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A study of forming pressure in the tube-hydroforming process

Fuh-Kuo Chen*, Shao-Jun Wang, Ray-Hau Lin

Department of Mechanical Engineering, National Taiwan University, Taipei, Taiwan, ROC

Abstract

The forming pressure required to produce a desired part using the tube-hydroforming process was investigated in the present study. The relationship between hydraulic pressure, outer corner radius of the deformed tube, tube thickness and tube yield stress was established based on a proposed theoretical model. In the theoretical model, the material hardening property was taken into consideration. Since the friction in the tube-hydroforming is smaller than that in the conventional stamping process, the die-closing force was also calculated according to the forming pressure predicted by the proposed theoretical model under frictionless condition. The finite element analysis was performed to validate the proposed theoretical model. In order to confirm the accuracy of the finite element simulations, two different finite element codes were employed to conduct the analysis and the simulation results were compared. The predicted values calculated by the proposed theoretical model were found to agree well with those obtained from the finite element simulation results.

Keywords: Forming pressure; Tube-hydroforming; Theoretical model; Finite element simulations; Material hardening

1. Introduction

Tube-hydroforming process has been widely used to produce automotive structural components due to the superior properties of the hydroformed parts in lightweight and structure rigidity. Compared to the traditional manufacturing process for a closed-section member, i.e., stamping and then welding process, tube-hydroforming leads to cost saving due to fewer die sets being required. In addition, most of tube-hydroformed structural members are of a one-piece type, resulting in a superior structural integrity to an assembly of stamping parts.

In a hydroforming process, the cross sections along the tube can be varied so long as the length of periphery of each deformed cross section is larger than that of the original tube within a certain limit. A set of dies with the die cavity identical to the part shape is designed to form the part. Most hydroformed parts have a complex shape and need a pre-bending process to bend the tube to the curvature of the parts. An additional pre-forming process may be also required to crush the tube so that the crushed tube can be fitted into the die cavity. The major deformation in the hydroforming process is expanding the tube itself to conform with the die cavity by increasing the internal hydraulic pressure to a designed value. For a part with larger difference between the perimeters of cross sections may require an axial feed to provide more tube material to the larger cross sections to fill the die cavity, in addition to the expansion by hydraulic pressure. The bursting failure mode may occur if the hydraulic pressure is too high [1,2]. The part corner with a smaller radius, on the other hand, may not be completely filled with insufficient internal hydraulic pressure. The underfilling problem is related to the formability of tube material [3,4]. The finite element method was also widely employed to simulate the hydroforming processes and determine the suitable process parameters [5,6].

The major concern of dimensional accuracy in a hydroformed part is the formation of an edge corner to a desired radius. It is known that the hydraulic pressure required in a tube-hydroforming process to manufacture a structural part depends on the minimum corner radius in the part shape. Therefore, a formability study of the tube material is essential to the tube-hydroforming design. The formability analysis of tube-hydroforming may include the relationship between hydraulic pressure, outer corner radius of the deformed tube, tube thickness and tube yield stress. The press tonnage required to hold the tube in place while it is internally pressurized, which is called die-closing force, is also very important in the tube-hydroforming design. A comprehensive knowledge of tube-hydroforming can be found in Ref. [7]. In Ref. [7], an empirical formula for predicting the pressure to form a corner radius is proposed. However, the author points out that the predicted pressure is an upper-

* Corresponding author. Tel.: +886 2 23661322; fax: +886 2 23631755.
E-mail address: fkchen@ntu.edu.tw (F.-K. Chen).

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bound value and is generally significantly higher than the actual pressure required.

In the present study, a theoretical model was proposed to construct the above-mentioned relationship and calculate the die-closing force. Two different finite element codes were employed to validate the proposed theoretical model. A comparison is subsequently made between the predicted values and the finite element simulation results.

2. Theoretical model

In order to simplify the analysis, a long, straight hydroformed part was considered, and the part shape can be formed by expansion alone without the need of axial feed. Thus, a state of plane strain deformation can be assumed in the theoretical model. In the analysis, the following assumptions are made:

(i) The tube is homogeneous and isotropic.
(ii) The elastic deformation is negligible and the tube is considered as rigid-plastic.

With the required internal hydraulic pressure calculated by the proposed theoretical model, the die-closing force exerted by the press can be predicted by carrying out a simple analysis. Both the theoretical model and the simple analysis are depicted in the following.

2.1. Relationship between hydraulic pressure and corner radius

A rectangular cross section, as shown in Fig. 1, was adopted to define the dimensions of the tube thickness (t), inner corner radius (r_i), and outer corner radius (r_o). For simplicity, a corner of an arbitrary die cross section, as shown in Fig. 2, was used to construct the theoretical model. In order to further simplify the analysis, the configuration was made symmetric with respect to the y-axis so that only one half of the configuration is necessary to be considered for the analysis. The die opening angle, as denoted by 2α in Fig. 2, was chosen as an arbitrary value to make the model sufficiently general. For the deformed tube at any stage of hydroforming after die-closing, the tube apex which is not in contact with the die surface can be assumed as a circular arc with a uniform thickness, t, and a mean radius of curvature, $\bar{r}$, as shown by A–B in Fig. 2. It is also assumed that a further

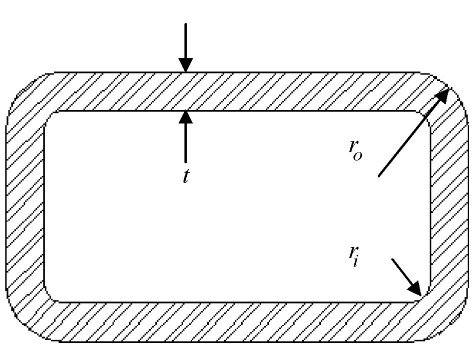

Fig. 1. Dimension of a rectangular cross section.

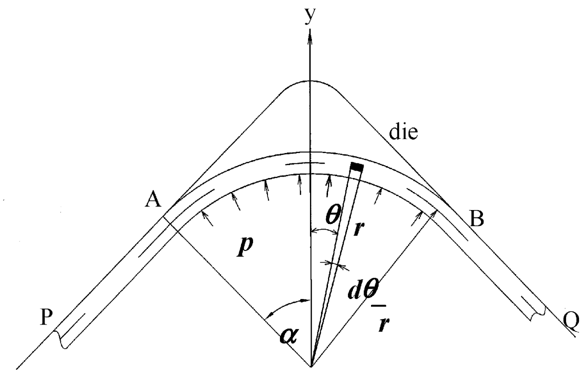

Fig. 2. A corner of an arbitrary die cross section.

deformation caused by an increment of internal pressure only occurs over the circular arc, and the remaining portion of the tube that is already in contact with the die surface, such as A–P and B–Q shown in Fig. 2, remains as it is. Therefore, it may be convenient to use polar coordinates (r,θ) for the analysis of the incipient plastic deformation of the tube apex.

Consider a differential element of the deformed tube at angle of θ, as shown in Fig. 2. The condition of force equilibrium in the r-direction results in

$$\frac{d\sigma_r}{dr} - \frac{\sigma_\theta - \sigma_r}{r} = 0, \tag{1}$$

where σ_θ and σ_r are the hoop and radial stresses, respectively. Since $\sigma_\theta > \sigma_r$ and σ_z is an intermediate stress in the plane strain condition, the Tresca yield criterion is given by

$$\sigma_\theta - \sigma_r = \bar{\sigma} \tag{2}$$

where σ_z is the axial stress and $\bar{\sigma}$ is the flow stress of the tube material. It is readily seen that the hydraulic pressure (p) required to cause the incipient yield of the tube apex is obtained by setting $\sigma_r = -p$ at $r = r_i$ and $\sigma_r = 0$ at $r = r_o$, resulting in

$$p = \bar{\sigma} \cdot \ln \frac{r_o}{r_i} \tag{3}$$

Rearranging Eq. (3) and taking exponential, we obtain the relationship between the outer corner radius of the deformed tube and the hydraulic pressure as

$$\frac{r_0}{t} = \frac{1}{1 - e^{-(p/\bar{\sigma})}} \tag{4}$$

It is to be noted from Eq. (4) that an exponentially proportional relationship exists between the hydraulic pressure required to expand the tube and the ratio of outer corner radius to the current thickness of the deformed tube, instead of the outer corner radius alone.

Both Eqs. (3) and (4) are applicable to a tube without work-hardening behavior. For tubes which work-harden, use of the stress–strain relations must be made to calculate the flow stress $\bar{\sigma}$. It is assumed that the tube work-hardens according to the stress–strain relations

$$\bar{\sigma} = K(\varepsilon_0 + \bar{\varepsilon})^n, \tag{5}$$