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REDUCING THE ENERGY CONSUMPTION IN THE HYDRAULIC CYLINDER ENDURANCE TEST

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Abstract: *From inventorizing the tests and verifications to which the hydraulic cylinders are subjected, it is found that the endurance test, which establishes the normal service life, is very energy-consuming. This energy consumption is caused both by the duration of the test and the fact that it is carried out at rated power. Starting from stand diagrams for energy recovery in endurance tests on rotary positive displacement machines, existing in the literature, the authors of this paper develop a similar diagram, intended for endurance tests on hydraulic cylinders.*

Keywords*: Energy consumption, endurance, hydraulic cylinders*

1. Introduction

Tests on general use hydraulic cylinders must contain the following information: name, destination, symbolization and hydraulic diagram; values of the functional parameters listed in the Table 1; conditions for use in hydraulic diagrams; data on braking at the end of stroke (type, adjustment mode, etc.); conditions for mounting (position, fastening mode, etc.), connecting and commissioning; admissible non - coaxiality of the drive force against the geometric axis of the cylinder; content in dust, water and aggressive substances in the environment in which the cylinders can operate normally; maintenance conditions; type of functional characteristics to be determined; reliability indicators.

Table 1: Functional parameters of hydraulic cylinders

Item no.	Parameter name		Symbol	Measurement units	
				SI	**Permissible**
1.	Rated pressure		p_n	N/m^2	bar
2.	Main dimensions	Rated bore of the cylinder (piston or plunger diameter)	D	mm	
		Rod diameter	d	mm	
		Piston stoke	L	mm	
		Active surface ratio (for differential cylinders)	φ	-	
	Main dimensions of telescopic cylinders	Active diameters of the extension steps 1...n	$D_1 ... D_n$	mm	
		Active diameters of the retraction steps 1...n	$d_1 ... d_n$	mm	
		Strokes of the pistons 1...n	$L_1 ... L_n$	mm	
		Total cylinder stroke	L	mm	
3.	Rated force	thrust		N	
		tensile		N	
4.	Piston speed	minimum	v_{min}	m/s	
		maximum	v_{max}	m/s	
5.	Total efficiency	thrust	$\eta=f(p)$ $\eta=f(v)$	- -	
		tensile	$\eta=f(p)$ $\eta=f(v)$	- -	

Table 1: Functional parameters of hydraulic cylinders (continued)

Item no.	Parameter name		Symbol	Measurement units	
				SI	SI
6.	Working fluid	Fluid type			
		Minimum kinematic viscosity	ν_{min}	mm^2/s	cSt
		Optimum kinematic viscosity	ν_{opt}	mm^2/s	cSt
		Maximum kinematic viscosity	ν_{max}	mm^2/s	cSt
		Minimum temperature	t_{min}	K	^{0}C
		Maximum temperature	t_{max}	K	^{0}C
7.	Ambient temperature	minimum	t_{min}	K	^{0}C
		maximum	t_{max}	K	^{0}C
8.	Cylinder mass (without working fluid)		m	kg	-

As part of the type, periodic and batch checks on hydraulic cylinders the tests indicated in the Table 2 shall be carried out.

Table 2: Tests and checks on hydraulic cylinders

Item no.	Technical condition to be verified		Checks		
			type	periodic	batch
1.	Appearance		x	x	x
2.	Dimensions of gauge and connection		x	x	x[1)]
3.	Functioning		x	x	x
4.	Quality of materials and dimensional checks on the main parts and subassemblies		x	x	-
5.	Cylinder mass (without working fluid)		x	x	-
6.	Pressure	minimum for piston displacement	x	x	x
		at the start	x	x	x
7.	Force	thrust	x	x	x
		tensile	x	x	x
8.	Piston speed	minimum	x	x	-
		maximum	x	x	-
9.	Tightness	internal	x	x	x
		external	x	x	x
10.	Braking at the end of stroke		x	x	-
11.	Pressure resistance		x	x	x[1)]
12.	Plotting the characteristic curves		x	x	-
13.	Functioning at cut-off temperatures		x	-	-
14.	***Functioning time (endurance)***		***x[2)]***	-	-
15.	Reliability		x	-	-

[1)] The check can be performed by sampling. The size of the sample lot and the acceptance conditions will be determined by the technical documentation.

[2)] 100000 cycles are performed, at rated power, then internal tightness is checked; no external leakage is allowed. The test is very energy-consuming. Stands with energy recovery are recommended.

2. Energy recovery in endurance tests on rotary positive displacement machines

During the endurance tests on rotary positive displacement machines energy consumption shall be reduced by simultaneous testing of two machines, hydraulically connected in a closed circuit, one operating as a pump and the other as a motor. The hydraulic power produced by the pump is re-used to drive the pump through the motor. Thus, the power supplied to the system must cover the difference between the power consumed by the pump and that supplied by the motor; this energy-saving process is called "recirculation of hydromechanical power" and it can be materialized with several types of diagrams, which differ by the power loss compensation mode.

2.1 Mechanical compensation of power losses by using one adjustable machine

In the case of mechanical compensation the energy source is an electromotor. If one of the machines under tests is adjustable, there is used the diagram in the Figure 1, characterized by the coupling of the two machines via the electric motor (np = nm = n).

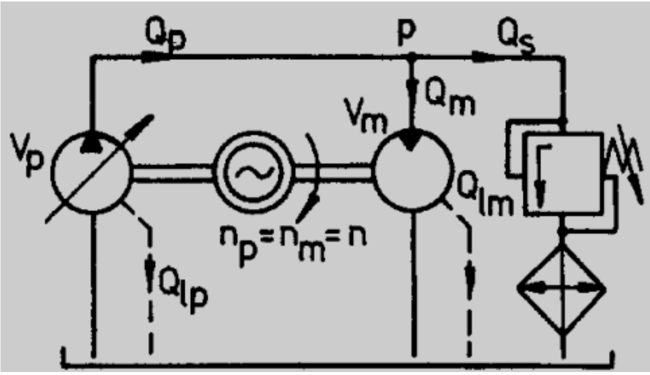

Fig. 1. Stand with mechanical compensation of power losses, an adjustable machine and a fixed machine

If the flow rate provided by pump, Q_p, is equal to the one used by motor, Q_m, pump discharge pressure, p, is practically null; as the pump capacity, V_p, increases relative to the motor capacity V_m, its discharge pressure increases to evacuate the excess flow through clearances and gaps. The normally closed valve limits the pressure p to the rated value specific to the tested machines.
The power supplied by electromotor, N_e, is the difference between the power absorbed by pump, N_p, and the one provided by motor, N_m:

$$N_e = N_p - N_m \tag{1}$$

Where
$$N_p = \frac{p \cdot Q_{tp}}{\eta_{tp}} = \frac{p \cdot n \cdot V_p}{\eta_{tp}} \tag{2}$$

and
$$N_m = p \cdot Q_{tm} \cdot \eta_{tm} = p \cdot V_m \cdot \eta_{tm} \tag{3}$$

It results:
$$N_e = p \cdot n \cdot V_m \left(\frac{V_p}{V_m} \cdot \frac{1}{\eta_{tp}} - \eta_{tm} \right) = N_{tm} \left(\frac{V_p}{V_m} \cdot \frac{1}{\eta_{tp}} - \eta_{tm} \right) \tag{4}$$

This power is minimal if the valve flow is null, so:

$$Q_p = n \cdot V_p \cdot \eta_{vp} = Q_m = \frac{n \cdot V_m}{\eta_{vm}} \tag{5}$$

It results:
$$\left(\frac{V_p}{V_m} \right)_{min} = \frac{1}{\eta_{vp} \cdot \eta_{vm}} \tag{6}$$

and
$$\left(\frac{N_e}{N_{tm}} \right)_{min} = \frac{1}{\eta_{tp} \cdot \eta_{vp} \cdot \eta_{vm}} - \eta_{tm} \tag{7}$$

For instance, for $\eta_{vp} = \eta_{vm} = 0.95$ and $\eta_{tp} \cong \eta_{tm} = 0.9$, it is required that $(V_p/V_m)_{min} \cong 1.1$ and $(N_e/N_{tm})_{min} \cong 0.33$.

$$\left(\frac{V_p}{V_m} \right)_{min} = \frac{1}{\eta_{vp} \cdot \eta_{vm}} = \frac{1}{0.95 \cdot 0.95} = 1.108$$

$$\left(\frac{N_e}{N_{tm}} \right)_{min} = \frac{1}{\eta_{tp} \cdot \eta_{vp} \cdot \eta_{vm}} - \eta_{tm} = \frac{1}{0.9 \cdot 0.95 \cdot 0.95} - 0.9 = 0.331$$

2.2 Mechanical compensation of power losses by using two fixed machines with equal capacities

If the machines have equal and constant capacity, $V_p = V_m = V$, the electromotor and the hydraulic motor must drive the pump through a speed multiplier with the transmission ratio $i = n_p / n_m > 1$ (Figure 2).

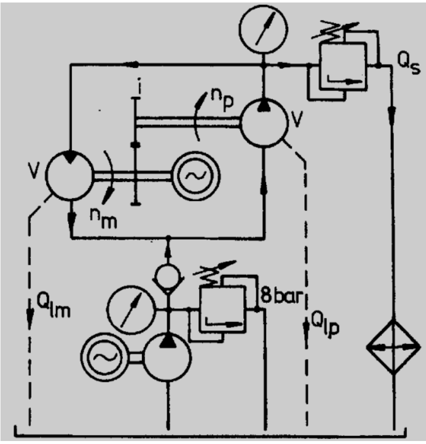

Fig. 2. Stand with mechanical compensation of power losses and two fixed machines

In this case
$$N_p = p \cdot i \cdot n_m \cdot \frac{V}{\eta_{tp}} \tag{8}$$

$$N_m = p \cdot n_m \cdot V \cdot \eta_{tm} \tag{9}$$

$$N_e = N_p - N_m = p \cdot n_m \cdot V \left(\frac{i}{\eta_{tp}} - \eta_{tm}\right) = N_{tm}\left(\frac{i}{\eta_{tp}} - \eta_{tm}\right) \tag{10}$$

So
$$\frac{N_e}{N_{tm}} = \frac{i}{\eta_{tp}} - \eta_{tm} \tag{11}$$

The flow discharged through the testing pressure control valve is calculated from the continuity equation:

$$Q_p = i \cdot n_m \cdot V \cdot \eta_{vp} = Q_m + Q_s = n_m \frac{V}{\eta_{vm}} + Q_s \tag{12}$$

So
$$Q_s = n_m \cdot V \left(i \cdot \eta_{vp} - \frac{1}{\eta_{vm}}\right) \tag{13}$$

Or
$$\frac{Q_s}{Q_{tm}} = i \cdot \eta_{vp} - \frac{1}{\eta_{vm}} \geq 0 \tag{14}$$

The minimum value of the transmission ratio is determined from the condition that the flow through the valve to be null ($Q_s = 0$):

$$i_{min} = \frac{1}{\eta_{vp} \cdot \eta_{vm}} > 1 \tag{15}$$

The minimum power provided by the electromotor is the same as the one in the previous case. If $i = 1.15$, $\eta_{tp} \cong \eta_{tm} = 0.9$ and $\eta_{vp} \cong \eta_{vm} = 0.95$, $Q_s / Q_{tm} = 0.04$ and $N_e / N_{tm} \cong 0.38$.

$$\frac{Q_s}{Q_{tm}} = i \cdot \eta_{vp} - \frac{1}{\eta_{vm}} = 1.15 \cdot 0.95 - \frac{1}{0.95} = 1.0925 - 1.0526 = 0.0399$$

$$\frac{N_e}{N_{tm}} = \frac{i}{\eta_{tp}} - \eta_{tm} = \frac{1.15}{0.9} - 0.9 = 0.377$$

2.3 Mechanical compensation of power losses by using an auxiliary pump

Hydraulic power loss compensation is performed by means of an auxiliary pump according to the diagram in Figure 3.

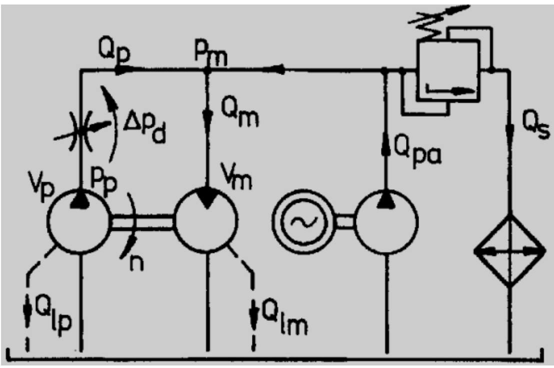

Fig. 3. Stand with hydraulic compensation of power losses

If the valve flow is null, from the continuity equation

$$Q_m = \frac{n_m \cdot V_m}{\eta_{vm}} = Q_p + Q_{pa} = n_m \cdot V_p \cdot \eta_{vp} + Q_{pa} \tag{16}$$

pump and motor speed result:

$$n_m\left(\frac{V_m}{\eta_{vm}} - V_p \cdot \eta_{vp}\right) = Q_{pa} \quad ; \qquad n_m = n_p = \frac{Q_{pa}}{\frac{V_m}{\eta_{vm}} - V_p \cdot \eta_{vp}} \tag{17}$$

Where Q_{pa} is the flow rate of the auxiliary pump.
From the system of equations

$$N_p = \frac{Q_p \cdot p_p}{\eta_{tp}} = N_m = Q_m \cdot p_m \cdot \eta_{tm} \tag{18}$$

$$p_p = p_m + \Delta p_d \tag{19}$$

one can deduce pressure drop on the motor, as a function of pressure drop across the throttle, Δp_d:

$$Q_p(p_m + \Delta p_d) = Q_m \cdot p_m \cdot \eta_{tm} \cdot \eta_{tp} \ ; \ p_m = \frac{\Delta p_d}{\frac{V_m}{V_p} \cdot \frac{\eta_{tp} \cdot \eta_{tm}}{\eta_{vm} \cdot \eta_{vp}} - 1} \tag{20}$$

The capacities of the tested machines are chosen or adjusted, so that

$$\frac{V_m}{V_p} \cdot \frac{\eta_{tp} \cdot \eta_{tm}}{\eta_{vm} \cdot \eta_{vp}} > 1 \tag{21}$$

so the motor capacity must always be higher than the pump capacity:

$$\frac{V_m}{V_p} > \frac{1}{\frac{\eta_{tp} \cdot \eta_{tm}}{\eta_{vm} \cdot \eta_{vp}}} > 1 \tag{22}$$

If the denominator of the equation (20) is very small, a low pressure drop across the throttle generates large test pressures.
The power consumed by the stand is equal to the power of the auxiliary pump:

$$N_{pa} = N_e = p_m \cdot Q_{pa} \cdot \frac{1}{\eta_{tpa}} = \frac{n_m \cdot \Delta p_d\left(\frac{V_m}{\eta_{vm}} - V_p \cdot \eta_{vp}\right)}{\eta_{tpa}\left(\frac{V_m}{V_p} \cdot \frac{\eta_{tp} \cdot \eta_{tm}}{\eta_{vm} \cdot \eta_{vp}} - 1\right)} \tag{23}$$

For instance, if $V_m \cdot \eta_{tp} \cdot \eta_{tm} / V_p \cdot \eta_{vm} \cdot \eta_{vp} = 1.025$, it results $p_m = 40 \cdot \Delta p_d$.

$$p_m = \frac{\Delta p_d}{\frac{V_m}{V_p} \cdot \frac{\eta_{tp} \cdot \eta_{tm}}{\eta_{vm} \cdot \eta_{vp}} - 1} = \frac{\Delta p_d}{1{,}025 - 1} = \frac{\Delta p_d}{0{,}025} = 40 \Delta p_d$$

A pressure drop of *10bar* across the throttle results in a motor test pressure $p_m = 200$ *bar*, and $p_p = 210$ *bar*.

$$p_p = p_m + \Delta p_d = 200 + 20 = 220\ bar$$

Admitting that $\eta_{tp} = \eta_{tm} = 0.9$ and neglecting the volumetric efficiencies, it results $V_m / V_p \cong 1.265$.
If $V_p = 125\ cm^3/rev$, $V_m = 158.1\ cm^3/rev$; at $n = 1000\ rev/min$ and $\eta_{vp} \cong \eta_{vm} = 0.95$, the auxiliary pump must supply the flow rate $Q_{pa} = 0.795\ l/s$.

$$Q_{pa} = Q_m - Q_p = n\left(\frac{V_m}{\eta_{vm}} - V_p \cdot \eta_{vp}\right) = 1\,000 \cdot 10^{-3}\left(\frac{158.1}{0.95} - 125 \cdot 0.95\right) = 47.67 \frac{l}{min} = 0.7945 \frac{l}{s}$$

For $\eta_{tpa} \cong 0.9$, the auxiliary pump consumes the power $N_{pa} = 17.65\ kW$, while the theoretical power of the tested motor is $N_{tm} = 47.43\ kW$, so $N_e / N_{tm} = 17.65 / 47.43 = 0.37$.

$$N_{pa} = N_e = p_m \cdot Q_{pa} \cdot \frac{1}{\eta_{tpa}} = 200 \cdot 10^5 \cdot 0.7945 \cdot 10^{-3} \cdot \frac{1}{0.9} = 176.5 \cdot 10^2 W = 17.65\ kW$$

$$N_{tm} = Q_m \cdot p_m \cdot \eta_{tm} = \frac{Q_{pa}}{\frac{V_m}{\eta_{vm}} - V_p \cdot \eta_{vp}} \cdot \frac{V_m \cdot p_m \cdot \eta_{tm}}{\eta_{vm}}$$
$$= \frac{0.7945 \cdot 10^{-3}}{10^{-6}\left(\frac{158.1}{0.95} - 125 \cdot 0.95\right)} \cdot \frac{10^{-6} \cdot 158.1 \cdot 200 \cdot 10^5 \cdot 0.9}{0.95} = 47.43\ kW$$

3. Stand with energy recovery for endurance tests on hydraulic cylinders

As an extension of the stand with mechanical compensation of power losses (Figure 1), used at the simultaneous endurance testing of two rotary positive displacement machines (a pump and a motor), the power recirculation stand shown in Figure 4 is developed, for performing endurance tests, with energy saving, on hydraulic cylinders.

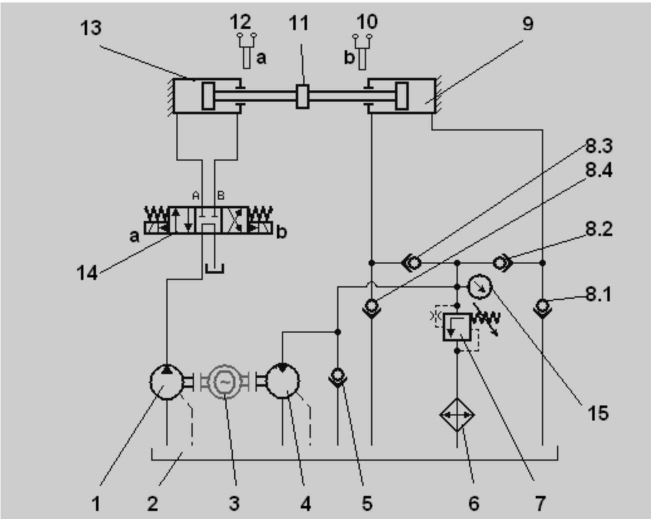

Fig. 4. Power recirculation stand for endurance tests on hydraulic cylinders

The stand in Figure 4 has the following advantages: it has a single pumping group for the test cylinder, and for the load cylinder the hydraulic oil supply is based on its operation in the pump mode; it has a single electrohydraulic directional valve for controlling the displacement of the two cylinders; it operates on the basis of "recirculation of hydromechanical power"; energy dissipation in heat is reduced, due to a much smaller flow discharged to the tank, through a single normally closed pressure valve; it requires small oil coolers.

The electromotor **3** of the stand has two drive heads, to which there are coupled a fixed positive displacement pump **1**, which absorbs from the oil tank **2**, and a fixed hydraulic motor **4**. The two hydraulic cylinders are identical; one of them is for testing **13**, the other is a load cylinder **9**. They have their rods fixed in the coupling **11** and they can move between two stroke limiters, namely towards the limiter **10**, when to electrically driven hydraulic directional valve **14** the electromagnet **a** is switched, and towards the limiter **12**, when to hydraulic directional valve the electromagnet **b** is switched. The check valve **5** allows hydraulic oil supply of the hydraulic motor from the tank in the non-actuated position of the hydraulic directional valve, and the check valves **8.4** and **8.2** allow, when the electromagnet **a** is switched, oil supply to the rod chamber of the load cylinder, and respectively oil discharge from the piston chamber of the load cylinder. The check valves **8.1** and **8.3** allow, when the electromagnet **b** is switched, oil supply to the piston chamber of the load cylinder, and respectively oil discharge from the rod chamber of the load cylinder. The stand is also equipped with a testing pressure control valve **7**, the pressure gauge **15**, which reads the adjusted pressure and the oil-water cooler **6**.

While the electromotor **3** is running and the directional control valve **14** is not actuated, the fixed pump is driven idle, the hydraulic cylinders **13** and **9** do not move, the hydraulic motor **4** is also driven by the electromotor and supplied through the check valve **5,** which opens.

The stand operates in two modes: **manual mode**, through which the hydraulic circuits are aerated / filled with oil and the test pressure is adjusted; **automatic mode**, through which the endurance test is performed at the set pressure.

In manual mode the electromagnets **a** and **b** are manually operated, and the testing pressure is adjusted by means of valve **7** and pressure gauge **15**.

In automatic mode the actuation of electromagnets **a** and **b** is made from the automation panel of the stand, depending on the signals received from the stroke limiters **10** and **12**, and the operation of the stand is as follows:

When the electromagnet **a** is switched on the spool valve of the hydraulic directional valve moves to the left position, the pump **2** sucks out of the tank and discharges in the chamber of the cylinder **13** piston, which expands its volume, and the cylinder **13** rod chamber collapses in volume, the oil being discharged through the directional valve to the tank. The effect of varying the volume of the two chambers is the displacement of the cylinder **13** rod to the right. The coupling **11** in this movement also drives the cylinder **9** rod. By displacement of this rod, the cylinder **9** rod chamber expands in volume and sucks oil out of the tank, through the check valve **8.4**, which opens, the cylinder **9** piston chamber collapses in volume and discharges oil through the check valve **8.2**, which opens, on two circuits: a larger part on the hydraulic motor **4** intake circuit, and a smaller part, equal to the difference between pump flow and motor flow, through the valve **7**.

When the electromagnet **b** is switched on the spool valve of the hydraulic directional valve moves to the right position, the pump **2** sucks out of the tank and discharges in the chamber of the cylinder **13** rod, which expands its volume, and the cylinder **13** piston chamber collapses in volume, the oil being discharged through the directional valve to the tank. The effect of varying the volume of the two chambers is the displacement of the cylinder **13** rod to the left. The coupling **11** in this movement also drives the cylinder **9** rod. By displacement of this rod, the cylinder **9** piston chamber expands in volume and sucks oil out of the tank, through the check valve **8.1**, which opens, the cylinder **9** rod chamber collapses in volume and discharges oil through the check valve **8.3**, which opens, on two circuits: a larger part on the hydraulic motor **4** intake circuit, and a smaller part, equal to the difference between pump flow and motor flow, through the valve **7**.

To avoid cavitational wear of the load cylinder it is recommended either to overfill it or that the oil tank be mounted above the hydraulic cylinders.

Conclusions

Two rotary positive displacement machines, a pump and a motor can be subjected to endurance tests at the same time, in advantageous conditions in terms of power consumption, by using three methods:

a) Mechanical compensation of power losses, based on the coupling of the drive axles of the two positive displacement machines, via a two-axis electromotor or an electromotor with a drive shaft and 1:1 ratio gear transmission. In this case the rotative speeds of the two machines are equal ($n_p = n_m$), and the geometric volume of the pump is higher than that of the hydraulic motor ($V_p > V_m$);

b) Mechanical compensation of power losses, based on the coupling of the drive axles of the two positive displacement machines, via an $i = n_p / n_m > 1$ ratio gear transmission. In this case the geometric volumes of the two machines are equal ($V_p = V_m$), and the pump speed is higher than the motor speed $n_p > n_m$;

c) Hydraulic compensation of power losses by using an auxiliary pump. In this case the rotative speeds of the two cars are equal ($n_p = n_m$), and the geometric volume of the hydraulic motor is greater than that of the pump ($V_m > V_p$).

In the *a)* and *b)* cases the power supplied by the electromotor represents the difference between the power absorbed by the pump and that supplied by the hydraulic motor, and in the *c)* case the power supplied by the electromotor is equal to the power consumed by the auxiliary pump.
The *a)* case for conducting endurance tests on rotary positive displacement machines can be extended to endurance tests on hydraulic cylinders, with the stand illustrated in the figure 4.

Acknowledgments

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