




Experimental Research of an Axial Piston Pump with Displaced Swash Plate Axis of Rotation

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Abstract. The article describes the influence of displacement of the swash plate rotation axis on the efficiency of axial piston pumps. The dead space volume was defined as the volume of the working chamber in the extreme position of the piston at the end of the pumping phase and its variability was determined as a function of the position of the axis of rotation of the swash plate and the swing angle of swash plate. The influence of swash plate rotation axis displacement on leaks between piston and cylinder was determined. It has been proven that displacement of the swash plate rotation axis reduces the dead space volume and leaks, thus improving the volumetric efficiency visible at small swash plate swing angles. The results of a comparative study of the volumetric and total efficiency of a modified design of a pump that allows the rotation axis to be repositioned are presented. The efficiency curves for the swash plate rotation axis crossing the shaft rotation axis and for the displaced swash plate axis of rotation are compared. The article also presents the author's construction of an axial piston pump with displaced swash plate rotation axis with a follow-up mechanism of capacity change controlled by a stepper motor. The diagram and view of the test stand are shown, as well as the results of tests of volumetric and hydromechanical pump efficiency.

Keywords: Hydraulics · Piston pump · Efficiency

1 Introduction

Piston pumps with swivel swash plate are often used in modern hydraulic systems. The design of such pump is schematically shown in Fig. 1. Shaft (1) is connected to drum (2) by means of a spline. In the drum there are pistons (4) in cylindrical chambers, finished with slippers (5). Pistons, during the rotation of the drum, due to the contact of slippers with the swash plate (6), perform the reciprocating movement in the cylindrical chambers. The face of the rotating drum contacts a fixed timing plate (3) with two kidney holes. One of these openings is connected to the suction port and the other to the pump delivery port. Thus, the cylinder chambers from which the pistons slide out are connected to the suction channel by means of a timing plate and the liquid is sucked in. As the shaft continues to rotate and the pistons pass through the dead-end, they begin to approach the timing plate - then the chambers are connected to the delivery channel and liquids

are pressed out of the chambers [1, 3]. Figure 1 also indicates the volume of dead space, i.e. the volume occupied by the fluid in the cut-off cylinder chamber when the piston is in its extreme position at the end of the pressing phase [4, 5].

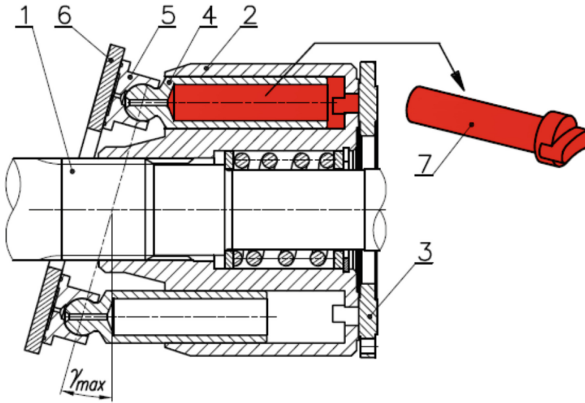


Fig. 1. Working mechanism of axial piston pump [8]: 1 - shaft; 2 - cylinder drum; 3 - timing plate; 4 - piston; 5 - slipper; 6 - swash plate; 7 - dead space volume; γ_{max} - max swing angle.

2 Materials and Methods

2.1 Displacement of Swash Plate Rotation Axis

The standard axial piston pumps have one thing in common: the swash plate axis of rotation crosses the shaft axis of rotation. The volume of the dead space increases as the swing angle of swash plate decreases (see Fig. 2a). If the axis of rotation of the swash plate is displaced as shown in Fig. 2b, the volume of this space will be constant regardless of the swing angle of swash plate. The displacement of the pump V_s is a function of swash plate swing angle γ , piston spacing D and piston diameter d_t [1–3]:

$$V_s = \frac{\pi d_t^2}{4} \cdot D \cdot \tan \gamma \tag{1}$$

If the axis of rotation of swash plate intersects the axis of rotation of the shaft, as in Fig. 2a, then for the angle γ_{max} the volume of dead space is minimal and increases with the reduction of the swing angle of swash plate γ according to the following relation [8]:

$$V_D = V_{Dmin} + \frac{\pi d_t^2}{8} \cdot D(\tan \gamma_{max} - \tan \gamma) \tag{2}$$

V_{Dmin} is the minimum dead space volume for maximum swing angle of swash plate γ_{max} . This space includes a part of the cylinder, the window of the cylinder drum and the empty space inside the piston.

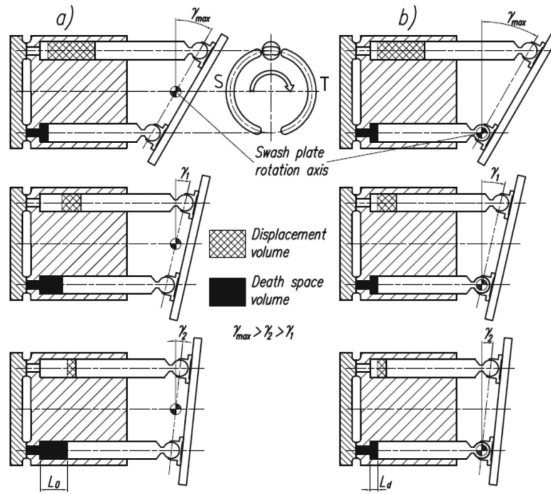


Fig. 2. Comparison of changes in the volume of dead space as a function of the swing angle of swash plate for a pump with the axis of rotation of the swash plate intersecting with the axis of rotation of the shaft a), and a pump with shifted axis of rotation of the swash plate b) [7].

2.2 Influence of Displacement of Swash Plate Rotation Axis on the Dead Space Volume

Figure 3 shows the situation when the axis of rotation of the swash plate is shifted in relation to its original position by a_x and a_y . The boundary condition is that the swash plate is positioned at its maximum swing angle to obtain a minimum volume of dead space V_{Dmin} . Thereby, regardless of the position of the axis of rotation of the swash plate, its position at the maximum swing angle will be the same. Consider the situation of the rotation of the swash plate relative to the displaced center of rotation from the angle γ_{max} to the angle γ in a system of coordinates with the center in the place where the center of rotation of the swash plate was originally located.

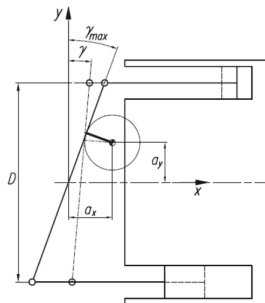


Fig. 3. Shift of the rotation axis of the swash plate by a_x and a_y in relation to the axis of rotation intersecting the axis of the shaft [8].

The volume of dead space as a function of the position of the center of rotation and the swing angle of the swash plate may be described as [8]:

$$V_D = V_{Dmin} + \frac{\pi d_t^2}{4} \left[\frac{D}{2} \tan \gamma_{max} - \left(\frac{D}{2} - C \right) \tan \gamma \right] \quad (3)$$

The coefficient C depending on a_x and a_y will be [8]:

$$\text{if } a_x \geq a_y \tan \gamma_{max} \quad C = \frac{\left| \frac{a_x}{\tan \gamma_{max}} - a_y \right| \sqrt{\frac{1}{\tan^2 \gamma} + 1}}{\sqrt{\frac{1}{\tan^2 \gamma_{max}} + 1}} - \frac{a_x}{\tan \gamma} + a_y \quad (4)$$

$$\text{if } a_x \leq a_y \tan \gamma_{max} \quad C = -\frac{\left| \frac{a_x}{\tan \gamma_{max}} - a_y \right| \sqrt{\frac{1}{\tan^2 \gamma} + 1}}{\sqrt{\frac{1}{\tan^2 \gamma_{max}} + 1}} - \frac{a_x}{\tan \gamma} + a_y \quad (5)$$

Example results of simulation of a change in the dead space volume as a function of the position of the center of rotation of the swash plate is presented in Fig. 4. The greatest influence on changes in the dead space volume has the movement of the axis of rotation in the direction a_y .

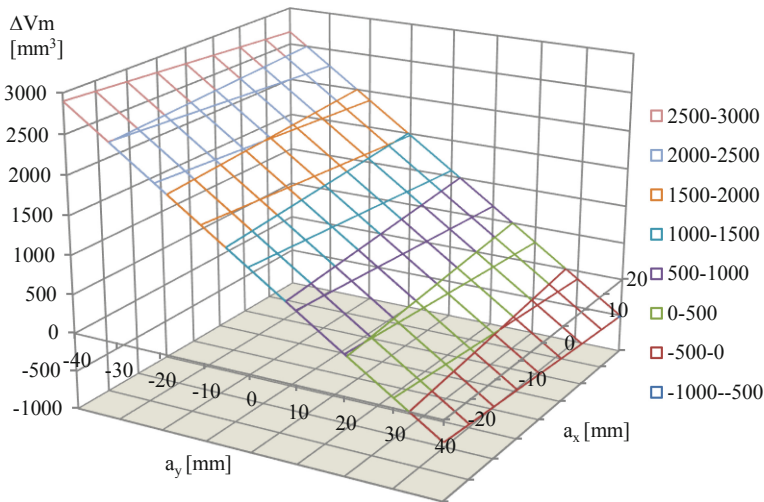


Fig. 4. Change of the dead space volume as a function of the position of the swash plate rotation axis for the pump with parameters: $D = 58$ mm; $d_t = 14,15$ mm; $\gamma = 1^\circ$; $\gamma_{max} = 15,65^\circ$; $V_{Dmin} = 4,27$ cm³; ΔVm - change of the dead space volume [8].

2.3 Influence of the Dead Space Volume on the Volumetric Flow Losses

At the end of pumping phase, the residual oil under high pressure is located in the dead space. Thus, when the suction manifold opens, the medium cannot be sucked at the

beginning of cycle, because the oil contained in the dead space expands, causing back-flow to the suction duct. Therefore, the volume of actually sucked oil is reduced, thus decreasing the volumetric efficiency. The higher delivery pressure and the larger dead space, there is the higher influence on the phenomenon mentioned above.

The change in the volume of the expanded oil at a pressure drop Δp will be [5]:

$$\Delta V_c = V_D \cdot \frac{\Delta p}{K_I - \Delta p} \quad (6)$$

Since the phenomenon of liquid expansion in the pump is adiabatic, the secant isentropic bulk modulus K_I should be used to describe it. The resulting flow rate losses for a pump driven at speed n with the number of pistons z can be determined as:

$$\Delta Q_s = \Delta V_c \cdot n \cdot z \quad (7)$$

These losses directly affect the reduction of the flow generated by the pump and thus the volumetric efficiency.

2.4 Influence of Displacement of Swash Plate Rotation Axis on the Length of Piston - Cylinder Gap

Figure 2 shows that the displacement of the swash plate rotation axis is carried out under the condition that the position of the swash plate for the maximum angle is the same as for a pump with a non-translocated rotation axis, causing the pistons to slide more into the cylinder drum, hence for the disc rotation angle γ_2 the distance of the piston face from the bottom of the cylinder chamber changes from L_0 to L_d . The difference is:

$$s_d = L_0 - L_d = -C \cdot \tan \gamma \quad (8)$$

The length of the piston-cylinder gap depends on the angle of rotation of the shaft [6], so greater insertion of pistons into the cylinder drum will result in an increase in the average length of the piston-cylinder gap and will reduce the leakage occurring in this node.

2.5 Influence of Displacement of Swash Plate Rotation Axis on the Volumetric Efficiency

Comparative tests were carried out in the Laboratory of Hydraulics of the Gdansk University of Technology on a pump with the possibility of moving the swash plate axis of rotation. The basic parameters of the pump were as follows:

- piston spacing $D = 58$ mm
- piston diameter $d_t = 14,15$ mm
- maximum swing angle of swash plate $\gamma_{max} = 15,65^\circ$
- minimal volume of dead space $V_{Dmin} = 4,27$ cm³

The results of the tests shown in Fig. 5 showed a noticeable increase in volumetric efficiency for small swing angles of swash plate. These tests confirmed the influence of the position of the swash plate axis of rotation on the volumetric efficiency and were an impulse to design a variable displacement pump with a displaced axis of rotation.

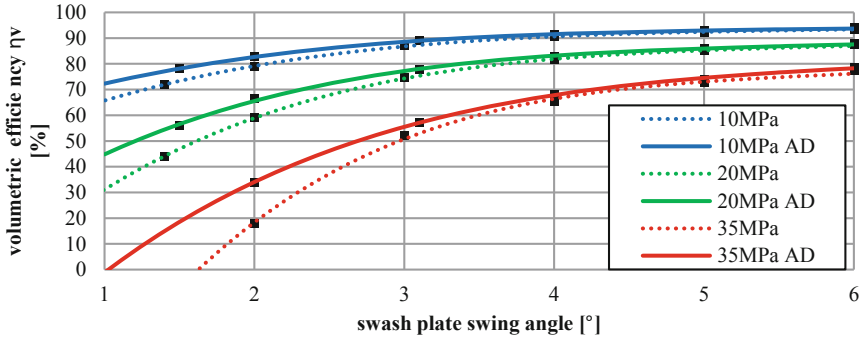


Fig. 5. Volumetric efficiency as a function of the swash plate swing angle for the pump with axis of rotation intersecting the axis of rotation of the shaft and for a pump with the axis displaced by $a_x = a_y \tan \gamma_{max}$ and $a_y = D/2$ (AD); $\nu = 40\text{cSt}$, $n = 1500$ rpm [8].

3 Results

3.1 Prototype Pump Construction

The pump shown in Fig. 6 is a prototype construction where the axis of rotation of the swash plate is displaced so that the volume of the dead space remains constant when the swash plate swing angle is changed. The swash plate (2) is fixed in the body by two bolts (3), which form its axis of rotation. The capacity is changed by using a piston (7) ended with a roller. The torque that loads the swash plate has the same turn every time, so there is no need for additional spring support of the plate. The change of the pump capacity is done by moving the control spool (10) by means of the stepper motor (11). This spool controls the flow of liquid through the moving cylinder (12) to the chambers of the actuator (7). Feedback is realized mechanically. Pusher (8) through lever (9) acts on cylinder (12) by moving it in the direction of the control slider movement. The movement of the cylinder continues until it reaches the appropriate position in relation to the control spool. The kinematic ratio of the mechanism, i.e. the ratio of piston stroke (7) to control spool stroke (10), depends on the length and position of the center of rotation of lever (9). A spring (13) acts on cylinder (12), which eliminates clearance in the mechanism and presses cylinder (12) against lever (9). This pump can have two directions of pressing and can be used for hydraulic closed systems. In such a construction, the pump has a symmetrical design, has two displacement pistons and two follow-up mechanisms. When changing the direction of flow, it is necessary to move the axis of rotation by means of hydraulically controlled bolts (3). Main parameters of tested pump were as follows:

- geometric displacement $q = 40 \text{ cm}^3/\text{rev}$
- pistons spacing $D = 67 \text{ mm}$
- piston diameter $d_t = 15,17 \text{ mm}$
- number of pistons $z = 9$
- dead space volume $V_D = 3,244 \text{ cm}^3$
- max swash plate swing angle $\gamma_{max} = 20^\circ$
- nominal pressure $p_{nom} = 35 \text{ MPa}$

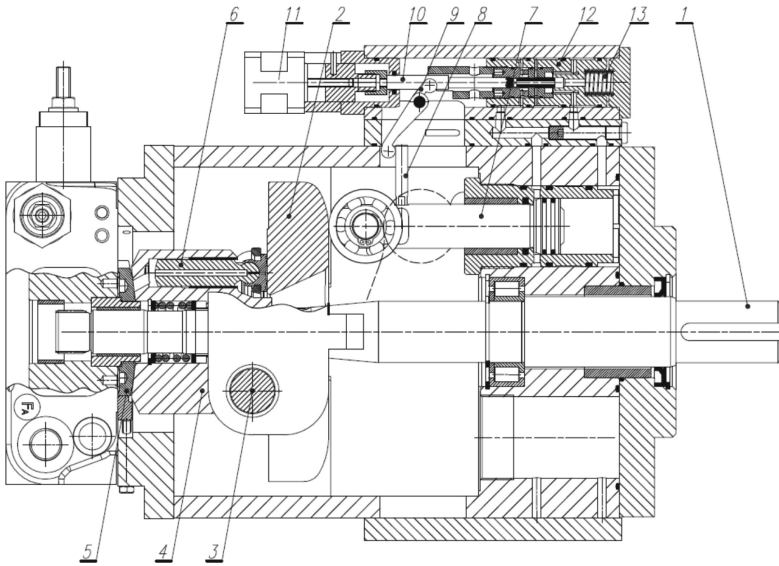


Fig. 6. Axial section of prototype piston pump: 1 - shaft; 2 - swash plate; 3 - bolt; 4 - cylinder drum; 5 - timing plate; 6 - piston; 7 - capacity change piston; 8 - pusher; 9 - lever; 10 - spool; 11 - stepper motor; 12 - cylinder; 13 - spring.

Figure 7 shows the operation of the mechanism of changing capacity when increasing the swash plate swing angle. The follow-up mechanism is delivered straight from the pump pressure collector.

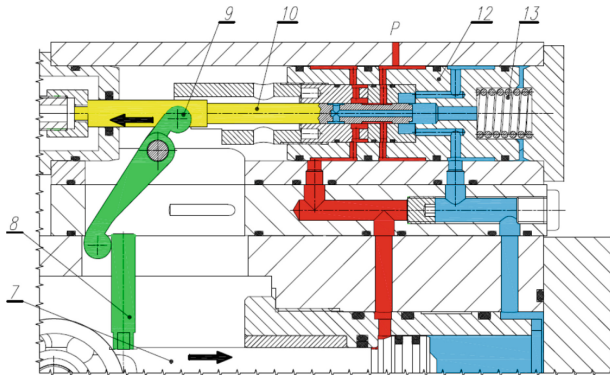


Fig. 7. The functioning of the follow-up mechanism: color red - powered; color blue - drain.

3.2 Test Stand

A scheme of the test stand is shown in Fig. 8. A view of the test stand is shown in Fig. 9. The tested pump was supplied with a supply pump. The suction pressure is regulated

by a relief valve (5) and the delivery pressure by a valve (6). During the tests, pressures in front of and behind the pump, oil temperature in the suction connection, flow rate, torque and speed were recorded. Used working fluid was mineral oil of viscosity class VG 46.

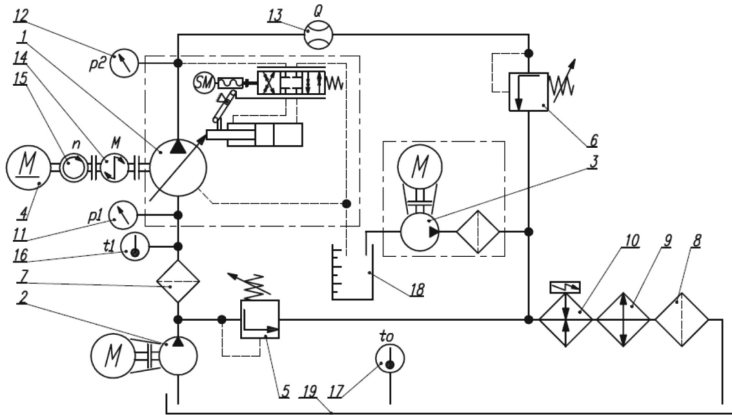


Fig. 8. Scheme of the test stand: 1 - tested pump; 2 - supply pump; 3 - filtering unit; 4 - 70 kW DC electric motor; 5 - suction pressure regulator relief valve; 6 - relief valve 0–40 MPa; 7, 8 - oil filters; 9 - cooler; 10 - 12 × 1,6 kW electric heaters; 11, 12 - pressure gauges; 13 - piston flow meter PT-200; 14 - torque meter HBM T1 500 Nm; 15 - optical speed sensor; 16, 17 - temperature transmitters; 18 - leakage measurement tank; 19 - oil tank 1700 l [9].

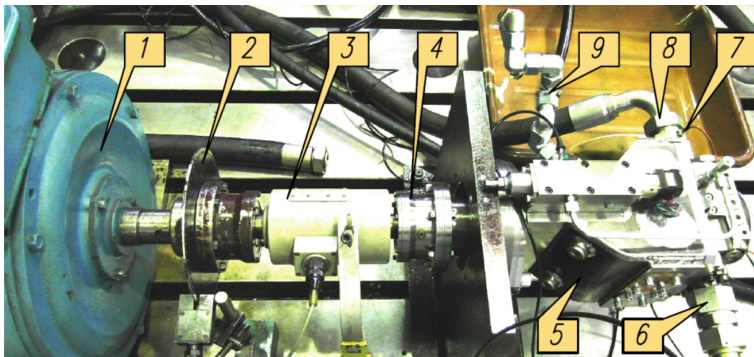


Fig. 9. View of the test stand: 1 - electric motor; 2 - speed measurement; 3 - torque measurement; 4 - clutch; 5 - tested pump; 6 - pressure line; 7 - temperature measurement; 8 - suction line; 9 - leakage line [9].

A list of used test apparatus is presented in Table 1.

Table 1. List of test apparatus.

Instrument	Range	Accuracy class	Max error
Manometer MPS-02	0–1,6 MPa	0,2	0,006 MPa
Manometer KFM	0–40 MPa	0,6	0,25 MPa
Thermocouple	0–70 °C	1	1,7 °C
Piston flow meter PT-200	0,3–200 l/min	0,2	0,4 l/min
Torque meter HBM T1	0–500 Nm	0,2	1,01 Nm
Incremental encoder	0–3000 rpm	–	rpm

3.3 Efficiency Characteristic

Volumetric and hydromechanical efficiency as a function of differential pressure and swash plate swing angle for viscosity 20 and 40cSt and rotational speed 2000 rpm are shown in Fig. 10 and 11.

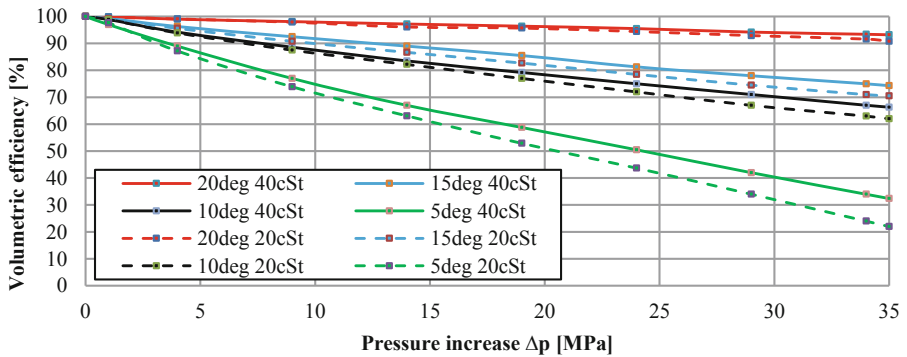


Fig. 10. Volumetric efficiency as a function of differential pressure for different swash plate swing angles and viscosities 20 and 40cSt for 2000 rpm [9].

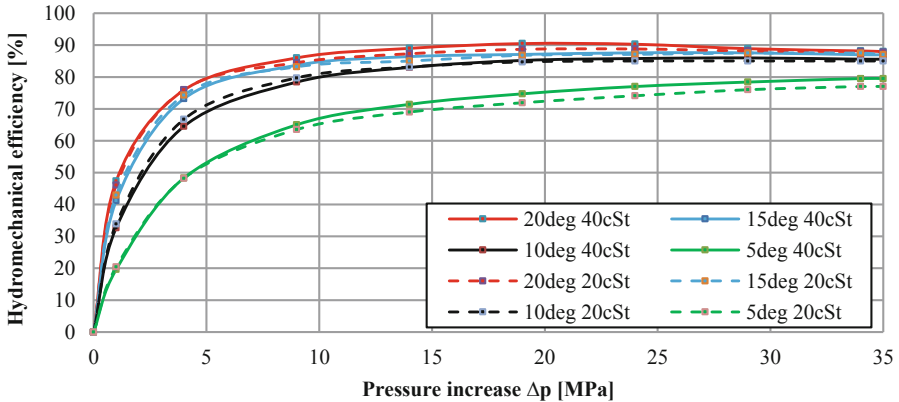


Fig. 11. Hydromechanical efficiency as a function of differential pressure for different swash plate swing angles and viscosities 20 and 40cSt for 2000 rpm [9].

4 Results

The prototype of an axial piston pump with a displaced axis of rotation of the swash plate, based on a commercial pump working mechanism (pistons with slippers, cylinder barrel, timing plate), has achieved a satisfactory volumetric efficiency, comparable to that of commercial pumps. Due to the fixed dead space volume, independent of the swash plate swing angle, the volumetric efficiency was improved for small swash plate swing angles and high pressures. The reason for the reduced hydromechanical efficiency is probably due to a shaft that is too long and can vibrate during pump operation. Tests have also shown problems with the follow-up mechanism, such as too much leakage through the directional valve and too high friction forces on the follow-up cylinder seals, causing it to lock. The results of the tests will be used to create a second, upgraded prototype.

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