# EFFECT ON SIMULATION LUBRICANT MODELING OF MATERIAL FORMING

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**Abstract:** Solid cylinder upsetting is analyzed using three different approaches for frictional boundary condition modeling. These are constant shear friction factor, experimentally measured frictional stresses, and analytical models accounting for lubricant entrapment and redistribution. All three approaches are implemented in the CFORM finite element code. The error between the three approaches and actual experimental measurements of the material deformation and interfacial pressures is investigated. It is shown that the constant shear friction factor is more accurate for solid film lubricants than for liquid lubricants. However, the calculations indicate that if accurate prediction of near net shape forming processes is to become a reality, improvements need to be made in the characterization of frictional boundary conditions. New theoretical developments applicable to arbitrary shapes and more accurate than the constant shear friction approach are needed.

**Keywords:** cylinder upsetting, interfacial pressures

## 1. INTRODUCTION

Frictional forces at the die-workpiece interface play a very strong role in material processing operations. These forces contribute strongly to the deformation characteristics, die wear and energy losses during forming. A common way of reducing the frictional forces is to use a proper lubricant (solid or liquid) between the dies and the workpiece. Due to the importance of these forces, accurate simulation of forming processes requires the use of models that accurately simulate the die-workpiece interface.

Due to the rapid development of computers and numerical methods the finite element method has become popular for the simulation of material forming problems. In some cases it is being used to address design issues associated with near net shape forming. A weakness of most of the finite element codes that have been developed for this purpose is the modeling of the interface conditions between the die and the workpiece. The models implemented in these codes are not representative of the state-of-the-art in the area of lubrication. This weakness has been noted by several researchers. As an example, one author writes [1]:

"Despite the importance of lubrication, workers involved in the design and analysis of metal forming processes are usually relatively un-informed about the subject. It is not uncommon that little or no attention has been given to lubrication in the design process or that sophisticated plasticity theory is combined with naïve assumptions about friction conditions."

Due to the fact that simulation of these processes is gaining widespread use in industry as a design tool for designing complex shapes and near shape forming operations, the implementation of models that are more accurate representations of the interfacial forces attains a greater importance. In the light of the above, the objective of the work described here was to investigate the effects of applying more realistic friction boundary conditions in the finite element analysis of a forging process and to determine the accuracy of the current approach.

## 2. MODEL 3 – THEORETICAL FRICTIONAL MODELING IMPLEMENTED IN THE FINITE ELEMENT ANALYSIS

Analytical modeling of the frictional stresses based on liquid lubricant entrapment and breakdown on the surface of solid cylinders upset between flat dies has been done by several authors [4].Oyane and Osakada derived expressions for the lubricant film thickness $h_d$, and the width of the unlubricated region W, as the lubricant breaks down. These parameters are shown in Fig. 1.

Plastic deformation begins at the center of the workpiece and spreads outward. The pressure distribution on the surface of the workpiece is parabolic with the maximum pressure occurring at the center. Since a depression is formed in the middle, entrapping the lubricant, the material around the depression must rise upward because it is incompressible. However, since the shape of the risen material is difficult to predict, and its hydrodynamic treatment is complicated, Oyane and Osakada further assumed that the surface rise around the depression could be neglected. Therefore, the outer surface is considered flat around the depression.

Analytical modeling of liquid lubricant entrapment and break down for the plane strain case has been done by Wilson [1].

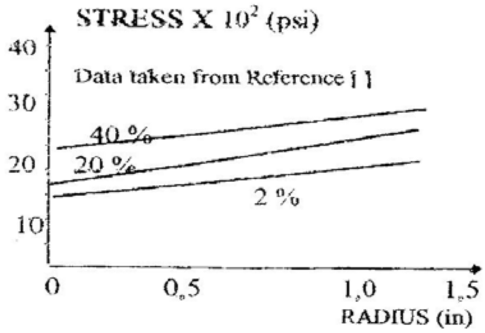


Fig.1 Radial distribution of experimentally measured frictional shear stresses at different reductions on the surface of aluminum cylinders with lead foil as lubricant

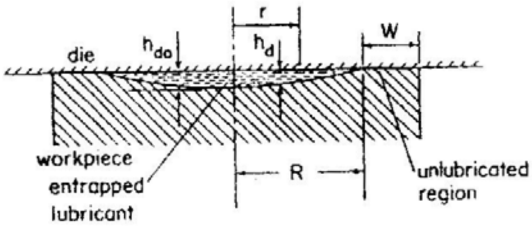


Fig. 2 Thickness of lubricant film entrapped and width of unlubricated region on surface of a solid cylinder upset between flat dies upset

Using Reynold's equation for the fluid between two surfaces, and taking advantage of axial symmetry, Oyane and Osakada calculated the lubricant film thickness at the center of the workpiece at the onset of deformation as

$$h_{d0} = \left( \frac{3\eta V R^2}{\sigma_0} \right)^{1/3}.$$

The dynamic viscosity of the lubricant is $\eta$ and $V$ is the approach velocity of the dies. The radius of the inner edge of the unlubricated zone of the workpiece is denoted by $R$. The boundary conditions require the pressure in the lubricant layer to be zero at the edge and $\sigma_0$ is the initial yield stress of the material at the center of the workpiece. The dimensionless lubricant film thickness along the surface of the workpiece is defined by the relation,

$$H = \frac{h_d}{h_{d0}} = \left[ 1 - \left( \frac{r}{R} \right)^2 + 2\left( \frac{r}{R} \right)^2 \ln\left( \frac{r}{R} \right) \right]^{1/3}$$

where $h_d$ is the thickness of the lubricant layer at any location $r$. The width of the unlubricated zone is given by,

$$W = \left(\frac{1}{1-P}\right)^{1/2} - \left(\frac{1}{1-P}\right)^{1/4}$$

where $P$ is the reduction in height given by

$$P = \frac{Vt}{z}$$

The initial workpiece thickness is $z$ and $t$ is the time measured from the instant the die and workpiece come into contact. The frictional stress on the surface of the workpiece in the lubricated region is,

$$\tau = \frac{\eta \dot{U}}{h_d}$$

where $\dot{U}$ is the workpiece surface velocity in the radial direction. The frictional stress in the unlubricated region is given by, $\tau = mk$ , where $m$ is a constant shear friction factor and, $k = \frac{\sigma_y}{\sqrt{3}}$

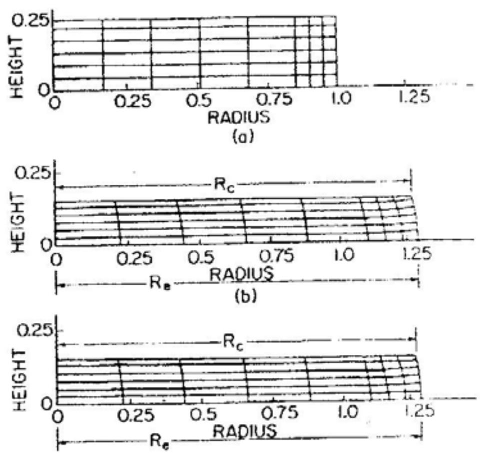


Fig. 3 Upsetting of cylindrical aluminum bilets with lead foil as lubricant. (a) undeformed configuration (uper right hand quadrant), (b) deformed configuration using experimentally determined friction, and (c) deformed configuration using a constant shear friction factor in the analysis

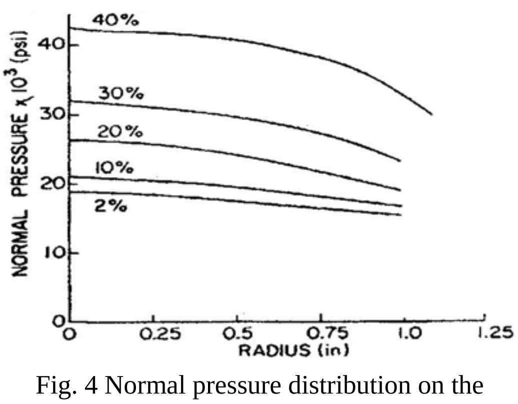


Fig. 4 Normal pressure distribution on the surface of a cylindrical aluminum billet lubricated with lead foil

The value of $m$ chosen depends on the friction condition in the dry region where metal contact occurs.

## 4. RESULTS

All calculations were performed using CFORM (Clemson Forming Process Design and Analysis Code). CFORM is based on the Eulerian coordinate approach. A rigid-viscoplastic constitutive law is used and axisymmetric and plane strain problems may be analyzed. Output includes flow stresses and strain, strain rates, velocities and the pressure distribution on the die. The theory and numerical methods used in the CFORM code, along with verification studies, are detailed in references [1, 2].

Table 2.
Percentage differences between the normal pressure distribution computed by the analyses and the experimental pressure distribution at different reduction (lead foil used lubricant)

| % Deformation | % Diference in Normal Pressures | |
|---|---|---|
| | $F_x^*$ | $F_m^{**}$ |
| 2 | 6,22 | 3,82 |
| 10 | 5,25 | 3,24 |
| 20 | 3,84 | 3,80 |
| 30 | 5,78 | 7,84 |
| 40 | 17,75 | 19,34 |

* $F_x$ = Experimentally Determined Friction Applied in the Analysis

** $F_m$ = Constant Shear Friction Factor Applied in the Analysis

The preceding discussion serves to point out one of the weaknesses of the constant shear friction factor method. Curves similar to Fig. 2 are usually unavailable for the various lubricants and arbitrary workpiece shapes used for forging and these curves are difficult to obtain.

Table 1 compares the differences obtained in the contact radius, $R_c$, and the radius at the equatorial plane, $R_e$, using experimental friction and the empirical constant shear friction factor. As can be seen, the differences are too small to draw conclusions about the advantages of one method over the other.

The contact normal pressures were computed in each case and these have been compared with experimentally measured [3] normal pressure distributions. Table 2 shows the percentage differences between the experimentally determined normal pressure distributions and the normal pressure distributions computed by the finite element analysis. Figure 4 shows the experimentally determined normal pressure distribution at various reductions on the surface of aluminum billets upset with lead foil as lubricant [3]. The results of the finite element analysis in which experimentally determined frictional stresses were used are shown in Fig. 4. The results of the finite element analysis using a constant shear friction factor are shown in Fig. 5. As Table 2 shows, the computed normal pressures are fairly close to the experimental values for the small deformations. At larger deformations they are significantly different from the experimental values. The percentage difference for the analysis using a constant shear friction factor was as high as 19.34 percent. Van Rooyen and Backofen report a reliability of ±7.0 percent for the normal stress. Note that the friction hill near the center of the workpiece predicted by analytical derivations [3] is confirmed by the finite element analysis as shown in Fig. 5 and 3.

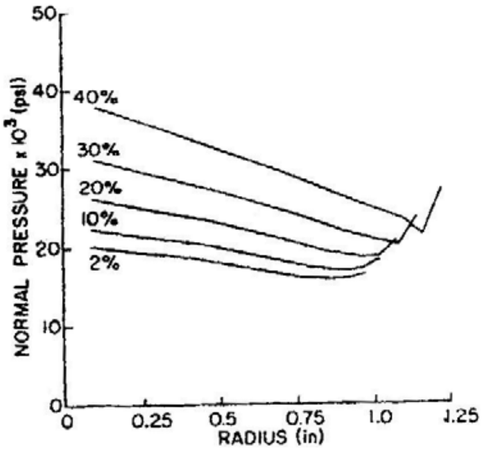


Fig.5 Normal presure distribution on the surface of a cilyndrical aluminum billet lubricated with lead foil, computed by the analysis using a constant shear friction factor

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