Preliminary Experimental Research on the Influence of Counterbalance Valves on the Operation of a Heavy Hydraulic Manipulator during Long-Range Straight-Line Movement

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Abstract: The effective use of robotic manipulators is particularly important when carrying out hazardous tasks. Often, for this type of mission, manipulators equipped with a hydraulic drive system are used, and their work results primarily from the implementation of precise movements through their effectors. In heavy manipulators, limiting the uncontrolled movement resulting from high inertia and relatively low stiffness has an impact on the improvement of the control precision. Therefore, the paper presents experimental studies that allow the assessment of the impact of the use of counterbalance valves on the precision and dynamics of a manipulator with a hydrostatic drive system. The tests were carried out for a wide range of effector velocities along a horizontal trajectory, on the basis of which, it was found that it was possible to improve the precision and dynamics of the work of such manipulators due to the precision of the trajectory and pressures in the drive system.

Keywords: manipulator; counterbalance valve; hydraulic drive system; accuracy

1. Introduction

Robots equipped with manipulators are used in many areas of life, especially dangerous ones such as, demolition, underwater construction, nuclear plant services, disaster area inspection, and rescue and EOD (explosives ordinance disposal) missions [1–10]. Heavy robots with manipulators with a hydraulic drive are usually used to carry out work that requires large lifting forces and reach [1–4,11–14]. Most often, the work of such manipulators can be divided into two phases. In the first one, the manipulator effector is required to quickly and efficiently reach the chosen load (execution of the delivery movement by the manipulator effector). In the second phase, during the loading and manipulation of the load, precision of the effector movement is required, regardless of its speed (execution of precise movement). Achieving high operation efficiency requires ensuring the shortest time of delivery movement. Most often, the fastest possible path for the effector from the transport to the working position is the horizontal trajectory. In heavy multi-DOF hydraulic-driven manipulators, achieving a horizontal trajectory always requires using at least two separate DOFs (i.e., arm and boom) controlled simultaneously by at least two hydraulic cylinders.

The trajectories for delivery movements are much longer in relation to the precise movements, and they even reach several meters. In order to ensure a high level of manipulation and work efficiency, heavy manipulators are characterized by a complex kinematic structure and have a large number of degrees of freedom, usually 6–9 DOF. They include, for example: base swing, boom lift, arm lift, forearm lift, grapple rotation and grapple opening, ref. [5] and may also have shoulder rotation, wrist rotation and independent fingers or clamps control. For this reason, controlling the manipulators with standard joysticks is much more complicated; moreover, it is not very intuitive and requires long
and very costly training of operators to acquire the right reflexes. In addition, regular and long-term training is needed to maintain their habits.

For these reasons, new solutions for human–robot interface [11–20] have been sought, which will allow less experienced operators to work effectively. Examples of such solutions are the master–slave type or hand tracking controllers, in which the operator directly controls the position of the tool [21–23]. Contrary to standard solutions using joysticks, this does not require controlling the operation of individual hydraulic cylinders and coordinating the movements of the manipulator elements. In this case, the manipulator control consists of controlling the direction of movement of the working tool (e.g., gripper), and the control signals of individual hydraulic cylinders (boom, arm and forearm) are generated automatically by the microcontroller. Due to this approach, any trajectories can be realized in a relatively simple way. Most often, horizontal trajectories are used to carry out tasks by the manipulator tool, which primarily results from the shortest and fastest path to the assumed target [24,25]. For the same reasons, controllers equipped with haptic feedback are also being developed [26,27].

As the standard, heavy hydraulic manipulators are equipped with proportional directional valves instead of servovalves, as they can control larger flows, are more reliable and durable, more resistant to contamination and less expensive. The disadvantage of such a solution is the lower response frequency and nonlinear static characteristic with deadband, which worsens the control accuracy [11,28]. Moreover, manipulators with a hydraulic drive have high inertia and a relatively lower stiffness, which may cause vibrations and make precise control difficult [11–14,26–28]. The results of the research [29–31] showed the possibility of limiting vibrations in machines and devices with a hydraulic drive system by using counterbalance valves. In this case, the research most often covers the control of one actuator, thus driving individual elements of the mechanism [32,33]. Research on counterbalance valves has also been carried out due to the possibility of limiting energy consumption during the standard cycle of operation of lifting devices (crane assembly) [34]. A positive effect on energy consumption was also demonstrated in [35], where the authors of the conceptual solution of the hydraulic drive system (EHA, electro-hydraulic actuator system), containing counterbalance valves, reduced the power demand by up to 50% compared to the classic solution. Research on the use of counterbalance valves has also included the precision of the control of the hydraulic receivers. The test results [36] showed that, for low power systems and slowly changing loads, it was possible to obtain high accuracy and the repeatability of the actuator piston rod movement by using counterbalance valves. The authors of [37] also referred to the precision of the control, where the influence of the counterbalance valve settings on the safety of the operation of hydraulic systems, with particular emphasis on the instability of the operation of such a system after stopping and restarting, was determined. The safety of the operation of hydraulic systems was also discussed in [38], which presented a control algorithm of a system equipped with counterbalance valves to reduce fluctuations in the lifted load by a lifting device.

Analyzing the above-mentioned literature, it can be concluded that the work on the use of counterbalance valves showed a beneficial effect due to the issues of energy consumption, work dynamics and control precision of particular hydraulic cylinders singly. In addition, lots of results [29,34,37] refer to the influence of counterbalance valves on oscillation minimization when the hydraulic cylinder is braking during lifting and lowering. However, no research results were found describing the impact of counterbalance valve application on the precision and dynamics of work when controlling several hydraulic cylinders simultaneously. This issue is especially important when executing movements with heavy multi-joint hydraulic-driven manipulators controlled in the master–slave system on a horizontal trajectory.

The system that controls the manipulator needs to keep precise track of the machine at all times. A low control precision can significantly hinder the implementation of tasks, extending the working cycle time and thus increasing the energy consumption of the process. The aim of the research was to determine the effect of introducing counterbalance
valves into the hydraulic system on the precision and dynamics of a heavy hydraulic-driven manipulator during the implementation of a horizontal trajectory by its effector.

2. Methods

The tests consisted of performing a horizontal rectilinear motion by the gripper of a heavy hydraulic manipulator at various speeds.

The tests were carried out on a robot manipulator with a maximum range of 4 m, load capacity of 200 kg and 6 DOF (Figure 1). The manipulator consisted of five sections: base swing, boom, arm, forearm and gripper. The manipulator was equipped with a hydrostatic drive system that allowed for the implementation of complex movements with the use of a proportional directional valve operating in the “load-sensing” system.

Two variants of its hydraulic system were used for the tests:
- Variant A, a hydrostatic drive system without counterbalance valves (Figure 2),
- Variant B, a hydrostatic drive system with counterbalance valves for boom and arm cylinders (Figure 3).

Figures 2 and 3 show the hydraulic diagrams of the valves with and without CV for the sections responsible for the movement of the boom and arm piston rod. It was assumed that the implementation of the rectilinear movement would be realized by combining the movements of the boom (1) and the arm (2), while maintaining the same position of the forearm (3) and the grapple rotation (4) (Figure 1).

The tested manipulator was equipped with measuring elements, enabling the assessment of the precision, dynamics and energy consumption of the work during the implementation of the assumed trajectory. The work precision was assessed on the basis of deviations of the gripper trajectory from the assumed horizontal trajectory. In turn, the dynamics of the work were assessed on the basis of the pressures in the boom and arm cylinders. The trajectories were determined on the basis of measuring the displacement of the cylinder piston rods by performing a simple kinematics task, and the pressure courses were obtained from pressure sensors installed at the cylinder stubs. The basic parameters of the hydraulic elements and measuring sensors of the tested manipulator are presented in Table 1.
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**Figure 2.** Diagram of boom and arm hydrostatic drive without counterbalance valves, where AA represents the hydraulic cylinders of the arm, BA represents the hydraulic cylinders of the boom, and PDV represents the boom and arm section of the proportional directional valve with “load-sensing” system.

**Figure 3.** Diagram of boom and arm hydrostatic drive with counterbalance valves, where AA represents the hydraulic cylinders of the arm, BA represents the hydraulic cylinders of the boom, CV represents the counterbalance valves, and PDV represents the boom and arm section of the proportional directional valve with “load-sensing” system.
### Table 1. Parameters of the main hydraulic and measuring elements.

| Parameter                  | Value/Specification                                                                 |
|----------------------------|-------------------------------------------------------------------------------------|
| Boom actuator, BA          | Piston diameter 100 mm, rod diameter 56 mm, stroke 550 mm                           |
| Long arm actuator, AA      | Piston diameter 63 mm, rod diameter 45 mm, stroke 600 mm                           |
| Counterbalance valve, CV   | Nominal flow 60 dm³/min                                                              |
| Proportional directional valve, PDV | Nominal flow 100 dm³/min limited by spools size to 40 dm³/min                   |
| Draw-wire encoder         | Measuring range 0–1250 mm, accuracy 0.02%                                          |
| Pressure sensor           | Measuring range 0–40 MPa, accuracy 0.05%                                           |

Within one measurement cycle, the gripper of the tested manipulator moved forward along the horizontal trajectory, and then, after a short stop, it returned to the starting point. In order to assess the impact of the use of counterbalance valves on the manipulator’s work, the following were carried out:

- Tests on the basic system for variant A of the hydraulic system of the manipulator (Figure 2),
- Tests on a modified system for variant B of the manipulator hydraulic system (Figure 3).

The tests were carried out for the same gripper speeds, i.e., 100, 200, 300 and 400 mm/s. The range of horizontal movement was 1800 mm and was made at a height of 500 mm above the plane of the manipulator base. The following indicators were adopted for the evaluation of the trajectory (Figure 4):

- The root mean square deviation of the gripper trajectory from the assumed horizontal trajectory \( y_{RMS} \),
- The absolute value of the maximum deviation of the gripper trajectory from the assumed horizontal trajectory \( y_{\text{max}} \),
- The deviation range \( \Delta r \). In turn, for the assessment of pressures, the following were adopted (Figure 5):
  - The root mean square pressure from the executed movement \( p_{RMS} \),
  - The maximum value of the pressure from the executed movement \( p_{\text{max}} \),
  - The pressure pulsation during the effector stop phase \( p_a \),
  - The pressure pulsation time, \( t_o \), when the gripper was stopped (where the pressure amplitude, \( \Delta p_{2\%} \), equal to 2% of the nominal pressure, \( p_N \), was assumed at the end of the pulsation time, \( p_N = 17 \) MPa, that is, \( \Delta p_{2\%} = 0.34 \) MPa),
  - The time of a single working cycle \( t_c \),
  - The frequency of pressure oscillations \( f \) during the effector stop phase,
  - The damping ratio \( \zeta \) of pressure oscillations during the effector stop phase, which was defined according to [39].

![Figure 4. The scheme of indicators to evaluate the trajectories, with (1) trajectory of effector and (2) reference trajectory.](image-url)
In order to implement the assumed horizontal trajectory, signals for controlling the boom and arm actuators of the tested manipulator were developed. They were based on the diagram in Figure 6. Based on the knowledge of the $x, y$ position coordinates of the gripper, the angles $\varphi_1$ and $\varphi_2$ were determined; the knowledge of which allowed determination of the necessary angular velocities $\dot{\varphi}_1$ and $\dot{\varphi}_2$, the velocity of the piston rods of the boom, $s_1$, and the arm, $s_2$, and, consequently, the required flow of these $Q_1$ and $Q_2$ cylinders.

According to the diagram (Figure 6), the first angles $\varphi_1$ and $\varphi_2$ were determined. They were determined based on the task of inverse kinematics [40], according to the scheme in Figure 7. It was assumed that the angle $\varphi_3$ between the arm and forearm was unchanged during the motion and equaled 180°.

Based on the cosine theorem (1), the relationship for the angle $\varphi_2$ of the arm was determined in accordance with the relationship (2).

$$
\cos \varphi_2 = \frac{x^2 + y^2 - a_1^2 - a_2^2}{2a_1a_2} = R
$$

(1)
\[ \varphi_2 = \arctg \frac{\sqrt{1 - R^2}}{R} \]  \hspace{1cm} (2)

In turn, the knowledge of the angle \( \varphi_2 \) allowed determination of the angle of rotation of the boom \( \varphi_1 \) (3).

\[ \varphi_1 = \arctg \frac{y}{x} - \arctg \frac{a_2 \sin \varphi_2}{a_1 + a_2 \sin \varphi_2} \]  \hspace{1cm} (3)

In the next stage, the angular displacements were differentiated, which made it possible to determine the dependence on the speed of the boom (4) and arm (5) piston rods based on the kinematic structure of the tested manipulator (Figure 8). A summary of the basic dimensions of the manipulator is presented in Table 2.

\[ s_1(t) = \varphi_1(t) \cdot h_1 \cdot \cos \left( \frac{\pi}{2} - \gamma_1 \right) \]  \hspace{1cm} (4)

\[ s_2(t) = \varphi_2(t) \cdot h_2 \cdot \cos \left( \frac{\pi}{2} - \gamma_2 \right) + \varphi_1(t) \cdot h_3 \cdot \cos \left( \frac{\pi}{2} - \gamma_3 \right) \]  \hspace{1cm} (5)

![Figure 8. Kinematic structure of the tested manipulator.](image)

| Parameter | Value, mm |
|-----------|-----------|
| \( w_1 \) | 370       |
| \( w_2 \) | 130       |
| \( h_1 \) | 1190      |
| \( h_2 \) | 500       |
| \( a_1 \) | 1400      |
| \( a_2 \) | 2500      |
The boom and arm cylinders were controlled by the sections of the proportional direct valve working in the load sensing system. The ideal flow characteristic of the proportional valve section was assumed, and the flow rate was proportional to the control signal. It was also assumed that the nominal flow rate generated by the $Q_N$ pump was equal to the sum of the flows of the $Q_1$ boom and $Q_2$ boom:

$$Q_N = Q_1(t) + Q_2(t)$$

For the signals $Q_1$ and $Q_2$ defined in this way, the control model was supplemented with an automatic control system according to the scheme in Figure 9. The positioning of the manipulator tool, $x_e, y_e$, was performed using the PID controller. The selection of the regulator settings was made on the basis of experimental tests carried out on a physical object in Figure 1. For this purpose, the commonly used Ziegler–Nichols method was used [41]. The values of the individual settings are shown in Table 3.

![Control system diagram](image)

**Figure 9.** Control system diagram, where PID1 is the PID controller of boom section and PID2 is the PID controller of arm section.

| Hydraulic Cylinder | Proportional Gain, $K_p$ | Integrated Gain, $K_i$ | Derivative Gain, $K_d$ |
|--------------------|--------------------------|-----------------------|------------------------|
| Boom (PID$_1$)     | 200                      | 3                     | 5                      |
| Arm (PID$_2$)      | 200                      | 3                     | 5                      |

**Table 3.** Parameters of the PID controllers.

3. Results and Discussion

The results obtained during the tests are shown in Figures 10–13. They contain the manipulator effector trajectories measured during the operation of the hydraulic system with and without counterbalance valves for the effector speeds of 100, 200, 300 and 400 mm/s.

![Manipulator’s effector trajectories](image)

**Figure 10.** Manipulator’s effector trajectories for a velocity of 100 mm/s.
Comparing the trajectories obtained for the low effector velocities, 100 mm/s and 200 mm/s (Figures 10 and 11), it was found that the use of counterbalance valves slightly deteriorated the accuracy of the effector guidance in relation to the system without them. Significant differences in trajectories can be noticed for effector speeds higher than 200 mm/s. Tool deviations for a manipulator without counterbalance valves for high speeds can be as high as 300 mm (Figure 13), while in the case of a manipulator with counterbalance valves, they oscillated around 100 mm. The greatest deviations were noted during the return of the working tool to the starting point for the manipulator without counterbalance valves. The use of these valves allowed for effective and controlled displacement of the cylinder piston rod, even in the case of the same loading force direction and the direction of rod movement of the piston rod.

The oscillations of the piston rods of the boom and arm cylinders can be seen on the pressure courses in their chambers (Figures 14 and 15). This disadvantageous phenomenon was especially visible for the higher speeds of the working tool of the tested manipulator. At speeds of up to 200 mm/s, the movement of the tool was slow enough...
that the system was able to catch up and minimize the resulting error. In this way, the unfavorable pressure pulsations in the system did not occur, which is shown by the $p_{\text{max}}$ (Figures 14 and 15). However, at higher effector speeds, significant pressure pulsations were found in the final stage of the return stroke for a manipulator without counterbalance valves (Figures 14 and 15). In this case, the maximum pressure values were several times higher than the average values. The use of counterbalance valves significantly reduced these pulsations (Figures 16 and 17), and their maximum values for the considered effector speeds did not differ significantly from each other.

Figure 14. Plot of boom cylinder pressure for manipulator without counterbalance valves and effector velocity 300 mm/s.

Figure 15. Plot of arm cylinder pressure for manipulator without counterbalance valves and effector velocity 300 mm/s.

Figure 16. Plot of boom cylinder pressure for manipulator with counterbalance valves and effector velocity 300 mm/s.
Comparing the values of the mean, \( y_{RMS} \), and maximum deviations, \( y_{max} \), as well as \( \Delta r \) (Figures 18 and 19), it can be concluded that the use of counterbalance valves in the hydraulic drive system significantly improved the precision of guiding the working tool in relation to the system without them. This was especially visible at speeds above 200 mm/s where the maximum deviations were almost 2.5 times smaller in favor of the system with counterbalance valves (Figure 18). Considering \( \Delta r \), it was found that the manipulator effector oscillations at a speed of 400 mm/s without counterbalance valves reached 600 mm (Figure 19). The use of counterbalance valves made it possible to reduce these by three times. Taking into account the \( \Delta r \) index, no significant differences were found in the accuracy of the displacement of the manipulator effector at speeds up to 200 mm/s (Figures 10 and 11), both for the manipulator with and without counterbalance valves.

In addition to the more precise guidance of the manipulator effector, the use of counterbalance valves significantly reduced the maximum pressures occurring in the cylinder chambers for the effector traveling at a speed of 400 mm/s from 16.5 MPa to 6.3 MPa, (Figure 20). Additionally, the use of these valves reduced by even 10-fold the pressure pulsations in the cylinder chambers for a sudden stop of the piston rod movement (Figure 21). By analyzing the RMS values of the pressures, it is possible to notice their increase for the manipulator with counterbalance valves in relation to the manipulator without them (Figure 22). It was particularly visible for the boom actuator for the entire range of the considered effector speeds. There was a reduction in the efficiency of the drive system with counterbalance valves of up to 30% for low effector speeds (up to 200 mm/s) and up to 15% for higher speeds (200–400 mm/s). However, due to the reduced pulsation, tasks can be carried out more efficiently because of the faster stabilization of the manipulator effector and reduced downtime between tasks.
Figure 19. Manipulator effector deviation range, $\Delta r$, for different speeds of movement.

Figure 20. Maximum pressures, $p_{\text{max}}$, for different speeds of the effector movement.

Figure 21. Pressure, $p_a$, for different speeds of movement of the effector.
Considering the stabilization time of the effector, it was found that the operating cycle time was extended by 15% for the manipulator without counterbalance valves for the effector speed of 400 mm/s (Figure 23). The use of counterbalance valves reduced these pressure pulsations, thereby shortening the working cycle time $t_c$ (Figure 24). For effector speeds of 100 and 200 mm/s, the phenomenon of pressure pulsation after stopping the effector was so small that the use of counterbalance valves did not significantly affect the total time $t_c$ of the working cycle. The same applies to damping pressure oscillations after stopping the effector (Figure 25). In the case of low velocity (up to 200 mm/s) of the effector, both for the variant with and without counterbalance valves, the system is quickly damped. Taking into account high velocity (above 200 mm/s), the use of counterbalance valves even increases the damping ratio several times and thus eliminates any possible effector vibration. Analyzing Table 4, it was found that the use of counterbalance valves does not cause changes in the pressure oscillations frequency for the considered actuators.
Figure 24. Time $t_c$ and $t_o$ for different speeds of movement of the effector for the drive system with counterbalance valves.

Figure 25. Damping ratio $\xi$, for different speeds of movement of the effector.

Table 4. Frequency of pressure $f$ for different speeds of movement of the effector.

| Hydraulic Cylinder | Variant of Hydrostatic Drive System     | Frequency, Hz |
|--------------------|----------------------------------------|---------------|
|                    |                                        | 100 mm/s | 200 mm/s | 300 mm/s | 400 mm/s |
| Boom               | with counterbalance valves             | n/a      | n/a      | 1.7      | 1.7      |
|                    | without counterbalance valves          | n/a      | n/a      | 1.7      | 1.7      |
| Arm                | with counterbalance valves             | n/a      | n/a      | n/a      | n/a      |
|                    | without counterbalance valves          | n/a      | 2        | 2        | 2        |
4. Conclusions

The presented research results confirm the influence of the use of counterbalance valves on the precision and dynamics of the manipulator’s work during the implementation of the horizontal trajectory by its working tool. On their basis, the possibility of controlling a manipulator with a classic drive system (without counterbalance valves) with an accuracy of ±100 mm for the effector speed not exceeding 200 mm/s was found. Carrying out tasks with similar accuracy for speeds greater than 200 mm/s required the use of counterbalance valves. Their use significantly limited the maximum pressures in the manipulator’s hydraulic system as well as the deflection of the effector. The use of counterbalance valves significantly reduced the pressure pulsations and effector oscillations, especially during the braking phase. Taking into account the energy aspects, a slight increase in the average pressure value in the system with counterbalance valves was noticed. As a result, the energy consumption of such a system may be increased. On the other hand, reducing pressure pulsation reduces downtime by 15% between movements, leading to faster task completion.

Taking into account such aspects as the limitation of manipulator oscillations and pressure pulsation in the hydraulic system and the improvement of the effector guidance accuracy, the use of counterbalance valves significantly improved the operation of the hydraulic manipulator. The use of counterbalance valves also improved the efficiency of tasks by reducing downtime between work cycles. Precise knowledge of the identified phenomena requires, in the next research, the development of a step model and simulations to be conducted, which will allow additional theoretical analysis of the chances and limitations of the proposed solution. It is also planned to conduct research on the influence of counterbalance valves on the operation of a heavy multi-DOF hydraulic-driven manipulator during the implementation of other trajectories.

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