LIFETIME OF SPHERICAL PNEUMOHYDRAULICS ACCUMULATORS

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Abstract: In the present paper is presented a theoretical research about the lifetime determination of pneumohydraulics accumulators under fatigue conditions. Spherical accumulators were analyzed because, as is known, they have indisputable advantages towards the cylindrical accumulators. The analysis presented is based on the methods of fracture mechanics of materials, considering that in the material structure were detected cracks with different shapes and sizes. The proposed method is illustrated with an example of calculation applied to a spherical shape accumulator.

Keywords: Lifetime, fatigue, crack, fracture mechanics of materials.

1. INTRODUCTION

The pneumohydraulics accumulators are essential components of hydraulics systems. Without these elements would not be possible to operate some basic equipment from different fields of activity: hydraulic presses, hydraulic hammers, die-casting machines etc.

Their particular importance in a hydraulic installation resulting from the accomplished functions: the accumulation of hydraulic power during periods when the system is not working at rated capacity, to provide backup power during periods of maximum demand of system; the elimination of shocks from pipes of hydraulic system; amortization of pressure pulsation; compensation of fluid losses from system; offsetting changes in the volume of fluid in systems operating at extreme temperature variations. In the fast automatic hydraulic systems requiring a big precision in transmission of commands, the use of accumulators is obligatory. Accumulators are provided with fast-action devices for coupling and decoupling from the system at a certain level of disturbance. Typically, the accumulators working pressure is between 200 and 300 bar, and their capacities are up to 50 liters.

Usually, for technological reasons, batteries are made from a cylindrical vessel. Inside of this vessel is a mobil partition wall (membrane or balloon) to create two rooms with variable volume (Figure 1) [1]. On one side of this mobil wall work the force generated by a liquid pressure (oil), and on the other side, work the force developed by a gas (air, nitrogen, etc.) pressure. The gas pressure is determined by the mobil partition wall position, respectively, the liquid pressure in the opposite room. Beside of cylindrical accumulators with membrane are used and spherical accumulators (Figure 1.b). They have the advantage, compared to the cylindrical, that at the same pressure and same diameter, in the spherical body is developing two times smaller stresses. Because of this, the weight of spherical accumulators is less than weight of cylindrical accumulators, for the same capacity and the same internal pressure.

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About the mechanical calculation of these devices, two aspects should be retained: high operating pressure, leading to the adoption of special methods of manufacture; changes in time of pressure, which requires verification of endurance through a calculation of fatigue.

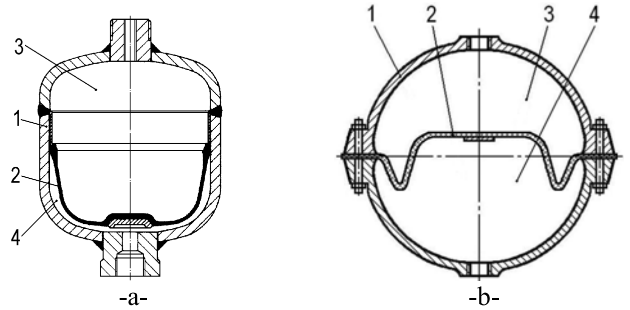

Fig. 1. Pneumohydraulics accumulators with membrane:
a - cylindrical accumulator; b - spherical accumulator;
1 - body; 2 - membrane; 3 - gas room; 4 - oil room.

On the other hand, taking into account the pretentious manufacturing technology and high price, is necessary to determine the remaining lifetime if in the structure of the material have ben discovered certain defects (for example, cracks). Of course, the adoption of such decisions can not be done only on the basis of clear methods of calculation, which must to take into account the duration of operation, the history of requirements and the effect of chemical aggression of the working environment on the accumulatror's body.

2. GENERAL CONCEPTS ON THE CALCULATION TO FATIGUE

The design of pneumohydraulics accumulators must consider, firstly, the requirement at variable pressure. This type of requirement can be tackled both in polycyclic domain, when the number of cycles is greater than 10^5, and in oligocyclic domain, when the number of cycles is less than 10^5. The majotity of machine's element are required in the polycyclic fatigue domain, while the process equipment, for instance the pressure vessels, are required in oligocyclic domain. As well, in the oligociclic domain (small number of cycles) falls and many other types of structures, such as spacecraft and aircraft cabins, fuel tanks of space ships, submarine's body etc. As required, the calculation of oligocyclic fatigue can be developed according to one of following two forms: verification calculation - for mechanical structures which are already staticaly sized, tacking into account the maximum stress of working cycle; sizing calculation - when is necessary to design a new structure.

It should also be noted that a fatigue calculation for a vessel type structure is carried out in certain areas, namely, those where action the maximum stresses. With these stresses is determined the number of cycles to failure of structures. On the other hand, must take into account that the wall of a pressure vessel may have different types of defects (pores, cracks, particles of slag inclusions etc.), which occur during the manufacturing process. Thus, because the phenomenon of fatigue of the construction material, these defects can grow until they reach a critical length; at this time occurs the rupture of material and equipment out of service. It also must not be lost sight of other influences, such as temperature effects and changes in temperature, residual stress action, working environment aggressiveness etc. Under these conditions appear necessary to corroborate these effects, in order to estimate by means of a suitable calculation method, the actual state of stress in the pressure vessel wall and its lifetime.

In the speciality literature there are presented many ways to solve the problem on the fatigue resistance determination. Among these may be mentioned: the use of materials fracture mechanics concepts; using analytical expression of the amplitude at breaking of materials [2, 3]; using relation between total deformation amplitude and number of cycles to failure, in areas of stress concentrators; using curve Wöhler [4]; using the "critical energy" priciple [5]. Further on, will be presented the stages of fatigue calculation of spherical hydropneumatics accumulators, using concepts of fracture mechanics of materials.

3. DETERMINATION OF ACCUMULATOR'S LIFETIME

Determination of accumulator's lifetime is a fatigue verification calculation. This calculation is applied in zones where stresses have maximum values, for example in the zones of welding joints or in the zones of orifices for pipe connection. Because in these zones are usually different types of defects in material structure, the lifetime determination of the accumulators can be made via the method that is based on the concepts of fracture mechanics of materials.

The application of this method assumes that the following facts are known: the material construction characteristics, parameters and working regime, history of requirements, type and location of defects in material structure. To apply the concepts of fracture mechanics of materials in the case of these structures, is the assumption that the critical crack depth is less than 50% of vessel wall thickness.

3.1. State of stress determination in the zone which is verified to fatigue

In this stage are determined the maximum stress values developed in the body of the accumulator under the action of inner pressure. For this, the pneumohydraulics accumulators can be considered into the category of thin-walled vessels or thick-walled vessels category [6]. In the first case, the ratio between the outer accumulator's diameter, d_e , and the inner accumulator's diameter, d_i , satisfies condition $d_e/d_i \leq 1.2$, and to determine the status of stress must be to apply the well-known relation from the shell's theory:

$$\sigma_1 = \sigma_2 = \frac{p_i \cdot d_m}{4 \cdot s} \tag{1}$$

In previous relationship σ_1 and σ_2 are, respectively, axial and circumferential stresses; p_i - maximum inner pressure; d_m - average diameter of shell; s - thickness of shell. In the case of thick-walled vessels, the ratio between inner and outer diameters of the battery satisfies the condition $d_e/d_i > 1.2$, and to determine the status of stresses must use the relations:

$$\sigma_1 = \sigma_2 = p_i \cdot \frac{0.5 \cdot \beta_r^3 + 1}{\beta^3 - 1}; \; \sigma_3 = -p_i \cdot \frac{\beta_r^3 - 1}{\beta^3 - 1}; \; \sigma_{ech} = 1.5 \cdot p_i \cdot \frac{\beta_r^3}{\beta^3 - 1} \tag{2}$$

In the previous relations σ_1, σ_2, σ_3 and σ_{ech} designate, respectively, axial, circumferential, radial and equivalent stresses; $\beta = r_e/r_i$; $\beta_r = r_e/r$; r_e , r_i , r - respectively, external, internal and commonly radius of thick-walled spherical body. If the body of accumulator has zones with stresses concentrators, must be calculated the maximum stresses in these zones with relation:

$$\sigma_{\max} = \sigma^{(p)} \cdot \alpha_k \tag{3}$$

in which:

$\sigma^{(p)}$ is the stress value calculated with one of previous relations;
α_k - stress concentration factor.

The establish of final state of stresses is obtained by superimposing stresses caused by inner pressure, temperature, transitional and residual stresses resulting from technological processes (rolling, forging, welding etc.).

3.2. The relation between the state of stress and critical crack length

Because in the structure of an equipment can be discovered (with help of non-destructive tests), cracks with various shapes and sizes, it is interesting to see which of these cracks are more dangerous, for the same status of stress. Usually, the cracks occur at the inner surface of vessels, in the heat affected zones of welds. In the case of elastic requirements, to establish the liaison between the critical crack length and the maximum stress which determine that crack length is used the relation of stress intensity factor [7, 8, 9]:

$$K = \sigma \cdot \sqrt{\pi \cdot f \cdot a} \tag{4}$$

in this relation:

σ is the tension from the material with cracks;

a – the length of crack;

f – shape factor of crack, which depend of ratio between the length of crack and the dimension of object with crack.

Knowing the type and size of crack, from the Relation (4) obtain the expression of critical crack length:

$$a_{cr} = \frac{1}{\pi \cdot f}\left(\frac{K_C}{\sigma_{cr}}\right)^2 \tag{5}$$

where:

K_C is fracture toughness of the material;

σ_{cr} - critical stress of building material; $\sigma_{cr} \leq \sigma_c$, because the requirements are in elastic domain;

σ_c - flow limit.

In this paper were studied the influence of maximum stress on the critical crack length, in the case of carbon steel, having three types of cracks: internal circular crack, internal elliptical crack and surface semi-elliptical crack. Research results are presented in Figure 2.

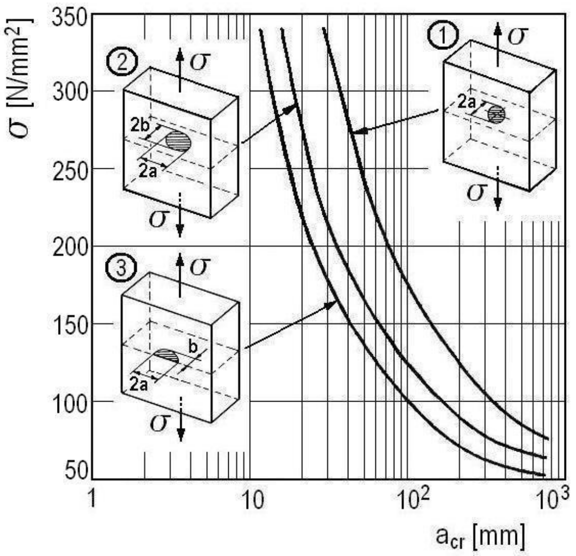

Fig. 2. The dependence between the critical length of crack and the stress whereat is set off the fracture, for carbon steel:
1- internal circular crack; 2- internal elliptical crack; 3- surface semi-elliptical crack.

It appears that in all cases, increasing the crack length corresponds to a decrease in maximum stress which cause the breaking. At the same time, it is apparent that the most dangerous structural defects is surface semi-elliptical crack, because critical length of this crack has the lowest value to the same value of maximum stress. This means that the results obtained in this case are covers for other types of cracks (internal circular crack and elliptical internal crack). Surface semi-elliptical crack is characterized by form factor:

$$f = \frac{1.2}{\Phi^2} \tag{6}$$

in which Φ is the elliptically integral of the second order:

$$\Phi = \int_0^{\frac{\pi}{2}} \sqrt{1 - \frac{a^2 - b^2}{a^2} \sin^2 \varphi \cdot \mathrm{d}\varphi} \tag{7}$$

For sizes of cracks which fulfill the conditions *a/b* = 4 and $\varphi=\pi/2$, value of elliptically integral is Φ=1.15. Because in practice it can meet differently concrete situations of work, has been calculate the critical length of semi-elliptical crack for different values of ratio *a/b*, as shown in Figure 3.

Calculation of crack propagation aims at quantitative assessment of the development of cracks which appear in the studied metallically structure, during the term of service. In this way is possible to avoid the risk of sudden fracture (catastrophic). To this end, knowing the critical crack length, a_{cr}, from which material breakage may occur, to prevent rupture of the material must determine the admissible length of the crack, with the relation:

$$a_{ad} = \frac{a_{cr}}{C_s} \tag{8}$$

in which the safety factor C_S= 3 ÷ 4.

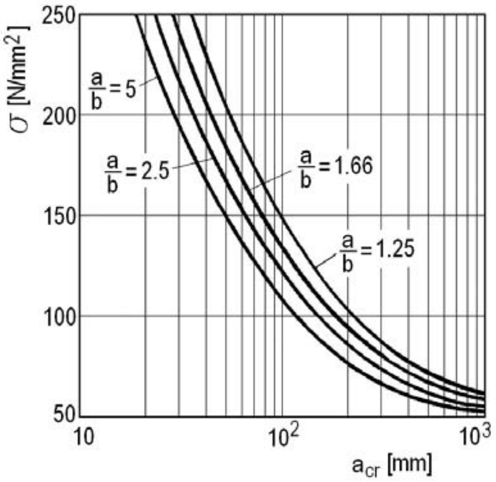

Fig. 3. The dependence between the critical length of the crack and the stress whereat is set off the material fracture, in the case of surface semi-elliptical crack.

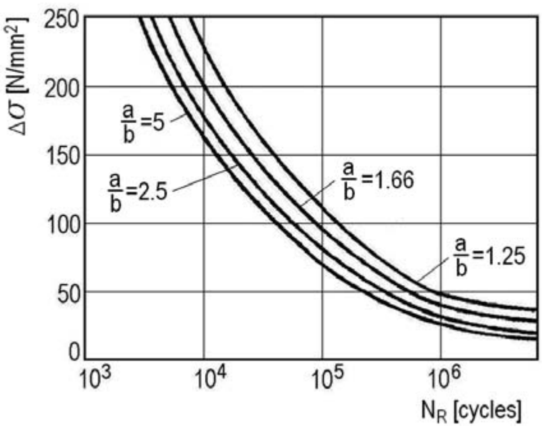

Fig. 4. The dependence between stress and number of work cycles, for a surface semi-elliptical crack with diferent values of ratio *a/b*.

3.3. Number of cycles to failure determination

After establish of critical length of crack, can proceed to determine the number of cycles to failure, from the Paris equation [8, 9]:

$$\frac{\mathrm{d}a}{\mathrm{d}N} = \beta \cdot (\Delta K)^m \tag{9}$$

in which:

$$\Delta K = K_{\max} - K_{\min} = (\sigma_{\max} - \sigma_{\min}) \cdot \sqrt{\pi \cdot a \cdot f} = \Delta\sigma_{ef} \cdot \sqrt{\pi \cdot a \cdot f} \tag{10}$$

By integrating Relation (9) is obtained the equation with which one can calculate the number of cycles to failure:

$$N_R = \frac{a_{cr}^{1-\frac{m}{2}} - a_i^{1-\frac{m}{2}}}{\beta \cdot \left(1-\frac{m}{2}\right) \cdot (\Delta\sigma)^m \cdot \pi^{m/2}} \tag{11}$$

in which:

a_i is the initially lengths of crack;

β and m – coefficients depending of material (m = 3; β = 3.492⊙10^{-12});

$\Delta\sigma = \sigma_{max} - \sigma_{min}$.

In Figure 4 are presented the results of calculations to determine the number of cycles to failure, in the case of surface semi-elliptical crack, for different values of ratio *a/b*. Further, knowing the number of cycles to failure, can determine the number of admissible operating cycles, N_{ad} and lifetime, D, for the analyzed structure:

$$N_{ad} = \frac{N_R}{C_N}, \quad D = N_{ad} \cdot \frac{z}{8760} \tag{12}$$

In previous relations C_N is the endurance safety coefficient (C_N =10) and z - frequency of cycles.

4. APPLICATION

The pressure casting equipment used in an enterprise for manufacture of fittings and sanitary facilities, include a spherical pneumohydraulics accumulator, with an outer diameter of 0.300 m and thickness of wall 0.010 m. The accumulator was made by welding two hemispheres from steel with the following mechanical characteristics: breaking limit σ_r = 510 N/mm^2, flow limit σ_c = 350 N/mm^2, modulus of longitudinal elasticity E = 2.12·10^5 N/mm^2; Poisson's coefficient μ =0.3; fracture toughness of material K_C =100 MPa·m$^{0.5}$. After making the accumulator on its inner surface, in the heat affected zone of the weld, surface semi-ellipsoidal cracks were discovered, with lengths ranging from 1 to 3 mm, characterized by a ratio *a/b*=5. Maximum working pressure of the accumulator is 20 MPa, and operating temperature is below 100 ^{0}C. Knowing that the accumulator will be required to fatigue after a pulsating cycle with a frequency z = 6 cycles/hour, is necessary to determine the accumulator's lifetime.

To apply the method based on the concepts of fracture mechanics of materials, first is necessary to determine the maximum stress values developed in the body of the accumulator under the action of inner pressure. The state of stress is calculated with Relation (1), because the value of ratio d_e/d_i < 1.2. Also, is determined the maximum stress value by applying Relation (3), assuming a stress concentration factor α_k= 1.5 in the zone of orifices for pipe connection. Next, using Equation (5) or the curves presented in Figure 3, is determined the critical crack length $a_{cr} \approx$ 0.28 mm.

Knowing the critical crack length can proceed to determine the number of cycles to failure from the relationship (11) or using the curves from Figure 4. Thus, under this application conditions, using the Relations (12), results that the number of cycles to failure is 3.324·10^6 cycles and the allowable number of cycles is 332,400 cycles. Because the frequency of cycles is z = 6 cycles/hour (1cycle/10 minutes), finally results that the accumulator's lifetime is approximately six years.

5. CONCLUSIONS

In the present paper proposes a method for calculating the lifetime of pneumohydraulics accumulators. In this calculation took into account the requirement of fatigue. But, in the same time, is considered the possibility that in the wall of these vessels can by different types of defects (pores, cracks, particles of slag inclusions etc.), which occur during the manufacturing process or during the operation.

Thus, because the phenomenon of fatigue of the material of construction, these defects may develop, causing breakage of material and final decommissioning of equipment. The method is based on the principles of fracture mechanics of materials and can be used to design pressure vessels. It determines the allowable number of cycles, and safe operating lifetime of pressure vessels required to fatigue.

Knowing the characteristics of building material and the history of requirements (duration of the previous cycles, frequency and amplitude of cycles, types and geometry of cracks) can determine the duration of remaining lifetime of equipments already in operation.

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