# Hydraulic actuators for fatigue tests of helicopter assemblies

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**Abstract:**

The new helicopter assembly ground-based fatigue test is rather expensive due to the construction of new test rigs. The significant part of the cost involves the expenses for hydraulic loading system: hydraulic cylinders, control rods and walking beams. The application of short stroke dynamic hydro cylinders significantly reduces the cost of dynamic rigs.

The analytic, design and fabrication methods to construct relatively cheap hydraulic dynamic actuators for fatigue tests have been developed. The methods of rod bearing, rod and piston sealing design are developed.

By means of these techniques, some new hydraulic cylinders with nominal forces 30, 50, 150 KN and stroke 40 and 60 mm were manufactured. They are successfully used in dynamic test rigs under 10 kHz.

## INTRODUCTION

Helicopter units are very complicated structures and their full-scale ground test is a very difficult engineering problem. They are required to load a specimen with forces and displacements in several points at several directions under a specific way and at high frequency. Electrohydraulic test rigs with servocontrol are often the only possible means to provide test programs.

The quality of dynamic test rig depends upon hydraulic actuators. Their design, weight and dimensions are the main factors determining the development of accurate high frequency loadings. The hydraulic actuator must have low friction force and durability under large-scale lateral rod loading. A cylinder with hydrostatic rod bearings is the most suitable for dynamic test rig. Our hydrostatic middle stroke actuator with rated loading 20 kH [1] is shown in Fig.1.

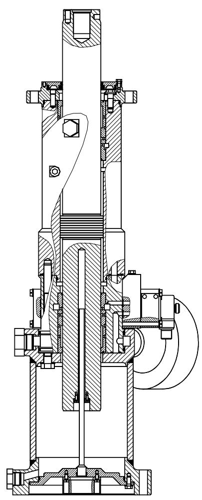

*Figure 1. Dynamic hydrostatic actuator*

The main advantage of this actuator type is absolute absence of contact friction; hence one can maintain high speed and loading frequency at practically unlimited resource. However this actuator is very complicated and expensive. Besides it is bulky and heavy, so it must be mounted only with fixed fastening of the case. The load forces must be transmitted to a sample through long beams and arms. Consequently test rig dimensions and price grow.

At the same time the greater part of assembly operation tests is fatigue tests in which speed and displacement are not very high. And we have possibility to mount actuators on the end bearings. The rod end with dynamometer is connected to the test specimen by spherical bearings. The other end of the actuator is connected with the base frame by spherical bearings too. So one does not have to use intermediate beams and arms, which reduces test rig displacements and its price.

As it is shown in [2] the actuator vibrates laterally with double frequency and the determinative lateral force loading the hydraulic actuator rod is inertia force:

$$R_{\max} \approx M \frac{rR_m}{h^2} \varphi_o^2 \omega^2 \qquad (1)$$

where: $M$ – actuator mass; $Rm$ – distance between mass center and lower bearing; $r$ - radius of sample arm; $\varphi_0$ – angular amplitude of sample arm; $\omega$ – angular speed; $h$- distance between axis of sample arm and lower bearing.

Hence (1) main requirements to an actuator are its minimum dimension and minimum mass. Minimal moving mass under the lowest friction and long life are desirable too.

Actuator determinative components are rod bearings, the noncontact piston, and rod seals.

## 1. ROD BEARINGS (ROD GUIDE SLEEVE)

The best material for hydraulic actuator rod bearing is bronze bearing with chromium-plated rod although there are many types of polymer materials. Rod bearing construction must provide its lubrication. The design of a drainage flute behind the bronze rod before a sealing makes it possible for the oil to flow through the bearing and decrease pressure at the output rod seals.

Due to the rod curving in the guide sleeves it rotates at the angle $\theta$. In upper bearing:

$$\theta = 2/3 * (1/6 * P * A * L / (E * J)), \qquad (2)$$

where: $P$ – lateral force on the end of the rod; $A$ – runout of rod; $L$ – distance between rod bearings; $E$ - modulus of elasticity; $J$ - moment of inertia.

In dynamic actuators the angle $\theta$ is sufficiently large because the lateral force is great. Therefore the formula where specific pressure is calculated as quotient from lateral force upon a bush to projection area is not correct. In this case the real contact area has to be taken into account. But calculations with Hertz contact formulas are not correct either because many simplifications are to be used and mainly because guide sleeves wear, which is the most intensive in the beginning, is not taken into account. On the other hand, as a rule lateral force loads actuator rod in rig test in one plane.

These are the points of the analytic technique presented in the paper:

1. Specific pressure is calculated according to projection of real contact area when the bush wear starts.
2. The value of specific pressures at steady-state wear must not be more than allowable one.
3. The initial allowable bush wear at which the specific pressure is of allowable value is preset.

In the beginning of actuator work, when the lateral force is being loaded, the contact between rod and bush is going on at the bush edge. But after some time hard chrome rod makes an indent in the bush. The contact area becomes sufficient and contact pressure is of acceptable value (Figure 2).

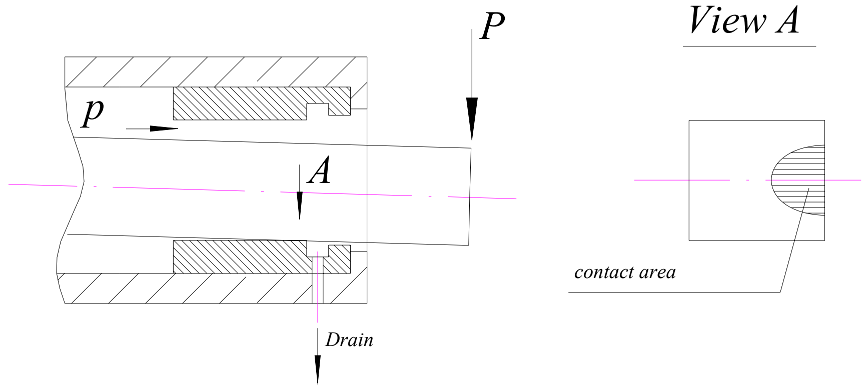


*Figure 2. Contact within rod and barrel*

Permissible level of pressure is selected as the pressure 2 kg/mm$^2$, because in this case a couple chromium-plated steel and a bush can work for a long time [3].

The initial permissible wear is chosen as 0.005 mm. This wear can not be measured by the ordinary means, and this wear indent is like a trace after friction contact.

Contact area is determined as embedding area of a cylinder into a cylinder with slightly greater diameter. The stress and strain by Hertz can be neglected as second-order terms.

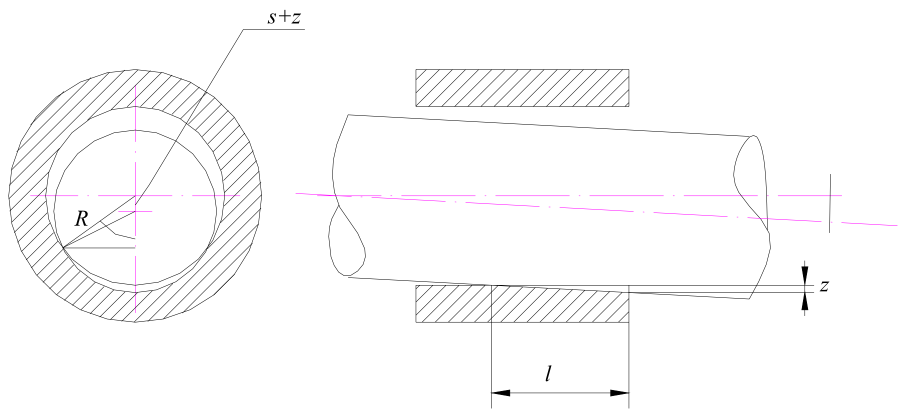


*Figure 3. Calculation of rod actuator bearing*

Contact area can be calculated on the basis of geometrical constructions in figure 3.

$$F = 2\int_0^l R\cos\varphi dx \tag{3}$$

After transformation into a row and rejecting of the second-order numbers we can derive the formula of contact projection area:

$$F = \frac{2lRs}{z}\left(\frac{\sqrt{z^2+2zs}}{s} - Arc\cos\left(\frac{s}{z+s}\right)\right) \tag{4}$$

In these formulas: $F$ - contact projection area; $R$ – radius of bearing bush; $\varphi$ – polar angle contact border; $l$ – contact length along the bush axis; $s$ – radial clearance; $z$ – depth of embedding on the bush edge.

The angle $\theta$ of rod decline into bush may be calculated by means of formula (2) and contrariwise $\theta=z/l$ (figure 3) we can calculate contact project area between the rod and bearing bush.

In Figure (4) one can see the relation between the depth of embedding on the bush edge for providing the initial contact area and the rod diameter to provide the initial pressure 2 kg/mm$^2$. Diagram is made for the lateral force 150 kg upon the short stroke actuator rod.


*Figure 4. Relation of embedding depth of the bush edge to the rod diameter*

The diagram shows that short stroke actuator with nominal force 30…50 kN must have rod diameter not less than 35…40 mm, if the lateral force is loading the rod, of course.

While wearing the contact project area grows and contact pressure decreases so that hydraulic actuator durability increases. In figure 5 we show analytic diagram of contact project area relationship to the depth of the bush edge embedding for example. An actuator has following parameters: nominal force – 50 kN, rod diameter – 40 mm, distance between point of loading lateral force equal 5% from nominal actuator force – 200 mm, distance between bearings – 120 mm, bearing length – 30 mm.

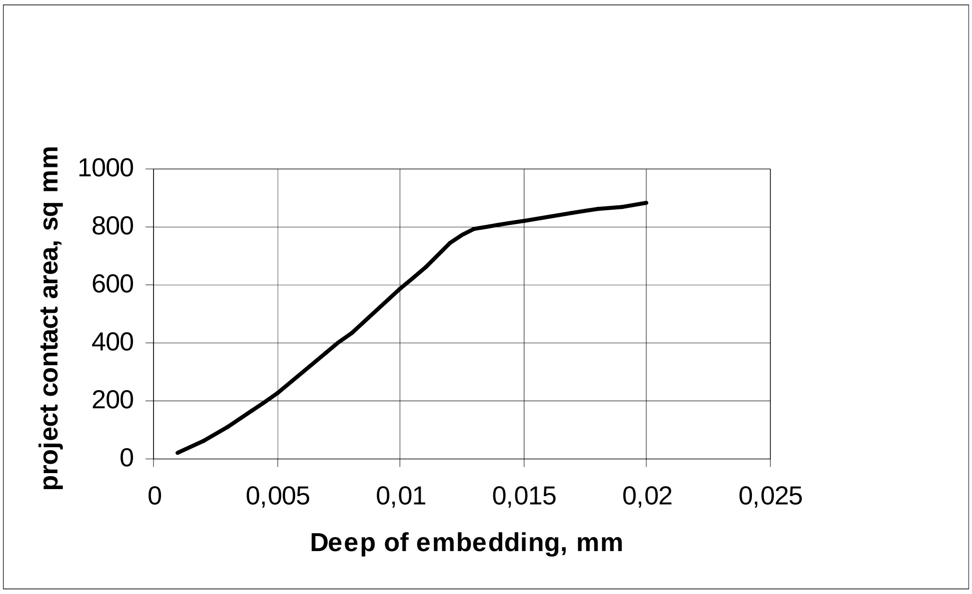


*Figure 5. Calculate diagram project contact area relationship on deep of embedding*

While the depth of embedding into the bush edge becomes 0.005 mm project contact area becomes equal 200 mm$^2$ and pressure becomes equal 2 kg/mm$^2$. While wear is going on contact project area grows to 800 mm$^2$ and pressure decreases to 0.5 kg/mm$^2$. As the process continues contact area length becomes equal to the bush length and liquid fluid grows from work chamber to drain. Besides some risk arises that piston-rod would touch the hydro cylinder body. Therefore the contact wear depth should not be more than 0.013 mm.

Experimental check of presented method has been made by checking hydraulic actuators after its long operation in dynamic test rigs. The hydraulic actuators manufactured by SHENCK and Moscow Helicopter plant were examined. All cylinders had operated about 3…7 thousands hours. During the survey it was found out that SCHENCK hydro actuators were in rather bad condition: wear contact deep was more than 0.02 mm, wear area came out of the bush limits. An analysis confirmed actuators bushes had been overloaded. Actuators rod diameter was only 25 mm and starting pressure was more then 5 kg/mm$^2$. The actuators of Moscow Helicopter plant had rod diameters 40…60 mm, start pressure less than 2 kg/mm$^2$ and their state was good.

## 2. ROD PISTON OF DYNAMIC ACTUATOR

The main requirements to rod piston of dynamic actuator are the absence of friction, minimum sizes, minimum flow-over between the chambers. If there is no piston sealing it is required to choose the clearance between the piston and actuator's body and piston length.

### 2.1. Clearance piston – actuators body

Contact between piston and cylinder body inboard surface is excluded from any normal lateral loadings. To avoid contact the following factors are taken into account.

1. The possible rod subsidence in bush bearings;
2. Deviations within bushes and inboard body surface;
3. The rod bend;
4. Guaranteed margin (reserve).

The rod subsidence in bush bearings can achieve a half of diameter clearance in bush bearings. Ordinary one equals 0.02…0.03 mm.

Deviations are defined 0.02 mm when the design and technology process are correct. That means fastening bearing cover to actuator body by pins, boring bearings on boring machine in one set without turn machine bench, correct positioning tool ware by inboard surface cylinders body; if necessary finish boring of inboard cylinder surface.

Rod bend is calculated as the bend of the beam on two supports. The rod lateral displacement between bearings is:

$$\upsilon = \frac{P}{6EJ_x}\left[alz - \frac{az^3}{l}\right] \tag{5}$$

where $\upsilon$ – rod lateral displacement; $P$ – lateral force on the end of rod; $E$ - modulus of elasticity; $J_x$ – moment of inertia; $a$ – distance from upper bearing to the rod end; $l$ – distance between bearings; $z$ – coordinate.

The maximum rod displacement from the lateral force is:

$$\upsilon_{\max\ Shtok} = \frac{\sqrt{3}\,Pal^2}{27\,EJ_x} \quad \text{when} \quad z = \frac{l}{\sqrt{3}} \tag{6}$$

But maximum rod lateral displacement is not always equal to the maximum piston lateral displacement. In the case of the rod moving from abutment to abutment, assuming that $a = z$ and defining maximum by the first derivative, the maximum piston lateral displacement yields:

$$\upsilon_{\max Porsh} = \frac{Pl^3}{24EJ_x} \quad \text{when} \quad z = \frac{l}{\sqrt{2}} \tag{7}$$

Calculations showed that the typical value of lateral displacement is 0.015…0.030 mm under lateral force on the rod end 5% from nominal actuator force. The commonly encountered result is 0.02 mm. It is true for ordinary short-stroke actuators. The only method to reduce piston lateral displacement from the bend is to enlarge the rod diameter.

The margin (Reserve) depends upon the proper manufacturing in the strict accordance to technology, personal skills of manufacturers and the terms of delivery. When the terms are short and definite, the mistakes are inadmissible, and the margin (reserve) is assumed to be not less than 0.02 mm.

Having summarized all the assumptions one can conclude that the radial clearance $s$ must be:

$$s = \Delta_{\text{subsidence}} + \Delta_{\text{deviations}} + \Delta_{\text{rod bend}} + \Delta_{\text{reserve}} \tag{8}$$

The typical short-stroke cylinders have s = 0.08…0.12 mm.

### 2.2. Piston length

Piston length is calculated from tolerable flow-over between chambers. The pressure cylinder body deformation and nominal servo valve flow are taken into account.

From field experience we know that the majority of hydraulic actuators are applied for loading 50…60% of nominal force and the pressure difference in chambers is about 60% of supply pressure. When the force on the rod is not applied, in chambers pressure becomes about 60% of supply pressure. As a rule, nominal flow servo valves are declared when pressure loss on two edges is 70 kg/cm$^2$. Therefore the main inside cylinders body pressure is defined as 120 kg/cm$^2$ and ordinary pressure difference between chambers is 120 kg/cm$^2$ (supply pressure normally equals 200… 210 kg/cm$^2$). Analysis of many cylinder types showed that typical flow-over between chambers is 5…10% of nominal flow servo valves when the loading control is good.

Piston length is calculated from

$$L = 8.18 \cdot 10^{6} \cdot \frac{1}{\nu\gamma} \cdot \frac{\pi D s_p^{3}}{q} \cdot p \qquad (9)$$

$$s_p = s + u_p \qquad (10)$$

where $L$ – piston length; $\nu$ - dynamic viscosity; $\gamma$ - relative density; D – piston diameter; s - radial clearance by formula (8); $q$ - tolerable flow-over; $p$ – difference chamber pressure; $u_p$ – increase of inside surface radius. The one is calculated by formulas of thick-wall vessel under pressure:

$$u_p = \frac{p}{E \cdot (D_n^2 - D_v^2)} \cdot ((1 - 2\mu) D_v^3 - (1 + \mu) D_v^2 D_n) \qquad (11)$$

where: $p$ – cylinder middle pressure; $E$ - body modulus of elasticity; $\mu$ – Poissonian coefficient; $D_v$ - body inner diameter; $D_n$ – body external diameter.

While using servo valve with nominal flow 60 l/min the piston length must be from 25 to 100 mm corresponding with nominal force and rod work stroke.

**OUTPUT ROD SEALING**

The sealing friction force can significantly decrease the regulation quality. It is especially noticeable in actuators with nominal force up to 10 kN. To reduce friction forces in dynamic actuators we used output sealing unloaded from pressure by means of drain groove in front of the sealing. Because of this groove the oil flows through the rod bearing which improves its operating conditions. Lower rod end has no sealing. So-called weep casing is made. When the rod moves down, oil flows through back-flow prevention valve from casing. To avoid evacuation the second back-flow prevention valve was mounted in the casing. It let the air flow into casing while the rod moves upwards (Figure 6).

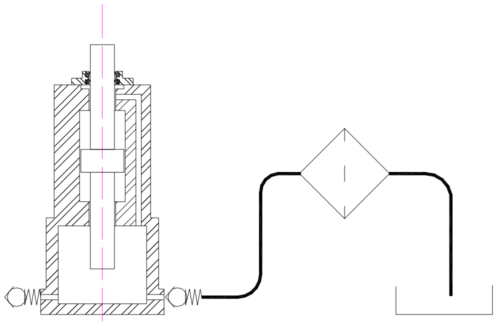

*Figure 6. “Weep causing” scheme*

As a result cheap long-lived small-sized short-stroke hydraulic actuators were designed and successfully used. Using hydraulic actuators cheap small-sized dynamic test rigs were designed. In Figure 7 our typical short-stroke hydraulic actuator is shown. The appearance of one of them is shown in Figure 8.

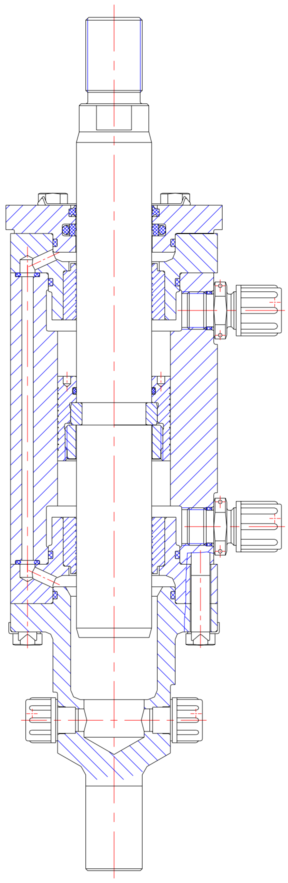

*Figure 7. Short-stroke hydraulic actuator 50 KN*60 mm*

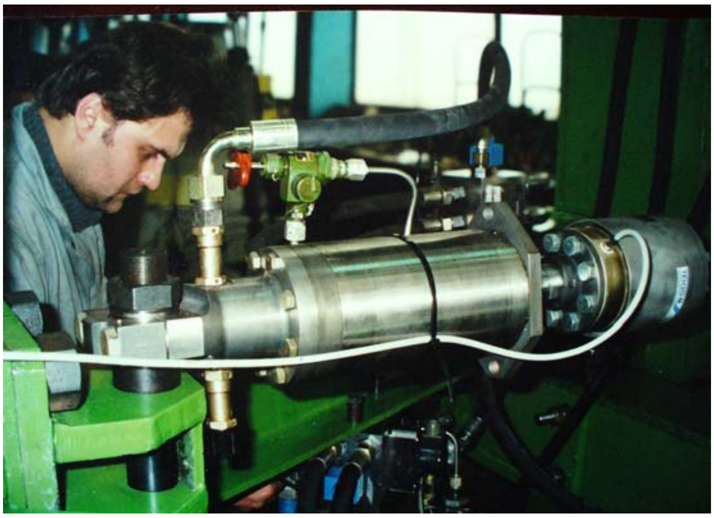

*Figure 8. Short-stroke actuator appearance*

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