Calculated and experimental diagnostics of the stress state of large-sized parts of the chemical industry machines and estimation of their longevity

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Abstract. A series of model and full-scale strain-gauge studies of the stress-strain state (SSS) of a large compressor connecting rod has been carried out. A computational model for assessing SSS is proposed, and its satisfactory agreement with the obtained experimental results is shown. The results of the SSS assessment are used to predict the resource for various operating options.

At a number of enterprises operating nitrogen-hydrogen compressors, there were cases of destruction of connecting rods [1], which creates a dangerous situation. Examination of the destroyed cars showed that all cases of breakdowns are identical and were the result of fatigue cracks in the piston heads. As a rule, the formation of cracks began from the inner surface of the head in its middle section. At the same time, experimental studies of the performance of structural elements that are large in size and work in chemical production are very difficult.

The calculation of the strength of the compressor connecting rod head contains a number of assumptions, it is influenced by the curvature of the head, the angle of sealing, the gap between the finger and the sleeve, and the design features of the heads. In this regard, it is advisable to conduct a study of the stress state of the head by experimental and computational methods using a numerical experiment in order to compare the results obtained. Taking into account that there were no surface damages arising from the action of contact stresses in the examined machines, they were not considered when assessing the probability of failure-free operation of the connecting rod. Previously, the probability of failure of elements of a mechanical system depending on the gaps was analyzed [2] and methods for evaluating mechanical systems by modelling their technical condition were considered.

Method
The method of the research was carried out on the crosshead of the connecting rod of the compressor 6CHBK-355 (piston force of 25 tons). The practice has shown that all of the incident cases of the formation and development of fatigue cracks, destructions of the heads in use were section heads coordinate 80-100$^0$ from the longitudinal axis of symmetry of the connecting rod (figure 1), indicating
the action of maximum stresses in these sections, therefore, the greatest interest is the distribution of stresses in these zones and not at the transition of the head in the beam of the rod.

Figure 1. The forces acting in the head of the connecting rod reciprocating compressor.

At the boundary of the contact and gap sections of the pin with the connecting rod bore at points A and B, the radial force NB, tangential force QB, and bending moment MV occur. Since the finger stiffness is several orders of magnitude greater than the rod head stiffness, in accordance with [3], a constant bending moment MV will act on the contact section of the VO. In this case, the curvature of the inner surface of the connecting rod head within the contact angle will be equal to the curvature of the finger. Then on the AB section (figure 1), the curvature is constant and the bending moment is [4]:

\[ M^* = \frac{E I_x (r - r^*)}{r^*} \]  

where \( E \) – elastic modulus, \( I_x \) is the moment of inertia, \( r \) is the average radius, \( \frac{1}{r^*} = \frac{1}{r_o + \frac{h}{2}} \) is the curvature of the head in the contact area, \( r_o \) the radius of the gudgeon pin, \( h \) – is the height of the section of the head.

From the condition of conjugation of sections, the bending moments are equal to \( M_B = M^* \). The moment in the current section, at the beginning of the reference angle \( \alpha \) at the border of the sections (contact angle \( 2\alpha_c \)) will be equal to:

\[ M = M^* + N_0 (1 - \cos \alpha) - Q_B r \sin \alpha \]  

Then the equation of the elastic line of the ring in accordance with [3] will have the form:

\[ \frac{d^2 W}{d \alpha^2} + W = -\frac{M}{E I_x} \]  

Substituting formula (2) in equation (3) we get

\[ \frac{d^2 W}{d \alpha^2} + W = -\frac{r^2}{E I_x} [M^* + N_B r (1 - \cos \alpha) - Q_B r \sin \alpha] \]  

where \( W \) is the radial displacement of the contact point \( W = -\frac{dv}{da} \) and \( r \) is the radius of the elastic line of the ring.

The angle of rotation of the normal \( \vartheta \) is determined from the expression [3]
\[ \theta = \frac{V'}{r} - \frac{1}{r} \frac{dW}{dx}, \]  

(5)

where \( V' \) is the circumferential offset of the ring point.

For an experimental assessment of the stress state of the connecting rod, full-scale strain-gauge tests were carried out at various operating modes of the compressor (figure 2).

The main tasks of experimental studies of the stress-strain state of the compressor connecting rods were: determining the magnitude and nature of the change in the loads acting on the crank mechanism of the compressor during its operation; determination of the nature of stress distribution in the connecting rod elements with the identification of the most loaded sections; comparison of the results of the numerical experiment with the data of experimental studies. Load cells on the connecting rod head were glued according to the scheme shown in figure 3 (a - inner surface of the connecting rod head; b - outer surface). The necessary measures were taken to protect the load cells from the corrosive environment, temperature influences and electrical interference. To exclude the effect of the arising temperature deformations, compensation strain gauges were used, which were glued to plates made of the same material as the connecting rod. The drying quality of the load cells was checked with a megometer. The places where the strain gauges were glued were covered with a lavsan film, on top of which a sealing coating based on bitumen masses was applied. After applying the coating, the insulation resistance, the integrity of the strain gauges and the absence of a short circuit to the connecting rod weight were additionally monitored. Strain gauge tests were carried out under various operating conditions of the compressor.

Figure 2. The location of the strain gauges on the inner (a) and outer (b) surfaces of the connecting rod head.

Main results
As noted earlier in [5], using the boundary conditions for \( \alpha = 0 \) and \( \alpha = \pi - \alpha^0 \), we can determine the values of the force factors \( N \) and \( Q \) as a function of the angle \( \alpha \). The stresses in the zone from the angle \( \alpha_0 \) to the angle of the transition of the head to the connecting rod were determined by numerical experiment using formula (4). At the same time, operating loads and gaps in the connecting rod – pin interface varied. The stress state of the connecting rod head was calculated for sections located within the angles from \( \alpha \) corresponding to the end of the contact zone to the angle \( \varphi = 135^\circ \). The specified loads corresponded to the operational ones: 180 kN at the nominal operating mode; 250 kN when working with overload and 100 kN when the machine is underloaded. The gaps in the connecting rod pin – connecting rod bore interface were also set in accordance with the operating parameters: the extreme
values were 0.05 mm and 0.15 mm. The average gap value was assumed to be 0.1 mm. Based on the results of analytical calculations, which varied the loads in the parts of the crank mechanism and the gaps in the coupling of the connecting rod bore-connecting rod pin, the stresses acting in the most loaded section of the connecting rod head were determined. The stress distribution on the inner surface of the connecting rod head depending on the load and the gap is shown in figure 3.

Figure 3. Stress distribution on the inner surface of the connecting rod head obtained by numerical experiment.

Calculation and experimental measurements of stresses showed acceptable agreement.

The characteristics of the connecting rod fatigue resistance are determined according to the recommendations [6, 7] and GOST 25 504 "Calculations and strength tests in mechanical engineering. Methods for calculating fatigue resistance characteristics". The connecting rod is made of 40Cr steel, for which the endurance limit based on reference data can be assumed to be equal to $\sigma_{\text{f}} = 280...290$ MPa.

For the fatigue estimation in probabilistic aspect, the methods described in [7] were used. The graph of dependence failure probability on the clearance value is shown in figure 4. It can be seen, that the failure probability is low enough.

Figure 4. Probability of failure of the connecting rod depending on the load and clearance (Values of piston forces: 1-250 kN; 2 - 180 kN; 3-100 kN).
Conclusions
The numerical experiment allows us to estimate the simultaneous effect of increase and gaps on stress growth, which is practically impossible to achieve by experimental studies. Experimental tests made it possible to determine the stresses in the connecting rod head during operation.

The stresses in the connecting rod head of an oppositional compressor determined by numerical research methods exceed the stresses determined experimentally by 7...22% and give a more conservative estimate of the stress state. The use of computational methods for determining stresses allows us to assess the impact of loads, gaps in the coupling of the pin-connecting rod head and take the necessary measures in advance to maintain the required level of probability of failure-free operation [6, 7].

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