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Enhanced energy efficiency of industrial application by direct driven hydraulic unit

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Abstract—Direct Driven Hydraulic (DDH) systems, which are characterized by a closed circuit type and a speed-controlled pump, offer a possibility of reaching higher energy efficiencies compared to the traditional open circuit type valve-controlled systems, and simultaneously offering high accuracy and dynamics. This study presents experimental results gained with a DDH system applied to an industrial position control application. The results include the system behavior regarding the accuracy of position control, pressures, power, and energy consumption with three different system structures: basic DDH, load-compensated DDH and load-compensated and damped DDH. It was found that compared to valve-controlled hydraulics, DDH system offered potential for significant energy savings, especially if combined with hydraulic load compensation. However, without damping, the motion involved marked vibrations in the end of the stroke. Vibrations were avoided by introducing damping, but at the cost of reduced energy efficiency.

Keywords—direct driven hydraulics; electro-hydraulic actuator; displacement-controlled hydraulics; load compensation

I. INTRODUCTION

The field of hydraulically operated stationary industrial applications has been dominated by valve controlled open circuit systems with large oil reservoirs. Advantages of these include simplicity of both the system structure and control and very fast response times. The disadvantages in turn include poor energy economy because of the loss-control principle and the environmental risk of the large oil volume. The poor energy economy of the basic form of valve-controlled systems has been tackled with compensation methods that typically base on variable displacement pumps and pressure accumulators. In spite of these, the loss-control principle remains the same causing relatively high energy losses typically leading to a need of a cooling system and a large oil volume.

An alternative solution for valve-controlled systems are purely pump-controlled systems that do not include any flow-controlling valves. These have been common in traction control of mobile machines for decades. In these the speed of the machine is controlled either with combination of variable displacement pump and fixed displacement hydraulic motor, or with both components being of variable displacement type. The prime mover in these systems is commonly an internal combustion engine that runs with approximately fixed speed, so the control of fluid flow and vehicle speed occurs on basis

of varying pump and hydraulic motor displacement.

In stationary industrial applications, where the prime mover is almost without exception an electric motor, the use of this type of system has been lesser, and valve-controlled systems have been favored. The reason for this is the fact that stationary applications are typically large open system type central hydraulic systems with several actuators, while pump-controlled are typically one-actuator systems.

However, over the last two decades purely pump-controlled systems have become more appealing also in stationary industrial hydraulic applications, the main reasons being the demand for better energy efficiency and the favorable technical and price development of electric drives. The latter has made it possible to implement pump-controlled drives that are not only more energy efficient than valve-controlled systems, but also are competitive with these in terms of dynamics. Simultaneously this development is leading to transition from central hydraulics to actuator specific hydraulic systems.

Research of using valveless pump-controlled systems in stationary industrial applications has been active during the last decade and several studies have been published. Some of these have compared the properties of different drive types [1, 2] and found the pump-controlled systems to be fully competitive or even superior to valve-controlled systems or purely electro-mechanical systems. Besides theoretical studies, e.g. [3, 4] also some applied studies have been published, e.g. [5, 6] which, in addition to the energy and dynamic benefits of pump-controlled systems, show that these systems are also beneficial from heat and noise generation perspective.

With pump-control, most of the problematics are either related to the way the actuator is run during the work cycle or the stability of the actuator velocity when an asymmetric actuator is used. If the cycle includes transitions between several operation points, it may lead to drastically reduced energy efficiency if the pump's controller is optimized only for static operation. Addressing these problems with different controllers and control strategies are discussed in [7, 8, 9].

This study presents a control solution and measurement results for a hydraulically operated stationary industrial application, where a single symmetric cylinder is used as an actuator. Proposed solution is based on the closed circuit type system, where a speed-controlled fixed displacement pump controls the actuator. In our research group, this system type

has been named Direct Driven Hydraulics (DDH). This solution enables better energy economy compared to valve-controlled systems since the system's input power is very accurately matched to the required actuator output power. High-energy efficiency is further assisted by the short pipelines between pump and actuator resulting in low friction losses of the fluid flow. Together, these reduce the need for a separate cooling system.

The initiator of this study was industry's need to cut down the energy costs of a material-handling machine, which consecutively lifts and lowers a fixed mass with a certain range and cycle time. The research questions of this study are whether this can be accomplished with a DDH system while fulfilling the required dynamic properties set for the machine and what kind of energy savings are achievable compared to traditional valve-controlled system.

Results presented in this paper were gained in applying an actuator-specific DDH to a full-scale test rig that simulates the material-handling machine in question. As a special feature, the test rig includes a possibility to use a hydraulic load compensation, which enables reaching even greater energy savings than with basic DDH. In addition, the load compensation circuit also contains an adjustable damping for both directions of movement. This enables better control of the dynamic behavior of the system. The experimental results of different system quantities and the energy saving potentials for all of these system options are presented in this paper. The presented energy consumption values for traditional valve-controlled system were gained from a Matlab/Simscape simulation, since measurements of the actual systems were not possible.

The study ends with an evaluation of DDHs suitability and benefits in stationary industrial applications, and its energy saving potential. Likewise, the benefits of load compensation and damping circuit are discussed. Recommendations for further research is presented.

II. METHODS

In the studied case, a mock-up of an industrial application, an actuator is required to shift a mass of 1325 kg continuously between two vertical positions. The distance between these positions is 260 mm and each of the movements, either lifting or lowering, should be achieved in less than one second.

A. Test Equipment

Fig. 1 presents the test setup, where the main components of the actuating circuit are permanent magnet synchronous motor (1), swashplate pump (2) and actuating cylinder (11). The pump, valves and other components of the system are mounted to a manifold, which is directly attached to the actuator cylinder. Further information on the design process of this circuit and the industrial case can be found in [10]. The test rig has been described in detail in [11]. Compared to the previous setup described in these papers the following changes were made: mass increased from 700 kg to 1325 kg, pump displacement reduced from 45 cm³/rev to 28 cm³/rev and damping option added to the load compensation circuit. The damping option (14, 15) comprises of throttle valves that

restrict the outflows from the chambers of the load-compensating cylinder (12). The inflows are free through check valves and the damping can be activated/deactivated with shut-off valves.

Fig. 1. Pump-controlled electro-hydraulic actuator for moving a large mass, featuring a load compensating circuit with throttle valves for optional introduction of damping.

The topology of the simulated traditional valve-controlled system with its component description is presented in [11]. For this study the accumulator preload pressure was raised to 80.6 bar and the setting pressure of pump from 80 bar to 110 bar.

B. Control System

Overview of the control system is given in Fig. 2. In the previous study [11], software supplied with the hardware was used for controlling the motion. Due to the limited capacity of the PLC, only 4 ms position control loop cycle time could be achieved. For the current study, a new position controller was built using Simulink and converted to structured text using code generation. This way a 2 ms position control loop cycle time was achieved. The controller employs a rate limiter in order to control the initial acceleration.

Fig. 2. Control system structure

C. Experiments

The motion sequence used in this study comprised of a rapid lift followed by 3 s standstill before the load was returned to its starting position for another standstill period, after which the next cycle began. The sequence consisted of five cycles and the same sequence was used regardless of the magnitude of load compensation and damping used in the test. The starting position of the motion was 20 mm and the end position was 280 mm corresponding to a stroke of 260 mm. The maximum stroke of the cylinder was 300 mm, so the actual motion left 20 mm piston travel unused in both ends. Fig. 3 presents part of a motion sequence recorded for CASE 1 (see next section).

Fig. 3. Lifting and lowering motion.

In connection with P(ID) control, a sudden change in the setpoint may cause excess accelerations and decelerations of the motor, and result in high pressure spikes in the system. To counteract this behavior, a rate limiter was used in the controller substituting the step change in the position reference with a steep ramp. Throughout the study, a target velocity of 500 mm/s was specified for the actuator in the control software. Three separate test cases were run; basic DDH (CASE 1), load-compensated DDH (CASE 2), and load-compensated and damped DDH (CASE 3).

III. RESULTS

This chapter presents the measurement results depicting the behavior of the system (pressure and motion response), as well as its power and energy consumption.

Fig. 4 presents the system behavior with plain DDH, i.e. when the load was not compensated. Because of this, the pressure in the lower chamber of the actuating cylinder (11 in Fig. 1) was high in both directions of the movement, and at mass holding stage, the average level was 140 bar. In comparison, the pressure in the upper chamber was virtually zero throughout the cycle. In the figure, position curves are shown for reference and in order to highlight the vibrations occurring at the end of the stroke.

Figs. 5 and 6 present the corresponding behavior with load-compensated DDH and load-compensated and damped DDH. Due to load compensation, the pressure in the lower chamber of the actuating cylinder was less than in the non-compensated case, but simultaneously pressure spikes emerged in the upper chamber. Pressure fluctuation in the upper and lower chamber were in opposite phase. The load compensation pressure decreases from 128 bar to 120 bar as the load compensation cylinder moves to its upper position.

Fig. 4. System response, pressure and position vs. time, CASE 1.

Fig. 5. System response, pressure and position vs. time, CASE 2.

Fig. 6. System response, pressure and position vs. time, CASE 3.

The use of damping, Fig. 6, increased the pressure in the compensation system that corresponded to the direction of movement and, therefore, also to the corresponding actuating cylinder chamber pressure.

Figure 7 shows the effective electric input power to the motor in relation to the hydraulic output power of the actuator cylinder for CASE 1 without load compensation. During lowering, the large potential energy of the mass could be recovered, as shown by the negative values in the graph.

Fig. 7. Input power to electric motor and actuator hydraulic output power. Negative values during lowering represent recuperation. Graphs shown for CASE 1 as an example.

In Fig. 7 it can be seen that for CASE 1, peak electric input power during lifting reaches 10 kW. In comparison, peak electric input power during lifting for CASE 2 and 3 was only 3.0 kW and 3.5 kW, respectively, because of load compensation.

Corresponding system response figures for simulated traditional valve-controlled system are not presented here since the behavior of the system was principally the same as in [11] only the pressure levels were higher. For this system, only the energy consumption values are presented here, see Table I.

Figure 8 depicts how the energy consumption evolves over five consecutive motion cycles. The energy value shown is the energy counter that represents the electric energy drawn from the DC bus or recovered to the bus [12]. The curve for CASE 1 shows that lowering of the load will return energy, while the curves for CASE 2 and CASE 3 show that energy is needed also for lowering, because load compensation was used in these two cases.

Fig. 8. Energy consumption trends over five motion cycles.

In Table I, the energy consumption of one motion cycle has been listed for each test case, as well as for the simulated reference system implemented with traditional valve-controlled hydraulic system.

TABLE I. ENERGY CONSUMPTION FOR ONE EXEMPLARY LIFTING-LOWERING CYCLE.

	Lifting (kJ)	Upper position (kJ)	Lowering (kJ)	Lower position (kJ)	Total (kJ)
CASE 1	5.31	1.02	-2.18	1.15	5.30
CASE 2	0.95	0.04	0.43	0.02	1.44
CASE 3	2.05	0.04	2.28	0.02	4.40
Trad.sys.	6.85	0.61	3.26	0.45	11.17

In the table, ‘lifting’ refers to the part of the motion cycle from standstill at the lowest position (at 20 mm) to the end of the lifting stroke (at least 98 % of the 260 mm stroke has been achieved). ‘Upper position’ pertains to the energy consumption needed to keep the load in the upper position (at 280 mm). ‘Lowering’ refers to the subsequent return part of the motion, from the upper position back to the lower position. Finally, the ‘lower position’ column shows the energy needed to keep the load in its 20 mm position waiting for the next cycle to start.

IV. DISCUSSION

Conventional P control results in non-smooth motion at the end of the stroke, as can be seen in the end of the lifting and the lowering motion in Figs. 4 and 5. In addition, the motion producing force is caused by fluctuating pressures in the actuator cylinder chambers. Furthermore, when load compensation is used, large pressure fluctuations are in the opposite phases in the lower and the upper cylinder chambers of the actuator, causing additional control challenges. By activating the damping feature of the load compensating circuit, much smoother motion with reduced pressure oscillations can be produced, Fig. 6. However, damping introduces dissipation and takes away much of the energy efficiency advantage given by the load compensation circuit, Fig. 8. Even more dissipation is needed if shorter rise times are pursued, otherwise vibrations will again occur at the end of the strokes.

In CASE 1, the peak electric power needed for producing the motion amounts to 10 kW, Fig. 7. By introducing load compensation (CASE 2 and CASE 3), this value could be reduced to 3.0–3.5 kW. This indicates that by adding a few

relatively inexpensive hydraulic components to the system, an economical way of downsizing the electrical drive and motor can be achieved, without sacrificing performance.

The energy consumption values, Table I, show that plain load compensation (CASE 2) results in least energy consumed, because of operating only against load inertia. When damping is activated (CASE 3), it results in much higher energy consumption that is close to the uncompensated case. However, the total amount of energy consumed depends on the throttling magnitude. The less throttling the less energy consumption, but on the other hand, the more oscillation in actuator motion and pressures. Therefore, a balance for acceptable values between these two should be determined.

In this study, the damping was realized with simple throttle valves that were active throughout the actuator motion although damping is actually needed only when the actuator closes its stopping positions. This caused gratuitous energy consumption during most of the actuator motion. To mend this, the damping should be activated only when needed. This could be accomplished, e.g., with replacing the throttle valves of the present damping system with proportional throttle valves which are held fully open until the actuator is very close to its stopping position.

Comparing the total energy consumption values of the traditional valve-controlled system and DDH variants reveals that any of the DDH systems is, in this respect, better than the traditional system. The only circumstances where the traditional system surpasses a DDH system energy wise, is when the load is held at position and when there is no load compensation (CASE 1). This is because the DDH system needs to maintain a load supporting pressure throughout this phase whereas in the valve-controlled system the load is locked at position with the controlling valve, and energy is only needed to run the idling pump. As a result, the energy saving potential when transferring from traditional valve-controlled system to a DDH system is at minimum 53 % with non-load-compensated variant (CASE 1) and 87 % with load-compensated variant (CASE 2).

Figures 4–6 show that the objective concerning the dynamics of the DDH systems was nearly reached as the required motion was accomplished in approximately one second (the rise times were between 1–1.1 s).

Consequently, from the point of view of energy saving, the objectives set for DDH system were reached, while the system dynamic performance requires improvement.

The main problems with presented DDH systems are related to strong oscillations in actuator position and pressures. Therefore, in the next stage of the research, a different control strategy will be pursued that will smoothly produce the transitions between standstill and full velocity, in order to overcome the vibration problems, while still maintaining good energy efficiency.

V. CONCLUSION

The objective of this study was to test the applicability of a direct driven pump-controlled hydraulic (DDH) system based

on a speed-controlled electric servomotor and a fixed displacement pump to a stationary industrial material-handling application that consecutively lifts and lowers a fixed mass with a certain range and cycle time. The requirements set for this system, which is suggested to replace the traditional valve-controlled system presently used in this application, were related to reducing energy consumption and reaching at least the same dynamic properties as the valve-controlled system.

Three different variants of DDH were measured; plain DDH, DDH with load compensation, and DDH with load compensation and vibration damping. All of these proved superior to simulated valve-controlled system in respect of energy consumption. The reached energy savings varied between 53–87 %. However, the dynamics of the system did not quite meet the set requirements. The reached rise times exceeded marginally the target, which was to carry out the 260 mm stroke in no more than one second.

In addition, the actuator tended to vibrate when reaching the target position, and the system pressures oscillated strongly also during the movement of the cylinder.

Nevertheless, the DDH is a promising alternative to traditional valve-controlled systems and therefore our research will be continued by applying a different motion control strategy (e.g. trajectory/motion planning, acceleration limitation) more suitable for the application in question. Another research subject to study more is the way the vibration damping should be applied during the motion of the actuator.

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NOMENCLATURE

E_{CASE1}	energy consumption for CASE 1	[kJ]
E_{CASE2}	energy consumption for CASE 2	[kJ]
E_{CASE3}	energy consumption for CASE 3	[kJ]
p_A	pressure in lower working chamber of actuator	[bar]
p_B	pressure in upper working chamber of actuator	[bar]
$p_{C,acc}$	pressure in compensation circuit accumulator	[bar]
$p_{C,r}$	pressure in compensation cylinder, rod side	[bar]
$p_{C,p}$	pressure in compensation cylinder, piston side	[bar]
P_a	hydraulic output power of actuator	[kW]
P_M	servo motor electric input power	[kW]
x_{ref}	actuator position command	[mm]
x_{meas}	measured actuator position	[mm]

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