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A COMPUTATIONAL STUDY OF THE ACTUATION SPEED OF THE HYDRAULIC CYLINDER UNDER DIFFERENT PORTS' SIZES AND CONFIGURATIONS

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Abstract

The discharged oil from hydraulic cylinder, during its operation, is highly restricted by the small sized outlets. As a result, a back pressure builds up and the piston motion, therefore, is slowed down; the system pump has to do additional work to overcome this hydraulic resistance so as to preserve the required speed. In this study the possibility of improvement of the actuation speed of the hydraulic cylinders was investigated and analysed. Both a four-port cylinder and a resized-ports cylinder were proposed as fast cylinders. FLUENT 6.3 was used for the simulation of the oil flow field of the hydraulic cylinders. Results showed that relation between discharge flow and the outlets diameters is best described by a power law having coefficients partially depending on the system pressure. It had also shown that for any given total outlet area, the actuation speed of the single outlet cylinders is always higher than that of the double outlets cylinders. In one case where the total outlet area is 3.93E-05m^2, the actuation speed of the single outlet cylinder is 21% higher than that of the double outlets cylinder; whereas, when doubling the total outlet area the different is reduced to just 6% . Resizing the outlet for small ports was more efficient than using multi-outlets; while for a large ports it shows no significant difference to use either one outlet port or multi-outlets. Both the solutions of resizing or ports addition need special valve to be fit to the cylinder so that the cylinder could be effectively operated under the control of the proportional valve.

Keywords: Computational hydraulics; Efficient hydraulic cylinders; High speed cylinders; Hydraulic control.

Nomenclatures

A	Total outlet area of the cylinder, mm^2
A_o	Orifice area, m^2
A_1	Piston side area of the cylinder, m^2
A_2	Rod side area of the cylinder, m^2
C_f	Flow coefficient, dimensionless
F_f	Frictional force, N
F_L	External load, N
$\dot{m}$	Mass flow-rate, kg/s
P_1	*Pump (supply) pressure, Pa*
P_2	*Tank side pressure, Pa*
p	Resisting pressure, Pa
p_1	Upstream pressure, Pa
p_2	Downstream pressure, Pa
Q	Volumetric flow rate, m^3/s
V_1	Upstream fluid velocity, m/s
V_2	Downstream fluid velocity, m/s
v	Cylinder actuation speed, m/s

Greek Symbols

ρ	Fluid density, kg/m^3

1. Introduction

Hydraulic system has benefits over pneumatic or electric systems, especially when heavy loads are involved, or when very smooth and precise position or pressure control is required [1]. Hydraulic actuators have several advantages including the fact that they produce less heat and electrical interference at the machine than do electric actuators.

But there still some problems encountered in power hydraulics such as the unjustified energy losses at throttles through the entire system. Cylinder actuators are the one of the hydraulic system components that causes a lot of energy losses in power transmission and control. A conventional double acting differential hydraulic cylinder has two ports of small cross sectional areas. When the cylinder is actuated by supplying pressure at either port; the piston starts to move away as a result of the force difference on the two sides of piston; the piston push the oil out of the cylinder through the other port; the oil flow is highly restricted by the small area of the outlet port. The small sized port acts as an orifice and resists the migration of the incompressible oil from the cylinder; consequently the piston motion is slowed down. The energy lost in this process is converted to heat within the oil and add an additional load to the pump.

Improving the efficiency of the cylinder actuators will make it possible to impart great benefits to all related applications. Energy loss in hydraulic actuators can be approximated experimentally or by using analytical or empirical formulas. CFD makes it easy to obtain a better solution for energy loss in cylinder actuators and various hydraulic components. Computational simulation provides fundamental indications during the preliminary stage of the design and minimizes the time of development and the cost of the subsequent experimental analysis. By means of CFD it is possible to examine the characteristics of the actuators; and optimise the ports numbers and sizes for the cylinder before manufacturing.

In this study, the speed improvement possibilities in the conventional single outlet port and the proposed double outlet ports hydraulic actuators were investigate and analysed. The hydraulic actuators were simulated on FLUENT 6.3 using 3D models.

Many studies have been carried for improving the performance and the energy efficiency of the hydraulic actuators. Most of these studies focus on improving the performance of the actuators by developing and enhancing the control algorithms rather than improving the actuator structure [1-8].

Andersson [9] invented and developed a poppet design called a "Valvistor". He proposed the hardware arrangement of an independent metering system. His intention was more to achieve a valve arrangement for controlling large flows. Jansson et al. [10, 11] do an early work in his Ph. D on the independent metering area. The main focus in his work is how to handle and control the extra degree of freedom in the independent metering system. The energy savings concern minimisation of the meter-out losses, not regeneration or recuperation. Eriksson et al. [12] presented an LQR-approach as another early work in the area of independent metering systems. He focuses on performance and the influence of friction in his work. Elfving [13] presented a physically based decoupling approach for the hydraulic cylinders. Elfving also briefly takes up the energy aspects of the hydraulic actuators. Linjama et al. [14] had carried out extensive work concerning digital valves utilizing independent metering to improve the energy efficiency of the hydraulic system.

Lew et al. [15] introduced a new concept for an Electronic Flow Control Valve (EFCV) with pressure compensation capability. They utilize a micro-controller and the embedded sensors inside the EFCV to estimates the actual flow rate by the quasi-steady flow rate equation, to provide flow control without the need of load, speed and displacement information from the power elements, like hydraulic motors or hydraulic cylinders. They also presented an innovative application of the EFCV, a self-sensing cylinder. Osman et al. [16, 17] studied the pressure drop in the hydraulic spool valve. They have showed that the outlet geometry of the outlet port of the valve has a significant effect on the pressure drop and energy loss. Osman et al. [18] showed the same results in their study of energy loss in hydraulic proportional control valve.

2. Mass Flow Rate Calculations

The ports in hydraulic cylinders act as orifices for oil flowing through it. Provided the fluid speed is sufficiently subsonic (V < Mach 0.3), the incompressible Bernoulli's equation for laminar flows, Eq. (1), can be used reasonably well in obtaining the pressure drop through the cylinder ports.

$$\Delta p = p_1 - p_2 = \frac{1}{2}\rho V_2^2 - \frac{1}{2}\rho V_1^2 \tag{1}$$

where: p_1 is the upstream pressure, Pa.; p_2 the downstream pressure, Pa.; V_1 the upstream fluid velocity, m/s; V_2 the downstream fluid velocity at vena contracta, m/s.; and ρ is the fluid density, kg/m^3. By considering the continuity of the flow, the velocities (V_1, V_2) can be replaced by the cross-sectional areas and the volumetric flow rate Q. Also, counting for the viscosity and the uncertainty of the actual flow profile at downstream of the orifice (vena contracta diameter), a flow coefficient C_f was introduced. Solving for Q, the Bernoulli's equation for laminar flow was reduced to suit a viscous turbulent flow as shown below.

$$Q = C_f A_o \sqrt{\frac{2(p_1 - p_2)}{\rho}} \tag{2}$$

Where A_o is the orifice area.

The mass flow-rate "$\dot{m}$" was found by multiplying volumetric flow rate, Q, with the fluid density, ρ.

$$\dot{m} = \rho\, Q = C_f A_o \sqrt{2\rho\,(p_1 - p_2)} \tag{3}$$

The flow coefficient C_f is tabulated in reference books and can be found experimentally; its value ranges from 0.6 to 0.9 for most orifices, and it depends on the orifice and pipe diameters as well as the Reynolds Number.

3. Modelling of the Actuators

During its operation, the conventional cylinder actuator is subjected to many forces as shown in Fig. 1.

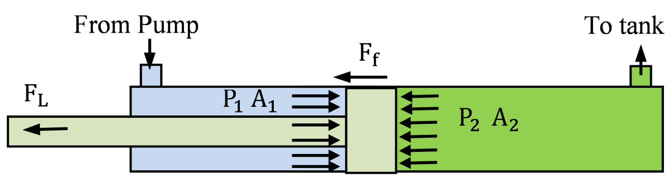

Fig. 1. A loaded conventional hydraulic actuator.

The actuator is connected to the external load, F_L through the piston rod. When the actuator is supplied with pump pressure, P_1 the equation of motion becomes:

$$P_1 A_1 - P_2 A_2 - F_f = F_L \tag{4}$$

The frictional force F_f is very small compared to other acting forces which are sometimes measured in kilo-Newton; so it is a negligible force. Solving for the tank side pressure, P_2 (commonly an unknown pressure), the motion equation becomes:

$$P_2 = (P_1 A_1 - F_L)/A_2 \tag{5}$$

A_1 and A_2 are the piston and rod side areas and are constant for any given cylinder. For a given supply pressure P_1, the load F_L determines the tank side pressure P_2. In the extension stroke the tank side pressure, P_2 could be greater than the pump pressure, P_1, when $P_1(A_1 - A_2) > F_L$; i.e. when the piston is moving. But still the force on the pump side is always greater than the force on tank side; it's why the regeneration can be possible.

Since the fluid in use is incompressible (hydraulic oil) the speed and acceleration of the cylinder actuator will depend on the flow rates allowed by the outlet port. Any attempt to increase the speed of the actuator should be associated with the increment of the flow rate. Increasing the speed of a given actuator having a given load and input pressure, means doing the required job in short time, hence energy is saved. The oil-zone at tank-side of the cylinder actuator is simulated in Fluent 6.3. The input pressure exerted by the piston onto this oil-zone was determined by Eq. (5).

The proposed actuators

To increase the cylinder speed, the existing two ports of the conventional cylinder were to be resized; or two new ports were to be added to the cylinder and the result, in that case, was a four ports cylinder as in Fig. 2. The proposed cylinders were equipped with a novel Self Actuated Flow Control Valve (SAFCV); otherwise, the additional oil flow couldn't be controlled. In the case of four ports cylinder, the proposed valve allows only one inlet and two outlets ports to be active at a time (each stroke). The flow through the new two ports bypasses the directional control valve. The new ports were connected to each other through the proposed valve. This "SAFCV" is a novel valve design that permits the flow from the tank side chamber of the cylinder and blocks the flow from the pump side. The proposed system allows the flow to inter the cylinder through one port only and facilitates for the flow to leave the cylinder through two ports each stroke. The proposed valve system has been experimentally tested in the Laboratory and it proved its functionality; but the details are not revealed here for patent requirements.

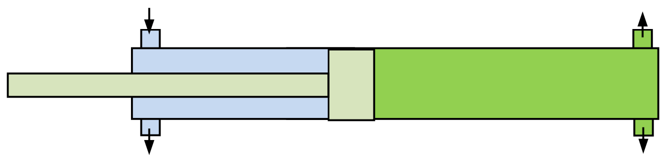

Fig. 2. Four port hydraulic cylinder actuator.

4. Computational Simulation

Fluent 6.3 software was used to simulate the oil flow inside the cylinder actuator. The flow variables and parameters such as the mass flow-rate and piston velocity were then computed. The cylinder ports diameters are usually small and the supply pressure is high; therefore, oil flow though these ports are assumed to be completely turbulent. Hence, the k-epsilon turbulence solver was chosen because of its numerical stability under a condition of large pressure gradient. Segregated implicit steady state equations were used. Second order equations are selected in the solution control. Convergence criteria of 10^{-6} for energy, dissipation rate, velocities and continuity were applied. A no slip velocity was taken as a boundary condition at the walls. Pressure inlet and pressure outlet boundary conditions were applied to all models of this study.

The fluid used was standard hydraulic oil having a viscosity of 46mm^2/s at 40°. Viscosity and density of the oil were all considered to be constant, i.e. the oil is Newtonian and incompressible. A conventional cylinder actuator of 80 mm bore was simulated. The outlet port diameter was varied from 5 mm to 10 mm. Inlet pressure of 100 bar exerted by the piston on the tank side fluid zone was kept constant for all the models; the pressure input was at the left. This pressure corresponds to the tank side pressure, P_2 of Eq. (2). The outlet port was considered open into the tank of atmosphere pressure.

4.1. Effect of the outlet size on the flow rates

Single outlet cylinders with various outlet diameters were simulated. The input pressure of 100 bar was kept the same for all models. Samples of pressure and velocity contours for 7 mm port cylinder were shown as in Figs. 3 and 4 respectively. A large variation in pressure and velocity were noticed at the outlet port.

Fig. 3. Static pressure contours for one outlet actuator.

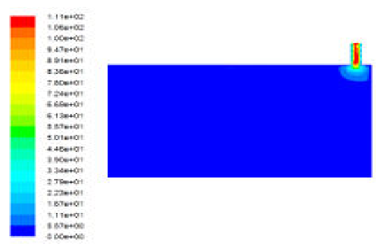

Fig. 4. Velocity contours for one outlet actuator.

After the simulation the mass flow rates for various cylinders plotted versus the outlet diameters as in Fig. 5. The results in Fig. 5 above agreed with the results that can be obtained by using Eq. (3) with C_f around 0.7.

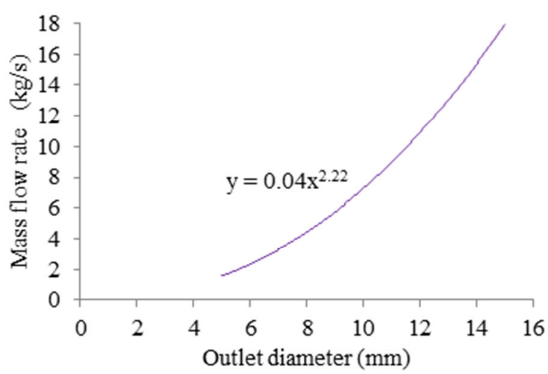

Fig. 5. Mass flow rate vs. outlet diameters.

For sufficient enough pump supply the mass flow rate increases rapidly with the increase of outlet diameter which agrees with the previous study [16]. The relation between flow rates and the outlets diameters was best described by a power law having coefficients partially depending on the operating pressures.

A new set of double outlet port cylinders with various outlet diameters were simulated under the inlet pressure of 100 bar. The pressure and velocity contours of one sample of cylinders were shown as in Figs. 6 and 7 respectively.

Fig. 6. Static pressure contours for double outlet ports actuator.

Fig. 7. Velocity contours for double outlet ports actuator.

The mass flow rates for cylinders with different outlet diameters were computed and plotted in Fig. 8.

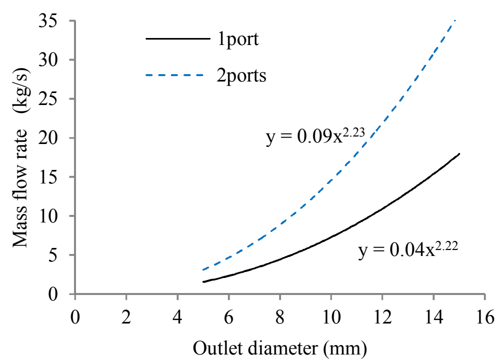

Fig. 8. Mass flow rate vs. outlet diameter in a constant inlet pressure condition.

The graph shows again power laws for both the single port and double ports cylinders. It's evident that addition of outlet port of the same size to the conventional cylinder actuator doubles the flow rates.

4.2. Effect of the pressure on the flow rates

To study the effect of the supply pressure on the flow rates; three models of cylinders with two equal outlets having 5, 7 and 9 mm diameters were simulated. The inlet pressure is varied by the increments of 10bars. The mass flow rates versus the inlet pressure were plotted as in Fig. 9.

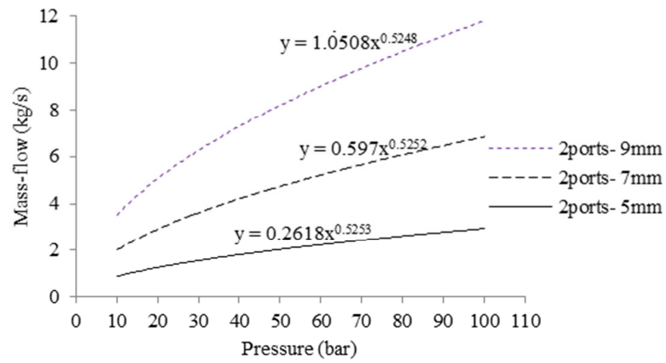

Fig. 9. Mass-flow rate vs. pressure for three actuators.

The mass flow rate trend lines shown in Fig. 9 follow the power law relation as:

$$\dot{m} = \alpha\, p^{0.525} \tag{6}$$

These power law relations for the three models have the same power of 0.525 regardless of the outlet sizes; which means their curve have the same trend line. But the equations have coefficient factors, *α,* of different values; these coefficients are dependent, among others, on the outlet size. It can be concluded that: for all outlet sizes the mass flow rate from the actuator is a function of ($p^{0.525}$).

4.3. Effect of port size on the actuation speed

To study the effect of outlet size on the actuation speed of the hydraulic cylinder; actuators having different outlets diameters ranged between 5 mm to 15 mm had been modelled and simulated. Some of these actuators have single outlet and the others have double outlets. These actuators were simulated on the Fluent under the inlet pressure of 100 bar.

The actuation speeds were then computed and plotted against the outlet diameters as shown in Fig. 10.

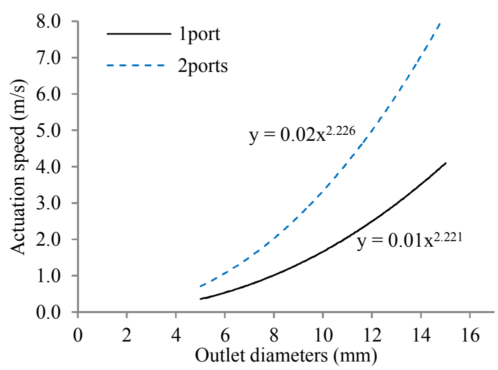

Fig. 10. Actuation speed vs. outlet diameter.

Graphical representation of the actuation speeds versus the total outlet areas was shown in Fig. 11.

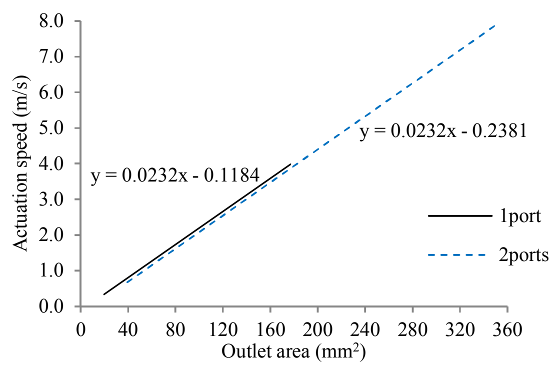

Fig. 11. Actuation speed vs. outlet area.

The speed-area equation of the single outlet cylinder is found to be:

$$v = 0.0232A - 0.1184 \tag{7}$$

While the speed-area equation for the double outlets cylinder is given by:

$$v = 0.0232A - 0.2381 \tag{8}$$

where: $\boldsymbol{v}$ is the actuation speed, m/s; and $\boldsymbol{A}$ is the total outlet area, mm^2.

It is evidence that the velocity-area curves for both the single and the double outlets models have the same slope of 0.0232; but their velocity intercepts are different. The velocity difference between the two types of the cylinders for any given outlet area value corresponds to the difference of the y-intercepts of the two curves; this is true only for the same input pressures. The single outlet curve has higher velocity intercept than the double outlets curve. This means that for any given outlet area the actuation speed of the single outlet cylinders is always higher than that of the double outlets cylinders.

5. Optimisation of the Outlet Size

The actuation speed of the hydraulic cylinder is highly affected by the outlet area as stated previous; the larger the outlet area the greater the speed of the actuator. There raised two alternatives to increase the ports areas: either by increasing the diameter of the existing outlet port and the result is a single outlet cylinder; or by addition of outlet port to make a double outlets cylinder. The two methods were studied to decide which of them is the best regarding the improvement of the actuation speed. A traditional cylinder actuator, having one outlet port of area 1.96E-05 m^2, corresponds to 5 mm diameter, was simulated. The cylinder was simulated on a pressure range of 10-100 bar. The mass flow rates were computed and the actuation velocities of the actuator were calculated.

Two different hydraulic cylinder models were then proposed for the purpose of finding the best way to improve the actuation speeds. In each model the outlet areas are doubled using one of the two methods mentioned above.

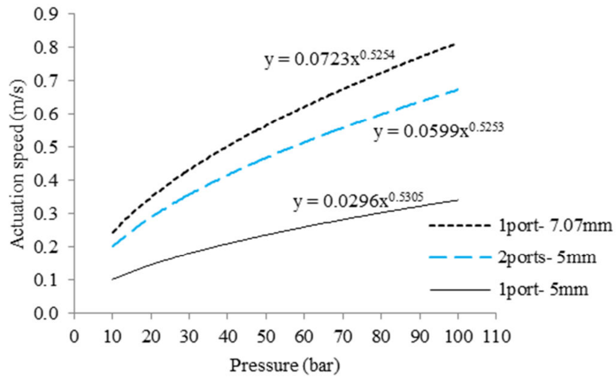

Fig. 12. Actuation speed vs. inlet pressure.

The first model was a cylinder having double outlet ports of 5 mm diameter each. This means that the total outlet area of the two ports together equals 3.93E-05 m^2. The second proposed model was a single outlet actuator of 7.07mm diameter which gives a total outlet area of 3.93E-05 m^2. The two models were simulated by applying the same boundary conditions and pressure range as for that of the traditional cylinder model simulated above. The mass flow rates were then computed; the actuation speeds were calculated thereafter. The actuation speeds and mass flow rates were plotted versus the inlet pressure as in Figs. 12 and 13 respectively.

The speed curves, shown in Fig. 12, follow the power law relation as:

$$v = \alpha\, p^{0.525} \tag{9}$$

The speed-pressure relationships for the three models also have the same power of 0.525 as for the different models previously presented in this study.

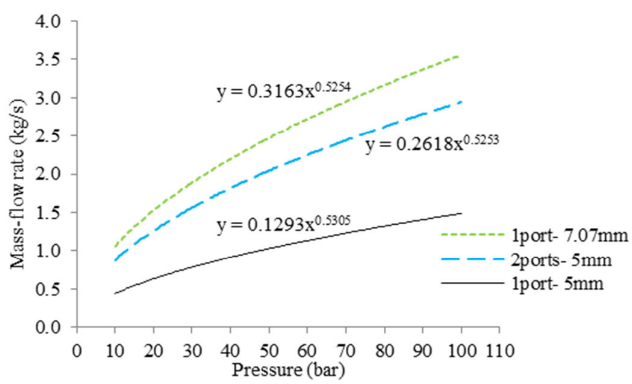

Fig. 13. Mass-flow rate vs. inlet pressure.

By the examination of the results in Figs. 12 and 13, it has been found that doubling the outlet area of the actuator by addition of outlet port improves the flow rates and the actuation speed by approximately 98%; while doubling the outlet area by increasing the port diameter improves the flow rates and the actuation speed by approximately 140%.

To affirm the reliability of the above results, more simulations were carried on another set of three cylinders with larger ports. The first model was a conventional single outlet cylinder having a port diameter of 7.07 mm. The second model was a single outlet cylinder of 10 mm diameter port. The third model was a double outlets cylinder of 7.07 mm diameter each. The last two models have the same total outlet areas of 7.85E-5 m^2, which is double the area of the first model. The flow rate and the actuation speeds were computed.

Graphical representation of the actuation speed versus the inlet pressure was presented in Fig. 14. The mass flow rate versus the inlet pressure was also plotted as in Fig. 15.

Again, examining the results in Figs. 14 and 15, leads to the conclusion: doubling the outlet area of the actuator by addition of one port improves the flow rates and the actuation speed by approximately 99%; whereas doubling the outlet area by increasing the existing outlet port diameter improves the flow rates and the actuation speed by approximately 110%.

These results showed that for the same total outlet areas, actuator with a resized outlet port performs better than actuator with double outlet ports. This is owed to the fact that; in the double ports actuator the oil flows over and contacts a large surface area of the outlet pipe; i.e. for the same total cross sectional areas the total circumferential area of the outlet ports for the double ports actuators is 1.414 times that of the one port cylinder. Therefore, the flow in the double ports actuator suffers more boundary layers resistance than in the one port actuator; as a result: the flow of the oil in the double outlets actuator slows down slightly, giving performance advantage to the actuators with one port.

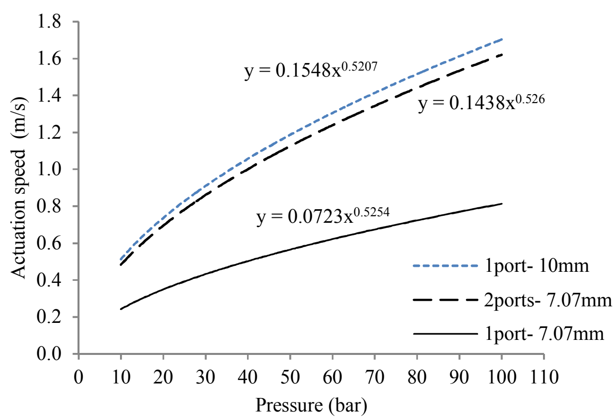

Fig. 14. Actuation speed vs. pressure.

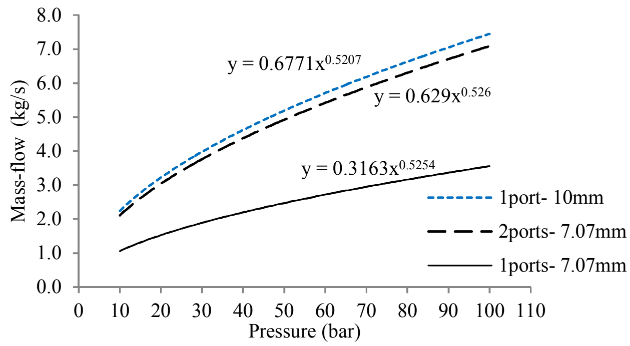

Fig. 15. Mass-flow rate vs. inlet pressure.

In the first case of the total outlet area of 3.93E-05 m^2 the actuation speed of the cylinder with one outlet port is 21% higher than that of the cylinder with double outlet ports. Whereas, in the second case of the cylinders having total outlet area of 7.85E-5 m^2 the actuation speed for the cylinder with one outlet port is just 6% higher than that of the cylinder with double outlet ports. It can be concluded that: to increase the outlet area for a small size ports it's more convenient to use one port rather than multiport, while for the large size ports it makes no significant difference to use either one port or multiport.

6. Conclusions

To study the effect of the outlet size on the flow rates and the actuation speed of the hydraulic cylinders, different types of cylinders were proposed and simulated on FLUENT 6.3. A single-resized-outlet and a double-outlet cylinder with different ports diameters were simulated under the inlet pressure of 100 bar. The relations between flow rates and the outlets diameters were best described by a power law having coefficients partially depending on the operating pressures; and that the flow rate is proportional to the number of ports.

Results also show that for a given total outlet area the flow rate and actuation speed of the single-resized-outlet cylinder were higher than that of the double outlets cylinder. In one of the cases where the total outlet area is 3.93E-05 m^2, the actuation speed of the resized single port cylinder was 21% higher than that of a cylinder with double outlet ports. When doubling the outlet area to 7.85E-5 m^2 the actuation speed for the cylinder with one outlet was just 6% higher than that of the cylinder with double outlets. The study concluded that: to increase the actuation speed of the hydraulic cylinder the outlet area must be increased by either resizing the existing ports or by addition of new ports to the conventional cylinder. Resizing the outlet for small ports cylinders was more efficient than using multi-outlet ports; while for large ports cylinders there was no significant difference in using either the one port or the multi-outlet ports.

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References

1. Tan, K.K.; and Putra, A.S. (2010). *Drives and control for industrial automation* (*advances in industrial control*). Springer, 15-16.
2. Chiang, M.H.; Yang, F.L.; Chen, Y.N.; and Yeh, Y.P. (2005). Integrated control of clamping force and energy-saving in hydraulic injection molding machines using decoupling fuzzy sliding-mode control. *International Journal of Advanced Manufacturing Technology*, 27(1-2), 53-62.
3. Liu, S.; and Yao, B. (2008). Coordinate control of energy saving programmable valves. *IEEE Transactions on Control Systems Technology*, 16(1), 34-45.
4. Beard, G.S.; and Stoten, D.P. (1996). Energy efficient pressure control circuit using MCS adaptive control of a hydraulic actuator. *International Conference on Control* (*UKACC*), 427(2), 1474-1478.
5. Mattila, J.; and Virvalo, T. (2000). Energy-efficient motion control of a hydraulic manipulator. *IEEE International Conference on Robotics and Automation*, 3, 3000-3006.
6. Yao, B.; and Liu, S. (2002). Energy saving control of hydraulic systems with novel programmable valves. *Proceedings of the 4th World Congress on Intelligent Control and Automation*, 4, 3219-3223.
7. Yao, B.; and DeBoer, C. (2002). Energy saving adaptive robust motion control of single-rod hydraulic cylinders with programmable valves. *Proceedings of the American Control Conference*, 6, 4819-4824.
8. Liu, S.; and Yao, B. (2002). Energy saving control of single-rod hydraulic cylinders with programmable valves and improved working mode selection. *National Fluid Power Association and Society of Automotive Engineers, Inc.,* 81-91.
9. Andersson, B.R. (1984). *On the Valvistor, a proportionally controlled seat valve*. Volume 108 of Linköping University. Linköping studies in science and technology. Dissertations. Division of Hydraulics and Pneumatics, Department of Mechanical Engineering, Linkoping University.
10. Jansson, A.; Krus, P.; and Palmberg, J.O. (1991). Decoupling of response and pressure level in a hydraulic actuator. *The 4th Bath International Fluid Power Workshop*.
11. Jansson, A. (1994). *Fluid power system design- a simulation approach.* Ph.D. Thesis, Linköping: Division of Fluid Power Technology, Department of Mechanical Engineering, Linköping University.
12. Eriksson, B. (1996). *Optimal force control to improve hydraulic drives.* Licentiate Thesis, Damek Research Group, Department of Machine Design,

13. Royal Institute of Technology, KTH, Sweden.

14. Elfving, M. (1997). *On fluid power control: A concept for a distributed controller of fluid power actuators.* Issue 658 of Linköping studies in science and technology: thesis. Division of Fluid and Mechanical Systems, Department of Mechanical Engineering, Linköping University.

15. Linjama, M.; Houva, M.; and Vilenius, M. (2007). On stability and dynamic characteristics of hydraulic drives with distributed valves. *Symposium on Fluid Power and Motion Control*.

16. Yuan, Q.; Schottler, C.; and Lew, J.Y. (2006). Electronic flow control valve (EFCV) with pressure compensation capability. *Proceedings of the Fifth International Fluid Power Conference*.

17. Osman, M.; Nagarajan, T.; and Hashim, F.M. (2010). Numerical investigation of pressure drop in hydraulic spool valve. *Proceedings of the 3rd International Conference on Solid State Science and Technology*.

18. Osman, M.; Nagarajan, T.; and Hashim, F.M. (2011). Numerical investigation of pressure drop in hydraulic spool valve. *Journal of Solid State Science and Technology*, 19(2), 48-60.

19. Osman, M.; Nagarajan, T.; and Hashim, F.M. (2011). Numerical study of flow field and energy loss in hydraulic proportional control valve. *National Postgraduate Conference*.

20. Scheffel, G. (2009). Energy efficiency in hydraulics. *Hydraulic Controls Division*, *Bulletin* HY11-3339/UK 05/2009, Parker Hannifin Corporation, Deutschland.