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Lubrication in tube hydroforming (THF) Part I. Lubrication mechanisms and development of model tests to evaluate lubricants and die coatings in the transition and expansion zones

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Abstract

The lubrication mechanisms that occur at the tool–workpiece interface for the transition and expansion zones are discussed. Suitable lubrication systems for the transition and expansion zones are reviewed based on the mechanics of deformation and material flow at the interface. Details of two model tests for evaluating the performance of tube hydroforming (THF) lubricants and die coatings are given. The optimization of die geometries for the model tests is based on sensitivity analysis through the finite element method together with experimental verification. The details of these tests are given and their development is discussed.

Keywords: Tube hydroforming; Lubrication; Finite element method

1. Introduction

Improvement of tribological conditions in tube hydroforming (THF) is extremely important, since the tribological factors highly influence part failure due to wrinkling, buckling, premature failure, uneven wall thinning, unacceptable surface integrity and part tolerance.

The lack of the knowledge pertaining to tribology of THF process has hindered optimal utilization of this technology in the manufacturing of complex parts despite the enormous advantages (light-weight structures, part consolidation, etc.) this technology offers over conventional stamping and welding operations.

It is therefore, necessary to investigate the tribological phenomena in THF, and accordingly, develop lubrication systems to enhance tribological performance. Development of robust lubrication systems will however, be achieved by considering (a) the dynamics and mechanics of the THF processes, (b) lubricant formulations (environmentally friendly products), (c) the physical–chemical properties of the interface, (d) the design of mating surfaces (die, coatings and tube surface textures) for lubricant retention and reducing surface wear and adhesion, and (e) development of lubrication tests that can emulate the realistic conditions occurring in THF.

2. Lubrication mechanisms in THF for the transition and expansion zones

In a typical THF process, the tube is first pressurized followed by feeding the material from the guiding zone to the transition and expansion zones (Fig. 1). To produce a sound part, a good combination of material feed and pressure is needed. These variables (pressure loading and feed), however, depend on the interface friction.

At different friction zones, the material flow, the relative velocity between die and tube and the state of stress are different. Therefore, different lubrication mechanisms are also expected. In the case of liquid lubrication, micro-plasto hydrostatic (MPHS) and micro-plasto hydrodynamic (MPHD) lubrication mechanisms can easily be encountered at the guiding zone. MPHD lubrication mechanism, which occur when the trapped lubricant permeates to the real contact surface, has a tremendous advantage as it can lower interface friction considerably.

Contrary to the guiding zone, the deformation mechanics at the transition and expansion zones seem not to favor the occurrence of MPHS and MPHD lubrication. This is due to the drop of the relative velocity between the tool and

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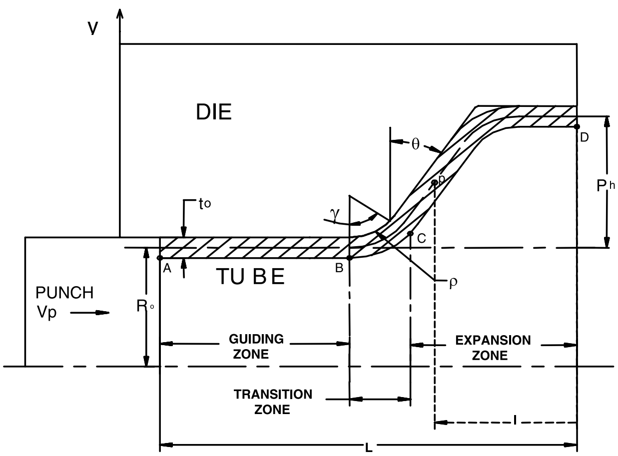

Fig. 1. Friction zones in THF.

tube with increase in the instantaneous radius of the expanding tube towards the expansion zone. In addition, a certain amount of lubricant trapped in the "pocket" of workpiece surface may escape the contact zone as the material enters the transition or expansion zones.

We can estimate the pseudo-tool–tube interface velocity distribution at the transition and expansion zone by assuming that (a) the volume flow rate of material from the guiding zone to the expansion zone remains unchanged and (b) the wall thickness distribution remains the same along all friction zones. In reality, uneven wall thinning and thickening can be encountered depending on the material flow and the level of interface friction. Due to geometrical differences, the velocities will vary from one region to the other.

Fig. 2(a)–(c) show longitudinal cross-section elements at the guiding, transition and expansion zones exhibiting MPHS lubrication mechanism. The MPHS lubrication mechanism can occur at the tool–workpiece interface when the local pressure q is large enough to generate hydrostatic effect, then the normal load (F_N) is carried by both the asperity contacts and trapped lubricants.

Based on the punch velocity V_p, material flow at the guiding, transition and expansion zones can be given by Eqs. (1)–(3), respectively:

$$\dot{m}_g = 2\rho V_p \pi t R_0 \quad (1)$$

$$\dot{m}_{tr} = 2\rho V_{tr} \pi t r_{tr} \quad (2)$$

$$\dot{m}_{exr} = 2\rho V_{exp} \pi t r_{exp} \quad (3)$$

where V_p is the velocity of the punch, and V_{tr} and V_{exp} the pseudo-velocities at the transition and expansion zones. Equating the material flow for the guiding, transition and expansion, the pseudo-velocities at the transition (V_{tr}) and expansion (V_{exp}) zones can be estimated by

$$V_{tr} = \frac{R_0}{r_{tr}} V_p \quad (4)$$

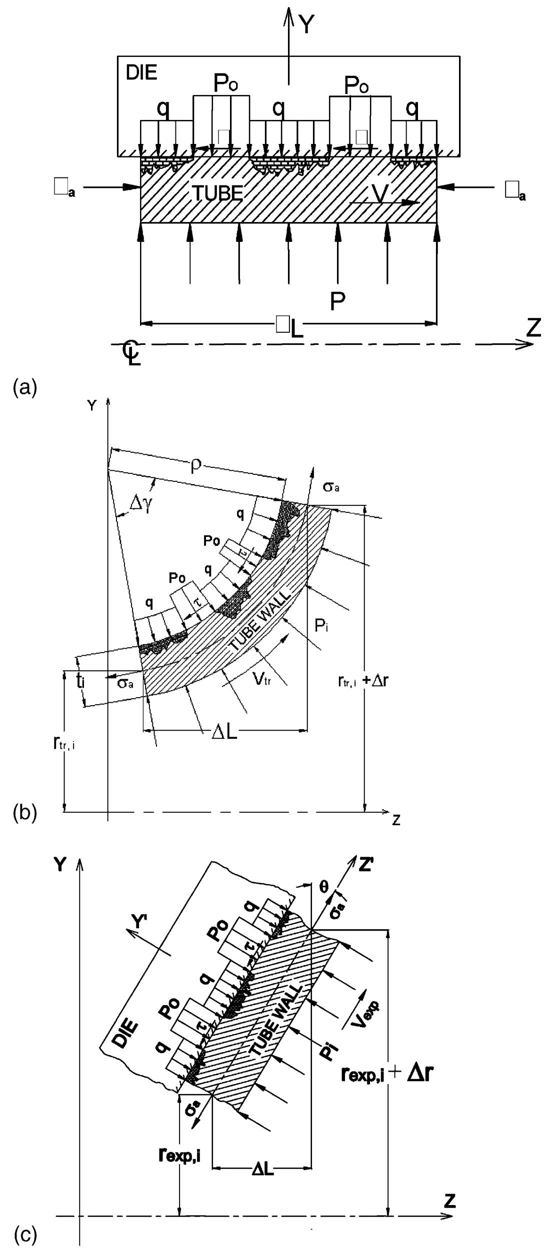

Fig. 2. MPHS lubrication: (a) an element at the guiding zone; (b) an element at the transition zone; (c) an element at the expansion zone.

$$V_{exp} = \frac{R_0}{r_{exp}} V_p \quad (5)$$

where R_0, r_{tr}, and r_{exp} are the tube radius at the guiding, transition, and expansion zones, respectively. Eqs. (4) and (5) show that the pseudo-velocities for the expansion and transition change with die geometry, and may acquire very low values compared to the velocity at the guiding zone making the occurrence of MPHD difficult.

Experimental evidence and mathematical models [1,2] have shown that relative velocity between tool and workpiece, v, and viscosity, η, are the most significant parameters for the occurrence of MPHD lubrication. The Reynolds equation, Eq. (6), is commonly used to determine the occurrence of MPHD:

$$\frac{dq}{dz} = 6\eta v \frac{h - h_m}{h^3} \tag{6}$$

where q is the local pressure, $h(z)$ the lubricant film thickness, η the viscosity, and v the relative velocity between die and tubular workpiece.

Eq. (6) shows that the local pressure, q, acting on a lubricant pocket increases with increasing tool–workpiece relative velocity, v, and lubricant viscosity, η. Therefore, low velocities encountered in the transition and expansion zones hinder the occurrence of MPHD lubrication. This suggests that liquid lubrication might not be a better choice for these friction zones.

The mechanics of deformation shows that surface expansion is dominant in the expansion zone. Therefore, dry film lubrication is more appropriate for this zone. More importantly, the lubricant needs to adhere firmly into the tubular specimen so that it can follow the surface expansion without breaking down. For dry film lubrication, the interface friction is dependent on the shear strength of the lubricant film adhered on the tube surface.

The difference in the lubrication mechanisms between the guiding, transition and expansion zones shows that different tribo tests that emulate realistic conditions in THF are absolutely necessary for identifying best lubricants and die coatings. It is the subject of this paper to present two model tests for the evaluation of lubricant and die coating performance at the transition and expansion zones. A test for evaluating the lubricants for the guiding zone, Fig. 1, is described elsewhere [3].

3. Model tests to evaluate the performance of lubricants and die coatings in the transition and expansion zones

The most reliable way to evaluate lubricants for a metal forming process is to test the lubricant during the actual THF operation. Thus, all relevant variables such as interface pressure as a function of pressure loading path, axial feed rate, die and tubular materials, etc. correspond to real conditions. However, testing a lubricant under production conditions require the measurement of a large number of variables such as interface pressure, tube wall thickness distribution, amount of galling, surface topography, surface coatings, etc. Taking these measurements at the plant level could be difficult and expensive. In addition, if the tested lubricant does not perform well, it may have a detrimental effect upon the product quality and the tooling. Thus, lubricants should be tested in production only if they have been proven to be effective in laboratory tests that emulate production conditions.

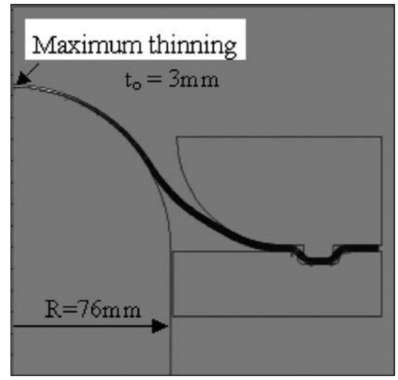

Fig. 3. Location of maximum thinning for $\mu = 0.0$.

4. Potential of limiting dome height (LDH) test for lubricant evaluations at the transition zone

The LDH test can be used to evaluate lubricant performance at the transition zone due to the similarities in the mode of material deformation encountered in the LDH test and that encountered in the transition zone of a THF process. In both, the LDH test and the transition zone of THF process, the blank is undergoing stretching and sliding along the die or punch surface.

The LDH test was essentially developed for testing formability of sheet metals. In this test, a round blank is held firmly around the periphery while a spherical punch is used to stretch form the blank, Fig. 3. The dome height depends on the ability of the material to distribute strain and on the limiting strain level. In the literature, there are many reports of correlation between the test and industrial applications [4,5]. This test, however, lost popularity in industry as a formability test because of lack of reproducibility despite its proven correlation with press performance [6]. One of the main factors leading to inconsistency of material formability results is the variation of interface friction encountered from one test laboratory to the other. This effect can, however, be exploited and make LDH a suitable test for lubricant evaluation.

We assume that the friction stress (τ_s) at the interface obeys Coulomb's law, i.e. $\tau_s = \mu P$, where μ is the coefficient of friction and P the interface pressure. Good estimation of friction coefficient in THF (transition zone) using LDH test will depend much on how close is the interface pressure induced by the LDH test to that encountered in the transition zone of THF process.

4.1. FEM simulations of LDH test results and discussion

The FEM simulations for LDH conducted have shown that this test is very sensitive to friction [7]. FE simulations were conducted in order to study the interrelationships of interface friction, forming load, interface pressure, and geometric variables. Friction coefficients of $\mu = 0.0, 0.05, 0.075$, and 0.10 were used in the simulations. AISI 1035 sheet material

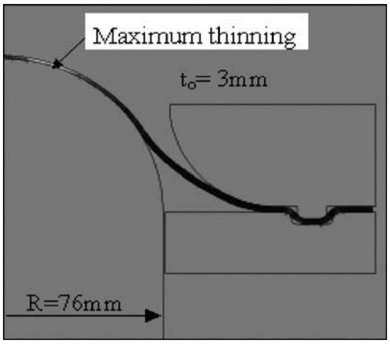

Fig. 4. Location of maximum thinning for $\mu = 0.10$.

with a thickness of 2, 3, and 4 mm were used. The punch radius was 76 mm and die corner radius was 50 mm (Fig. 3).

4.2. Friction sensitivity to geometric variables and forming load in LDH test

Fig. 3 shows the LDH FE simulations when a friction coefficient of $\mu = 0.0$ was applied. As expected, under this condition, the maximum thinning of the dome wall is observed to occur at the apex of the dome. A 3 mm steel sheet was used for this simulation. Changing the interface friction to $\mu = 0.075$ caused the location of maximum thinning to shift 20 mm away from the apex (Fig. 4). The general trend of how the location of maximum thinning changes with change in interface friction is shown in Fig. 5. Friction on the punch surface hinders free thinning at the apex and therefore, the position of maximum strain, which corresponds to the location of maximum thinning, moves away from the apex. For a friction range of $\mu = 0.00$–0.10, the location of maximum thinning moved a distance of about 27 mm away from the apex of the punch, with the punch radius of 76 mm.

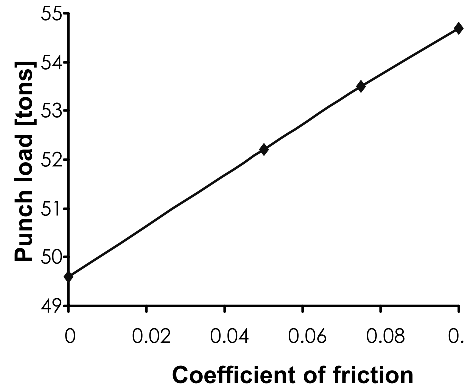

Fig. 6. Influence of friction on punch load.

The FE simulations show a small increase in the forming load with increase in the interface friction (Fig. 6). A blank of 3 mm thickness (AISI 1035) was used for this case. This shows that the interface friction is not sensitive to the forming load.

By exploiting the sensitivity of friction to the geometrical variables observed in Fig. 5, the LDH test can be used to rank lubricant performance with ease. Furthermore, by combining FE simulations and experiments, friction coefficients for tested lubricants can be estimated.

5. Development of a model test to evaluate the performance of lubricants and die coatings in the expansion zone

Many parts in THF undergo calibration process. In order to establish a model test for the expansion zone, which will be used to rank lubricants and determine the coefficient

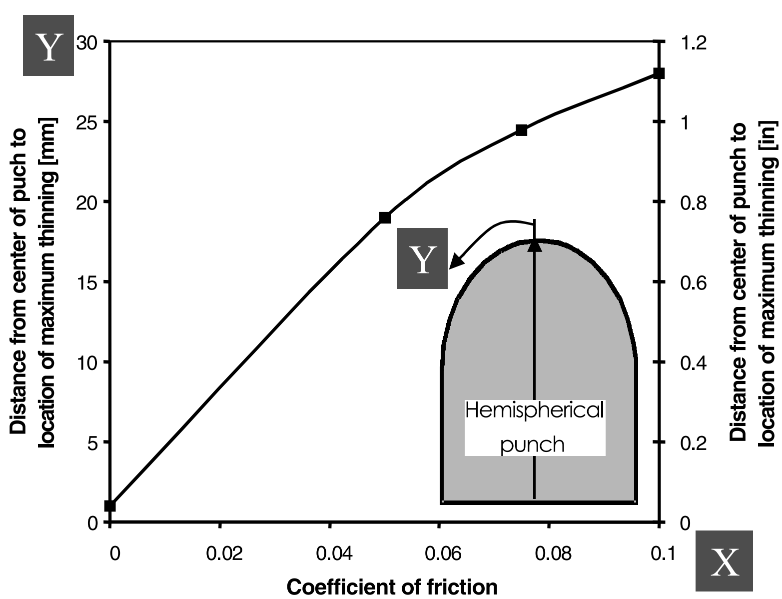

Fig. 5. Influence of friction on the location of maximum thinning.

of friction, several requirements must be fulfilled: (a) the model test must emulate the conditions in the expansion zone and (b) the test must give discriminatory results for different coefficients of friction. Based on plasticity theory and the finite element analysis, we studied the interrelationships between geometric and process variables pertaining to tube expansion of different cross-sections. Emphasis was given to major variables that influence tribological aspect in THF such as (i) the influence of the pressure loading path on wall thickness distribution, (ii) the influence of friction on the wall thickness distribution, (iii) the influence of friction on die corner fill, and (iv) influence of friction on the pressure at which necking starts.

For a lubrication test to be discriminating enough, the die geometry should allow a high increase in the outside surface of the tube. This is necessary, because thinning of the lubricant will increase with increase in the surface area. More importantly, the die geometry should allow the tube material to slide over the surface of the die. FEM simulations were used in the determination of the best die geometry. The die geometrical configurations considered in this study are given in Fig. 7. The finite element software DEFORM 2D and PAMSTAMP were used for the simulations.

5.1. Square die geometry

Attempt to model friction test using square die geometry has been reported in [8]. This die geometry was used here as a base for comparison with other geometries regarding sensitivity to friction. If the tube were to fill the square cavity completely, the perimeter would increase by 27% (Eq. (7)). This value gives an indication of maximum contact area of the tube which will interact with the die and hence provide us with tribological information:

$$\frac{\text{perimeter}_{\text{square}}}{\text{perimeter}_{\text{tube}}} - 1 = \frac{4D}{\pi D} - 1 = 0.27 \qquad (7)$$

5.2. Triangular die with flat surfaces

The basis for considering this geometry is the fact that there is a significant increase in the surface area or perimeter in the cross-section as compared with a square die (Eqs. (7) and (8)). If the tube were to fill the triangle cavity completely, the perimeter would increase by 67%. A tube diameter of 57 mm was used. This is twice as large as that obtained with the square die. It should, however, be noted that 67% refers to gross surface area increase. In reality, local surface area increase is dependent on the flow mode of material on the die geometry and material properties such as strain hardening [9]. Therefore, thinning of lubricant will depend on the local surface expansion. The difference in the surface area or perimeter for a square die and equilateral triangular die suggest that the triangular die is superior to the square die:

$$\frac{\text{perimeter}_{\text{triangle}}}{\text{perimeter}_{\text{tube}}} - 1 = \frac{3L}{\pi D} - 1 = \frac{3D\sqrt{3}}{\pi D} - 1 = 0.66 \qquad (8)$$

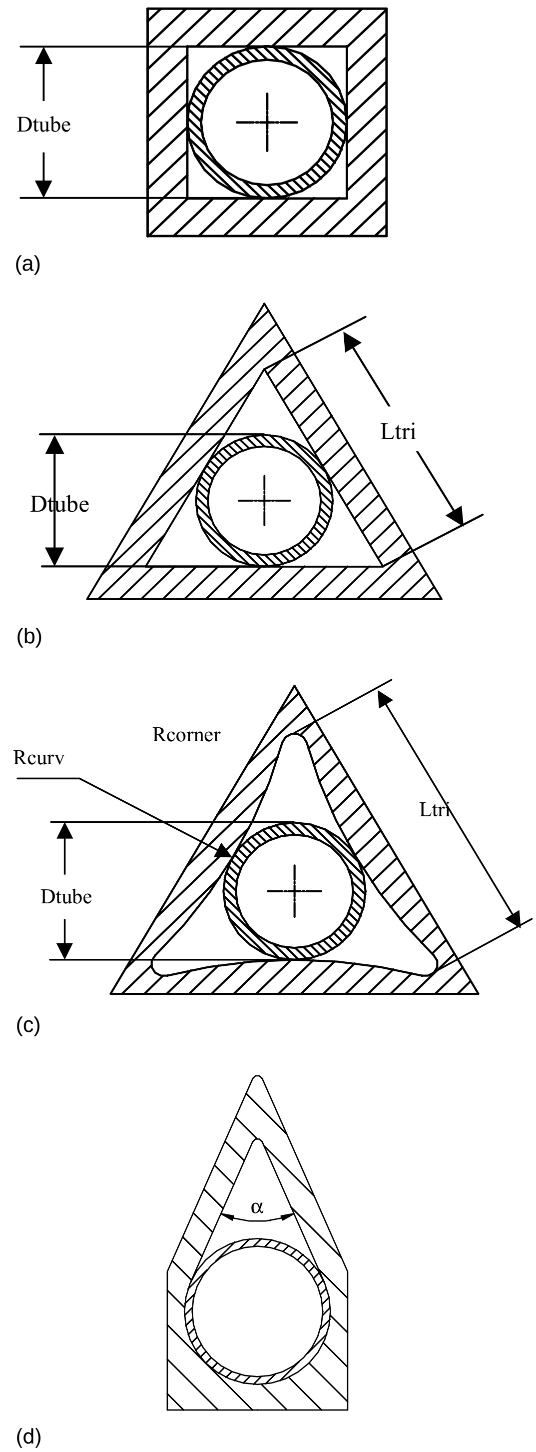

Fig. 7. Die geometrical configurations: (a) square die; (b) triangular die with flat surfaces; (c) triangular die with curved surfaces; (d) pear-shaped die.

5.3. Triangular die with curved surfaces

The curved surface triangular die allows a higher increase of the surface area (or perimeter in cross-section) than the triangle. For the geometrical variables of $R_{curv} = 200$ mm, $D_{die} = 57$ mm, $R_{corner} = 5$ mm and $L_{triangle} = 115$ mm, the perimeter of the curved surface triangle increases up to 93%:

$$\frac{\text{perimeter}_{\text{curv–triangle}}}{\text{perimeter}_{\text{tube}}} - 1 = 0.93 \quad (9)$$

By varying the radius of curvature R_{curv}, corner radius R_{corner} (Fig. 7), the increase in perimeter can be varied and also different configurations of the die can be obtained. It is anticipated that with this geometry, the change in the thickness distribution, corner fill and corner radii will be affected by the friction in a larger degree than with the triangular die with flat surfaces. Note that the perimeter increase given in Eqs. (7)–(9) are for quantitative purpose to select the best geometry. The feasibility of expanding the tube to fill the die cavity will depend on the formability of the tubular material and the size of tube wall thickness used.

5.4. FE simulations

FE simulations, using the commercial code DEFORM 2D, were first conducted for the three geometries, square, triangular dies with flat surface and triangular die with curved surface. Friction coefficient were varied from $\mu = 0.04$ to 0.1. The aims of the simulation were to (a) establish relationship between friction, interface pressure and geometric variables and (b) select the best die geometry, i.e. the most discriminating one.

FE simulations were conducted based on the following conditions: plane strain; the number of elements in the tube is 600; the material is SS304; tube o.d. = 57 mm; tube wall thickness = 2 mm; maximum pressure to the tube = 1200 MPa.

5.4.1. FE simulation results for square, triangular and curved die geometries

The summary of FE simulation results is given in Table 1. The change in the corner fill ΔA is obtained by subtracting the corner fill A for $\mu = 0.04$ from corner fill A for $\mu = 0.1$. Similarly, the change in maximum thinning Δtw is obtained by subtracting the maximum wall thinning tw by subtracting the corner fill A for $\mu = 0.04$ from corner fill A for $\mu = 0.1$.

Table 1
Sensitivity of friction to geometric variables for three die geometries[a]

Die type	Change in the corner fill ΔA from $\mu = 0.04$ to 0.1 (mm)	Change in maximum wall thinning Δtw from $\mu = 0.04$ to 0.1 (mm)
Square	–	0.07
Triangular with flat surfaces	0.6	0.08
Triangular with curved surfaces	0.7	0.09

[a] Corner fill A can be seen in Fig. 9.

It can be observed that the change in both corner fill and maximum thinning for the interface friction range of $\mu = 0.04$–0.1 are very small (Table 1). Thus, it will be difficult to model friction test with these geometries.

5.5. Pear-shaped die

The unique feature for the pear-shaped die geometry (Fig. 1) is that the material is confined to flow in one direction only. With square and triangular dies, the material was confined to flow to all the die corners. Thus, with the pear-shaped die, more material is expected to flow towards the expansion zone. FE simulations were conducted for three different die geometries with die angles of $\alpha = 90^\circ$, 58°, and 48°. From the simulation results, wall thickness distributions, corner fill and pressure at necking were determined. The FE simulation results for a friction range of $\mu = 0.0$–0.15 are summarized in Table 2.

5.5.1. Pear-shaped die with die angle $\alpha = 58^\circ$

The die angle in this geometry is chosen such that the tube can expand about 25 mm towards the die corner. Simulation conditions similar to the ones used in the pear-shaped die with the die angle of $\alpha = 90^\circ$ were used.

5.5.1.1. Wall thickness distribution. Fig. 8 shows the wall thickness distribution for $p_i = 200$ MPa. A considerable increase in the difference in the wall thickness for different coefficient of friction is observed with this geometry. The difference between $\mu = 0.04$ and $\mu = 0.1$ is now 0.2 mm which is 100% more compared to the results obtained with triangular dies. However, the absolute value itself is still very low.

5.5.1.2. Corner fill. The chart for the corner fill given in Fig. 9 shows that distance to the die varies over 2 mm for

Table 2
Sensitivity of friction to geometric variables for pear-shaped die geometries with different die angles (200 MPa)

Die type	Change in the corner fill ΔA from $\mu = 0.04$ to $\mu = 0.1$	Change in maximum wall thinning Δtw from $\mu = 0.04$ to $\mu = 0.1$	Change in pressure at necking from $\mu = 0.04$ to $\mu = 0.15$
Pear-shaped die with $\alpha = 90^\circ$	0.04 mm	0.14 mm	–
Pear-shaped die with $\alpha = 58^\circ$	2.0 mm	0.2 mm	–
Pear-shaped die with $\alpha = 48^\circ$	Necking occurred	0.6 mm	180 MPa

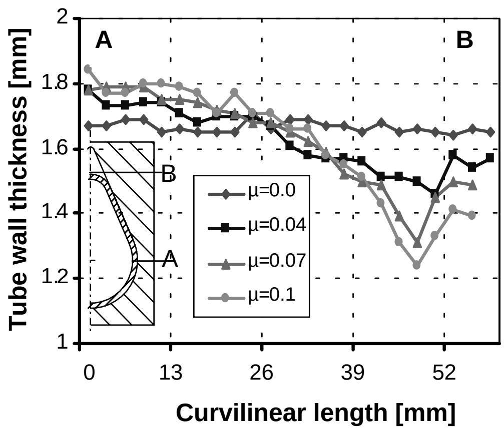

Fig. 8. Pear-shaped die with $\alpha = 58^\circ$ wall thickness distribution ($p_i = 200$ MPa).

a coefficient of friction range of $\mu = 0$–0.1. This result indicates that it is possible to develop a practical expansion test using the pear-shaped dies because the influence of geometrical variable on friction is measurable. The results demonstrate that the size of the die angle has a significant influence on the flow of material to the die corner under the pear-shaped die geometry.

5.5.2. *Pear-shaped die with die angle* $\alpha = 48^\circ$

Similar FE simulation conditions as the ones used for the pear-shaped dies with $\alpha = 90^\circ$ and 58° were used.

5.5.2.1. Wall thickness distribution. Fig. 10 shows the wall thickness distribution for 200 MPa. The wall thickness distribution shows a similar trend like the other simulations. But the difference in the wall thickness is now large, compared to the other die geometries. For a coefficient of friction range of $\mu = 0$–0.1, wall thickness changed by about 1.2 mm.

It can also be observed in Fig. 10 that with a coefficient of friction $\mu = 0.07$, necking occurred. This suggests that necking can also be used as a criterion for the evaluation of lubricants.

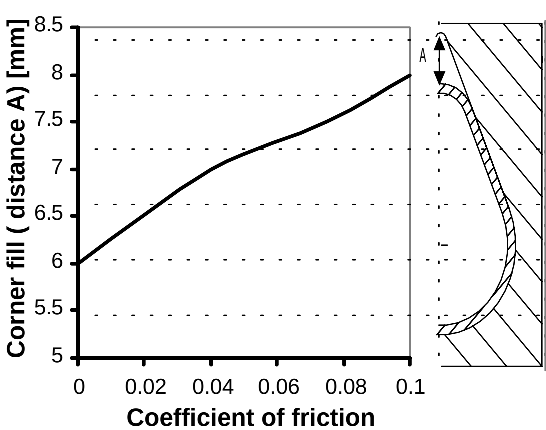

Fig. 9. Pear-shaped die with $\alpha = 58^\circ$ corner fill ($p_i = 200$ MPa).

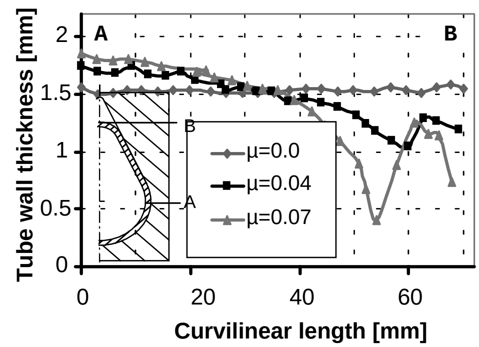

Fig. 10. Pear-shaped die with $\alpha = 48^\circ$ wall thickness distribution ($p_i = 200$ MPa).

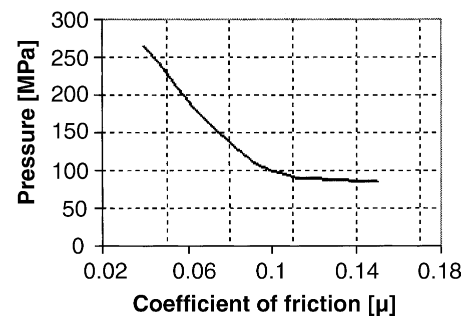

Fig. 11. Bursting (necking) pressure for different coefficients of friction.

5.5.2.2. Pressure at necking (bursting pressure). Another way to rank different lubricants is to use the bursting (or necking) pressures. Fig. 11 shows the bursting pressure over the coefficient of friction. It should be noted that, bursting of tube is not given by DEFORM 2D, we, therefore take pressure at necking as equivalent to pressure at bursting. Fig. 11 shows a significant increase in bursting pressure with increase in the coefficient of friction. The pressure gradient indicates that, pressure can be used as a criterion for the evaluation of lubricants using the pear-shaped die geometry.

5.6. *Preliminary test results*

Based on pear-shaped FE simulations, the tooling was manufactured. Details on the pear-shaped tooling will be given in Part II of this paper series. Fig. 12 shows the change in protrusion heights for lubricated and non-lubricated

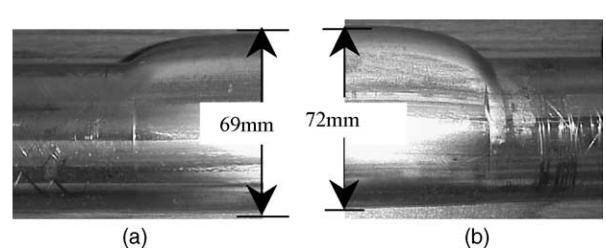

Fig. 12. Protrusion height for tube specimens after test: (a) non-lubricated; (b) lubricated.

samples. The test results show good agreement with FE simulation.

6. Conclusions

Lubrication mechanisms and their effect on interface friction have been discussed. It is observed that due to difficult of attaining hydrodynamic effect at the transition and expansion zone, a dry film lubrication seem to be the best choice for these region.

FE analysis of LDH test have shown that, this method (LDH) is friction sensitive and can be applied for performance evaluation of lubricants and die coatings for the transition zone, provided the pressure induced at the interface is similar to the ones in THF operation. Variation of the thickness of the blank is used for adjusting the pressure levels.

In order to come up with the best model test for evaluating lubricants and die coatings in the expansion zone, different die geometries were investigated by the FEM. These geometries included (a) square die, (b) triangular die with flat surfaces, (c) triangular die with curved surfaces, (d) pear-shaped die with a die angle of 90°, (e) pear-shaped die with a die angle of 58°, and (f) pear-shaped die with a die angle of 48°.

The pear-shaped die (die angle $\alpha = 48°$) has shown to be the best for building a tube expansion test model. With this geometry, lubricants' die coatings can be ranked and the apparent friction coefficient can be determined based on four criteria: (a) corner fill measurement, (b) bursting pressure (pressure at necking), (c) wall thickness, and (d) corner radius.

The preliminary test results from pear-shaped friction tooling have shown good agreement with the FE simulation. Detailed experimental results for low carbon steel and stainless steel tubes will be given in Part II of this series on "Lubrication for THF". LDH test results will also be presented in Part II.

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