Drawing simulation by static implicit analysis with the artificial damping method

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Abstract. Wrinkling during draw is typically a local instability problem. When the structural instability is localized, there will be a local transfer of strain energy from one part of the structure to neighboring parts, and global solution methods, which is typically represented by the arc length method, may not work. So, this type of problems has to be solved either dynamically or with the artificial damping. On the other hand, the essential nature of the buckling behavior can be regarded as a static problem, even though it may be possible to raise some side issues due to the inertia effect. In this study, we traced the local buckling behavior of anisotropic elasto-plastic thin shells in Numisheet2014 BM4 using the artificial damping method.

1. Introduction
Advanced can manufacturing technology and cost control efforts have resulted in a consistent reduction of the net metal weight used, which has led to beverage cans with thinner sidewalls, reduced neck diameters, and smaller base diameters. However, small-diameter cans increase the likelihood of wrinkling during production. Wrinkling during redraw is typically a local instability problem that may be triggered by a small, local disturbance. When the structural instability is localized, there will be a local transfer of strain energy from one part of the structure to neighboring parts, and global solution methods, typified by the arc length method, may not work. This class of problems has to be solved either dynamically or statically with the aid of artificial damping. The latest general purpose finite element codes provide automatic mechanisms for stabilizing unstable quasi-static problems through the automatic addition of viscous damping to the model. When local instability occurs, the deformation rate of that portion begins to increase and, consequently, locally released strain energy is dissipated due to the appended artificial damping effect. In the phenomena of the real cases, the energy generated by the artificial damping is equivalent to the energy released as the inertia effect. Namely, replacing the kinetic energy with the artificial damping energy makes it possible for the unstable phenomenon to be analyzed stably in order to obtain the equilibrium solution within the static analysis.

In this study, we traced the local buckling behavior of anisotropic elasto-plastic thin shells in Numisheet2014 BM4 using the artificial damping method.
2. Simulation model and result

A schematic view of tools used for the drawing process, including their dimensions, is shown in Figure 1. The drawing occurs continuously during one punch stroke, and the punch speed is 140 mm/sec. The recommended friction coefficient is 0.03, and the blank holding force is given as 8900 N. AA5042 aluminum alloy \((t = 0.2083\, \text{mm})\) is considered as sheet materials. This material is assumed to have the YLD2000 anisotropic yield criteria\(^1\).

The anisotropic elasto-plastic material behaviors were implemented in Abaqus by using the UMAT user-defined material subroutine. In this study, we traced the local buckling behavior of anisotropic elasto-plastic thin shells using the artificial damping method. The artificial damping method is a technique used to achieve total energy balance in static analysis, compensating for strain energy changes due to local elastic instability. Abaqus provides an automatic mechanism for stabilizing unstable quasi-static problems through the automatic addition of volume-proportional damping to the model, while the artificial damping method or its comparable quasi-static capabilities are available in other codes. Via Eq.(1), viscous forces are added to the global equilibrium equations, as shown in Eq.(2):

\[
F_v = cM^*v
\]

\[
P - Q - F_v = 0
\]

where \(M^*\) is an artificial mass matrix calculated with unity density, \(c\) is a damping factor, \(v = \Delta u/\Delta t\) is the vector of nodal velocity, \(\Delta u\) is the vector of incremental displacement, \(\Delta t\) is the increment of time (which may or may not have a physical meaning in the context of the problem being solved), \(P\) is the total applied load, and \(Q\) is the internal force. The automatic seamless simulation provided good agreement with the experimentally observed buckling behavior, which covered the entire drawing process well, as shown in Figure 2.

3. Eigenvalue analysis

Using implicit codes, it makes it possible to calculate eigenmodes at any midpoint in the analysis. Figure 3 shows the results from the eigenvalue analysis for AA5042 of \(H_1 = 5.207\, \text{mm}\). The eigenvalue analysis was performed at the very early stage of the punching process (2 mm of stroke). The magnitude of this punch stroke is small enough to assume a nearly uniform
circumferential stress distribution on the shell. The generated plastic strain at this point is around 1-2%.

The curves in Figure 3 represent the relationship between the normalized eigenvalue and the number of circumferential full-waves \( n \), where \( m \) is the number of radial half-waves. In the vicinity of the lowest critical value, multiple eigenmodes with different numbers of waves \( (n=9-12 \text{ for } m=1) \) exist. The proximity of the eigenmodes must have impaired the reproducibility of experiments, where our history analysis of the punching process provides a unique buckling mode with \( n=12 \).

The buckled area on the shell can be regarded as a circular plate with a hole at the center and with the inner and outer boundaries clamped. Timoshenko[3] suggested that such a circular plate, when subjected to radial loading, buckles in multiple waves along the circumference. For the ratio of the inner and outer radius approaching unity, the buckling conditions for a compressed ring are analogous to those of a long, compressed rectangular plate. In the case of the rectangular plate, a typical deformation mode indicates several half-waves in the direction of compression but only one half-wave in the perpendicular direction (in Figure 4). The lengths of the half-waves are supposed to approach the width of the plate, because a buckled plate subdivides approximately into squares. A single eigenvalue will be extracted for this deformation mode. However, manufacturing experience indicates that the real process conditions, such as the mechanical properties of the aluminum sheet, the tooling geometry, and the contact conditions (including lubrication), cause multiple buckling modes with different numbers of circumferential waves in close proximity. This suggests that the effective width of the buckled area is influenced by the above-mentioned factors. In fact, the conference organizer reported that the number of circumferential full-waves was found to be \( n=13 \) in their benchmark experiment, whereas our analyses provides \( n=12 \).

4. Conclusion
In this paper, we traced the local buckling behavior of anisotropic elasto-plastic thin shells in Numisheet2014 BM4[2] using the static implicit analysis with artificial damping method. We estimated the buckling mode by eigenvalue analysis and the effective width of the buckled area by the basic buckling theory for shell. However the relationship between the eigenmodes obtained...
from the linear eigenvalue analysis and the deformation modes actually observed in the buckling process requires further investigation.

References
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