An Experimental-Numerical Study on Curling Process for Metallic Caps Production

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Abstract. In the field of food industry, adoption of metallic caps for the storage of agri-food products is widespread. This type of caps is obtained through a sequence of forming processes executed on a tinplate. First, the plane metal sheet is deep drawn to obtain a cup whose edge is slightly shaped inward, then a curling process forms the edge into a hollow ring. This paper is focused on the curling, which allows to assess the final cap shape immediately before the application of the cups on the jars. Since the process is generally made on packaging lines with high production rates, the design of a proper curling machine needs to consider the effect of fatigue as well as maximum load values. Because of the different cap dimensions and shapes contemporary worked on the line, the loads definition is not easy. The scope of this work is the development of a model for the prediction of curling loads for different cap shapes and sizes. The analytical solutions, available in literature, together with numerical simulations of the process were compared to experimental compression tests made on two different caps type. Considerations on caps geometry and size influence on curling force represent a contribution to the fitness for service evaluation as well as a starting point for the design of new curling machines.

1. Introduction

Metallic capsule processing for food industry is a long-established practice that interests a large variety of products according to the type of materials and closing system, as for example eurocap, snap-on and twist-off capsule. The term “twist-off” is used to denote special lug caps for glass jars or bottles made of printed or lacquered tinplate. Compared to conventional screw caps, these lug caps, clamped under spring tension, bring about a particularly tight closure of the glass containers. A polymeric sealing compound is injected in the cover area opposite the mouth edge of the glass to ensure a tight seal.

This type of caps is manufactured on a fully automatic production line. As reported in Figure 1, the manufacturing process starts from the plane metal sheet cutting, then a deep drawn process allows to obtain a cup shell with straight edge. In the following steps, caps edge is cleaned and slightly shaped...
inward before the execution of the curling in which the edge is formed into a hollow ring. A last process produces the lug that allows mechanical coupling between caps and jars thread.

![Figure 1](image1.png)

*Figure 1*. Sequence of curled capsule production phases, from metal sheet to lug forming.

Because of the complex load combination, repeated with high frequency in the time, the machine is subjected to an intense fatigue regime. The field experience shows as the heaviest load of the process is represented by the curling load that is executed by a punching mechanism like that reported in Figure 2 (a). The mechanism is made by a groove track carved on a wheel profile in which a pin is forced to move. Wheel rotation forces an alternating motion of a shaft on whose end a punch is placed. As shown in Figure 2 (b) the printed workpiece, put in a fixed die, is constrained by a holder and the punch is forced on its free edge. The punch cavity, hemispheric or toroidal, is characterized by its radius and angular extension.

![Figure 2](image2.png)

*Figure 2*. Example of curling machine mechanism (a) and schematic of die, punch and workpiece coupling (b).

Since curling process consists in a controlled plastic collapse of the shell edge, the maximum curling force depends on material properties, shell geometry, punch cavity shape and punch to workpiece friction properties. Generally, a numerical study can help to understand the deformation mechanisms of the manufacturing process [1]. Masmoudi et al. [2] experimentally and numerically investigated the external curling of thin tubes. By forcing copper tubes on a conical die, they obtained a load-displacement curve that divided in characteristic stages associated with tube deformed profile. In a first phase, when the contact was between the tube bottom edge and die interface, load value increased rapidly until a maximum corresponding to the tube end plastic bending. In this phase they found the role of friction resulted fundamental for the maximum load magnitude. The following plastic collapse of the edge on the die conical surface, was associated with a progressive falling in load curve until a minimum value corresponding to the first detaching between tube-end and die surface. Subsequently the inversion process proceeded with the slipping of tube lateral face on the die surface, which caused a new increasing in load curve. In this condition friction was found less significant in relation to the load values, although it affected the final tube deformed shape. Still on external tube curling, the effect on die geometry was
studied by A. EI-Domiaty [3] that investigated both conical and flat die. Independently by the die shape a critical punch entrance radius can be defined as function of tube geometry, material and friction conditions. Design curves, normalized respect to the material, were presented to find the critical radius respect to tube geometry. An analytical formulation to define punch entrance radius was proposed by You-Min Huang et al [4] that resulted from the combination of tube geometry and extreme values of radius corresponding to minimum value of punch radius required to flare, and maximum value of punch radius required to curl. Although the mentioned studies provided evidence on process dynamics and punch design, few investigations about internal curling process of printed workpiece were made. The influence of the mentioned process parameters on the curling load as well as its variation with the capsule geometry, results not investigated. Nevertheless, estimation of internal curling load in general cases can be found in literature. V. Boljanovic et al. [5] estimated the force needed to produce a circular edge of a initial deep-drawn cylindrical workpiece by assuming the formation of plastic hinge caused by the application of bending moment to the curled material. This hypothesis provides an easy approximated solution for curling force definition. Since maximum machine force is generally much greater than the force needed to execute the curling process, more precise formulation is not needed from technological point of view even because curling machines are usually designed as displacement-controlled mechanism. On the other hand, this approach does not consider the influence of the production ratio on material fatigue to which the machine is subjected. In this regard a more precise definition of curling force is needed. A.N. Kinkade [6] deeply investigated both external and internal quasi-static inversion mechanism of tubes. Starting from an energetic approach, he proposed an analytical solution taking account of all the material deformation contributes and friction forces. Obtained results were very well correlated with all published tests. The aim of the present work is to provide a simple method for the maximum curling load definition through experimental compression tests, numerical simulations and an analytical approach.

2. Materials and Methods

2.1. Experimental Methods

Experimental campaign was conducted with the purpose to reproduce inward curling of caps edge through quasi-static compression tests (Figure 3).

![Figure 3. Experimental test equipment (a) and detailed identification of capsule, die and punch (b).](image-url)

The geometry of the tested capsule is reported in Figure 4 and the corresponding dimensions for both the type of capsules adopted can be found in table 1.
Figure 4. Sketch of caps geometry before and after curling process.

Both the caps type, which will be identified as Type-A and Type-B, were made of the same TH415CA tinplate material with a thickness $t$ equals to 0.18 mm. The two type of caps differ in diameter $D$, initial height $h_1$ and final height $h_2$. Each type of cups was tested under two different compression velocity, 1 mm/min and 10 mm/min respectively.

| Type | $D$ [mm] | $h_1$ [mm] | $h_2$ [mm] | $t$ [mm] | $d$ [mm] |
|------|----------|------------|------------|---------|---------|
| Type - A | 82 | 14.2 | 10.7 | 0.18 | 1.8 |
| Type - B | 58 | 12.4 | 9.5 | 0.18 | 1.8 |

Tested caps were cut on a meridian plane and the section acquired through a ‘Lext Olympus OLS 5000’ con-focal microscope. Measurements on section capsule were compared to the industrially processed capsule dimensions with the purpose to validate the experimental campaign. Analysis on the load vs displacement curves were conducted and the maximum value of load was estimated both for type A and B capsules.

The mechanical properties of TH415CA material tin plate, in terms of yield strength $\sigma_y$, tensile strength $\sigma_R$, modulus of elasticity $E$ and Poisson’s ratio $\nu$ are reported in Table 2.

| $\sigma_y$ [MPa] | $\sigma_R$ [MPa] | $E$ [GPa] | $\nu$ |
|------------------|------------------|----------|-------|
| 415              | 450              | 200      | 0.29  |

2.2. Numerical Model
To investigate the deformation process involved in curling and justify the load vs displacement curve experimentally obtained, a 2D axisymmetric FEM model was developed. By adopting 1260 quad full integration elements, a half meridian section of the capsule was modelled as an elastoplastic isotropic material with TH415CA material properties while the punch and die were modelled as rigid parts.

Constraining condition were imposed by clamping the die and by suppressing all the degrees of freedom, except for the displacement in compression direction, to the punch. The boundary conditions on the capsule consisted in the imposition of a surface to surface contact formulation among capsule, die and punch. Since both the industrial curling process and compression tests were conducted by adopting any kind of lubrication, the friction coefficient in numerical model was set equal to 0.3 as
estimated by [2]. The inward curling of the capsule was obtained by imposing to the punch a displacement equal to the caps free edge difference of height \( s = h_2 - h_1 \) after and before the process.

The outcomes of simulations were compared to the experimental results in terms of load vs displacement curve and value of maximum load both for type A and B capsules. A 3D swipe of the capsules model in three steps of deformation process is here reported in Figure 5.

![Figure 5. Capsules numerical model 3D swipe before curling process (a), during process (b) and completely curled (c).](image)

2.3. Analytical Methods

An analytical approach to the problem was formulated to extrapolate the maximum curling force through an energetic approach. A perfectly plastic and isotropic material behaviour was assumed as experimentally measured, the invariance of capsule wall thickness (\( t \)) during the deformation was assumed as well as the constancy of the total material volume involved in process.

The total deformation work \( W_{\text{tot}} \), generically defined by the volume integral of the product of stress \( \sigma \) and deformation \( \varepsilon \) regimes, has been considered as the sum of two main contribution.

\[
W_{\text{tot}} = \int \sigma \cdot \varepsilon \, dv = W_b + W_c \tag{1}
\]

First contribution is the meridian bending deformation work \( W_b \) and the second is represented by the circumferential diameter reduction work \( W_c \) (Eq.1). The work contribution in meridian bending was evaluated by considering an angular extension \( \beta \in [0, \pi/2] \) of punch profile, characterized by the radius \( r = d/2 \), which is equal to half of the external curl diameter \( d \). In Figure 6 a schematic half section of capsule wall is reported.

![Figure 6. Capsule meridian half-section scheme.](image)
Meridian bending was studied by supposing that material was forced to assume the punch shape. Deformation regime through the capsule wall thickness was considered as constant and equal to the maximum value measured at the outside of the wall, as reported in Eq. 2

\[
\varepsilon_b = \frac{\Delta l}{l} = \frac{2\pi \left( r + \frac{t}{2} \right) - 2\pi r}{2\pi r} = \frac{t}{2r}
\]  

(2)

Considering the stress constant and equal to the yielding point \( \sigma_Y \), the bending work can be expressed in function of \( \varepsilon_b \) an \( \beta \), as reposted in Eq. 3.

\[
W_b = \sigma_Y \int \varepsilon_b dv = \frac{\sigma_Y \pi D \beta t^2}{2}
\]

(3)

The corresponding values of load were estimated as the derivative of the work respect to the punch angle \( \beta \), as reported in Eq. 4.

\[
P_b = \frac{dW_b}{d\beta} = \frac{\sigma_Y \pi Dt^2}{2r}
\]

(4)

Where the material direction flow \( s \) was expressed as the product of the punch radius \( r \) by the punch angular extension \( \beta \). The work contribution in diameter reduction was expressed by considering the actual diameter of section on a normal plane as function of the punch angle \( \beta \).

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\[
\varepsilon_c = \frac{\Delta l}{l} = \frac{2\pi (R - u) - 2\pi R}{2\pi R} = \frac{r}{R} (1 - \cos \beta)
\]

(5)

In Figure 7, schematics of a capsule normal and meridian section were reported. At the beginning of the process the external capsule radius \( R \) resulted equal to half of the undeformed diameter \( D \); during compression, flowing on the punch surface, capsule radius \( R \) decreased by an amount \( u \) dependent from the angle \( \beta \). This reduction corresponds to a deformation regime \( \varepsilon_c \) estimated in Eq. 5.
Since the stress regime is constant and equal to the yielding point, the capsule diameter reduction work can be expressed as in Eq. 6.

\[ W_c = \sigma_y \int \varepsilon \, dv = \sigma_y 2 \pi r^2 \left( \beta - \sin \beta \right) \] (6)

The derivative respect to the punch angle gives the circumferential reduction loads (Eq.7)

\[ P_\beta = \frac{dW_c}{d\beta} \cdot \frac{d\beta}{ds} = \sigma_y 2 \pi r \left( 1 - \cos \beta \right) \] (7)

The influence of friction was not considered in Eq.4 and Eq.7. It can be included by considering the friction load contribute as a rate of the total load dependent to the component \( N \) normal to the punch contact surface (Eq. 8).

\[ N = P_\tau \cdot \sin \beta \] (8)

**Figure 8.** Capsule meridian section with total curling load decomposition in friction and normal to contact surface contribution.

In Figure 8 a scheme of total load decomposition \( P_\tau \) with respect to the capsule half meridian section is reported. By considering friction influence, the total curling load can be expressed as reported in Eq. 9, which resulted only dependent by material properties, capsule dimension, punch radius, and angular extension.

\[ P_\tau = \frac{P_b + P_c}{1 - \mu \sin \beta} \] (9)

The proposed analytical solution was adopted to estimate, both for type A and B capsule, the maximum curling load value. To validate the analytical approach, the computed values were compared to the experimental and numerical maximum loads.

3. Results and Discussion

In Figure 9 the type-A cap section is reported in three different compression stages. Image (a) shows a caps section before the compression test, the slight shape inward of the edge is clearly visible. In image
(b) a section in the first plastic deformation phase is reported, and in image (c) a section of a completely curled cap is highlighted.

Figure 9. Meridian section of tested type-A capsule before the test starting (a), during compression (b) and after compression test execution (c).

The comparison of the measurements acquired on the tested cap with those obtained on the production line, showed the deformed sections are perfectly comparable. Since the tests looked reproducing the curled cap shape, an analysis of the load vs die displacement curve was made. In Figure 10 load vs displacement curves are reported for tests conducted on type-A caps.

Figure 10. Load vs displacement compression curves of type-A capsule.

Tests A1 and A2 were performed with a compression velocity equal to 10 mm/min and tests A3 and A4 with a velocity of 1 mm/min. The curve trends show a good repeatability of the results independently by the compression velocity. Same results in terms of repeatability were obtained from the type-B caps.
load vs displacement curve, here not reported for the sake of brevity. Both for types A and B caps the average maximum load $P_{\text{max}}$ was computed and reported in table 3.

Table 3. Type-A and type-B maximum average compression load.

| Type  | $P_{\text{max}}$ [N] |
|-------|----------------------|
| Type-A| 3350                 |
| Type-B| 2465                 |

In Figure 11 numerically obtained load vs displacement curve for type-A capsule was compared to that one experimentally measured in A1 compression test. A good correlation between experimentally and numerically obtained curve can be recognize up to the maximum load peak reaching, while the scattering of the FEM curve, over the maximum load, resulted due to discretization and contact algorithm formulation dependency.

![Comparison between numerically obtained for type-A capsule (green curve) and A1 experimentally measured (dotted curve) load vs displacement curve.](image)

Figure 11. Comparison between numerically obtained for type-A capsule (green curve) and A1 experimentally measured (dotted curve) load vs displacement curve.

Same comparisons were made for type-B caps and the obtained results, here not reported for the sake of brevity, showed same correlation between experiments and simulation outcomes. Maximum values of load carried out by simulations, for type A and B capsule, are reported in table 4.

Table 4. Type-A and type-B maximum numerically computed load.

| Type  | $P_{\text{max}}^{\text{FEM}}$ [N] |
|-------|----------------------------------|
| Type-A| 3470                             |
| Type-B| 2552                             |

Since the good results obtained from the comparison of the curves, together with the small percentage of deviation in maximum loads, respect to the corresponding values experimentally obtained, a detailed study about the capsule deformation regime was conducted. In Figure 12, snapshots of deformed capsule from the stages of deformation indicated in Figure 11, are reported. Image (A) shows as initial
interaction between capsule and punch surface is localized in a very small region on the top of the capsule free edge. During these phase capsule wall resulted subjected to a distributed bending regime. With the increasing of the punch displacement, a progressive contact of the capsule wall to the die surface prevented radial displacements (B) by producing a localized bending clearly visible in image (C). This corresponded to a rapid increasing in load values up to the reaching of the maximum one in correspondence of first plastic hinge formation occurrence (D). The subsequent material plastic collapse produced the contact also between capsule wall and punch; this phase was characterized by a decreasing in load values until a local minimum (E). From this point the process consisted in a progressive sliding of the capsule material on the punch profile (F). The corresponding linear increasing in load values depended on the rising of contact surface and of material volume involved in process.

![Figure 12. Capsule deformation regime in different stage of compression from initial contact between capsule and punch (A), to the plastic hinge formation (F).](image)

Maximum values of curling load were also obtained by adopting the proposed analytical solution (Eq. 9) for both type A and B of capsules. The corresponding values are reported in the table 5 together with the values of load obtained from experimental tests and numerical simulations.

| $P_{\text{max}}$ [N] | Experimental | Numerical | Analytical |
|----------------------|--------------|-----------|------------|
| Type-A               | 3350         | 3470      | 3352       |
| Type-B               | 2465         | 2552      | 2548       |

Experimental, numerical and analytical maximum curling load resulted soundly comparable. In table 6, the deviation magnitude $e_{\%}$ of numerical and analytical simulation was computed respect to the experimentally obtained load. Both for type A and B capsules deviations resulted lower than 3%.
Table 6. Deviation magnitude of numerical and analytical load values compared to the experimentally obtained.

| Type    | $e_{\text{exp-numerical}}$ | $e_{\text{exp-analytical}}$ |
|---------|--------------------------|-------------------------------|
| Type-A  | $\approx 3\%$           | $>2\%$                         |
| Type-B  | $\approx 3.5\%$         | $\approx 3.2\%$               |

4. Conclusions

The conducted experimental campaign provided an estimation of the maximum curling load for the two typologies of examined capsules. Subsequently, the developed numerical model allowed to deeply investigate the phenomena involved in process. Deformation regime and stress distribution on capsule wall material were investigated. Finally, thanks to an energetic approach to the problem, a simple analytical solution for the maximum curling load estimation was developed. The proposed approach, supported by numerical and experimental benchmarks, by considering material properties, cap geometry and die angular extension, provided a simple approximated evaluation of maximum curling load.

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