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TESTING OF REGULATING AND NON-REGULATING HYDRAULIC PUMPS

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This article deals with the testing of regulating and non-regulating hydraulic pumps in laboratory conditions. The laboratory test was carried out on the hydraulic device consisting of two separate hydraulic circuits (for testing the non-regulating hydraulic pump and regulating hydraulic pump). A throttle valve or proportional throttle valve is used for loading. During the laboratory tests, the flow efficiency of the non-regulating hydraulic pump at rotation speed n = 1,500 rpm (nominal speed) was η_{pp} = 98.91% (0 MPa) and η_{pp} = 94.97% (20 MPa), and for the regulating hydraulic pump, the values were η_{pp} = 97.57% (0 MPa) and η_{pp} = 94.54% (20 MPa). The determined values of flow efficiency of hydraulic pumps determine the correct operation of the experimental hydraulic laboratory equipment.

Keywords: fluid flow measurement; hydraulic systems; hydraulic equipment

Hydraulic circuits are an integral part of agricultural and forestry tractors. Flow efficiency is the most important technical parameter of the hydraulic pump (Inaguma, 2012; Kosiba et al., 2022). The manufacturers of lubricants have to increase the quality of fluids used in the hydraulic systems of tractors. According to Nedeljkovic et al. (2018), hydraulics pumps are among the most frequently used machines in agriculture. The hydraulic system and hydraulic fluid must be monitored during the whole operation (Tkáč et al., 2018; Fangwei et al., 2017). The hydraulic laboratory device is based on the hydraulic circuits of agricultural tractors. With laboratory equipment, it is possible to ensure the repeatability of measurements of flow characteristics.

The calculated flow efficiency defines technical service durability and operational capability (Renn and Tsai, 2005). The importance of technical diagnostics of a hydraulic pump is in the monitoring of flow characteristics, which can be considered as a predictive maintenance of hydraulic pumps (Hryciów et al., 2021). The failure of a hydraulic pump is signalled by a decrease in flow characteristics (Wu et al., 2012). An important parameter in evaluating the flow characteristics is dynamic viscosity, which can significantly change the flow efficiency (Trivedy and Bhatt, 2017; Kučera and Hnilicová, 2022).

The main aim of this publication is to measure the hydraulic pump flow characteristics on the hydraulic laboratory device (regulating and non-regulating hydraulic pump). The hydraulic device is designed to test both types of hydraulic pumps simultaneously. The regulating and non-regulating pumps have their own hydraulic section (tank, cooler, and filtration). The first is used for the testing of hydraulic pumps with an inclined plate (regulating) and the second for the testing of gear (non-regulating). The innovativeness of the hydraulic device lies in its dual use. Within one test, it is possible to test a regulating and a non-regulating hydraulic pump.

Material and methods

Hydraulic device for dual testing of non-regulating and regulating pumps

The hydraulic device is shown in Fig. 1. The hydraulic circuit A is used for testing the regulating hydraulic pump (HP1). An electric motor with an output of P = 30 kW is used for driving the hydraulic equipment. The circuit contains a pressure valve (PV_1) in the form of a safety valve, i.e., after exceeding the pressure of p = 20 MPa, the fluid is released back into the tank (T_1). A turbine flow feeler (Q_1) is used to measure the hydraulic fluid flow. A 10 µm filter (F_1) and a cooler (C_1) are connected to the outlet, which maintained the setting temperature of t = 40 °C. A regulating hydraulic motor (HM) is used to drive the second hydraulic circuit (B).

The hydraulic circuit B is used for the testing of non-regulating hydraulic pumps. The circuit contains a pressure valve (PV_2) in the form of a safety valve, i.e., after exceeding the pressure of p = 20 MPa, the fluid is released back into the tank (T_2). A turbine flow feeler (Q_2) is used for measuring the flow characteristics. A mechanical throttle valve (RV) or a proportional throttle valve (PRV) can be used for deriving the load of the hydraulic pump. A three-way valve (TV) is used for connecting the selected type of loading. When using a mechanical throttle valve (RV), an increase in load pressure is achieved by a mechanical change in the flow cross-section. The proportional throttle valve (PRV) is used

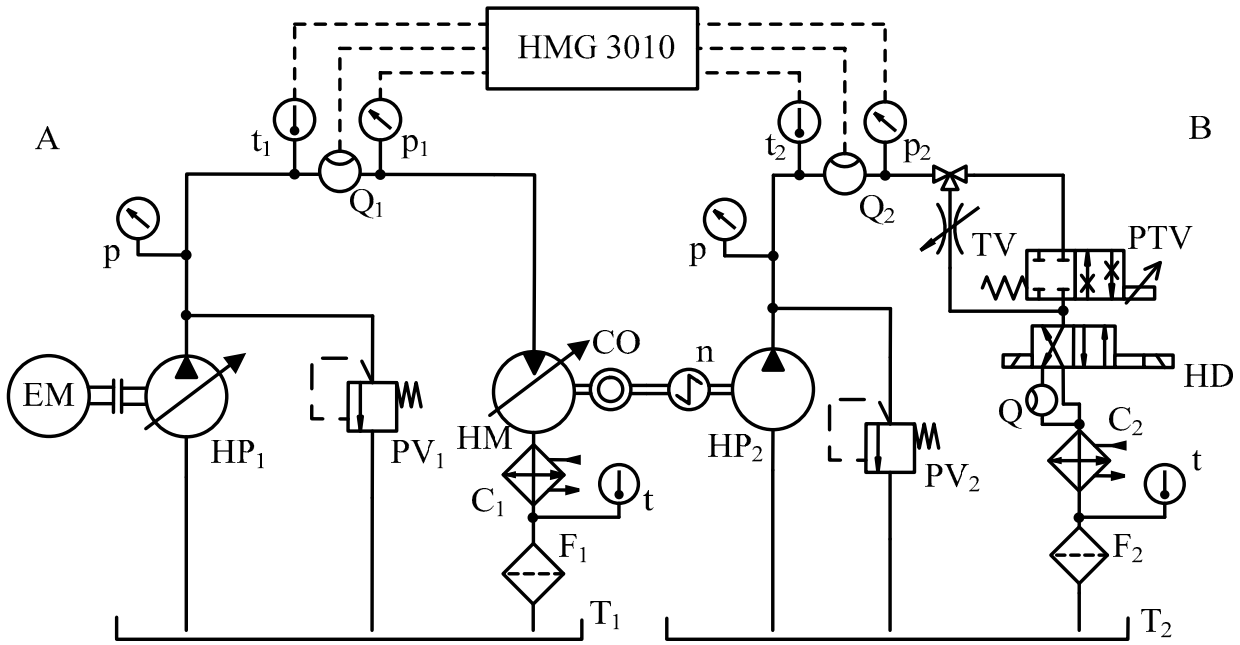


Fig. 1 Design of the laboratory equipment
Source: Kosiba, 2017
A – primary circuit; B – secondary circuit; EM – electric motor; HP1 – regulating hydraulic pump; HP_2 – non-regulating hydraulic pump; HM – hydraulic motor; F_1, F_2 – filter; T_1, T_2 – tank; C_1, C_2 – cooler; CO – coupler; TV – throttle valve; PTV – proportional throttle valve; p – pressure sensor; t – temperature sensor; Q – flow rate sensor; n – speed sensor

for changing the fluid flow (increasing the load pressure) and it is controlled by an electromagnet, which acts on the compression spring.

The other alternative for testing of non-regulating hydraulic pumps (hydraulic circuit B) is to use the volume flow feeler (Q). Flow control through the volume flow indicator is by means of a two-position distributor. Using a digital starter, the two-position distributor is switched to the position of fluid discharge through the flow indicator during t = 1 min. The mentioned measuring of hydraulic fluid flow is possible using the mechanical throttle valve (TV) or proportional throttle valve (PTV).

To monitor the temperature in both hydraulic sections, expansion thermometers are placed in both tanks. To monitor the pressure in the hydraulic circuit, manometers (p) or pressure feelers are used. An inductive rotation speed feeler is used for scanning the speed on the mechanical clutch (SP). A portable data recorder HMG 3010 is used for capturing the values of flow rates, temperatures, and pressures, as well as rotation speed values.

The HM 46 hydraulic fluid is used in the hydraulic circuit (circuit A and B). Its basic technical indicators are given in Table 1.

Table 1 The basic technical indicators of the fluid

Parameter	Unit	Value
Kinematic viscosity at 40 °C	$mm^2 \cdot s^{-1}$	48.48
Density at 15 °C	$kg \cdot m^{-3}$	0.876
Flash point	°C	220
Pour point	°C	-27

Determination of flow efficiency

The real flow through the hydraulic pump is obtained when the theoretical flow is reduced by the loss of fluid flow:

$$Q_S = Q_T - Q_{LF} \quad (1)$$

where: Q_S – actual flow ($dm^3 \cdot rpm$); Q_T – theoretical flow ($dm^3 \cdot rpm$); Q_{LF} – loss flow ($dm^3 \cdot rpm$)

Theoretical hydraulic pump flow Q_T (Dobrota et al., 2010):

$$Q_T = V_G \cdot n = q_p \cdot 2 \cdot \pi \cdot n = q_p \cdot \omega \quad (2)$$

where: V_G – geometric volume (dm^3); q_p – fluid flow per revolution ($dm^3 \cdot rpm$); n – rotation speed (rpm); ω – angular speed ($rad \cdot s^{-1}$)

According to Dobrota et al. (2010), the real flow of fluids Q_S depends on geometric volume and dynamic efficiency. After substituting into Eq. (2), we obtain:

$$Q_s = Q_T - Q_{LF} = Q_T - \frac{V_G \cdot \Delta p}{\mu} = Q_T - \frac{V_G (p_t - p_v)}{\mu} \quad (3)$$

where: Δp – pressure drop on the hydraulic pump (MPa); p_p – pressure at the inlet of the hydraulic pump (MPa); p_v – pressure at the outlet of the hydraulic pump (MPa); V_G – geometric volume of the hydraulic pump (m^3); μ – dynamic viscosity (Pa · s)

Flow efficiency is defined based on the formula:

$$\eta_p = \frac{Q_s}{Q_T} = \frac{Q_T - Q_{LF}}{Q_T} = 1 - \frac{Q_{LF}}{Q_T} \quad (4)$$

After substituting and expressing Eqs (3) and (4), we obtain the relation for calculating the flow efficiency (Dobrota et al., 2010):

$$\eta_p = 1 - \frac{V_G(p_p - p_v)}{Q_T \cdot \mu} \quad (5)$$

After substituting and expressing Eqs (2), (3), and (5), we obtain the relation for the calculation and determination of flow efficiency. The actual fluid flow Q_S was not based on Eq. (4) but on the actual determination of the hydraulic pump:

$$\eta_p = \frac{Q_s}{V_G \cdot n} \quad (6)$$

or

$$\eta_p = \frac{Q_s}{V_G \cdot n} \cdot 100 \quad (7)$$

Mechanical efficiency is given by the formula (Baroška, 2012):

$$\eta_m = \frac{p_T}{P} = \frac{p_p \cdot V_G \cdot n}{P} = \frac{p \cdot Q_T}{P} \quad (8)$$

where: p_T – hydraulic pump theoretical power (W); P – hydraulic pump power input (W); p_p – pressure at the outlet of the hydraulic pump (MPa); V_G – geometric volume (dm^3); Q_T – theoretical flow rate (m^3·rpm)

Gross efficiency is given by the relation:

$$\eta_c = \eta_p \cdot \eta_m \quad (9)$$

After substituting and expressing Eqs (5), (8), and (9), we obtain the relation for the calculation of gross efficiency:

$$\eta_c = \frac{p \cdot Q_T}{P} = \frac{P_G}{P} \quad (10)$$

where: P_G – output power of the hydraulic pump (W)

Dimensioning of hydraulic circuit accessories

The hydraulic components of the designed hydraulic device were based on formulas according to Baroška (2012), Tottem and De Negri (2012), Will and Gebhardt (2011), Sigloch (2009), and Hujo et al. (2022).

Tube sizing

The infer surface of the tube:

$$d_p = \sqrt{\frac{4 \cdot Q}{\pi \cdot w}} \quad (11)$$

where: Q – fluid flow (m^3·rpm); w – flow resistance (rpm)

Tube wall thickness:

$$S = \frac{p_{max} \cdot d_p}{2 \cdot \sigma_{per}} \cdot k \quad (12)$$

where: p_{max} – maximum tube pressure (MPa); d_p – inner surface of the tube (m^2); σ_{per} – permissible tensile stress (MPa)

Tank sizing

The volume of the tank:

$$V_{tan\,k} = V_{HM} + V_{HG} + V_H + V_{20\%} + V_p \quad (13)$$

where: V_{HM} – hydraulic motor volume (m^3); V_{HG} – hydraulic pump volume (m^3); V_H – volume of hydraulic components (m^3); $V_{20\%}$ – oversizing (%); V_p – tube volume (m^3)

The volume of the cooler:

$$P_v = \frac{\Delta T \cdot c_{01} \cdot \rho_{01} \cdot V_{tan\,k}}{t \cdot 60} \quad (14)$$

where: ΔT – temperature difference (°C); C_{01} – heat capacity (°C); ρ_{01} – density (kg·m^{-3}); $V_{tan\,k}$ – tank volume (m^3)

Specific cooling capacity:

$$P_{01} = \frac{P_v}{T_1 - T_2} \quad (15)$$

where: P_v – cooler power (W); T_1 – maximum temperature (°C); T_2 – minimum temperature (°C)

Cooling power:

$$P_0 = P_{01} \cdot 1.1 \quad (16)$$

where: P_{01} – specific cooling capacity (W·°C^{-1})

Equations (11)–(16) were used to determine the experimental laboratory device components:

- inner surface (minimum) – 9.86 mm^2;
- tube wall thickness (minimum) – 1.21 mm;
- tank volume (minimum) – 57 dm^3;
- cooling power (minimum) – 4.66 kW;
- specific capacity (minimum) – 116.5 × 10^{-3} W·°C^{-1};
- cooling power (minimum) – 121.5 × 10^{-3} W·°C^{-1}.

The catalogue data of the hydraulic parts:

- tube diameter d = 14 mm;
- wall thickness s = 1.5 mm;
- tank volume $V_{tan\,k}$ = 80 dm^3.

An EMMEGI cooler of type 2024K with a cooling capacity of P_0 = 0.2 kW·°C^{-1} is used for cooling.

Results and discussion

Figure 2 shows the effect of rotation of a pair of gear wheels in a non-regulating hydraulic pump on fluid flow. Figure 3 shows the pressure dependence of operation of the non-regulating hydraulic pump at rotation speed n = 1,500 rpm. Fluid flow values were recorded at a sampling frequency (Hz) of 1 ms, for t = 10 s. The measured non-regulating hydraulic

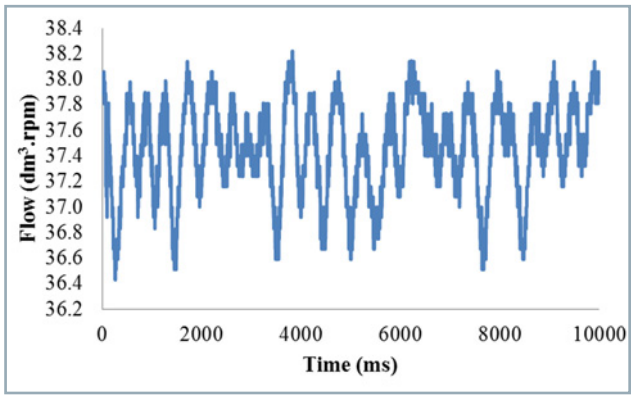


Fig. 2 Flow characteristic at rotation speed n = 1,500 rpm

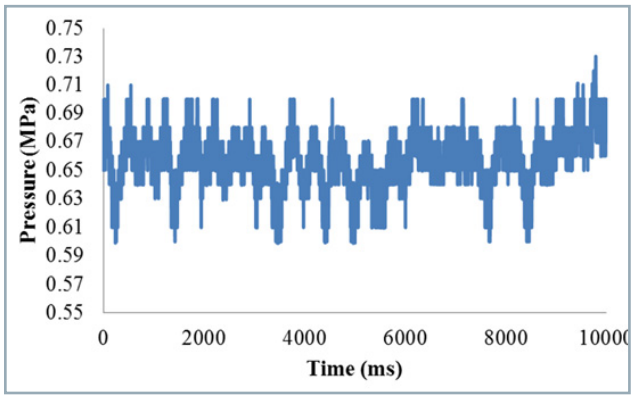


Fig. 3 Pressure dependence at rotation speed n = 1,500 rpm

pump has 12 pairs of teeth on the gears. Measurement was performed at the nominal speed of rotation. Table 2 shows the basic statistical evaluation of flow and pressure characteristics of hydraulic pump operation.

Due to the rolling of gears, the values of fluid flow rates and pressures change over time. Based on the theoretical elaboration of Hao et al. (2016) and Casoli et al. (2005), it is possible to determine the coefficient of loss of fluid flow.

Discharge of non-regulating hydraulic pump:

$$q = 2z_m(V_1 - V_{los}) = 2z_m\left[\frac{n}{360}\cdot\pi\cdot r_{a1}^2 - \int f(x_0, y_0)dx - V_{los}\right] \quad (17)$$

where: z_m – number of gears; V_1 – tooth gap volume (m^3); n – rotation speed (rpm); V_{los} – lossy volume (m^3); ra_1 – gear head radius (m)

Lossy volume:

$$V_{los} = \int_{xj1}^{xj2}\left[f(x_2, y_2) - f(x_1, y_1)dx\right] \quad (18)$$

The time changes of flow can be determined according to Eq. (19):

$$\bar{Q} = \frac{\omega}{2\pi}\cdot q \quad (19)$$

where: ω – angular velocity (rad.s^{-1})

Fluid loss coefficient:

$$k_Q = \frac{Q_{max} - Q_{min}}{\bar{Q}} \quad (20)$$

where: Q_{max} – maximum fluid flow (m^3·rpm); Q_{min} – minimum fluid flow (m^3·rpm)

Figure 4 shows the hydraulic fluid flow of the regulating hydraulic pump at speed n = 1,500 rpm. The measured regulating hydraulic pump has 7 pistons. Figure 5 shows the pressure dependence of operation of the regulating hydraulic pump at rotation speed n = 1,500 rpm. Fluid flow and pressure values were recorded at a sampling frequency (Hz) of 1 ms during t = 10 s. Table 3 shows the basic statistical evaluation of the measurement of flow and pressure characteristics.

According to Changbin et al. (2014), it is possible to determine the fluid flow through one cylinder:

$$Q_p = sign(p_{p0} - p_{pi})\cdot C_p\cdot S_p\sqrt{\frac{2(p_{p0} - p_{pi})}{\rho}} \quad (21)$$

where: p_{p0} – pressure at the outlet of the piston chamber (MPa); p_{pi} – instantaneous pressure in the piston chamber (MPa); C_p – piston chamber flow coefficient; S_p – piston area (m^2); ρ – fluid density (kg·m^3)

Table 2 Basic statistical evaluation of measured data

	Average	Median	Modus	Min.	Max.	Variance	Standard deviation	Coeff. of variation
	$\bar{x}$	Me	Mo	x_{min}	x_{max}	σ^2	σ	V_k
Flow (dm^3·rpm)	37.46	37.89	37.93	36.43	38.22	0.12754	0.35712	0.9534
Pressure (MPa)	0.66	0.67	0.068	0.6	0.73	0.0003	0.01737	2.64361

Table 3 Basic statistical evaluation of measured data

	Average	Median	Modus	Min.	Max.	Variance	Standard deviation	Coeff. of variation
	$\bar{x}$	Me	Mo	x_{min}	x_{max}	σ^2	σ	V_k
Flow (dm^3·rpm)	72.79	72.8	72.5	69.1	75.99	1.31689	1.14756	1.71809
Pressure (MPa)	0.99	0.96	0.95	0.5	1.75	0.06847	0.2616	26.4945

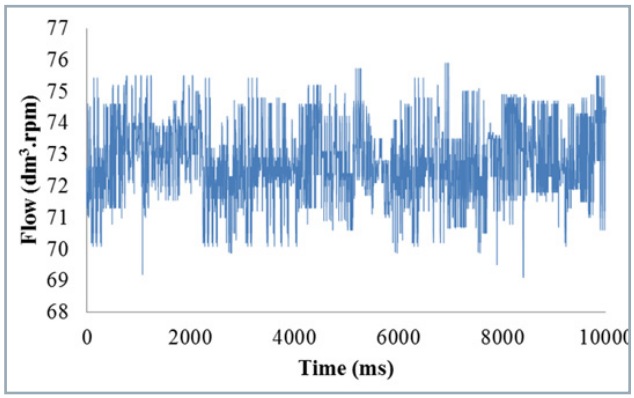


Fig. 4 Flow characteristic at rotation speed n = 1,500 rpm

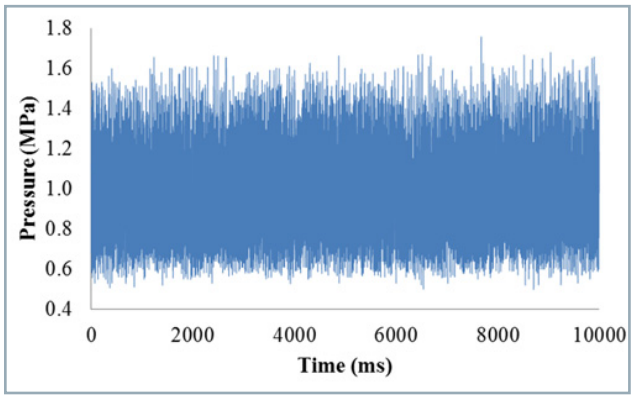


Fig. 5 Pressure dependence at rotation speed n = 1,500 rpm

Measurement of operation of the designed laboratory device

The operation of the experimental hydraulic device was verified based on the flow characteristics of the regulating and non-regulating hydraulic pump depending on the change of load pressure at their rated rotation speed and subsequent calculation of their flow efficiencies.

The verification of hydraulic fluid flow is performed at nominal speed (rpm) and with a gradual increase of loading pressure in the interval from p = 0 MPa to p = 20 MPa. The geometric volume of the non-regulating hydraulic pump is VG = 0.025 m^3. Table 4 shows the flow rate characteristics of the non-regulating hydraulic pump with basic statistical evaluation.

The flow efficiency of the non-regulating hydraulic pump at pressure p = 0 MPa reached η_p = 98.91% at fluid flow Q = 37.25 dm^3·rpm, and at pressure p = 20 MPa reached η_p = 94.97% at fluid flow Q = 35.75 dm^3·rpm.

The verification of the hydraulic fluid flow of the regulating hydraulic pump is performed at rotation speed n = 1,500 rpm and with a gradual increase of loading pressure in the interval from p = 0 MPa to p = 20 MPa. Table 5 shows the hydraulic fluid flow of the regulating hydraulic pump with basic statistical evaluation.

The flow efficiency of the regulating hydraulic pump at pressure p = 0 MPa reached the value η_p = 97.57% at fluid flow Q = 73.18 dm^3·rpm, and at pressure p = 20 MPa reached η_p = 94.44% at fluid flow Q = 70.84 dm^3·rpm.

A three-way valve is used to choose how the hydraulic pumps will be loaded. Throttle loading was used by Inaguma and Yoshida (2013) and Rana (2015) in their hydraulic device. A proportional throttle valve was used for loading by Kim and Kim (2013) and Murrenfoff (2014).

In the non-regulating hydraulic pump, flow efficiency was found to be η_{pp} = 98.91% at pressure p = 0 MPa and rotation speed n = 1,500 rpm (nominal rotation speed), and η_{pp} = 94.97% at pressure p = 20 MPa. In the regulating hydraulic pump, flow efficiency was found to be η_{pp} = 97.57% (0 MPa) and η_{pp} = 94.45% (20 MPa). The decrease of flow efficiency is in accordance with the results published by Dobrota et al. (2010) (non-regulating hydraulic pump) and by Bergada et al. (2012) (regulating hydraulic pump). The dependence of pressure change at the output of the regulating hydraulic pump coincides with the data found by Kučera and Aleš (2017).

Table 4 Flow characteristics of non-regulating hydraulic pump at rotation speed n = 1,500 rpm

Pressure (MPa)	Average	Variance	Standard deviation	Coeff. of variation	Probability of normal distribution 68.27%			
					average	variance	standard deviation	coeff. of variation
	$\bar{x}$	σ^2	σ	V_k	$\bar{x}$	σ^2	σ	V_k
0	37.28	0.04225	0.20555	0.55136	37.25	0.03741	0.19342	0.51924
2	37.12	0.04185	0.20457	0.55111	37.1	0.0371	0.19261	0.51917
4	37.01	0.04005	0.20012	0.54073	36.98	0.03682	0.19189	0.51889
6	36.88	0.0396	0.199	0.53958	36.84	0.03627	0.19045	0.51696
8	36.47	0.03715	0.19274	0.5285	36.44	0.0359	0.18947	0.51996
10	36.32	0.03622	0.19032	0.524	36.3	0.03412	0.18472	0.50886
12	36.15	0.03579	0.18918	0.52333	36.16	0.03196	0.17877	0.4944
14	36.05	0.0342	0.18493	0.51299	36.02	0.02987	0.17283	0.47982
16	35.9	0.03298	0.1816	0.50586	35.91	0.0282	0.16793	0.46764
18	38.88	0.03052	0.1747	0.44933	35.88	0.02749	0.1658	0.42644
20	38.78	0.02814	0.16775	0.43257	35.75	0.02622	0.16193	0.41787

Table 5 Flow characteristics of regulating hydraulic pump at rotation speed n = 1,500 rpm

Pressure (MPa)	Average	Variance	Standard deviation	Coeff. of variation	Probability of normal distribution 68.27%			
					average	variance	standard deviation	coeff. of variation
	$\bar{x}$	σ^2	σ	V_k	$\bar{x}$	σ^2	σ	V_k
0	73.2	0.06985	0.26429	0.36105	73.18	0.05844	0.24174	0.33034
2	72.84	0.06988	0.26435	0.36292	72.85	0.05801	0.24085	0.33061
4	72.62	0.07005	0.26467	0.36446	72.65	0.05869	0.24226	0.33346
6	72.41	0.06822	0.26119	0.36071	72.4	0.0578	0.24042	0.33207
8	72.12	0.06744	0.25969	0.36008	72.09	0.05633	0.23734	0.32923
10	71.95	0.06522	0.25538	0.35494	71.86	0.05399	0.23236	0.32335
12	71.78	0.06536	0.25566	0.35617	71.71	0.05395	0.23227	0.3239
14	71.63	0.06421	0.2534	0.35376	71.59	0.05287	0.22993	0.32118
16	71.42	0.0638	0.25259	0.35366	71.45	0.05247	0.22906	0.32059
18	71.19	0.06387	0.25273	0.355	71.2	0.0526	0.22935	0.32212
20	70.8	0.06341	0.25181	0.35567	70.84	0.05187	0.22775	0.32151

Conclusion

Hydraulic pump operating conditions significantly affect the pump efficiencies. It is very important to understand how pump efficiencies depend on hydraulic pump operating conditions (Kosiba et al., 2013, Aliboev, 2016). As summarized in the results and discussion, the laboratory tests of hydraulic pumps are extremely important. The proposed laboratory hydraulic device enables the laboratory tests of two types of hydraulic pumps – the non-regulating and the regulating hydraulic pump.

Based on the methodology and theoretical calculations, an experimental hydraulic device was constructed. During the testing, flow efficiency was found to be η_{pp} = 98.91% at pressure p = 0 MPa and η_{pp} = 94.97% at pressure p = 20 MPa. In the regulating hydraulic pump, at rotation speed n = 1,500 rpm (nominal rotation speed), flow efficiency was found to be η_{pp} = 97.57% (at pressure p = 0 MPa) and η_{pp} = 94.54% (at pressure p = 20 MPa).

The determined values of the flow efficiency of the hydraulic pumps determine the correct operation of the designed experimental hydraulic laboratory equipment.

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