



RESEARCH ARTICLE

Hydraulic Cylinder Testing Systems with Energy Recovery

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ABSTRACT

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The article analyzes hydraulic systems of benches for testing piston hydraulic cylinders. Particular attention is given to benches with energy recovery during testing, and their disadvantages are identified. Materials and Methods. An original design of a bench for testing hydraulic cylinders with energy recovery is proposed. Results and Discussion. The operating algorithm of the bench is presented, and its advantages over currently known test benches are indicated.

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INTRODUCTION

Testing is a critical stage in the development of positive-displacement hydraulic machines because it must identify and eliminate design shortcomings and process errors, determine the characteristics, energy performance, service life, reliability, starting and maneuvering properties, noise characteristics, and other operating indicators of the hydraulic system. Tests are conducted to confirm durability, that is, to assess the probability that hydraulic machines will achieve their assigned service life or service life before the first failure. Bench endurance testing is also relevant under assembly-production conditions, since hydraulic machines supplied by manufacturers often do not achieve the required service life, which leads to failure of the entire hydraulic system (Nikitin, 2007; Ponamorenko, 1969).

An important condition for testing is to ensure the closest possible correspondence between the loading modes of the hydraulic machines being tested and their operating conditions in service.

Figure 1 shows a test bench diagram proposed by the standard (GOST 18464-96, 2001) for testing hydraulic cylinders for operation under load, rod sealing, pushing and pulling force on the rod, piston (plunger) speed, overall and mechanical efficiency, service life, time to failure, and braking. The main disadvantage of this scheme is that the load on the hydraulic cylinder being tested is generated by throttling the working fluid in pressure valve 14. This causes several problems, including the release of a large amount of heat, which leads to heating of the working fluid and therefore requires cooling. The main negative effect of this testing method is significant energy loss and, consequently, low profitability of the testing process (Bashta, 1974).

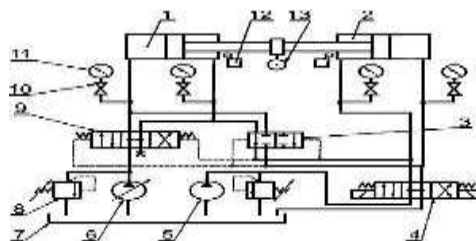


Figure 1: Diagram of a Test Bench for Piston Hydraulic Cylinders

Hydraulic cylinder under test; 2 – loading hydraulic cylinder; 3 – two-position hydraulically controlled directional control valve; 4 – three-position solenoid-operated directional control valve; 5 Loading pump;

6 Drive pump; 7 – hydraulic tank; 8 – pressure relief valve;

9 Three-position hydraulically controlled directional control valve;

10 Damping valve; 11 – pressure gauge; 12 – switch; 13 – dynamometer

The same standard (GOST 18464-96, 2001) proposes a scheme for testing piston hydraulic cylinders with energy recovery. A diagram of a test bench for testing hydraulic cylinders with energy recovery is shown in Figure 2.

In this case, the load on hydraulic cylinder 10 under test is created by loading hydraulic cylinder 11, in which force arises from static pressure generated by loading pump 2. However, the use of this energy-recovery scheme leads to a testing mode in which the operation of hydraulic cylinder 10 under test differs substantially from its operation under real service conditions. As a result, the mechanical motion of the hydraulic cylinder under test leads to increased and nonuniform friction, while static-pressure loading is applied against the direction of rod movement; the piston seals being tested therefore operate in a mode that does not correspond to the technical operating specifications.

The main aim of this study is to develop a test bench for piston hydraulic cylinders that brings the operating conditions of the hydraulic cylinders under test as close as possible to real service conditions and minimizes energy losses during testing.

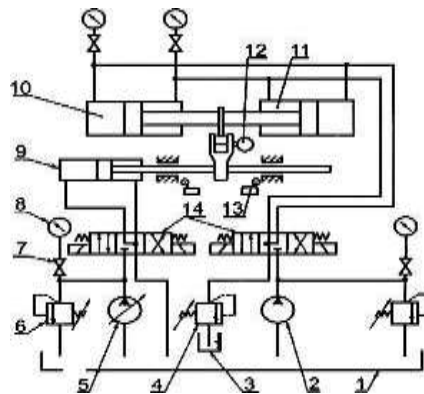


Figure 2: Diagram of a Test Bench for Hydraulic Cylinders

1 Hydraulic Tank; 2 – Loading Pump; 3 – Measuring Tank; 4 – Back-Pressure Valve;

5 Drive Pump; 6 – Pressure Relief Valve; 7 – Damping Valve; 8 – Pressure Gauge; 9 – Drive Hydraulic Cylinder; 10 – Hydraulic Cylinder Under Test; 11 – Loading Hydraulic Cylinder; 12 – Dynamometer; 13 – Switch; 14 – Directional Control Valve

MATERIALS AND METHODS

The development of an energy-recovery bench for endurance testing of hydraulic cylinders was based on the previously proposed method for testing positive-displacement rotary hydraulic machines with energy recovery (Bashta, 1974; Rybak et al., 2021; Zharov et al., 2011; Ustyantsev, 2012). In this case, one hydraulic machine under test, for example a hydraulic pump, is loaded by another hydraulic machine under test, a hydraulic motor, and vice versa. This method significantly reduces the dimensions and simplifies the test-bench scheme because no special loading systems are required. The method increases the efficiency of the test system and, consequently, reduces energy consumption. Rybak et al. (2011) propose a bench scheme for testing hydraulic machines with different working volumes, where their operation is coordinated by introducing a mechanical transmission, while the hydraulic line is a pressure line with adjustable load.

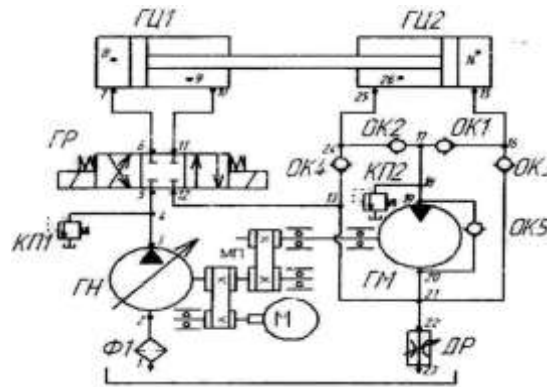


Figure 3: Schematic Hydraulic Diagram of a Bench for Testing Piston Hydraulic Cylinders with Energy Recovery

This study proposes a scheme for testing hydraulic cylinders in which the pressure line of a bench for testing rotary hydraulic machines is used to test piston hydraulic cylinders (Chukarin et al., 2009; Rybak et al., 2009). The schematic hydraulic diagram of the bench for testing piston hydraulic cylinders is shown in Figure 3. The essence of this hydraulic-cylinder testing scheme is that a pair of piston hydraulic cylinders, HC1 and HC2, is installed in the pressure hydraulic line of the rotary hydraulic-machine testing scheme between the hydraulic pump and the hydraulic motor.

RESULTS AND DISCUSSION

A preliminary analysis of the operation of the proposed energy-recovery bench for testing hydraulic cylinders showed its technical advantage over the standard schemes currently used. However, the hydraulic-cylinder testing system shown in Figure 3 has a substantial disadvantage: the pistons of the hydraulic cylinders under test are connected directly to each other, which causes the operating mode of the cylinders during testing to differ substantially from their operation under real service conditions. In this cylinder connection scheme, the hinges connecting the cylinder body and rod do not operate under the required load, and the rod and piston seals are not properly tested because the load on the cylinder rods acts strictly along the cylinder axis. To eliminate this disadvantage, a mechanical intermediate element, a rocker arm, is proposed for introduction into the bench design between the cylinders under test. This makes the testing conditions closer to the real operating conditions of hydraulic cylinders [10]. A diagram of the proposed energy-recovery hydraulic-cylinder testing system is shown in Figure 4.

The bench operates as follows. Electric motor 1 converts electrical energy into rotational motion of shaft 2 and then, through second mechanical transmission 3, drives shaft 4 of hydraulic pump 5. The mechanical energy supplied from the shaft of electric motor 1 is converted by hydraulic pump 5 into hydraulic energy of the working fluid. Through the line, the fluid enters directional control valve 7, which directs it, for example, through the line into the rod chamber of the first hydraulic cylinder under test, 9, which in this case functions as a hydraulic motor.

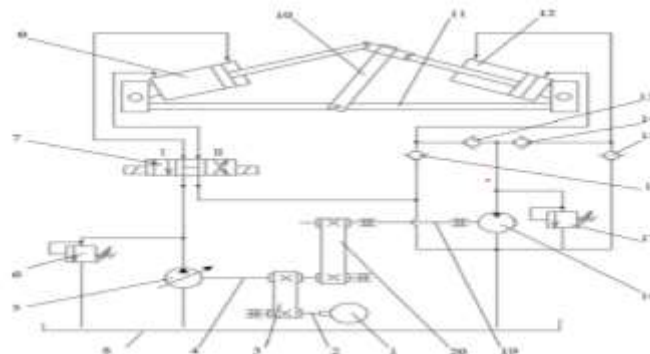


Figure 4: Hydro-Kinematic Diagram of a Bench for Testing Piston Hydraulic Cylinders with Energy Recovery

The hydraulic energy of the working fluid, through the operation of first hydraulic cylinder 9, is converted into the mechanical energy of piston displacement. Through first mechanical transmission 10 installed on beam 11, this piston is mechanically connected to the rod of second hydraulic cylinder 12 under test. The mechanical energy of the piston displacement of second hydraulic cylinder 12, which in this case functions as a hydraulic pump, is converted into hydraulic energy of the working fluid located in its rod chamber. Through the line, the working fluid is directed through check valve 14 to the inlet of hydraulic motor 18. In hydraulic motor 18, the hydraulic energy of the working fluid is converted into the mechanical energy of rotation of shaft 19 of hydraulic motor 18. Through mechanical transmission 20, this energy is transmitted to shaft 4 of hydraulic pump 5, where it is combined with the rotational energy of shaft 2 of electric motor 1 and is again converted by hydraulic pump 5 into the energy of the working fluid. During the movement of the piston of second hydraulic cylinder 12 from the extreme right position to the extreme left position, its piston chamber is filled with working fluid by suction from the low-pressure hydraulic line through check valve 16.

To move the piston of first hydraulic cylinder 9 from the extreme left position to the extreme right position, directional control valve 7 directs the fluid supplied to its inlet through the line into the piston chamber of first hydraulic cylinder 9. At the same time, the working fluid displaced from the piston chamber of second hydraulic cylinder 12 flows through the line and check valve 13 to the inlet of hydraulic motor 18; its rod chamber is filled with working fluid by suction from the low-pressure hydraulic line through check valve 15.

In the proposed scheme, mechanical transmission 20 creates a condition under which the delivery of hydraulic pump 5 slightly exceeds the required flow rate of hydraulic motor 18 at the specified rotational speed of shaft 4 of hydraulic pump 5, corresponding to the rotational speed of shaft 2 of electric motor 1. This leads to an increase in pressure in the pressure line connecting second hydraulic cylinder 12 to hydraulic motor 18 up to the setting pressure of relief valve 17. Excess working fluid is discharged through relief valve 17 from this line into the low-pressure hydraulic line. Thus, by adjusting relief valve 17, the pressure in the line at the outlet of second hydraulic cylinder 12 can be changed, thereby regulating the loading of hydraulic cylinders 9 and 12 under test.

Check valves 14, 15, 16, and 13 provide unloading conditions for the second hydraulic circuit when it reaches the specified operating modes and hydraulic motor 18 operates as a hydraulic pump. Relief valve 6 discharges excess working fluid from the outlet of pump 5 into hydraulic tank 8 if the pump is overloaded.

Thus, when piston hydraulic cylinders are tested using the proposed testing method and scheme (Rybak et al., 2023), they are tested under conditions close to their real operating conditions and with significant energy savings.

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