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# Experimental Tests of the Piston Axial Pump with Constant Pressure and Variable Flow

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**Abstract:** Constant pressure variable flow reciprocating axial pumps (CPAP) are used in various applications, where a constant output pressure is maintained when the flow rate changes. When the hydraulic system is at rated pressure or less, the swash plate has maximum tilt, and the pump delivers maximum flow. The swash plate comes into this position thanks to the action of a reactive piston in which there are two springs. However, when the pressure rises above the nominal pressure value, the piston of the hydraulic pressure transducer (HPT) distributes the fluid under pressure to the hydraulic cylinder (HC), which causes a decrease in the tilt angle of the swash plate and a decrease in flow. The CPAP was selected as a component of the hydraulic system of the aircraft for the experimental tests in this paper. The experimental tests covered the structural and working parameters of the pump and analyzed their performance, efficiency and reliability. Experimental tests of structural and operating parameters of the CPAP were carried out in the Laboratory for Hydraulics and Pneumatics "PPT-Namenska" Trstenik on the hydraulic system, which simulated the real conditions prevailing in the hydraulic system of the aircraft. A system was used for data acquisition and recording of pump characteristics, which were obtained during experimental testing. The results of the measurement and testing of the structural parameters of the CPAP are shown in tabular form, and the experimental tests of static characteristics and dynamic behavior are shown diagrammatically.

**Keywords:** axial piston pump; pressure; hydraulic system; aircraft; experimental test



**Citation:** Petrović, R.; Banaszek, A.; Andjelković, M.; Qananah, H.R.; Alnagasa, K.A. Experimental Tests of the Piston Axial Pump with Constant Pressure and Variable Flow. *Designs* **2024**, *8*, 5. https://doi.org/10.3390/designs8010005

Academic Editor: Wenbin Yu

Received: 10 November 2023
Revised: 23 December 2023
Accepted: 27 December 2023
Published: 30 December 2023



## 1. Introduction

The operation of the pump and the reliability of the aircraft's hydraulic system are significantly affected by the following parameters: flow ($Q$), pressure ($p$), number of revolutions of the pump drive shaft ($n$), type of hydraulic oil (viscosity) and oil temperature ($T$) [1,2]. The operating parameters should be adjusted to maximize efficiency while meeting system requirements. The appearance of vibrations and noise during pump operation is particularly noteworthy. It is essential to address any unusual vibrations or noises, as they may indicate mechanical problems that require attention.

The paper specifically considers the design of the hydraulic system of the aircraft so that the entire system is designed efficiently to minimize pressure losses [3]. Safety mechanisms are built into the hydraulic system to ensure reliable operation, even in the event of pump failure. Test and simulation tools were used to model different operating scenarios and optimize pump parameters [4,5]. This can help identify potential problems and improve pump performance under different conditions.

Finally, the paper points out the necessity for the pump and hydraulic system to be designed in compliance with all aviation regulations and standards, as non-compliance can lead to safety and performance issues.

## 2. Mathematical Model

The hydraulic pressure transducer (HPT) registers any change in pressure in the pressure line that causes the displacement of the piston by the size $x$ and is constantly in balance with the spring. The disturbance force $\Delta F$ acts on the piston HPT and is equal to the difference of the force that occurs due to the action of the pressure $p_n$ on the surface of the piston $A_v$ and the force of the spring, which is equal to the product of the stiffness $C_1$ and the preload of the spring $\varepsilon_I$ [6–11].

Balance of static forces:

$$\Delta F = F_v - F_0 = p_n \cdot A_v - C_1 \cdot \varepsilon_1 \tag{1}$$

Balance of dynamic forces:

$$\begin{aligned}
&\Delta p_n \cdot A_v - C_1 \cdot \varepsilon_1 - m_1 \cdot \tfrac{d^2x}{dt^2} - f_{t1} \cdot \tfrac{dx}{dt} - C_1 \cdot x = 0 \\
&\Delta p_n \cdot A_v - C_1(x + \varepsilon_1) - m_1 \cdot \tfrac{d^2x}{dt^2} - f_{t1} \cdot \tfrac{dx}{dt} = 0 \\
&\Delta F = m_1 \cdot \tfrac{d^2x}{dt^2} + f_{t1} \cdot \tfrac{dx}{dt} + C_1 \cdot x
\end{aligned} \tag{2}$$

Spring deflection contains two quantities: the preload $\varepsilon_1$ and the deflection due to piston displacement (HPT).

Using the Laplace transform, we obtain the transfer function of the pressure transducer

$$\Delta F_{(s)} = m_1 \cdot s^2 \cdot x_{(s)} + f_{t1} \cdot s \cdot x_{(s)} + C_1 \cdot x_{(s)} \tag{3}$$

The pressure from the pressure line moves the piston HPT, changes the size of the flow opening, and thus generates flow toward the hydraulic cylinder $Q_c$.

The hydraulic cylinder drives the swash plate with the flow, which comes from the HPT. From the flow balance, taking into account all leakages as well as the part of the flow due to the compressibility of the fluid, it follows:

$$Q_c = Q_v + Q_{lc} + Q_{lk} \tag{4}$$

$Q_c = K_1 \cdot x$; the flow of fluid coming into the cylinder from the HPT;
$Q_v = A_c \cdot \frac{dy}{dt}$; the flow necessary to drive the piston of a hydraulic cylinder;
$Q_{lc} = K_2 \cdot p_c$; the flow that determines the compressibility of the fluid;
$Q_{lk} = \frac{V_c \cdot dp_c}{B \cdot dt} = K_3 \cdot \frac{dp_c}{dt}$ flow due to leakage past the piston.

From the balance of forces on the swash plate, it follows:

$$\sum F = 0 \Rightarrow F_c - F_{a2} - F_{t2} - F_v - F_{o2} = 0 \tag{5}$$

$F_c$ $(A_c \cdot p_c)$, force due to pressure $p_c$;
$F_{a2}$ $(-m_2 \frac{d^2y}{dt^2})$, inertial force;
$F_{t2}$ $(-f_{t2} \cdot \frac{dy}{dt})$, force due to viscous friction;
$F_v$, reaction on the swash plate due to the action of the pistons $F_v \approx 0$;
$F_{o2}$ $(-C_2(y + \varepsilon_2))$, spring force (the stiffness of the spring $C_2$, the previous deflection $\varepsilon_2$ and the deflection caused by the displacement of the piston by the size $y$) [12–14].

$$A_c \cdot p_c - m_2 \frac{d^2y}{dt^2} - f_{t2} \cdot \frac{dy}{dt} - C_2(y + \varepsilon_2) = 0 \tag{6}$$

The spring force is equal to the product of the stiffness $C_2$, and the deflection consisting of the two lengths of the previous deflection $\varepsilon_2$ and the displacement of the piston $y$.

A change of the tilt angle of the swash plate as a function of the stroke of the piston HC for the size $y$ is:

$$\gamma = \gamma_{max} - K_4 \cdot y \tag{7}$$

$\gamma_{max}$ is the maximum angle of the swash plate, $\gamma_{max}$ = 19° = 0.3333 rad, $K_4$ is the coefficient of change of the angle of the swash plate depending on the stroke of the piston, $y$.

$$Q_p = Q_{tp} + Q_{cp} + Q_{kp} + Q_c \tag{8}$$

CPAP hydraulic systems must provide a flow that will satisfy the following requirements: actuator needs, needs conditioned by external and internal leaks, needs conditioned by the compressibility of the hydraulic fluid and the flow going to the pressure transducer [15].

$$Q_p = Q_{tp} + K_6 \cdot p_n + \frac{V_{tp} \cdot dp_n}{B \cdot dt} + K_1 \cdot x \tag{9}$$

After the Laplace transform

$$Q_p = Q_{tp} + K_6 \cdot p_n + K_7 \cdot s \cdot p_n + K_1 \cdot x; \tag{10}$$

where:

$Q_p$—required pump flow, $m^3/s$;
$Q_{tp}$—flow directed towards actuator, $m^3/s$;
$Q_{kp}$—losses, $m^3/s$;
$Q_c$—flow through HPT, $m^3/s$;
$K_6$—internal and external leakage coefficient;
$V_{tp}$—total volume of hydraulic fluid under pressure;
$K_7$—compressibility coefficient of the total volume of the pump.

The $K_6$ loss coefficient includes losses inside the pump, between the distribution plate and the cylinder block, between the pistons and the cylinder block and leakage through the pedal. This coefficient includes the external losses that occur on the components closest to the pump. In this case, linearization of non-linear characteristics, which significantly affect the damping of the system, was performed [16].

In addition to the mentioned linearization, other linearizations related to the flow amplification coefficient of the hydraulic pressure transducer and the flow of the constant pressure pump are also possible. Linearization is also possible for HPT and HC viscous friction as well as for HC flow loss [17–19].

## 3. Measured and Adopted Structural Parameters of CPAP

Piston axial pumps with constant pressure (Figure 1) belong to the group of pumps with automatic regulation of the flow depending on the pressure, which changes within the limits set in advance [20].

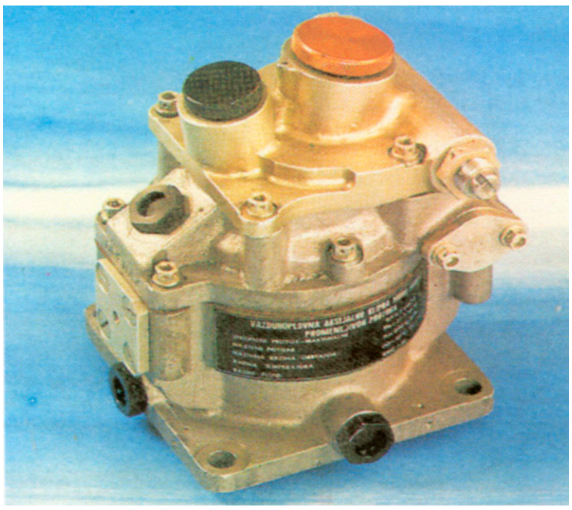

**Figure 1.** Piston axial constant pressure pump.

The principle of regulating the operation of the pump is given in Figure 2. The flow of fluid through the regulating distributor (pos. 5) depends on the pressure that opens the distributor [21]. The fluid flow is given by the input characteristic through the choke (pos. 4), which depends on the opening and fluid pressure in the cylinder (pos. 2).

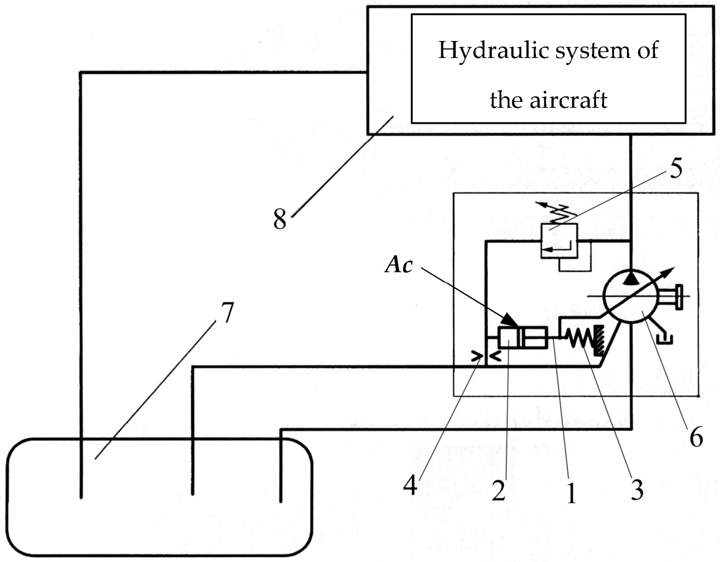


**Figure 2.** Functional diagram of a constant pressure pump. 1—piston rod; 2—cylinder; 3—spring; 4—muffler; 5—distributor for regulation; 6—hydraulic pump; 7—pressure tank; 8—actuators of working fluid.

On the one hand, a force acts on the swash plate, the intensity of which depends on the pressure and the area of the piston, Ac. On the other hand, the spring force acts on the swash plate, the intensity of which depends on the stiffness of the spring and the length by which the spring is compressed. The piston rod (pos. 1) changes the angle of inclination of the swash plate and thus regulates the output flow.

The hydraulic system in an aircraft plays a crucial role in various operations, including controlling flight surfaces, landing gear, brakes and other critical functions. The design of an aircraft's hydraulic system is carefully engineered to ensure reliability, safety and optimal performance. Operational and structural parameters of the aircraft hydraulic system powered by a piston axial pump with constant pressure and variable flow are shown in Table 1.

Hydraulic pumps are responsible for generating the necessary pressure to move the hydraulic fluid through the system. Engine-driven pumps, electric pumps or air-driven pumps are commonly used for redundancy. Hydraulic pumps are responsible for generating the necessary pressure to move the hydraulic fluid through the system. Engine-driven pumps, electric pumps or air-driven pumps are commonly used for redundancy.

The hydraulic reservoir stores hydraulic fluid and allows for thermal expansion and contraction. It typically includes a fluid level indicator, breather and filter to maintain the cleanliness of the fluid. Filters are used to remove contaminants from the hydraulic fluid, preventing damage to components. Regular maintenance and replacement of filters are essential for system reliability. Accumulators store hydraulic energy and can provide emergency power in case of pump failure. They also help to smooth out pressure fluctuations in the system. Check valves permit fluid flow in one direction only, preventing backflow and maintaining system pressure. They are crucial for preventing unintended movements of hydraulic actuators. Pressure relief valves prevent system overpressure by diverting excess fluid back to the reservoir. They ensure that components and lines are not subjected to pressure beyond their design limits.

Aircraft hydraulic systems include sensors and indicators to monitor fluid levels, pressure and system health. Automated control systems may adjust hydraulic pressures based on flight conditions. Critical components often have redundancy to enhance safety. Emergency procedures are in place to handle hydraulic system failures.

The hydraulic system is integrated into the overall avionics and control systems of the aircraft to ensure seamless operation.

**Table 1.** Operational and structural parameters of the aircraft hydraulic system powered by a piston axial pump with constant pressure and variable flow.

| | Title | Value |
|---|---|---|
| $\varepsilon_1$ | HPT spring preload | $\varepsilon_1 = 1.4 \times 10^{-3}$ m |
| $c_1$ | HPT spring stiffness | $c_1 = 5.5 \times 10^4$ N/m |
| $m_1$ | piston mass HPT | $m_1 = 3 \times 10^{-3}$ kg |
| $f_{t1}$ | piston viscous friction force HPT | $f_{t1} = 70$ Ns/m |
| $K_1$ | HPT flow amplification coefficient | $K_1 = 0.170$ m$^2$/s |
| $A_v$ | piston surface HPT | $A_v = 8 \times 10^{-6}$ m$^2$ |
| $K_2$ | flow loss coefficient at HC | $K_2 = 2 \times 10^{-12}$ m$^5$/Ns |
| $K_3$ | flow coefficient due to compressibility | $K_3 = 4 \times 10^{15}$ N/m$^5$ |
| $A_c$ | piston surface HC | $A_c = 1.26 \times 10^{-4}$ m$^2$ |
| $\varepsilon_2$ | spring bias HC | $\varepsilon_2 = 9 \times 10^{-3}$ m |
| $c_2$ | spring stiffness HC | $c_2 = 2 \times 10^4$ N/m |
| $f_{t2}$ | force of viscous friction of the piston HC | $f_{t2} = 100$ Ns/m |
| $B$ | compressibility modulus of the fluid | $B = 1.4 \times 10^{15}$ N/m$^2$ |
| $m_2$ | piston mass HC | $m_2 = 8.52 \times 10^{-1}$ kg |
| $K_4$ | the coefficient of change of the angle of the swash plate | $K_4 = 27.5$ rad/m |
| $K_5$ | HC flow gain coefficient | $K_5 = 1.14 \times 10^{-3}$ m$^3$/s rad |
| $K_6$ | coefficient of total leakage of the pump | $K_6 = 4 \times 10^{-13}$ m$^3$/Ns |
| $K_7$ | compressibility coefficient of the entire container | $K_7 = 9.33 \times 10^{14}$ m$^5$/N |
| $n$ | pump shaft rotation frequency [9] | $n = 1500$ r/min |
| $q$ | specific pump flow | $q = 15 \times 10^{-6}$ m$^3$/r |
| $\gamma_{max}$ | maximum angle of the swash plate | $\gamma_{max} = 19^\circ = 0.33$ rad |
| $Q_n$ | nominal flow rate of the pump | $Q_n = 3.7 \times 10^{-4}$ m$^3$/s |
| $p_n$ | nominal pressure of the pump | $p_n = 200 \times 10^5$ Pa |
| $p_{max}$ | maximum pump pressure | $p_{max} = 211 \times 10^5$ Pa |
| $\Delta\gamma$ | swash plate sensitivity threshold | $\Delta\gamma = 1 \times 10^{-3}$ rad |

## 4. Experimental Tests of Static Characteristics and Dynamic Behavior of CPAP

Experimental tests of static characteristics and dynamic behavior of pumps are essential for understanding their performance, efficiency and reliability in various operating conditions. These tests involve measuring and analyzing the pump's behavior under different parameters.

The pressure generated by the pump at different flow rates under stationary conditions was measured. The efficiency of the pump was determined by measuring the input and output power at different operating points. The pump discharge pressure at different flow rates was measured. Sudden changes in flow rate and pressure were simulated to observe pump response. The time required for the pump to reach a steady state after a disturbance was measured. The pump was subjected to input signals of different frequencies and amplitudes. How the pump responded to different frequencies was analyzed to identify resonant frequencies and potential instabilities.

The sudden closing and opening of the valve were simulated to evaluate the response of the pump to transient pressure changes. Pressure peaks were measured, and the pump's ability to cope with the effects of hydraulic shocks was observed. We reduced the pressure at the pump suction to cause cavitation and used accelerometers and vibration sensors to analyze vibrations in the pump. Vibration frequencies and amplitudes were analyzed to identify potential mechanical problems. The temperature of the critical components of the pump during continuous operation was observed. An assessment was made of the pump's ability to dissipate heat and avoid an excessive temperature rise.

After a longer period of operation, an inspection of the pump components and an assessment of their wear and tear during operation was carried out. An assessment of the durability and performance of the pump over time was performed. Pump efficiency was tested at different operating speeds to understand the speed-dependent performance.

Pressure transducers, flow meters, temperature sensors and vibration sensors were used for accurate data collection. A robust data acquisition system was used to capture and analyze data in real time. Safety protocols were implemented to prevent accidents during testing, especially in high-pressure environments.

Test equipment and instruments are certified for accurate and reliable results. By systematically conducting static and dynamic tests, engineers can gain a comprehensive understanding of a pump's performance characteristics, identify potential problems and optimize its design for efficiency and reliability.

For experimental research on the static and dynamic characteristics of the CPAP, a hydraulic system was installed in the Laboratory for Hydraulics and Pneumatics "PPT-Namenska" Trstenik, which is shown in Figure 3. The experimental hydraulic system is similar to the real hydraulic system of the "Orao" aircraft [22–24]. During the experiment, the pump is driven by an electric motor. In the aircraft hydraulic system, the pump is driven by a gear reducer shaft from an airplane jet engine.

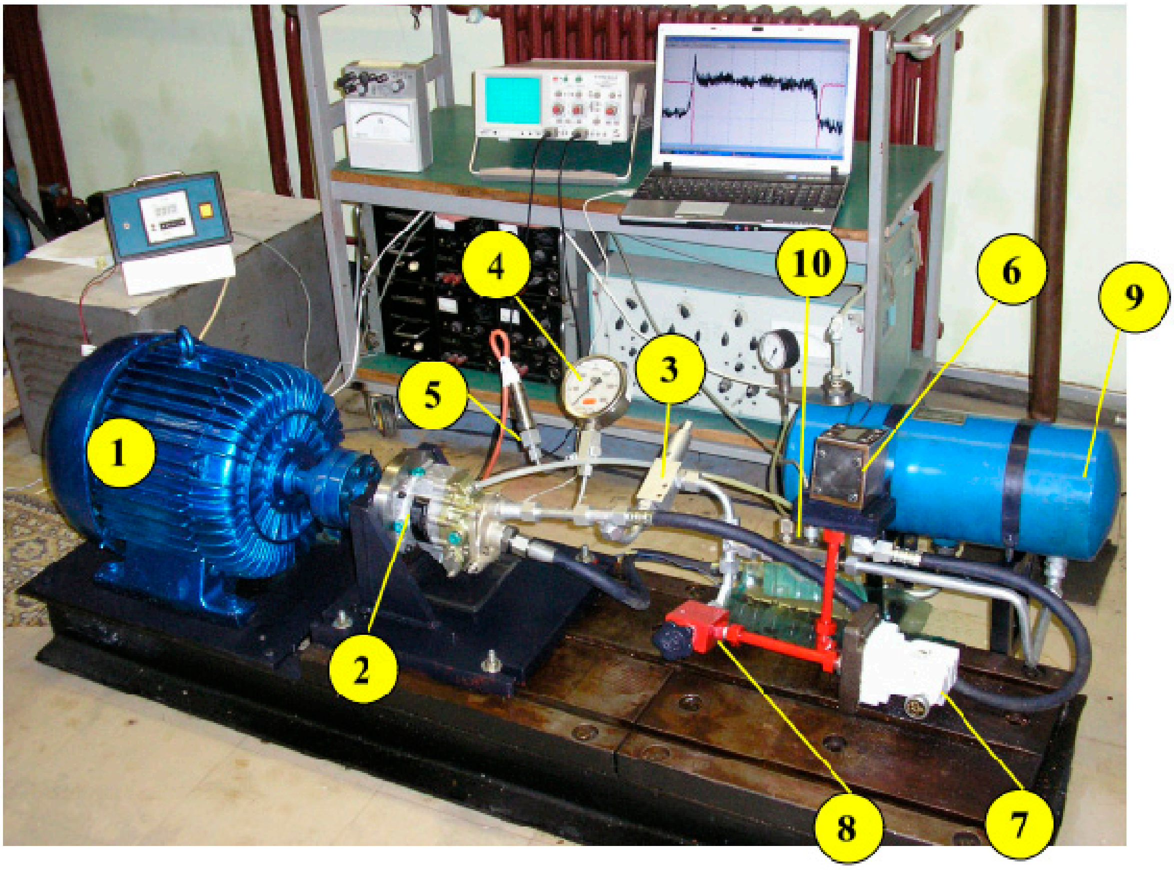


**Figure 3.** The laboratory hydraulic system for testing CPAP. 1—electric motor; 2—hydraulic pump; 3—safety valve; 4—manometer; 5—pressure transducer; 6—electromagnetic distributor; 7—pressure regulator; 8—muffler; 9—flow converter; 10—pressure tank.

In addition, the laboratory hydraulic system contains far fewer components than the hydraulic system of the aforementioned aircraft. The pump works in a pressurized system with volume regulation in laboratory conditions. Energy losses are small, and the working fluid heats a little [25,26].

A pneumatic aggregate is connected to the hydraulic system, which has the task of maintaining a certain overpressure in the hydraulic reservoir. The pressure in the tank is constantly maintained within the limits of 0.2 to 0.3 MPa, which is enough for the high-quality operation of the pump. In these conditions, cavitation does not occur even with the fastest flow change processes.

In the tank of the experimental hydraulic system, there is about $7 \times 10^{-3}$ m$^3$ of working fluid, which participates in the transformation and transmission of power, which is approximately 7.4 kW. The experimental hydraulic system contains mineral-based hydraulic fluid, Hidraol 15, with a kinematic viscosity of $\nu = 15 \times 10^{-6}$ m$^2$/s at a temperature of 40 °C. The applied Hidraol 15 is similar in characteristics to the hydraulic working fluid used in aircraft systems. The pump under test is designed to work with "Aero Shell 40" oil, which is used for the temperature range from −55 °C to 135 °C.

In times when the system does not need hydrostatic energy, the pump reduces the flow to approx. $2 \cdot 10^{-5}$ m$^3$/s. Then, it absorbs far less power than when excess fluid is returned to the tank after pressure reduction. In the hydraulic system (Figure 4), there is an electromagnetic distributor (pos. 6) with which the flow of the pump can be translated very quickly from the maximum to the minimum value. There is a safety valve in the pressure line (pos. 3), which, if necessary, relieves the system to the set level. The indirect action pressure regulator (pos. 7) enables precise pressure regulation in the pressure line, while small pressure variations are regulated by means of a variable resistance damper (pos. 8).

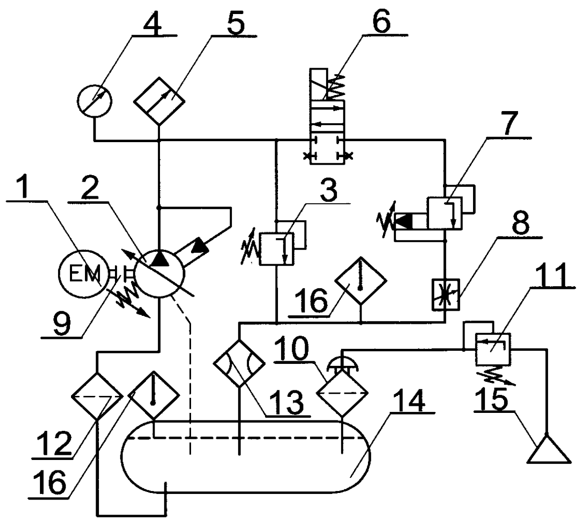


**Figure 4.** Scheme of the laboratory hydraulic system for testing CPAP. 1—electric motor; 2—pump CPAP; 3—safety valve; 4—manometer; 5—pressure transducer; 6—electromagnetic distributor; 7—pressure regulator of indirect action; 8—damper of variable resistance; 9—elastic coupling; 10—intake filter; 11—air pressure regulator; 12—intake filter; 13—flow converter; 14—pressure tank; 15—pressure air source; 16—temperature converter.

Four parameters were included in the experimental tests:

- Pressure in the discharge line;
- Pump flow;
- Temperature of the working fluid in the tank;
- Temperature at the outlet of the variable resistance choke.

### 4.1. Changes in Pump Flow When the Pressure Changes from the Set Point to the Maximum Set Value

In the hydraulic system of the aircraft, the pressures change from the minimum value $p_{min}$ is dictated by the actuators with their resistances, through the nominal value, $p_n$, to the maximum value $p_{max}$. Then, the flow changes from the nominal value, $Q_n$, to the minimum value, $Q_{min}$.

The experiment was carried out in such a way that the working pressure was set to $p_r$ = 3 MPa, and then by instantly closing the electromagnetic distributor (pos. 6) (Figure 4), which was in the flow position, the pressure increased to the level of the set pressure interval of $p_n$ = 20 MPa up to $p_{max}$ = 21 MPa. After a certain time, the electromagnetic distributor was opened again, and thus, the characteristic of the pump for the given operating mode was obtained. The recorded characteristic of the pump shows that within the set pressure interval, from $p_n$ = 20 MPa to $p_{max}$ = 21 MPa, the flow varies within the limits of $Q_n$ to $Q_{min}$ (Figure 5). In the diagram, two transition modes can be observed at the jump signal of closing and opening of the pressure line.

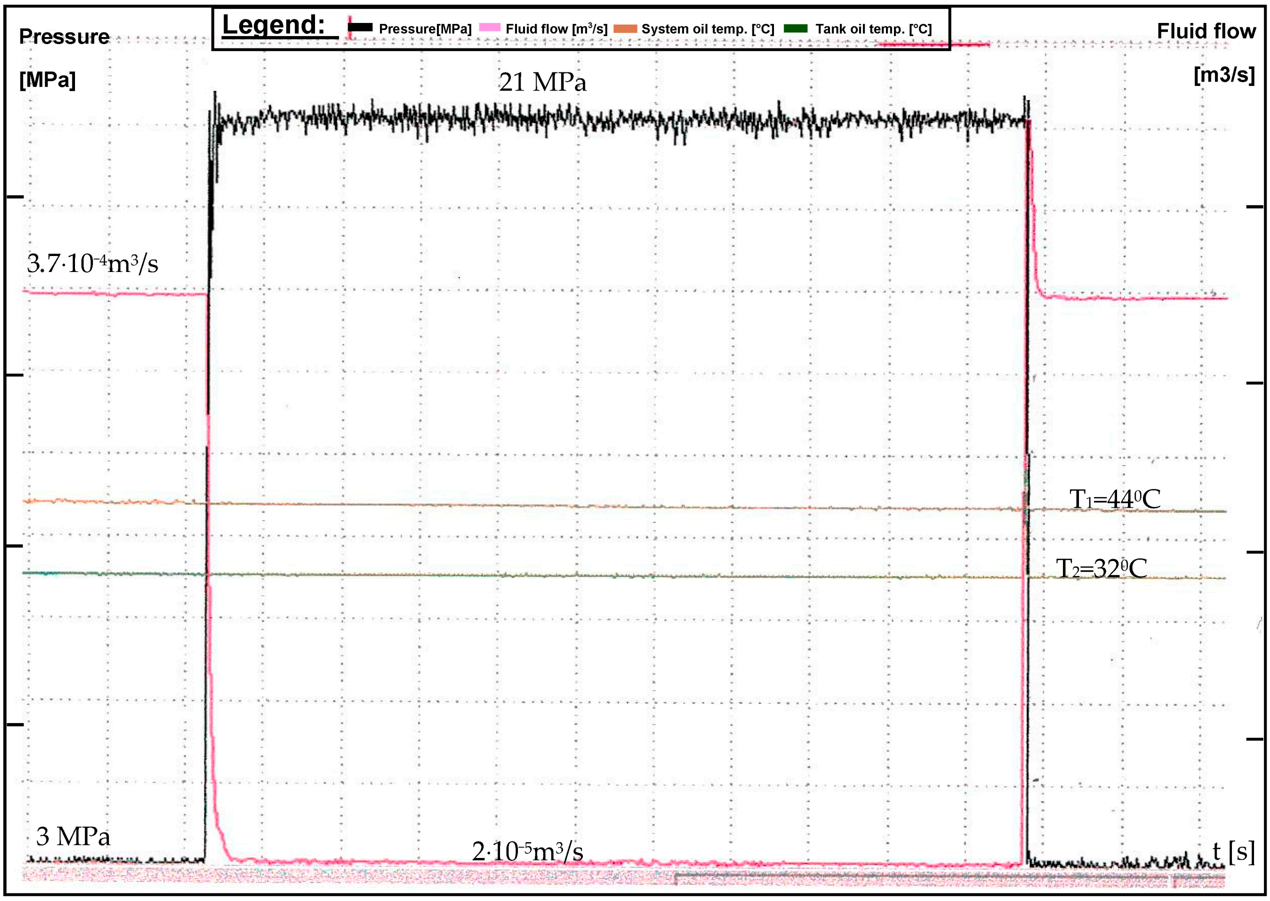


**Figure 5.** Diagram of the change in flow when the pressure changes from $p_r$ = 3 MPa to $p_{max}$ = 21 MPa (pressure range from 3 to 23 MPa, the cut-off value is 2 MPa).

When using the pressure regulator (pos. 7), the working pressure is set to the value $p_r$ = 16 MPa. By activating the electromagnetic distributor (pos. 6), the pressure increases to $p_{max}$ = 21 MPa, and the pump flow changes within the limits of $Q_n$ to $Q_{min}$ (Figure 6).

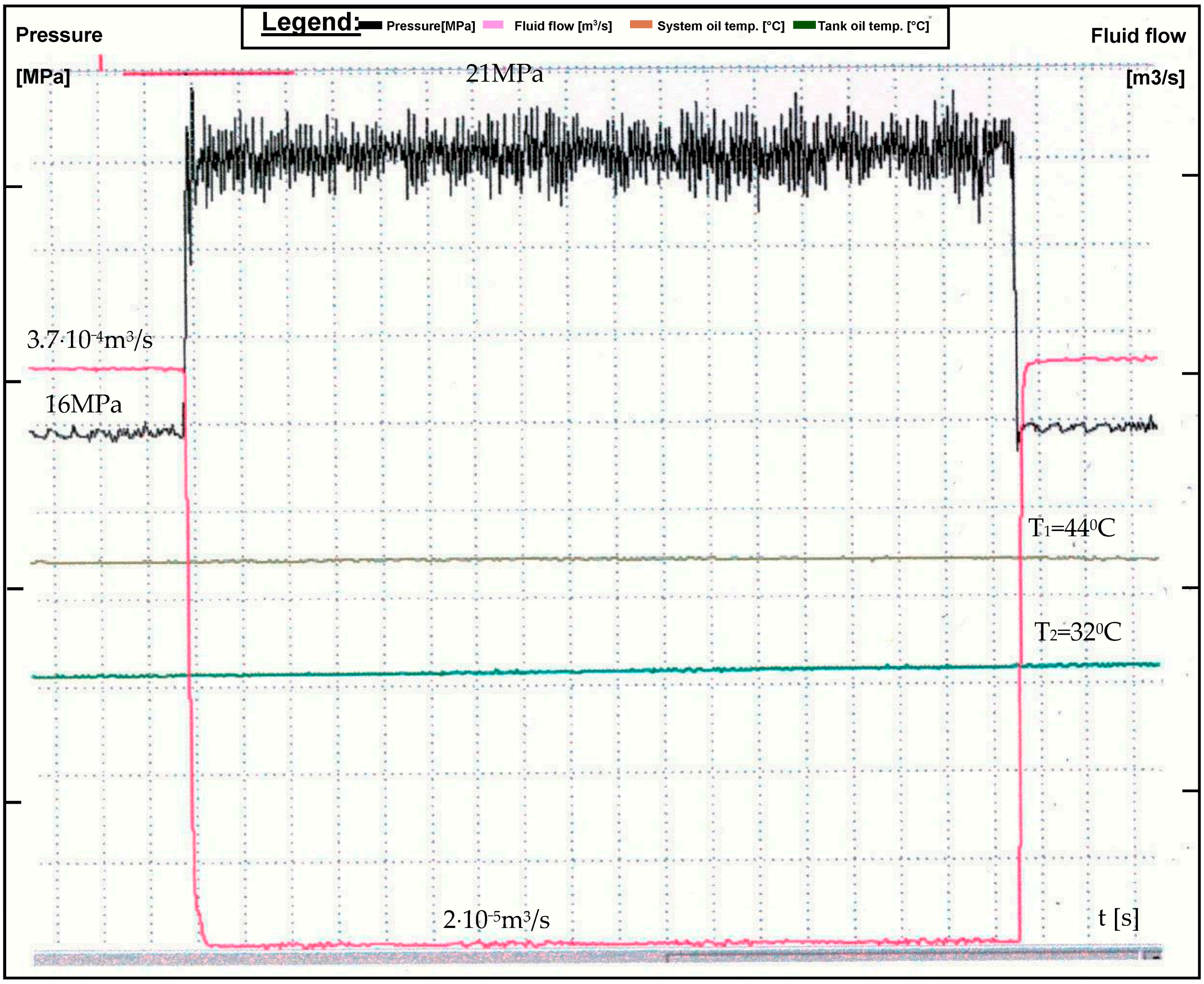


**Figure 6.** Diagram of the flow rate change when the pressure changes from $p_r$ = 16 MPa to $p_{max}$ = 21 MPa (pressure range 6 to 23 MPa; the value of the partition is 1.7 MPa).

It can be seen that changes in flow and pressure are strictly dependent and that a change in one quantity causes a precise and rapid change in the other quantity, regardless of the starting pressure level.

### *4.2. Changes in Pump Flow when Changing Two Maximum Pressure Values*

In order to check the stability of the pump operation with an increased maximum pressure, a new characteristic of the pump was recorded. By adjusting the HPT, the maximum pressure was increased to the value $p_{max}$ = 23 MPa. The change in maximum pressure was performed by changing the preload of the HPT spring, which caused an increase in the maximum pressure, and the nominal pressure was set to $p_n$ = 22 MPa. All other parameters are as in Section 3, and the recorded characteristic is shown in Figure 7. The diagram also shows the usual operating mode when the pump works with nominal pressure $p_n$ = 20 MPa and maximum pressure $p_{max}$ = 21 MPa. It is obvious that the pump works stably even with a different maximum pressure, without a large pulsation of pressure and flow, excluding certain noises in the electronic system that occur when recording the diagram.

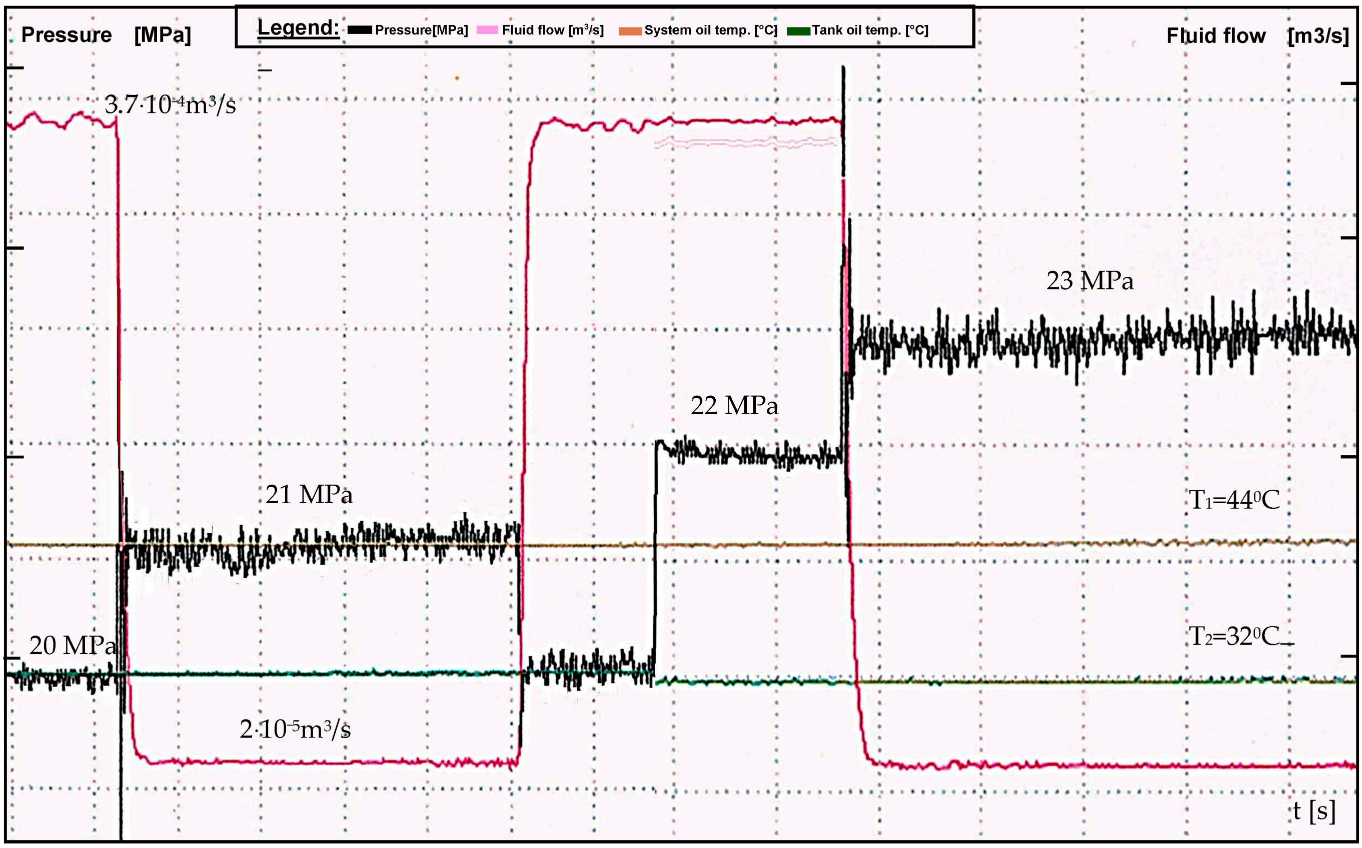


**Figure 7.** Diagram of the change in flow when the pressure changes in two levels, from 20 MPa to 21 MPa and from 22 MPa to 23 MPa (the pressure range is from 13 to 23 MPa; the value of the division on the diagram is 1 MPa).

## 5. Conclusions

The paper analyzed the performance of the pump with theoretical expectations and industrial standards. The efficiency of the pump in different operating conditions was analyzed. An assessment of the performance of the axial piston pump in terms of constant pressure and variable flow was performed. The reliability of the pump was evaluated based on the experimental results. An analysis of the wear of the parts observed during the test period was performed. Energy consumption was analyzed in all potential efficiency improvements. Areas for further research and development have been identified.

The methods of testing and presenting the basic performance in the stationary state according to the international standard ISO 4409:2019 were used [27]. The specified methods for measuring the performance characteristics of hydraulic components are covered by the ISO 10767-1:2015 standard [28].

The paper carried out a detailed analysis of the hydraulic system requirements, including pressure and flow requirements at different points.

The specific needs of the aircraft hydraulic system are essential for proper pump selection and optimization. The choice of the test stand and the appropriate pump size was made based on the pressure and flow of the system. Factors such as the number of actuators, the size of the hydraulic lines and the overall layout of the hydraulic system were taken into account. For the experiment, a reciprocating axial pump with the possibility of variable displacement was selected. This allows the pump to adjust its power according to the different demands of the hydraulic system, improving efficiency and reducing energy consumption. Pressure control valves are built into the hydraulic system to regulate and maintain constant pressure. These valves ensure system pressure stabilization and prevent excessive fluctuation. Flow control devices are incorporated to manage and distribute the flow within the hydraulic system. This ensures that each component receives the necessary flow rate while maintaining the required constant pressure.

The dynamic response of the pump to rapid changes in load or demand is taken into account. A pump with a fast response time allows the hydraulic system to quickly adapt to different conditions, improving the overall performance of the system. Measures have been implemented to control and manage the temperature within the hydraulic system. The influence of heat on the viscosity of the hydraulic fluid and, consequently, on the efficiency of the pump was analyzed.

The constant pressure piston axial pump belongs to the group of complex components of the hydraulic system of the aircraft, in the internal structure of which numerous and complex interactions take place. The aforementioned circumstances impose the need and justify the obligation to constantly improve the characteristics in order to increase the performance of the complete hydraulic system on the plane. It was found that hydro systems on aircraft are very complex, with a large number of very strict requirements that are placed on hydro pumps, which are generators of hydrostatic energy in the systems. Analyses have shown that piston axial pumps with constant pressure and variable volume, which work in two modes, are the most suitable. Thus, depending on the pressure feedback, the pump operates in constant flow or constant pressure mode. It was also concluded that hydraulic pumps with a folding plate have an advantage in application because they have a small inertial mass of the parts that participate in the process of switching the pump from one mode of operation to another and vice versa. Due to these facts, vane pumps have a lower time constant, which is a very important characteristic of aircraft hydraulic systems.

**Author Contributions:** Conceptualization, R.P. and A.B.; methodology, R.P. and A.B.; software, A.B. and R.P.; validation, A.B. and R.P.; formal analysis, A.B. and R.P.; investigation, A.B. and R.P.; resources, A.B. and R.P.; data curation, A.B. and R.P.; writing—original draft preparation, R.P., A.B., M.A., H.R.Q. and K.A.A.; writing—review and editing, R.P., A.B., M.A., H.R.Q. and K.A.A.; visualization, A.B.; supervision, R.P.; funding acquisition, R.P. All authors have read and agreed to the published version of the manuscript.

**Funding:** This paper is funded by the University "Union-NikolaTesla", Faculty of Information Technology and Engineering.

**Institutional Review Board Statement:** Not applicable.

**Informed Consent Statement:** Not applicable.

**Data Availability Statement:** Data are contained within the article.

**Acknowledgments:** The authors would like to thank the Editor-in-Chief, Editor and anonymous reviewers for their valuable reviews.

**Conflicts of Interest:** The authors declare no conflict of interest.

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