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Resolution of Stress into Components for Strength Assessments of Permanent Joints

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Abstract—The formulation of permanent joints is considered. To improve joint creation, resolution of their stress–state into components during the design stage is proposed.

Keywords: permanent joints, stress concentration, stress state, flexure, resolution, components, hybrid solder joint

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Improvement in permanent joints, including welds, is based not only on technological measures but also on effective assessment of their strength and, in particular, the stress concentration during operation. Solder joints, which are widely used in engineering, are characterized by high static and cyclic strength [1]. However, they exhibit considerable chemical and mechanical heterogeneity. That may produce a nonuniform stress distribution even for joints of the same shape—for example, in butt joints of the same type of workpiece [2]. In creating optimal structures using permanent joints, detailed analysis of their stress fields is required. Finite-element software is used for that purpose. Knowing the stress distribution, it is simple to establish the hazardous zones but difficult to establish the causes of the nonuniformity [3].

Another approach to determining the stress state of a structural element is the decomposition principle: resolution of a complex problem into a set of simpler problems [4].

For example, the stress concentration coefficient K_σ in a plate with a hole increases with distance between the hole and the central axis of the plate, as established experimentally in [5]. However, the underlying causes were not considered in [5]; that hinders the search for remedies.

Since similar structural elements may appear in permanent joints, it is expedient to investigate why the stress distribution changes for different hole positions. To that end, we will resolve the model of the stress state into simpler subcomponents. That eliminates expensive experiments to assess the carrying capacity of permanent joints with holes.

Consider the cross section of a plate with a hole (diameter D) at a distance y_h from the axis (Fig. 1). The hole shifts the cross section's center of gravity a distance y_c in the opposite direction. Regarding the hole as an element with negative area, we determine the displacement of the center of gravity from the formula [6]

$$y_c = \frac{F_h y_h}{F_l + F_h} = \frac{\delta D y_h}{\delta b - \delta D} = \frac{D y_h}{b - D}, \qquad (1)$$

where $F_l = \delta b$ is the cross-sectional area of the plate with no hole; $F_h = \delta D$ is the area of the hole; δ is the plate thickness.

Under the action of tensile force P, an additional flexural torque is produced by the displacement of the center of gravity

$$M_{fl} = P y_c = \sigma\delta b y_c = \sigma\delta\frac{D y_h}{b - D}, \qquad (2)$$

where σ is the stress created by the force P.

The normal stress due to flexure at the hole is tensile and is added to the stress due to the tensile longitudinal force P (Fig. 1). The additional flexural stress at point A, which is at the edge of the plate, is greater than that at point B, on account of the higher K_σ at point A, as established in [5].

The additional normal stress at point A is close to the maximum flexural stress

$$\sigma_{fl.max} = \frac{M_{fl}}{W}, \qquad (3)$$

where $W = J/h_{max}$ is the drag torque of the cross section with a hole relative to the axis passing through the

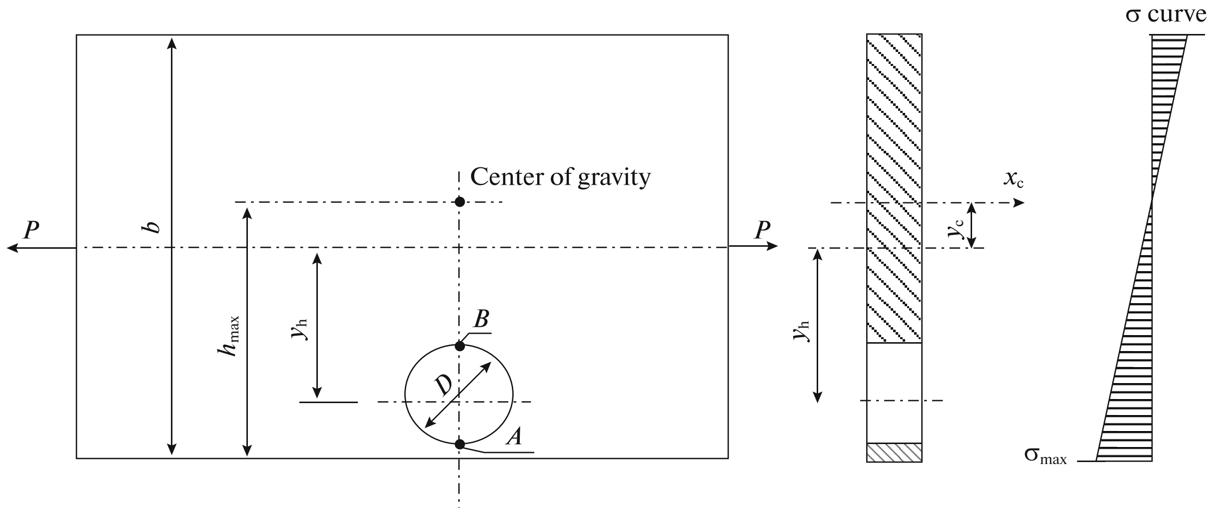

Fig. 1. Loading of a plate with a displaced hole.

center of gravity; and J is the moment of inertia of the cross section relative to the axis passing through the center of gravity [7]

$$J = J_{pl} - J_{ho} = \frac{\delta b^3}{12} + \delta b y_c^2 - \frac{\delta D^3}{12} - D\delta(y_h + y_c)^2, \quad (4)$$

where J_{pl} is the moment of inertia of the intact plate; and J_{ho} is the moment of inertia corresponding to the part of the plate removed to make the hole.

More precisely, the additional stress at point A may be determined on the basis of the linear distribution of the normal stress

$$\frac{\sigma_{fl.max}}{0.5b + y_c} = \frac{\sigma_{fl.A}}{y_c + y_h + 0.5D}. \quad (5)$$

In determining the additional flexural stress from Eqs. (1)–(5), we obtain an unwieldy final expression.

We now consider the most hazardous hole position: its emergence at the edge of the plate (Fig. 2) [5]. The cross section with the hole remains rectangular. Therefore we find that

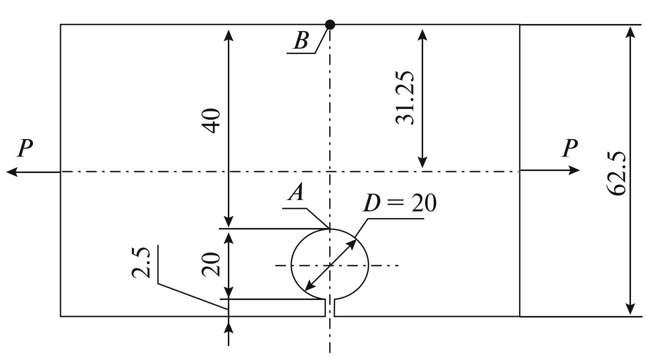

Fig. 2. Loading of a plate with a hole displaced to its edge.

$$y_c = 11.25 \text{ m}; \quad M_{fl} = Py_c = 11.25\sigma\delta b;$$

$$W = \frac{\delta AB^2}{6} = \delta\frac{40^2}{6} = 266.7\delta;$$

$$\sigma_{fl.A} = M_{fl}/W = 2.64\sigma.$$

Experiments show that the maximum stress appears at point A with $K_\sigma = 7$ [5].

Summing the maximum normal flexural stress and the stress from the longitudinal tensile force, we obtain $\sigma_{max} = 4.2\sigma$.

Thus, from the overall maximum stress in the plate with the displaced hole, it follows that 60% of the stress is due to flexure and only 40% is due to stress concentration.

When the hole is not displaced, the stress concentration coefficient $K_\sigma = 3$ [8]. That differs by only 6.7% from the stress concentration coefficient with a displaced hole: $K_\sigma = 2.8$.

To refine the results, we calculate the normal stress by the finite-element method for the plate in Fig. 2, using Solidworks software (Fig. 3). We find that the maximum stress when $P = 10$ MPa is 74.9 MPa. The stress concentration coefficient $K_\sigma = 7.49$; that is 7% more than the experimental value in [5]. This indicates that the solution is highly accurate.

In Fig. 4, we show the tensile stress distribution in a flexurally deformed plate with a hole at the edge.

In Fig. 5, for comparison, we show the stress distribution in a symmetric plate with a hole at the edge. In the absence of flexure, the maximum longitudinal normal stress is reduced to 33.1 MPa (by a factor of 2.26). The maximum stress in the symmetric sections is practically the same as with a central hole [8]. That indicates an effective means of eliminating flexure in structures.

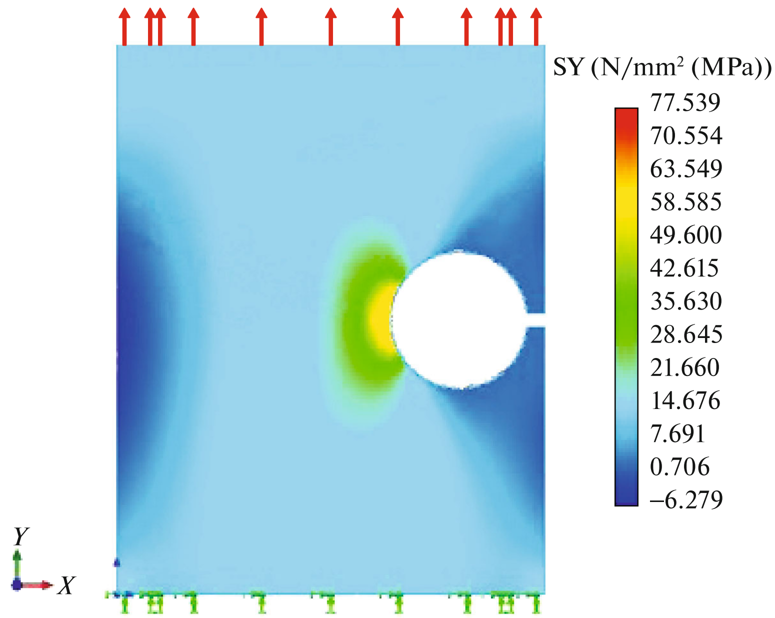

Fig. 3. Distribution of the longitudinal normal stress in a plate with a displaced hole.

The decomposition principle also permits more accurate assessment of the strength of joints combining solder seams of both butt and lap type. We see that soldered structures are largely ineffective in view of the different deformability of the butt and lap seams, leading to failure of the sections with butt seams in joints of collar and step type, regardless of the total solder area [9].

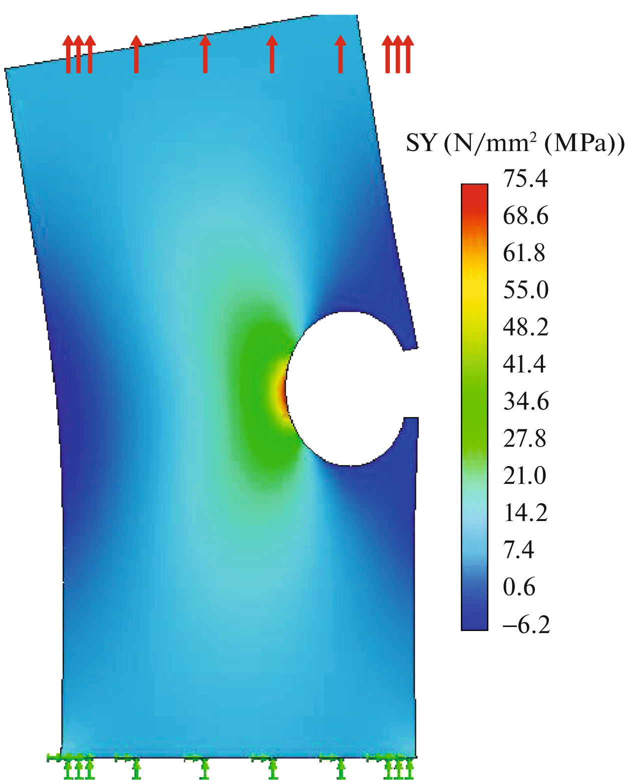

Fig. 4. Distribution of the tensile stress in a deformed plate with a hole displaced to its edge.

The stress distribution in mixed solder joints in Fig. 6 shows that the stress is 7.25 MPa in butt joints and 1.65 MPa in lap joints. Therefore, the mixed joints PN-2, PV-1 and PN-5, PV-2 and 2PN-3, 3PV-1 and PV-2, PV4 and PV-1, 2PN-1 and PV-2, and 2PN-4 recommended in State Standard GOST 19249–73 must be ruled out, since they do not offer greater carrying capacity than simple butt joints.

The decomposition principle is useful not only in design but also in tests of solder joints undertaken to develop recommendations for testing under cyclic loads [10].

Lap joints are widely used with flexural stress due to asymmetric loading [8, 11]. The decomposition principle was used to decrease the flexural stress in contact point welding with glue in [12]; the components being joined were very thick. The decomposition principle was used on account of the complexity of the problem and the indeterminacy of the solutions: instead of one complex problem, three simpler problems were solved. It was found that influence of flexure on the cement layer could be decreased by tapering the edges in the lap zone. That reduced the total thickness of the welded parts and the eccentricity of the operational forces. In addition, the power of the welding equipment was decreased, and the strength of the weld point was increased by 28% [12].

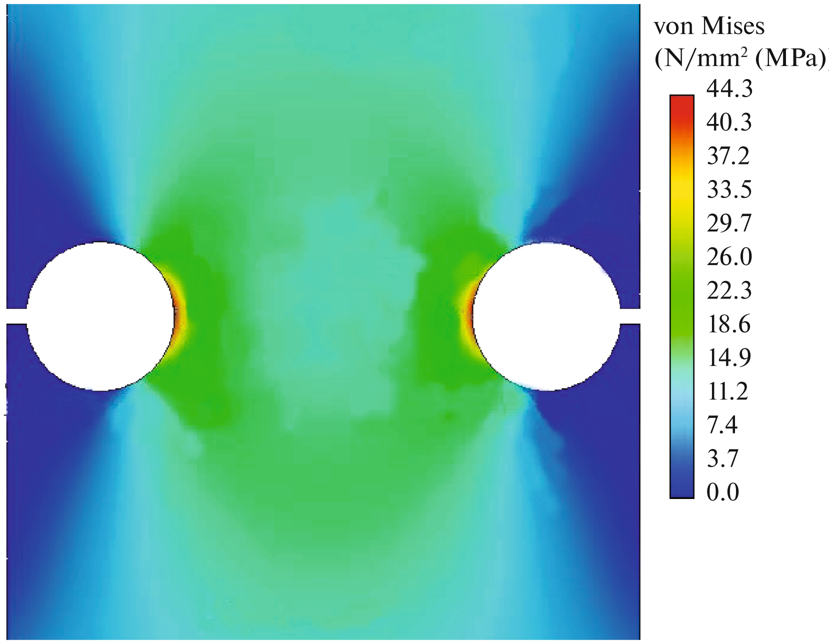

Fig. 5. Distribution of the stress in a symmetric plate with a hole displaced to its edge.

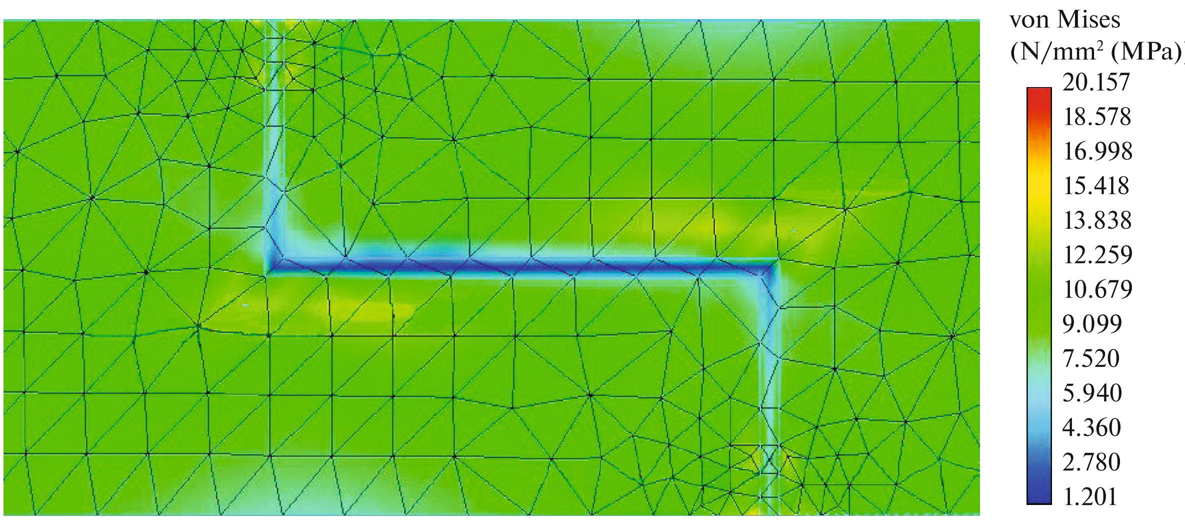

Fig. 6. Stress distribution with mixed solder joints.

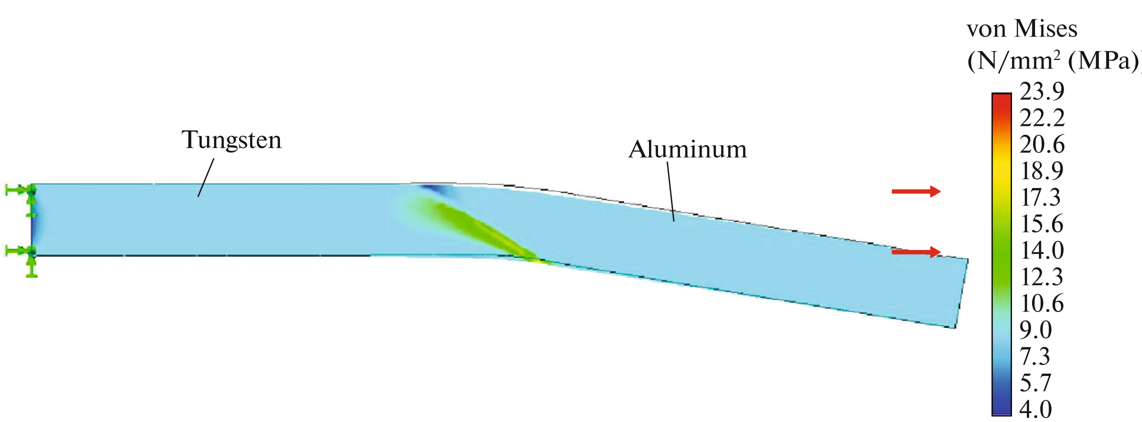

Fig. 7. Stress distribution in a deformed oblique solder joint between tungsten and aluminum.

Thus, in developing the optimal configuration of a part with permanent joints, it is expedient to employ the decomposition principle in analysis of the stress–strain state, so as to distinguish between the stresses due to flexure or shear. To that end, we have developed software based on the finite-element method to detect even barely perceptible flexure—for example, in oblique joints of materials differing in elastic modulus (Fig. 7).

The effectiveness of this approach was confirmed by the assessment of unidirectional butt welds. It was found that convexity of the weld not only acts as a stress concentrator at the boundary with the basic metal but also generates additional stress in the core of the weld [13]. In addition, the additional stress is greatest when the height of the convexity is half the thickness of the plates in the butt weld.

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