DIAGNOSTICS PROCEDURE FOR INSPECTION
OF BOILER COMPONENTS

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At present, a large number of vessels operating under pressure in industry have been working for periods longer than the design life. Nonstandard situations occur in many cases in which further service of a vessel becomes very problematic and is complicated by several factors.

Firstly, the state of the metal of the vessel in service greatly differs from the state of the metal of a new vessel and, even more so, from the state of rolled plates (prior to the fabrication of the vessel). Therefore, it is very risky to compare the characteristics of the metal of a specific vessel, which had been in service for the design life, with the tabulated values of strength presented in GOST 1050-88 or GOST 380-88 for rolled plates.

Secondly, the parameters of corrosion or mechanical damage of vessels are usually not included in conventional standards. Of course, this does not yet mean that equipment is not suitable for service; simply, there were no procedures which would make it possible to evaluate the actual risk resulting from this type of damage. Current design calculations using, for example, the finite element method, make it possible to determine these risks and select measures for preventing them – a decrease of working pressure or temperature; hardening of the structure during repair.

The development of a method of evaluating the risk from damage (depressions) consisted of several stages. Initially, it was proposed to restrict the permissible depth \( f \) of a depression by the following norm: 25 mm per 1 m of length \([1]\). However, without considering the wall thickness is not possible to evaluate bending stresses formed in the wall. It is obvious that for the same curvature of the wall in the area of the depression (characterized approximately by the depth to length ratio), the bending stresses on the surface will be proportional to wall thickness.

We shall examine a region of a wall in the form of a hinged strip (of unit width) loaded with uniformly distributed load \( q \), with the deflection

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f = \frac{5ql^4}{384EI}.
\]

Here \( L \) is the length of the strip; \( E \) is the Young’s modulus (for steel \( E = 0.2 \) MPa \((2 \times 10^6 \) kgf/cm\(^2\)); \( J = 1 \cdot S^3/12 \) is the moment of inertia of the section; \( S \) is the wall thickness.

Bending stresses \( \sigma = M/W \), where \( M \) is the bending moment; \( W \) is the moment of resistance to bending.

At \( L = 1 \) m = 100 cm, \( f = 2.5 \) cm, after the simplest transformations we obtain \( \sigma = 2400 \) S. Consequently, already at \( S = 0.65 \) cm the bending stresses will reach 1560 kgf/cm\(^2\) and thus (according to GOST 14249-89) exceed the permissible stresses for steel St3sp, i.e., 154 MPa \((1540 \) kgf/cm\(^2\)).

According to the second procedures \([2]\), the permissible depth of the depression (air hole) is restricted in per cent of the smallest size of its base. It is evident that this approach also does not take into account the real stress state in the wall of the vessel and evaluation is purely of qualitative nature.

The most extensively justified procedure, published in AOZT DIEKS in 1996 \([3]\), proposes to take into account the ratio of the depth of the depression to the wall thickness.

The Scientific and Production Association Tekhkranënergo developed a new method of calculating stress concentration resulting from the presence of a depression. The method is based on analysis of the stress state by the finite element method

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which makes it possible to calculate with sufficient reliability the stress concentration caused by the depression. The procedure takes into account not only the wall thickness and the depth of the depression but also its length and width, the vessel diameter, and even the orientation of the main axis of the depression in relation to the longitudinal axis of the vessel.

Calculations show that for a circular depression, the results obtained using the two procedures (AOZT DIEKS and Scientific and Production Association Tekhkranenérgo) are in almost complete agreement, but the latter procedure can be used to solve more complicated problems in examining more general defects typical of vessels subject to mechanical damage.

We shall examine analysis of the residual efficiency of a vessel on a specific example of an air collector with diameter $D = 1200$ mm, length $L = 3000$ mm, wall thickness $S = 8$ mm, investigated by Scientific and Production Association Tekhkranenérgo in 1998.

In diagnostics of the air collector, produced from steel St3sp (GOST 380-89), working at a temperature of $t = 20–30$ °C and a pressure of $p = 0.8$ MPa, examination of the wall of the shell showed a depression with length $A = 100$ mm, width $B = 40$ mm and a depth of $4$ mm, oriented along the generating line of the vessel.

The permitted stresses for this steel at the given temperature (according to GOST 14249-89) is $\sigma = 154$ MPa. The permissible pressure for the vessel without the depression (according to GOST 14249-89) is $p = 1.6$ MPa (16.1 kgf/cm²). Thus, the vessel without the depression would be capable of efficient service. However, in the presence of a depression it is necessary to examine the question of efficiency of the vessel. We shall examine various variants of solving this problem.

1. At the norm of permissible depth of the depression being $25$ mm per $1$ m $[1]$ (i.e., $2.5$ mm per $100$ mm of the length of the depression) and at the actual depth of $4$ mm, the vessel should be rejected. The possibility of its further service (even at reduced pressure) is not examined in this case.

2. At the norm of the permissible depth of the depression of $5\%$ of the smallest dimension of its base $[2]$ and at a width of the depression of $40$ mm, the permissible depth of the depression is $2$ mm. This means that at the actual depth of the depression of $4$ mm it is not permitted to operate the vessel without repair.

3. The procedure developed by AOZT DIEKS makes it possible to evaluate the degree of overloading of the vessel using the coefficient

$$\lambda = \frac{1.105}{h/S + (h/S)^2 + 1},$$

where $h$ is the depth of the depression.

For the given values of $S = 8$ mm, $h = 4$ mm, $\lambda = 0.68$, the permissible pressure $[p] = 16.1 \cdot 0.68 = 11$ kgf/cm² = $1.1$ MPa, i.e., it is higher than $8$ kgf/cm² (0.8 MPa).

In this case, the vessel can be used in service without any repair.

4. In calculations using the procedure developed by the Scientific and Production Association Tekhkranenérgo, it is necessary to use the tabulated values of the stress concentration factors which depend on the wall thickness, the depth, length, and width of the depression, and also on the diameter of the vessel and the angle of inclination of the major axis of the depression to its generating line. All the dimensions are used in the fractions of the wall thickness, $S = 8$ mm. At the shape of the depression close to circular, with the transverse dimension $d = A = B = 100$ mm = $12.5 \times S$, and the depth $h = 4$ mm = $0.5 \times S$ (vessel diameter $D = 1200$ mm = $75 \times S$), the tabulated value of the stress concentration factor in the center of the depression is $K_c = 0.52$. Consequently, the permissible pressure for the given vessel with the depression of these dimensions and shape should be decreased by a factor of $1.52$, to $[p] = 16.1/1.52 = 10.6$ kgf/cm² = $1.06$ MPa, i.e., higher than $8$ kgf/cm² (0.8 MPa).

It is evident that the results of the calculations obtained using the last two procedures for the depression of approximately circular shape are almost identical. However, for an elongated depression at $A = 100$ mm, $B = 40$ mm, the angle of inclination of the axis of the depression to the generating line of the vessel $\alpha = 0°$, the stress concentration factor in the center of the depression is $K_c = 3.46$. This means that the vessel can be either rejected or the permissible pressure must be reduced to $[p] = 16.1/3.46 = 4.6$ kgf/cm² (0.46 MPa).

In this case, the calculated stress in the wall of the shell with radius $R$ at a working pressure of $8$ kgf/cm² is

$$\sigma = pR/K_c/S = 8 \cdot 60 \cdot 3.46/0.8 = 2080 \text{ kgf/cm}^2 = 208 \text{ MPa},$$

which is considerably higher than the permitted stress.

Does this mean that further service of the vessel is not possible? According to the calculations carried out on the basis of the permitted stresses, the vessel should be rejected. However, if service is characterized by static loading (and the material of welded pressure vessels is always ductile), the stress concentration may not be dangerous because this stress level is obtained