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The Research of Hydraulic Pressure Intensifier for Use in Electric Drive System

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ABSTRACT This paper discusses the proposal of use of a hydraulic amplifier as a drive system's supporting element—for example, the drive of work equipment in unmanned ground vehicles (UGVs) is powered by electricity. Such a solution can provide very high forces on actuators of UGV (cutters, spreaders, hydraulic wrenches, and chisels) with relatively low electricity demand. The experimental tests of the amplifier in laboratory conditions for various power supply and load configurations of the amplifier are presented. The obtained test results allowed to estimate its usefulness in the solutions proposed above and to determine the further path of detailed research for selected electrical system solutions equipped with hydraulic amplifiers.

INDEX TERMS Actuators, drive system, hybrid power systems, hydraulic amplifier, hydraulic drive, pressure amplifier.

I. INTRODUCTION

Trends in construction of modern drive systems show that vehicles with hybrid propulsion systems are offered more often [1]. Possibility of movement utilizing electric features (accumulated in the battery's cell) has become very important in urban areas where the concentration of exhaust gases is still increasing [2]–[5]. Hybrid propulsion systems, which combines engine and electric motors, are the most popular kind of drive systems in modern vehicles [6]–[9].

The vehicles and mobile applications powered by electricity are more and more popular. The Unmanned Ground Vehicles (UGV) are a specific type of vehicle in this group (Fig. 1). Besides the drive system, there is also work equipment (e.g. manipulator) that need to be powered in these types of vehicles [10], [11]. It seems to be difficult to obtain high force from an electric motor, which is additionally inviable from an economic point of view. The high power consumption significantly reduces the battery life resulting in a shorter working time of the platform. Therefore, utilizing a combination of electric and hydraulic drive systems appear to be a good idea to resolve work equipment's power supply problem.

The possibility of generating high oil pressures in the hydraulic system results in obtaining high forces in actuators. Hydraulic pressure intensifiers emerging on the market open new opportunities in construction of special, hybrid (changing electric in hydraulic power) drive systems. This solution will significantly increase the working capacity of Unmanned Ground Vehicle with relatively low electricity consumption.

The article presents the research results on hydraulic amplifiers for purpose of evaluating its suitability for the UGV work equipment's drive system.

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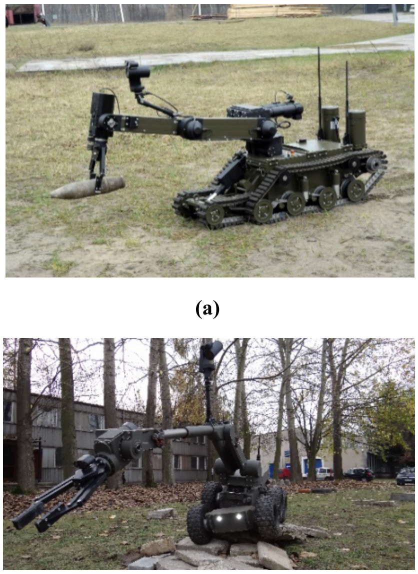
II. OBTAINING OF HIGH PRESSURE IN HYDROSTATIC DRIVE SYSTEMS

There are two ways of obtaining high pressure in hydrostatic drive systems (Fig. 2): usage of high-pressure hydraulic pumps or hydraulic pressure amplifiers. Construction of a hydraulic system is determined by the chosen method. When a high-pressure pump is used, all hydraulic components in the pressure line (e.g. hoses, valves, and actuator) have to be designed for high pressure working, resulting in an increased cost of the hydraulic system. When the pressure intensifier is used, a single component, the actuator, works with high pressure, making the hydraulic system cheaper. What is more, the system with only one component operating on high pressure is safer, because of lower damage probability. Hydraulic pressure intensifier using is good solution when only one actuator in system requires high pressure.

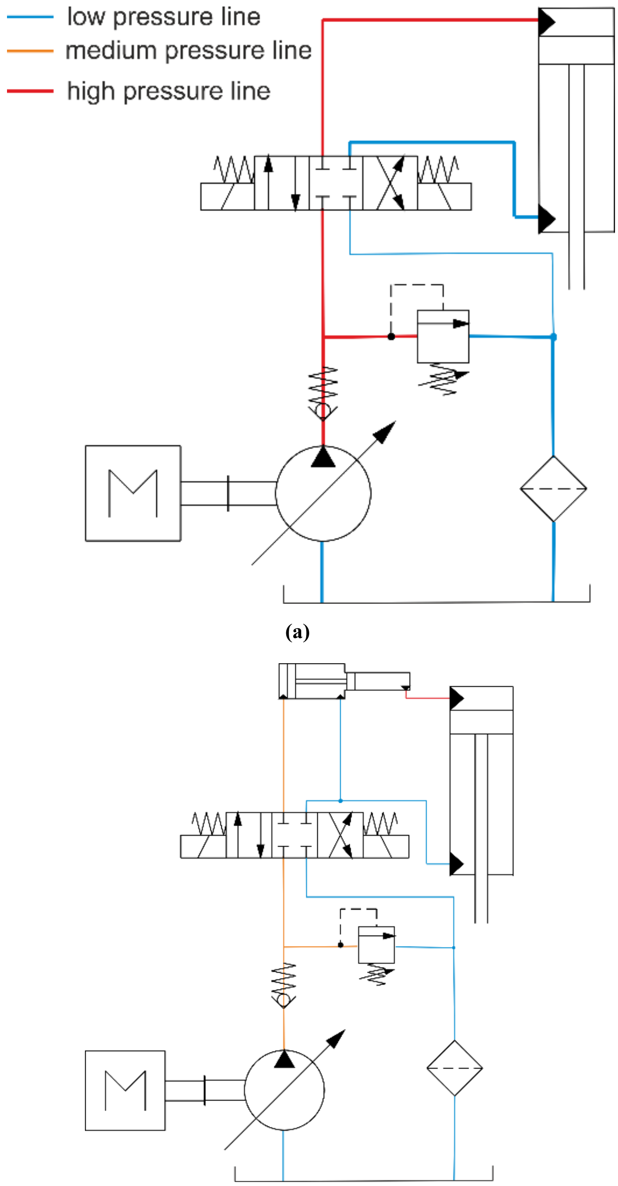
A hydraulic pressure booster's principle of operation is the same as the piston pump. The pressure acting on the surface of the piston causes the force. Then, the force moves the piston rod connected to the smaller piston which, moving in the



(a)

(b)

FIGURE 1. **Unmanned Ground Vehicles: (a) PIAP's INSPECTOR [13], (b) PIAP's IBIS [14].**



(a)

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FIGURE 2. **High pressure hydraulic systems: a) with high-pressure hydraulic pump, b) with pressure intensifier.**

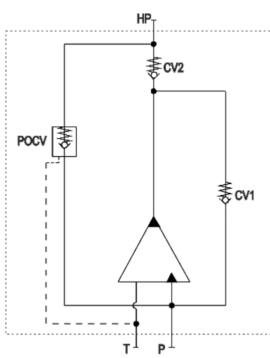


FIGURE 3. **Hydraulic pressure intensifier hydraulic scheme (description in text).**

cylinder, generates high pressure. Different kinds of hydraulic pressure boosters are describe in [15].

The construction diagram of the pressure amplifier is shown in Fig. 3. The hydraulic fluid is connected to the P (Pressure) port of entry, while the check valves, CV1, CV2 and POCV (pilot open) are connect to the HP (High-Pressure) port. Check valves are closed after the pressure value have compensated in the P and HP lines. High pressure in the HP line is received through oscillatory movement of the piston, which moves until the pressure reaches its maximum value. The movement is resumed when the pressure value in the high-pressure line decreases. POCV is open (and the high-pressure is decreasing) when there is a flow in T line.

The basic parameters for the selection of hydraulic pressure intensifier is gain factor:

$$p_g = i \cdot \Delta p = i\left(p_p - p_t\right) \tag{1}$$

In which: p_g – gain pressure; i – gain factor (describe in catalogue cart); Δp – pressure difference (between P and T line); p_p – supply pressure; p_t – pressure in T line.

The hydraulic pressure intensifier's basic information are shown in Table 1.

III. STAND FOR TESTING THE PROPERTIES OF A HYDRAULIC PRESSURE INTENSIFIER

To evaluate the possibility of using hydraulic amplifiers as high-pressure generators in hydrostatic drive systems, a Scanwill MP-T (Table 1; Fig. 4) amplifier calibrated at gain factor $i = 5$ was tested at the Institute of Machine Design, Military University of Technology.

The research was performed on a Bosch Rexroth DS 4 laboratory stand (Fig. 5a). This workstation [16] features a hydraulic power unit with a variable vane pump PV7-A supplied by an electric motor (power - 3 kW). In addition, the stand contains a hydraulic press with an 80 kg pre-load (Fig. 5b).

TABLE 1. Pressure intensifiers information.

Name	Gain factor	Max supply pressure [bar]	Max gain Pressure [bar]	Flow [dm³/min]	Weight [kg]
		miniBOOSTER			
HC1-9	5	207	500	0,4	0,75
HC2D2	12	207	800	0,6	4,15
HC8	20	207	2000	0,3	4,5
HC6H	25	207	5000	1,0	11
		Scanwill			
MP-L	2	200	400	2	9
MP-T	5	200	800	0,3	1,3
MP-2000	16	125	2000	0,1	2,7
MPL-4000	20	200	4000	2	9
		Parker			
SD 500	4	125	500	-	3

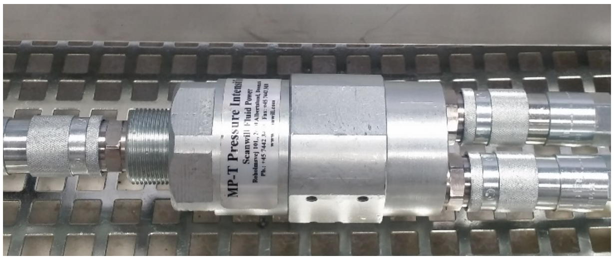
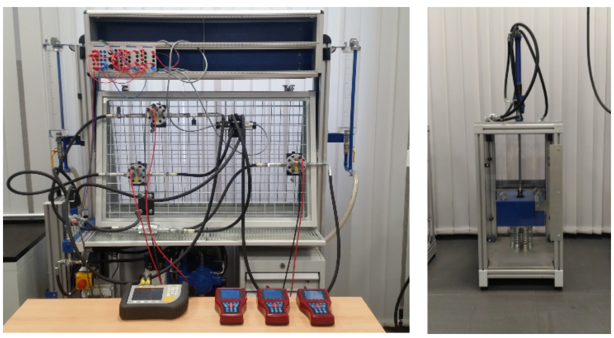


FIGURE 4. Scanwill MP-T pressure intensifier.



(a) (b)

FIGURE 5. Bosch Rexroth DS 4 workstation: a) stand, b) hydraulic press.

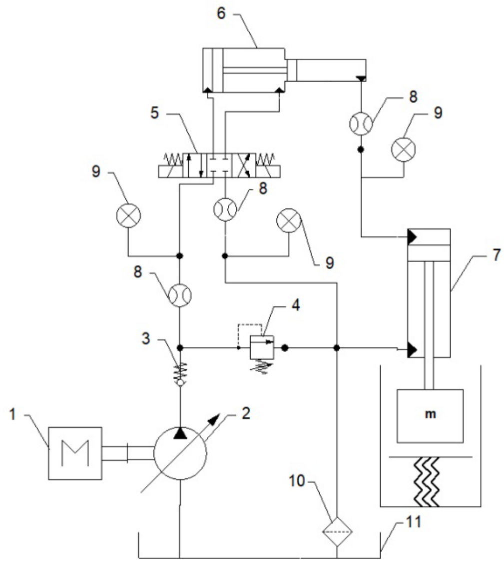


FIGURE 6. Hydraulic system loaded by throttle valve: 1 – motor, 2 – hydraulic pump, 3 – check valve, 4 – pressure relief valve, 5 – shut off valve, 6 – pressure intensifier, 7 – throttle valve, 8 – flow sensor, 9 – pressure sensor, 10 – filter, 11 - tan.

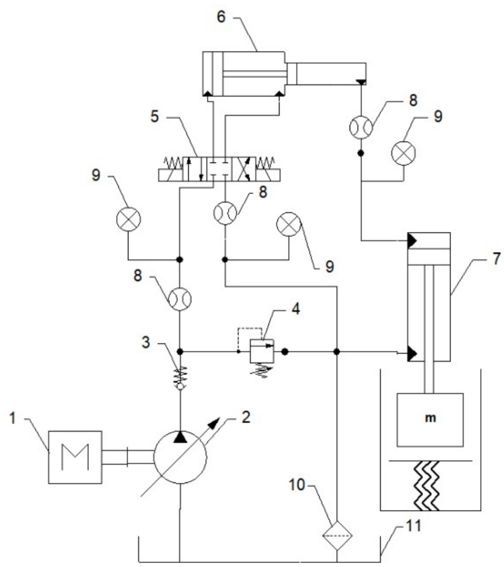


FIGURE 7. Hydraulic system: 1 – motor, 2 – hydraulic pump, 3 – check valve, 4 – pressure relief valve, 5 – directional control valve, 6 – pressure intensifier, 7 – hydraulic cylinder, 8 – flow sensor, 9 – pressure sensor, 10 – filter, 11 - tank.

Two configurations of the hydraulic system were used for the research conducted. First, testing of the hydraulic pressure intensifier was performed with a manual control throttle valve as an intensifier load (7 - Fig. 6). The load was generated by reducing flow through the throttle valve, as long as the valve was totally closed. The supply pressure was provided by a manual control pressure relief valve (4 - Fig. 6). The hydraulic pressure intensifier's oil supplies were controlled by a shuttle valve (5 - Fig.6) opening and/or closing.

For identifying the amplifier's performance parameters in the second testing configuration, a hydraulic press was used as a loading element (Fig. 7). This allowed the determination of the performance characteristics of the amplifier for a smoothly increasing load. As in the preliminary research, the pressure relief valve was a pressure limiter. Flow direct control was provided by 4/3-way directional control valve (5-Fig. 7) with max. operating pressure at 350 bar.

As part of the conducted tests, the hydraulic oil's pressure and flow values were recorded, both in the hydraulic amplifier's supply line, in the high-pressure line and in the sink line.

In both cases, the measuring system consisted of three gear volumetric flow sensors (with a measuring range of $0 \div 30$ dm^3/min) and three pressure sensors, two sensors in the CAN system with a measuring range of $0 \div 600$ bar and an analog sensor with a measuring range of $0 \div 400$ bar. The measuring elements have been placed in the following configuration:

- in the supply line, before the pressure amplifier: flowmeter and CAN pressure sensor;
- in the high pressure line: flowmeter and CAN pressure sensor;
- in the sink line: flowmeter and analog pressure sensor.

For the purposes of recording the results, Parker Service Master Plus data acquisition systems and Hydrotechnik 4010 were used for pressure value registration and recording of flow rates.

IV. RESEARCH AND ANALYSIS OF RESULTS

Tests of the hydraulic amplifier were carried out for two different parameters regarding the hydraulic power unit using the throttle valve and hydraulic press as loading elements.

In the first attempt, the hydraulic pump displacement was set to $Q_1 = 8$ dm^3/min and the pressure relief valve p_{w1} - to an opening pressure of 30 bar. For such parameters of the hydraulic system, the hydraulic amplifier was first loaded with a throttle valve (until the valve was completely closed). Then, the load was simulated by the hydraulic press (until the springs supporting the weight of the press cylinder were blocked). The results of the experimental tests have shown that the values of pressure in the hydraulic system were specified. The curves of the recorded pressure values are shown in Fig. 8. The values of the hydraulic amplifier's supply pressure are marked as follows: 1 - for throttle valve's load, 2 - for hydraulic press's load. The pressure values in the gain line are marked as follows: 3 - for throttle valve's load, 4 - for hydraulic press's load. The theoretical pressure values in the pressure gain was determined as follows: 5 - for throttle valve's load, 6 - for hydraulic press's load. The theoretical pressure values in the pressure gain was calculated using (1).

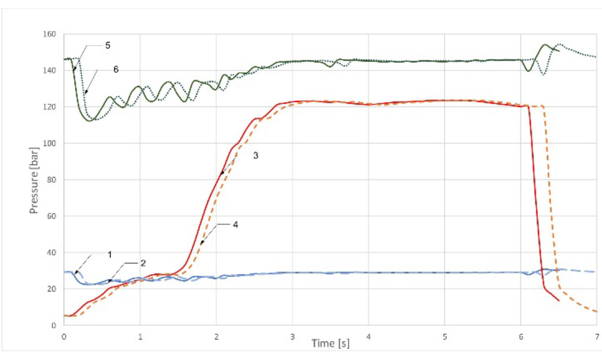


FIGURE 8. Pressure curves – the load simulated by throttle valve and hydraulic press for $Q_1 = 8$ dm3/min, $p_{w1} = 30$ bar (description in text).

Results from the tests indicate an increase of pressure in the HP line. In the case of loading the system by the throttle valve, the pressure reached a maximum value of 123 bar, while the load simulated by the hydraulic press reached a maximum value of 124 bar. The conducted research allowed the determination of the actual amplification factor, which is variable depending on the load size. For both studied cases, this coefficient ranges from 4.2 to 4.3.

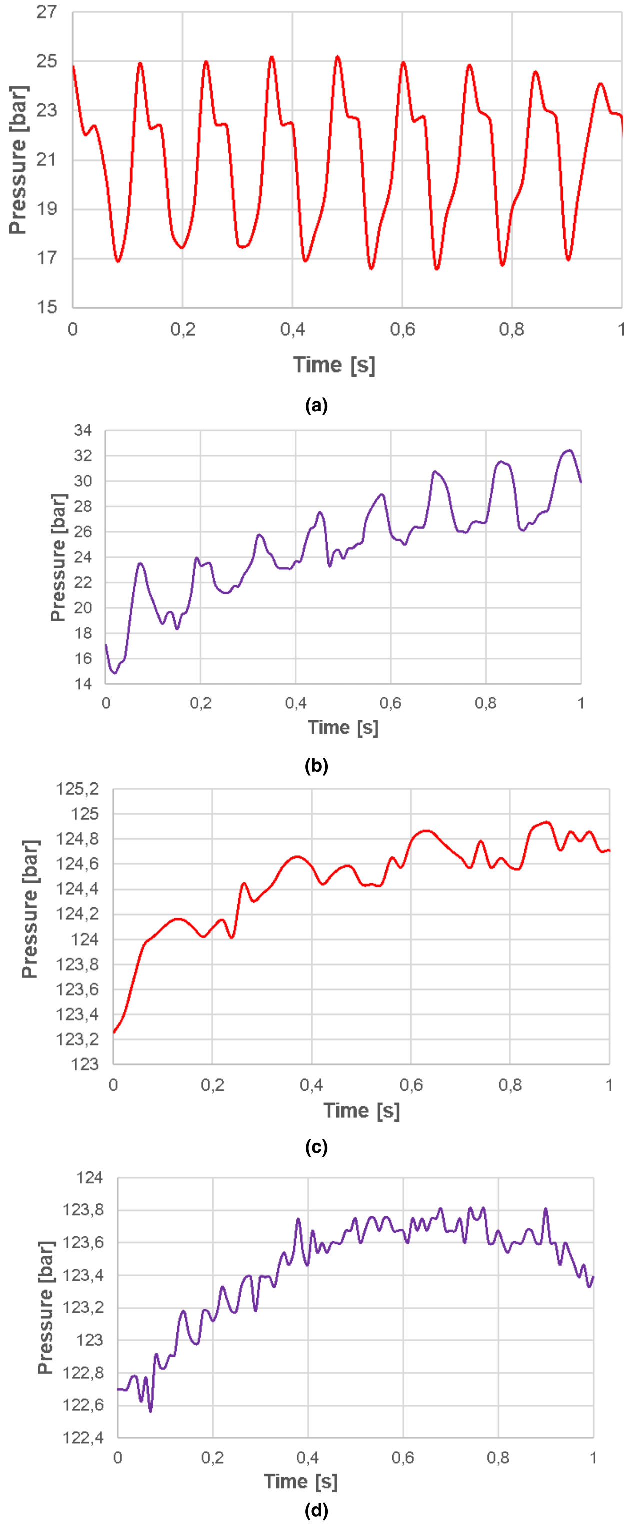


FIGURE 9. Pressure pulsation in HP line: (a) pressure compensation phase – load simulated by the throttle valve, (b) pressure compensation phase – load simulated by the hydraulic press, (c) maximum working pressure phase – load simulated by the throttle valve, (d) maximum working pressure phase – load simulated by the hydraulic press.

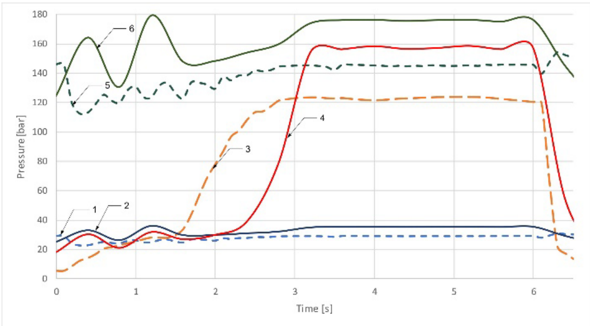


FIGURE 10. Pressure curves – the load simulated by throttle valve for various system parameters (description in text).

The obtained test results confirmed the pulsating nature of work of the hydraulic amplifiers (Fig.9). The nature of pulsation is various and depends on the phase of the amplifier's operation. The largest oscillations are observed in the phase of pressure compensation in the hydraulic amplifier's supply line and the high-pressure line. When the maximum operating pressure is reached, these fluctuations stabilize. In the case of loading the system with a hydraulic press in the pressure compensation phase, pressure pulsation occurs at the level of 8.3 Hz with an amplitude of approx. 5.5 bar, whereas after reaching the maximum value in the high-pressure line, the oscillation disappears (pressure fluctuations in the range $0 \div 1.2$ bar are observed). In the case of the load performed by the throttle valve in the pressure compensation phase, pressure pulsation is also observed at the level of 8.3 Hz with an amplitude of approx. 8 bar. Similar to the load case with a hydraulic press, when the maximum pressure has been reached, the pulsation disappears - pressure fluctuations do not exceed 1.4 bar.

In the second attempt, the hydraulic pump displacement was increased to the value of $Q_2 = 11$ dm^3/min, and the pressure relief valve was set to the opening pressure at the level $p_{w2} = 35$ bar. The results of tests carried out for various system parameters ($Q_1 = 8$ dm^3/min, $p_{w1} = 30$ bar, $Q_2 = 11$ dm^3/min, $p_{w2} = 35$ bar) and the same nature of the load (the throttle valve) of the hydraulic amplifier are shown in Fig. 10.

The values of the hydraulic amplifier's supply pressure were marked as follows: 1 - for Q_1, p_{w1}; 2 - for Q_2, pw_2. The pressure values in the gain line are marked as follows: 3 - for Q_1, p_{w1}; 4 - for Q_2, p_{w2}. The theoretical value of pressure gain was marked as follows: 5 - for Q_1, p_{w1}; 6 - for Q_2, p_{w2}. The theoretical pressure values in the pressure gain was calculated using (1).

With the increase of the flow rate, the nature of the pressure's curves change - for the flow rate Q_1 this curve is exponential, while the flow rate Q_2 has linear characteristics. For the flow rate Q_2, the gain factor takes a much wider range of values - from 4.4 to 5.5. Due to the different amplifier's supply pressures, there are large differences between the maximum gain pressures - for the supply pressure value p_{w1} the gain pressure value reaches 123 bar, for the value of the input pressure p_{w2} the gain pressure value reaches 158 bar.

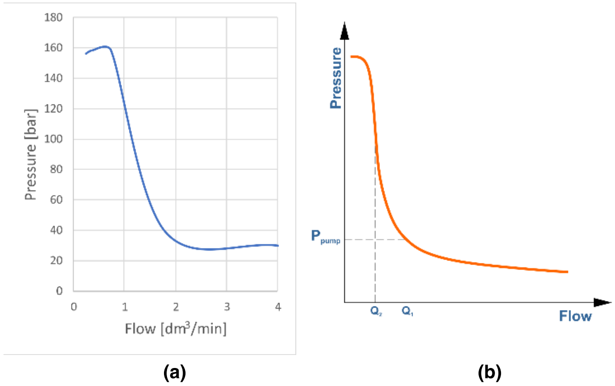


FIGURE 11. Connection between gain pressure and flow: (a) practical, (b) theoretical [17].

The conducted tests allowed the determination of the connection between the gain pressure and the flow rate of hydraulic oil supplying the amplifier. The determined characteristics were compared with the example characteristics of the hydraulic amplifier given by the manufacturer (Fig. 11). Subsequently, the nature of the curves were defined as hyperbolic assuming the highest pressure (158 bar) for the smallest flow rate (0.4 dm^3/min).

V. CONCLUSIONS

The research allowed the formation of preliminary conclusions regarding the nature of the work of hydraulic amplifiers:

– pressure boosting takes place from the moment of pressure compensation in the supply line and high pressure line;
– the actual value of the gain factor is not constant and depends on the nature of the load and the flow rate value;
– the construction of the hydraulic pressure amplifier has entailed the pulsating nature of its operation and hence pressure pulsations in the system, which may exclude the use of a pressure amplifier in certain application areas.

The above conclusions make it advisable to carry out tests for selected hydraulic amplifiers. Particular emphasis should be placed on the area from the start of pressure compensation in the hydraulic amplifier's supply and gain pressure line until the maximum pressure in the system is stabilized. This researches will allow to provide some solutions to lower the pressure pulsation. The impact of using pressure amplifiers to the system dynamic response will being examined too. For the purposes of these tasks, it may be reasonable to build a specialized workstation dedicated to research on high-pressure (up to 700 bar) systems.

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