accumulators" Vol 17, 2011.

[9] V.Irizar, P.Rasmussen, O. Olsen and C.Andreasen "Modeling and Verification of Accumulators using CFD" 15th Scandinavian International Conference on Fluid Power, SICFP'17, Linköping, Sweden, June (7-9), 2017.

[10] S.Mameic and M.Bogdevieius "Simulation of dynamic processes in hydraulic accumulators" ISSN (1648-4142), 2010.

[11] K. Sui and J.Wang "Calculation and analysis on residential pipeline pressure pulsation," Environmental Technology, No. z1, pp (203-206), 2013.

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