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OPTIMUM FORM OF THE HYDRAULIC CYLINDER ROD CROSS-SECTION

In the article optimum form of the hydraulic cylinder rod cross-section is suggested for decreasing stresses.

A hydraulic cylinder rod is known to experience bending strain besides compressive strain. In this case the diagram of normal stress distribution along cross-section will be asymmetrical. Thus for decreasing stresses in section we should admit more rational asymmetrical cross-section with reference to one of its main principal axis of gravity. A tee-form section is a classical example of the above-mentioned sections, and at bending strain it is symmetrical H-beam section. Unfortunately, the application of sections of these types for hydrocylinder rod is ineffective now from the point of view of tightness, which is attributed to irrational forms of mating seals surfaces. The idea is the following - by leaving the form of rod cross-section round it is quite possible to get the optimum section by certain interior material grooves or, in other words, to transform the existing classical sections qualitatively in the specific unknown quantity corresponding to it quantitatively.

It is known that reliability of machinery hydraulic cylinders in supporting power on many occasions is restricted by rod strength due to the condition of its trouble-free state in dangerous section with x position is described by non-excess or "supporting power - load"

$$\sigma_i(x) \le [\sigma]_i. \qquad (1)$$

Here the current operational stresses $\sigma_i(x)$ play the role of load and admitted ones $[\sigma]_i$ those of supporting power. It is not a secret either that rod destruction (advent of permanent) on many occasions occurs for reasons of irreversible changes in microstructure of their material as a result of cyclical loading by loads of variable sign, that is due to fatigue breakdown. With sufficient degree of authenticity negative stresses σ_{min} of compressions arising in its dangerous section are determined by a formula

$$\sigma_{min}(x) = -\{P_i/F + [M_Q(x) \pm M_{Ri}(x) + P_i y_T(x) + P_i e_i(x)\ ]/W\} \qquad (2)$$

and positive $\sigma_{max}(x)$ stretching - from expression

$$\sigma_{max}(x)=P_j/F-[M_Q(x)\pm M_{Rj}(x)-P_j e_j(x)]/W, \quad (3)$$

where: $P_{i,j}$ is hydrocylinder longitudinal pushing (compressing), drawing (extending) effort; $M_Q(x)$ is bending moment from lateral load (hydraulic cylinder weight); $M_{Ri,j}(x)$ is frictional moment in hydrocylinder radial bearings, attributed to kinematics of machine hydraulic drive and by $P_{i,j}$ effort action; F is rod cross section area; W is axial torque of rod section resistance; $y_T(x)$ is hydraulic cylinder full deflection as a result of its operational longitudinal-transversal loading; $e_{i,j}(x)$ is eccentricity of application of longitudinal compressing effort $P_{i,j}$ in hydraulic cylinder supports.

Since on most occasions of hydrocylinder utilization $P_i>P_j$, it is clear that $|\sigma_{min}(x)|>|\sigma_{max}(x)|$, that is the cycle of rods loading is evidently asymmetrical with negative average stresse $\sigma_m(x)$ equal to

$$\sigma_m(x)=[\sigma_{max}(x)+\sigma_{min}(x)]/2. \quad (4)$$

Taking into account the connection of amplitude stresses $\sigma_a(x)$ characterized by relationship

$$\sigma_a=[|\sigma_{max}(x)|+|\sigma_{min}(x)]/2, \quad (5)$$

with σ_{-1} - the limit of endurance and by stresses $\sigma_{max}(x)$ and $\sigma_{max}(x)$, in the final form the condition (1) of the rod trouble-free state can be represented as

$$\sigma_i(x)=\sigma_a(x)+\varphi_s\sigma_m(x)\leq[\sigma]=\sigma_{-1}/k_{si}, \quad (6)$$

where: φ_s is the coefficient of rod material sensitivity to asymmetry; k_{si} is the ultimate factor of safety. When the stress $\sigma_a(x)<0$, coefficient φ_s should be set equal to 0, that is the expression (6) in view of formula (5) is expressed somewhat simpler

$$[|\sigma_{max}(x)|+|\sigma_{min}(x)|]/2\leq\sigma_{-1}/k_{si}. \quad (7)$$

The analysis of relationships (2) and (3) shows that the only variable in them is the value of the full hydrocylinder deflection $y_T(x)$, and this means that compression stresses $\sigma_{min}(x)=var$, and stretching stresses $\sigma_{max}(x)=const$. That is in the final form the condition of hydraulic cylinder trouble-free state can be denoted by relationship

$$\sigma_i(x)=|\sigma_{min}(x)|\leq[\sigma]_i=2\sigma_{-1}k_{si}^{-1}-\sigma_{max}(x). \quad (8)$$

In other words as a result of exterior actions on hydrocylinder and interior functional interactions of its elements current operational stresses $\sigma_i(x)$ are not constant in time, but continuously increase reaching limiting values with accumulation of defects. In this case the function $\sigma_i(x)=f(t)$ can illustrate the hydraulic cylinder evolution in supporting power (Fig. 1). Now we will make it clear.

The hydraulic cylinder operation stars with $\sigma_1(x)$ stresses and continues as a rule till the meaning $[\sigma]_1$, later parametric conventional failure follows. Its further application is fraught with a risk of full, often evident failure arising behind the σ_{-1} limit, that is why it is expedient to restrict admitted hydraulic cylinder resource in stresses by the difference $\{[\sigma]_1-\sigma_1(x)\}$.

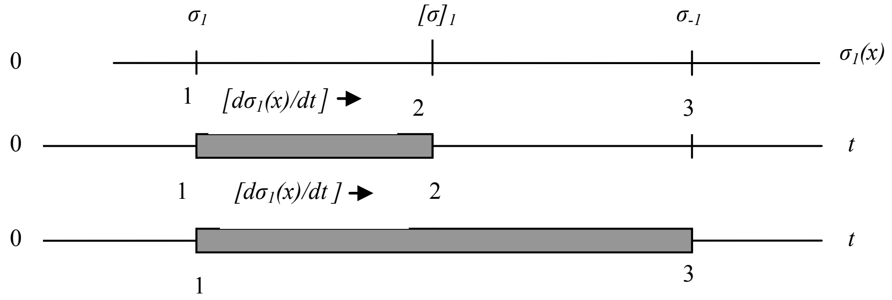

Fig. 1. The scheme of hydrocylinder evolution in the process of operation

Hence in case where some average speed $d\sigma_i(x)/dt$ of stress accumulation is known it is not difficult to determine both admitted

$$t_{1-2} = \{[\sigma]_1 - \sigma_1(x)\}/(d\sigma_1(x)/dt), \tag{9}$$

and maximum permissible service life of hydrocylinder

$$t_{1-3} = \{\sigma_{-1} - \sigma_1(x)\}/(d\sigma_1(x)/dt). \tag{10}$$

From the above-mentioned it follows the hydraulic cylinder modernization in the direction in question may aim at either decreasing starting stresses $\sigma_1(x)$ (Fig. 2a) or increasing the admotted ones $[\sigma]_i$ (Fig. 2b), or reducing velocity $d\sigma_i(x)/dt$ of their accumulation. In the ideal case it is expedient to apply all the mentioned arrangements as a complex (Fig. 2c).

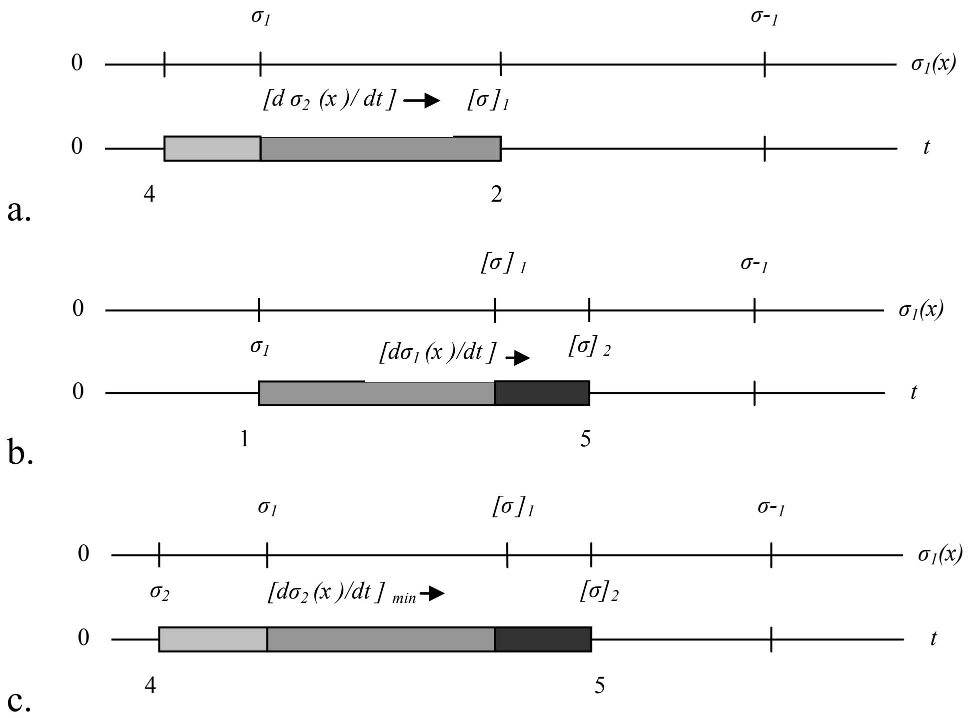

Fig. 2. The scheme of modernized hydrocylinder evolution

It being known that if the first and the last suggestions have their specific engineering applications, the second one is frequently achieved by increasing accuracy of evaluation of k_{si} hydraulic cylinder rod ultimate factor of safety with the other known conditions for accepted level of authenticity according to the formula

$$k_{si}=(1-\omega_{-1}{}^2\Lambda^2)^{-1}+\{(1-\omega_{-1}{}^2\Lambda^2)^{-1}[(1-\omega_{-1}{}^2\Lambda^2)^{-1}-\{(1-\omega_a{}^2\Lambda^2)\}^{½} \quad (11)$$

with the obliged satisfaction of condition

$$k_{si}>\sigma_{-1}/(\sigma_{-1}-\mu_{-1}\Lambda), \quad (12)$$

where: ω_{-1} ω_a are variation factors of endurance limit σ_{-1} and $\sigma_a(x)$ amplitude stresses limit, respectively; Λ is quantile of normal distribution; μ_{-1} is average square deviation of endurance limit σ_{-1} .

Let us consider as an example the most interesting version of hydraulic cylinder complex modernization (Fig. 2b), comparing it with the assumed basic one (Fig. 1). So the same normal stresses arise at tensile and compressive strains at any point of rod cross-section of ring section, and at bending strain normal stresses arise increasing with ranging from neutral axis. It is evident that stress epure for symmetrical cross-sections is symmetrical relative to the main axis of gravity. In the case where normal force and bending moment are present at cross-section the total epure of stresses is asymmetrical relative to the neutral line, that is rod material at section is distributed irrationally. It points to the necessary of applying in similar cases asymmetrical cross-section relative to one of its main axes of gravity. A classic example of such section is a tee-section for which in case of bending with tension and compression it is possible to select dimensions satisfying the condition of equal strength both for stretched fibres and fibres under compression. However, from the point of view of tightness it is doubtful that hydraulic cylinder rod might be of such cross-section. Nevertheless we'll try to get ideal section from ring section close to the tee-section as to its geometrical characteristics. To solve the given problem of tee „transformation" we'll formulate the main requirements imposed for the promising section for the purpose of satisfaction of supporting and sealing power of a modernized hydrocylinder. Firstly, the moment of resistance for the most loaded fibres should be as large as possible. Secondly, the form of cross-section from the outside should be described by smooth convex curve. Thirdly, cross-section area should not be more than that of the analogue (ring section). And lastly, an orifice or orifices inside the section may be of arbitrary shape.

Cross-section with concentric round orifice inside satisfies in full measure the above-mentioned conditions for which double effect is achieved. Firstly, moment of resistance increases for more loaded fibres with equal cross-section areas, and secondly, in cases where longitudinal compressive (extending) force is applied to with eccentricity, section centre of gravity is shifted in the direction of its decreasing up to zero.

Much more promising cross-section is round cross-section with coaxial orifice made in the form of a circle with segments cut off at the top and at the bottom, the segment with larger area being cut off from the more compressed fibres. And lastly, the synthesis of the two previous solutions makes it possible to receive much more pronounced dual effect.