Methods of measuring stress and strain of massive engineering designs taking into account operational factors

Dashchenko A.F.

Doctor of Engineering Science
Odesa National Polytechnic University, Odesa, Ukraine
aml28@mail.ru

Kolomiets L.V.

Doctor of Engineering Science
Odesa National Polytechnic University, Odesa, Ukraine
odivt@list.ru

Limarenko A.M.

PhD in Technical Sciences
Odesa National Polytechnic University, Odesa, Ukraine
aleksandr.limarenko@rambler.ru

Abstract
A complex method of research of heavy machinery of massive parts by the example of frame hydraulic press is considered. The analysis of the existing methods of determination of stress and strain is conducted and their shortcomings are identified. An experimental determination of the stresses that occur in the bed in a production environment is carried out. To determine stresses and strains in the metal frame, experimental and numerical research methods based on strain measurement and computer simulation are suggested. The finite element method uses mathematical apparatus of solid mechanics. Analysis of the results of numerical and physical experiments confirmed the validity of the proposed approach to the definition of stress and strain state of massive and complex geometry of heavy engineering machinery parts.
Science-based research of frame of powerful hydraulic press enables to make recommendations for optimizing powerful machine parts designs at the design stage and reduce metal consumption of these

products in the mass production of machine-building enterprises.
Keywords: NUMERICAL METHOD, STRAIN GAUGE SENSOR, MASSIVE PART, STRESS, STRAIN, GEOMETRIC SHAPE

The frame of powerful hydraulic presses have quite high levels of specific loads. The ability to create a powerful hydraulic press and to ensure its reliable operation at long operation is determined by the ability to provide strength and rigidity of body parts.

Literature on the strength and stiffness of engineering constructions is almost inexhaustible. Various laws of load distribution, the strength characteristics of the material, the stress values in the areas of concentration are considered, proposed and justified in order to establish different criteria for the limit state of bearing elements presses. Questions of press strength and stiffness are subject of a number of research and development [1].

The loads are transmitted to the press frame during pressing, and are divided into central and eccentric.

Central load corresponds to a symmetrical load on the frame axis.

Eccentric load frame is the result of displacement of the workpiece in one direction or another depending on the configuration of the molded article.

Let us consider the typical cases encountered in design practice in the selection of load circuits.

- The cylinder is supported on the frame (traverse) through its flange.

In this case, the load distributed evenly over the surface in contact of the cylinder bed (traverse) with the flange is reduced to two concentrated loads, each of which is equal in magnitude to half of the pressing force and is applied in the center of gravity of half rings, which are contact surfaces.

- The cylinder is attached to the frame (yoke) with screws.

In this case, a uniform load on the screw axis location circle is centered at two forces, each of which is equal to half of the largest pressing force and is applied in the center of gravity of semicircles.

- When pressing force transmission through the die plates positioned on the frame (traverse), concentrated load is replaced by a uniformly distributed load, which is symmetrical with respect to the press axis and are arranged on two-thirds of the length of the bolster.

- If the pressing force transferred by the tool, the contact surface of which with the frame (traverse) is a circle, uniformly distributed load in a circle is provided in two concentrated loads, each of which is equal in magnitude to half of the pressing force and is applied in the center of gravity of the semi-circle, and also is an integral part said circle.

Complexity of choice of design scheme is due to the fact that the frame is compound or continuous solid complicated structure, the dimensions of which are of the same order in the length and width [1]. In addition, sections of the sizes vary sharply. Thus, the base frame can be compared to neither timber nor the frame nor the plate of constant cross section.

The frame press is a basic part, so the method of its calculation should prevent premature failure due to the appearance of fatigue cracks or permanent deformations.

Despite the fact that the frames have accumulated considerable experience in design, their calculation is performed in most cases approximately. This is due to the complexity of the description of the stress-strain states of these parts the frames due to their design features.

Improving methods for calculating frame, more detailed account of the nature of their construction operation, identification of stress points make it possible to analyze more effectively their stress-strain state and offer the best design solutions.

In connection with the development of computer technology in design practice, numerous methods are widely adopted. One of the most effective and the most developed methods in the engineering practice is the finite element method (FEM).

The method consists in the fact that a given system (design) is broken down into individual elements of finite size - finite elements. These elements can be either flat or spatial, have enough diverse geometric shape, but only such shape allows you to explore the state of stress of the element under the action of external loads and the overall strength of interaction with neighboring elements.

As a tool program, complex ANSYS is used by many researchers.

The studies were conducted on the basis of a hydraulic press P7640A model (Fig. 1). The press has the following dimensions: width 1250 mm, height 2400 mm, weight 9800 kg.

In order to develop a methodology for finite element analysis, frame of the press is selected. Geometrical model of the bed is made on the basis of working drawings that will take into account the design features of the actual elements. Geometric model press frame (Fig. 2) consists of 346 dots, 5098 lines, 1671 surfaces and 53 volumes.

Figure 1. General view of a hydraulic press P7640A

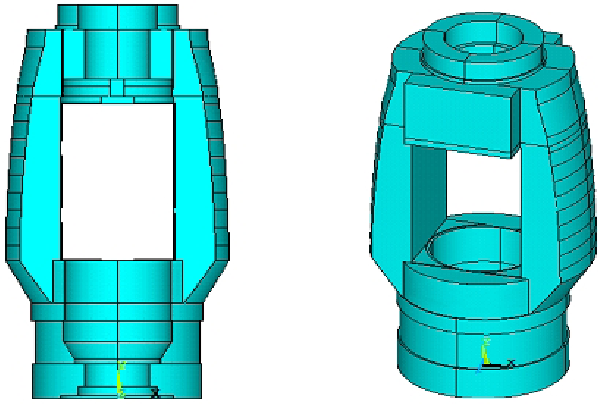


Figure 2. The geometric model of the press frame P7640A

Geometric model should match the type of item that will be used for finite element modeling [2, 3]. The paper does not consider beam and plate elements, because they do not provide the model correspondence to actual design. For calculation of the basic parts of powerful hydraulic presses, volume elements that provide an accurate account of the geometry of the base member (frame) are most effective. For frame as spatial body, where the state of stress in the dangerous points is three-axis, three-dimensional finite elements are used. Approximation of frame model is conducted by volumetric element SOLID 45 from the library of standard finite element software ANSYS [2, 3, 4]. The generated by program finite element model of the frame contains 81,297 items.

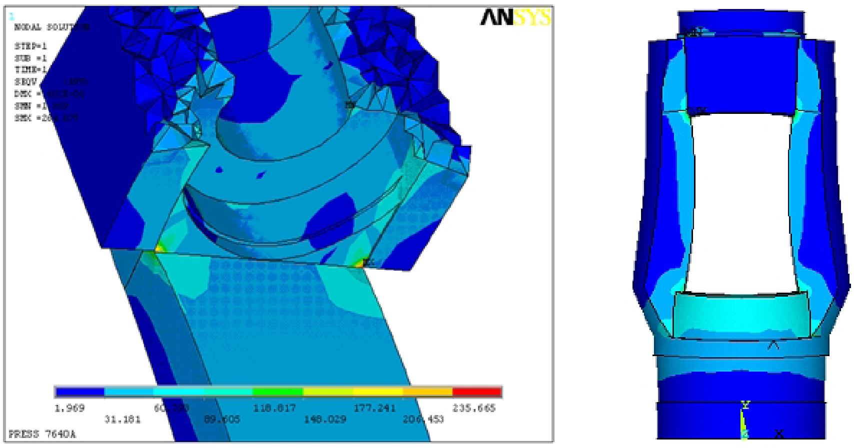
The frame is made of steel 35L according to GOST 977-88, i.e. in the program, the following mechanical properties are specified: modulus of longitudinal elasticity, MPa; Poisson's ratio, μ = 0.333.
Distributed load was applied to the press frame (beam and the lower part, where the cylinder is arranged). The intensity of a distributed load corresponds to the operating pressure p=500kPa. On the basis of fixing base frame press, the actual design was enshrined; that is the prohibition of all movements along the axes X, Y, Z.

As a result of static analysis of the frame press parameters, stress-strain state and their values were obtained (see. Table 1).

The equivalent stress in the frame is shown in Fig. 3.

Table 1. The maximum values of the parameters of the stress-strain state frame in specific sections

parameter	value	accommodation
UXmicron	0.17238	traverse
UY micron	0.40189	traverse
UZ micron	0.2355	traverse
USUM micron	0.4019	traverse
σ_x MPa	95.052	cylinder
σ_y MPa	165.088	traverse
σ_z MPa	104.695	cylinder
τ_{xy} MPa	66.211	stand adjacent to the cylinder
τ_{yz} MPa	54.591	stand adjacent to the traverse
τ_{xz} MPa	71.538	cylinder
σ_{equiv} MPa	235.665	traverse



a) b)

Figure 3. The equivalent stress σ_{equiv} concentration of stress at the attachment bolts and racks (a), stress field in the frame (b)

Let us determine the actual safety factor n for working pressure p = 50 MPa (material of frame is steel St. 25 L) with yield strength σ_T=280MPa.

For the upper arms, the greatest stress is σ_{equiv} = 235,665 MPa, stock factor:

$$n = \frac{\sigma_T}{\sigma_{equiv}} = \frac{280}{235.665} = 1.19,$$

For rack, the greatest stress is σ_{equiv} = 148,029 MPa, stock factor:

$$n = \frac{\sigma_T}{\sigma_{equiv}} = \frac{280}{148.029} = 1.89.$$

For most of the cylinder, stress is σ_{equiv} = 177,241 MPa, stock factor:

$$n = \frac{\sigma_T}{\sigma_{equiv}} = \frac{280}{177.241} = 1.58.$$

Considering that the calculation of the frame by finite element method is very accurate, and the increased stress is local, it can be assumed that the strength of the frame press P7640A when workloads is provided.

In accordance with the nature of the stress-strain state in case of symmetric load, the stresses and strains that occur at the frame press points located on one side of a vertical y-axis were studied. For this purpose, on the surface of the frame 12 load cells are symmetrically mounted.

In the experiment, the one-element gages brand 21KPA-5 were used. Fastening to the press frame is carried out by a cold curing adhesive - superglue.

Meter used for measuring deformations digital MDD – 1 for fixing the static strain gages by means included by the half-bridge circuit. During measuring the compressive force pressing successively increased from zero to 5 MN to the registration of readings every 1 MN.

The threefold repetition of the experiment showed coincidence readings for each of the strain gages glued.

The voltage value at each point is the average of the measured values for each load level. For example, the cross section of the frame, in which the strain gauge is No 1:

$$\sigma_y = \frac{1.6+1.2+1.2+1.2+1.6}{5} = 1.36\ kg/cm^2 = 0.136 \text{ MPa}$$

For obtaining voltage values at the maximum possible for the press P7640A tractive effort 10 MN, value σ_y obtained actually should be multiplied by 500. For example, a strain gauge No 1:

$$\max \sigma_y = 500 \cdot \sigma_y = 500 \cdot 1.36 = 680\ kg/cm^2 = 68 \text{ MPa}.$$

Analysis of experimental data has allowed establishing the following pattern: at points in the positive direction of the longitudinal axis (strain gauges 7, 8, 9, 10, 11, 12), the voltage is greater than the voltage at the symmetric points (strain gauges 1, 2, 3, 4, 5, 6). This suggests that an eccentric load is applied there, giving rise to a bending moment. The coincidence of theoretical and experimental results can be assessed as satisfactory. A more detailed experimental study requires special manufacturing equipment, providing a central load, which eliminates bending moments M_x and M_z.

When pressing situation arises, the pressing force is applied to the eccentric bearing surfaces of the press. Eccentric load application is equivalent to the combined effect of centrally applied load P and torque M (Fig. 4).

$$M = P \cdot e$$

where e — eccentricity of force P.

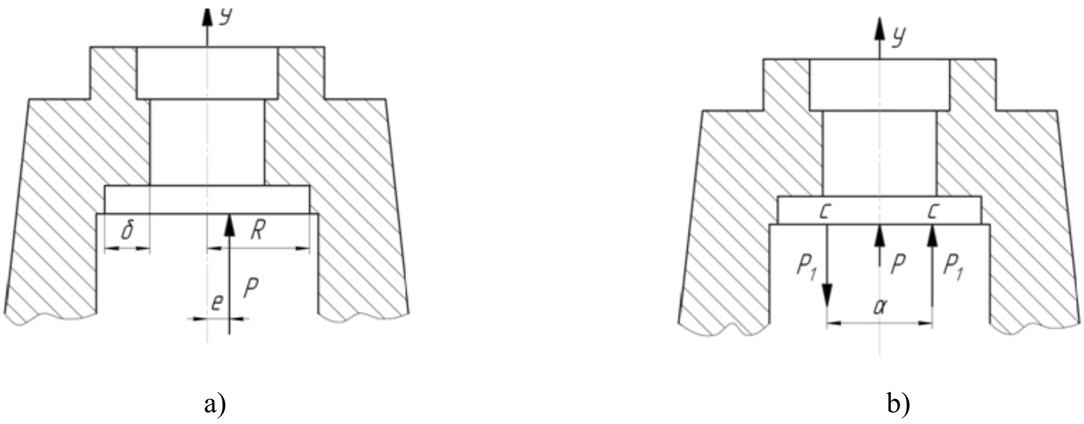


Figure 4. Scheme of the bed load with an applied eccentric load: the force applied to the eccentricity (a); central force applied to the pair (b)

The state of stress caused in the bed by centrally applied force of pressing was investigated previously by finite element method.

Research of the stress condition of the frame press, caused by the bending moment, is only possible with the help of the finite element method and is independent enough. Indeed, it is necessary to determine the arising voltage for arbitrarily oriented plane of action of the moment, and then to vary the solutions for different points in the plane of the provisions and to make a decision from the central action of pressing force.

Let us assume that the bending moment M is implemented by force P_1, which is the resultant contact loads transmitted by half die plate on the annular supporting surface of the upper crosshead. Contact loads are assumed to be uniformly distributed. Each force is applied in the center of gravity of a half ring bearing surface, which is determined by the position of $0.5a = 0.5\pi R\left(1 - \delta^2 / 12R^2\right)$,

where δ — the width of the ring bearing surface;

R — midline radius of the circular bearing surface.

Since $P_1 \cdot a = P \cdot e$, so $P_1 = \frac{e}{a} P$,

The strength of the P_1 acting on its supporting surface of half-ring creates the same effect as an applied central force $2P_1$.

Supposing that the bending moment M acts in the xy plane, which coincides with the symmetry planes of the supports. From the combined action of the force F and moment M, the total equivalent stresses take place in the right half of the frame

$$\sigma = \sigma(P) + \sigma(2P_1) = \sigma(P)\left[1 + \frac{2P_1}{P}\right] = \sigma(P)\left(1 + \frac{2e}{a}\right).$$

Applying strength condition

$$\sigma(P)\left(1 + \frac{2e}{a}\right) \le [\sigma],$$

to the point where the maximum stresses occur in the upper beam, we obtain the limit value of the eccentricity

$$_{\max} e_1 = \frac{a}{2}\left(\frac{[\sigma]}{\sigma(P)} - 1\right).$$

If the bending moment M acts in a plane zy, traverse the upper half, which cut off the xy plane, the total stress from the force F occur. In the sections of the rack, total equivalent stresses occur

$$\sigma = \sigma(P) + \sigma(M) = \sigma(P) + \frac{P \cdot e}{W_X},$$

where W_X — time support resistance

From conditions of strength

$$\sigma(P) + \frac{P \cdot e}{W_X} \le [\sigma],$$

applied to the rack point where there is the greatest stress, we obtain the limit value of eccentricity

$$_{\max} e_2 = \frac{1}{P}\left([\sigma] - \sigma(P) \right) W_X$$

From two values $_{\max} e_1$ and $_{\max} e_2$, we need to focus on a smaller one.

Section resistance modulus of rack of press P7640A is estimated:

in the upper part = W_X m³;

in the middle part = W_X m³;

at the bottom = W_X m³.

Here is an example to determine the maximum eccentricity press P7640A for the pressing force $F = 6$ MN, which corresponds to the pressure in the cylinder $p = 30$ MPa.

The allowable stress for steel 25 L with safety factor $[n]=1.4$ is

$$[\sigma]=\frac{\sigma_T}{[n]}=\frac{280}{1.4}=200 \text{ MPa.}$$

The average radius of the support ring surface is

$$R=\frac{1}{4}(0.5+0.3)=0.2 \text{ m,}$$

width of the bearing surface is

$$\delta=\frac{1}{2}(0.5-0.3)=0.1 \text{ m.}$$

The coordinate of the point of application of force P_1 is

$$\frac{a}{2}=\frac{2R}{\pi}\left(1-\frac{\delta^2}{12R^2}\right)=0.1246 \text{ m.}$$

In the most severely stressed point of the upper beam equivalent stress is

$$\sigma(P)=3.6\cdot p=3.6\cdot 30=108 \text{ MPa.}$$

Limit eccentricity is

$$_{\max}e_1=\frac{a}{2}\left(\frac{[\sigma]}{\sigma(P)}-1\right)=0.1246\left(\frac{200}{108}-1\right)=0.106 \text{ m}$$

The highest voltage equivalent at rack points, which are the farthest from axis x, occur in the lower sections:

$$\sigma(P)=2.92\cdot p=2.92\cdot 30=87.6 \text{ MPa.}$$

Limit eccentricity is

$$_{\max}e_2=\frac{a}{2}\left(\frac{[\sigma]}{\sigma(P)}-1\right)=0.1246\left(\frac{200}{108}-1\right)=0.106 \text{ m}$$

$$_{\max}e_2=\frac{1}{P}\left([\sigma]-\sigma(P)\right)W_X=\frac{200-87.6}{6}\cdot 8.166\cdot 10^{-3}=0.153 \text{ m.}$$

Finally, $e = 0.106$ m should be taken. It should be noted that in reality the force application point P_1, that is the resultant additional pressure forces on the bearing surface of the upper beam is located a little further from the longitudinal axis through the uneven distribution of pressure forces. Therefore, the actual values of the forces must be less than the expected values and the effect of the eccentricity must be a little weaker. Therefore, the expressions obtained above for the ultimate are understood eccentricity that goes to the margin.

We used software package ANSYS, where volume finite elements were used. The results make it possible to assess the rationality of the study design (in this case, the frame press) and change its shape and size to reduce metal consumption.

Analysis of the stress state of the frame press P7640A according to the results of finite element analysis on the PC has found the following:

- The upper cross member is deformed as a ring with a complex cross-sectional shape.
- Interaction of the upper beam with struts action is equivalent to the support forces and moments acting in the xy plane. Characteristic manifestation of the edge effect causes a significant circumferential normal stresses.
- The cylinder under internal pressure, force and torque efforts at the interface with the stands is deformed as a typical manifestation of the edge effect.
- Analysis of the field equivalent stress for the press P7640A leads to the conclusion that the construction of the frame press is not equal strength. The design needs to be optimized by reducing the edge effect in the upper beam and the frame [5, 6]. For this, the design of struts should be facilitated, the struts junction with the upper beam and on the smooth surfaces of the cylinder with a smaller mass of metal in these parts of the structure should be performed. This will reduce the bearing moment acting on the upper beam and the cylinder, and also reduce stress.

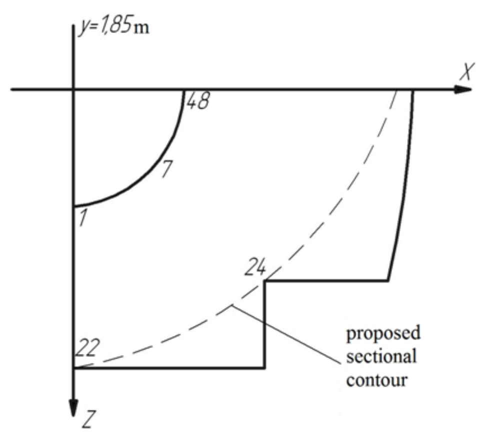


Figure 5. The cross section of the traverse, the proposed

Lines shown in Fig. 5 correspond to the weight reduction by 11% of the total weight of the frame.

Conclusion

If for any reason, e.g. the technology requirements for casting machine beds, their machining, operating conditions, etc., it is not possible to reduce significantly the maximum stresses in the upper beam and the cylinder, it should be borne in mind that the most intense local portions are local and sufficiently small. Therefore, press should be subjected to tension with such pressure, at which in the most stressed areas of construction, voltage would reach the yield strength. After unloading in these areas of construction, residual stresses, which are combined with the stresses during operation, will reduce the values of the peak stress and smooth them.

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