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Experimental and numerical studies on failure modes of riveted joints under tensile load

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ABSTRACT

Though engineers are not advised to utilize riveted joints in tension, rivets will inevitably bear tensile load in realities. Hence, it is interesting to investigate the failure modes of riveted joints when they are under tensile load. Following previous studies (Chen et al., 2011), in this work, the authors selected three sizes of riveted joints to conduct the riveting and tension processes experimentally and numerically. Three kinds of failure modes including pull through, shank breaking, and head breaking were observed. Simulations are able to give almost the same results as those from experiments. Furthermore, three formulas were proposed to calculate the maximum tensile strength of riveted joints. Though the values calculated from these three formulas are approximate, they have the same order of magnitude as real ones. Moreover, they could be utilized to estimate which kind of failure mode may take place when riveted joints were under tensile load.

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1. Introduction

As one of the most important mechanical joints, riveted joints are extensively utilized in various configurations for a wide range of industrial products, such as aircrafts, vehicles, bridges, and many other products. In the famous Eiffel Tower in Paris, about 2.5 millions rivets were utilized to connect 15,000 iron pieces (http://corrosion-doctors.org/Landmarks/eiffel-history.htm). In modern Boeing 747 airplanes, abundance of rivets can also be found to join many parts because of their high reliability. To study the riveting process of riveted joints and their strength, engineers usually use a series of practical experiments which are time-consuming and costly. In recent years, with the development of finite element method (FEM) and computing technology, FEM softwares are able to deal with most mechanical problems, such as bulk forming, sheet forming, and many other processes. In addition, with FEM software, a better insight of the process is possible by analyzing the various mechanical parameters of the models.

There is a general piece of advice: "Never use a rivet in tension" (Slagter, 1995). However, in reality, tensile loads are inescapable. For example in airplanes, due to the upward suction explain force, the rivets that hold on the skin to the stringers and ribs undergo tensile stresses. Hence, rivets are mainly under three kinds of loads: shearing load, tensile load, and the combination of these two. When a rivet is under large shearing force, the rod of the rivet will break because of the force. However, there are several failure modes when a rivet is under tensile load. The first kind is that the rivet head is pulled through sheets. The second one is that the rivet rod breaks under tensile load which is similar to the pure extension process of a rod billet. The third mode is that the rivet head breaks. Assuming that the plate material was rigid-perfectly plastic and the tension and compression yield stresses were equal, Slagter (1995) studied the pull-through strength of flush head rivet. He deduced a very complex mathematical function by neglecting a few details and compared it with some experimental results found in literatures. The function was capable to give a nearly correct but not exact result, because a few details were neglected. During 1920s, Wilson and Oliver (1930) did much work on solid rivets. In their experimental reports, the last two kinds of failure modes in tension were both found. Langrand et al. (2001) used Arcan experimental tools to measure the tensile strength of riveted joints. The rods of rivets broke in their study and the corresponding simulations were developed in FEM software. Gurson damage criterion was utilized to depict the fracture process of the rivet rod. However, a systematic

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and comprehensive investigation of the failure modes of riveted joints in tension is very scarce if any in the literature. Hence, in the current article, the main work is to develop the experimental and numerical models to analyze these three failure modes.

2. Finite element method

For a more complete description of the scientific approach of FE approach of metal-forming process, the reader may refer to Wagoner and Chenot (2001).

2.1. Constitutive law

For cold-forming processes, an elastic plastic law must be chosen. A widely used approximation, which reveals accurate enough in most applications, is based on an additive decomposition of the strain rate tensor $\dot{\varepsilon}$ into an elastic part $\dot{\varepsilon}^e$ and a plastic one $\dot{\varepsilon}^p$:

$$\dot{\varepsilon} = \dot{\varepsilon}^e + \dot{\varepsilon}^p \tag{1}$$

In our case, where material rotations remain relatively small, the use of a non-objective derivative of the stress tensor is acceptable. The hypo-elastic law is written as:

$$\frac{d\sigma}{dt} = \lambda^e trace(\dot{\varepsilon}^e)I + 2\mu^e \dot{\varepsilon}^e \tag{2}$$

where λ^e and μ^e are the Lamé coefficients.

The plastic component of the strain rate tensor obeys the flow rule:

$$\sigma' = \frac{2}{3}\sigma_0(\bar{\varepsilon})\frac{\dot{\varepsilon}^p}{\dot{\bar{\varepsilon}}} \tag{3}$$

where σ' is the deviatoric stress tensor, σ_0 is the yield stress, $\dot{\bar{\varepsilon}}$ is the equivalent strain rate, and $\bar{\varepsilon}$ is the equivalent strain.

Different laws can be used to describe work-hardening. To represent our materials accurately, the following form is chosen:

$$\sigma_0(\bar{\varepsilon}) = \sqrt{3}k(1 + a\bar{\varepsilon}^n) \tag{4}$$

where a, n, and k are the material parameters. To obtain the exact values of these three parameters of each kind of material, uniaxial tension test was conducted to get the load-displacement curve of this process and the curve was transferred to the true stress strain curve to obtain the initial material parameters. The initial material parameters were improved by iterative computation with Forge2009®.

2.2. Friction

At the interface between part and tool, the friction shear stress can be modeled by a Coulomb law:

$$\tau = -\alpha_f|\sigma_n|\frac{\Delta v}{|\Delta v|}\ with\ |\tau| \le \frac{\sigma_0}{\sqrt{3}} \tag{5}$$

where α_f is the friction coefficient, σ_n is the normal stress, and Δv is the tangential velocity.

2.3. Virtual work principle

For an incompressible or quasi-incompressible flow, it is desirable to utilize a mixed formulation. In the domain Ω of the part, for any virtual velocity and pressure fields, v^*, p^*, this formulation can be written as:

$$\int_\Omega \sigma' : \dot{\varepsilon} * dV - \int_\Omega p div(v*)dV - \int_{\partial\Omega_c} \tau v * dS = 0 \tag{6}$$

and the mass conservation equation can be written as:

$$-\int_\Omega (\kappa div(v) + \dot{p})p * dV = 0 \tag{7}$$

where κ is the compressibility coefficient of the material.

The most widely used time integration method remains the simple one-step Euler scheme. In this simple approach, the total time is decomposed into small increments Δt and the displacement field is proportional to the velocity field at the beginning of the increment:

$$\Delta u = \Delta t \quad v \tag{8}$$

In the same way, the stress increments are introduced, so that the Eqs. (9) and (10) are rewritten as:

$$\int_\Omega (\sigma' + \Delta\sigma') : \dot{\varepsilon} * dV - \int_\Omega (p + \Delta p) div(v*)dV - \int_{\partial\Omega_c} \tau v * dS = 0 \tag{9}$$

$$-\int_\Omega \left(\frac{\kappa}{\Delta t} div(\Delta u) + \Delta p\right) p * dV = 0 \tag{10}$$

2.4. Finite element approximation

Many different finite element formulations were proposed, and developed at the laboratory level, but it is now realized that the discretization scheme must be compatible with other numerical and computational constraints, among which the following can be quoted:

- Remeshing and adaptive remeshing.
- Unilateral contact analysis, iterative solving of nonlinear and linear systems.
- Domain decomposition and parallel computing.
- Easy transfer of physical internal parameters, when multi-physic coupling must be taken into account.

To-day, a satisfactory compromise is based on a mixed displacement and pressure formulation using tetrahedral elements, and a bubble function for the velocity in order to stabilize the solution for incompressible or quasi-incompressible materials. The displacement field is discretized with shape functions N_n, in terms of nodal displacement vectors U_n:

$$\Delta u = \sum_n \Delta U_n N_n(\xi) \tag{11}$$

The pressure field is generally expressed in terms of different shape functions:

$$p = \sum_m p_m M_m \tag{12}$$

Using isoparametric elements, the mapping between the physical space with coordinates **x** and the reference space with coordinate ξ is:

$$x = \sum_n X_n N_n(\xi) \tag{13}$$

The strain increment tensor can be computed with the help of the conventional **B** discretized linear operator:

$$\Delta\varepsilon = \sum_n U_n B_n \tag{14}$$

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